SUBMERSIBLE VEHICLE SYSTEMS DESIGN

Written by
a Group of Authorities

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Preface

The purpose of this book is to present, for the first time, a comprehensive and cohesive work on the major elements of manned submersible design. It is intended to be of use to individuals and organizations concerned with the design, construction, operation, and/or certification/classification of these underwater vehicles as well as to those who may be involved in the planning or management of ocean systems utilizing them. The book should also be of use to those with paralleling interests in unmanned submersibles, either remotely operated vehicles (ROVs) or untethered autonomous underwater vehicles (AUVs), since numerous subjects discussed are pertinent to the design of all underwater vehicles.

The title of the book, *Submersible Vehicle Systems Design*, is indicative of its orientation and scope. The use of *Submersible Vehicle* identifies its focus on the design of small, low speed, short endurance, underwater craft which are heavily dependent on external sources of support, such as surface ships, to accomplish their mission. In contrast, the term "submarine" is associated, by popular usage, with large, high speed, long endurance, self-sufficient vehicles used almost exclusively for military missions. However, the type of submersible considered in this book and the submarine have certain characteristics in common—both are untethered, or "free swimming," manned vehicles possessing neutral buoyancy in the so-called submerged design condition. Consequently, a number of design principles are valid for both of these categories of vehicles. Indeed, much of the technology applied to the design and construction of these submersibles was developed initially for submarines.

The *Systems Design* portion of the title indicates the book's orientation toward the "systems approach" to submersible design as presented in Chapter I—"The Basic Design Process." This chapter focuses on the design of a submersible not as an isolated system but as one of perhaps several individual systems comprising the total, or mission, system. Emphasized is the fundamental that it is the design of the mission system which must satisfy a given set of mission requirements in an optimal manner. *Design*, in the title, also indicates the book's orientation toward a synthesizing procedure in which knowledge and techniques pertinent to the design process are assembled and utilized as "tools" without undue concern for associated theory. Consequently, this book is not intended to be a treatise on hydromechanics, structural analysis or other theoretical areas of interest. Rather, it is concerned essentially with the application of theory—the presentation of theory itself being held to a minimum consistent with the clear development of the subject at hand. Where appropriate, theory and other background material are found in references at the end of the chapters.

The book is composed of eight chapters—seven of them providing essential input to the submersible design process discussed in Chapter I. Very brief overviews of these chapters are given to provide an overall perspective of the book.

Chapter I, "The Basic Design Process," is initially concerned with the entire design process for the mission system of which the submersible is but one of its individual systems. The submersible system is then isolated from the mission system for a detailed consideration of the design process involved and of essential input to this process. Basic design procedures, involving the conceptual and preliminary design phases, are discussed with guidelines for conceptual design being presented.

Chapter II, "Characteristics and Development of Submersibles," is introduced with a historical review of submarine development to provide background on the source of much of the technology which has been applied to submersible development. Modern submersible development and various types of submersibles, both manned and unmanned, are presented to give an overview of this field of technology. Specific examples of submersibles designed and constructed in the United States and abroad are given and data listed to provide the reader with some appreciation of the range of submersible characteristics.

Chapter III, "The Environment," provides essential information on the nature of the environment in which submersibles operate—this environment being subdivided into the atmosphere, air/sea interface and the water column. The physical properties of sea water and the dynamical processes occurring in these subdivisions of the environment are sources of so-called mission external design constraints on the submersible design which must be thoroughly understood by the designer. Of pertinent, but more general, interest is the Chapter's section on the geography of the world's oceans which discusses the nature of the sea floor areas of the world.

Chapter IV, "Materials," contains a detailed discussion of materials used for submersible structural, buoyancy/ballast, and other systems. As was the case for environmental factors, the characteristics of materials in the presence of these factors, particularly pressure, temperature, and salinity, become mission external design constraints which must be understood by the designer in making correct selections of materials for particular applications. The Chapter contains extensive data and other information on materials to provide design guidelines.

Chapter V, "Hydromechanical Principles," focuses on engineering principles associated with the hydrostatic and hydrodynamic naval architectural aspects of submersible design. Various types of submersibles from the hydromechanical viewpoint, both manned and unmanned, are discussed as introductory background. Details of submersible hydrostatics are presented based on the type of submersible having these criteria for the so-called submerged design condition—neutral buoyancy, zero trim/list, and positive statical stability. Hydrodynamic principles presented are those related to the resistance/propulsion and motion stability/control aspects of design.

Chapter VI, "Structural Principles," is concerned with engineering principles underlying the structural design of submersibles. In this regard, three categories of structure are considered—the pressure hull, exostructure or main structure external to the pressure hull, and appendages to the main body of the submersible. Examples of various types of these structures are provided.

Chapter VII, "Submersible Vehicle Support Systems," considers systems other than the submersible which may comprise a mission system—specifically, land, air, and sea transportation systems, handling systems, navigational and positioning systems, and maintenance and repair facility systems. The Chapter's purpose is twofold—1) to aid in the selection of support systems to form, with the submersible system, a mission system meeting mission requirements within specified constraints and 2) to provide information on the constraints, called mission internal design constraints, which these systems place on each other, particularly on the submersible.

Chapter VIII, "Design and Operating Safety," is the third chapter in this book to discuss sources of mission external design constraints placed on the design of a submersible—the sources, in this instance, being rules and regulations pertaining to safety considerations for the design and, additionally, the operation of the submersible's systems. Background is given on entities concerned with submersible safety, such as the U.S. Navy and classification societies, and specific safety requirements of these entities are given.

Acknowledgments

This book is the result of the volunteer efforts of a number of individuals who are especially qualified to write on the subject material of their particular chapters or chapter sections. These authors, identified at the beginning of their respective chapters, deserve the gratitude of all who read this book for sharing their considerable knowledge on various aspects of submersible design. The Editor also wishes to record his appreciation to all the authors. It was indeed a pleasure to work with them in completing this book.

The Editor is greatly indebted to the book's Control Committee. Their individual and combined expertise in submersible and submarine design, construction, and operation were invaluable in completing the extensive and exacting review process. The Committee's unstinting commitment to this publication, in terms of time, effort, and support of the Editor's activities, is sincerely appreciated.

The input and support of the Society's MS-2 (Submersibles) Panel of the Marine Systems Committee are acknowledged with appreciation. In particular, the Editor would like to sincerely thank the Panel's chairman, Mr. John Pritzlaff, for his untiring support and for facilitating the publication of this book in numerous ways.

The Editor is indeed grateful for the encouragement and fiscal support of Perry Offshore, Inc. (then Perry Oceanographics, Inc.) and for the good offices of Mr. H. A. (HAP) Perry in this regard. Mr. Ralph Draper, then an employee of Perry Oceanographics, Inc., devoted considerable time and effort in advancing the best interests of this book in many ways which are very much appreciated.

The late Dr. Edwin A. Link of Harbor Branch Foundation made the Foundation's submersible personnel and facilities available to the Editor during early days of this undertaking. His interest in and support of this effort are gratefully acknowledged.

A primary reference for this book is *Manned Submersibles* by R. Frank Busby of Busby Associates, Inc. Mr. Busby facilitated access to certain material in his book and provided encouragement and advice on this work for which the Editor is grateful.

The Editor is indebted to Mr. Terence Flanagan of Flanagan Associates and his staff for their part in bringing this book into being. Their work is much appreciated.

Numerous other individuals and organizations have aided and abetted in the development of this project—from offering moral support to providing advice on chapter content and granting permission to use photographs and other material. The Editor sincerely thanks all who were so involved.

Finally, the Editor acknowledges with deep gratitude the essential support provided by his wife, Ann. Her actual and supportive involvement were manifested in many ways which served the very best interest of this undertaking.

E. Eugene Allmendinger
Editor

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Chapter I
The Basic Design Process

E. E. Allmendinger

1. Introduction

THIS CHAPTER is concerned with the basic design of manned submersible vehicle systems and, in a general sense, with the overall systems of which they are a part. There are many ways of pursuing the design process, each varying in detail according to time-honored procedures followed by design organizations and to the simplicity or complexity of the overall system in question. Consequently, no attempt is made herein to present *the* way of pursuing this process. Rather, guidelines are presented illustrating fundamentals and concepts that may be applied, in one form or another, to a broad spectrum of approaches to the design of these systems. Certain of these fundamentals are also useful in the design of unmanned submersibles, or remote-controlled vehicles as they are usually called.

Section 2 presents perspectives of the submersible and its supporting systems, the total system being called the *mission system*, and the overall design and associated processes involved in the development of this system. The pertinence of this perspective becomes apparent when one considers the fact that small submersibles, unlike large submarines, are incapable of operating alone. They must be assisted by other systems in order to accomplish underwater tasks. At the very least, they must be supported by shore-based facilities, housing maintenance, repair and administrative activities, and by ships providing transportation, launching and retrieval services, underwater navigational support, and at-sea maintenance and repair facilities. More complex mission systems may also include other support systems such as land-transport vehicles, air-transport vehicles, satellites, seafloor-based transponders, and even submarines. The submersible and other systems constituting the mission system, whatever it may be, must work in concert with each other to accomplish the mission tasks in the most efficient manner possible. Thus, it is important to view this collection of individual systems as a whole—being aware of their interactions and remembering that the cost of accomplishing underwater tasks is the cost of the mission system and not the submersible's cost alone.

Section 2 presents, in Fig. 1, a way of viewing the submersible and its supporting systems, both as individual systems and as they are combined to form the mission system. This view facilitates discussions of the design goals which involve design optimization and consideration of design constraints. The background of design optimization is developed as based on cost-effectiveness criteria and is diagrammed in Fig. 2. Constraints on the design of the mission system, as will be seen, include *mission external* and *mission internal design constraints* and *design criteria*. They are indicated in Fig. 1.

Section 2 also discusses three processes associated with mission system design or the design of any of its individual systems including the submersible—the *pre-design*, *design*, and *post-design* processes which are diagrammed in Fig. 3. The pre-design process involves the potential user/owner of the system to be designed and is concerned with the development of the primary input to the design process—the *mission statement* and *mission requirements*. The design process conceives the mission system that is feasible and meets the mission requirements in as optimal a manner as possible. It is divided into *basic*, *contract*, and *detail design* phases—basic design being subdivided into *conceptual* and *preliminary design* stages. An overview of conceptual design, essentially involving feasibility studies, is diagrammed in Fig. 4. Finally, the post-design process is composed of three activities which are discussed briefly—*construction* and *alteration* of "new" and "existing" individual systems of the mission system, *test* and *evaluation*, and *operation*. Experiences gained from these activities form, collectively, important "feedback" input to the design of future systems.

Succeeding sections of this chapter focus exclusively on the basic design of manned submersibles. Section 3 provides a description of submersibles in general—considering these systems, in turn, to be composed of a number of individual systems as illustrated in Fig. 1. The titles of these systems are derived from the U.S. Navy's "Ship Work Breakdown Structure" (SWBS)—this document also being followed, to the extent feasible, in discussions of system components.

Section 4 discusses inputs to submersible design, dividing them into two categories—inputs which furnish *guidance* for the design and those which impose *constraints* on it. Guidance inputs discussed include *mission/performance requirements*, the latter being derived from the former, and *post-design experiences*. Constraint inputs, as discussed for mission systems, are imposed by *mission external* and *internal design constraints* and *design criteria*. As will become evident, constraints come from many sources, making a comprehensive treatment of them beyond the scope of this chapter. Consequently, the presentation of this subject is detailed only to the extent necessary to convey an impression of the nature of the various constraints involved. Other chapters of this publication present details of the more important sources of constraints.

Section 5 concerns the basic design of manned submersibles, focusing primarily on their conceptual design. The development of conceptual design alternatives and the bases for selecting the optimum alternative, *technical feasibility* and *cost-effectiveness*, are discussed. The *empirical* and *systematic parametric analysis* approaches to conceptual design are considered. The empirical approach is modelled by the *design spiral*, and the parametric approach is illustrated with an example of a design optimization program contained in the Appendix to this chapter.

Submersible design-related references are given at the end of the chapter. It should be noted that *Manned Submersibles*, by R. Frank Busby, is considered by the author to be a companion piece to this and other chapters. It is an excellent source of general and detailed information and data on these vehicles.

2. An Overall Perspective

2.1 Overview of Mission and Submersible Systems

A system is any object or process, or group of objects or processes, created to serve some useful purpose or mission, the mission being defined by a set of mission requirements. Such a system may be called a *mission system* to differentiate it from other system categories to be discussed presently. It is often convenient to picture a system as isolated from its surroundings by a boundary for the purpose of studying its internal behavior and interaction with its surroundings. Figure 1 provides such a diagram, in this instance showing a mission system designed to serve one or more underwater missions. Note that the mission system boundary encloses *M individual* or (I) *systems*, each enclosed by its own boundary, illustrating the aforementioned fact that the mission is accomplished by a submersible supported by other systems. The mission system diagram in Fig. 1 provides an example of these (I) systems in which it is assumed that mission requirements dictate the needs, among others, for the submersible's transportation by air and on the surface of the sea as well as for its launching, recovery and support while submerged. As shown, (I_1) is the submersible, (I_2) a transport aircraft, (I_3) a surface support ship, and so forth—the last individual system, whatever it may be, is designated (I_M).

The design of many mission systems begins with most of the potential (I) systems already existing and some nonexisting. In the foregoing example, for instance, the aircraft and surface ship may already exist while the submersible does not exist. Existing systems require from no to extensive alterations to convert them to (I) systems, within a particular mission system, with commensurate amounts of design effort, construction and costs involved. Obviously, the extent of alterations required is a primary consideration in choosing between candidate (I) systems. Nonexisting (I) systems, of course, must be designed and constructed "from scratch" and are often the most costly of these systems.

Fig. 1 shows the manned submersible system isolated from the mission system. Again, note that the submersible's boundary encloses *N individual* or (S) *systems*. Examples of (S) systems shown in the figure include (S_1) hull structure, (S_2) propulsion plant, (S_3) electrical plant systems, and so forth—the last system being (S_N). The "*N*" term can be any relatively small number depending on the accounting system used by the design organization. The U.S. Navy's SWBS is used herein in which *N* is either 6 or 7 as indicated. Note also that in the submersible system shown in Fig. 1 the (S) system boundaries enclose "*i*" (SS) *subsystems*. For example, the (S_1) hull structure system would enclose the (S_1S_1) shell plating (the pressure hull), (S_1S_2) longitudinal and transverse framing (the exostructure), and so forth according to the SWBS. Although not shown in the figure, subsystems in turn can be broken down into yet smaller systems. One can visualize this "boxes within boxes" procedure extended until every last nut and bolt of every (S) system is isolated, itself, as a system. It will suffice to say here that systems breakdown becomes increasingly fine as the design process advances.

The design of any (I), (S), or smaller system also begins with at least some of its individual systems existing. These existing systems, requiring no or

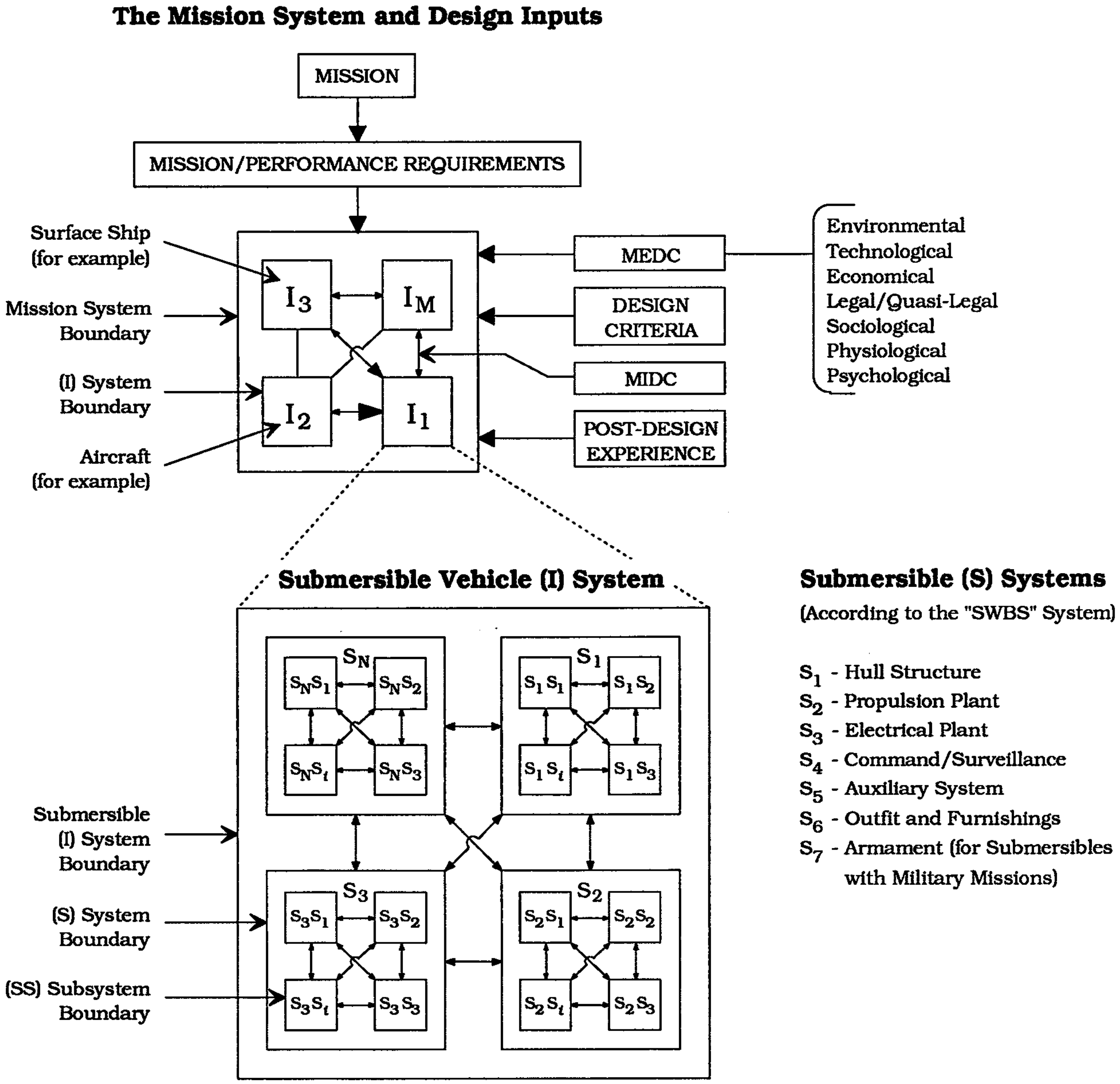


Fig. 1 The mission system and its subsystems

small modifications, are often referred to as "off the shelf" items. Their use significantly reduces costs.

2.2 The Design Goal

It is the mission system, and not necessarily any of its individual (I) systems, which should achieve the design goal of *satisfying a given set of mission requirements in as optimal a manner as possible considering the design constraints acting.* This statement introduces the topics of *design optimization* and *design constraints* which are discussed briefly in the following paragraphs.

2.3 Design Optimization

Optimizing a mission system's design is usually based on maximizing its *cost-effectiveness* (*CE*) to the extent possible given the constraints acting. This process may vary from relatively simple to complex depending on the nature of the mission system. In general, it requires extensive economics and experiential data and the use of sophisticated techniques, the descriptions of which are beyond the scope of this presentation. The purpose here is to convey an appreciation for what this process involves and for some of the major design considerations entering into it.

Maximizing cost-effectiveness essentially means arriving at one of many solutions to the design problem, posed by mission requirements and design constraints, which identifies the mission system best able to accomplish the mission task(s) at least cost. For a mission system including a manned submersible as one of its systems, one way of expressing this statement as an equation is

$$\frac{1}{CE} = \frac{\text{cost}}{\text{mission task}} = \frac{\text{dive days}}{\text{mission task}} \times \frac{\text{operating costs / year}}{\text{dive days / year}} \tag{1}$$

The first term of the equation's right side is the measure of the system's expeditiousness in performing given underwater tasks. The second term is the mission system's *cost per dive day*, which is "bottom line" economic data for system operations. Optimization results from minimizing $1/CE$ or maximizing CE itself. Figure 2 shows a breakdown of considerations involved in mission system cost-effectiveness studies.

Dive days per mission task, as indicated in Fig. 2, can be expressed as *hours per mission task* (once the submersible is on task site) divided by *on task-site hours per dive day*—this ratio reflecting the effectiveness with which tasks are accomplished. It is evident that this ratio should be as small as possible considering capital and operating-cost constraints. It is also evident from Fig. 2 what characteristics of the submersible and other (I) systems are involved in attempting to minimize *hours per mission task* and

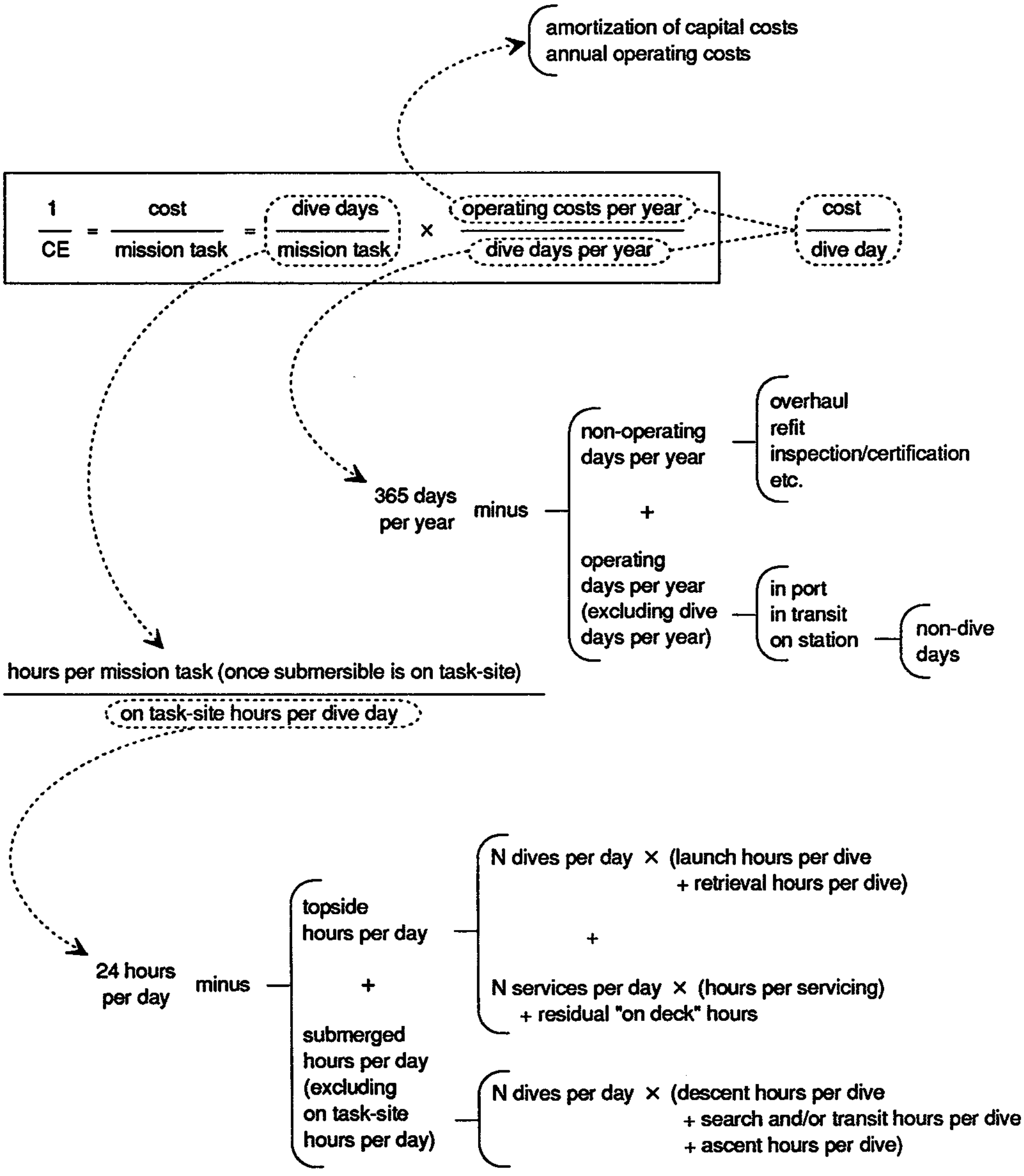


Fig. 2 Mission system cost-effectiveness

maximize *on task-site hours per dive day.* In the first instance, for example, effective on-site utilization of time is dependent on the capabilities of instrumentation and equipment carried on the submersible to perform tasks and on the expertise (personnel training) with which they are used. In illustration, consider a manipulator. Its manipulative capabilities and the expertise with which they are used by the human operator are important influences. But there may be antagonistic considerations met in attempting to minimize *hours per mission task* in this illustration—the most capable, and also available and reliable, manipulator may also be the most expensive alternative which acts adversely on the second term of the equation. Cost-effectiveness considerations in feasibility studies are based on the fine art of making judicious compromises in order to arrive at the optimum overall solution to the design problem.

Maximizing *on task-site hours per dive day* to the extent feasible involves, in turn, minimizing to the extent feasible *topside* and *submerged hours per day*—this time being necessary but unproductive. As seen in Fig. 2, *topside time* primarily includes *launch/retrieval* and *servicing times.* Most of these systems launch submersibles quite expeditiously if there is reasonable compatibility between these (I) systems, although some systems require more launching crew than others. Retrieval time begins when the submersible surfaces and ends when it is on deck—a much more time-consuming evolution than launching unless the submersible is tethered, particularly in adverse sea-weather conditions. Submersible surface characteristics, such as freeboard, are helpful in reducing retrieval time as is the method of "pick-up" employed by launch/retrieval systems. In this regard, the single-point lift over the stern is considered by many as the superior method. Surface communication linkages, stressing effectiveness in poor sea-weather conditions and providing for emergency situations, can also reduce retrieval time. Servicing time primarily involves energy replacement and pre- and post-dive check-out of the submersible's systems. Energy replacement time can be reduced by submersible characteristics such as a storage system using replaceable, modular battery packs. Minimizing servicing of submersible systems emphasizes the importance of design features stressing simplicity, ruggedness, reliability, maintainability, and accessibility of these systems. It must be remembered that servicing instruments and equipment at sea is apt to be much more difficult and time-consuming than at a shore base.

Submerged time, excluding *on task-site time,* includes *descent, search* and *transit,* and *ascent times.* Reduction of descent and ascent times focuses entirely on the submersible, if untethered, and its characteristics, such as a streamlined form, which increase descent and ascent velocities. These characteristics become increasingly significant as the maximum operating depth increases. Tethered submersibles have added concerns related to the characteristics of tether-management systems. Reduction of time expended in searching for the task site or in submerged transit between sites draws attention to the capabilities of underwater navigation systems which involve the characteristics of communication linkages between the submersible and other (I) systems such as the surface support ship and bottom-anchored transponders.

The second term of the cost-effectiveness equation is *operation cost per year* divided by *dive days per year* or *cost per dive day.* Operating cost per year, as indicated in Fig. 2, usually includes amortization of capital cost of the (I) systems—either as they are newly constructed, or altered, existing systems. It is dependent, of course, on the choice of (I) systems composing the mission system from an economic perspective. Emphasized is the need for these choices to consider both *capital* and *annual operating costs* of these systems as well as their collective ability to satisfy the mission requirements. In this regard, it must be remembered that the objective is to minimize the *cost per mission task,* which does not necessarily mean minimizing these costs for each (I) system of the mission system.

Dive days per year, of course, are the only potentially productive days of the year—the only days when submerged operations are conducted. Consequently, their number should be maximized to the extent feasible which, in turn, requires that the remaining days' number should be minimized to the extent possible. As shown in Fig. 2, these days are divided into *non-operating* and *operating days,* excluding dive days. Non-operating days, or *layup days,* include those which must be set aside for activities such as *overhaul, refit, inspection/certification* procedures, and so forth as well as *idle days* resulting from unemployment. If the "must" activities are not conducted on a yearly basis, the number of days so utilized should be averaged over the years involved. Thus, if they occur in a block of time over two years, each year should be charged with one-half of their number. Idle days can be included, if desired, based on experience—remaining constant for studies leading to selecting the optimum design.

Operating days per year, excluding dive days, include *in-port, in-transit* to and between stations, and *on-station nondive days.* In-port days include those necessary for preparing for the voyage and providing personnel with a respite from rigors of the sea. Their number is dependent on the characteristics of the port and support ship, principally from the view of cargo handling and fueling, and on management policy regarding shore leave. In-transit days depend in number on voyage distances, average sea-weather conditions enroute, and on the support-transport ship characteristics regarding its speed and seakeeping abilities. On-station days, in general, are those days when the transport-support ship is in the desired location for deploying the submersible. The number of nondive days on station depends on sea-weather conditions, the motion and position-keeping characteristics of the ship, the capabilities of the launch/retrieval system, and the material condition of the submersible.

Minimizing the number of unproductive days per year to the extent possible involves two sets of considerations: those focusing on the design of the mission system and those concerned with all aspects of its management. The first set of considerations is of interest here.

The design of the mission system should lead to the selection of its (I) systems and their characteristics which tend to reduce the number of unproductive days. In this regard, important considerations include (1) basic design philosophy, (2) speed of transit to, from, and between stations or operating areas, (3) endurance of the support system, (4) maintenance and repair facilities, and (5) limiting weather/sea-surface conditions for operation and survival.

Consideration (1) focuses on the virtues of keeping (I) system components as simple and rugged as possible, thereby promoting system safety and reliability. As will be seen, this philosophy encourages use of the design criterion that designs, particularly of critical systems, be based on state-of-the-art technologies—that is, on proven technologies. The contributions of this philosophy to increasing dive days per year, as well as to operational safety, are obvious.

Consideration (2) focuses on the speed requirement for the (I) system providing transportation—usually a surface ship. This speed should be as high as feasible, and the ship's seakeeping characteristics should permit its maintenance under reasonably adverse sea conditions. This consideration may also lead to the choice of more than one system for transportation, particularly if mission requirements emphasize high in-transit speeds or short response times. The use of an aircraft to fly a submersible and associated gear to a distant port to be placed on board a surface ship is a case in point.

Consideration (3) involves the ability of the support system to remain at sea for a specified length of time, which is usually given as a mission or performance requirement. It is concerned with characteristics such as fuel, freshwater, provisions and stores capacities, and habitability. Habitability has to do with all aspects of design relating to human comfort and is directly related to endurance of personnel and the level of work performance. It increases in importance as endurance increases.

Consideration (4) includes facilities that are vital to the success of the mission as well as to reducing the unproductive, on-station time—facilities that maintain the submersible in a state of operational readiness. They may be an integral part of the support system or another (I) system closely associated with the submersible. In the latter instance, these facilities are housed in portable vans which are outfitted at the shore facilities and secured to the deck of the support ship for the cruise. In both instances, the facilities include personnel trained in specific areas of technology such as mechanical and electronic technicians. These persons, including the pilot, are referred to as the "submersible's crew" to differentiate them from the "ship's crew" and other groups of personnel on board.

Consideration (5) involves two environmental factors: weather conditions, with visibility and wind forces being principal concerns, and sea-surface conditions as measured by sea-state numbers from 0 to 9, severity of conditions increasing as these numbers increase. One or both factors, if adverse, can cause a serious increase in unproductive in-transit and on-station time as well as endangering survival of any transport-support system subjected to these elements. Consequently, the design or choice, or both, of (I) systems should consider ways and means of reducing this unproductive time to the extent possible. The three (I) systems primarily involved are the *transport-support, submersible,* and *launch-recovery systems.*

The major transport-support system has been, and is currently, a mono-hull surface ship. From the perspective in question, important characteristics include (1) its seakeeping characteristics permitting speed to be maintained under reasonably adverse sea conditions, (2) its motion characteristics enabling it to provide a relatively stable working platform at higher sea states, and (3) its survival characteristics permitting it to proceed in-transit or remain on station with safety at high sea states. In recent years, safety and economic concerns with limiting operating and survival sea states have given rise to consideration of nontraditional transport-support systems, including semisubmersible ships and submarines. Semisubmersible ships are more "transparent" to wave action than mono-hull ships thereby furnishing more stable working platforms under given sea conditions. The use of submarines leads to a "a break with the surface" altogether. This system choice may become inevitable when operating areas involve other surface conditions such as broken or solid ice fields.

Submersible design, from weather-sea considerations, should be concerned with increasing the *dive time per on-station time ratio* to the extent possible. This concern focuses on an extremely critical phase of manned submersible operation: its retrieval after launching. This phase is always carefully considered, in the light of anticipated weather-sea conditions, in making decisions to initiate a dive or to terminate it once in progress—decisions, of course, which directly affect the dive time per on-station time ratio. From this view, design features include those

(1) enhancing the submersible's surface condition, primarily providing adequate freeboard and height of access hatch above the waterline, (2) improving surface communications and means of being located, and (3) facilitating launch-recovery procedures through the reduction of its size and weight to the extent possible. As will be seen, (1) and (3) present the designer of manned submersibles with distinct challenges.

Launch-recovery system design, like submersible design, should also be concerned with increasing the dive time/on-station time ratio and for the same reason. Several types of this system exist—perhaps, as previously indicated, the most common being the single-point lift, constant-tension, A-frame system located at the support ship's stern. Of the three systems being considered, it is the one which usually sets limiting sea states for operation owing to, essentially, two factors: (1) the requirement for it to function safely and effectively at the interface between the submersible and support systems which usually have highly mismatched motion characteristics in a seaway and (2) the necessity for most types of this system to have divers in the water when handling untethered submersibles. Factor (1) creates the potential for developing dangerously high dynamic loads and submersible-ship collisions if extreme care is not exercised by the system's operator. Factor (2) can lead, of course, to hazardous situations for the divers in all but the lowest sea states. Motion mismatch must usually be accepted if the support system is a conventional surface ship. Consequently, improvements in the design of the launch-recovery system that address this problem, as well as obviating the need for divers, are required to raise on-station productive time. One possibility involves the use of a remote handling system in which the submersible is launched and recovered while submerged at a depth of minimal surface-wave action—a solution which also removes the need to emphasize the submersible's surface operating characteristics. The choice of other transport-support systems, such as semisubmersible ships and submarines, may also be considered in an effort to raise the mission system's productivity from this point of view.

2.4 Design Constraints

There are numerous design constraints imposed on the mission system and its individual systems. They may be placed into one of three categories: *mission external design constraints*, *mission internal design constraints*, and *design criteria*.

Mission external design constraints (MEDC)—As shown in Fig. 1, sources of MEDC lie outside, or external to, the mission system boundary, indicating that they are independent of all mission considerations; that is, they exist irrespective of the mission system in question. MEDCs imposed on system design may come from a few or several of the following sources: (1) *environmental*, (2) *technological*, (3) *economical*, (4) *legal/quasi-legal*, (5) *political*, (6) *sociological*, (7) *physiological*, and (8) *psychological*. The designer's control over MEDCs varies from none to considerable. For instance, he has no control over the environmental parameters such as pressure, temperature, salinity, and energy-propagating characteristics. He must accept them for what they are, clearly recognizing the constraints they impose on the design. On the other hand, the designer can, for example, exercise some control over technological constraints associated with materials by selecting a specific material for, say, the pressure hull of the submersible. However, once the material has been selected, the design is constrained by its metallurgical properties.

MEDCs from several of these sources usually vary with the passage of time. For instance, technological, economic and legal considerations and situations may change between the inception of a system's design and its construction. Anticipating these changes and allowing for whatever impact they may have on the design often improves the system's performance. Changes in technological constraints are particularly important to anticipate. In this regard, envision a "Technology Status Scale" marked from left to right with status indicators "Concept," "Research and Development," "Experimental," and "State-of-the-Art." The scale indicates that all systems at any level of the "boxes-within-boxes" scheme advance, with time, from birth as a concept on paper through the intermediate status levels where they are either promoted or discarded, the promoted systems eventually reaching the final status level. "State-of-the-art" status means that the system has accumulated a favorable history of operating experience. Because safety and reliability are key attributes associated with this status, the choice of critical systems is always based on it. Noncritical systems may be selected on the basis of a lower status level for compelling reasons. As an example of anticipating changes in these levels, consider a relatively new battery which has advanced from "concept" to "experimental" status and is currently installed on a test vehicle where "debugging" is in progress. Its high energy-density (watt-hours per pound), as compared with lead-acid and silver-zinc batteries, makes it a potentially attractive energy source for a submersible. The question faced by the designer is: should the battery, considered to be a critical system, be incorporated into the submersible's design now in anticipation of its achieving "state-of-the-art" two years hence when it will be installed, or should the design use a current "state-of-the-art" battery? Correct answers to such questions have the potential for placing the submersible, or any finished product, well ahead of its competition in performance. These answers, of course, involve cost considerations. In this regard, it should be observed that system costs tend to decrease with increase in time in "state-of-the-art" status—due to increasing confidence in and demand for the system. In summary, then, although design conservatism may indicate otherwise, the designer should at least be fully aware of this consideration and should exercise anticipatory judgment as warranted.

Questions similar to those preceding also serve to focus attention on technological areas whose progress along the "Technology Status Scale" requires expediting. In this instance, the basis for decisions lies in the ability to anticipate future missions and their mission requirements. The difficulty of this task increases with the number of years involved in the projection. Nevertheless, projections based on, say, five years will provide the indispensible lead-time required for this process.

Mission internal design constraints (MIDC)—The sources of MIDC lie inside, or internal to, the mission system's boundary—or, inside the boundary of any (I) or lower-level system. These sources are the interactions between the systems that influence each other's designs. In Fig. 1, MIDC are symbolized by arrows, or vectors, between the boxes representing system boundaries. The strength of these vectors, indicated by the size of the vector's tip, may vary from zero for "decoupled" systems to considerable for "coupled" systems. For example, in Fig. 1, the submersible (I_1) and aircraft (I_2) are "coupled," or "interactive," systems, as are the submersible (I_1) and the surface ship (I_3) systems. On the other hand, the (I_2) and (I_3) are "decoupled," or "non-interactive," systems. If coupled, "existing" systems often exert stronger influences on "new" systems than vice versa. For instance, consider the MIDCs acting between the (I_1) and (I_2) systems. The (I_2) system imposes strict weight and size constraints on (I_1), whereas (I_1) may only require minor alterations in (I_2)—perhaps modifications in handling equipment and tie-down gear. Consequently, these constraints are represented in Fig. 1 by the heavy-tipped vector from (I_2) to (I_1) and the light-tipped vector from (I_1) to (I_2). As a second example, consider two (S) subsystems of a manned submersible—man and the life-support systems which are not shown in Fig. 1 but are here arbitrarily designated as (S_5S_1) and (S_5S_2), respectively. The "existing" system, (S_5S_1), exerts considerable influence on the design of (S_5S_2) by reason of man's physiological requirements. In turn, (S_5S_2) can only impose limited constraints on (S_5S_1), requiring that man be "altered" by increasing his knowledge and skills so that he can operate the (S_5S_2), or other (SS) systems, correctly. MIDC vector notations between these systems would be similar to those between the (I) systems of the first example.

A major consideration in selecting "existing" (I), (S), or lower-level systems is that minimal MIDCs will be exerted on them by other systems. This would mean that a minimum of alterations would be required to convert them for use in the mission system which would involve commensurately low conversion costs.

Design criteria (DC)—These criteria furnish standards for judgments regarding design procedures and for selecting system components. The use of specified structural formulations and the requirement that critical system components be based on state-of-the-art technology are examples. The basis for optimization, either cost-effectiveness or minimum weight, may also be considered as a design criterion. These criteria may be generated by the user/owner, the designer, or classification-certification bodies.

2.5 The Design and Associated Processes

These processes, diagrammed in Fig. 3, may be referred to in chronological order as the (1) *pre-design*, (2) *design*, and (3) *post-design process*. In this section, they are discussed briefly from the perspective of the mission system to provide an overview of the entire subject. The remaining sections of this chapter focus essentially on the first phase of the design process, called *basic design*, as it applies specifically to the submersible system.

The pre-design process—This process leads to the formulation of the basic input to the design process—the *mission(s) statement(s)* and associated *mission requirements*. It is undertaken by the potential user or owner of the mission system to be designed, independently or in consultation with the system's designer.

The mission is a short, concise statement of *what* the user/owner wants to accomplish—in this case, underwater. Missions may be categorized in several ways. The most general way is by the titles of the user community being served: (1) *industrial/commercial*, (2) *military*, (3) *scientific*, or (4) *recreational*. A subcategorization may be based on terms describing the general nature of the underwater tasks to be performed; for example, (1) *inspection*, (2) *survey*, (3) *monitoring/sampling*, (4) *construction*, (5) *maintenance/repair*, (6) *search*, (7) *salvage*, (8) *rescue*, and so forth. Mission system design may be based on single or multimission requirements, the latter including two or more categories of underwater tasks.

Mission requirements, also called user/owner requirements, elaborate on the mission statement by specifying conditions under which the mission task(s) are to be carried out. They include information such as land-base locations, task-site locations

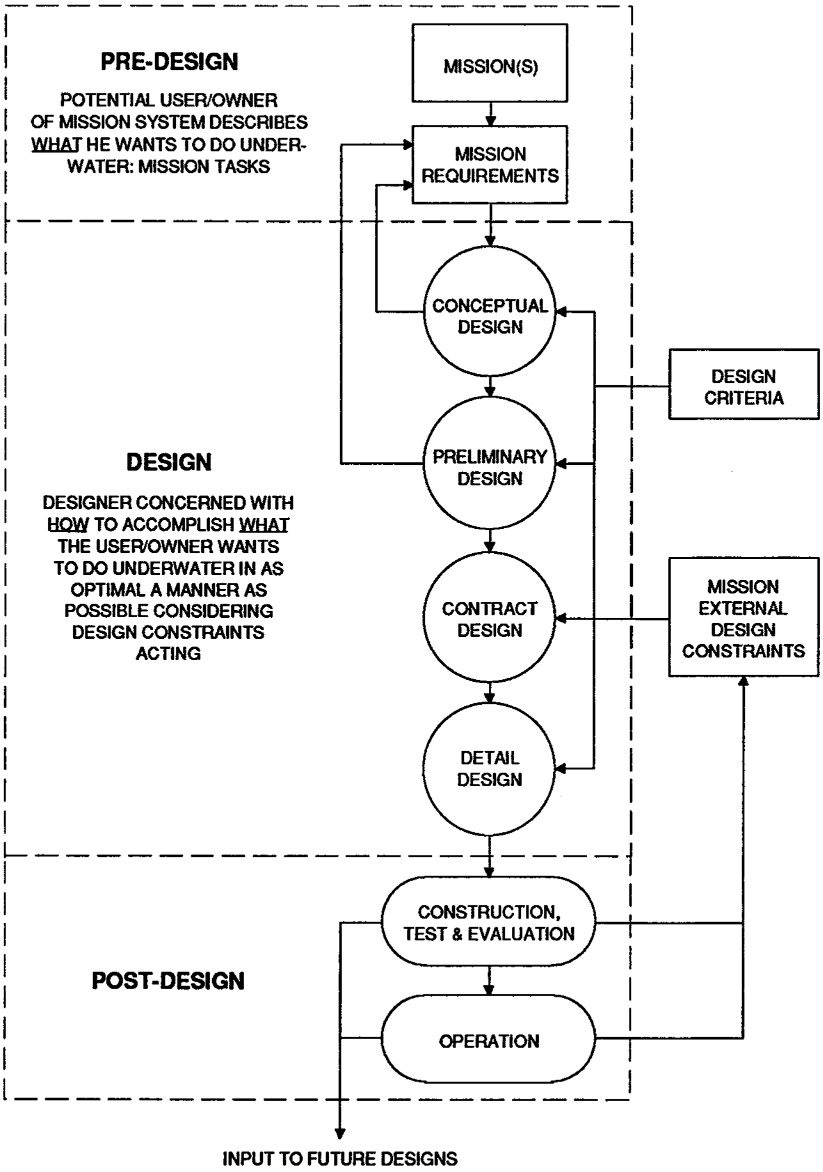


Fig. 3 Design and associated processes

and maximum depths, descriptions of work objects and tasks to be performed, and such other information as the user/owner may wish to stipulate. These requirements become the basis for developing the *performance requirements* during conceptual design which set forth capabilities that the various (I) systems must possess to accomplish the mission as prescribed. For example, the locations of land bases and task sites lead to the development of capabilities which the transportation system, say, a surface ship, must possess including its range, endurance, speed, and seakeeping qualities. As will be seen, the mission requirements may be altered during basic design by changes made in performance requirements—this process is indicated by feedback lines from the conceptual and preliminary design as indicated in Fig. 3. These changes should be made in consultations between the user/owner and the designer.

Inherently, single-mission systems are more efficient than multimission systems, with degradation of efficiency increasing as the number of missions increases. In multimission systems, each mission has its own set of mission and performance requirements—none, a few or several of these requirements being common to all sets. A composite set of unhomogeneous requirements results in design compromises and trade-offs in arriving at the characteristics of the various (I) systems which reduce the mission system's ability to perform any one mission well. Consequently, if multimission systems are deemed necessary, their designs should be based on the highest degree of commonality possible between the sets of requirements. Otherwise, a *primary mission* should be identified from among the several missions with the composite set of requirements favoring this mission.

The design process—This process is composed of three design phases: (1) *basic design*, (2) *contract design*, and (3) *detail design*. The levels of labor, preparation time, and costs associated with each phase increase exponentially as the process progresses. Design creativity and flexibility are essentially limited to the basic design phase—the mission system, including its submersible system, being well defined at the beginning of contract design.

The design process, at this point, is discussed from the perspective of the mission system (the design of its submersible system is considered specifically in Section 5). As has been seen, this system is composed of (I) systems which may be labelled as *new construction* or *existing systems*. Design efforts focus on the total, or "from-scratch," design of new construction systems and on the selection of suitable existing systems and their alteration, if required, to convert them to (I) systems. The submersible, herein, is considered to be a new-construction (I) system.

Basic design is the primary design phase concerned with *how best* to accomplish *what* the potential user/owner of the mission system wants to do underwater as set forth by his mission requirements. In general, this phase involves:

1. Development of performance requirements, or (I) system capabilities, from the mission requirements.
2. Determination of the principal characteristics of the (I) systems required to achieve these capabilities, which enable new construction systems to be designed and existing systems to be selected and altered if necessary.
3. Estimation of capital and operating costs of (I) systems.
4. Identification of one or a few mission systems from among a series of alternatives which optimally meet performance requirements to the extent possible considering design constraints acting.
5. Refining and firming-up characteristics and cost estimates of the (I) systems composing the optimum conceptual design(s).

The *conceptual design stage* of basic design is concerned with the first four of these functions—function four, herein, is based on cost-effectiveness as the optimization criterion. The *preliminary design stage* of this phase is concerned with the fifth function.

Conceptual design, diagrammed in Fig. 4, is the first attempt to translate all the user/owner's mission requirements into performance requirements and characteristics of the (I) systems composing the mission system. It is often viewed as consisting of *feasibility studies* and *completion* of the *optimum conceptual design(s)* once they are identified by these studies. Feasibility studies are concerned with the development of those performance requirements and associated system characteristics which have significant impact on the cost-effectiveness optimization process as discussed in Subsection 2.3 and diagrammed in Fig. 2. Completion of the optimum design(s) involves developing other major performance requirements and characteristics—those which are essential in defining (I) systems but which do not have significant impact on the optimization process.

Feasibility studies create an orderly series of mission system alternatives, all of which meet the mission requirements and are technically feasible, and select one or a few of these alternatives as the conceptual design(s) to be carried on into preliminary design. These alternatives are shown in Fig. 4 as MS_1, MS_2 ... MS_i where "i" can be any small to large number depending on the simplicity/complexity of the mission and optimization techniques used; for example, techniques using longhand or computer methods. As indicated in the figure, each alternative is composed of a unique combination of (I) systems. Each (I) system, in turn, has unique capabilities, characteristics required to achieve these capabilities, and costs associated with providing these characteristics. Certain unique (I) systems may appear in more than one combination but individual combinations are not duplicated in other alternatives. Mission system alternatives may be generated by (1) varying performance requirements with the cascading effect of varying (I) systems' capabilities, characteristics and costs and (2) varying (I) systems' capabilities, characteristics, and costs in ways each of which meet a specific set of performance requirements. Alternatives generated in this manner contain information necessary to conduct the search for the optimum, cost-effective mission system(s).

Combinations of (I) systems forming mission system alternatives may vary in *number*, *type/capabilities*, and *status*. These features are related to functions required to accomplish mission tasks, the time frame for completing these tasks, if critical, and economic considerations. A *mission profile*, derived from mission requirements, is useful in providing functions and time-frame information—its usefulness increasing as the complexity of the mission increases. This profile is a chronological listing of all events occurring from the mission's beginning to end and including time-frame data where necessary. For example, the profile of a certain demanding mission,

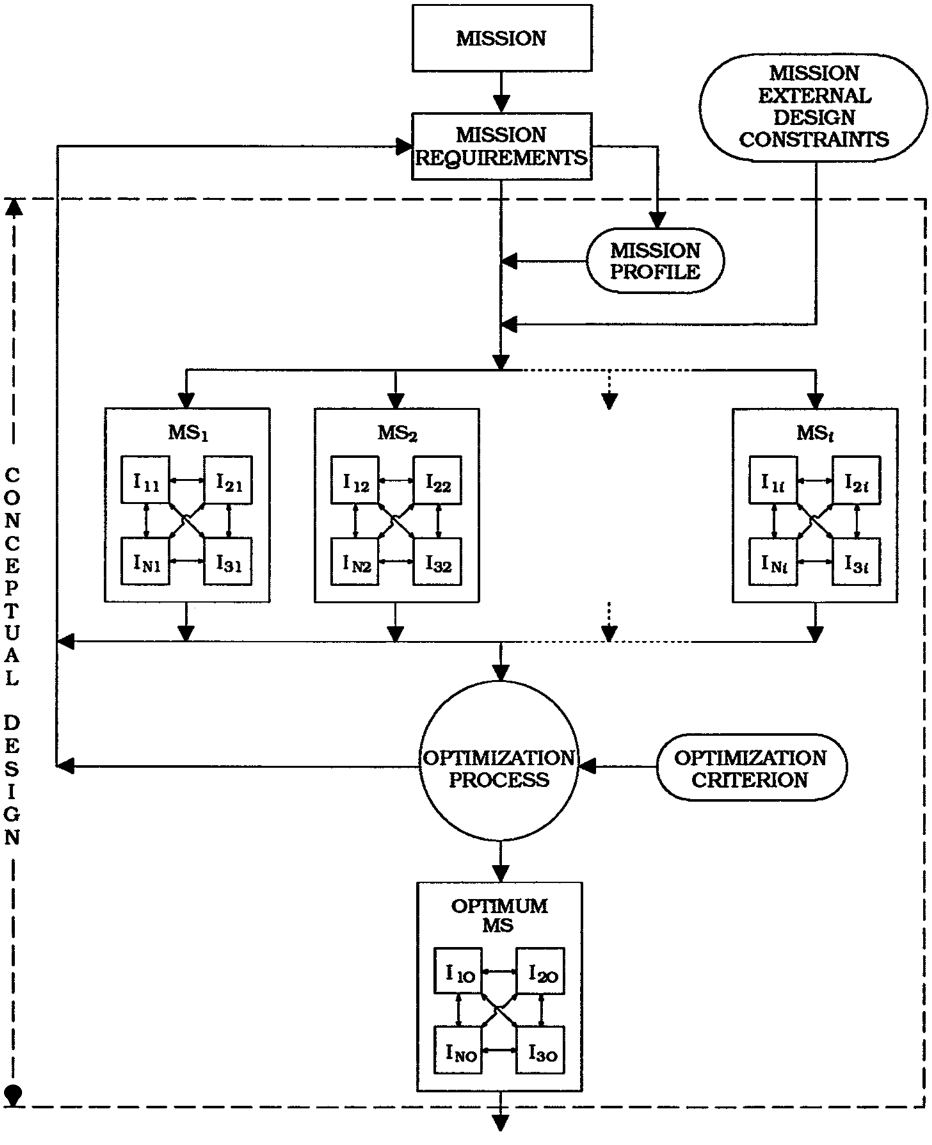


Fig. 4 Conceptual design of mission system

requiring short response times, identified the number, type, and certain capabilities of (I) systems needed to serve the submersible transportation function. In this instance, four types of (I) systems, a special flatbed trailer, an aircraft, a surface ship, and a submarine, were necessary to provide land, air, sea-surface, and sub-sea transportation. *Status* refers to the *new construction* or *existing state* of the (I) systems and whether they are to be *owned*, *leased*, or *chartered*—all options affecting the "operating cost per year" term of the cost-effectiveness equation in Fig. 2.

Figures 3 and 4 show a "feedback line" from conceptual design to the mission requirements which indicates that certain variations in performance requirements may require changes in certain mission requirements, a procedure which should involve consultations between the user/owner and designer. The "feedback line" is somewhat more detailed in Fig. 4, which shows two inputs to this line, one from the mission system alternative "exit line" and the other from the optimization process. In the first instance, it will be recalled that all of these alternatives must meet performance requirements *and* be technically feasible from the producibility/availability and operability points of view. Thus, in forming these alternatives, it may be found that certain performance requirements cannot be met for technical feasibility reasons—that they must be altered to make the alternatives in question viable. In the second instance, cycling the alternatives through the optimization process in searching for the optimum mission system(s) may reveal that changes in some performance requirements will result in a better solution to the design problem from the cost-effectiveness view. In either case, these changes may also necessitate changes in the mission requirements so that they can be met by the altered set of performance requirements.

The other "exit line" from the optimization process, as indicated in Fig. 4, leads to the optimum mission system(s) identified by this process in a manner suggested by the cost-effectiveness equation of Fig. 2. The conceptual design(s) of the selected mission system(s) can now be completed by developing the primary characteristics of the (I) systems not considered in the feasibility studies. The identification of more than one optimum conceptual design means that the "optimization curve" is reasonably flat over a limited range of alternatives—that there is little to choose between them. In this instance, the user/owner may select one of them based on subtleties not heretofore considered, or more than one may be carried forward into preliminary design. The mission system(s) carried forward should have associated performance requirements firmed up and provide baseline (I) system characteristics which are further developed and refined during the preliminary design stage.

The conceptual design of the submersible system, which, in general, proceeds as described above, is discussed in Section 5. Approaches and techniques applicable to the designs of both the submersible and its mission systems are discussed in that section.

Preliminary design, as noted, is the second stage of basic design and is concerned with this phase's fifth function. Starting with baseline data provided by the selected conceptual design(s), it refines and firms up the major characteristics of (I) systems, the MIDCs they exert on each other, and cost estimates associated with them. In summary, it provides a precise engineering definition of the mission system and assurance that the design goal can be attained.

Figure 3 shows a "feedback line" from preliminary design to the mission requirements, indicating that refinements in the design may reveal technical-feasibility reasons for altering performance and, consequently, mission requirements that were overlooked in conceptual design. These discoveries should be few in number and minor in their effect on the requirements. A major discovery of this nature would, most likely, raise doubts about the validity of the selected conceptual design(s) and be cause for repeating the conceptual design stage or terminating the design altogether.

Preliminary design of the submersible system proceeds, in general, as discussed above. It is discussed further in Section 5.

Contract design requires yet further refinement of design and additional detail. It yields contract plans and specifications necessary for interested parties to bid on the construction of new (I) systems or the alteration of existing (I) systems. It also provides contractual documents for the construction and alteration work.

Specifications delineate quality standards of material and workmanship as well as setting forth performance expectations for the (I) systems and their subsystems. They also describe tests and trials which must be performed successfully before acceptance of the systems.

Detail design is the final phase of the design process and entails the development of detailed working plans from which the (I) systems are constructed or altered. In one sense, it is really not a design phase since all the creative design effort is done in the preceding phases, the design being unequivocally defined prior to entering this phase. It does, however, require the greatest amount of work of all the phases and is often undertaken by entities building or altering the system.

The post-design process—This process is composed of three activities: *construction/alteration, test and evaluation,* and *operation.* Although this publication is not directly concerned with these activities, there are two reasons for including them in this overall perspective: (1) They constitute important sources of input to new designs and feedback to the design in question, and (2) they are subject to certain existing MEDCs and will influence the future impact of MEDCs on following designs.

Regarding (1), post-design experiences may reveal that items composing the (I) systems fail to meet or exceed design expectations. Failure in this regard results in the redesign of the offending item or, at least, a lowering of confidence in it which may lead to restrictions being placed on performance. For example, the maximum operating depth of a submersible may be made shallower than stated in the performance requirements due to the discovery of metallurgical deficiencies in the pressure hull's material. Meeting or exceeding expectations, of course, raises confidence in the item which effectively documents the soundness of the current design in this regard and encourages its use in future designs.

Regarding (2), the principal MEDCs of interest here are in the legal/quasi-legal category—rules and regulations of classification societies and various federal agencies including certifying authorities. These MEDCs impact on all aspects of the design and post-design processes. Post-design activities furnish feedback to these MEDCs, causing them to be maintained or altered as experience warrants. In this sense, then, the post-design process is also a source of input to future designs through its influence on future MEDCs.

3. The Submersible System

3.1 Overview

An appropriate introduction to the basic design of manned submersibles requires an elaboration of Fig. 1—specifying and briefly describing (S) systems composing the submersible. No standardized approach to this process exists. Submersible design organizations label and assemble the (S) and subsystem "boxes" within the submersible system's boundary in numerous ways. Without deprecating any approach, the one used herein parallels in composition and terminology the U.S. Navy's SWBS (Navships 009-039-9010) to the extent feasible for submersible design. The primary reason for this choice is that this system is also used in Chapter V, "Hydromechanical Principles," for weight-accounting purposes. The SWBS is discussed in that chapter.

Paralleling SWBS for this purpose leads to the identification of seven (S) systems. They, together with the major systems comprising them, are summarized in the following outline:

1. Hull Structure System
   a. Structural System
   b. Special Purpose System
2. Propulsion Plant System
   a. Energy Generating System
   b. Propulsion System
   c. Special Purpose System
3. Electrical Plant System
   a. Electrical Power Generation System
   b. Power Distribution System
   c. Lighting System
   d. Special Purpose System
4. Command and Surveillance System
   a. Command and Control System
   b. Navigation System
   c. Communication System
   d. Surveillance System
   e. Special Purpose System
5. Auxiliary System
   a. Human System
   b. Life Support System
   c. Air, Gas and Miscellaneous Fluids Systems
   d. Submersible Control System
   e. Mechanical Handling System
   f. Special Purpose System
6. Outfit and Furnishings System
   a. Fittings System
   b. Hull Compartmentation System
   c. Preservatives and Covering Systems
   d. Furnishings
   e. Special Purpose System
7. Armament System

Several of the above (S) systems have components whose weights vary during a particular dive as well as those whose weights remain constant, or fixed, during the dive. In design weight-accounting procedures, the variable components are listed separately from their systems' fixed-weight components—this list often being called the *load to submerge*. A summary follows to provide examples of components included in the load to submerge, accompanied by the caveat that the components actually utilized vary from design to design.

1. Variable load
   a. Human systems and effects
   b. Expendable items—fuel, oxidants, lube oil, fresh water and supplies
   c. System operating fluids which vary during the dive owing to compression/expansion
   d. Mission system variable solid weights—weights on/off loaded during the dive
2. Variable ballast/trim
   a. Seawater
   b. Solid weights—used to compensate for items of mission system weights on-loaded during dive
3. Main ballast
   a. Seawater
4. Descent/ascent weights
   a. Seawater
   b. Solid weight—single weights or pellets

Under variable load, it can well be argued that human systems, persons on-board, represent fixed weights during a dive, as indeed they do. However, these systems are not considered as such by SWBS because their weights are accounted for as items of variable load. Although human systems are included herein because of their pertinence to the presentation of manned submersible design, the SWBS practice is continued in the interest of consistency.

The above (S) systems all have SWBS titles. However, several of their systems and smaller systems do not have these titles in the interest of facilitating a discussion of them pertinent to submersible design. Nevertheless, the reader should easily be able to relate their titles to SWBS groups if the need arises.

The following sections briefly discuss the (S) systems and the systems composing them. Other chapters of this publication elaborate in detail on this discussion.

3.2 Hull Structure System

Several SWBS groups compose this system. They can be combined into the *structural* and *special purpose systems.*

Structural systems—All structural elements composing the submersible are contained within this system. In brief, they can be divided into the *pressure hull* and *external structure.*

The pressure hull is the main, pressure-resisting shell structure of the submersible. It houses all human systems as well as instrumentation and equipment to which they must have direct access. This hull is usually a ring-stiffened cylinder, closed with hemispherical heads, for shallow-depth submersibles although other geometries may be used. An unstiffened sphere, or combination of spheres, is used for deep-depth submersibles because of the structural efficiency of this shape. Penetrations of the pressure hull include access hatches, viewports or panoramic windows, and through-hull fittings for electrical connectors and mechanical shafting. Usually, no bulkheads or major tankage are within this hull, although there might be for very large submersibles.

External structure consists of the *exostructure*, *foundations*, *outer hull*, if used in a design, and *structural appendages*. The exostructure supports all external items and, in some instances, the pressure hull itself. Its framing system carries and distributes loads to which the submersible is subjected during handling and operations. Foundations, attached to the exostructure, provide direct support for system components.

The outer hull is composed of relatively thin metal or fiberglass plating and a supporting framing system. Its surface, alone or in combination with the exterior surfaces of the pressure hull, tankage, and/or fixed buoyancy material, forms the envelope surface on "enclosed envelope" submersibles. This type of envelope, as contrasted with "non-enclosed envelopes" sometimes used, provides a fair surface required to reduce drag at higher speeds, to minimize the dangers of entanglement and mud/silt entrapment and to protect enclosed systems.

Structural appendages are those structural items extending beyond the enclosed form of the hull or beyond the principal dimensions of the exostructure in the case of an open-frame submersible. Examples include the fairing or "sail" protecting the access hatch and structural castings for the stern planes and rudder—the latter two appendages themselves are parts of the ship control system.

Special purpose system—Fixed ballast and *fixed buoyancy* are included in this system. Items of ballast and buoyancy remain fixed during a particular dive. They may be altered in magnitude or location or both between dives to compensate for changes made in various items of weight between dives.

Fixed ballast, such as pig lead, is used to achieve equality of total submerged weight and displacement when, otherwise, displacement would exceed weight, a situation often encountered in the design of shallow-depth submersibles. Fixed ballast is also used to obtain satisfactory stability and trim and to provide for a design weight margin. For very large submersibles, as for submarines, it may also be used to compensate for weight alterations made to the vehicle during its lifetime.

Fixed buoyancy, such as syntactic foam or hollow spheres, is used to achieve equality of total submerged weight and displacement when, otherwise, weight would exceed displacement, a situation which is always encountered in the design of deep-depth submersibles. Like fixed ballast, it is also used to achieve satisfactory stability and trim. It may also be the means of compensating for weight alterations made during the submersible's lifetime.

3.3 Propulsion Plant System

The propulsion plant is composed of all systems required to move the submersible in the ahead and astern directions under normal or emergency conditions. The maneuvering system, to be discussed later, is included in this (S) system, but only to the extent it is used for propulsion as well as maneuvering. To the extent that it is used exclusively for maneuvering, SWBS considers it to be a part of the Ship Control System within the Auxiliary System.

Energy Generating System—Electrical energy is used for propulsion and many other purposes on a submersible. In this regard, SWBS divides the total energy requirement into two parts—energy for propulsion and energy for all other needs, the latter being considered within the electrical plant system if it is used exclusively for nonpropulsion purposes.

Submersibles may be supplied with electrical energy for propulsion and other needs from sources stored on supporting (I) systems or on board. In the first instance, energy is received through an electric cable which is a part of the tether linking the submersible and support systems. In the second case, a primary or secondary energy source must be stored on board. Examples of these sources include, respectively, chemical energy stored in fuel oil and electrical energy stored in batteries. Energy generating systems involving energy stored in its primary form must utilize either open- or closed-cycle energy conversion systems. Open-cycle systems, such as diesel-generator sets, require access to the atmosphere's oxygen and, hence, can only be operated on the surface for propulsion and for charging the batteries that supply energy when submerged. Closed-cycle systems require both oxidants and fuel to be stored on board, thus permitting system operation while submerged and thereby obviating the necessity for batteries as the main energy source. Stirling engine-generator sets and fuel cells are examples of this system.

The vast majority of untethered submersibles store energy on board in its secondary form in lead-acid, silver-zinc, or nickel-cadmium batteries located outside the pressure hull in pressure-resistant or pressure-compensated containers. The choice between them depends on energy density, battery characteristics, and cost trade-offs. In this system, the primary energy source and energy conversion systems are carried on board the support ship with the submersible's batteries being charged between dives.

Propulsion System—This system may be subdivided into *propulsion thrusters, drives*, and *support systems.*

Propulsion thrusters most commonly used are free or nozzled propellers, although other thrust-producing devices have been proposed and occasionally used such as water jets and cycloidal propellers. Free propellers may have relatively large diameters, which improves system efficiency over a range of speeds. However, the unprotected blades are subject to damage and entanglement. Nozzled propellers have relatively small diameters, but performance is enhanced for high-thrust, zero to low-speed situations which characterize the operation of most submersibles. The nozzle also affords protection for the propeller.

Propulsion drives consist primarily of ac and dc electric motors and electro-hydraulic units, listed in order of decreasing efficiency. The dc motor drives have been used predominantly and electro-hydraulic drives only occasionally. As compared with ac motors, dc motors are more complex, are limited in power output, require more maintenance, and need protection from seawater. On the other hand, they have better speed control, produce higher torque, and can be operated directly from the battery—that is, a dc–ac inverter is not required, thus saving weight, space, and cost. Most propulsion drives are located outside the pressure hull to avoid mechanical, through-hull penetrations and seal problems. The motors are enclosed in pressure-resisting or pressure-compensated containers, the latter option reducing sealing difficulties.

Support systems include all electrical, hydraulic, and mechanical power transmission and conversion items used for propulsion and the propulsion control system. Electric transmission and conversion items include cable from the battery to the motor and an inverter, if required. These items for hydraulic systems include piping, flexible tubing, and motor-pump units. Mechanical transmission items include shafting, reduction gears, clutches, and couplings where these items are not combined to form self-contained thruster units.

Special purpose system—This system is composed of propulsion-plant operating fluids, spare parts, and tools. Operating fluids include air, hydraulic oil, oil in the battery compensating systems, and battery electrolyte.

3.4 Electrical Plant Systems

The electrical plant is composed of systems necessary to provide and distribute electric power to meet all of the submersible's power needs, excluding propulsion, and to provide for lighting. In this regard, the nonpropulsion electrical power needs are referred to as the "ship's service load" or "hotel load."

Electrical power generation system—This system may be divided into the *main* and *emergency subsystems*. The main subsystem is composed of energy conversion systems, such as auxiliary diesel-generator sets, and battery packs which are used exclusively to meet ship's service loads. If they are not used exclusively for this purpose but also for propulsion, they are included in the propulsion plant's energy generating system. The previous discussion of these systems is applicable here.

The emergency electric power generating subsystem is always located on board both tethered and untethered submersibles, separate from and independent of the main system in untethered vehicles. It serves critical systems in case of main power failure. Examples include primary and secondary batteries located as close to the system they serve as possible. The simplest illustration is the primary battery in a flashlight used for emergency lighting.

Power distribution system—This system provides for the distribution of power to all nonpropulsion systems on the submersible. It includes the main cable wireways, emergency power cable system, switching gear and panels, and so forth.

Lighting system—The lighting system includes lighting distribution and fixtures, excluding the signal, anchor, and navigating lights, which are in other systems.

Special purpose system—This system includes electrical plant operating fluids, such as the electrolyte in the battery, and the plant's spare parts and tools.

3.5 Command and Surveillance System

The nature and extent of systems composing this system vary greatly depending on demands placed on them by mission performance requirements. They vary from relatively simple systems composed of a few components, for the less demanding missions, to extremely complex systems composed of numerous, technically sophisticated components for highly demanding missions. This fact must be borne in mind by the reader for the following text.

Command and control system—This system contains subsystems facilitating command and control of the submersible. Included are data display and processing systems, digital data switchboards and digital data communication systems, command and control testing systems, and command and control analog switchboards.

Navigation system—Safe and efficient surface and submerged operations heavily depend on strongly coupled functions of navigation, searching, obstacle avoidance, and communications. The first three functions are of primary concern for submerged operations and are the responsibility of the navigation system. The fourth function is important in surface, as well as submerged, operation, brief though it may be. Though closely allied with navigation, communications is in a separate system, to be considered shortly.

Navigation subsystems operate independently or interact with other (I) systems of the mission system. Some examples of independent systems are the onboard magnetic compass, gyro compass and fathometer; examples of interacting systems include the submersible's transponder/pinger interacting with seafloor-based transponders or a surface ship's transceiver/hydrophone system or both. These systems may be subdivided according to general types, with examples, as follows:

(1) Electrical—navigation lights and gyrocompass.
(2) Electronic—radio navaids, such as Loran C and Omega, and radar navigation systems.
(3) Acoustic—fathometer and sonar systems such as obstacle avoidance sonar and short- and long-baseline navigation systems.
(4) Visual/acoustic—acoustic imaging systems.
(5) Visual—direct visual through viewports and windows and indirect visual using periscope and fiber-optic systems.
(6) Other—magnetic compass, depth gauge, trim indicator and chronometer.

Communication system—This system is divided into *interior* and *exterior communication subsystems*. Interior subsystems facilitate exchange of information between persons on board the submersible and include sound-powered telephones, announcing systems, alarm systems, and switchboards for these systems.

Exterior subsystems provide for communication between persons on board the submersible and those on other (I) systems, usually those on a surface support ship or working as divers outside the submersible. Tethered submersibles invariably use "hard-wire" telephone systems for both submerged and surface communications. Untethered submersibles use acoustic underwater telephones while submerged and, while on the surface, use radio, "plug-in hard-wire" telephones, and visual and audible systems.

Surveillance system—This system is of primary interest for marine vehicles having military-oriented missions including, of course, submarines. To the extent that it may be utilized on some submersibles, the system is subdivided into *surface* and *underwater subsystems*. Surface subsystems include surface and air search radar; underwater subsystems include active, passive, active/passive, and classification sonar.

Special purpose system—This system is comprised of electronic test and monitoring equipment, system operating fluids, spare parts, and tools.

3.6 Auxiliary System

The auxiliary system is composed of numerous systems which, except for the human and life-support systems, fit within specified SWBS groups and are given the same titles as these groups. As has been noted, the human system is not included in SWBS as a system. The life-support system is scattered among several SWBS groups. It is treated as a single system herein in the interest of clarifying the discussion.

Human systems—In the broadest sense, these systems include all persons *on board* and *external to* the submersible who are responsible for its safe and effective operation and who utilize its services in accomplishing mission tasks. Although on-board persons are of primary concern in submersible design, it is important here to acknowledge that external persons also have some influence on the design. Many persons could be included but, to make the subject manageable, these persons can be called, collectively, the *submersible crew* to *distinguish* them from the "ship's crew," the "scientific party," the "diver group," and so forth. The submersible crew accompanies the submersible at all times during operations. It is composed of one or more operators and mechanical, electrical/electronic, and acoustical technicians. The technicians maintain their vehicle in a ready-to-dive condition. The designer should keep in mind their abilities and the rigors of working at sea in matters pertaining to the simplicity, ruggedness, and accessibility aspects of the design.

On-board persons can be categorized as *crew* and *noncrew*. Crew, or pilots as they are usually called, are the equivalent of ship captains, being in command and having responsibility for all aspects of the submersibles' operations. One or two pilots are on board, depending on the mission's duration, and are always under about one atmosphere of pressure. Noncrew include all other persons on board under one-atmosphere or ambient-pressure conditions in the case of diver lock-out vehicles. Their training must include a working knowledge of the submersible's emergency features.

On-board persons, as human systems, affect submersible design and create MIDCs by virtue of their physiological and psychological characteristics. Many of the quantifiable characteristics are given as so-called "standard man" data. This system's effects on design can be viewed from the perspective of three considerations: *life protection*, *comfort*, and *support*—the latter being considered as a subsystem of the

Auxiliary System. Though there is some overlap between these considerations, a separate discussion of them ensures that all effects of "human systems" on design are included.

Briefly, life protection implies protection of the human system from the sea environment, under one-atmosphere or ambient-pressure conditions, and from the detrimental or dangerous characteristics of other systems. Examples include the necessity to enclose human systems in pressure hulls, the location of potentially dangerous systems (such as batteries and high-pressure air systems) outside the pressure hull, and the need to incorporate escape or rescue systems into the design.

Life comfort has to do with the productivity of the human system existing in a foreign environment, a consideration which becomes increasingly important as the mission duration increases. The "habitability" or "human-engineering" aspects of design address this consideration. It is one of the basic factors in determining the size and shape of the pressure hull through cubic space per person and dimension requirements. It also influences, for example, the pressure hull's internal arrangement, decor, noise attenuation measures, and atmospheric control, a subject included in the next system to be discussed.

Life support system—This system can be divided into *atmospheric, food and water*, and *waste-management subsystems*, the components of which are scattered among several SWBS groups in the Auxiliary and Outfit and Furnishings systems. Briefly, the atmospheric-management system is responsible for the proper breathing gas composition, carbon dioxide removal, contaminant removal, atmospheric monitoring, and temperature/humidity control. Managing ambient-pressure environments is more complicated than managing one-atmosphere environments owing to the use of helium-oxygen, rather than air, as the breathing gas. Emergency provisions are made for two situations: a prolonged submerged period during which the atmospheric-management system continues to function normally and a second situation in which components of this system either fail or are unable to cope with an on-board casualty. The first situation is met by simply providing a specific margin of life-support gases and other materials. For example, it is usually specified that a short-mission-duration (eight to ten hours) submersible have enough of these materials on board to remain submerged for 72 hours. The second situation is met by components such as face-masks connected to a separate source of breathing gas and emergency carbon dioxide absorbents.

Food and water management systems used depend primarily on the mission duration. On short-mission, eight- to ten-hour-duration submersibles, food and water is brought on board in portable containers such as lunch boxes and canteens. Despite the informality of this system, it is important that it also include some kind of emergency rations which will last as long as the emergency atmospheric management materials.

This system increases in complexity as the mission duration and number of on-board persons increase. In long-endurance (say, 15 to 30 days) submersibles, a food system similar to that employed by airlines is used—prepared meals are stored on board and heated. Consequently, both a cold food storage space and a heating device must be provided. Freshwater tankage must also be provided for potable and shower water if ambient-pressure systems are included in the design. In the latter instance, the on-board capacity of freshwater is enormously increased because hot showers are the primary means of warming divers after extra-vehicular excursions.

The waste-management system, quite naturally, parallels the food and water subsystem. For short-mission-duration submersibles, human and other wastes are kept in portable containers. On long-duration submersibles plumbing fixtures, holding tanks, and trash bins must be provided; nothing is discharged into the sea environment.

Air, gas, and miscellaneous fluids system—This system includes, essentially, the compressed air and hydraulic fluids subsystems. Compressed air may be used for several purposes, although its primary use is to discharge water from the main ballast tanks. It is stored under high pressure in flasks located outside the pressure hull, the air lowered to safe, usable pressures for specific purposes by reduction valves. Air flasks, like secondary batteries, are also secondary sources of energy, in this case, "pneumatic energy." They must be recharged by air compressors which are usually located on the support (I) system.

The hydraulic-fluid subsystem transmits power to all systems except propulsion. The rudder, stern planes, and manipulator are examples of systems which may be hydraulically powered.

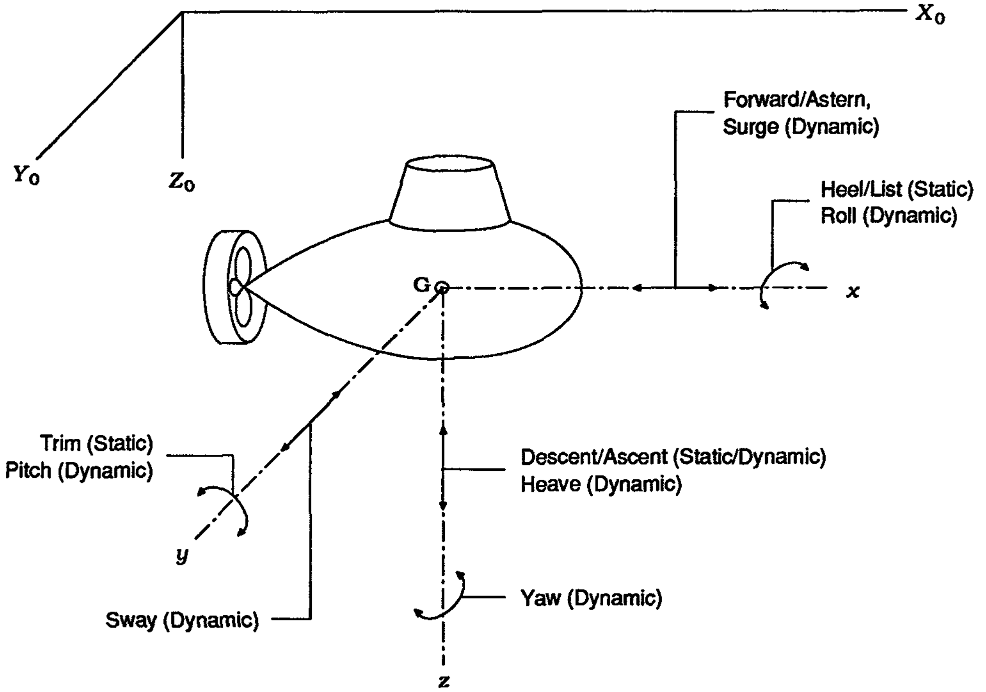
Submersible control system—This system is composed of *static* and *dynamic control subsystems*. Maneuvering the submersible is their function, "maneuvering" being defined as the *controlled change or retention of the vehicle's motion or position*. In this regard, Fig. 5 shows a submersible in hydrospace with reference to X_0, Y_0, Z_0 earth axes with x, y, z body axes attached to the vehicle. The extent of a submersible's maneuvering ability is called *degrees of freedom*. It is possible to have a total of six degrees of freedom, three linear degrees in the x, y and z, or Z_0 directions and three rotational degrees about these axes. Most submersibles have four to six degrees of freedom, achieved by a great variety of static and dynamic control systems. Consequently, these systems are discussed in general, rather than specific terms.

Static control subsystems employ static forces of weight and displacement in accomplishing their functions. They are effective in three degrees of freedom: *descent/ascent* in the Z_0 direction, *trim and buoyancy control* about the y-axis and in the Z_0 direction, respectively, and list about the x-axis.

Descent/ascent may be accomplished by the *main ballast* and *descent/ascent weight systems*, to the extent that they are used in a design. Use of a main ballast system enables the submersible to float freely on the surface before diving with a certain amount of its displacement-producing volume, called *volume of reserve displacement*, above the waterline. Complete flooding of the main ballast tanks reduces the volume of reserve displacement to zero. Other means must be used to cause the vehicle to sink in the water column. Usually, descent weights are used in the form of additional water ballast or solid weight. The water is expelled by high-pressure air or the weight dropped just prior to reaching the desired depth. Deeper-depth submersibles must use a droppable, solid weight because of the difficulty of expelling water against high hydrostatic heads. Droppable weight, in the form of steel pellets, can be released gradually to control the rate of descent and thus is often used. If the total submerged weight and displacement are equal at the desired depth, the submersible is said to be in a condition of *neutral buoyancy*, a basic design condition for most untethered submersibles. Ascent may be initiated by expelling water from the main ballast tanks or by dropping an ascent weight, the latter method being used for deep-depth submersibles. Ascent weight is always some form of solid weight or steel pellets which can be released gradually to control the rate of ascent. If both systems are used, main ballast is not blown until the submersible is at or near the surface.

Descent/ascent, of course, may be inclined as well as vertical using propulsion thrust and planing forces to help "drive" the vehicle down or up. For deeper-depth submersibles, this is a high energy-consuming procedure and is usually not used, at least for untethered vehicles. A few designs have also used a spiral descent/ascent path achieved by eccentric location of the descent/ascent weights.

An emergency-ascent system is also incorporated into most designs. These systems may involve the release of the pressure hull from the rest of the



AXES — X_0, Y_0, Z_0 — Earth Axes (fixed to earth)
x, y, z — Body Axes (fixed in submersible - origin at center of gravity (G) or other point)

CONTROL — Static Control Systems: Effective in three degrees of freedom
Dynamic Control Systems: Effective in six degrees of freedom

Fig. 5 Degrees of freedom—static/dynamic control systems submersible or the release of some large item of weight such as the battery pack or droppable keel.

Trim and buoyancy control involve the *trim* and *variable ballast/buoyancy systems*, whose functions are to create or prevent trim and an imbalance between total submerged weight and displacement respectively. This imbalance is called *negative buoyancy* if weight exceeds displacement and *positive buoyancy* if displacement exceeds weight. For example, the accomplishment of many mission tasks is facilitated by purposely creating trim fore or aft or negative buoyancy so that the submersible "sits hard" on the bottom. On the other hand, unwanted trim and positive/negative buoyancy can be prevented by these systems if within their capabilities, a process called compensation.

The trim system accomplishes its functions by shifting weights fore and aft to generate trimming or trim-compensating moments. These weights are contained within the system and, hence, are not considered variable loads. They include pumpable fluids, such as mercury, movable trim weights, and movable weights of another (S) system such as the battery pack. It should be noted that if mercury is used, the system must be outside the pressure hull since this material is exceedingly dangerous to humans.

The variable ballast/buoyancy system makes adjustments either in weight, by admitting or expelling seawater from variable ballast tanks, or in buoyancy by increasing or decreasing displacement of oil-inflatable/deflatable bladders. Some submersible designs may have reason to use automatic depth control and "hovering" systems. The latter system precisely maintains neutral buoyancy, allowing the submersible to remain suspended in the water column.

The trim and variable ballast/buoyancy systems may be either separate but closely coupled systems or be combined into one system. Small submersibles use separate systems. Large submersibles may use one system, similar to that used on submarines, which employs interconnected/sea and fore, aft, and mid-length (auxiliary) trim tanks. Just fore and aft tanks may also be used.

List involves the *list system*, a system similar to the trim system in composition and functions except that list weight is moved transversely to generate listing or list-compensating moments. Usually, the list weight is mercury. The weight of items in other systems is not used for this purpose.

Dynamic control subsystems employ dynamic forces of thrust and lift and it is convenient to refer to them as *thrust-* and *lift-producing* systems. Thrust-producing systems, such as small motor/screw propeller self-contained units, can provide the submersible with as many degrees of freedom as required. These degrees of freedom are referred to in the dynamic sense as *surge*, *sway* and *heave* in the x, y, and z-axis directions and *roll*, *pitch*, and *yaw* about these axes. In general, lift-producing systems, such as the stern planes and rudder, provide two degrees of freedom—pitch about the y-axis (dive/rise inclinations) and yaw about the z-axis (steering). Additional degrees may be achieved, say, for example, in the z or Z_0 directions by the combined use of stern and forward-located (bow or sail) planes.

The thrust-producing system's functions include the controlled change or retention of a submersible's motion at zero to low speeds. In performing these functions, it may either augment or replace certain static control systems. Zero speed, of course, implies position change or retention at specific X_0, Y_0, Z_0 coordinates in the water column. As noted, position changing is necessary to accomplish many mission tasks. Position retention, say, in a current field exerting forces on the submersible, is an equally important ability. Low speed implies that this system is effective in performing its maneuvering functions at relatively low speeds only—say, at speeds up to a few knots.

Thrust-producing systems for maneuvering parallel those used for propulsion; self-contained, motor/screw propeller units are the most prevalent. These units may be fixed or rotatable, operating in the open or enclosed by shrouds or ducts penetrating the submersible's hull. The design objective is to use the minimum number of these, or other, units required to achieve the desired degrees of freedom and to have them located so as to reduce cross-coupled, or induced, motions to a minimum.

The lift-producing system's functions include the controlled change or retention of a submersible's motion at higher speeds—speeds in excess of a few knots. Higher speeds are necessary, of course, to provide the rapid waterflow over control surfaces required to generate lift forces. The design of this system is intimately associated with the hydrodynamic design aspects of the submersible as a whole. In this regard, performance requirements place emphasis on submersible characteristics which facilitate either the change from or retention of, say, horizontal, straight-line motion. For example, these requirements may stress the need for frequent path changes, accomplished quickly with control forces and moments of reasonable magnitudes—one case being the quick entrance into, as well as exit from, a turning circle of small diameter. Such a submersible would possess a minimum level of dynamical stability[1] and ample movable control surfaces at the stern. On the other hand, the retention of horizontal, straight-line motion would be stressed in missions requiring the submersible to travel between Points A and B with little or no path deviations in between. In this instance, the submersible would possess a higher level of stability. If the vehicle's "bare hull," or unappended body, lacked this stability by

[1]Dynamical stability and associated equations of motion are subjects included in Chapter V, "Hydromechanical Principles."

itself, as it often does, the stability of the submersible as a whole is raised by adding fixed fins or stabilizing surfaces at the stern—in effect, by adding tail feathers to the arrow. These fins, as well as movable control surfaces, are components of this system.

Lift-producing systems include vertically oriented rudders, horizontally oriented planes at the stern and often at the bow, "X" configured rudder/plane combinations, and stabilizing fins. Rudders and planes may be all-movable or partially-movable control surfaces made of flat plates or having NACA sections and profiles. Most movable control surfaces are hydraulically operated.

The propulsion thruster may also be involved in maneuvering. For instance, in one system the motor/propeller unit is rotated about a vertical axis to provide angled thrust for steering. In another system, a shroud surrounding the propeller can be tilted, thereby deflecting the propeller race to produce angled thrust vectors.

Mechanical handling system—This system includes *anchor handling* and *storage* and *mooring systems*. Small submersibles usually are not provided with these systems. They anchor by sitting hard on the bottom with negative buoyancy or by attaching themselves to an object with grabber manipulators. Large submersibles may use these systems. The SWBS places handling of the submersible itself in the special-purpose systems.

Special purpose systems—This group is composed of *handling*, *rescue* and *salvage*, *fluids/spare parts and tools*, and *mission systems*. The first two systems, in effect, are small parts of larger systems with the same titles, most of their components being on other (I) systems or external to the submersible. The external components are reviewed prior to discussing the system on the submersible.

Handling systems include those involved in handling the submersible during its *transportation*, *launch and recovery*, and *towing* to the extent that these activities are provided for in the design of the mission system. Transportation may include landborne, airborne, and seaborne surface and subsurface (I) systems, whatever they might be. Handling systems on these systems may include, for example, cranes, winches, ramps, rails, tiedown fittings, and shock-absorbing mounts.

Launch and recovery of a submersible may be accomplished on the surface or remote from the surface, at a depth where surface-wave action is negligible. The surface launch and recovery system is based on the surface support ship and is one of several types, including stern-mounted A-frames, articulated- or nonarticulated-boom cranes, overhead rail cranes, docking-well cradle lifts, and ramps. Remotely, these activities may be conducted by heave-compensated elevator systems suspended from a surface ship, underwater platforms, and launch and recovery systems on submarines.

Towing systems are used primarily for large submersibles which cannot be carried on the deck of a ship. They may also be used for small submersibles in unusual circumstances. A towing winch and cable are primary components of this system.

A submersible, large or small, cannot be burdened with weight and space requirements of many of the handling systems' components for the three activities. Consequently, they are held to a minimum—including, for example, padeyes, fairlead fittings, special socket fittings for launch and recovery, and shock-absorbing mounts. A design for launch and recovery may use a single-point lift, multi-point lift, or cradle system in interfacing with the external components of this system. As implied, single- and multi-point lift submersibles are lowered and raised by cable attachments at one or more points on top of the submersible. Padeyes or socket fittings located at these points are securely fixed to the exostructure or pressure hull which distributes lowering/raising, as well as other handling, loads. In cradle systems, these loads are introduced into the exostructure by bottom skegs or another structure of the submersible.

Rescue and salvage systems may involve interrelated or separate activities. Rescue systems have the function of extricating persons from a submersible unable to surface and from which they are either unable to, or elect not to, escape. Salvage systems have the function of retrieving the submersible. Rescue may be accomplished through salvage, persons remaining on board while the submersible is being brought to the surface.

Rescue and salvage systems on a submersible, as noted, are also small parts of much larger systems. External components of these systems include dedicated rescue and salvage ships such as the U.S. Navy's Auxiliary Submarine Rescue (ASR), ships of convenience (any ship available which can accommodate other system components), one-atmosphere and ambient-pressure rescue chambers, other manned submersibles including dedicated rescue submersibles such as the U.S. Navy's Deep-Submergence Rescue Vehicles (DSRVs), unmanned submersibles or Remote-Operated Vehicles (ROVs), and divers and diving facilities.

The extent of rescue and salvage systems on a submersible varies with its size. A small submersible, for instance, cannot tolerate the weight of a heavy skirt and associated structure required for the mating of a rescue bell or another submersible to make a dry transfer of personnel. A wet transfer, from submersible to rescue vehicle through the water, may be used for personnel at ambient pressure on board the striken submersible, although this practice is not considered desirable. Consequently, for small submersibles, rescue is usually effected by salvage. The function of rescue/salvage systems on board the vehicle is to facilitate these operations. These systems include components from other submersible systems as well as their own; underwater telephones, pinger/transponders, marker buoys, external lights, and salvage padeyes are examples of such systems.

A large submersible can usually afford, in terms of weight and space, to carry more extensive rescue/salvage systems. In addition to components mentioned, its design may include a mating skirt, permitting dry transfer of personnel, and salvage connections and fittings. In this case, rescue and salvage may be separate events. A large submersible may also be able to afford the luxury of carrying an escape capsule which, like the pressure hull in some designs, can be released from the rest of the structure. In both instances, rescue occurs on the surface.

Fluids/spare-parts and tools include those items which are associated with the auxiliary system. Fluids of this system whose weights remain fixed during the dive include, for example, mercury in the trim/list systems and oil in the variable buoyancy system. As noted in Subsection 3.1, the weights of these fluids remaining in their systems are included as "system weights." Examples of fluids whose weights vary during the dive are seawater in the main and variable ballast/trim systems and air in the high-pressure air system. Again, as noted, the weights of these fluids are not a part of their "system's weight" but are included in load to submerge.

The mission system is part of the submersible's payload which also includes noncrew and effects. It is composed of items internal and external to the pressure hull which either are fixed or vary during a dive. Fixed items are those which are used exclusively for accomplishing mission tasks, for example, sensors, manipulators, sample bins, coring devices, strobe lights, video equipment, still and movie cameras, and diver support and protection gear. Variable mission loads are items which are on- or off-loaded during the dive such as equipment being transported to a seafloor oil/gas installation. Note that the items of some (S) systems may be used in accomplishing both mission tasks and operation of the submersible, in which case they are not included in the mission system. For example, obstacle-avoidance sonar may be used in accomplishing these tasks, but it remains a part of the navigation system.

3.7 Outfit and Furnishings System

This system, in general, includes a large number of system components required to complete the submersible and to make it habitable, this number increasing as submersible size increases. Following is a listing of these systems and components which does not require extensive discussion:

Fittings include hull fittings, rails, stanchions, lifelines, and so forth.

Hull compartmentation is comprised of items such as nonstructural bulkheads, floor plates and gratings, ladders, and nonstructural closures. The bulkheads and closures are found only on large submersibles.

Preservatives and coatings includes paint, zincs for cathodic protection, deck covering, hull insulation material, and sheathing and fittings for any refrigerated spaces which might be on board.

Furnishings include items of furniture, fixtures, and equipment enabling each space within the pressure hull to serve its function—the functions relating to submersible control, mission task activities, eating, sleeping, and sanitary needs. One space must serve all functions on small submersibles; spaces serving each function may exist on large submersibles.

Special purpose system includes components such as fluids, spare parts, and tools associated with this (S) system.

3.8 Armament System

This system applies only to submersibles having offensive/defensive military missions requiring the launching of weapons. It is included here primarily in the interest of presenting the complete SWBS system. If applicable to a design, it includes components for handling, storage, protection, launching, and guidance of these weapons. The weapons, themselves, are included as an item of variable load in the load to submerge group.

4. Inputs to Submersible Design

4.1 Overview

Inputs to submersible design, as for other (I) systems of the mission system, were introduced in Section 2. As noted, these inputs can be placed into one of two categories: inputs which *furnish guidance* for the design and those which *impose constraints* on this process. They are summarized within these categories in the following outline:

1. Guidance Inputs
   a. Mission/performance requirements
   b. Post-design experience
2. Constraint Inputs
   a. Design criteria
   b. Mission external design constraints
   c. Mission internal design constraints

This section elaborates on these inputs as they pertain to the design of manned submersibles.

4.2 Mission/Performance Requirements

A set of *mission requirements* includes those pertinent to both the design of the submersible and other (I) systems composing the mission system.

They are generated by the prospective owner/user of this system and provide the basic guidance for designing and constructing nonexisting (I) systems and for designing alterations to be made to existing entities to convert them to (I) systems. In this and following sections, it is presumed that the submersible, initially, is a nonexisting system—one to be designed from scratch and constructed.

Mission requirements pertinent to submersible design primarily convey information on tasks to be performed underwater including details of the task sites (location, depth at site, etc.) and the nature of task work (details of work objects such as well heads, data to be acquired, and payload). *Performance requirements* are derived from the mission requirements and set forth the capabilities that the submersible must have to meet the mission requirements (speed, maneuvering, endurance, etc.). These requirements provide the basis for developing the submersible's characteristics and associated (S) systems as well as being primary input to cost-effectiveness considerations. As will be seen, performance requirements are developed during the feasibility studies of the conceptual design and are reasonably firmed up by the end of this stage. They are shown as the first consideration, or "spoke," of the design spiral in Fig. 7. As indicated in this figure by the two-way arrow, changes in these requirements during conceptual design may lead to adjustments in mission requirements to obtain a better solution to the design problem. Although changes are actually developed during the design process, it is more useful here to consider them as design input because of their very close association with the mission requirements.

The relationship between mission and performance requirements is illustrated by two examples involving the mission requirements for (1) task-site depth and (2) details of work objects. Task-site depth leads to the performance requirements for *maximum operating depth*, D_o, and *collapse* or *design depth*, D_c. The term D_o is at least equal to the maximum site depth while

$$D_c = FD_o$$

where F is known as the *factor of safety* which accounts for structural design uncertainties. Its value, usually between 1.5 and 2.5, is determined by the designer or imposed by legal/quasi-legal MEDCs of classifying societies or certifying agencies. Details of work objects relate to several performance requirements including, for example, maneuvering and manipulator capabilities. Maneuvering capabilities may be expressed as degrees of freedom (six degrees total—three in translations and three in rotation) or degrees of trim and list required, say, to attain a favorable submersible/work object orientation to facilitate work. Manipulator capabilities are given as degrees of motion freedom (shoulder, elbow, wrist, etc.) required to work on the object effectively. Manipulator-managed tools must also mate with the work object and be capable of performing the required task expeditiously.

A representative list of mission and performance requirements is given in Subsection 5.3.

4.3 Post-Design Experiences

The design of a new submersible is guided by experiences gained during the construction, testing, and operation of existing submersibles, the extent of this guidance increasing as the similarity between the new and existing designs increases. These experiences, if favorable, are key factors in selecting components for submersible (S) systems from a group of candidates all of which may enjoy state-of-the-art status. As has been noted, these experiences also provide input to the development of legal/quasi-legal MEDCs. As an example, the aforementioned ABS publication "Rules For Building and Classing Underwater Systems and Vehicles" is based in part on experiential input. Past and current experiences are documented in test reports, user reports and dive logs.

4.4 Design Criteria

These criteria, as already indicated, permeate all aspects of submersible design, providing bases or standards for making design decisions and judgments aside from those imposed by other design constraints acting. They may be established by the owner/user, designer, or other entities, two major groups being criteria for design optimization and criteria for selecting the submersible (S) system's subsystems and components. For manned submersibles, optimization of the conceptual design is usually based on either the *cost-effectiveness* or *minimum weight criterion*. These criteria are not necessarily mutually exclusive but do establish priorities for design considerations. For example, a mission requirement for very large payloads may invoke the minimum weight criterion for the design. Consequently, a highly efficient material (high strength/density ratio) may be used for the pressure hull, a material which is also very expensive and one which would probably not be used under the cost-effectiveness criterion. Cost-effectiveness, though, is generally the criterion used and is discussed in detail in Subsection 5.3. The second group of design criteria is closely related to technological areas and considerations pertinent to submersible design. These criteria are, in effect, the bases for technological MEDCs created by the owner/user or designer as well as by other entities such as classifying societies and certifying agencies. They include technological status, such as state-of-the-art, minimum cost and minimum

Chapter II
Characteristics and Development of Submersibles[1]

E. E. Allmendinger and J. A. Pritzlaff

1. Introduction

SINCE ANCIENT TIMES, man has had the desire and the need to penetrate aquatic environments for military, scientific, industrial, and recreational purposes. In pursuit of these activities, he has developed an amazing array of underwater vehicles, which, in general, are referred to as submarines and submersibles. *Submarine* is the term usually reserved for the large, self-sufficient, manned underwater vehicle that has been used almost exclusively for military missions. In contrast, *submersible* is used for the relatively small, manned or unmanned underwater vehicle that is heavily dependent on supporting systems, such as a surface ship, to accomplish peaceful underwater missions.

History, from the fifth century B.C. until fairly recent times, is replete with legendary and factual accounts of underwater vehicles, their builders, and their exploits. Perhaps the underwater adventures of Alexander the Great (356–323 B.C.) provide a logical starting point. One drawing depicts him observing the wonders of the Aegean Sea from a diving bell apparently made of glass (Fig. 1). Despite this ancient precedent for peaceful underwater pursuits, it was not until recent years that underwater vehicles were used extensively for other than military missions.

Submersible development has drawn extensively on submarine technology. Consequently, it is appropriate to review briefly some highlights of submarine history.

One of the earliest underwater vehicles designed for warfare was a leather-covered rowboat built by a Dutch scientist, Cornelius van Drebble, around 1620. It is said that he successfully demonstrated his boat on the Thames River with no less important a personage on board than King James I of England, diving to depths of 3 to 5 m and remaining submerged for a few hours.

More than a century later, the *Turtle* (Fig. 2), built by American colonial David Bushnell during the Revolutionary War, made the first recorded attack by

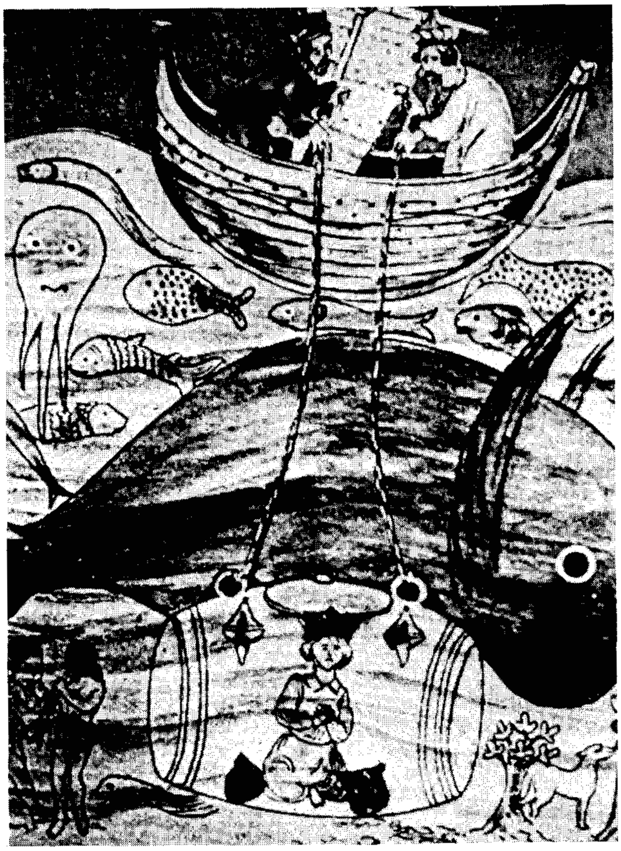


Fig. 1 Alexander the Great observing the wonders of the Aegean Sea from inside a glass diving bell (322 B.C.)

[1]This chapter contains material from an article written by E. E. Allmendinger for the *OCEANUS* magazine of the Woods Hole Oceanographic Institution, which has given permission for its use herein.

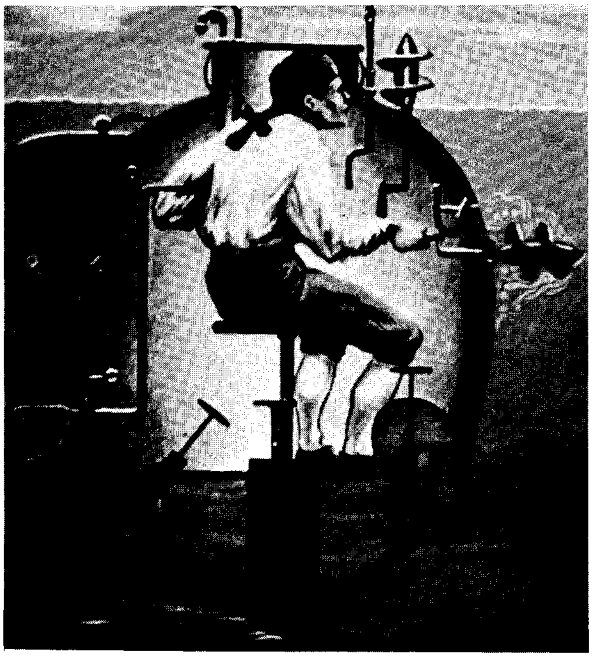


Fig. 2 Bushnell's *Turtle*, built during the American Revolution, carried only one man, who had to crank the propeller, adjust ballast, and steer by looking through tiny portholes

a submersible on an enemy warship, the *HMS Eagle*, in New York Harbor in 1776. The plan was to bore into the ship's hull to attach a spar torpedo. Success was thwarted when the operator found the ship's bottom to be copper-plated. A few feet from his position was the unplated rudder post—and possible success. The attempt, however, frightened the British into moving the fleet's anchorage to more protected waters. Submersibles, although certainly not yet an integral part of any navy, were indeed on their way to becoming viable naval vessels.

Robert Fulton's *Nautilus* (Fig. 3), the first of several submarines to bear the same name, was sail-powered on the surface and man-powered submerged. Fulton successfully demonstrated its military attributes to Napoleon, blowing up a barge on the Seine River in 1800. The Emperor, however, remained indifferent to this unorthodox method of naval warfare, which forced Fulton to turn his attention to England. He built a larger craft and in 1806 attempted to sell it to the British Admiralty through the good offices of William Pitt. Again rejected, a frustrated Fulton returned home to be recorded in history for building the steamboat *Clermont*.

The *Hunley* was one of several "David-class" submersibles built by the Confederate Navy during

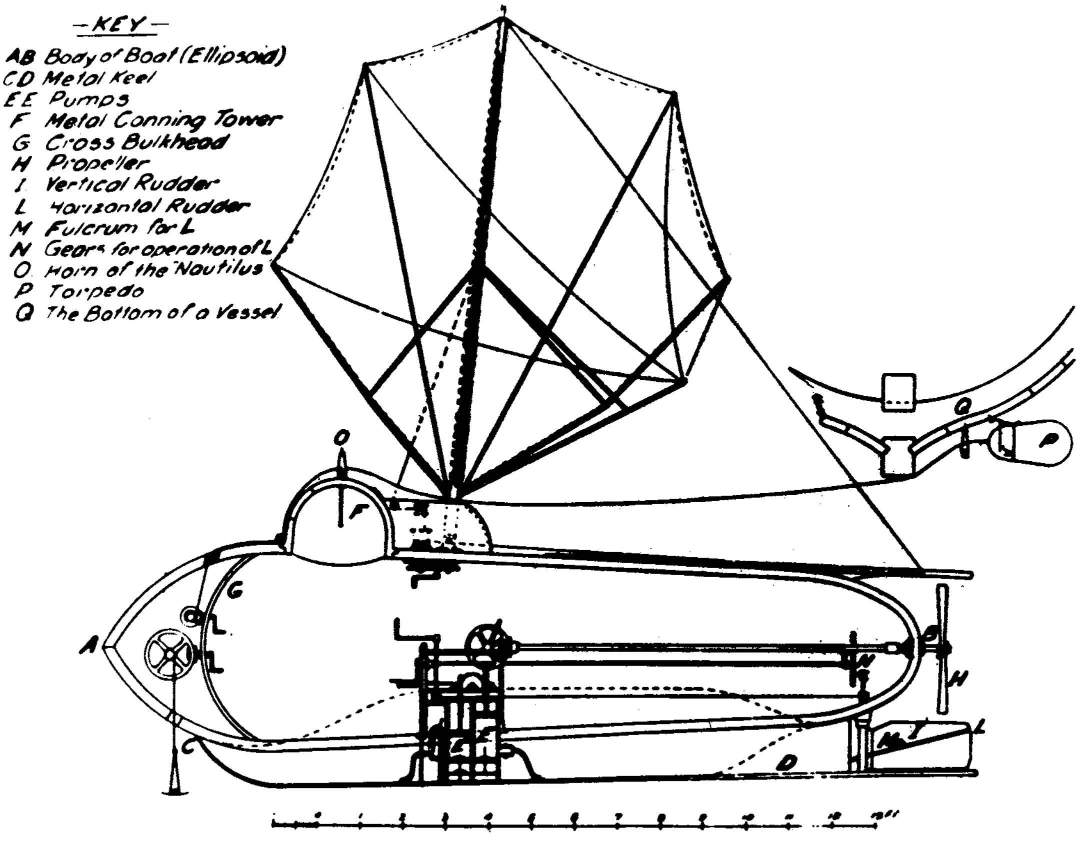


Fig. 3 The *Nautilus*, built by Robert Fulton, was demonstrated to Napoleon in 1800

the Civil War in a desperate attempt to break the Union Navy's blockade of southern ports. This craft made the world's first successful attack on an enemy warship, sinking the Federal corvette *Housatonic* in Charleston Harbor in 1864. It was a costly victory, however, both vessels being lost in the encounter.

The development of the modern submarine began in the late 1800's and early 1900's, the U.S. Navy commissioning its first submarine, the *USS Holland*, in April 1900 (Fig. 4). Although the *Holland*'s performance and safe operation were jeopardized by the use of a gasoline engine for propulsion, its hull shape was remarkably similar to that required for minimum submerged resistance. The submarine came of age in September 1914 when the German U-9 astounded the world by sinking the British cruisers *Aboukir*, *Cressy*, and *Hogue* within a few minutes. Since that fateful day, the submarine increasingly has been recognized as a major arm of the world's leading navies, with all aspects of its design, construction, and operation undergoing continuous improvement.

The U.S. Navy's so-called "fleet boat" epitomized the United States' submarine development stage at the end of World War II. To this point in its history, the submarine might more appropriately have been called a "submersible torpedo boat," since it was essentially a surface vessel that could operate submerged for relatively short intervals of time. This severe constraint was imposed, of course, by limited battery energy and the need for frequent access to the atmosphere to operate the diesel engines and to replenish the crew's on-board supply of oxygen. Burdened with this constraint, designs emphasized surface-operating characteristics as embodied in the

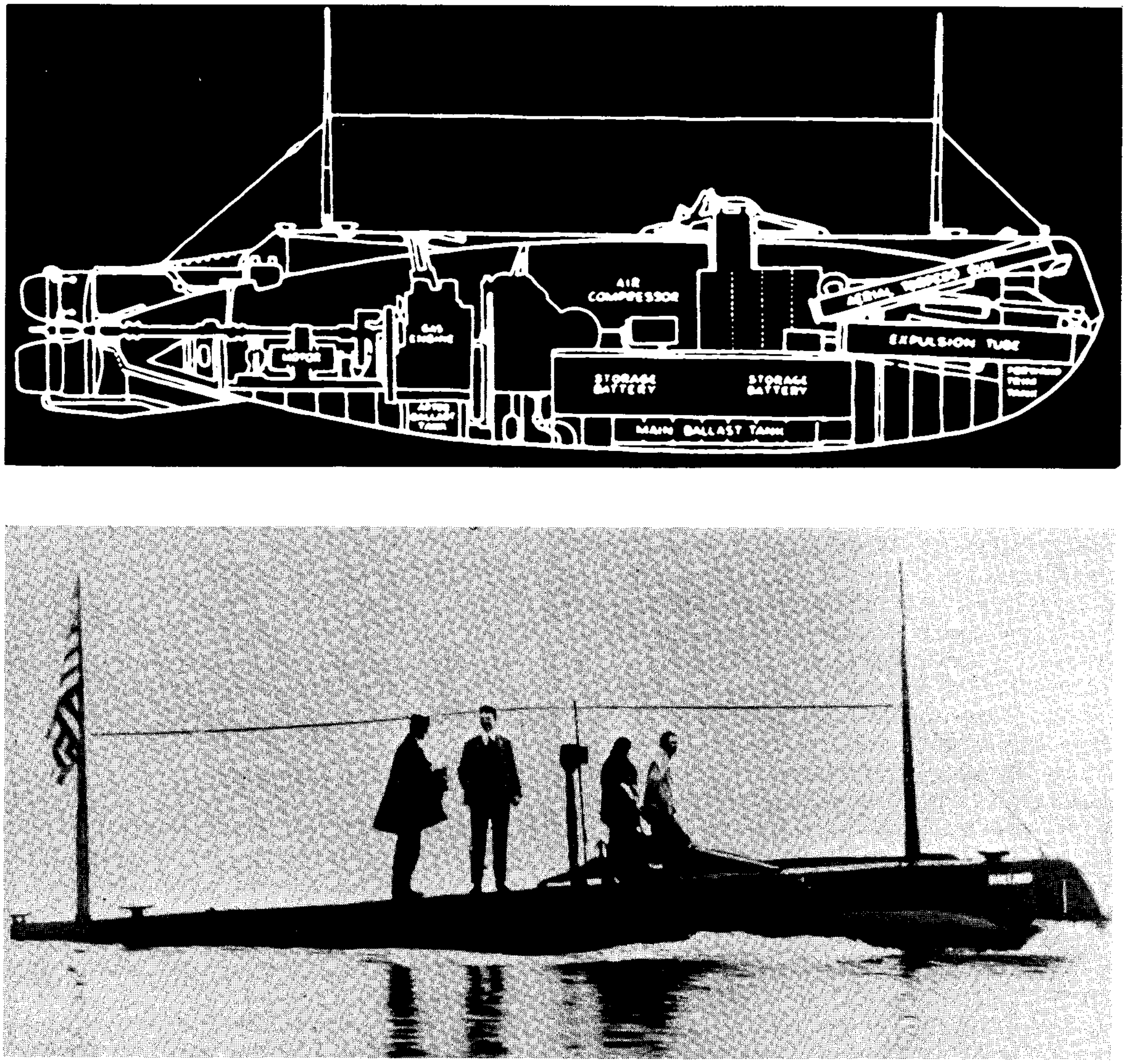


Fig. 4 *USS Holland* (SS1). The cross-sectional plan (***top***) shows that both gun and torpedo were housed within the hull. Conventional surface-ship design persisted in the form of masts for which a submarine had no use

"fleet boat"—a long, slender hull to reduce wave-making resistance encountered at or near the surface with a superstructure and numerous appendages to improve sea keeping and facilitate surface operations.

The advent of nuclear power and the less-heralded oxygen generator in postwar years removed the need for extensive time on the surface, thus making possible the development of the "true submarine" as initially envisioned by Jules Verne—a vessel that could operate submerged for almost unlimited periods of time. No longer obliged to acknowledge wave-making resistance, which disappears at deep depths, and surface operating priorities, designs could now stress submerged performance. This led to the development of a "cod's head and mackerel tail"[2] hull form, uncluttered with extensive superstructure and appurtenances in order to minimize submerged resistance and improve high-speed maneuvering. The *USS Albacore*, with its streamlined hull, and the *USS Nautilus*, the world's first nuclear submarine, initiated the "true submarine" trend.

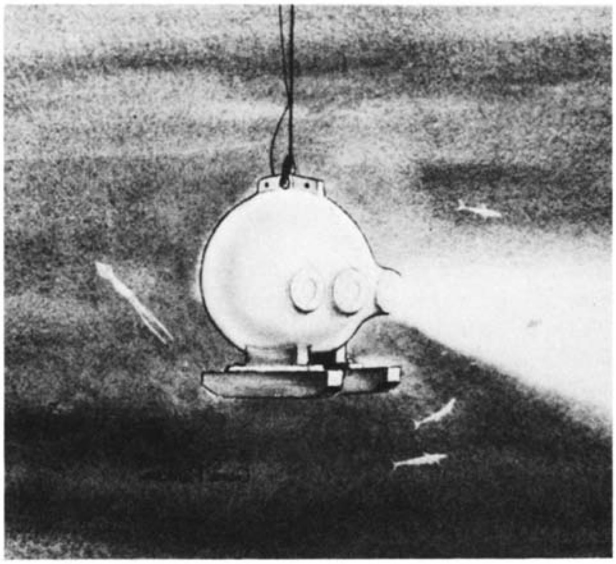


Fig. 5(*a*) Bathysphere (1930)

2. Modern Submersible Development

2.1 Development History

Modern submersible development began in the early 1930's, initiated principally by scientific research interests. These interests, of course, had existed for many preceding decades, but had been pursued primarily from on board such famous surface ships as the *Beagle*, *Challenger*, and *Meteor*. Now, scientists were becoming even more inquisitive, wanting to see for themselves just what was transpiring beneath the waves. A tethered submersible called a bathysphere (deep sphere) (Fig. 5) was built in 1930 to serve this purpose.

The bathysphere was a thick-walled steel ball with fused quartz viewports. This cast steel ball was 1.37 m (4.5 ft) inside diameter with a shell thickness of 31.75 to 38.1 mm (1.25 to 1.5 in.). The sphere was lowered by a surface support ship using a 22.2-mm (3/4 in.) wire rope. William Beebe, a zoologist, and Otis Barton used the bathysphere in 1934 to descend to a record depth of 923 m (3028 ft) off the coast of Bermuda to study and photograph deep-sea life. In 1948, Otis Barton built the benthoscope, a design very similar to the bathysphere, in which he made tethered dives to 1372 m (4500 ft) off the coast of Southern California. The benthoscope was a sphere of 1.37 m (4.5 ft) inside diameter and a shell thickness of 44.5 mm (1.75 in.).

World War II years saw underwater science and technology focus on submarine and anti-submarine warfare, and numerous developments of this period had direct impact on the designs of modern submersibles. The creation of a variety of new instruments and marine hardware made possible a dramatic increase in knowledge of the oceans, perhaps one of the most significant advances being made in the field of acoustics.

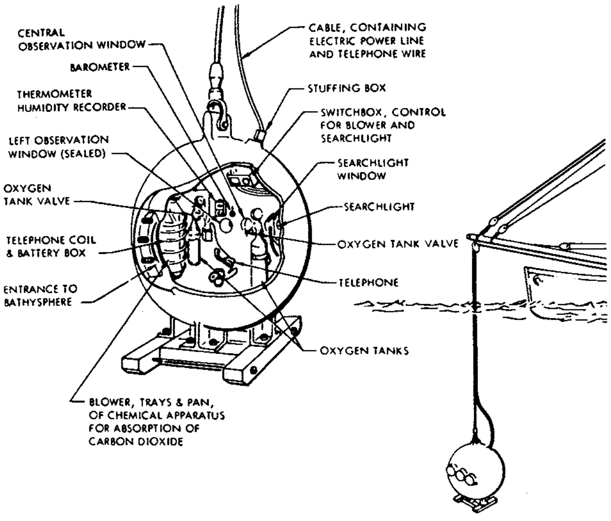


Fig. 5(*b*) Cutaway view of bathysphere used by William Beebe in his record dive of 1934, marking the beginning of modern submersible development (courtesy of National Geographic Society)

Postwar years marked the advent of a strange new submersible called a bathyscaphe (deep boat) developed by Auguste Piccard, a Swiss scientist, who also developed the stratospheric balloon (Fig. 6). The design of his first bathyscaphe, called *FNRS-2* after

[2]A phrase often used in describing the ideal hydrodynamic hull-form for deep submergence propulsion.

the Belgian research society funding the project (Fonds National Belge De La Recherche Scientifique), embodied principles used in the balloon, *FNRS*. After taking cosmic ray measurements at a record altitude of 16 280 m (53 400 ft) in the balloon, Piccard felt it was necessary to take comparable measurements in the deep ocean. The "balloon" of the submersible was a large, thin-skinned steel tank filled with gasoline, which served as the buoyancy material and from which was "hung" a heavy pressure-resistant steel personnel sphere.

Piccard's first bathyscaphe, *FNRS-2* (Fig. 6), was built in 1948. It consisted of a 2-m-inside-diameter (6.58 ft) steel sphere 89.9 mm (3.54 in.) thick. The light sheet metal float contained 30 m^3 (1059 ft^3) of gasoline. Vertical control of the craft was achieved through the use of iron shot as the drop weight. Gasoline, although a liquid, is more compressible than sea water, and as the craft descended, buoyancy was lost due to this compression. As the gasoline compressed, displacement volume was lost, and the rate of descent of the bathyscaphe would increase to unsafe levels if weight was not dropped to compensate for the loss of buoyancy (displacement). As with any material or structure to be used in the deep ocean, its bulk modulus must be compared with that of sea water at the operational depth.

The descent of *FNRS-2* was controlled by iron shot dropped from large storage tubs through an electromagnetic "choke" valve. With safety of the craft and personnel in mind, Dr. Piccard used a well-proven, fail-safe concept and an early form of failure mode and effect analysis to achieve a safe return to the surface in the event of a power failure. The bathyscaphe was battery-powered, and this direct-current power was used to control the release of the iron shot. With a magnetic "choke" valve energized, a magnetic field was generated that froze the iron shot into a solid plug in the narrow opening of the shot tub. For normal release the electromagnet was de-energized for a short period and shot fell out. In the event of a power failure, the valves would be de-energized automatically and all of the shot would be released. In fact, to make the craft absolutely safe, the shot tubs themselves and the heavy main power batteries, were held in place with electromagnetic latches. All of the bathyscaphes developed during the 1950's and the 1960's used this same fail-safe design and control principle.

Although designed before WWII (1938–1939), the *FNRS-2* was not completed nor were tests started until 1948. *FNRS-2* made an unmanned test dive to 1388 m (4554 ft) off Dakar West Africa (Senegal). Because of a lack of proper "system" design and support equipment (due to limited research-type funding), the craft had to be towed on the surface when full of gasoline. High seas that built up during the test dive prevented the gasoline from being pumped out of the float, and storm-condition towing damaged the float beyond repair. Because funds were not available to continue the project, work on *FNRS-2* was stopped. The French Navy took an interest in the program, and the *FNRS-2* personnel sphere was used for an improved bathyscaphe, *FNRS-3*, which made dives to 4052 m (13 290 ft) in 1954. Because Dr. Piccard wanted more control over the bathyscaphe project, he left the *FNRS-3* program to build a new bathyscaphe in Trieste, Italy. This craft, named *Trieste* (Fig. 7), made dives to 3692 m (12 100 ft) in the Mediterranean (1956).

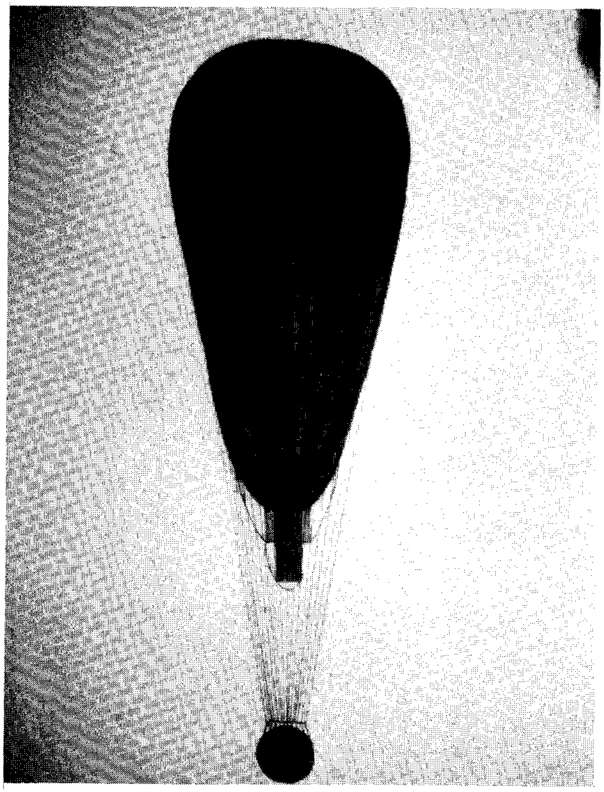


Fig. 6(*a*) Stratospheric balloon by Auguste Piccard

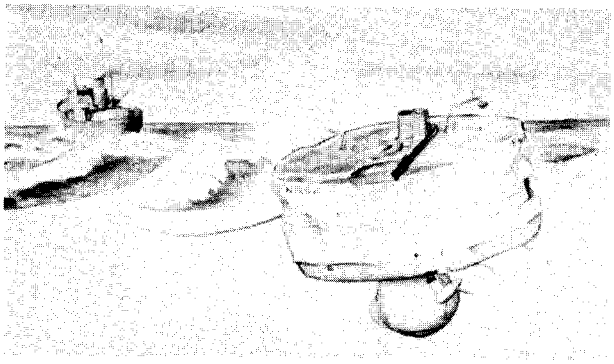


Fig. 6(*b*) Piccard's first bathyscaphe, *FNRS-2* (1948)

Recent history of submersibles in the United States covers a period of about 37 years—from the early 1950's to the present.[3] The period may be spoken of in terms of the "first and second generation" of submersibles, the first generation lasting until about 1973 and the second generation from 1973 to the present. Initiation of the first generation began in Italy in 1952 with the building of the bathyscaphe *Trieste* by Piccard. This vehicle was purchased by the U.S. Navy in 1953 and eventually, on January 23, 1960, made a historic dive to 10 915 m (35 800 ft) in the Mariana Trench off Guam. This record depth has never been reached again by a submersible. During the same time frame the French retired *FNRS-3* and built a new bathyscaphe, *Archimede* (Fig. 8). It made dives in 1962 to 9604 m (31 500 ft) in the Kurile Deep of the Kamchatka Trench off Japan.

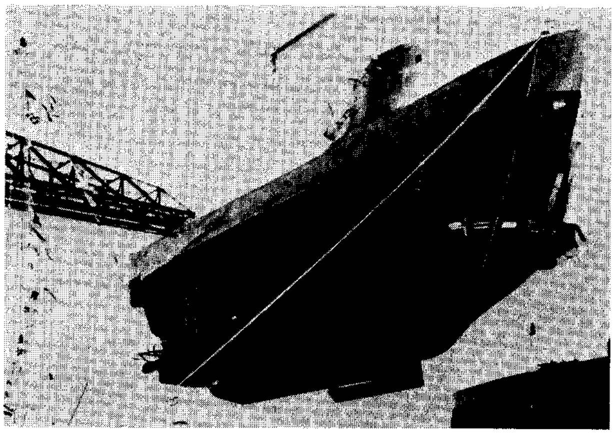


Fig. 8 The French bathyscaphe *Archimede*

[3]The "present" is mid-1988.

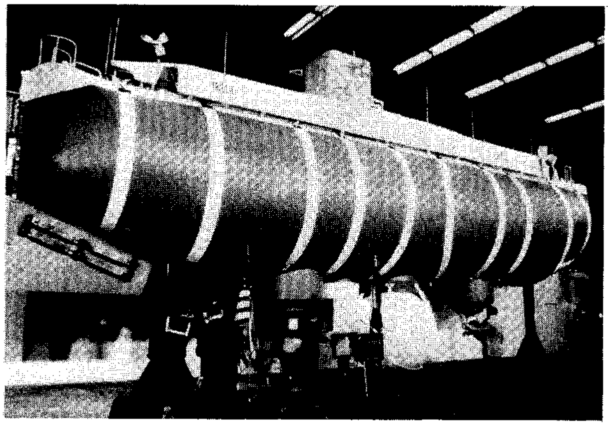


Fig. 7(*a*) The bathyscaphe *Trieste* (1960 configuration) on display at the Navy Yard Museum, Washington, D.C.

The 1950's also saw the development of underwater systems by Jacques-Yves Cousteau, including his soucoupe (diving saucer), *Denice*, which was one of the first shallow-diving submersibles to be used in the United States during the mid-1960's (Fig. 9). The soucoupe could take two people to depths of 300 m (1000 ft). With a weight of 4.2 tons, this craft was easy to handle by a shipboard crane and with the support ship *Calypso*. The diving saucer has made hundreds of dives on a worldwide basis.

To this point in its history, the submersible had been strictly a manned vehicle. The year 1960 marks the debut of the unmanned submersible. Until then, most underwater research data had been acquired by suspending individual instruments from a surface ship, but increasingly sophisticated research now demanded that two or more instruments record data in a carefully coordinated manner. The solution, developed by Fred Spiess of Scripps Institution of

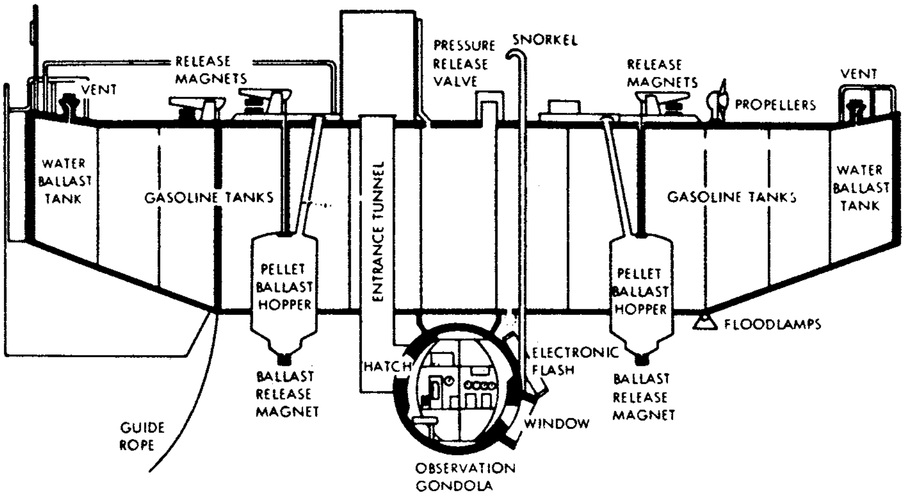


Fig. 7(*b*) Cutaway of the *Trieste*, which made the world's record dive to 10 915 m (35 800 ft) in the Mariana Trench off Guam on January 23, 1960 (from Terry, *The Deep Submersible*, 1966)

Oceanography in California, was to mount all required instruments on a single frame, the total assembly being called a *fish*, which was towed behind a ship at specified depths. The fish enjoyed considerable success, especially when deployed from uniquely equipped research ships such as the *Mizar*, which is known for its successful role in the undersea searches for the ill-fated nuclear submarines *USS Thresher* and *USS Scorpion* and for the H-bomb lost off Palomares, Spain. Those successes notwithstanding, the unmanned submersible would not come into its own until the mid 1970's.

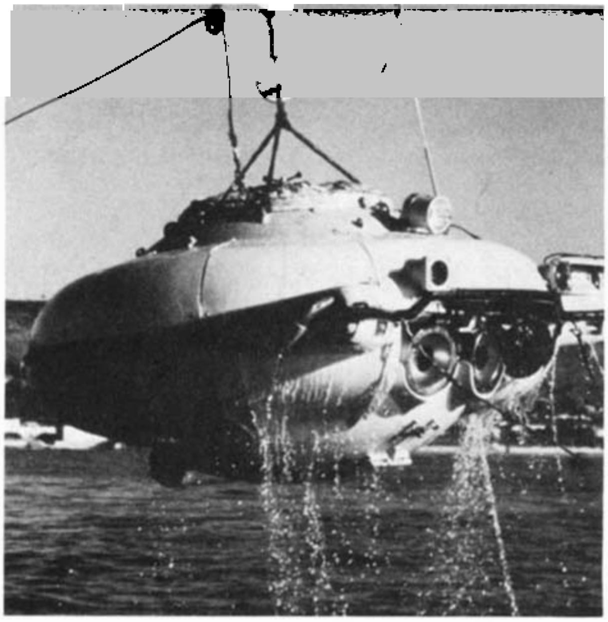


Fig. 9(*a*) Cousteau's diving saucer *Denice*

Returning to manned submersibles, the Navy modified *Trieste*, to become *Trieste II*, to locate, inspect, and photograph the wreck of the *USS Thresher* in 1963 and to revisit the site in 1964. Modifications included a different float designed for faster towing, an improved maneuvering system and the installation of the original 20 000 ft rated steel sphere.

A portion of the first generation, lasting from about 1963 to 1973, is sometimes rather descriptively subdivided into two phases: the periods of "great expectations," ending in 1969, and the "doldrums," ending about 1973. Perhaps the key event initiating the first phase was the tragic loss of the *Thresher* on April 10, 1963.

The *Thresher*'s legacy was to focus national attention on the marine environment—on how much remained to be learned about it and how little work could be accomplished in its domain. The following years saw a great flurry of government and private activity and the formulation of a "national ocean program." One of the most prestigious and comprehensive documents produced was the Stratton Commission report, *Our Nation and the Sea*, which advocated the development of underwater work systems with capabilities down to 6098 m (20 000 ft). Commensurately, the heady "great expectation" years witnessed both large and small companies, many of them aerospace-oriented, racing to establish themselves in some area of the submersible field. The thought prevailed that the federal government would support "inner space" research and development in a manner paralleling the support for its "outer space"

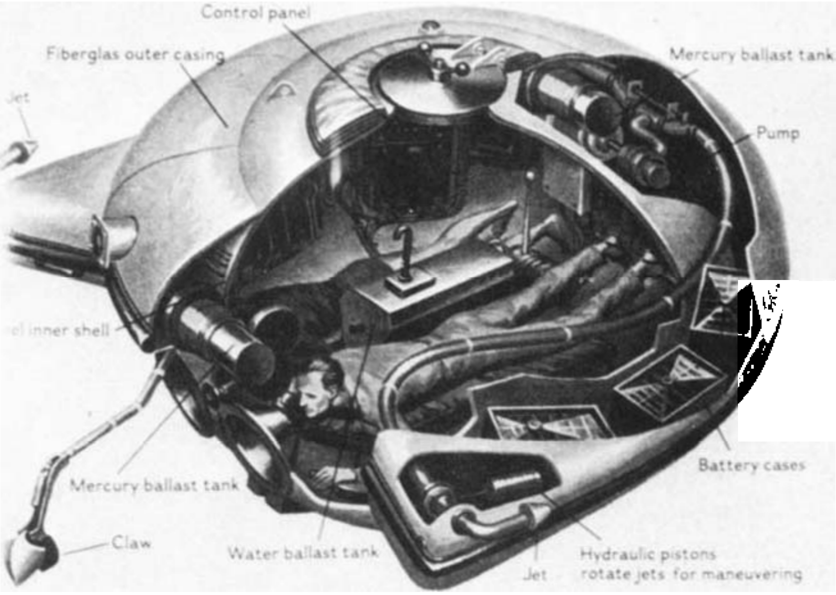


Fig. 9(*b*) Cutaway view of the *Denice*

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program—that there might well be a "wet NASA" established.

But Vice President Agnew in the spring of 1969 made it clear in a speech that the government had no such intention. In addition, increasing pressures of the Vietnam War caused a reassessment of the Navy's submersible interests and a retrenchment in funding from this major source of support. Thus it was, with few other customers in sight, that the great expectations faded, many companies withdrawing from the field and some smaller ones failing in the process.

Problems of this phase were compounded by the fact that many submersibles were built on speculation or to demonstrate company capabilities in the submersible field with the thought of improving future "bidder's list" standings. Designs were often based on mission requirements unrelated to well-identified markets; submersibles were constructed and then went looking for work. It is little wonder, then, that the "doldrums" would follow. This phase saw most submersibles laid-up or scrapped, the industry in general reaching a low ebb of activity.

Although the submersibles of this era found few markets, they did represent the forward-looking views and designs of their creators. For example, a group of engineers within the Electric Boat Co. got together and built *Star I*. The Company began to think that there might be something to this small submersible business and went on to produce *Asherah* for Dr. George Bass of the University of Pennsylvania for undersea archeological research in the Mediterranean off Turkey and the Greek Islands. Electric Boat continued in the small submersible business for a while, producing *Star II* (still operating in the Hawaiian Islands) and *Star III* (Fig. 10), now on exhibit at San Diego's Sea World. Two later and larger submersibles were built by Electric Boat for the U.S. Navy in 1968, the *Turtle* (Fig. 11) and *Sea Cliff*.

Fig. 10 The *Star III* by Electric Boat Division, General Dynamics Corporation, Groton, Connecticut

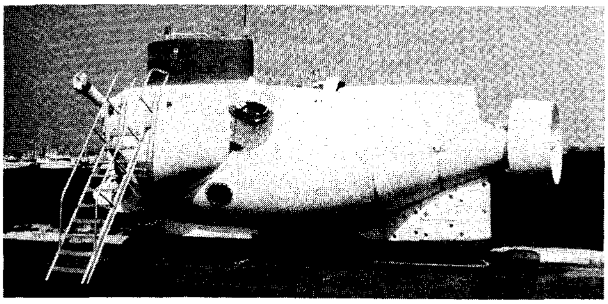


Fig. 11 The *Turtle/Seacliff* by Electric Boat Division, General Dynamics Corporation, Groton, Connecticut

Many other aerospace and industrial companies entered the market place during the early years of the submersible development industry in the mid 1960's. Examples of craft built during this period are discussed in the appendix to this chapter.

The second generation of submersible activity, beginning about 1973, has seen submersibles come into their own, establishing themselves as necessary and effective components of systems for accomplishing a wide variety of underwater industrial, scientific, and military tasks. Their successes, it must be remembered, were, and continue to be, based on invaluable experiences gained in the design, construction, and operation of first-generation submersibles. Thus the phase of "great expectations," in which almost all of the first-generation submersibles were built, was not a lost cause. The successes, for the most part, were also based on well-defined markets and missions for submersible services that promoted a trend toward building specialized vehicles rather than the less-efficient, general-purpose submersibles of the first generation.

Most markets for submersible services have been created by the activities of the expanding offshore oil and gas industries. These activities have generated a wide variety of tasks, including (1) seafloor surveying for suitable footings for platforms and for pipeline routes, (2) assisting in the installation of seafloor structures, platforms, and pipelines, (3) monitoring underwater activities, (4) inspection, maintenance, and repair of underwater structures and pipelines, and (5) providing assistance and mobile support for divers.

Notable scientific activities also have required submersible services. Some outstanding examples include (1) the 1974 French-American Mid-Ocean Undersea Study (FAMOUS) of the Mid-Atlantic Ridge (*Alvin, Archimede*, and *Cyana*) and (2) the 1979 East Pacific Rise study, locating thermal vents surrounded by an amazing array of benthic organisms (*Alvin*).

Chapter III
The Environment

J. D. Irish and W. S. Brown

1. Introduction

THE MARINE ENVIRONMENT consists not only of the water column, but also the atmosphere above the water's surface and the seafloor and sub-seafloor (that is, the sediment and rocks). A comprehensive study of the marine environment is enormous in scope and thus is often subdivided into its disciplinary components—biological, chemical, geological and physical oceanography, meteorology, and geophysics. This chapter will focus on those features of the marine environment which are of particular interest to designers and operators of submersibles. It is limited to basic introductory information, but the reader will be referred to oceanographic literature for further details in the areas of discussion.

Every aspect of submersible design is influenced to varying degrees by one or more environmental factors. As noted in Chapter I, these factors comprise one of the several groups of so-called *external design constraints*—that is, constraints which are independent of or external to the mission on which the design is based. For example, temperature, salinity, pressure, and density gradients in the water column are encountered by all submersibles as they move through the water column irrespective of their individual missions. These external constraints may be subdivided into those which are fixed by mission requirements and those which are variable. An example of the former is the mission requirement of a maximum operating depth which fixes the pressure on which the design is based. But the same mission requirements for maximum operating depth and locations of operating areas do not fix specific values of temperature, salinity, and density of the sea water which will be encountered. In the latter example, the mission requirements indicate only ranges of these parameters that the submersible must be designed to accommodate.

A summary of those design considerations which are influenced by environmental factors is presented in Table 1 of Chapter I. The following brief overview is intended to introduce the submersible designer to the elements of the marine environment which are most influential on submersible design. More detailed descriptions of some of the more important elements appear in later sections.

It is convenient to view the marine environment from the perspective of its vertical profile, which is composed of three basic regimes: atmosphere, water column, and sub-seafloor. Of course, the characteristics of the water column are of primary interest in submersible design, but not to the exclusion of the other two environments and the interfaces between them. Figure 1 summarizes the various factors influencing submersible design and operations. Our overview starts at the top and works downward through the water column to the sub-seafloor.

1.1 The Atmosphere

The atmosphere, much like the water below it, is a fluid in constant motion. Many of the same dynamical principles control the ways in which both fluids move, despite the large density differences between the two. The interface between the atmosphere and ocean is characterized by a continual exchange of energy between the atmosphere and the upper portion of the water column. Of particular interest are the exchanges of thermal energy (heat) which influence sea and atmospheric temperature, and the kinetic energy associated with winds which are responsible for the generation of surface waves and currents.

Certain atmospheric factors such as wind, fog, rain, and icing at higher latitudes affect submersible operations on the sea surface. These factors influence design considerations related to retrieval and become more critical in abnormal conditions where considerable time may elapse between the time a submersible surfaces and the time it is picked up. Heeling moments due to wind place constraints on the design of the extent and distribution of the

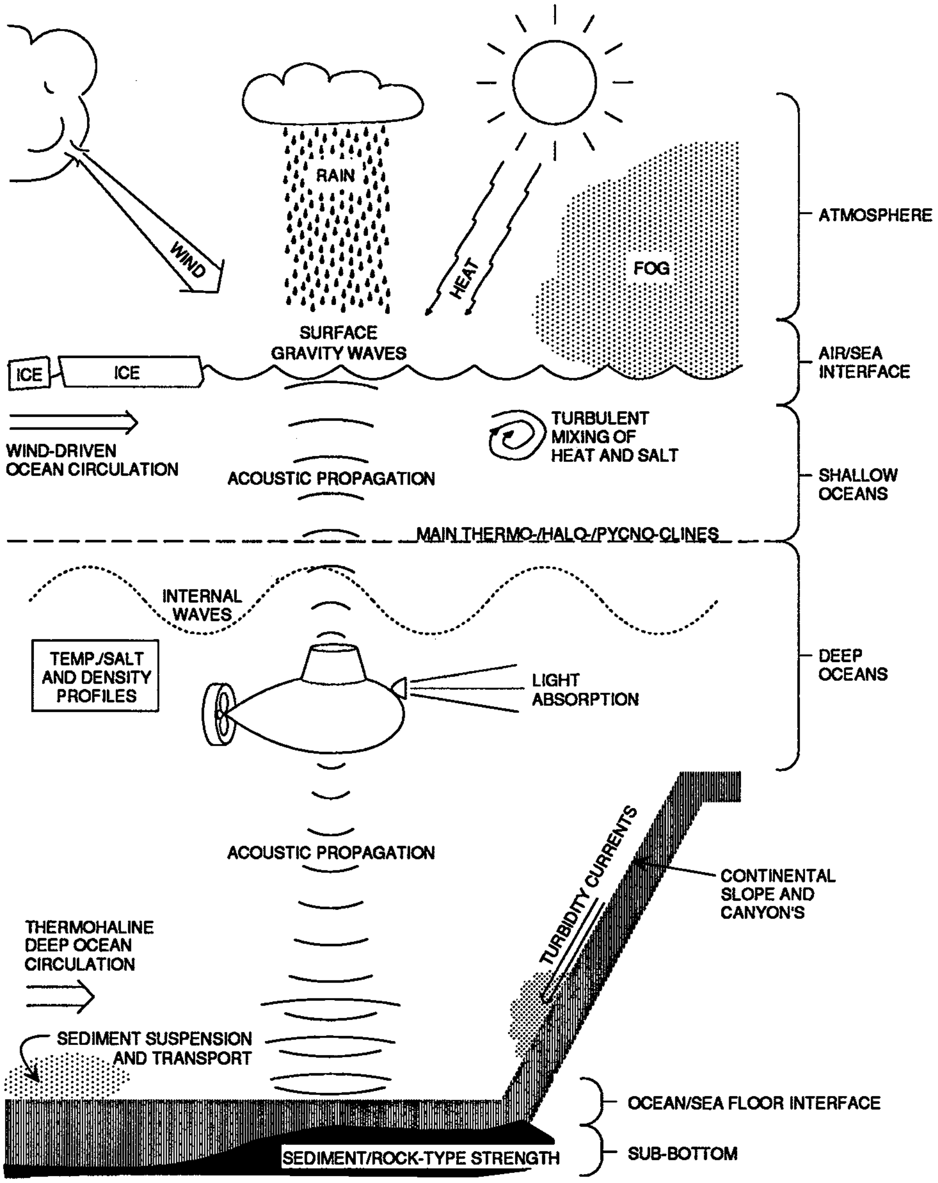


Fig. 1 Factors affecting submersible design for various regions of ocean profile as shown at right

submersible's lateral area above the water line as well as provide adequate dynamic stability to limit heel angles. Fog and rain, or adverse weather in general, reduce or obliterate visual communications, stressing the need for other, highly reliable means of contacting the support ship. The possibility of icing and its effects on stability must concern the submersible designer.

1.2 The Air/Sea Interface

The interface between the atmosphere and ocean can be open water, partial ice cover, or solid ice cover. The air/open water interface is the interface of primary interest for manned submersible design and is the one discussed here. Partial and solid ice cover interfaces usually dictate the use of unmanned submersibles (see Subsection 3.5).

Surface waves are one of the primary concerns in submersible design. As discussed in Chapter I, the sea-state limitations imposed by surface waves play a major role in establishing the cost-effectiveness of the mission system, that is, the percentage of at-sea time a submersible can be used. Surface waves must be considered in connection with dynamic stability requirements to limit the angle of heel due to wave action and on freeboard requirements to protect the access hatch from waves and to facilitate retrieval procedures. Surface waves produce direct wave-slap forces on the submersible which need to be considered, and create splash-zone corrosion considerations if the vehicle is not hosed down between dives.

The magnitude of expected surface currents affects the maneuvering requirements of a submersible design. The problem may not be severe if the support vessel is also being affected by the same uniform surface currents. In addition, currents can affect the submersible during the descent or ascent so it may not arrive precisely at the dive target or support ship and thus may have to spend valuable time positioning itself.

1.3 The Water Column

For the purpose of discussion, the water column is subdivided into zones or layers, and each of the following environmental factors is discussed in terms of its principal effect on submersible design.

1.3.1 Surface Gravity Waves—Shallow and Deep Layers

Although surface gravity waves are most apparent at the air/water interface, their effects can extend to great depths in the ocean. The amplitude of surface waves and their associated currents are attenuated so that their effects become negligible (less than 5 percent) by a depth $L/2$, where L is the wavelength. For typical wind waves (period of 20 s) in deep water this depth is about 300 m (984 ft) and almost never exceeds 1000 m (3284 ft). Consequently, the submersible experiences forces in the shallow-depth layer (several hundred meters) due to surface wave action. This motion may be severe enough to exert considerable influence on the design of the launch/retrieval system, including that portion on the submersible, and put limits on the sea states in which safe operations can be considered.

Tides and tsunamis are other surface waves which have such long wavelengths (long periods) that their amplitudes are not attenuated appreciably with depth even in the deepest oceans. However, because their periods are so much longer (minutes to a day) than typical wind waves, the water velocities associated with them are generally small. An important exception to this case occurs in coastal regions where tidal currents can be significant.

1.3.2 Internal Gravity Waves—Shallow and Deep Layers

The vertical density gradients found in the ocean allow for another class of waves: internal gravity waves whose maximum velocities are found at the depths of the maximum density gradients. The vertical excursions associated with these waves can be very large (10's of meters) with periods of minutes to an hour.

1.3.3 Wave Generation Drag—Surface and Deep Layers

The low air/water density ratio permits surface gravity waves to be generated at the interface by a submersible moving near the surface. Internal gravity waves are also generated by submersibles moving along strong density gradients in the water column. The term "dead water" is used in association with such gradients found in or near fjords or estuaries.

The generation of any of these waves creates a *wave-making resistance* and increases the energy required to move the submersible. This resistance tends to be greatest where the density gradient is largest, but the amount of energy used in making waves is dependent on the geometry and design speed of the submersible. For example, submersibles with missions requiring high speeds in the shallow ocean (where density gradients are large) should have geometries which minimize wave-making resistance. Conversely, submersibles with missions requiring high speeds in the deep ocean (where density gradients are small and wave making is less significant) should have geometries which minimize frictional and form resistances. The submersible geometries in these two cases are considerably different.

1.3.4 Temperature, Salinity, Pressure, and Density (Specific Weight) of Sea Water—Upper and Lower Layers

Sea water temperature, salinity, and density vary substantially in the upper layer of the water column which extends down to about 1000 m (3284 ft). These variations, in general, are due to the influences of heat exchange and precipitation/evaporation across the air/sea interface at a particular latitude and season. In the lower layer, below 1000 m, these characteristics tend to be relatively constant. The upper layer may be subdivided further into regions of seasonal and main thermoclines (haloclines)—where temperature (salinity) gradients are largest—extending, respectively, from the surface to about 60 m and from this depth to about 1000 m.

Density or specific weight of sea water is a function of temperature, salinity, and pressure. Consequently, this property will vary nonlinearly in the upper layer of temperature and salinity variations down to about 1000 m. In the lower layer, it may be approximated as a linear or exponential function of depth.

Hydrostatic pressure in the water column is the integral of the density times gravity from the surface down to the depth of interest (that is, the weight of the water above). Therefore, the *in situ* pressure includes the effects of temperature, salinity, and sea-water compressibility, and varies nearly linearly with depth. (A unit volume of sea water moved from the surface to a depth of 9000 m is compressed by about 4 percent.)

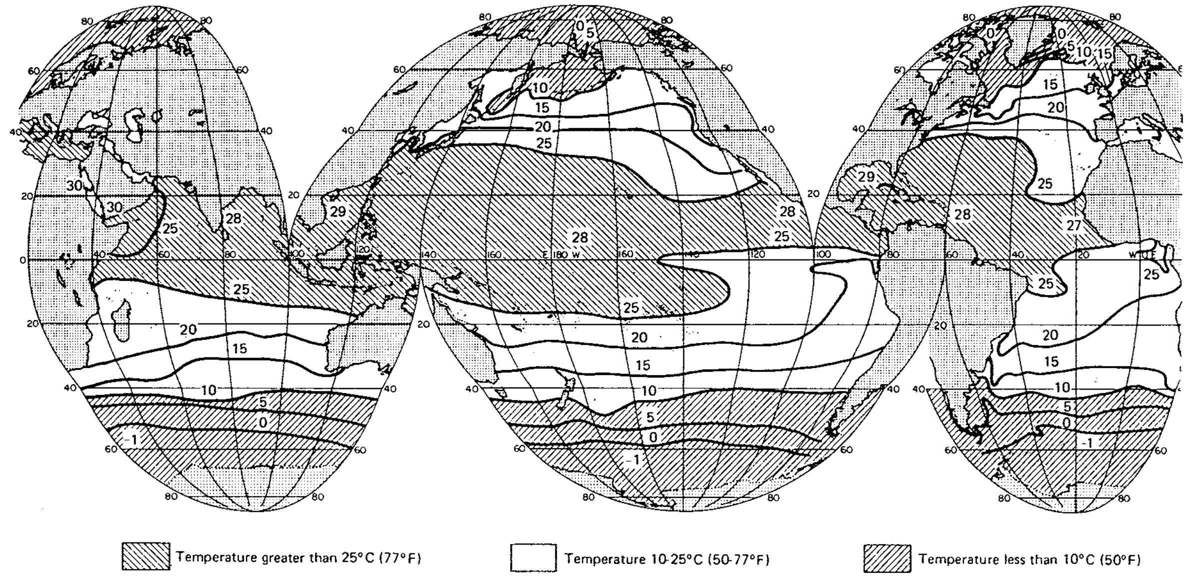


Fig. 2 Sea-surface temperatures in August. (From Sverdrup et al., 1942)

These environmental factors, individually and collectively, influence most aspects of submersible design and operation—including shell structure strength, corrosion considerations, life-support systems, acoustical systems, electric/mechanical systems exposed to sea water, and buoyancy control systems. Specific references to these factors are made in other chapters of this book.

2. Physical Properties of Sea Water

Sea water can be characterized by its temperature, salinity (the dissolved solids), and pressure. From these three quantities and the equation of the state of sea water, the ocean scientist or engineer can calculate other desired quantities, such as the density, sound velocity, heat capacity, or electrical conductivity. The oceanographic tables of yesteryear have been replaced by the microcomputer for the calculation of these derived quantities.

2.1 Temperature

The temperature, T, of a water parcel is expressed in degrees Celsius (°C) and gives an indication of the energy or work that has been done on or associated with that water parcel. Temperature is now being easily and accurately measured by electronic thermometers employing thermistors or platinum resistance probes as the sensing elements. These sensors are capable of resolving microdegree temperature fluctuations, and are stable to millidegrees over a period of months. Oceanographic temperatures are referenced to the International Practical Temperature Scale of 1968 (IPTS-68). (See Mackenzie, 1971, for a discussion of and conversion between this standard and the older IPTS-48.)

Oceanographic temperature tends to vary systematically with depth and latitude. The surface water is warmer at low latitudes (maximums of 25 to 30°C), where there is an excess of incoming solar radiation which heats the surface water. Surface water is cooler at the poles (minimums equal to the freezing point of saltwater—about -2°C), where energy is lost to the atmosphere by radiation. The distribution of temperature with latitude is not smooth, due to the general oceanographic circulation patterns, and will vary with seasons. (Maps showing global sea surface temperature distributions can be found in *The Oceans*, 1942.) Figure 2 shows typical summer sea surface temperatures.

Figure 3 shows typical vertical distributions of oceanic temperature. There is a relatively shallow region of uniform temperature (from 0 to 10's of meters thick) associated with the mixed layer at the surface (caused by surface wave mixing). Below this, the temperature decreases sharply in a *seasonal thermocline* (found largely at mid-latitudes), which varies in depth and intensity. It can disappear entirely during the winter, because its strength depends on the weather during the past year. The vertical gradients in the seasonal thermocline are about 0.05°C/m. Below the seasonal thermocline (which is generally confined to the upper 100 m), there is the *main* (or permanent) *thermocline* which is found between 100 and 1000 m and is maintained by the general oceanographic circulation and mixing. The temperature gradient here (0.02°C/m) is less than in the seasonal thermocline. Finally, the temperature

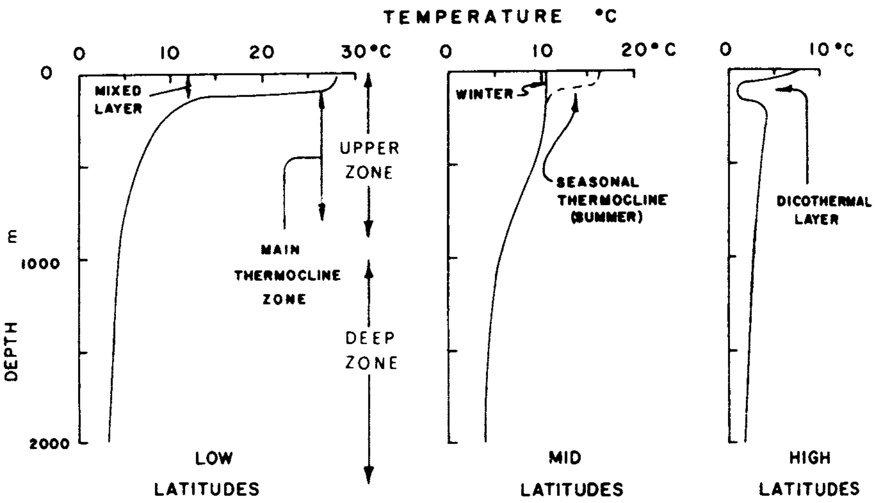


Fig. 3 Typical vertical distribution of oceanic temperature

in the deep ocean is nearly uniform and is shown with a temperature gradient of 0.001°C/m.

Water is slightly compressible, so that the volume of a parcel of water moved from the surface to the ocean floor (5000-m depth), where the pressure is over 7000 psi (almost 500 times atmospheric pressure), will decrease by about 2 percent. If this compression is done adiabatically (without loss or gain of heat), the work done by the pressure moving the water molecules closer together will raise the temperature of the water parcel at a rate of about 1.4×10^{-4}°C/m. For example, if a parcel of sea water at 5.000°C/m at a 4000-m depth were raised adiabatically to the surface, the temperature would decrease to 4.574°C. Conversely, if a parcel with temperature of 5.000°C at the surface were moved to a 4000-m depth, the temperature would increase to 5.438°C. In the North Pacific, the temperature below 3500 m increases with depth at a rate which is about the adiabatic gradient (1.5×10^{-4}°C/m), indicating that the water is very well mixed, but the compressibility effects seemingly cause the temperature to increase with depth.

To compare the temperature of two parcels of water, the oceanographer refers to the potential temperature, or θ, which is the *in situ* temperature, T (the temperature of the parcel at its site in the ocean as measured by a thermometer), which has been corrected for these compressibility effects. Generally, temperatures are corrected to the surface or zero relative pressure, but to compare two parcels at great depth, it is sometimes more accurate to correct to a common or a standard non-zero pressure surface. This is because our knowledge and estimation of the compressibility effects is not perfect, and the results are sometimes better if only small corrections were made to each observation, rather than comparing two numbers after subtracting large corrections. (Note that oceanographers refer to zero pressure as the pressure at the ocean surface, which is really one atmosphere absolute pressure, or about 14.78 psi.)

2.2 Salinity

The salinity of sea water, S, is the total amount of dissolved material or "salt" in the water. It is "the total weight of solid material (in grams) found in 1 kilogram of sea water when all the carbonate has been converted to oxide, the bromide and iodine replaced by chlorine, and all organic matter completely oxidized" (Forsch, Knudsen and Sorensen, 1902). The

Table 1 Composition of sea water—concentration of constituents in sea water having a chlorinity of 19‰

	g/kg	g/unit of chlorinity
Chloride, C^-	18.890	0.99894
Sodium, Na^+	10.560	0.5556
Magnesium, Mg^{++}	1.273	0.06695
Sulphate, $(SO_4)^{--}$	2.649	0.1394
Calcium, Ca^{++}	0.4104	0.02106
Potassium, K^+	0.380 mg/kg	0.02000
Carbon, as $(HCO_3)^-$ or $(CO_3)^{--}$	28	0.00735
Bromide, Br^-	65.9	0.00340
Strontium, Sr^{++}	8.1	0.00070
Boron, as H_3BO_3	4.6	0.00137
Silicon, as silicate	0.01–4.5	—
Flouride, F^-	1.4	0.00007
Nitrogen, as $(NO_3)^-$	0.01–0.80	—
Aluminum, Al^{+3}	0.5	—
Rubidium, Rb^+	0.2	—
Lithium, Li^+	0.1	—
Phosphorus, as $(PO_4)^{-3}$	0.001–0.1	—

salinity is usually expressed in grams of solids per kilogram of sea water or parts per thousand, which is abbreviated ‰. The composition of the various constituents of sea water are summarized in Table 1. (Note that sodium and chlorine make up more than 85 percent of the dissolved solids.) The concentrations of the dissolved elements are surprisingly constant, except in coastal regions where there is appreciable input of fresh water or in regions containing an odd mixture of elements, such as near hydrothermal vents in the seafloor. Therefore, there is an implicit assumption that the ionic composition of the ocean is fairly constant, as given by Table 1.

Historically, the salinity was determined by the chemical titration of a sample collected in a bottle lowered on a wire from the ship. This titration determined the chlorinity of sea water, which could be directly related to salinity by

$$\text{salinity} = 1.80655 \times \text{chlorinity} \qquad (1)$$

Presently, instead of directly measuring salinity, we use various sensors to measure the electrical conductivity of sea water and then calculate the salinity from the equation of state. The electrical conductivity of sea water is about 4 siemens/meter or S/m, where a siemen is inverse ohms or the SI unit of conductivity. The stability of conductivity sensors for use at sea is improving, and current instruments are stable enough that we can simply and reliably make measurements of *in situ* conductivity. Considerable work was done in the '70s on the definition and measurement of conductivity and on the equation of state of sea water used to calculate salinity which lead to the adoption of the Practical Salinity Scale of 1978 (PSS-78). (For details, see *UNESCO Technical Papers in Marine Science*, 1981.)

The surface distribution of salinity is less zonal and shows less variation than the temperature distribution. The amount of salt is largely controlled by evaporation and precipitation in open ocean regions, fresh river runoff in coastal zones, and ice formation and melting in polar regions (both sea ice and glaciers). The lowest salinities (0 to 30‰) are found in estuaries and polar regions. Coastal salinities are typically higher (30 to 34‰), but less than open ocean salinities (33 to 37‰). The average oceanic salinity is 34.7‰. Semiclosed evaporation basins such as the Mediterranean and the Red Sea exhibit such high salinities as 39 and 41‰. The outflow of the salty Mediterranean waters into the North Atlantic makes the Atlantic slightly saltier than the Pacific. The highest open ocean salinities are found in the center of the oceanic gyres (see the section on wind-driven circulation), and there is a relative low salinity band along the equator due to excess rainfall. (See *The Oceans*, 1942, for charts showing the distribution of surface salinity; see also Subsection 3.5.) Figure 4 shows sea surface salinities for the northern hemisphere during the summer.

The salinities also vary vertically, as shown in Fig. 5. As with temperature, there is a surface mixed layer with relatively low salinity gradients. Below this there is a strong halocline where the salinities decrease and have a destabilizing influence. The depth of the halocline roughly agrees with the depth of the thermocline. There tends to be a salinity minimum around 800 to 1000 m at mid latitudes. At low latitudes there tends to be a high surface salinity, and low surface salinities are found at high latitudes.

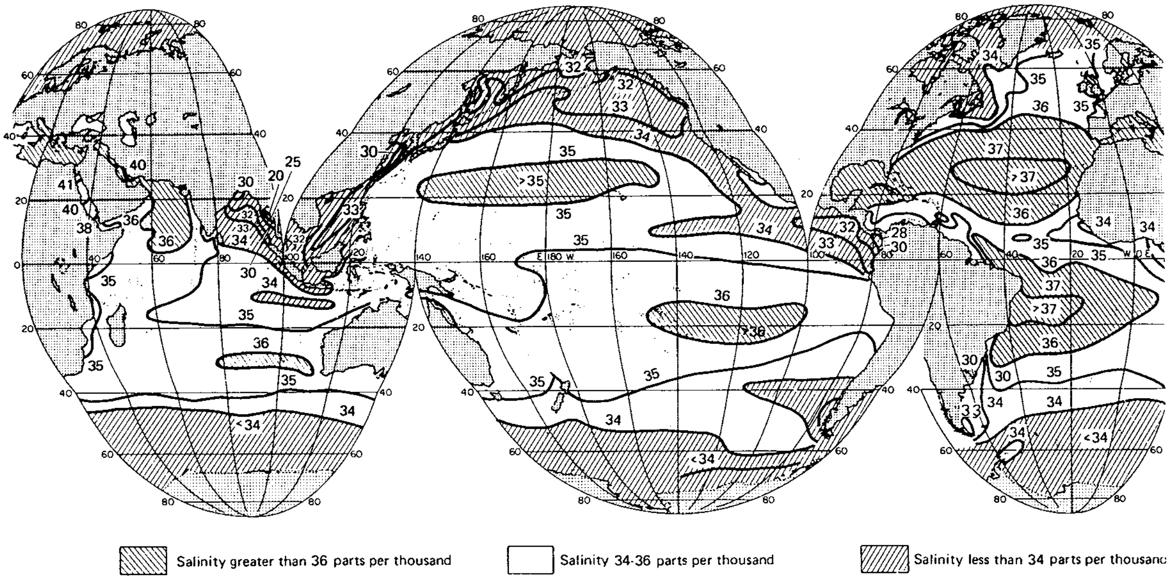


Fig. 4 Salinity of sea-surface water in northern summer. (From Sverdrup et al., 1942)

2.3 Density

The most important dynamic property of sea water is its density, ρ. This is sometimes expressed as the specific volume (the inverse of density), $\alpha = 1/\rho$. Density is generally calculated from the equation of state of sea water, which is expressed as a function of temperature, T, salinity, S, and pressure, P. (Oceanographers often use the decibar as a pressure unit since 1 decibar is nearly equal to the pressure due to 1 m of sea water. Note 1 decibar = 10^{-4} pascals, SI unit of pressure.) Typical values of surface oceanic density vary from 1020 to 1030 kg/m^3. Oceanographers tend to get tired of writing the 1000 every time, so have expressed density more conveniently in terms of the *in situ* density anomaly, σ_{STP}, which is defined as

$$\sigma_{STP} = \rho - 1000 \qquad (2)$$

The currently accepted reference for calculations of sea water properties is the 1980 equation of state of sea water (EOS-80) as given by *UNESCO*, 1981.

As with temperature, pressure effects on density are important since the volume of a parcel of water decreases as the depth or pressure increases. Hence a parcel with a density anomaly of 28.106 kg/m^3 at the surface will have a density anomaly of 46.644 kg/m^3 at 4000 m. As can be seen by this example, the pressure effect can be significant and especially in the deep ocean can obscure the effects of temperature and salinity variability. To compare the density of two water parcels, oceanographers have defined two further quantities. Sigma-*t*, σ_t, is defined as the density anomaly with the *in situ* salinity and temperature, but zero relative pressure. This removes the largest effect of the pressure on the volume, but does not consider the change in temperature. Sigma-θ, σ_θ, is the density anomaly with the *in situ* salinity, the potential temperature, and zero pressure. Thus we have a quantity which is defined in terms of the intrinsic properties of water properties at zero pressure or at the surface where the water probably acquired these characteristics. The external influence of pressure on the volume and temperature of the water has been removed, and the density of two parcels of water can now be truly compared.

An important feature of the ocean is its vertical density gradient or stratification. Normally, density increases with depth because there is the tendency for heavier parcels to sink below lighter parcels. When a parcel is displaced from its equilibrium position, there is a restoring force (due to the density difference and gravity) which tends to return the parcel to its equilibrium position. The strength of this restoring force is related to the vertical potential density gradient. (Note that this restoring force gives rise to a class of oscillations called internal waves which are discussed further below.) Oceanographers express the strength of this restoring force as the Brunt-Vaisala frequency, or natural frequency of oscillation,

$$N(z) = \sqrt{-\frac{g}{\rho}\frac{\partial \sigma_\theta}{\partial z}} \qquad \text{(units are rad/s)} \qquad (3)$$

It is the slope of the vertical density profile corrected for the effects of compressibility. If N is positive, heavier water is on the bottom and the water column is stable. If N is zero, the water is of neutral stability; it is an indication of well-mixed water. If N is imaginary, then there is heavier water sitting on top of lighter water, or the stratification is unstable, and

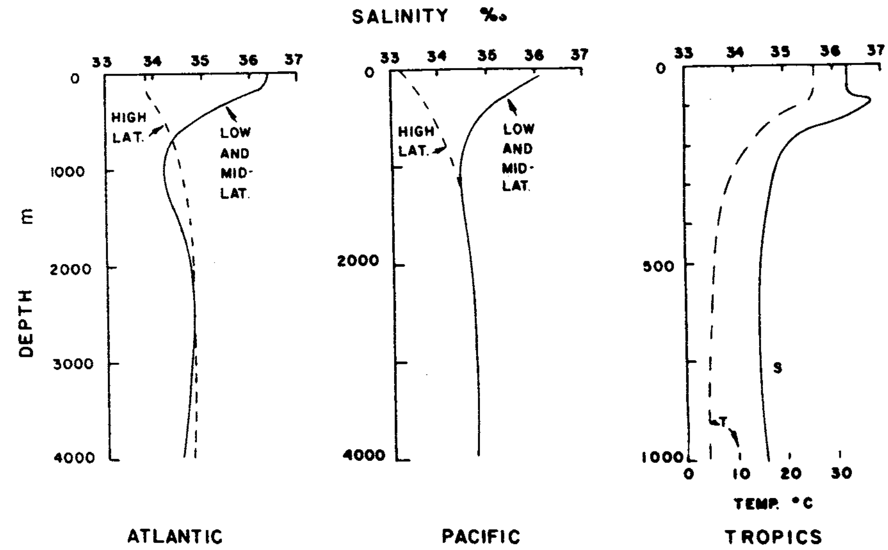


Fig. 5 Vertical variation of oceanic salinities

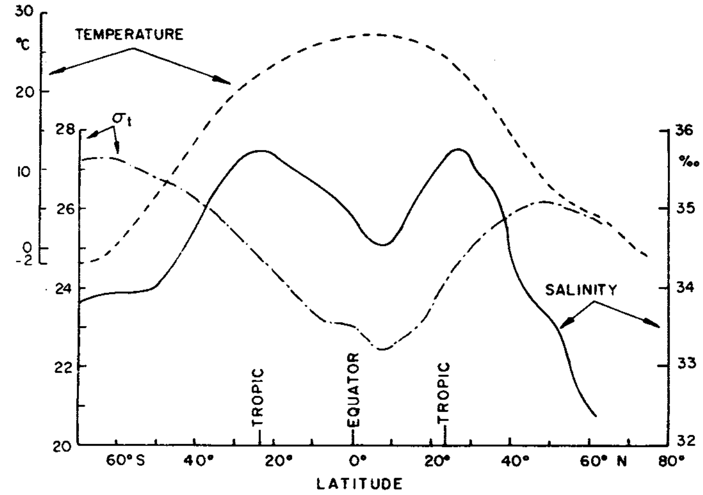


Fig. 6 Latitudinal variation of oceanic temperatures, salinities, and densities

one would expect vertical water motion. Therefore, one can express the vertical profile of density by either density as a function of the depth, or by the Brunt-Vaisala frequency, N, as a function of the depth.

Figure 6 shows the variation of temperature, salinity, and density (sigma-t) with latitude. Density is largely controlled by temperature except in polar regions where salinity variations become significant. Representative vertical profiles of density are shown in Fig. 7. The regions of high-density gradient (high N) are called pycnoclines and indicate regions of high stability. These regions strongly resist displacement, but can trap internal waves, which can in turn interact with a submersible operating in or passing through these regions.

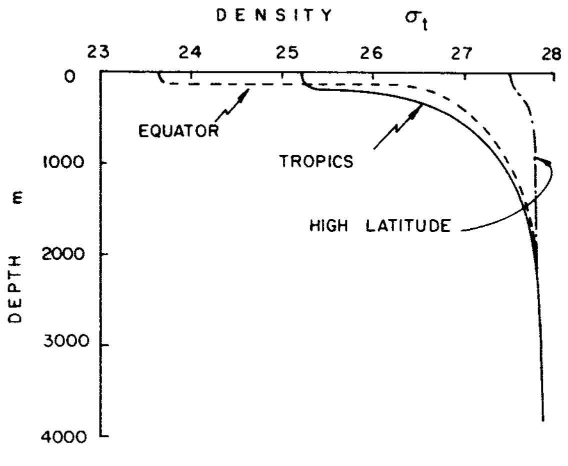


Fig. 7 Vertical profiles of densities at various locations

2.4 Temperature-Salinity Relationship

Water from different parts of the ocean tends to have different and distinctive temperature and salinity characteristics. Oceanographers have classified "water masses" by their T-S relationship. Figure 8 shows the principal water masses of the world defined and shown for comparison. Superimposed on the T-S plot are lines of equal density, sigma-t. Oceanographers use such relationships to study water motion and mixing, but they can also give submersible designers a summary of the temperatures and densities they will encounter in different parts of the world's oceans.

2.5 Specific Heat

The specific heat of sea water, C_P, is defined to be the heat in joules required at a constant pressure to raise the temperature of one kilogram of sea water one degree Centigrade. Thus the units are J/(kg°C). C_P is a function of temperature, T, salinity, S, and pressure, P. For sea water, the specific heat increases with temperature and decreases with salinity and pressure. Typical values are around 4000 J/(kg°C). The best empirical fit at zero pressure is given by Millero et al., 1973, which extends to low temperatures. The pressure effect has not been directly measured, but has been estimated by Fofonoff and given in the UNESCO papers, 1983.

2.6 Sound Velocity

Because sea water is compressible, it can support waves of compression and expansion, or small per-

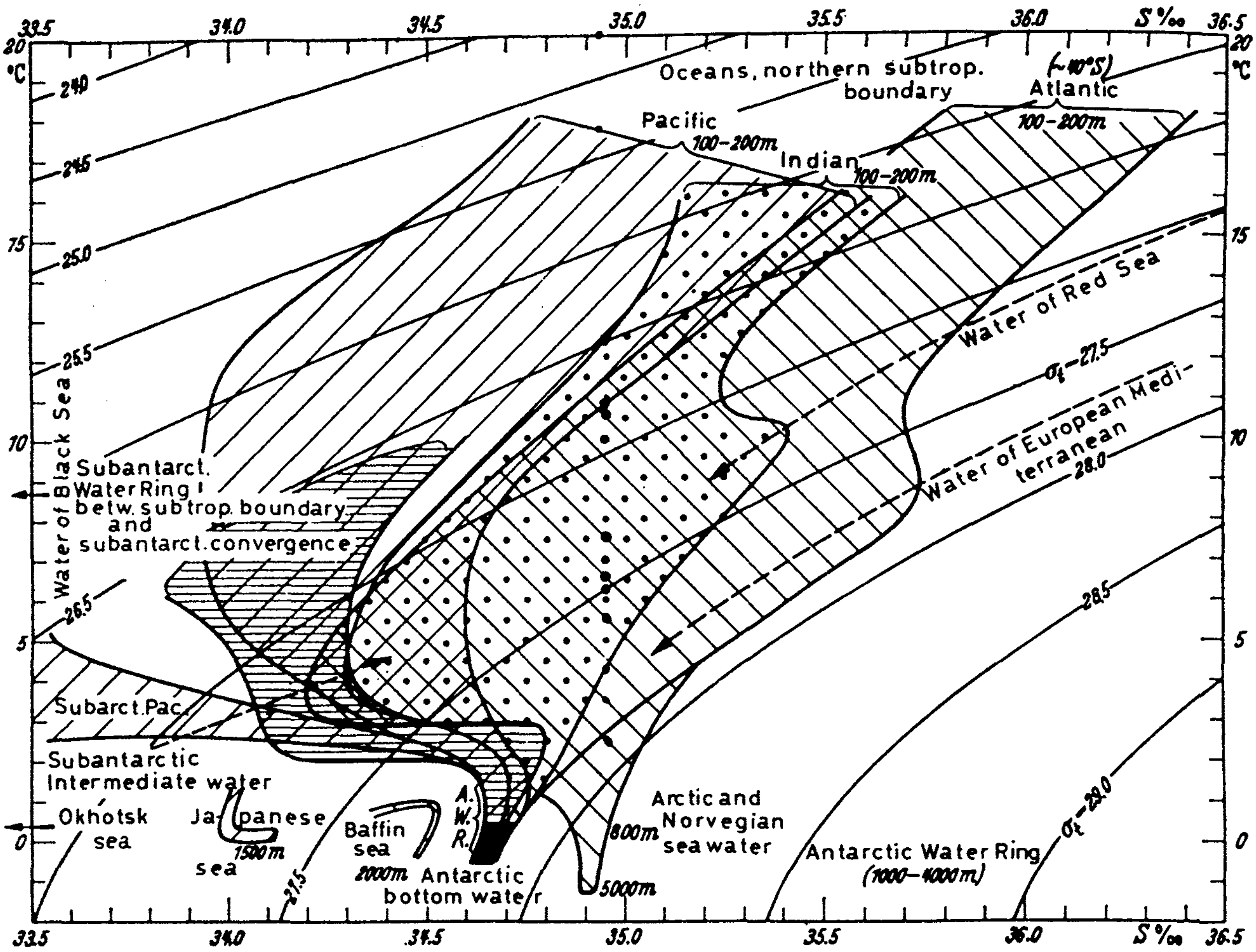


Fig. 8 Relation between temperature and salinity below the disturbed near-surface layer for the entire world ocean

turbations or fluctuations in pressure about the mean. We call these pressure oscillations sound, and they travel in the ocean with a velocity of about 1500 m/s, which is about five times faster than in air. The ocean is relatively opaque to all forms of electromagnetic radiation (from long radio waves through short ultraviolet), but transmits these sound oscillations much better than the atmosphere. Here acoustics are a valuable tool for communication and getting a "view" through the ocean. The velocity of sound is a function of the temperature, salinity, and pressure, increasing with all three factors. The most recent work by Chen and Millers, 1977, used standard sea water and so is consistent with the PSS-78 and in good agreement with values computed from the EOS-80.

Temperature effects dominate the sound velocity profile in the upper ocean, causing a decrease in sound velocity with depth. Pressure effects dominate in the deep ocean, increasing sound velocity with depth. Therefore, the resulting profile has a minimum at about 1200 m depth. The variations in sound velocity with depth have little effect on vertical or near-vertical transmissions of sound, as in communications between a submersible and a mother ship directly above. However, the sound velocity profile and the minimum or "sound channel" has important implications for horizontal or near-horizontal transmission of sound.

Sound traveling from a point source will spread out spherically, so that the sound energy will decrease as the square of the distance from the source. The limit of sound transmission is reached when the energy density decreases until it is equal to the ambient noise level (see below). If the energy were confined between two vertical surfaces, then in cylindrical spreading the reduction in energy would be proportional to the distance, so it will travel further before decreasing to ambient noise level. Sound travels along paths or "rays" which are governed by Snell's law:

$$C_1 \sin\theta_2 = C_2 \sin\theta_1 \qquad (4)$$

where C is the velocity of sound, θ is the angle of the ray from vertical, and the subscripts refer to the different layers. Hence it is obvious that the rays are

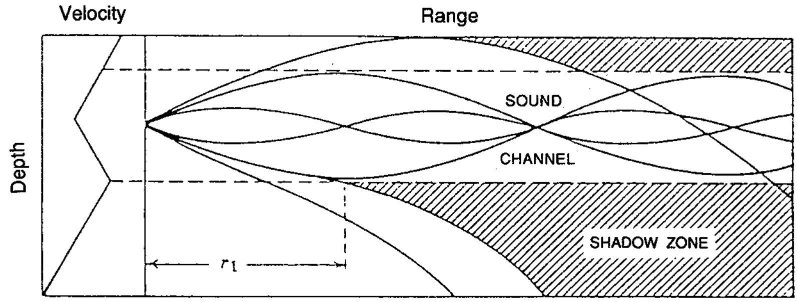


Fig. 9 Acoustic ray trace showing sound channel and shadow zone

bent toward lower velocity, and that a ray path from a source at the depth of the sound channel will be refracted toward the sound channel. Hence the rays are concentrated at that depth and consequently travel a large distance. The energy contained within a solid angle marked by four rays being emitted from the source is constant, so when the energy is concentrated in the sound channel, the spreading is closer to cylindrical than spherical. Hence, it is not surprising that large distances can be obtained by transmitting near the depth of the sound velocity minimum.

There are other source velocity profiles which are of greater concern to the submersible operator. Consider the case of a mixed layer where the sound velocity is controlled by pressure and increases with depth. Any sound transmitted into this region will be bent toward the surface. If this layer is above a layer with decreasing temperature which dominates over pressure, then rays penetrating into this layer will be refracted toward the bottom. This results in a shadow zone where acoustic communication or detection is impossible (Fig. 9).

2.7 Hydrostatic Pressure

For most practical purposes, the vertical equation of motion reduces to

$$\frac{\partial P}{\partial z} = \rho g \tag{5}$$

and integrating from the surface (where we take the pressure to be zero) to depth z gives

$$P(z) = \bar{\rho} g z \tag{6}$$

where $\bar{\rho}$ is the average or integrated density to depth z. This states that the pressure at a depth is due to the weight of water over it, and ignores any dynamic effects. P is sometimes called the hydrostatic pressure, because if all motion were to cease, the equations of motion would simply reduce to equation (6). This also gives a method to convert pressure (as measured by a pressure sensor) to depth. If one assumes the density of an average ocean profile, then equation (6) converts gage pressure to depth. For example, a pressure of 4062 dbars (5891 psi) converts to a depth of 4000 m (see Sanders and Fofonoff, 1976, or UNESCO, 1983, for further details and limitations).

2.8 Acoustic Ambient Noise

The limitation to acoustic transmission and detection in the ocean is the ambient noise level. This is due to a number of factors which add up to a background spectrum, as shown in Fig. 10. The noise level decreases with increasing frequency. Chief sources of low frequency noise (1 to 100 Hz) are seismic activity and explosions. In mid-frequency ranges (10 to 1000 Hz), ship noise is predominant except during rainstorms, when the noise of the rain hitting the surface becomes significant. At high frequencies (100 to 10 000 Hz), wind-generated noise dominates and is a function of wind speed.

2.9 Freezing and Sea Ice

The freezing temperature of sea water is dependent on the salinity and pressure of the water. Millero and Leung, 1976 (or *UNESCO*, 1983), give an empirical form to laboratory measurements at low pressure. For example, sea water of 35‰ at the surface has a freezing temperature of −2.54°C. As sea water freezes, the salt settles out in brine channels, which, because it is thus concentrated salt, is more dense and settles. Thus cold, salty water tends to form in areas where sea ice is being formed.

In regions where ice is melting, the salinity will be low, and a low salinity or freshwater layer is found at the surface. This reduces the density considerably and has implications for the buoyance requirements of a submersible which is operating here.

2.10 Coefficient of Thermal Expansion

One of the unique properties of fresh water is the reversal in the sign of the coefficient of thermal expansion at 4°C. Thus, water starting to freeze at 0°C is less dense than water at 4°C. As the salinity increases, this temperature of maximum density

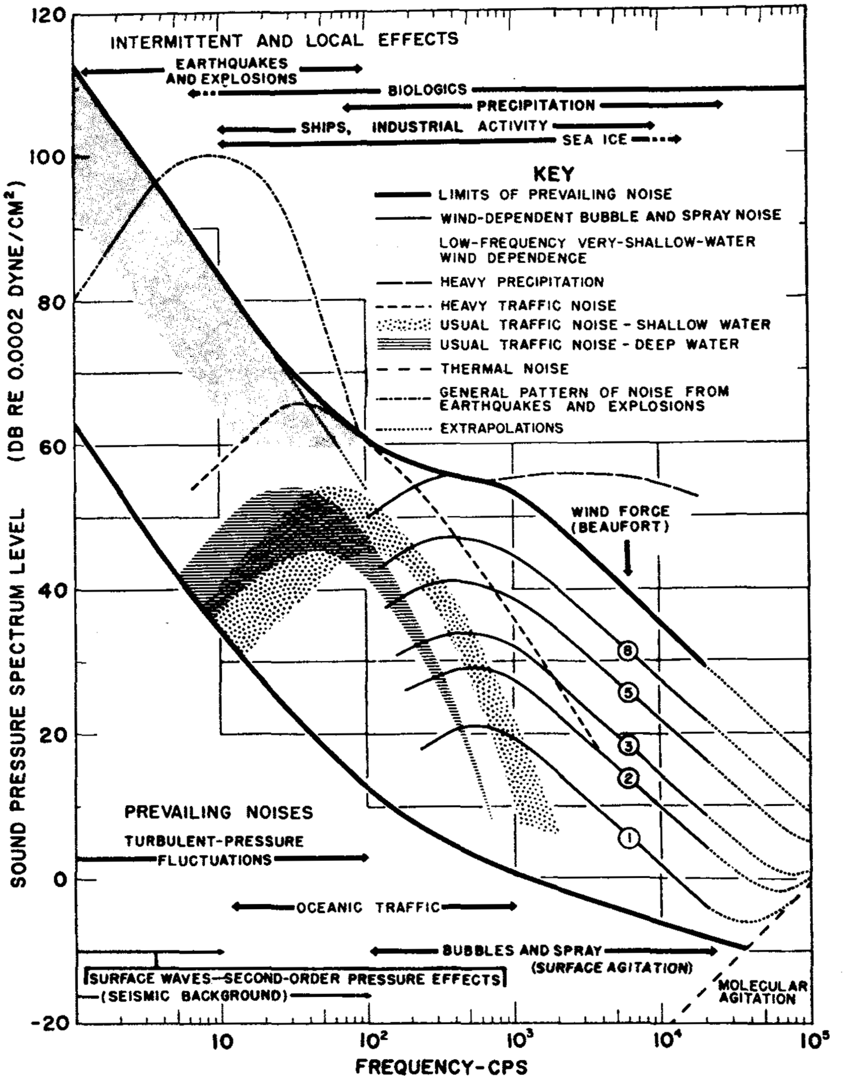


Fig. 10 Background spectrum of ambient noise level in the ocean

decreases. At a salinity of 24.7‰, the freezing point and temperature of maximum density are equal at −1.33°/C. For salinities greater than 24.7‰ the water continues to decrease in density with decreasing temperature until the freezing point is reached. A typical value of the coefficient of thermal expansion is 2×10^{-4}°C. This value increases with temperature and pressure.

3. Dynamical Processes

3.1 Surface Gravity Wind Waves

Waves are an efficient method of transporting energy from one point to another. The waves transport energy but, unlike ocean currents, do not transport the water itself. Figure 11 summarizes the energy density spectrum (proportional to wave amplitude squared) of surface waves according to wave frequency. The major energy in surface waves appears at the once- and twice-a-day tidal frequencies and in the 1- to 40-s wind wave frequency band. Waves are like simple harmonic oscillators where the density discontinuity at the air/sea interface and the force of gravity form the restoring force analogous to the mass and spring used in introductory physics classes. In addition to gravity for the long-period waves such as tidal motion, the effects of the earth's rotation become apparent in the restoring force. To gain an understanding of the principles involved, we shall first examine a single frequency, small amplitude, simple sinusoidal wave at the ocean's surface and then expand our description of the full ocean spectrum shown in Fig. 11 as a sum of these simple waves.

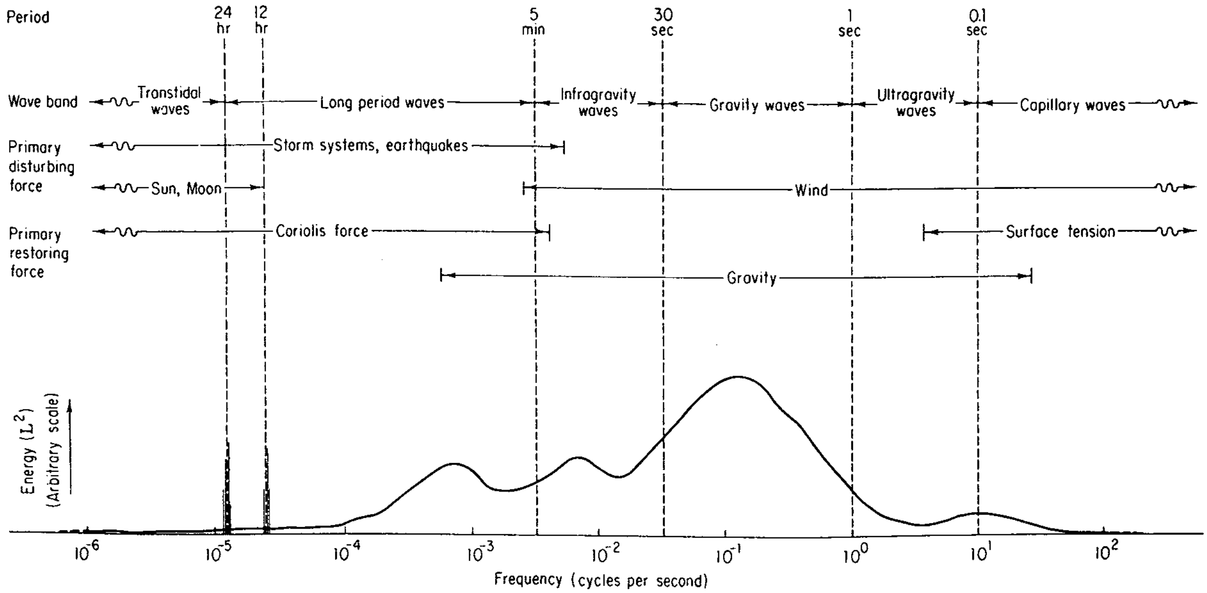


Fig. 11 Estimated schematic representation of the energy contained in the surface waves of the oceans

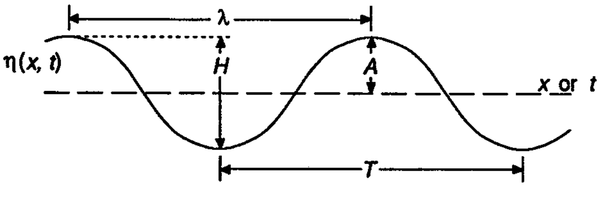
Waves can be characterized by their period, wavelength, speed, and amplitude. A simple sine wave, Fig. 12, has a wave height, H, equal to the distance from crest to trough and is twice the amplitude, A. The wavelength, λ, is the distance from crest to crest or trough to trough. The wave period, T, is the time between successive crests as seen at a fixed location. It is generally more convenient to refer to the wavelength and period in terms of the wave number, k, and frequency, ω, as

$$k=\frac{2\pi}{\lambda} \quad \text{and} \quad \omega=\frac{2\pi}{T} \tag{7}$$

The wave phase speed, C (often referred to as celerity), is the distance the wave crest travels in a period, or

$$C=\frac{\lambda}{T}=\frac{\omega}{k} \tag{8}$$

If we align the x-axis with the direction of wave propagation, the waveform of a simple progressive



λ = wavelength
H = wave height
A = wave amplitude
T = wave period
η = waveform

Fig. 12 Characteristics of a wave

wave can then be described as a function of space, x, and time, t, in the general form

$$\eta(x,t)=A\cos(kx-t-\phi) \tag{9}$$

where ϕ is an arbitrary phase chosen at $x = T = 0$.

Assuming that the wave height is much less than the wavelength or water depth, and making other suitable simplifications to the Navier-Stokes equations, a set of simple solutions (often called the Airy solution) can describe the water velocity distribution associated with the wave in equation (9). (For the development of these solutions, see Kinsman, 1965, etc.)

The frequency, ω, of an Airy wave is related to its wave number, k, by the dispersion relationship,

$$\omega^2 = gk\tanh(kh)$$

$$C^2 = \frac{g}{k}\tan(kh) \tag{10}$$

where g is the acceleration due to gravity and h is the water depth. The phase velocity can be determined from equation (10), but not simply, since we cannot get an explicit solution to ω/k.

The energy in the surface waves travels at the group velocity, C_g, which can be determined from the dispersion relationship by

$$C_g=\frac{\partial\omega}{\partial k} \tag{11}$$

Again, like equation (10) this does not have a simple solution because k appears as the argument of the hyperbolic tangent as well as a multiplier.

To simplify the above equations, we note that the hyperbolic tangent has two limits. When kh is small

Table 2 Sinusoidal progression waves

Term	General Expression	Deep Water	Shallow Water
Surface Elevation $\eta(x, t)$	$\eta = A\cos(kx - \omega t)$		
Phase Velocity C	$\sqrt{\frac{g}{k}\tanh(kh)}$	$\sqrt{\frac{g}{k}} = \frac{g}{\omega}$	$\sqrt{gh}$
Group Velocity C_g	$\frac{g}{2\omega}\frac{\sinh(kh)\cosh(kh) + kh}{\cosh^2(kh)}$	$\frac{1}{2}C = \frac{1}{2}\frac{g}{\omega}$	$C = \sqrt{gh}$
Orbital Velocity $w(x, z, t)$	$A\omega\frac{\cosh[k(z+h)]}{\sinh(kh)}\cos(kx - \omega t)$	$A\omega e^{kz}\cos(kx - \omega t)$	$A\sqrt{\frac{g}{h}}\cos(kx - \omega t)$
Orbital Velocity $u(x, z, t)$	$A\omega\frac{\sinh[k(z+h)]}{\sinh(kh)}\sin(kx - \omega t)$	$A\omega e^{kz}\sin(kx - \omega t)$	$A\omega(1 + z/h)\sin(kx - \omega t)$
Pressure Deviation $\Delta p(x, z, t)$	$\rho gA\frac{\cosh[k(z+h)]}{\cosh(kh)}\cos(kx - \omega t)$	$\rho g\eta e^{kz}$	$\rho g\eta$
Bottom Pressure $\Delta p(x, -h, t)$	$\frac{\rho gA}{\cosh(kh)}\cos(kx - \omega t)$	0	$\rho g\eta$

(less than 0.3), then $\tanh(kh)$ is approximately kh. When kh is large (greater than 1.5), then $\tanh(kh)$ is approximately 1. The case where the wavelength is small (large k) as compared with the depth h (kh is large), the solution simplifies to the "deep water" case of $\omega^2 = gk$. When the wavelength is large compared with the depth (kh is small), then the solution simplifies to the "shallow water" case of $C^2 = gh$. If kh is neither large or small, then the full dispersion relationship must be used.

The Airy solutions can then be simplified to deep and shallow water approximations, as well as the full solutions for intermediate water. These relationships as a function of time and depth in the water are tabulated for the three cases in Table 2 and discussed below and illustrated in Fig. 13.

3.1.1 Shallow Water Solutions

When the wavelength is greater than 20 times the depth ($kh < 0.3$), then the shallow water approximations apply and the depth becomes the controlling factor. The phase velocity and group velocity are the same $\sqrt{gh}$ and are not dependent on k. The horizontal component of velocity is not a function of depth, but is constant from top to bottom. The vertical component of water motion decays linearly from its maximum at the surface to zero at the bottom. The pressure under a shallow-water wave also is not a function of depth (no attenuation), but is just the hydrostatic pressure due to the amount of water above. These solutions describe long waves such as tides and tsunamis and normal wind waves when they propagate into shallow water near the coast.

When two progressive waves traveling in opposite directions are added, or a wave is reflected, a standing wave results. The velocity under a standing wave is quite different and is illustrated in Fig. 14.

3.1.2 Deep Water Solutions

When the water depth is greater than one quarter the wavelength ($kh > 1.5$) then the deep water approximations apply and the water depth becomes unimportant. The horizontal and vertical components of velocity are equal and the orbits become circles which decrease exponentially as a function of depth. Their motion becomes negligible (decays to $1/e$ of the surface value) at a depth equal to one half the wavelength. Therefore, for a typical 10-s swell, $C = 16$ m/s and $\lambda = 156$ m. The wave particle velocity is 1.6 m/s at the surface and decays to 0.7 m/s at 78 m = wavelength/2.

The group velocity now becomes half the phase velocity, and we have what are referred to as "dispersive" waves. The energy travels only half as fast as the wave form, so if one has a finite "packet" of energy, one will see individual waves appear at the back of the packet, grow in amplitude as they move up in the packet, and then die out as they move out the front of the packet. The packet itself moves with the group velocity, and the individual waves move with the phase velocity. Longer wavelength waves travel faster than shorter wavelength waves. A storm generates a whole "spectrum" of frequencies: the longer (lower frequency) waves travel faster than the shorter (higher frequency) waves, and the waves "disperse." This becomes evident when watching the swell generated by distant storms. On successive

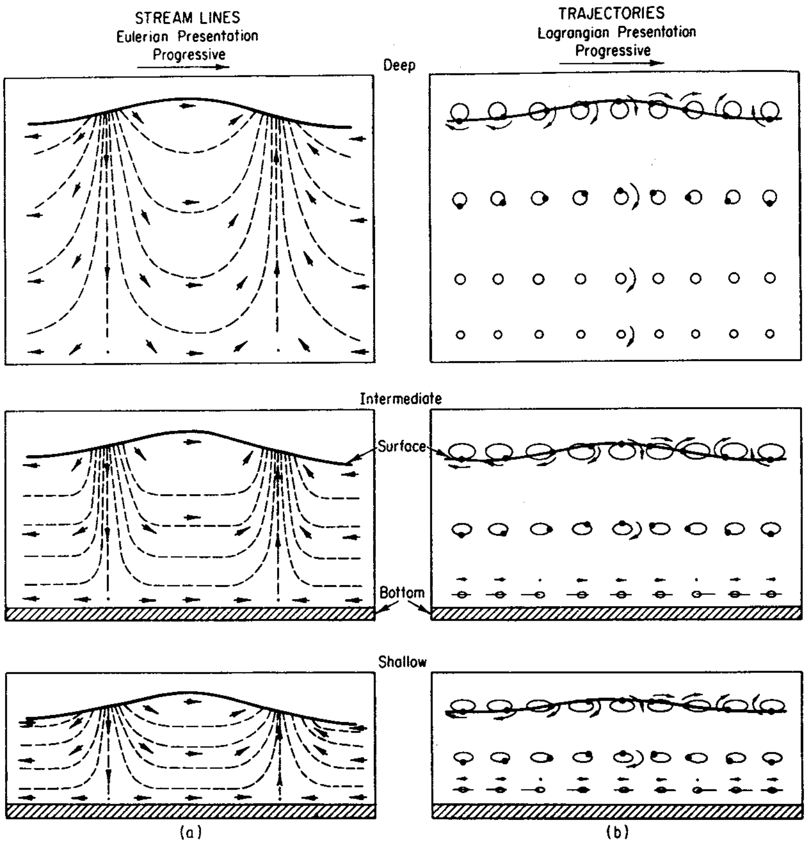


Fig. 13 Relationships of Airy solutions in deep, intermediate, and shallow waters

days, the period of the observed swell will increase in frequency, because the slower, shorter wavelength, higher frequency waves will travel slower and arrive later than the faster, longer wavelength, lower frequency waves. Dispersion can easily be observed by throwing a rock into a still pond and watching the resultant disturbance spread out with time.

3.1.3 *Wave Energy*

The energy in the Airy wave is equally partitioned between potential energy (displacing the sea surface from its equilibrium position) and kinetic energy (water particle motion). The average kinetic energy (KE) per square unit of surface area from the surface to the bottom is found by integrating from the surface to the bottom and over one wavelength. Similarly, the average potential energy (PE) per surface area is found by integrating the displacement, and is found equal to the kinetic energy.

$$\mathrm{KE} = \mathrm{PE} = \frac{1}{4}\rho g A^2 \tag{12}$$

Hence the total energy is ½($\rho g A^2$) or, in terms of the wave height,

$$\text{total energy} = \frac{1}{8}\rho g H^2 \tag{13}$$

This could be thought of as the work required to lift a sheet of water of thickness H through a vertical distance of ⅛ the wave height, or the energy in a layer of water of depth H moving at a velocity of the square root of ¼(gH). These results are independent of the wave period or frequency.

The energy in small amplitude surface waves travels at the group velocity, C_g. The rate of transmission of energy per unit of wave crest for deep water waves becomes

$$\text{energy flux} = \frac{1}{32}\rho g^2 H^2 T \tag{14}$$

The flux of power in a given length of wave of wave crest s is about $10^4\ H^2Ts$, so for a typical 4-ft wave with a 10-s period along a 100-ft-long section of wave crest, the flux of power is 4.5×10^{10} ergs/s or about 0.02 hp.

3.1.4 *Significant Wave Height*

When one observes the sea surface, one is looking at the sum of many individual waves. The eye is very

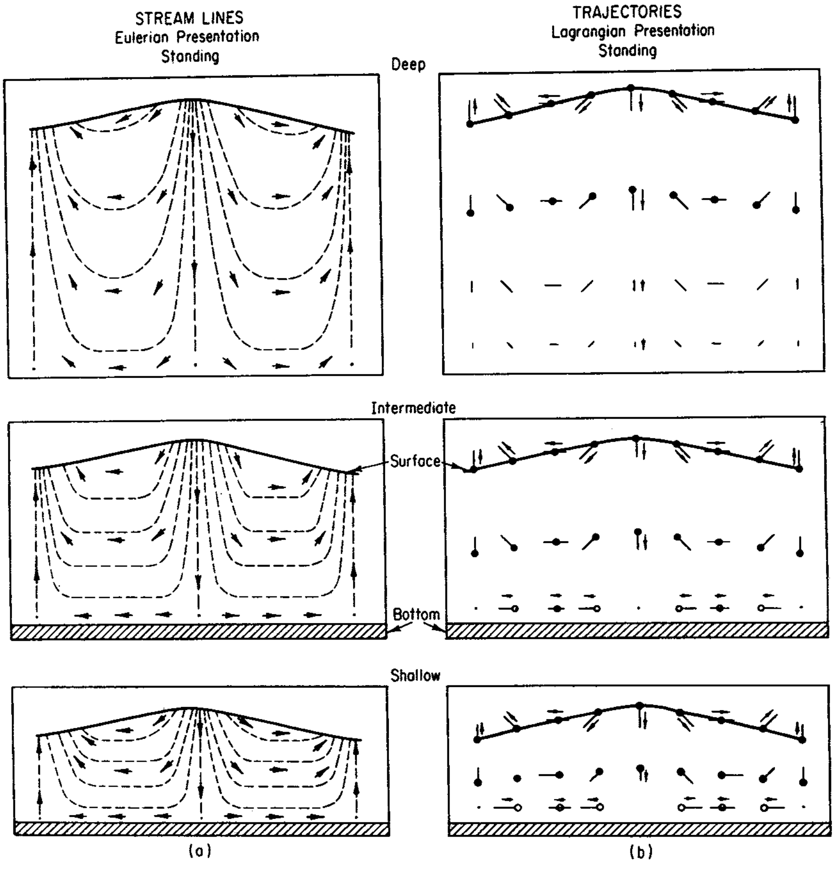


Fig. 14 Summary of orbital velocities beneath a standing wave

poor at estimating the wave height of an individual wave in this complicated sum, but if an experienced observer were to estimate the average wave height, he or she would come close to the significant wave height, $H_{1/3}$, which is defined as the mean of the highest one-third of the waves present. This parameter does not have any "significance" in itself, but it is used by certain models in forecasting ocean waves. An approximation of the significant wave height as a function of wind speed is

$$H_{1/3} = 1.82 \times 10^{-2} U^2 \tag{15}$$

where H is in feet and U is in knots measured 64 ft above the sea surface.

3.2 Energy Density Spectrum of Surface Waves

Figure 11 gives the spectrum of sea-surface elevations and illustrates that the sea surface is not the simple Airy wave discussed above, but something more complex. However, studies have shown that if we consider the sea surface as the sum of a large number of different Airy waves of different frequency traveling in different directions, these simple solutions apply surprisingly well, except in shallow water regions where the depth changes rapidly and the wave height increases rapidly, and friction and dissipation effects become important. The spectrum shown does not consider any directionality, but could be obtained from a measurement of sea surface elevation at a single point. Measurements of directional spectra require either several sensors in an antenna or direction sensitive instrumentation. Directional wave spectra (Fig. 15) show a much more complicated picture of the wave energy and its direction of propagation.

3.3 Wind-Generated Waves

Wind blowing over the ocean's surface creates the oscillations we call waves. To describe statistically the sea surface, one looks at the frequency domain description or the power spectral density of the sea surface, which shows the amplitudes of the various sinusoids. These are added together at each frequency to describe the observed sea surface. Figure 16 shows a number of spectra measured at different wind speeds. It is clear that the faster

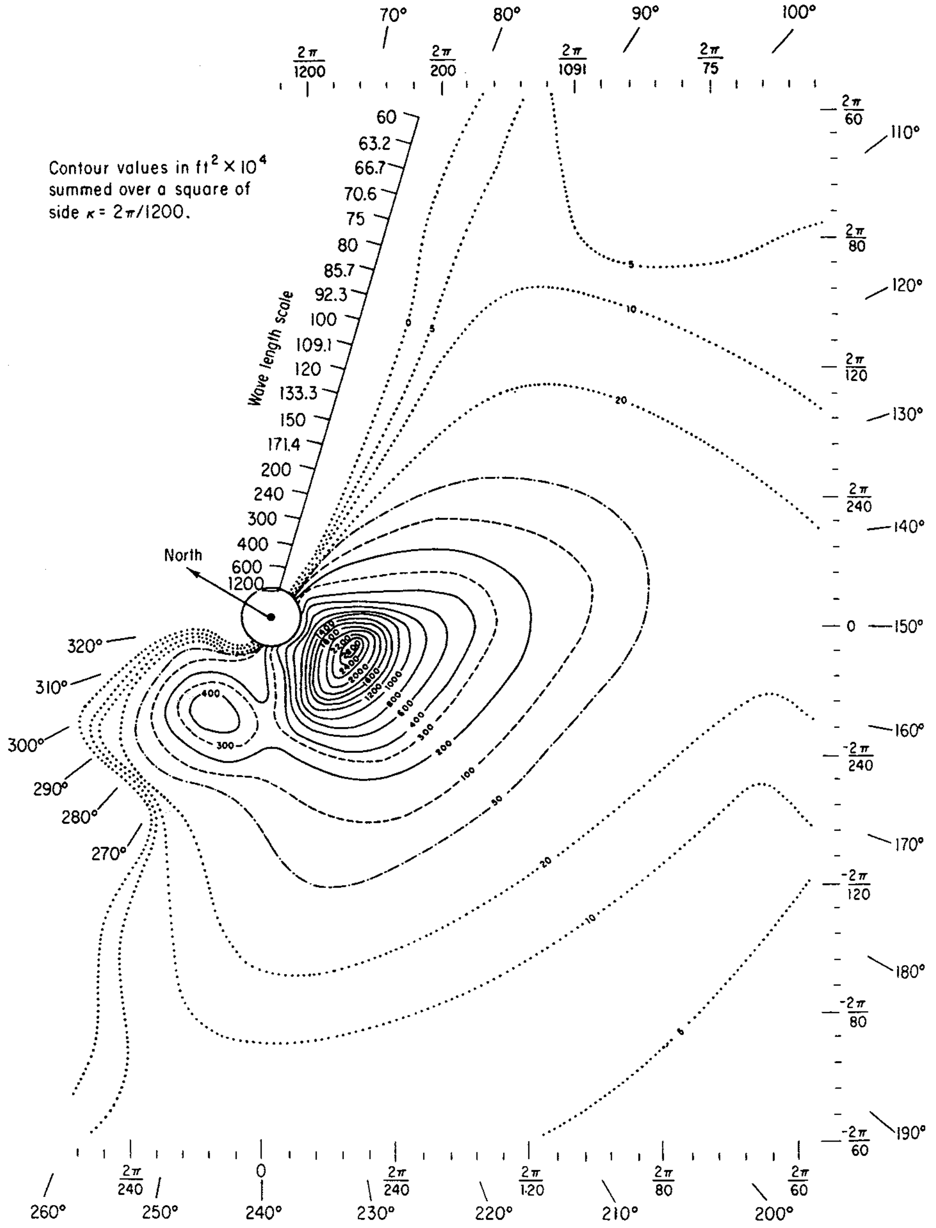


Fig. 15 Smoothed directional spectrum of wind-generated waves

wind speeds produce spectra with higher energy waves at lower frequency. When the wind blows for a length of time, the sea builds up to a state in which the waves break, having too much height for their wavelength. Therefore, we see spectra which have less energy (lower amplitudes) at higher frequencies and higher amplitudes at lower frequencies. The higher frequency end of the spectrum (at higher frequencies than the peak) is described by a frequency to the fifth-power law. The slope of the waves can be determined from our simple wave theory, and the mean square slope increases with frequency, so the higher frequency waves have the greater slope, and they break the earliest. Thus the spectrum can be described as a saturation spectrum, because if more energy is input at a specific frequency, the waves cannot get higher without breaking, and the spectrum reverts to the equilibrium. This can be modeled as

$$\text{energy density} = \frac{Bg^2}{f^5} \tag{16}$$

The low end of the spectrum is dependent on the fetch and duration of the wind. It is obvious that it

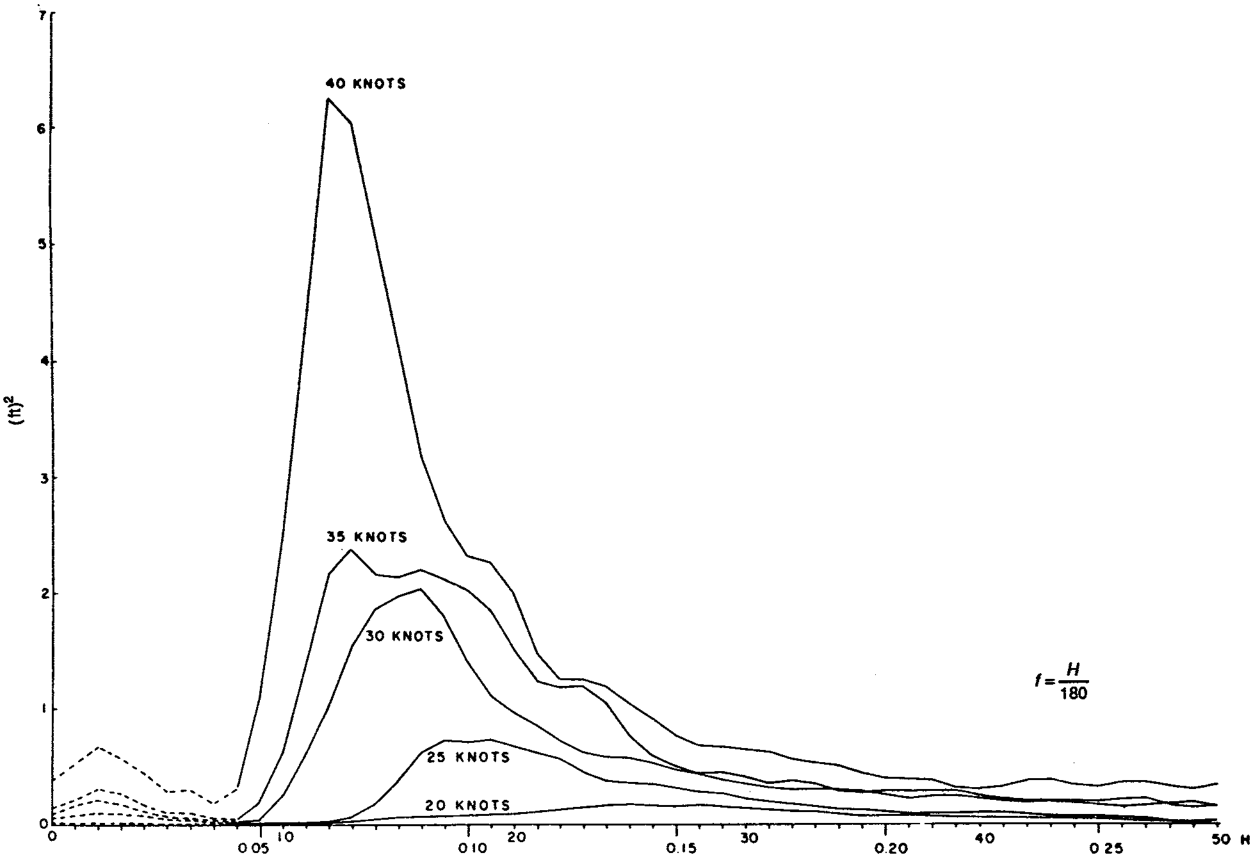


Fig. 16 Averages of selected spectra for winds from 20 to 40 knots

takes some time for the ocean to reach an equilibrium state or fully developed sea once the wind has started blowing. Therefore, the duration of the wind is important. Also, the wind has to blow over an area or distance, called fetch, before the equilibrium state is reached. If there is not enough duration or fetch, the spectrum does not reach equilibrium, and the waves are said to be duration or fetch limited. Figure 17 shows an example of combined duration and fetch as a function of integrated spectrum E and wave frequency f. As an example, if a 32-knot wind has blown for 20 h over a 200-nautical-mile fetch, the 20-h line intersects the 32-knot cumulative spectrum line at $f = 0.09$ Hz and $E = 46$ ft^2. The sea is duration limited, because if it blew for 28 h, we would get an E of 80 ft^2, which is nearly the maximum. (Note that the significant wave height for the 20-h wind is 2.83 times the square root of E—about 18 ft.) Also by the same arguments, the 200-nm fetch is limiting the waves. A fetch of greater than 500 nm would be required for our 32-knot wind to avoid fetch limitations.

Assuming a fully developed sea, the spectrum should cut off at some low frequency point which is a function of the wind. This is often modeled as another term in equation (16) as

$$\text{energy density} = \frac{Bg^2}{f^5}\exp\left(-\frac{D}{(fU)^4}\right) \tag{17}$$

where B and D are both numerical constants ($B = 8.27 \times 10^{-7}$ and $D = 62.368$), f is the frequency in Hz, and U is the wind velocity in knots. The energy density is then ft^2/Hz. This can be integrated to get the total energy, E, in a fully aroused sea, as used in Fig. 17.

3.3.1 *Shoaling Effects*

All waves propagating in the ocean will either die out gradually due to friction or eventually reach the coast and be dissipated. There is no effective change in swell as they move toward the coast until they reach shallow enough water that they "feel" the bottom. Then the depth controls the wave velocity, and refraction can occur. The depth at which the waves are influenced by the bottom is dependent on the wavelength as discussed above. As the bottom shoals, the wave velocity decreases and the wavelength shortens (the wave period or frequency is conserved). If the wave is traveling perpendicular to the bathymetry so that no refraction occurs, the

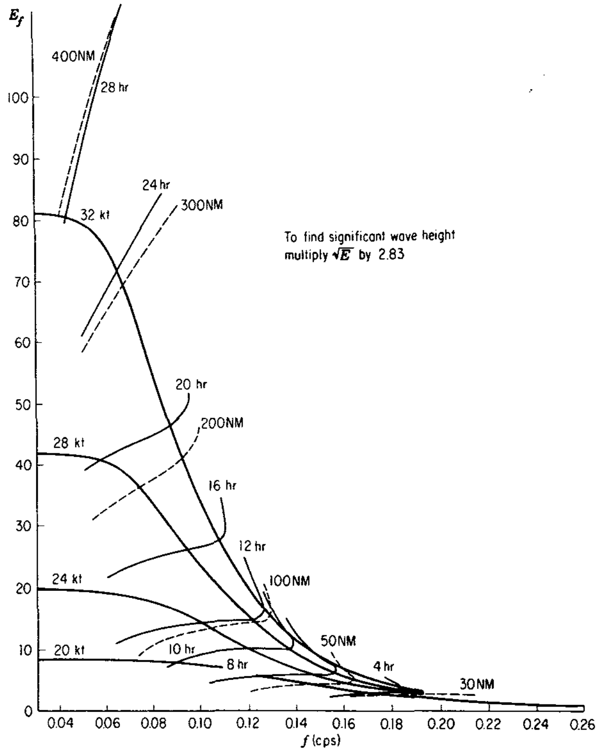


Fig. 17 Example of combined duration and fetch

wavelength decreases so that the energy per square unit surface area increases, and the wave height must increase proportionally.

It is possible to draw orthogonals perpendicular to the wave crest or to trace the path of a point (the path is called a ray) on the wave crest as it travels toward the beach. As the wave approaches the beach at an angle, the part of the crest in shallower water will travel slower, and the wave crest will become more parallel with the coast. Diagrams of the paths of waves can be made knowing the bathymetry, as shown in Fig. 18. Again ignoring friction, the energy between two rays and between two crests remains constant. Therefore, as the distance between rays decreases, the energy density increases, as does the wave height. At a headland or around islands, the wave energy is focused and wave heights are generally greater. If the bathymetry is such that the waves diverge, then the energy density and wave height decrease. Fishermen who anchor over Scripps Canyon have discovered and used this fact. Generally, these effects are only important in shallow water and should only be of concern when operating in coastal regions.

3.4 Internal Waves

In addition to surface waves at the air/sea interface, the density structure within the water column can support a class of waves called internal waves. These waves differ from the surface waves discussed above in that the maximum particle displacements appear below the surface, and the typical frequencies are much lower. Internal waves can exist on a density gradient because when displaced from equilibrium, the density difference when acted on by gravity causes a restoring force. Because the restoring force is smaller and the dynamics different, the frequencies of internal waves are much lower. The lowest frequency is the local inertial frequency: $f = 2$ sin(latitude) cycles per day. Thus at 30°N, the inertial frequency would be one cycle per day. The highest frequency possible is the local Brunt-Vaisala frequency (see properties section), which is typically one cycle per hour (cph) in the deep ocean and up to 10 cph in the thermocline.

Interfacial waves are often confused with internal waves. Interfacial waves are waves which propagate along a density interface. A good example of these are in the Norwegian fjords, where one often encounters a layer of fresh water overlying a layer of salt water. Fishermen often have encountered this interface at about keel depth, and have referred to experiencing this effect as running into "dead water," because their boat slowed down as energy was abstracted from its forward motion to generate waves along the interface. Our surface gravity waves are really interfacial waves, propagating along the interface between air and water. The theory for surface waves can be converted to interfacial waves by substituting a reduced gravity which is multiplied by the density difference divided by the density. Since the density of air is nearly zero compared with water, this factor is nearly one, and so is ignored.

Internal waves, instead of propagating along interfaces, propagate along rays called "characteristics." The slope of the characteristic is a function of the internal wave frequency, ω, the inertial frequency, f, and the Brunt-Vaisala frequency, N.

$$\text{slope} = \frac{\omega^2 - f^2}{N^2 - \omega^2} \qquad (18)$$

Energy propagates along this characteristic, which is in the direction of the group velocity. However, unlike surface waves, the phase velocity is at right angles to the group velocity. Thus, at the limit where the frequency approaches f the slope approaches zero, and the energy is trapped in a layer of the ocean; we have what are referred to as inertial currents where the water moves in an inertial circle with a frequency equal to f. As the frequency increases, the slope becomes non-zero, and now energy can propagate down into the ocean from a source at the surface. At the upper limit, N, the group velocity is vertical, the energy is trapped between the surface and bottom, and the water particle motion is merely vertical.

Next Page

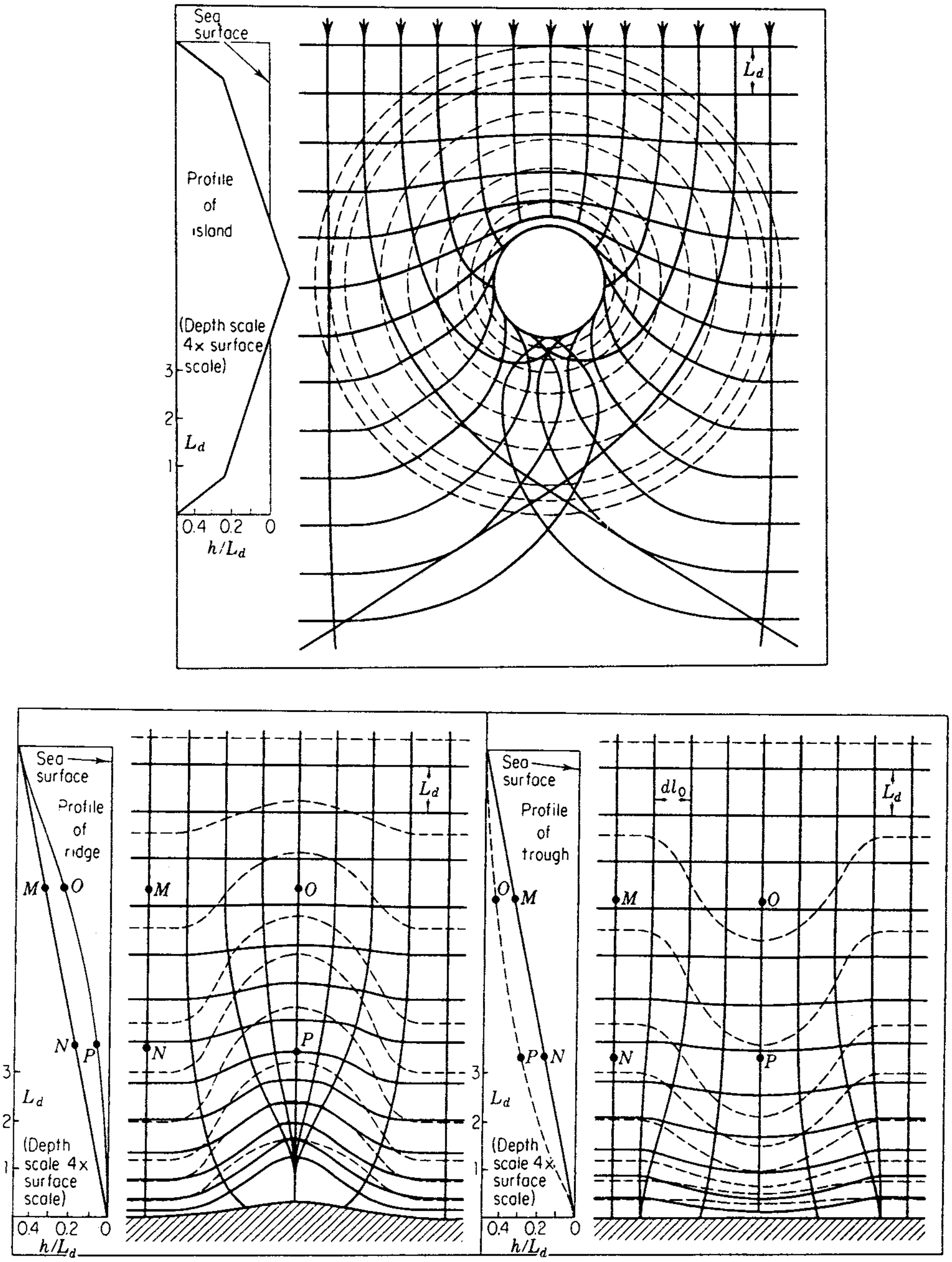


Fig. 18 Wave refraction around a circular island, over a ridge, and over a trough

As in our discussion of surface waves, the observed oscillations are really a sum of different waves propagating in different directions. To describe these waves statistically, Garrett and Munk formulated a model which collected recent observations and created a description of "open-ocean" (away from any boundaries or sources) internal waves. From this theory, a typical root-mean-square (rms) variation for an internal wave is 15 m vertical motion with a period of about 1 h. Since open-ocean internal waves have such a long period, they are probably not of strong importance to the submersible designer. However, there are several regions where they are very important motions.

Tidal currents interacting with the continental shelf break or seamount can scatter energy from the surface or barotropic tide to an internal or baroclinic tide. This energy will propagate in a beam along the characteristic out into the ocean. Along the west coast of the United States, the characteristic slope is

Chapter IV
Materials

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1. Overview

THE KEEN DESIRE to explore the ocean depths is stimulated not only by our curiosity into unknown areas, but also by the natural resources which the ocean gives us. In order for us to successfully explore and exploit marine resources, a system consisting of various subsystems (such as deep-diving vehicles and supporting facilities) is indispensable. The successful development of such a system will depend upon the availability of materials. Extensive study has shown that advanced materials with a diversity of properties will be required. Consequently, the selection of materials for such a system cannot be discussed without reference to a particular marine application and the material requirements of that application.

Among various subsystems, the requirements imposed upon the materials for submersible vehicles are the most demanding. The submersible hull must endure extreme pressure and, at the same time, carry a maximum payload for a crew and necessary equipment. Thus, hull material must have a high strength-to-density ratio, and a high fracture toughness. Furthermore, since a submersible will be operated in a very hostile environment, the seawater, the hull material must be corrosion-resistant. Hull materials, then, are advanced materials with which the ocean industry has as yet had no prior experience. For good maneuverability and good payload, ballast-buoyancy-trim materials will be used, also new to the ocean industry. However, there is no single hull or ballast-buoyancy material which satisfies all requirements: each material has both advantages and disadvantages, depending upon the mission requirements.

In Section 2 of this paper, material considerations are discussed in a general context—in terms of their environmental, manufacturing, and other characteristics. Section 3 then deals more specifically with the individual materials considered for use in a particular hull, with special reference to their mechanical, environmental, and fabricable properties. These materials include steels, aluminum alloys, titanium alloys, glass reinforced plastics, acrylic plastics, and glass. Section 4 covers ballast-buoyancy-trim materials and Section 5, other material considerations. Life Support System Materials are covered in the discussion of Life Support Systems in Chapter VII.

2. Material Considerations in General

Much of the information presented below comes from a book by Masubuchi [1].

2.1 Environmental Considerations

In this section, two factors related to the environmental effects on materials are discussed: corrosion and stress corrosion cracking.

2.1.1 Corrosion-Control Problems in the Marine Industry

Corrosion is often considered only in terms of rusting and tarnishing. Corrosion damage, however, also occurs in other ways, for example in failure by cracking, or in loss of strength or ductility. In general, each type (with some exceptions) occurs via an electrochemical mechanism. According to Uhlig [2], the five main types of corrosion classified with respect to outward appearance or altered physical properties are uniform attack (or general wasting), pitting, dezincification and parting, intergranular corrosion, and cracking.

Various control measures which have been developed over the years for protecting structures from corrosion have offered impressive economy to the shipbuilding and ocean engineering industries. De-

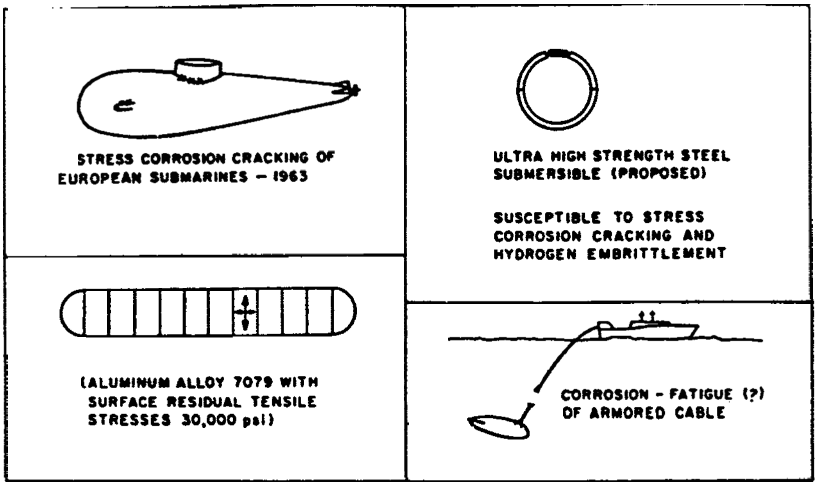

Fig. 1 Types of current or possible future corrosion control problems [3]

spite this wealth of information, corrosion is still one of the major problems facing ocean engineers.

Brown and Birnbaum [3] have summarized their views on corrosion control for structural metals in the marine environment. Their important conclusion is: "With new materials and new geometries of components and structures for various deep ocean projects, the primary need for corrosion control has shifted from one of maintenance economy (though of course this is as desirable as ever) to one of survival of the component or structure." Starting from this standpoint, let us review the current and possible future corrosion control problems.

Figure 1 shows various types of corrosion control problems facing the Navy [3]. Engineers engaged in the design and fabrication of ocean engineering structures with high-strength materials will probably face similar problems.

The first case in Fig. 1 is stress corrosion cracking of high manganese austenitic steels used in nonmagnetic European submarines. Service failures occurred at welds. On the basis of experience to date,

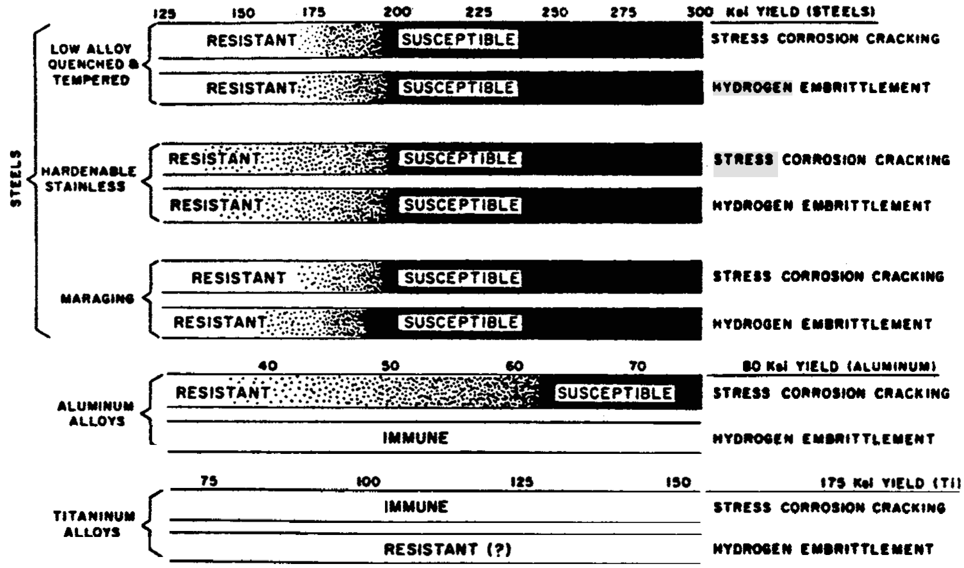

Fig. 2 Estimated strength ranges for susceptibility to cracking in sea water, fresh water, or humid atmospheres [3]. Note: Readers are cautioned that titanium alloys were regarded to be immune to both stress corrosion cracking and hydrogen embrittlement when this figure was originally drawn; however, it has been found that some titanium alloys, especially those with higher strength levels, are sensitive to both stress corrosion cracking and hydrogen embrittlement

it appears that an acidified salt solution containing hydrogen sulfide will crack more steels, and in general crack them more quickly, than any other medium. However, it is not entirely certain that this is the case with all steels.

The second case is a submersible of aluminum alloy 7079. This alloy has a high strength-to-density ratio, but it is susceptible to selective corrosive attack and stress corrosion. Even though the aluminum plates are protected by a coat of paint, cracking may occur at breaks in the coating, especially when there are tensile residual stresses on the surface.

The third case is the stress corrosion cracking of a proposed submersible of very high-strength steel (designated H-11). This steel is highly susceptible to stress corrosion cracking. If one attempts to counter stress corrosion in this material by the application of cathodic protection, a phenomenon intrudes to defeat this. This phenomenon is hydrogen embrittlement, to which all the high-strength steels (low-alloy, hardenable stainless, precipitation hardening, and maraging) are susceptible when they are heat-treated to sufficiently high strengths.

The current estimate of the strength ranges for the three classes of steels at which they become susceptible to stress corrosion cracking is given in Fig. 2. (The precipitation-hardening steels are lumped with the other hardenable stainless steels in this figure.) In preparation of this figure, only data for sea water, salt solutions (used in the laboratory), distilled water (representing condensate), marine atmosphere, and humid air were used. Omitted were data for the acidified salt solutions containing hydrogen sulfide and nitrate solutions, on the grounds that these are unrealistic for the marine environment. Figure 2 also indicates the range at which hydrogen embrittlement under sustained load might be expected to become a practical problem. Aluminum alloys which have the face-centered cubic lattice structure do not show hydrogen embrittlement, although some aluminum alloys, especially those with higher strength levels, are sensitive to stress corrosion cracking. Readers are cautioned that titanium alloys were regarded to be immune to both stress corrosion cracking and hydrogen embrittlement when Fig. 2 was originally drawn; however, it has been found that some titanium alloys, especially those with higher-strength levels, are sensitive to both stress corrosion cracking and hydrogen embrittlement.

The fourth illustrative case in Fig. 1 is one of corrosion fatigue of armored cable. It is obvious that if corrosion produces a pit, this can be expected to act as a stress raiser, which could accelerate the fatigue process, and this is observed. Perhaps somewhat surprising, however, is the observation that although the titanium alloy B120VCA is essentially inert to salt water, when it is fatigued in salt water the life is sharply reduced compared with the life in air. It might be supposed that adequate cathodic protection should remove the "corrosion" component of cathodic protection, and this is observed for this alloy (Fig. 3). But in the case of steels of almost any strength level, the application of cathodic protection beyond a certain level (defined for only a very few conditions for a very few steels to date), once again a form of hydrogen embrittlement, appears to intrude to counter the effect of corrosion protection and place a limit on the available effectiveness of cathodic protection. An example of data showing this is given in Fig. 4.

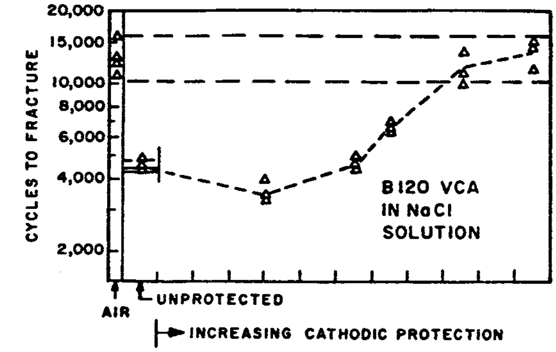

Fig. 3 Cathodic protection of high-strength titanium alloy against corrosion fatigue [1,4]

2.1.2 *Corrosion in the Marine Environment*

The following discussion deals with corrosion in the marine environment, first with respect to the general wasting of steel and then in terms of the various types of corrosion in other metals and alloys. Data presented here can be found in "Guidelines for Selection of Marine Materials," prepared by Tuthill and Schillmoller [5].

Corrosion of steels in sea water—It may be assumed that the rate of corrosion of steel in sea water is to a large extent governed by the oxygen content

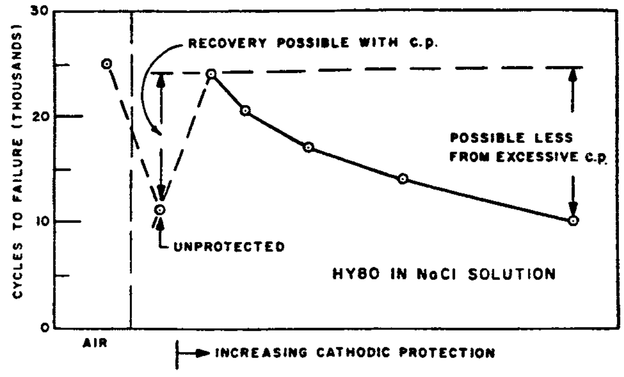

Fig. 4 Effect of cathodic protection on corrosion fatigue of steel at low-cycle frequency (100 cpm) [3]

of the sea water. The pattern of corrosion on carbon steel in aerated sea water is shown in Fig. 5, where the average corrosion rates typical of steel piling in the atmosphere, in the splash zone, in the tidal zone, in clean sea water, and in the mud of the sea bottom are plotted. The differences shown are due in part to the availability of oxygen and in part to other factors.

Capson [6] has proposed that the corrosion rate of steel in the atmosphere is directly related to the rate at which a ferrous corrosion product is leached or washed from the barrier film of rust. When one of the products of corrosion is soluble, the formation of a fully protective barrier film is almost impossible.

It has been postulated that in the marine atmosphere, small amounts of copper and nickel in the low-alloy steels enhance their corrosion resistance by altering the structure of the barrier film of the corrosion product so as to produce a tighter, denser film with less tendency to be removed by leaching or spalling.

In the splash zone, rust film that has little opportunity to become dry does not develop protective properties. Such unfavorable conditions exist, especially above high-tide level, aggravated further by the high oxygen content of the splashing sea water. As shown in Fig. 5, the corrosion rate in this region is several times greater than that in other regions.

Corrosion in the tidal zone reaches a minimum as a result of the protective action of oxygen concentration cell currents. Steel surfaces in the tidal zone in contact with highly aerated sea water become cathodic to the adjacent submerged surfaces where the oxygen content is less, especially when these surfaces are covered with oxygen-shielding organisms. The current that flows from the anodic submerged surfaces to the cathodic tidal zone areas is sufficient to provide substantial cathodic protection.

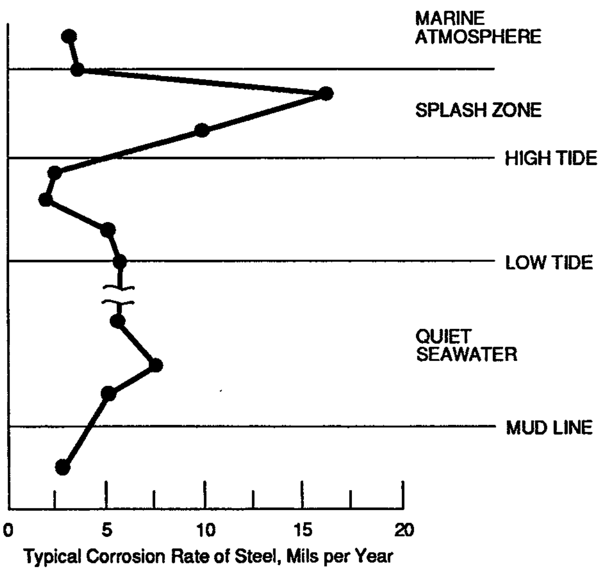

Fig. 5 Corrosion of steel piling [6]

The attack of submerged surfaces is governed principally by the rate of diffusion of oxygen through layers of rust and marine organisms. It usually is in the range from 3 to 6 mils per year and is substantially independent of water temperature and tidal velocity, except where industrial pollution leads to higher rates.

The rate may go up in the vicinity of the mud line because marine organisms can generate additional concentration cell and sulfur compound effects.

2.1.3 Stress Corrosion Cracking

Stress corrosion cracking differs from other types of metal attack in that it is a form of localized failure which is more severe under the combined action of stress and corrosion than would be expected from the sum of the individual effects of stress and corrosion acting alone. Stress corrosion cracking involves a brittle-type fracture in a material that is otherwise ductile. The surface direction of the cracks is perpendicular to the direction of the load. Stress corrosion cracking should not be confused with other types of localized attack, such as pitting, galvanic attack, intergranular corrosion, impingement, or cavitation.

Causes of stress corrosion cracking—There are many variables affecting the instigation of stress corrosion cracking. Among these variables are alloy composition, tensile stress (internal or applied), corrosive environment, temperature, and time.

Metallurgical factors: Pure metals, it is generally believed, do not crack as a result of stress corrosion. Alloys prepared from pure metals may crack, however. In recent studies, a significant improvement has been demonstrated in the cracking resistance of some alloys made from extremely pure metals. Some alloys in a particular base-metal system are more resistant to cracking than others. Such metals include aluminum, copper, and magnesium-base alloys. In these cases, cracking resistance improves as the alloy content is reduced and the composition approaches that of a pure metal [7].

Effect of stress: Since no cracking has been observed with metal surfaces in compression, only tensile stresses at the surface of the metal cause the cracking. These stresses may be due to strains within the materials (residual), or they may arise from an external load (applied). Usual causes of internal or residual stresses include

- deformation of the metal near welds, rivets, bolts, or in press or shrink fits
- an unequal cooling of a section or structure from a relatively high temperature

- the crystal structure within the metal going through a phase change or rearrangement which involves volume changes [7]

Applied stresses are the result of operating conditions and include

- differential thermal expansion
- dead loading
- pressure differentials [7]

Environmental effects: The corrosivity of a chemical medium cannot be used an an indication of its capability of promoting stress corrosion cracking. Many rather corrosive solutions do not cause cracking. In fact, the environments which are most conducive to stress corrosion cracking are those which produce highly localized attack without a significant general surface corrosion. Examples of environments known to cause stress corrosion cracking in some of the more common metals and alloys are shown in Table 1.

Figure 1, which shows four examples of corrosion control problems in marine structures, covers stress corrosion cracking by sea water of high-strength steels and a high-strength aluminum alloy. Figure 2 also shows susceptibility to cracking in sea water, fresh water, or humid atmospheres of high-strength steels, aluminum, and titanium alloys.

Effects of temperature: Temperature also has the effect of increasing stress corrosion cracking sensitivity. As a general rule, the susceptibility of a material to cracking increases with an increase in the temperature. An exception, however, is the cracking of low-alloy and carbon steels in hydrogen sulfide. Under such conditions, an inverse relationship exists and the cracking may be attributed to a secondary effect of corrosion; embrittlement is due to the hydrogen released during corrosion. The embrittling effect of hydrogen decreases as the temperature increases, and the tendency to crack therefore also decreases [7].

Table 1 Materials and environment which cause stress corrosion cracking [1,7]

Material	Environment
Aluminum	Air, sea water, sodium chloride solutions
Copper-base alloys	Ammonia, steam
Steel	Alkalies, nitrates, hydrogen cyanide, hydrogen sulphate, anhydrous liquid ammonia, sodium chloride solutions, marine atmosphere
Stainless steels	Caustic, chloride solutions
PH stainless steels	Chloride solutions, marine atmosphere
Magnesium-base alloys	Chloride-chromate mixture, moisture
Nickel (commercial purity)	Aqueous or fused caustic at elevated temperature
Monel, Iconel	HF vapors
Titanium alloys	Red fuming nitric acid, HCl, dry molten chloride salts, salt water

Susceptibility of materials to stress corrosion cracking—The factors controlling stress corrosion cracking were described qualitatively in the foregoing. The engineer's role centers around the task of avoiding stress corrosion cracking in marine structures. To solve this problem, some factor which expresses the susceptibility of materials to stress corrosion cracking must be introduced. The recent development of fracture mechanics enables one to do this. The concept of the threshold level of stress-intensity factor K_{Iscc} was introduced in the same way as that of fracture toughness (the latter to be discussed in detail in Section 2.3). There are three phases of failure by stress corrosion cracking in structural components or in laboratory test specimens: First, formation of a small pit by corrosion processes; second, formation and rapid propagation of a crack; and third, final failure by fracture processes [8]. K_{Iscc} is defined as the threshold value of the stress-intensity factor below which a stable crack growth occurs. Present design criteria against stress corrosion cracking do not permit the combination of a crack size and applied stress which exceeds K_{Iscc}. Thus, a material must undergo the test determining K_{Iscc}.

For several years, data have been reported for a variety of structural materials defining K_{Iscc}. In fracture toughness tests, structural designers have been given a potential for implementing "safe-life" or other design concepts based on crack growth laws. These design concepts will be discussed in Section 2.3. K_{Iscc} values for several high-strength materials which are considered for use in future deep-submersible hulls, as well as some relevant comments, are given below.

There are two testing methods for defining K_{Iscc} values. One is the initiation method as shown in Fig. 6 [8]. This method employs a cantilevered, dead-weight loading system by which crack growth results in increasing applied K_I; for this reason, a system of bracketing data derived from specimens with and without crack growth is necessary to define the threshold K_{Iscc} value. The second type, the arrest method, utilizes a bolt-loaded (constant deflection), modified wedge-opening-loaded specimen that has a decreasing K_I field with increasing crack length [9]. By this method, the specimen is loaded to an applied K_I well above the expected K_{Iscc}, and the crack is allowed to propagate until it arrests at K_{Iscc}.

To evaluate the relative susceptibility of materials to stress corrosion cracking, the ratio analysis diagram (RAD) [10] is used (details will be discussed in Section 2.3). Since the parameter K_{Iscc} describes the applied K_I level for the beginning of crack growth, equations such as the surface-flow equation and the through-crack equation apply only for the stress corrosion cracking initiation; nothing is implied regarding crack growth as a function of time. The

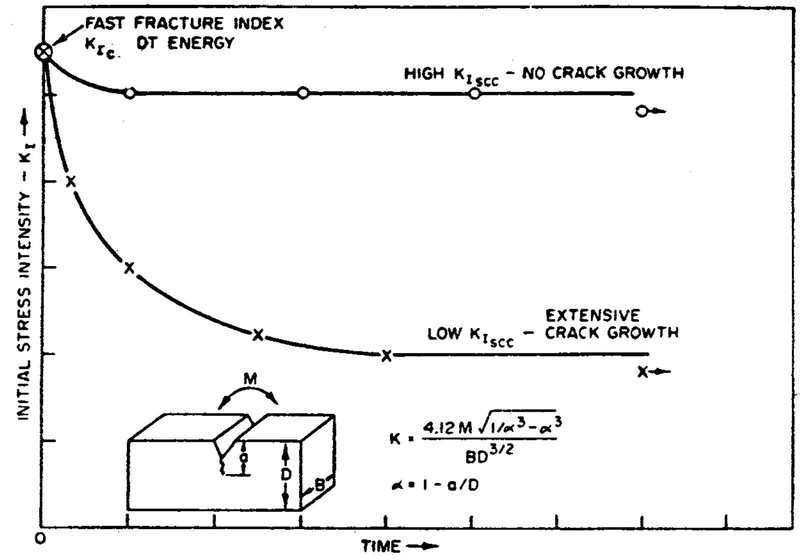

Fig. 6 Graph of K_{Iscc} determined by cantilever test method. Purpose of test is to determine threshold level for crack propagation [8]

available analysis tools are the stress intensity for stress corrosion cracking initiation and the final crack size for failure as given by the fracture-resistance parameter K_{Ic}. Without repeating the detailed stress-intensity analyses of references such as [11] and [12], RADs for steel, S-PH stainless steels, and titanium alloys are shown in Figs. 7 to 9.

In Fig. 7, the ratio of K_{Iscc} to yield strength is overlaid on the RAD. This diagram illustrates the comparison of stress corrosion cracking sensitivity with fracture toughness. Such comparison is necessary for an interpretation of the severity of stress corrosion crack growth on structural integrity. If the final failure is by fracture, the life of the structure is determined by the size of the defect or crack that can be tolerated for a given loading system, which in turn is defined by the fracture-resistance property. Structures designed with materials of high fracture resistance, K_{Ic}, have the capability of containing large flaws. The problem of stress corrosion cracking, then, is related to inspectability and maintenance. However, those structures designed with materials of

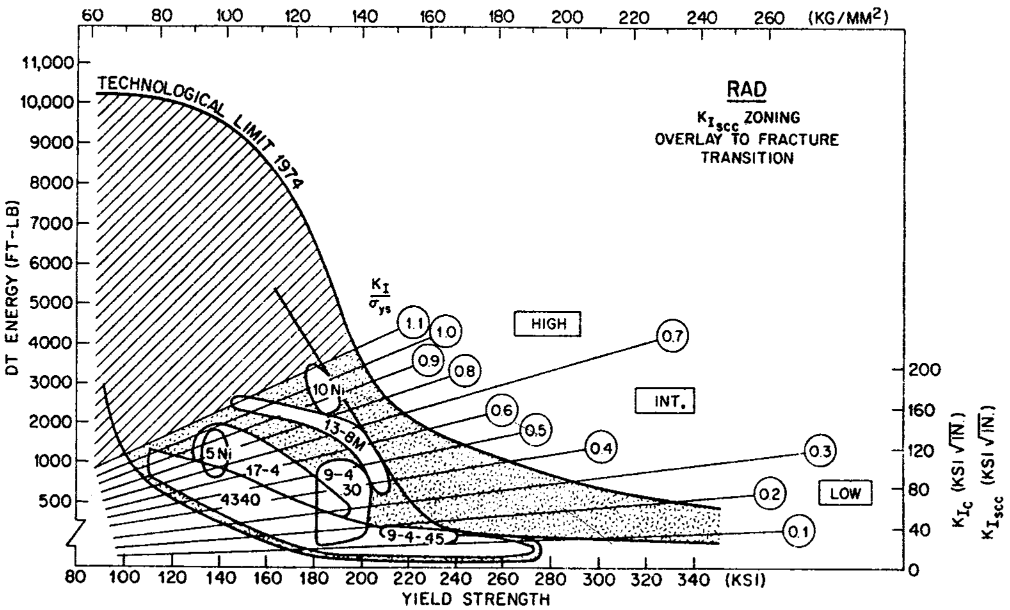

Fig. 7 Ratio analysis for SCC of various types of steel [8]

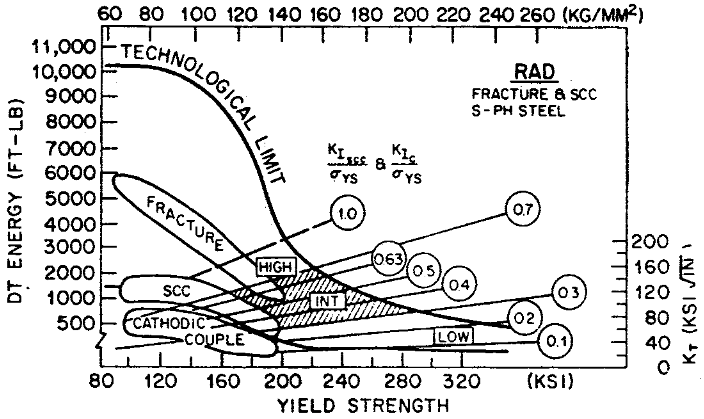

Fig. 8 Ratio analysis showing fracture and SCC properties for S-PH, a high-strength stainless steel [8]

low K_{Ic} may suffer sudden crack extension after a short period of crack growth. As indicated in Fig. 7, steels which belong to the former group include 10Ni steel (HY-180 level steel), 13-8M steel, and 5Ni steel (HY-130 level steel). The latter group consists of HP 9-4 steels and AISI 4340 steel; 7-4 HP steel may be on the borderline between both groups. As a whole, these materials cannot attain the higher fracture resistance levels indicated by the technological limit (TL) line for the case of stress corrosion cracking.

Figure 8 shows RAD for high-strength, precipitation-hardening stainless steel [8]. In this figure, several factors are presented simultaneously. Fracture toughness is in the high-resistance range for lower-strength levels and in the intermediate resistance range for the higher-strength levels. Similar effects are shown in the salt-water stress corrosion cracking properties of this steel by the data zone coded "SCC" and "cathodic-couple." A comparison of the fracture toughness and stress corrosion cracking sensitivity reveals that the latter value is considerably lower than the former. For this high-strength stainless steel to be used for applications in a sea-water environment, then, a very careful analysis of potential crack growth must be performed. The most important single aspect is determining a maximum for the yield-strength value which will absolutely preclude brittle fracture. The stress corrosion cracking properties dictate that a maximum allowable yield strength less than that for fracture would be necessary to absolutely prevent stress corrosion cracking under linear-elastic conditions. It is noted that crack growth in test specimens was present for all strength levels regardless of the stress state—linear-elastic or plastic loading. It is also apparent that cathodic protection systems which depend on sacrifice zinc or aluminum bars to prevent general corrosion aggravate the stress corrosion problem by lowering the K_{Iscc} value.

Figure 9 shows a stress corrosion cracking RAD for titanium alloys [8]. Two of the alloy systems—721 (Ti-7Al-2Cb-1Ta) and 811 (Ti-8Al-1Mo-1V)—are well known for their sensitivity to crack growth in salt water. The other two alloys—6-4 (Ti-6Al-4V) and 6-6-2.5 (Ti-6Al-6V-2.5Sn)—are not sensitive to salt water in all heat-treated conditions; accordingly, the data zone may well be representative of either the fracture-resistance properties or the sustained load cracking properties of the alloys.

Avoiding stress corrosion cracking [1,7]—Earlier in this section, we discussed the many variables contributing to stress corrosion cracking. Since many factors are involved, the reduction or elimination of one contributing factor may help the overall situation. In this section, we will touch on some of the methods for reducing the contributing factors.

Reduce tensile stresses: The methods of lowering tensile stress lead K_I to be lower than K_{Iscc}. Stresses due to operating conditions often can be reduced by changing the design. Such changes may include lowering the operating pressures, avoiding misalignment of bolted or welded connectors, avoiding differential thermal expansion, and eliminating heavy loads on thin sections. Another possible method is increasing the metal thickness in loaded thin sections or in pressure vessels, when other design considerations will allow such actions. This will lower the applied stress by distributing it over a larger cross-sectional area. Residual stresses, however, cannot be reduced by increasing the metal thickness. These stresses may be eliminated by stress-relief anneal after fabrication and installation. A final method for reducing residual tensile stress is to put the surface layers of the metal in compression.

Alter the corrosive environment: The method for altering the corrosive environment is perhaps the most obvious method of limiting stress corrosion cracking. For example, the elimination of chloride ions by ion exchange from aqueous solutions allows the use of stainless steel in high-temperature waters.

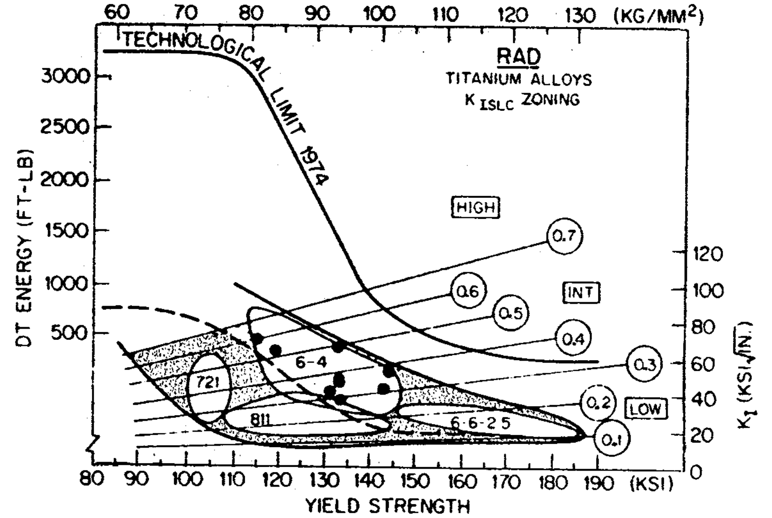

Fig. 9 RAD comparison of SLC properties of Ti-6Al-4V alloys with the data zone for salt water SCC [8]

However, the addition of inhibitors to solutions to reduce stress corrosion cracking has not been promising in general. Design considerations can again aid in reducing cracking. For instance, placing internal heaters away from the side or bottom of vessels and near the bulk of the solution minimizes the tendency for concentration in relatively stagnant areas. A concentrated solution introduced into a dilute solution near the center of the vessel rather than near the edge or bottom permits the solution to become diluted before it reaches the vessel walls.

Lower the surrounding temperature: If the operating temperature of the bulk environment is fixed and cannot be lowered without loss in efficiency, the method for lowering the surrounding temperature is impossible. However, the raised local temperature associated with heat-transfer surfaces or the introduction of hot solutions into cooler ones can be avoided many times by the vehicle design.

Use of a cathodic protection system: Cathodic protection can be applied to the relief of stress corrosion cracking as well as to other types of corrosion. It does, however, aggravate the stress corrosion cracking problems in some materials, such as high-strength precipitation-hardening stainless steel, as shown in Fig. 8.

2.2 Manufacturing

Basically, manufacturing processes for submersible hull construction consist of three stages—forming, welding, and machining (cutting or milling), though details depend largely upon the shape and structure planned by the designers. Fabricators of submersible hulls obtain the hull material in the form of rolled or forged plates. These plates are cut and formed based on the hull design. The formed pieces are then assembled and welded to a structure. Finally, the hull is machined, if necessary, in order to keep circularity or sphericity according to design requirements.

In this section, the three manufacturing stages described above are discussed. Emphasis will be put on welding since it is most responsible both for the successful construction of the hull and for the successful achievement of missions imposed on the deep submersible. Section 2.2.1 covers forming, 2.2.2 welding. Machining, however, is not discussed, because it does not affect the capability of submersible hulls to any great extent.

2.2.1 Forming

Submersible-hull fabricators receive hull material in flat plate form. Plates are cut and formed in accordance with the designed shape. The shapes of submersible hulls are classified into two types: cylindrical and spherical. The former type, similar in shape to a submarine, may be appropriate for shallow depths or larger submersibles. The latter is adapted to most submersibles that have been launched up to the present time. Because a spherical hull has a higher buckling strength than a cylindrical one, it is suitable for deeper depths. Proposed and current shapes, along with structures of submersible hulls, are presented in Chapter I. Figures 10 and 11 are schematic drawings of these two hull shapes [13,14].

In order for a submarine-type hull to be built, two kinds of forming methods are essential: bending and spinning. A mid-ship cylindrical part is formed to a ring by bending a flat plate. It is then seam-welded and stiffened with ring-shaped stiffeners. The coni-

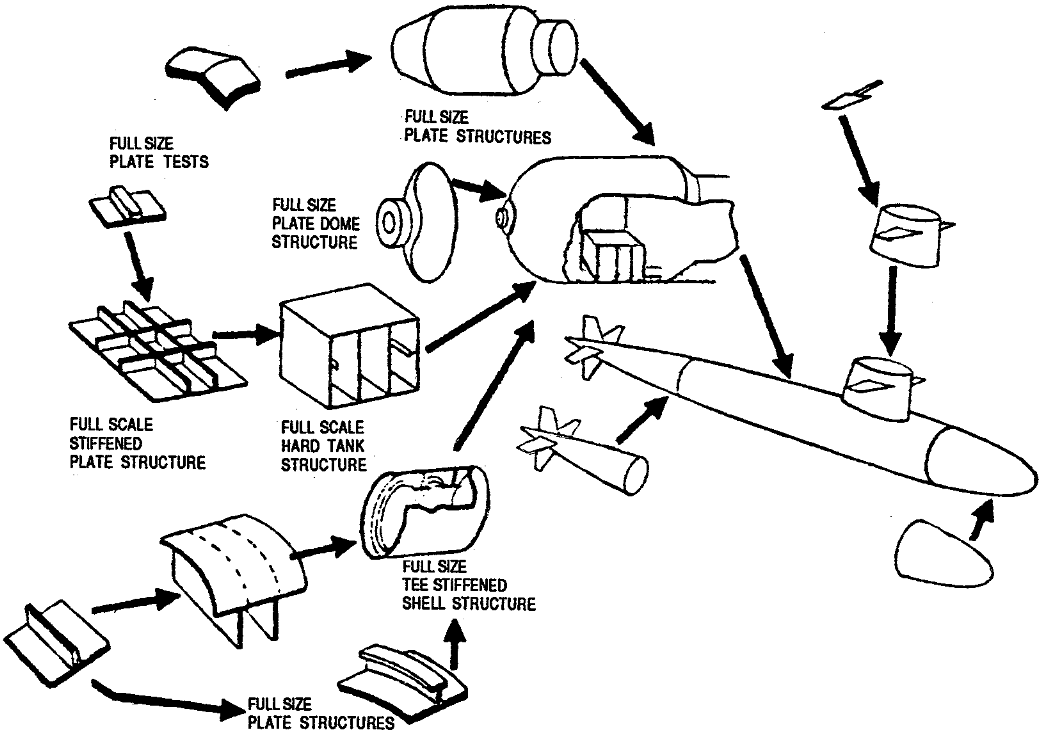

Fig. 10 Structure development and certification process for advanced submarines [13]

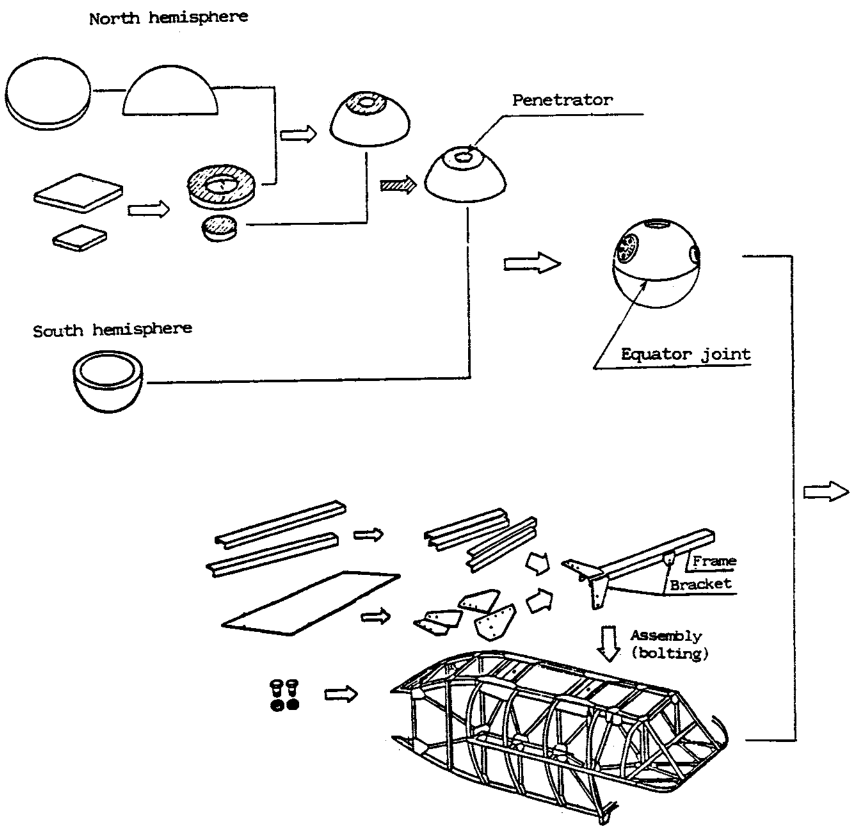

Fig. 11 Structure development for deep-diving submersible [14]

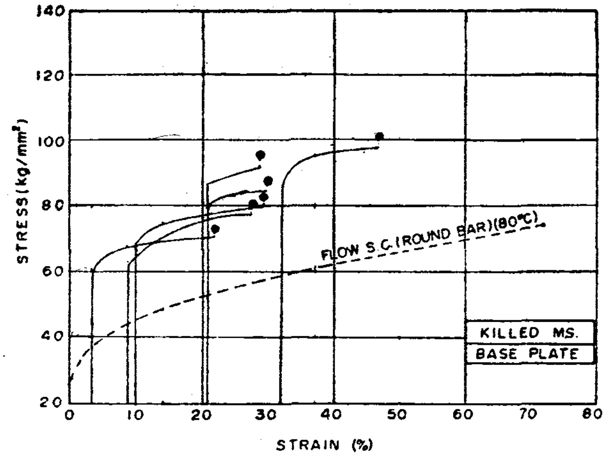

Fig. 12 Fracture stress curves of base plate [15]

cally shaped aft part is formed in a similar manner. A round head of the fore part is fabricated either by welding bent pieces or by spinning.

The method of bending a flat plate to a ring, called roll bending, can be performed by any of several types of roll bending machines [14]. A typical one consists of three rolls in a pyramidal arrangement, the upper roll adjustable to control the degree of curvature. In another type, one roll is directly above a second at the front. These two rolls grip and move the plate against an adjustable third roll at the rear, which deflects the plate and controls the curvature. Bending rolls are available in a wide range of sizes, some capable of bending plates up to 6 in. thick.

Since bent plates are heavily worked plastically, plasticity effects can change the mechanical properties of the plate. Although a number of papers have been published on this subject, there are unfortunately little data on high- and ultrahigh-strength materials for deep submersible use. Therefore, data on mild steel alone are presented in Figs. 12 and 13. Figure 12 shows the effect of prestrain on the fracture stress [15]. Test specimens with various levels of prestrain were pulled, and stress-strain curves were recorded. One can see that the fracture stress increases while the ductility decreases as the amount of prestrain increases. Figure 13 presents the effect of prestrain on the Charpy V-notch test data [16]. It is shown in the figure that the prestrain has a bad effect on absorbed energy and brittle-ductile transition temperature. Tensile prestrain degrades the notch-toughness of a material more than compressive prestrain does. Although these data were obtained from a mild steel (yield strength is about 40 ksi), much the same results would be obtained from higher strength materials. Thus, one must pay attention to the effect of bending on the properties of hull materials.

The effect of spinning on the material properties will be almost the same as that of bending because spinning is also the process of forming with plasticity. The process of spinning will be described later with hot-spinning.

In the case of the construction of a spherical submersible hull, again two methods of forming are considered: bending and spinning. Bending in this case, however, is different from that mentioned above. Bending of the plate is used when the fabrication system in which bent pieces with double-curvature are welded is employed. It is very difficult to make a plate with double-curvature, since roll bending, which is suitable for single-curvature bending, is no longer feasible. The multi-point press process may be the only way to form a double-curved plate. In this process, a plate simply supported on a die is pressed at several points in such a way that the deformed plate gives a planned shape. The pressing at each point must be programmed based on the plasticity analysis. The difficulty involved in this process is the calculation of the amount of plastic deformation and elastic spring-back.

Spinning is a metal forming process in which parts with rotational symmetry are produced from flat blanks or preforms with the use of a mandrel and

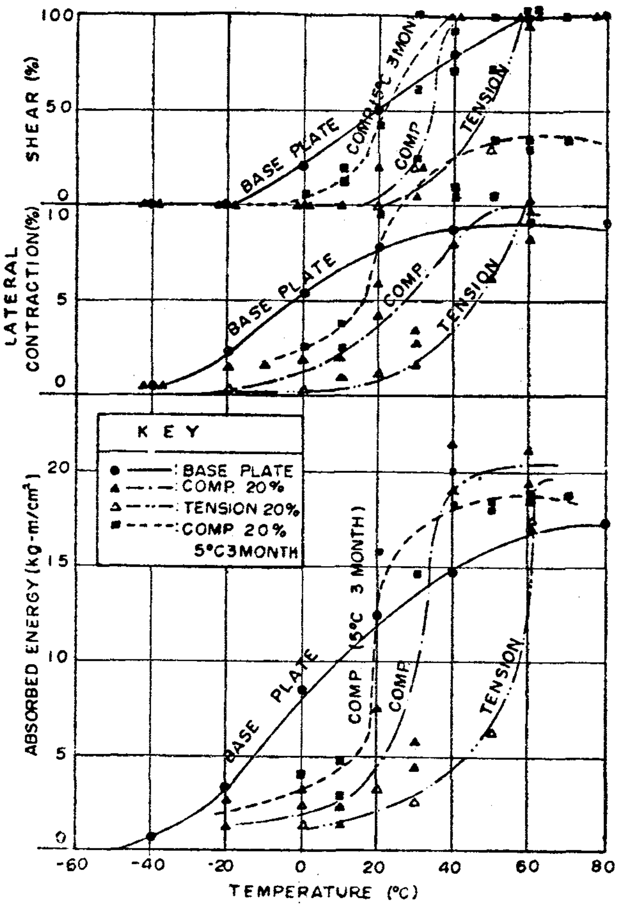

Fig. 13 Transition curves of prestrained steel (Charpy V-notch test) [16]

tools or rollers. There are two basic types of forming using this method: manual or conventional spinning, as shown in Fig. 14*a*, and power spinning, as shown in Fig. 14*b* [17]. The spinning process may also be classified into cold spinning and hot spinning, depending on the temperature of the material being formed. As can be understood from Fig. 14, spinning involves large amounts of plasticity. Thus, while cold-spinning requires a material to be thin or very ductile or both, hot-spinning is recommended for very thick materials with less ductility. Since the hull materials for deep submersibles are expected to be very thick, hot-spinning must be used. With the hot-spinning process, by which a plate is heated for easy working, a thick plate is formed into a semi-sphere. Two semi-spheres are then formed into a sphere by welding the circular seams between them.

These plastic workings greatly affect the properties of materials. The effect of multi-point press may be basically the same as that of roll bending, though because a plate is pressed locally, the strain distribution is not uniform. We do not have much data regarding the effect of hot-spinning on the submersible hull materials, and so mild steel data are used as an example.

Figure 15 shows the effect of prestrain at an elevated temperature on the fracture stress [18]. In this figure, the prestraining history is represented by a broken line. The upper portion of the figure shows the test results where prestrain was given at room temperature (R.T.) and fractured at the same temperature. Total strain and fracture stress, of course, remained constant without regard to the amount of prestrain because each specimen experienced the same stress-strain history as it would have if a specimen without prestrain were pulled until it fractured. The middle section of the figure illustrates that when specimens were prestrained at 300°C, prestrain increased the fracture stress slightly, but decreased the total strain. Finally, the bottom portion of the figure shows the prestrain at 600°C decreased both the total strain and the fracture stress. It must be noted that the residual strain (the difference between fracture strain and prestrain) is greatly decreased as the amount of prestrain increases. Thus, hot-spinned materials will lose ductility. The loss of ductility, in general, causes low fracture toughness and low fatigue strength. Therefore, a designer must keep in mind the fact that the probability of hull failure can increase by hot prestraining.

2.2.2 *Welding*

Welding is the most important part of manufacturing in relation to material properties, for its heating and cooling sequence can cause many problems—residual stress, welding distortion, degradation of notch toughness, and material discontinuity, among others. In the first part of this section, various welding processes feasible for high-strength materials will be described, followed by a discussion of the problems caused by welding.

Welding processes—More than 40 welding processes are used in present-day metal fabrication. Figure 16 is a master chart of welding processes prepared by the American Welding Society (AWS) [1,19]. (There are several new welding processes which are not shown in Fig. 16, including electroslag welding, ultrasonic welding, cold welding, friction welding, electron beam welding, plasma-arc welding, and laser welding.) Among various welding processes, those feasible for high-strength materials and large structures include arc welding processes, electron beam welding processes, and laser welding processes. The arc welding processes, commonly used in today's ocean engineering industry, contain metal-electrode arc welding processes: shielded metal-arc welding, submerged-arc welding, gas tungsten-arc welding (GTA), and gas-metal arc welding (GMA).

Shielded metal-arc welding: Used exclusively with steels, this is an arc welding process wherein

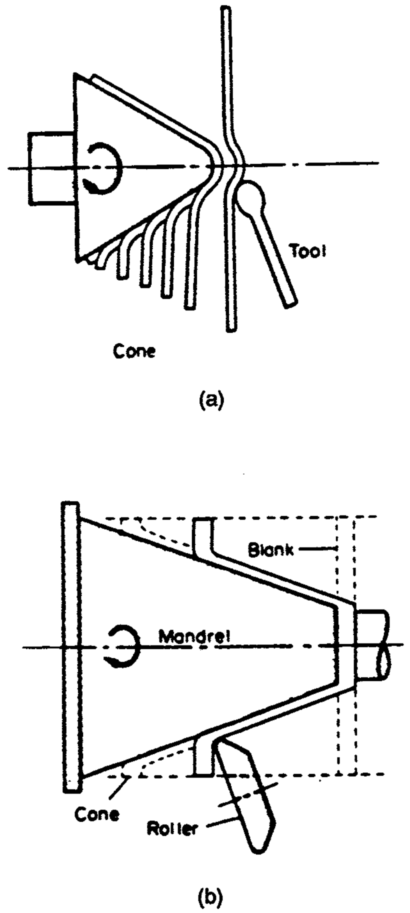

Fig. 14 Spinning processes [17]: (*a*) conventional spinning of sheet metal; (*b*) power spinning of conical part

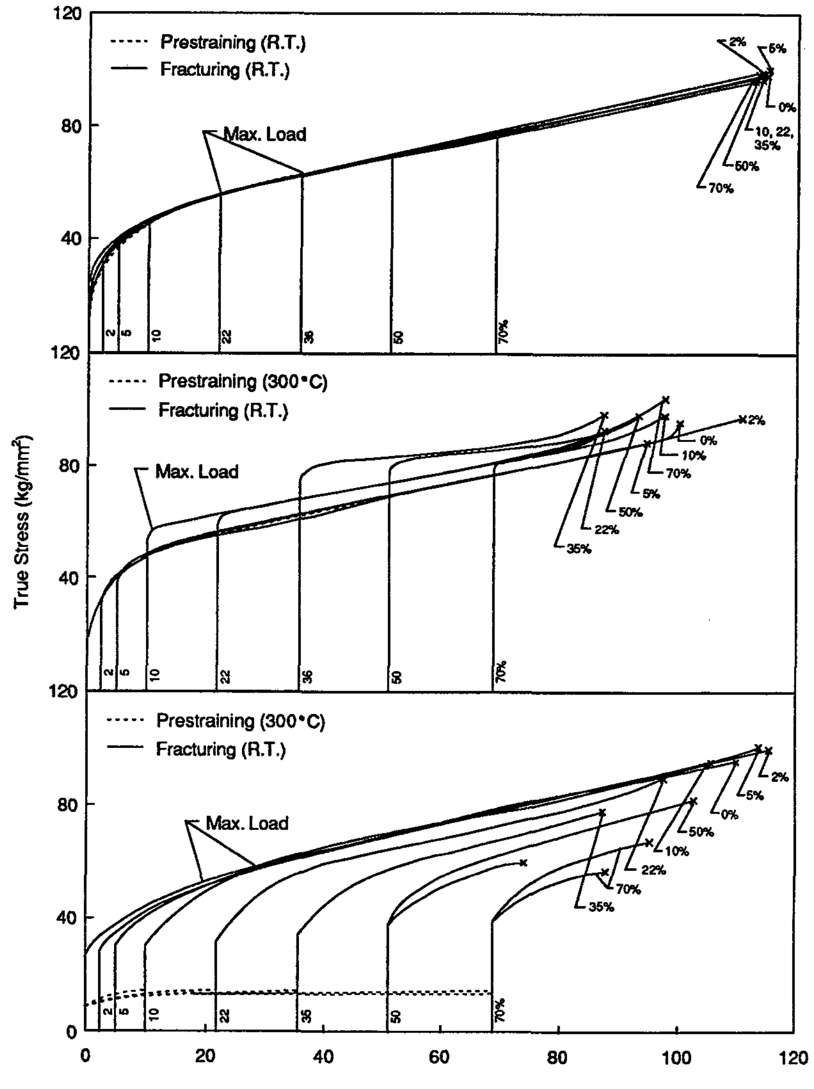

Fig. 15 Effect of prestrain on flow stress curve (tensile prestraining) [18]

coalescence is produced by heating with an electric arc between a covered electrode and a workpiece. Shielding to prevent contamination from the air is obtained from the decomposition of the electrode covering. Filler metal is obtained from the electrode. In practice, the process is limited primarily to manual covered electrodes in which the welding operator manipulates the electrodes. Some attempts have been made to mechanize this process, but they have not been successful.

Figure 17 schematically represents the shielded metal-arc process. Equipment to operate the electrode usually consists of an electric power supply specifically designed for the process, insulated electrode holders of adequate electric and thermal capacity, cable, and grounding clamps. The process may use either alternating current or direct current with the electrode either positive or negative. Currents between 15 and 500 A with arc voltages between 14 and 40 V, depending upon the covering characteristics, are normal.

Coating electrodes for manual arc welding have been classified by AWS and the American Society for Testing and Materials (ASTM) [19]. The electrodes are grouped according to their operating characteristics, the type of coating, and the strength level of the weld metal. AWS-ASTM electrodes are known by code names, such as E-70XX and E-100XX. For example, E-100XX electrodes produce deposited metal having a minimum specified ultimate tensile strength of 100 000 psi. The last two digits refer to the type of electrode coating and operating characteristics.

Electrodes as large as 5/16-in. diameter may be employed, depending upon the plate thickness and

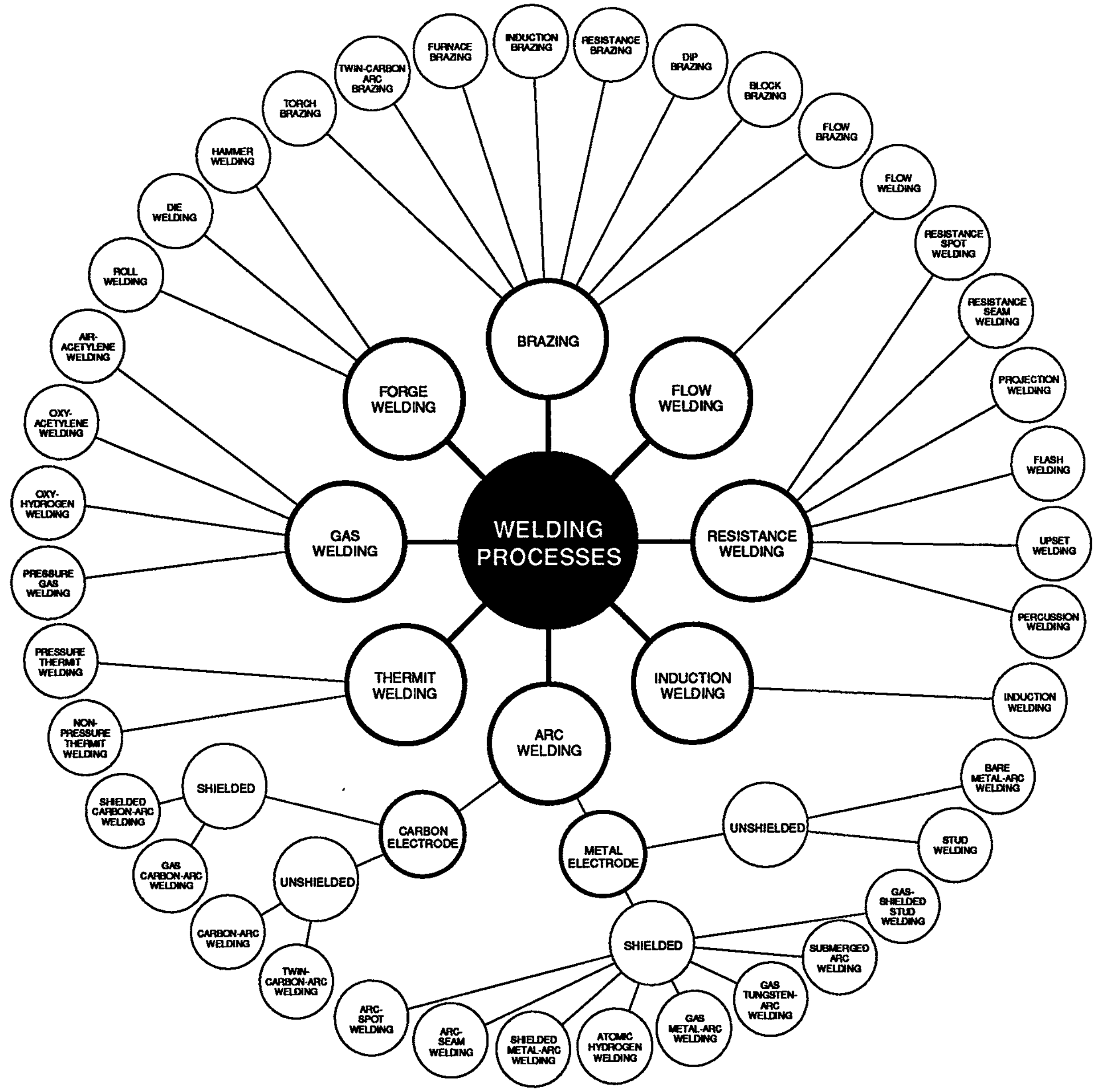

Fig. 16 Master chart of welding processes [1,19]

type of joint. Welds are built up in relatively thin layers, approximately ten layers per inch of thickness. This will permit a partial progressive grain refinement of proceeding layers, resulting in an improvement in ductility and impact resistance of the weld metal.

Submerged-arc-welding: This is an arc welding process wherein coalescence is produced by heating with an electric arc or arcs between a bare metal electrode or electrodes and the workpiece. The welding is shielded by a blanket of granular, fusible material on the workpiece. Pressure is not used, and filler metal is obtained from the electrode and sometimes from a supplementary welding rod. Figure 18 shows how a submerged-arc groove weld is made.

The fusible shielding material is known as "flux," "welding composition," or "melt." This is a finely crushed mineral composition and will be referred to as "flux" in this book. Flux is the basic feature of submerged-arc welding and makes possible the special operating conditions which distinguish the process. Flux, when cold, is a nonconductor of electricity, but in the molten state it becomes highly conductive.

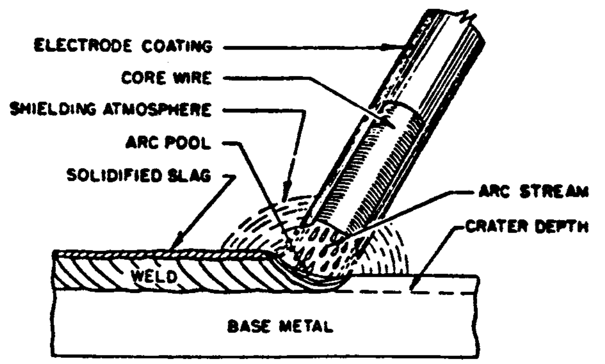

Fig. 17 Schematic representation of shielded metal-arc process [1]

In submerged-arc welding, there is no visible evidence of the passage of current between the welding electrode and the workpiece. The electrode does not actually contact the workpiece; instead, the current is carried across the gap through the flux.

The flux completely covers both the tip of the welding electrode and the weld pool. Thus, the actual welding operation takes places beneath the flux without sparks, spatter, smoke, or flash. No protective shields, helmets, smoke collectors, or ventilating systems are needed, with the exception of goggles, which may be worn as routine protection for the eyes.

In its molten state, the flux provides exceptionally suitable conditions for unusually high current intensities. Thus, great quantities of heat may be generated. The insulating qualities of the flux enable the intense heat to be concentrated in a relatively small welding zone, where the welding electrode and base metal are rapidly fused. High welding speeds are possible under these conditions, and deep penetration can be obtained by this concentrated heat. Consequently, a relatively small welding groove can be used, permitting the use of smaller amounts of filler metal.

The wide use of submerged-arc welding stems from its ability to produce satisfactory welds at high rates of deposition. Currents used in submerged-arc welding are much higher than those employed in manual shielded metal-arc welding. The maximum electrode usually employed is $5/16$-in. diameter and the maximum current (for single electrode) is approximately 2000 A.

It should be pointed out that high amperage results in a very coarse columnar structure in the weld metal with an enlarged heat-affected zone (HAZ) in the base metal as compared to multipass welding wherein the energy input is relatively small. This is important in connection with vessels which will operate at low temperature. The coarse columnar structure of the high-amperage weld usually results in lower impact resistance as compared to multipass welds. Therefore, particular consideration should be given to the welding procedure in terms of the service requirements involved.

During the last ten years, many studies have been done on how to improve notch toughness of two-pass submerged-arc deposited metals in heavy mild steel ship plates and how to improve notch toughness of submerged-arc deposited metals in high-strength, notch-tough steels. Both topics will be discussed later in Section 2.2.2.

Gas-shielded-arc welding: In the gas-shielded-arc welding process, coalescence is produced by fusion from an electric arc maintained between the end of a metal electrode, either consumable or nonconsumable, and the part to be welded with a shield or protective gas surrounding the arc and weld region. The shielding gas may or may not be inert, pressure may or may not be used, and filler metal may or may not be added.

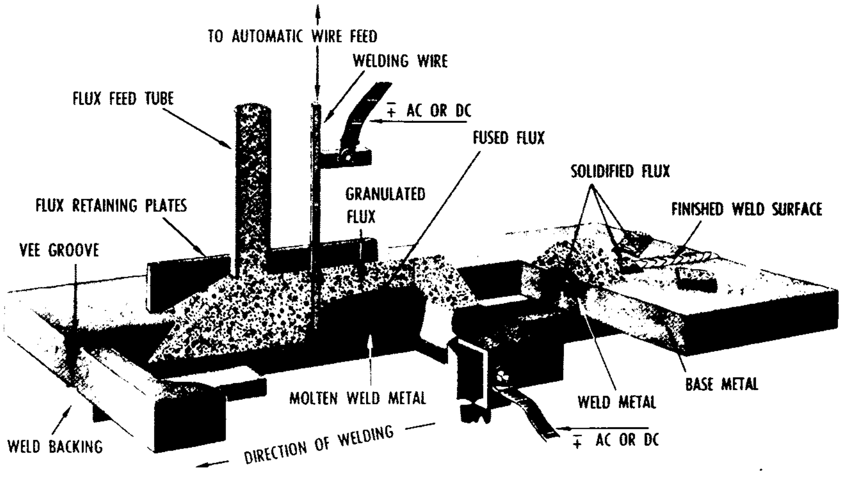

Fig. 18 Processes of a submerged-arc groove weld

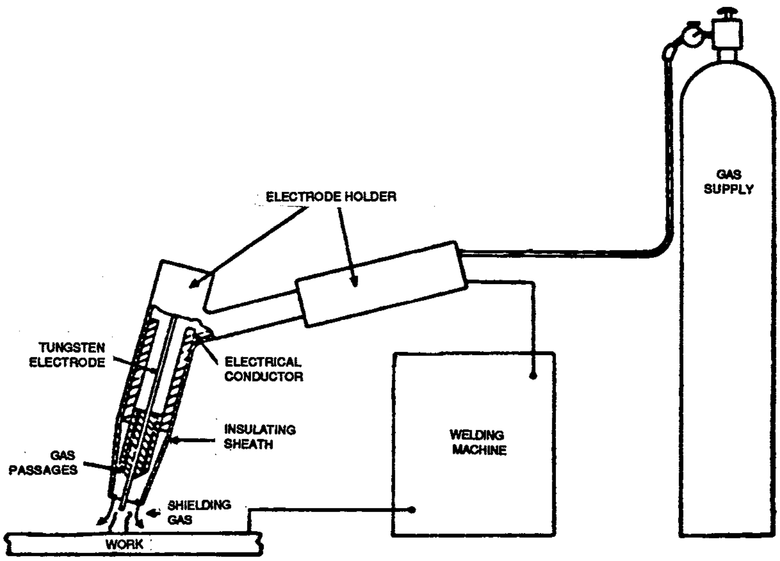

Fig. 19 Schematic diagram of gas tungsten-arc welding

At this time, there are two different types of gas-shielded-arc welding. The first type, gas tungsten-arc (GTA) welding, employs a tungsten electrode or an electrode of some other refractory, high-melting point material, such as graphite, which will not melt or be vaporized too rapidly in the intense heat of the arc. Gas metal-arc (GMA) welding, the second type, employs a continuously fed electrode which melts in the intense arc heat and is deposited as weld metal.

Figure 19 shows basic features of GTA welding. The shielding gas is fed through the electrode holder, generally referred to in this process as a torch. Argon, because it is an inert gas, is commonly used for the shielding gas; the use of helium as the shielding gas is rather rare. A small percentage of oxygen is often added to the inert shielding gas. Carbon dioxide and a mixture of oxygen and carbon dioxide have been found to be effective as the shielding gas with welding carbon and low-alloy steels. More recently, fluxes have been added as a core within a tubular sheath or as a granular magnetic material which adheres to the filler wire surface.

The GTA welding produces a weld of high quality in terms of notch toughness. This is considered to be the most advanced metal-arc welding process. Thus, it may possibly be used for welding high-strength submersible hulls. The GTA process, however, has a disadvantage, for a very low deposition rate is not very attractive to fabricators.[1]

Figure 20 shows basic features of gas metal-arc welding. The filler wire, which is manufactured in a coil form, is fed mechanically into the welding arc. The arc travel is controlled manually in the semi-automatic process and mechanically in the automatic process.

A bare wire is commonly used for the electrode, but flux-covered wires are also used.

Various gases are used for shielding. These include

- pure inert gases such as argon and helium,
- mixtures of argon, carbon dioxide, oxygen, and other gases, and
- carbon dioxide.

Because of the high cost of inert gas, carbon dioxide and a mixture of oxygen and carbon dioxide are widely used for welding carbon steel and low-alloy high-strength steels. Argon and a mixture of argon and carbon dioxide are often used for welding quenched-and-tempered steels and ultrahigh-strength steels. Inert gases, which may contain small portions of other gases, are used almost exclusively for welding stainless steel, aluminum, titanium, and other nonferrous metals.

Compared with GTA welding, GMA welding is characterized by a high deposition rate (although much less than that of the submerged-arc welding). Therefore, the GMA process is chiefly used for welding heavy plates, while the GTA process is mainly used for welding thin sheets.

A group of GMA processes generally called the narrow gap process has been developed during the last 20 years [21]. The narrow gap process differs from conventional welding procedures in the type of joint design used. The new procedure uses a square-butt joint with a narrow root opening (approximately ¼ in.). The use of this type of joint results in a weld deposit with a small fusion zone. A typical weld deposited by this process in a 2-in.-thick HY-80 steel is shown in Fig. 21.

[1]The deposition rate of the GTA process can be increased by the use of the hot-wire technique.

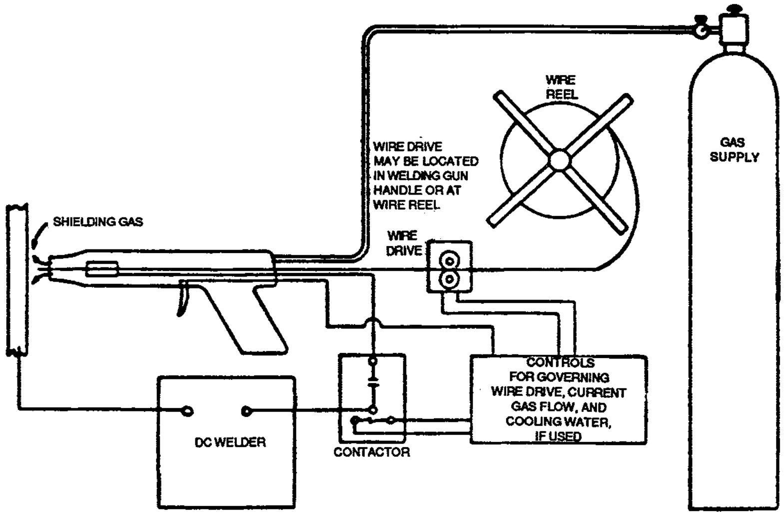

Fig. 20 Schematic diagram of gas metal-arc welding

The very high depth-to-width ratio and the very narrow, uniform heat-affected zone are apparent. A mixture of argon and carbon dioxide is used as the shielding gas. Welds are deposited from one side of the plate using a specifically designed guide tube that extends into the joint. Welds can be made in all positions—the weld shown in Fig. 21 was made in the vertical position.

Electron beam welding: Heat for coalescence in electron beam welding (EB) is obtained from the impingement of a beam of high-velocity electrons on the workpiece to be welded. The EB welding process was first developed to obtain ultra-high purity welds in reactive and refractory metals used in the atomic energy and rocketry fields. Eventually, however, it may be applied to other structures as well, due to the increasing use of higher-strength advanced materials in various structures. (Conventional metal-arc welding processes cannot produce notch-tough welds comparable to the base plate.)

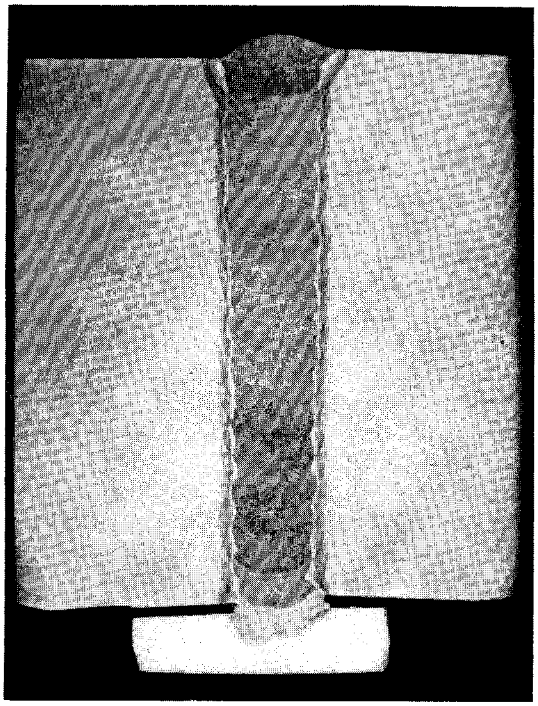

Fig. 21 A weld deposited by the narrow gap process

Figure 22 [22] shows the electron optical system employed. A high-voltage current passes through the filament, heating it to about 4000°F, and causes high-velocity electrons to be emitted. These electrons are concentrated and focused on the workpiece by means of the control electrode, accelerating anode, and focusing coils. Welding must be done in a vacuum chamber so that a concentrated electron beam cannot be scattered by atmospheric particles.

EB welding machines with a vacuum chamber are classified into two types: hard-vacuum environment and soft-vacuum environment [22]. In hard-vacuum environment EB welding equipment, the pressure in the chamber is kept as low as 0.3×10^{-3} torr. The pressure in the chamber of soft-vacuum EB machines is kept at less than 10^{-4} torr. Diffusion pumps are required to attain such a low pressure, along with mechanical pumps or Rootes pumps.

The necessity of using the vacuum chamber imposes a serious limitation on the size of the workpiece. This limitation is the principal reason for EB welding's limited use, despite its excellence. Much effort, however, has been devoted to the development of EB machines in which the workpiece can remain outside the vacuum chamber. Some nonvacuum type EB machines have been developed, but they are not yet practical enough. These machines enable the electron beam to emerge through a small orifice in

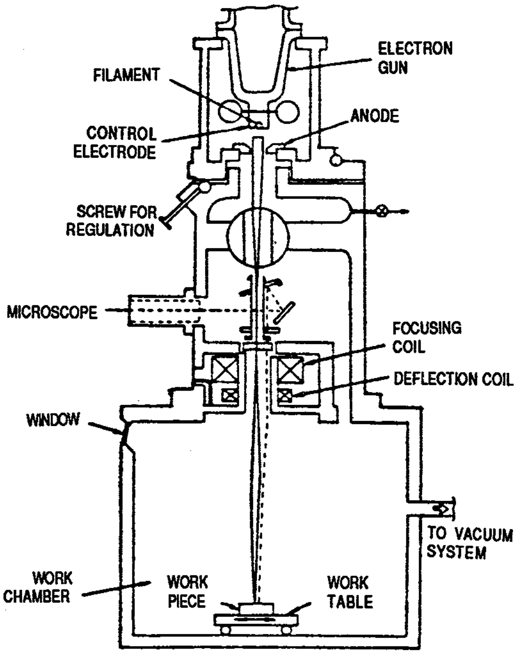

Fig. 22 Schematic representation of EB welding machine [23]

the vacuum chamber in order to strike the workpiece outside. High-capacity vacuum pumps control the leakage through the orifice.

Another attempt to avoid the limitation from a vacuum chamber involves the development of a mobile-chamber-type EB welding process [23]. As shown in Fig. 23, the principal mechanism of this process is the application of a small, movable chamber equipped with an electron gun, which provides an evacuated space between the moveable chamber and the workpiece. This machine enables one to apply the EB welding process to large structures; however, the difficulty of sealing still remains.

EB welding machines are classified into two types—high-voltage and low-voltage. High-voltage equipment employs 50 to 100 kV, while low-voltage machines use from 10 to 30 kV. High-voltage machines provide a well-focused narrow beam, which leads to deeper penetration. However, the X-ray emission by high voltages and greater electron velocities necessitates expensive protective shielding. This problem is not serious for low-voltage equipment, which yield sufficient spot concentration and penetration while soft X-rays can be absorbed by the walls of the evacuated chamber.

Although it has the disadvantage of size limitation of the workpiece, EB welding has great advantages over the conventional processes. One is its ability to handle welding materials which are difficult to weld by other processes, such as ultrahigh-strength materials. Another is its high efficiency: EB welding saves time. For example, today's high-power EB welding machine can weld 20-cm-thick steel plate or 30.5-cm-thick aluminum plate with one pass [24]. Figure 24 demonstrates an EB-welded joint of 150-mm-thick steel plate.

As the feasibility of conventional metal-arc welding processes decreases with increasing yield-strength and plate thickness of materials (which is the case in welding deep submersible hull materials), EB welding is expected to be used more extensively in this area.

Laser welding [1,20]: The application of laser beams to welding is rather new. Laser welding is accomplished by focusing the high-intensity light beam from the crystal (usually a ruby) at or near the location on the workpiece where local fusion is to be made. The objective is to fuse the point of contact between the two workpieces, thereby forming a weld. As with other welding processes, the amount of metal melted depends upon the intensity and total energy transferred by the laser beam. The power density of the laser beam is in the range between 10^5 and 10^{13} W/cm^2, which is about 10^3 times higher than that of metal-arc welding and as high as that of EB welding.

The high-power density enables one to make a weldment with lower total heat input, hence less welding distortion. Low-heat input leads to a small HAZ—hence, a sound weldment, free from the HAZ degradation problem.

Other advantages of laser beam weldings are as follows:

- Since the laser delivers its energy in the form of light, it can be operated in any transparent environment (air, vacuum, inert gas, or even certain liquids) and through transparent windows; also, it need not be in close proximity to the workpiece.
- There is no need for mechanical contact of any kind with the workpiece, nor any requirement that the material being worked on be a conductor of electricity.
- Since the laser beam is almost perfectly collimated and monochromatic, very simple optical systems can be used to bend, direct, and focus the laser beam with high precision.

Because of these great advantages, the laser beam welding process will be applied to a very wide range of industries in the future.

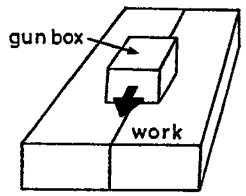

Fig. 23 Schematic drawing of movable chamber type EB welding machine [23]

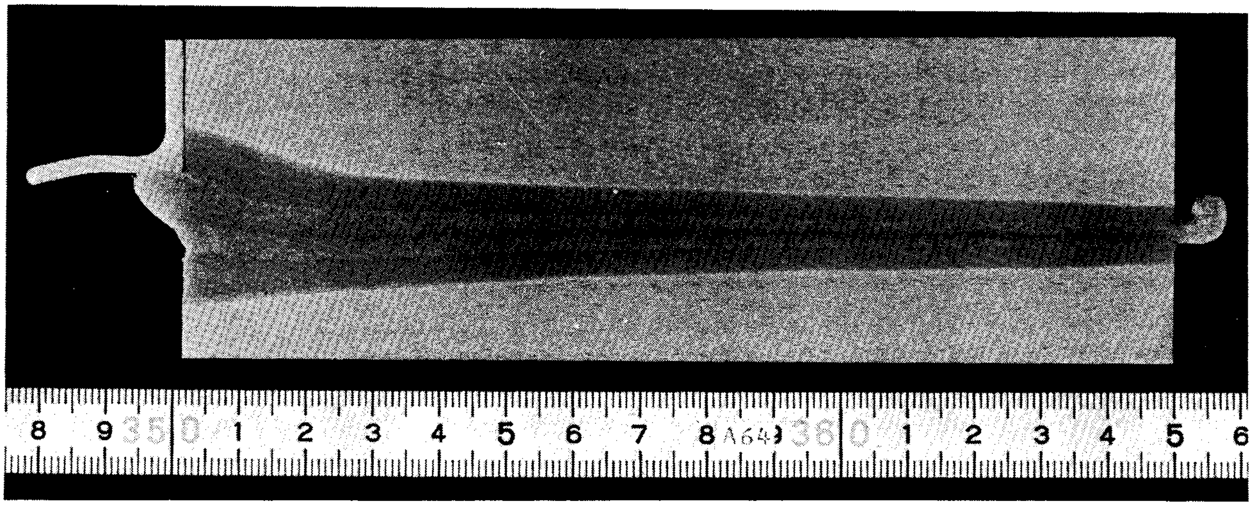

Fig. 24 EB welded 150-mm-thick (6-in.) steel plate

Despite its potentialities, however, the laser beam welding process has not been used for the fabrication of large structures. There are two reasons for this: first, the low capacity of the state-of-the-art welding machine to emit sufficient energy output for welding thick plates means that today's laser welding machines can only weld a plate with the thickness of a small fraction of an inch. This limitation will be overcome by increasing the energy output capacity. Second, current welding machines are unable to make continuous weldings, given the beam's pulsed nature. To solve this problem, an attempt has been and is being made to develop a device capable of very high pulse repetition rates.

When these two problems are overcome, laser welding is expected to be the most feasible process available for the welding of ultrahigh-strength materials.

Problems associated with welding—Welding causes many problems: welding heat changes material properties in the HAZ; in general, it lowers notch toughness; absorption of hydrogen causes hydrogen embrittlement or hydrogen cracking. The use of filler metal causes a discontinuity in material properties at the weld. (No filler metal which matches the base metal in both strength and toughness is available for today's ultrahigh-strength materials.) Thermal plasticity causes high tensile residual stresses near the weld, which have a bad effect on, among other things, the fracture strength of the material.

Since the effects of residual stress will be described in Section 2.3, hydrogen embrittlement and notch toughness of welds should now be discussed.

Hydrogen embrittlement is one of the most common and serious types of time-dependent fractures [1]. In laboratory testing, the presence of hydrogen results in a decrease in the ductility of unnotched tensile specimens and hence a decrease in the tensile strength of notched specimens. In service, failure can occur without warning, minutes to years after a static load has been applied to a structure containing hydrogen.

Where welding is involved, hydrogen is supplied from moisture either in the base material or in the filler metal through the decomposition caused by the high temperatures characteristic of welding. This hydrogen can cause cracking in the HAZ. It is well known that cracking of this type disrupted the introduction of HY-80 steel to submarine fabrication.

Characteristics of hydrogen embrittlement: As well as steels having a body-centered cubic (BCC) crystal structure, titanium and zirconium and their alloys also are susceptible to hydrogen embrittlement. The hydrogen can be introduced into these materials during processing, cleaning (acid pickling), and electroplating operations.

There are two characteristics of hydrogen embrittlement. First, it is not a form of stress corrosion cracking. In fact, fracture often occurs when the metal serves or has served as a cathode, during electroplating or in a cathodic "protection" operation. Second, embrittlement results from hydrogen contents that are greater than the equilibrium solubility limit (about 10^{-3} ppm by weight in iron and 20 to 35 ppm in titanium and zirconium at room temperature at 1 atm hydrogen pressure). The excess hydrogen which causes embrittlement can be as low as 1 ppm in high-strength steel and 35 ppm in titanium and zirconium.

Hydrogen embrittlement of steels: Figure 25 shows hydrogen-induced delayed-fracture characteristics of various steels [25]. Curves in the figure were estimated by Masubuchi and Martin [25] from

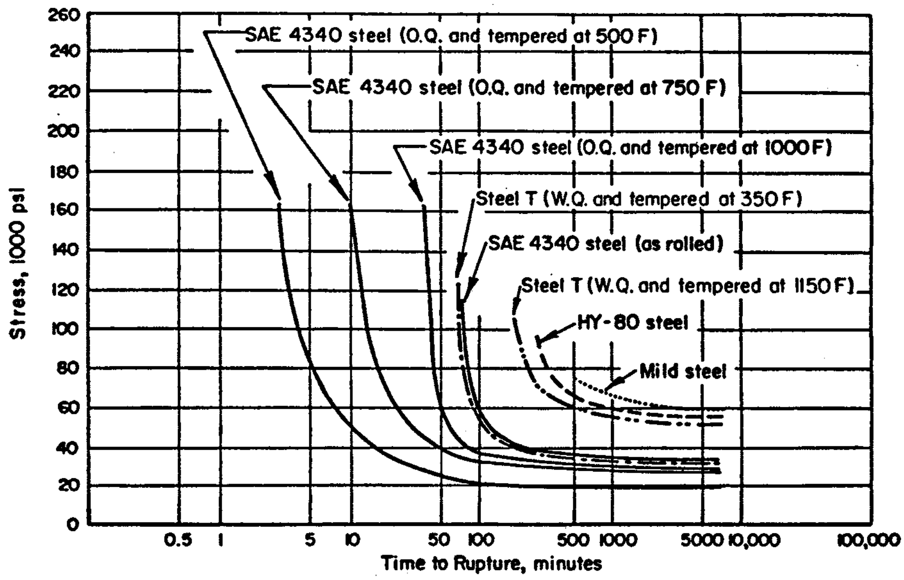

Fig. 25 Hydrogen-induced delayed fracture characteristics of various steels [25]. Note: Curves are estimated from results obtained by Simcoe *et al.* [26]. The commercial high-strength structural steel is identified above as Steel T. Table 2 contains approximate tensile strengths of these steels

experimental results conducted by Simcoe and others [26] on SAE 4340 steel quenched and tempered at difficult strength levels.

Figure 25 shows a general tendency of steel to become more susceptible to hydrogen cracking as the strength level increases. For example, SAE 4340 steel

Table 2 Summary of results of hydrogen-induced cracking tests on welded specimens made from various materials

Base Plate and Heat Treatment	Approximate Tensile Strength, psi	Number of Specimens Tested	Plate Thickness, in.	Hydrogen Charging Period, h	Summary of Test Results
1. Mild steel	25 000	6	½ to 2	up to 379	Very small cracks in the heat-affected zone of two specimens, but no cracks in four other specimens hydrogen charged up to 126½ h.
2. HY-80 (quenched and tempered)	100 000	4	½ to 2	up to 216	Small cracks in two specimens, but no cracks in two other specimens.
3. Commercial high-strength structural steel (water quenched and tempered at 1150°F)	120 000	1	¾	4½	No cracking.
4. Commercial high-strength structural steel (water quenched and tempered at 350°F)	150 000	5	½, ¾	up to 24	Several cracks were found in three specimens.
5. SAE 4340 steel (as rolled)	150 000	1	¾	14	No cracking.
6. SAE 4340 steel (oil quenched and tempered at 1150°F)	175 000	1	½[a]	6¾	Several transverse cracks.
7. SAE 4340 steel (oil quenched and tempered at 750°F)	220 000	1	½[a]	6	Fairly systematic cracks.
8. SAE 4340 steel (oil quenched and tempered at 600°F)	240 000	1	¾	1	Systematic cracks.
9. SAE 4340 steel (oil quenched and tempered at 500°F)	260 000	25	¼ to ¾	up to 16	Extensive cracks in all specimens except one which had been mechanically stress-relieved; cracks were found after hydrogen charging for a few hours or less

[a]Ground from ⅝ to ½ in. thick

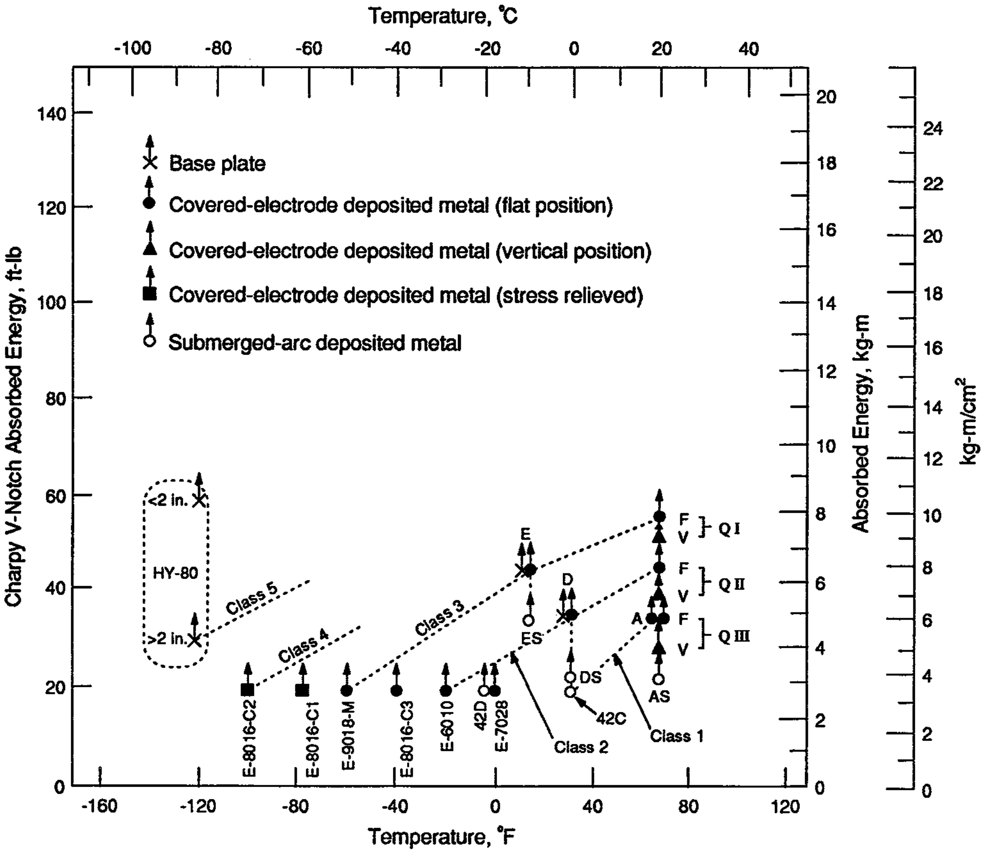

Fig. 26 Notch-toughness required by various specifications: 1. International specification for ship steel—A, D, E (AS, DS, ES are 70% of A, D, and E, respectively; submerged-arc welding). 2. AWS: E-7028, E-6010, E-8016-C3, E-9018-M, E-8016-C1, E-8016-C2. 3. IIW, Commission II: QI, QII, QIII for flat and vertical positions. 4. IIW, Commission XII: 42C, 42D (submerged arc-welding).

oil quenched and tempered at 500°F has the tensile strength of about 260 000 psi. When hydrogen is charged to this steel while it is subjected to tensile stress of 80 000 psi, it takes only 5 minutes before the steel fractures. When the stress is lowered to 40 000 psi, for example, it takes about 15 min before the steel fractures. When steel with a lower strength level is subjected to stress and hydrogen, it takes a longer time before the steel fractures. For example, when HY-80 steel is subjected to a tensile stress of 80 000 psi, it takes about 400 min before it fractures.

Masubuchi and Martin [25] investigated hydrogen-induced cracking characteristics of weldments in various steels. Table 2 summarizes their experimental results as follows:

- Mild steel was immune to hydrogen embrittlement. Of six specimens which were charged with hydrogen up to 379 h, very small cracks were observed in two specimens. No cracks were observed in the other four specimens, hydrogen charged up to 126½ h.
- As the strength level of steel increased, weldments became more susceptible to hydrogen embrittlement.
- Weldments made in SAE 4340 steel quenched and tempered at a very high-strength level were very susceptible to hydrogen embrittlement. Extensive cracks were obtained in all specimens except one which had been mechanically stress relieved. Cracks were found after hydrogen charging for a few hours or less.

Notch toughness of weld metal [1,21]: In order for brittle fracture of welded structures to be avoided, it is important that both the base plate and weld metal have adequate notch toughness. It is not difficult to obtain weld metal with notch toughness equivalent to ordinary carbon steel. However, it becomes a serious consideration when welding high-strength, quenched-and-tempered steels. These steels have both high yield strength and excellent notch toughness that is difficult to match in the weld metal. The case of HY-80 steel provides a good example of this.

Table 3 shows the requirements of AWS-ASTM specifications on weld metal notch toughness in butt joints with requirements on HY-80 steel base metal. AWS-ASTM specifications include electrodes which produce welds with from 60 000 psi to 120 000 psi

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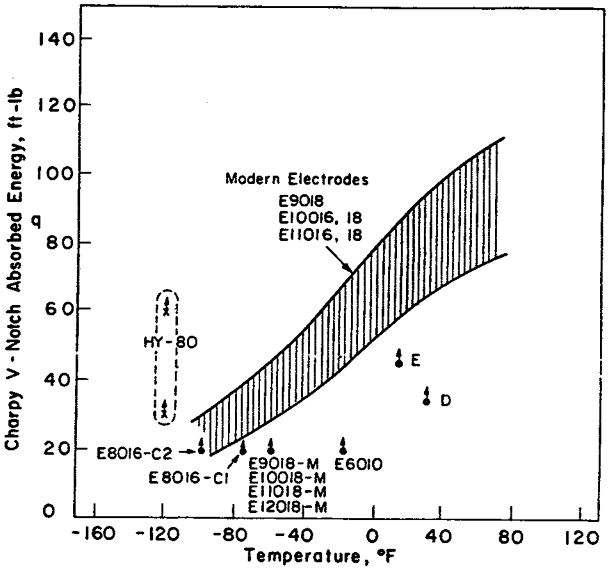

Fig. 27 Notch-toughness of weld metals of low-hydrogen high-strength steel electrodes

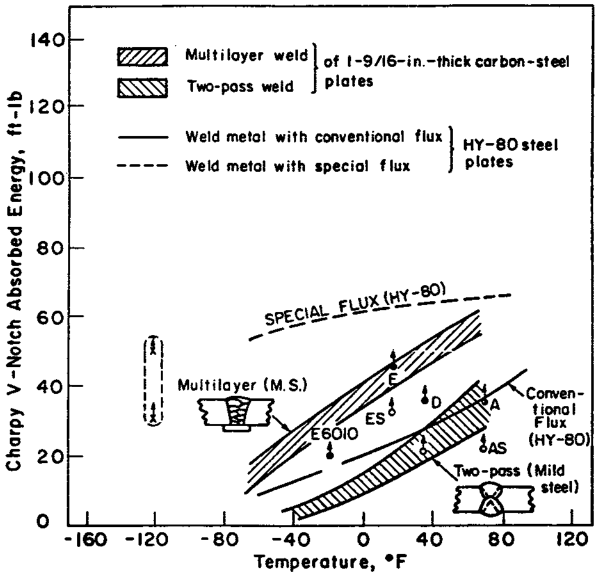

Fig. 28 Notch-toughness of submerged-arc deposited metals in mild steel and high-strength steels

tensile strengths. These electrodes include those for steels up to the HY-80 class.

In Fig. 26, the requirements of HY-80 base plates and those on covered-electrode deposited metals are compared. Taking into consideration the general trend of notch toughness increasing with temperature, the figure reveals that current AWS-ASTM specifications do not claim the weld notch toughness to be as high as that for HY-80 base plate.

Figure 27 illustrates this situation more specifically. The shaded area of the figure represents the scatter band of notch toughnesses of modern electrodes. Though these electrodes satisfy the requirements regarding yield strengths, even the maximum value of notch toughness does not match the required minimum notch toughness. For this reason, many research programs have been conducted to develop flux-wire combinations which produce weld metals as tough as HY-80 base plates. As an example, notch toughness data on weld metals made by the use of special flux-wire combinations developed at the Battelle Memorial Institute [27,28] are presented in Fig. 28.

This graph shows the Charpy V-notch transition curves of both the weld metal which had the best notch toughness and a weld metal made with conventional wire and flux. The welds were made in ½-in.-thick HY-80 steel plate by the multilayer tech-

Table 3 Notch toughness requirements of various specifications

Base Metal or Weld Metal	Specifications and Class	Charpy V-Notch: Minimum Energy Absorption, ft·lb	Charpy V-Notch: Temperature, °F
	U.S. Navy HY-80 Steel		
Base Metal	Thickness 2 in. or less	60	−120
	Thickness over 2 in.	30	−120
	AWS-ASTM		
Covered-Electrode-Deposited Metal	E6012, E6013, E6020, E7014, E7024	not required	
	E7028	20	0
	E6010, E6011, E6027, E7015, E7016, E7018	20	−20
	E8016-C3, E8018-C3	20	−40
	E9015-D1, E9018-D1, E10015-D2, E10016-D2, E10018-D2	20[a]	−60
	E9018-M, E10018-M, E11018-M, E12018-M	20	−60
	E8016-C1, E8018-C1	20[a]	−75
	E8016-C2, E8018-C2	20	−100

[a]Stress-relieved condition.

Chapter V
Hydromechanical Principles

E. E. Allmendinger, M. De La Vergne, and H. Jackson

1. Overview

THE SUBJECT of *hydromechanics*, as applied to the design of submersibles or other types of marine vehicles, involves the field of engineering concerned with the mechanics of the fluid, water, in motion. It is customary to consider this subject as being divided into *hydrostatics* and *hydrodynamics*—the former often being viewed as but a special case of the latter in which there is no relative motion between the vehicle under study and the surrounding water particles. As will be seen, hydrostatic and hydrodynamic design considerations are interrelated to the extent that it is expedient to present their underlying principles in the same chapter entitled, simply, Hydromechanical Principles.

Section 2 of this chapter provides definitions and introductory material intended to facilitate the presentation and understanding of hydromechanical principles as they apply specifically to the design of the so-called *manned, neutral-buoyancy submersible* with three-dimensional mobility and as they may be adapted for use in the designs of a broad spectrum of both manned and unmanned submersible types with one-, two-, or three-dimensional mobility. A rather extensive discussion of hydromechanical categories of submersibles is included in support of both functions of the introductory material. The section is concluded with an overview of submersible operating conditions which, collectively, form a *drive profile* and the control systems required to transit this profile safely and effectively.

Section 3 concerns *hydromechanical principles-hydrostatics* primarily as they are directly applicable to the design of manned, neutral-buoyancy submersibles. These principles relate to design considerations, including

1. statical equilibrium and stability fundamentals
2. statical stability details
3. dynamic stability
4. weight-displacement-volume relationships
5. hydrostatic characteristics
6. static control systems

In several instances, it will be necessary to discuss these considerations with reference to three major operating conditions:

- surface condition,
- transition condition—passing through the air/water interface on diving/surfacing, and
- submerged condition.

For example, statical stability details differ significantly among these conditions and, consequently, must be discussed for each one.

Section 4 concerns *hydromechanical principles-hydrodynamics*, also as directly applicable to the design of manned, neutral-buoyancy submersibles. The section is introduced with a review of the equations of motion followed by a presentation of hydrodynamic principles relating to the following design considerations:

1. resistance and propulsion,
2. propulsors, and
3. stability and control.

It will be necessary to discuss them with reference to

- low and high forward speeds and
- low and high vertical speeds

because they and related submersible characteristics differ substantially between the low- and high-speed regimes. For example, minimizing resistance is not critical for designs whose performance requirements include low forward or vertical speeds or both. Consequently, these designs are not obliged to use streamlined, enclosed envelopes of the type discussed in Chapter I. On the other hand, minimizing resistance is of great importance for high forward or vertical speeds or both, with streamlining in the direction of motions being mandatory.

Note should be taken of two additional hydrodynamic design considerations not listed above: *motion of a submersible in waves* and *tether dynamics*. Motion in waves primarily pertains to launch/retrieval dynamics, particularly to the retrieval phase, and the relative motions of two bodies of vastly different masses and shapes represented by the submersible and its support ship. In most instances, little can be done by the submersible's designer to overcome the severe dynamic mismatch between these bodies, the burden for safe and rapid handling of the submersible falling essentially on the design and expert operation of the launch/retrieval system located on the support ship. Consequently, this subject is not addressed in Section 4. The reader should examine reference [1] or other works on naval architecture for more detailed information on motion in waves.

The subject of tether dynamics is important in the designs of all manned and unmanned underwater-vehicle systems employing tethers, primarily from the points of view of drag and tether management and attachment to vehicle to minimize restrictions on maneuverability. However, since this publication is essentially concerned with untethered, manned submersibles, tether dynamics is not discussed in Section 4, either.

No attempt is made in this chapter to derive mathematical formulations associated with hydromechanical principles presented; reference [1] and other works on naval architecture are useful for these derivations. Other references given provide additional background material on subjects discussed.

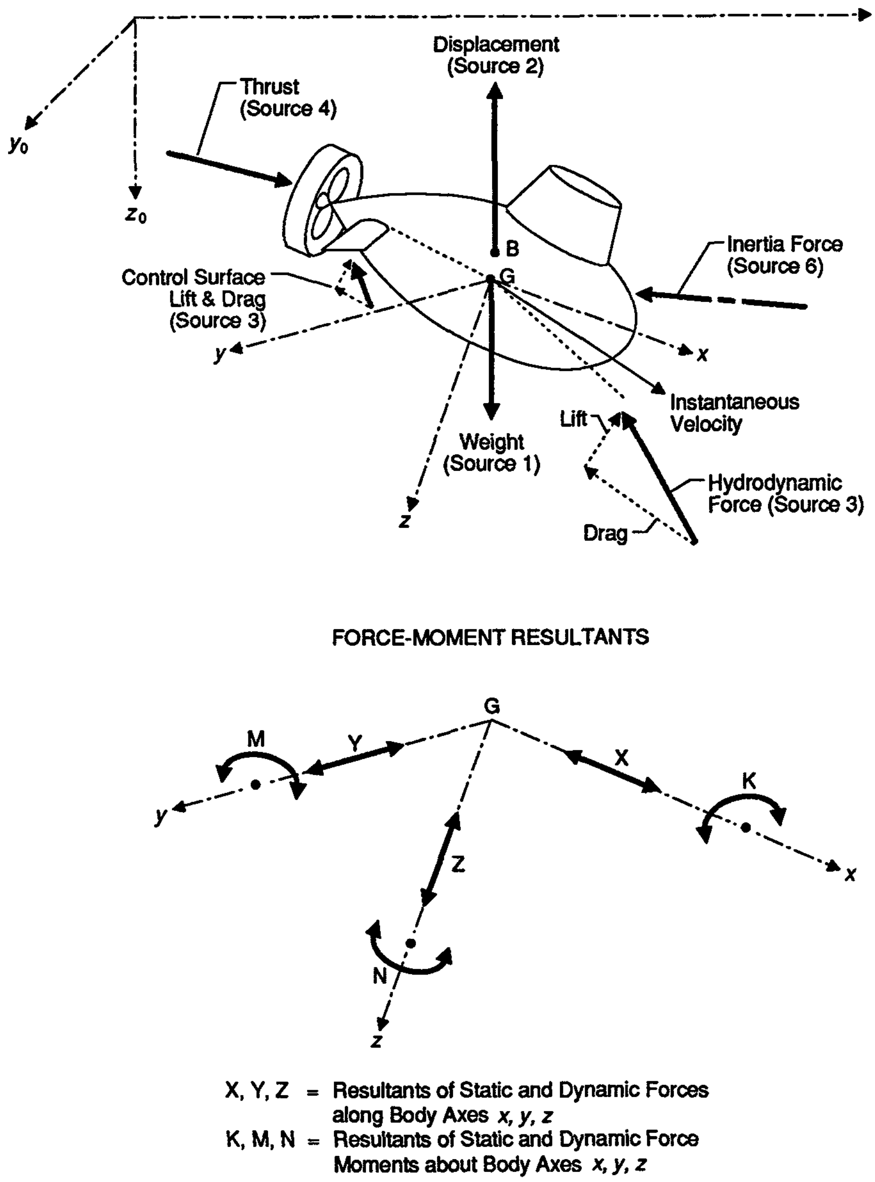

Fig. 1 Forces/moments acting on a submersible

2. Introduction

2.1 Forces and Moments Acting on Submersibles

Hydromechanical principles pertinent to submersible design concern the behavior of these vehicles when acted on by forces/moments from a few or all of the following force sources:

1. Earth's gravitational field (weight)
2. Resultant of hydrostatic pressure on the submersible (displacement)
3. Relative motion between a submersible's body, its appendages, and the surrounding water particles (lift and drag or resistance)
4. Reactions to time-rate-of-change in linear and angular momentum of water particles caused by the submersible's thrust-producing devices (thrust and torque)
5. Contacts with objects, waves, and wind (contact forces)
6. Resistance to changes in motion evidenced by linear and angular accelerations (inertia)

These forces, except for those of Source 5, are shown acting on a neutral buoyancy submersible in Fig. 1, in which the vehicle is arbitrarily oriented in space at a particular instant with reference to the x_0,y_0,z_0 earth axes[1] as indicated with x,y,z body axes attached to it. They, and associated moments, may be categorized as either *static* or *dynamic*. Forces/moments from Sources 1 and 2 and the objects of Source 5 may be called *static* in the sense that they exist in the absence of relative motion between the submersible and surrounding fluid field. On the other hand, forces/moments from Sources 3 and 4, and the waves and wind of Sources 5 and inertia of Source 6 may be called *dynamic* because their existence does depend on the presence of relative motion between submersible and surrounding fluid or change in motion. From another perspective, they may be categorized as follows, with force/moments from some sources being in one or more of these categories:

1. Control forces/moments, used by the operator for control of the vehicle:
   a. Source 1: changes in weight/moments
   b. Source 2: changes in displacement/moments
   c. Source 3: thrust/moments
   d. Source 4: lift/moments
   e. Source 5: purposeful contact with objects
2. Induced forces/moments, induced or created by changes in position or motion of the vehicle:
   a. Sources 1 and 2: changes in weight, displacement, weight/displacement moment
   b. Source 4: lift-drag/moments, induced by angles of attack, and resistance
   c. Source 6: inertia forces/moments
3. External forces/moments, over which the operator has no direct control:
   a. Source 1: basic weight
   b. Source 2: basic displacement
   c. Source 5: accidental contact with objects, waves and wind

Brief comments on the static-dynamic categories follow with remarks on control, induced and external forces/moments included. Details are given in Sections 3 and 4 of the chapter.

2.1.1 *Static Forces/Moments*

Examples of these forces are shown in Fig. 2.

Weight W of Source 1, as was seen in Chapter I, is the sum of the weights of all items comprising the submersible, condensed into seven weight groups, and the loads it carries in various operating conditions. The weights of the seven groups remain constant during a particular dive while the loads vary.

Displacement Δ of Source 2 is the product of the volume of displacement $\overline{V}$ and the specific weight γ of the surrounding water. $\overline{V}$ is the sum of the volumes of all water-excluding items of the submersible and the loads it carries, but note that items and loads in containers do not contribute to $\overline{V}$. Both components of Δ may vary during a dive—$\overline{V}$ because of compressibility of the individual volumes under hydrostatic pressure and γ because of changes in water temperature and salinity with changes in depth, in operating areas, or in both. At deep depths, changes in pressure also alter γ. If changes in $\overline{V}$ and γ offset each other, an *isoballast* condition is said to exist; no changes in ballast or displacement are required to maintain the desired relationship between W and Δ.

The difference between W and Δ, or their resultant, is called *buoyancy*.[2] *Positive buoyancy* exists when W is less than Δ, *negative buoyancy* when W is greater than Δ, and *neutral buoyancy* when W equals Δ. The term *neutral buoyancy* is used in identifying the type of submersible whose total submerged weight and displacement are equal in the so-called *design condition*, a term to be discussed presently.

W and Δ have collinear lines of action when the submersible is at rest. When inclined away from this position by forces/moments from other sources about either the x or y axes, W and Δ form righting moments (*RM*) which resist these inclinations. This resistance is called statical stability, which varies

[1]The x_0,y_0,z_0 earth axes are fixed in the earth with the $+z_0$ direction being vertically downward; these axes provide a fixed reference frame for hydromechanical studies. The x,y,z body axes are fixed in, and move with, the submersible, their origin being at some convenient point such as the vehicle's center of gravity G. Both sets of axes form right-hand, orthogonal axes sytems.

[2]The term "buoyancy" is often used for "displacement" instead of being reserved for use as the "resultant" of W and Δ. The latter definition will be used herein. A contradiction seems to exist, perhaps for psychological reasons, in the direction sense assigned to buoyancy; "positive buoyancy" is vertically upward in the negative z_0 direction, whereas "negative buoyancy" is vertically downward in the positive z_0 direction.

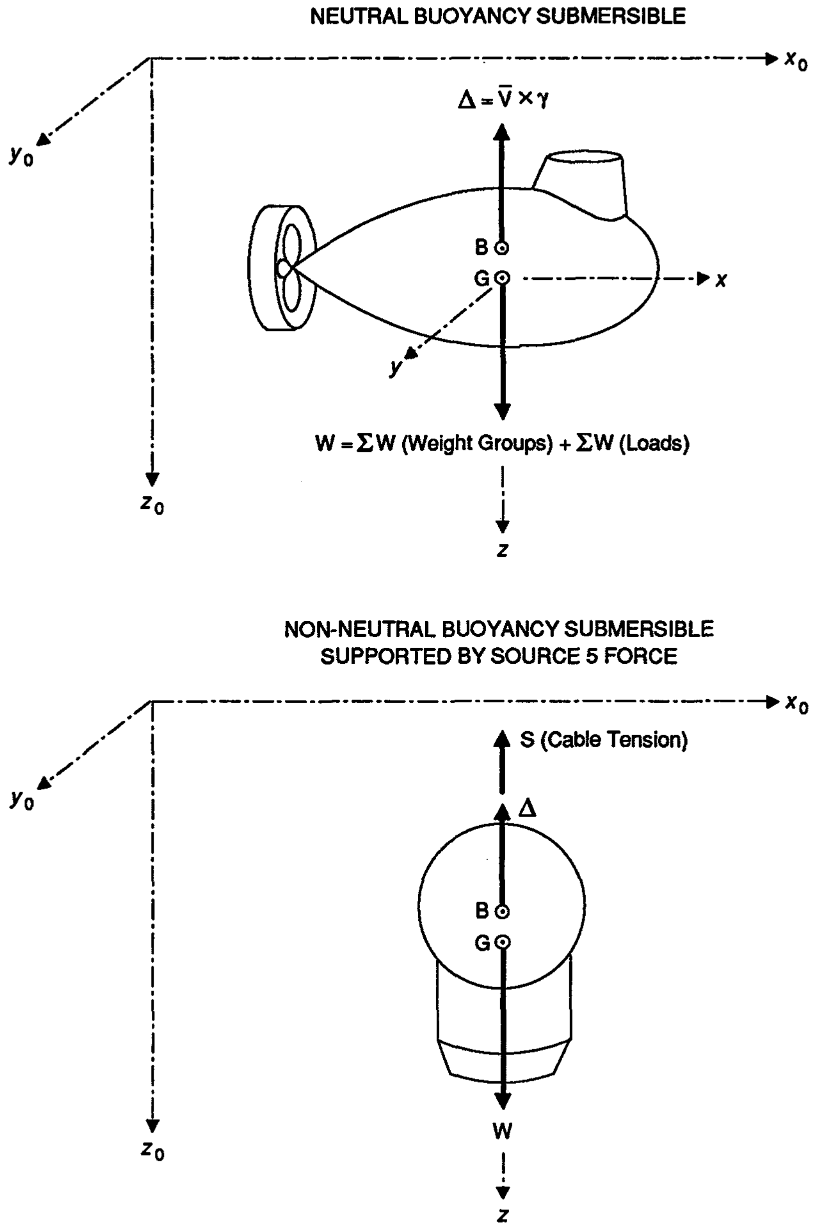

Fig. 2 Examples of static forces

with the angle of inclination θ about the x-axis or α about the y-axis. A plot of RM versus θ or α is the statical stability curve. The development of RM and a typical statical stability curve are shown in Fig. 3.

Source 5 objects include the following: items attached to the submersible, such as lowering/lifting and mooring cables or tethers; objects to which the submersible may be attached by its "grabber" manipulators or other devices, such as the columns or struts of underwater structures; the seafloor; and objects contacted accidentally. The presence of any Source 5 forces/moments significantly alters the "free-floating" characteristics of the vehicle.

2.1.2 Dynamic Forces/Moments

Examples of these forces/moments are show in Figs. 4 to 7.

Lift (L) and drag (D) forces/moments of Source 3 are, respectively, normal and parallel to the direction of fluid flow, as shown in Fig. 4. L generally exists when there is a non-zero angle of attack α developed between the body's axis and the direction of fluid flow. The angle is non-zero because of the body's asymmetry with respect to its axis, inclination of a symmetric body to the direction of flow, or both. In this case, the bodies are the submersible's hull and

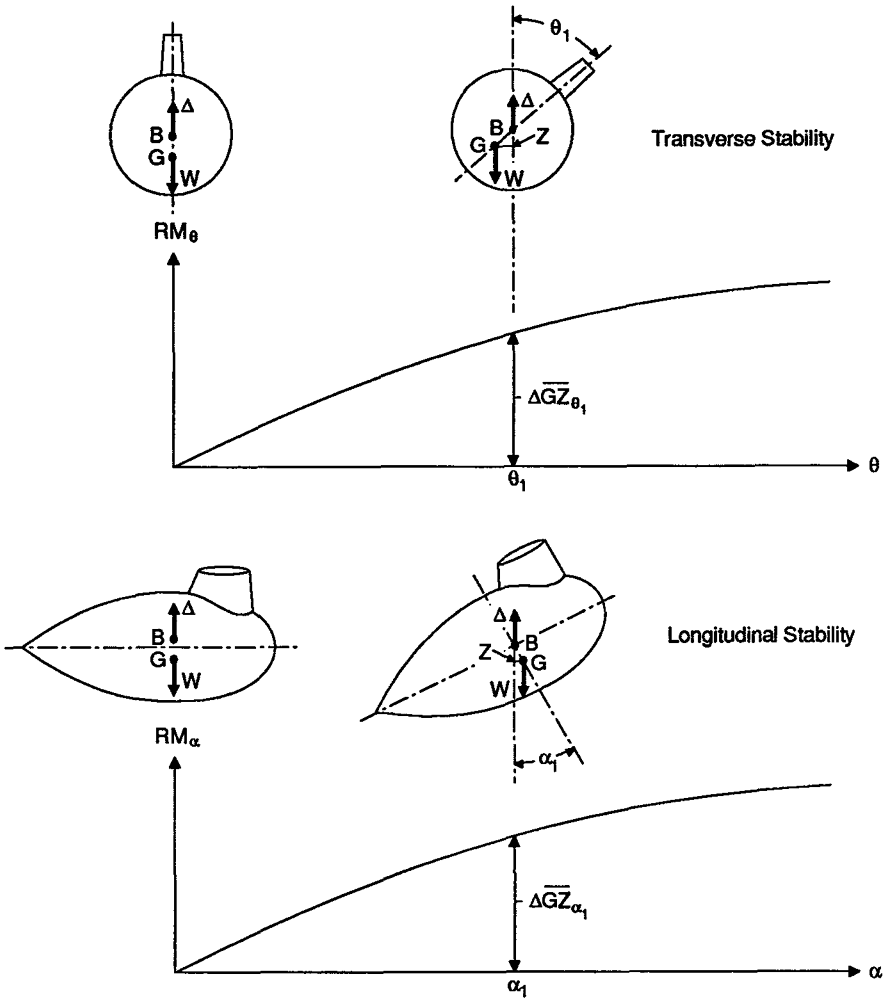

Fig. 3 Statical stability curves of righting moments

its appendages, the latter including items such as the sail or access-hatch shield, fixed fins used for motion stabilization, and movable control surfaces such as rudders and planes. *L* forces/moments are used to control the submersible's motion at the higher forward speeds; *D* forces/moments always accompany *L*. They are also present for zero or near-zero values of α, in which case drag is usually called resistance *R* by naval architects. This force is important in resistance and propulsion, or speed and power, studies. The nature of *R* is outlined in Fig. 5, and details of its components are discussed in Section 4.

Thrust *T* of Source 4 is created by thrust-producing devices on the submersible such as screw and cycloidal propellers and waterjets. Free, shrouded or ducted screw propellers are most frequently used. These devices increase the linear momentum of the water particles as they pass through them, and thrust is the reactive force to that causing the increase in the particles' momenta. *T* is used to produce forward speed and, in some instances, vertical speed in descent/ascent operations. *T* and its moments are also used to control the submersible's motion at zero and low forward speeds when *L* forces/moments are ineffective. Propellers also increase the angular momenta of water particles passing through them; torque *Q* is the reactive moment to that causing this change. *Q* is an unwanted moment which tends to rotate the submersible in a direction opposite to that in which the propeller is rotating. If pronounced, this moment must be compensated for in some manner. Using two coaxial counter-rotating propellers is one way of achieving this compensation. Various types and arrangements of thrust-producing devices are shown in Fig. 6.

Wave and wind forces of Source 5 act on the submersible during its surface operations. These forces are usually of greater concern during the submersible's surfacing/retrieval phase than in its launch/diving phase because the former requires more time than the latter, considerably more time under unusual or emergency conditions. Wave-wind

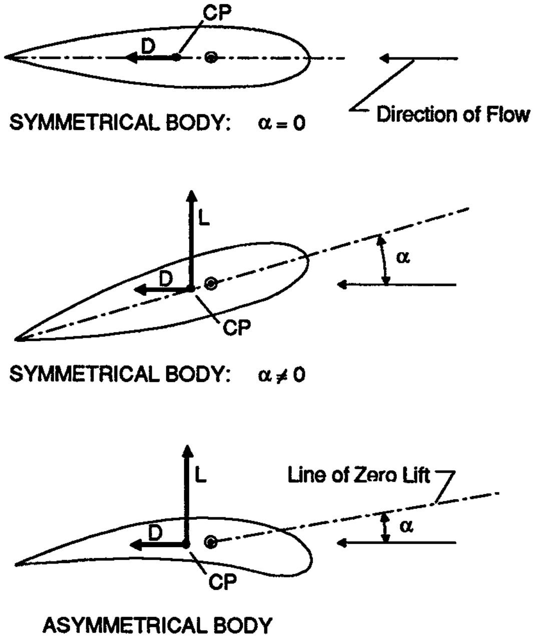

Fig. 4 Examples of lift and drag forces illustrated by all-movable control surfaces

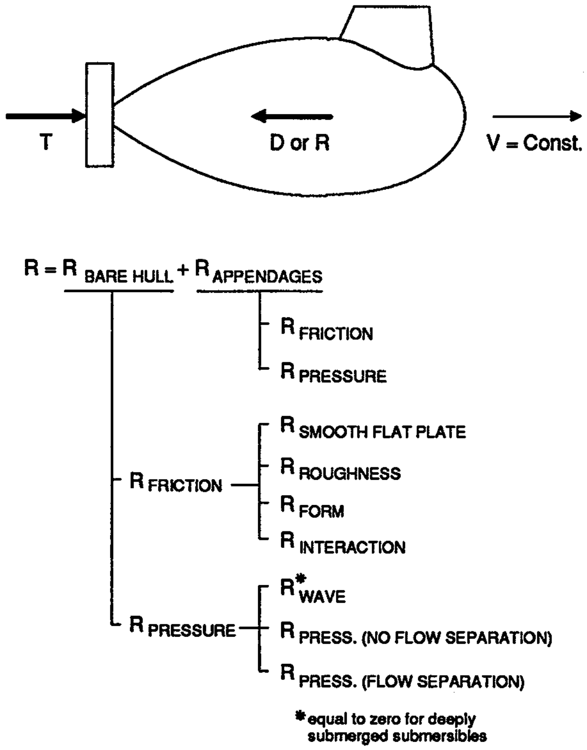

Fig. 5 The nature of resistance

forces/moments are of concern in providing for safe angles of submersible inclincation and for safe and efficient launch/retrieval systems.

An introductory discussion on safe angles of inclination necessitates a brief review of the second type of stability to be introduced in this section, *dynamic stability* or *righting energy*, and the wave-wind-generated *heeling moment* and *heeling energy*. Dynamic stability is closely associated with statical stability, the latter having been defined as the resistance to inclinations about the x and y axes with x-axis inclinations being of primary interest here. As was seen, a curve of *RM* versus angle of heel θ can be drawn. Dynamic stability is the area under this curve to any specified θ_{rad}, as shown in Fig. 26 (see page 225), which illustrates a more advanced discussion of the subject. This area is the righting energy or energy required to heel the submersible to that angle. Mathematically, the dynamic stability curve is the first integral of the statical stability curve. Both wave-wind forces are assumed to strike the exposed, above-water structure broadside. They act in conjunction with the water resistance force, applied at the center of lateral resistance, to form heeling moments. These moments vary with the strength of the wave-wind force, the lateral area of the exposed structure, the height of the center of this area above the water line, and θ. A heeling moment (*HM*) curve can be drawn on the same coordinates as the statical stability curve, as shown in Fig. 26, and the area under this curve is heeling energy. The heeling energy curve is the first integral of the heeling moment curve. As indicated in Fig. 26, the submersible will heel over to a maximum angle of heel, θ_M, at which the heeling energy equals the righting energy—assuming that the resistance to rolling is negligible. It will suffice to say here that θ_M must be limited to acceptable values by providing adequate dynamic stability and reducing the submersible's lateral area to the extent feasible, which usually presents no problem for small vehicles.

Safe and efficient launch/retrieval systems, as noted in Chapter I, primarily involve the design and operation of the main portion of the system[3] located on the support ship because there is little the submersible designer can do to overcome the dynamic mismatch between vehicle and ship. Freeboard is one feature which can be incorporated into the design to offset at least partially the effect of wave action in hampering the attachment of the lift-line to the submersible, which is often accomplished by divers on the vehicle's topside structure.

Inertia forces/moments of Source 6 may be introduced with the observation that applied forces/moments imparting linear and angular accelerations to a submersible are always larger than the products of the vehicle's mass/mass moment of inertia times

[3]The most widely used launch/recovery is a stern-mounted, hydraulically operated, single-point lift, A-frame system. Remote systems have been designed in which the submersible is launched/retrieved at depths at which surface wave action is negligible.

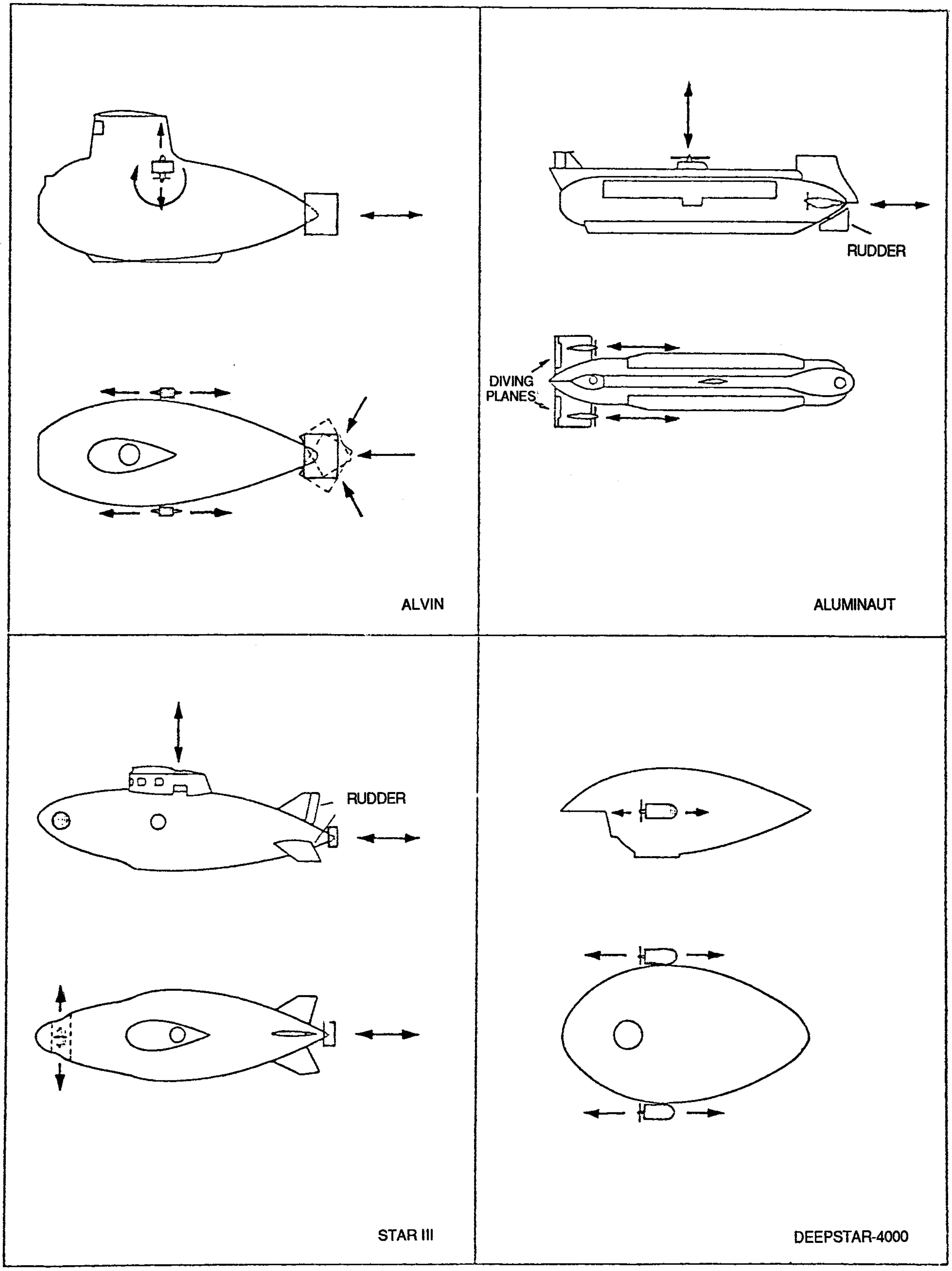

Fig. 6 Examples of types and arrangements of thrusters

these accelerations. In this regard, three masses and three mass moments of inertia must be considered, respectively, with reference to linear and angular accelerations: those of (1) the submersible and its loads, (2) the water entrained in the submersible's envelope and its appendages, and (3) the water adjacent to the submersible which is accelerated by its motions. Masses and mass moments of inertia (1) and (2) are constant for a given vehicle envelope, enclosed or open, and for a specific condition of loading. These quantities for (3) vary with type of motion involved: different quantities of water are linearly accelerated in the x, y, z directions and angularly accelerated about these axes, as illustrated in Fig. 7. It is evident from this figure, for example, that large quantities of water are accelerated by the submersible's motion along the y-axis or about the z-axis while small quantities are accelerated by its motion about the x-axis. The third mass and mass moments of inertia may be viewed either as added mass or added mass moments of inertia or, when multiplied by accelerations, as added hydrodynamic forces and moments created by the submersible's accelerations through the water. If the former view obtains, the sum of (1), (2), and (3) is referred to as the virtual mass or virtual mass moment of inertia of the vehicle. Because of the variations in (3), these quantities will vary with the motions involved.

The sum of masses (1) and (2) leads to use of the term *dynamic displacement*, Δ_D, to differentiate it from the usual *static displacement*, Δ, as previously defined. For neutral-buoyancy submersibles, then,

$$W = \Delta$$

$$W + W_{EW} = \Delta + \Delta_{EW} = \Delta_D \qquad (1)$$

where W_{EW} and Δ_{EW} are weight and displacement of entrained water. Dividing Δ_D by the gravitational acceleration g yields the sum of masses (1) and (2). Entrained water, sometimes called "free-flooding" water, is always present in varying amounts in enclosed envelope submersibles, its volume equal to the envelope's volume minus the volume of the water-displacing items within the envelope plus the free-flooding volume of the appendages. For relatively "closely-packed" envelopes, the volume of entrained water can be about 8 percent to 15 percent of $\overline{V}$. Entrained water may also be present in open-frame vehicles, for instance in nonpressure resistant containers and in the frame's tubular structure.

Mass and mass moments of inertia (2) and (3) will vary with the type of envelope: enclosed envelope submersibles entrap more water and accelerate more adjacent water than do open-frame vehicles. An appreciation for the latter situation may be gained by imagining a light, solid door and a screen door of the same size immersed in a pool of water, each door

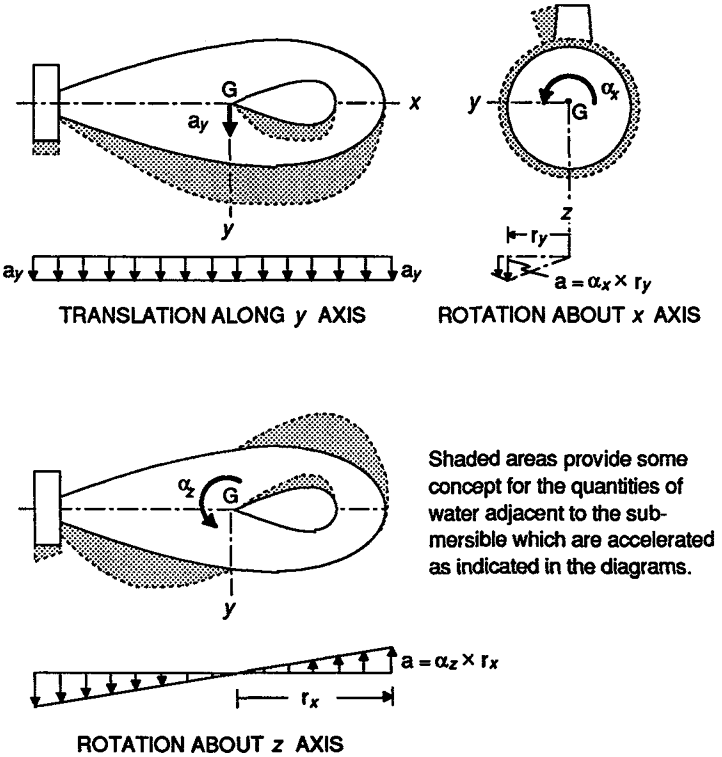

Fig. 7 Concept of added mass and mass moment of inertia

being given separate translational and rotational motions involving the same linear and angular accelerations. Obviously, in both instances, a much larger mass of adjacent water will be moved by the solid door than by the screen door, with the result that a much greater force and moment must be applied to it than to the screen door. Said another way, the screen door, or open-frame submersible, is much more "transparent" to the adjacent water as far as these motions are concerned. A design implication is that enclosed-envelope submersibles require greater control forces/moments than do similar-sized, open-frame submersibles for comparable maneuvering capabilities.

2.2 Equilibrium and Nonequilibrium States

Forces/moments from various sources act on a submerged submersible so as to place it in states of *equilibrium* or *nonequilibrium* which may be expressed in simplest manner by the equations

$$\begin{aligned} X&=0 \text{ or } \neq 0 \qquad K=0 \text{ or } \neq 0 \\ Y&=0 \text{ or } \neq 0 \qquad M=0 \text{ or } \neq 0 \\ Z&=0 \text{ or } \neq 0 \qquad N=0 \text{ or } \neq 0 \end{aligned} \tag{2}$$

As indicated in Fig. 1, X, Y, and Z and K, M, and N are, respectively, the algebraic sums, or resultants, of all forces in the x-, y-, and z-axis directions and moments about these axes. If the resultants of all of the equations (2) are zero, the submersible is said to be in a state of *static* or *dynamic equilibrium* in which it is either at rest, with no motion relative to the x_0,y_0,z_0 earth axes, or in steady-state motion with respect to these axes—that is, proceeding in a straight-line with constant speed. Inertia forces/moments, of course, do not exist for these equilibrium states. If the resultant of one or more of equations (2) is not zero, the submersible is said to be in a state of *static* or *dynamic nonequilibrium*. For dynamic conditions, the non-zero resultants cause non-steady-state motion of some type, with linear and angular accelerations creating inertia forces/moments.

Hydrostatic studies, then, are concerned with the behavior of a submersible in states of

1. Static equilibrium—the vehicle at rest with reference to the x_0,y_0,z_0 axes
2. Static nonequilibrium—the vehicle being artificially restrained from motion for the purpose of analyzing static forces/moments acting and their affects

Examples of studies 1 and 2 above include, respectively, those required to find equilibrium condition points on the equilibrium polygon and those associated with statical stability studies in which the submersible is inclined away from its position of static equilibrium to ascertain righting moments developed at various angles of heel.

Hydrodynamic studies are concerned with the behavior of a submersible in states of

1. Dynamic equilibrium—the vehicle possessing steady-state motion or the vehicle at rest with the surrounding water possessing this motion, as would be the case of a submersible holding position in a steady current field
2. Dynamic nonequilibrium—the vehicle possessing some type of nonsteady-state motion with reference to the x_0,y_0,z_0 axes.

Examples of studies in 1 and 2 above include, respectively, those necessary to establish speed-power relationships in resistance and propulsion studies and those involved in maneuvering and control investigations utilizing equations of motion.

Equations of motion are used to analyze the motion of a submersible and forces/moments causing it, which is described by the path generated by the body axes origin with reference to the earth axes origin, as indicated in Fig. 8. Motion can be further characterized in terms of *degrees of freedom*. In general spatial motion, the vehicle is free to move in translation in the x,y,z directions and in rotation about these axes, thus possessing six degrees of freedom. In such motion, there is a large number of cross-coupling effects which results in a complex set of six equations of motion for

1. surge (in the x-direction)
2. sway (in the y-direction)
3. heave (in the z-direction)
4. roll (about the x-axis)
5. pitch (about the y-axis)
6. yaw (about the z-axis)

As the degrees of freedom of motion are reduced, the equations of motion are progressively simplified until they parallel the familiar mechanics equations of

$$F = ma \quad \text{and} \quad M = I\alpha$$

for one degree of freedom only, for example, the submersible being constrained to translate in x-direction or to rotate about the z-axis without other motions being involved. These simplified equations are very useful in submersible design. The most simple equation is the basis for speed-power studies, in which case the inertia force ma is zero. The simplified moment equation is useful in maneuvering/control studies for zero and very low forward or descent/ascent speeds. An intermediate level of simplification of these equations is obtained by restricting the submersible's motion to the horizontal or vertical planes, in both cases the submersible being restricted to three degrees of freedom:

- horizontal plane (surge, sway, yaw)
- vertical plane (surge, heave, pitch)

with three equations of motion involved for each plane.

It should be noted, with reference to vertical plane and spatial motion, that statical stability is one of the factors entering into the equations of motion and is

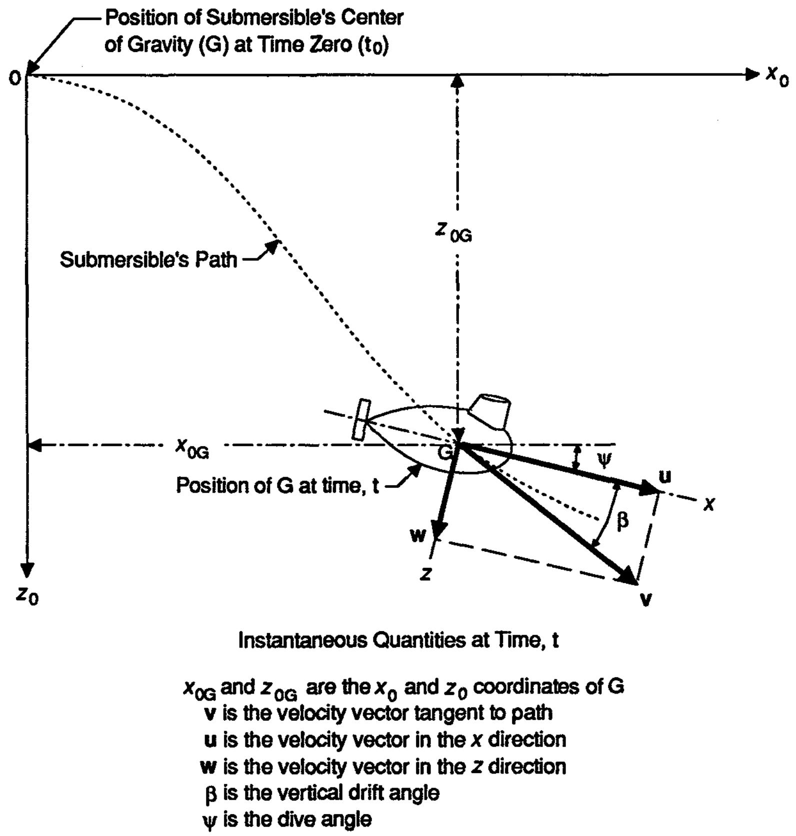

Fig. 8 Vertical plane motion of a submersible with reference to earth axes

here called *metacentric stability*. As for static inclinations away from the equilibrium position, the metacentric stability's weight-displacement moment also resists inclinations away from the horizontal by dynamic forces/moments. For low speed submersibles, the lower limit of metacentric stability must be such as to provide for safety and an adequately stable platform for conducting underwater tasks. Its upper limit should be such as to avoid requiring excessive thrust forces/moments for maneuvering. For higher-speed submersibles, metacentric stability is an important factor in motion stability considerations, as will be seen.

Motion stability is the third type of stability to be introduced in this section. It is briefly discussed at this point with respect to vertical plane motion involving forward (x-direction) and vertical (z_0-direction) speeds.

Motion stability with respect to forward speed is demonstrated in Fig. 9, which depicts a submersible moving in a vertical plane with constant speed on a *straight-line, horizontal* path at a *specified depth* (words in italics are characteristics of the initial motion). A very short-duration force is shown disturbing the vehicle's initial motion. The question arises concerning the kind of motion the submersible will have after the disturbing force ceases, assuming, in this instance, that all control surfaces are fixed in the neutral position. The answer involves the concept of *controls-fixed motion stability*, the level and kind of stability possessed by the vehicle depending on the number and type of initial motion characteristics retained after the disturbance. In this regard, the lowest to the highest level of stability retains one, two, and three characteristics, respectively:

- Level 1, straight-line stability, retains only the straight-line characteristic
- Level 2, directional stability, retains the characteristics of both straight-line and horizontal initial motion
- Level 3, positional stability, retains the characteristics of straight-line and horizontal motion at the initial specified depth

It should be understood that a directionally stable submersible must also possess straight-line stability and that a positionally stable submersible must also be directionally stable. This third level of motion stability is unachievable without the use of control forces/moments. In Fig. 9, note that directional stability is associated with either a smooth or oscillatory transition between the initial and final paths depending on whether the so-called stability indices are real or complex (for details see the Maneuvering and Control chapter of reference [1]).

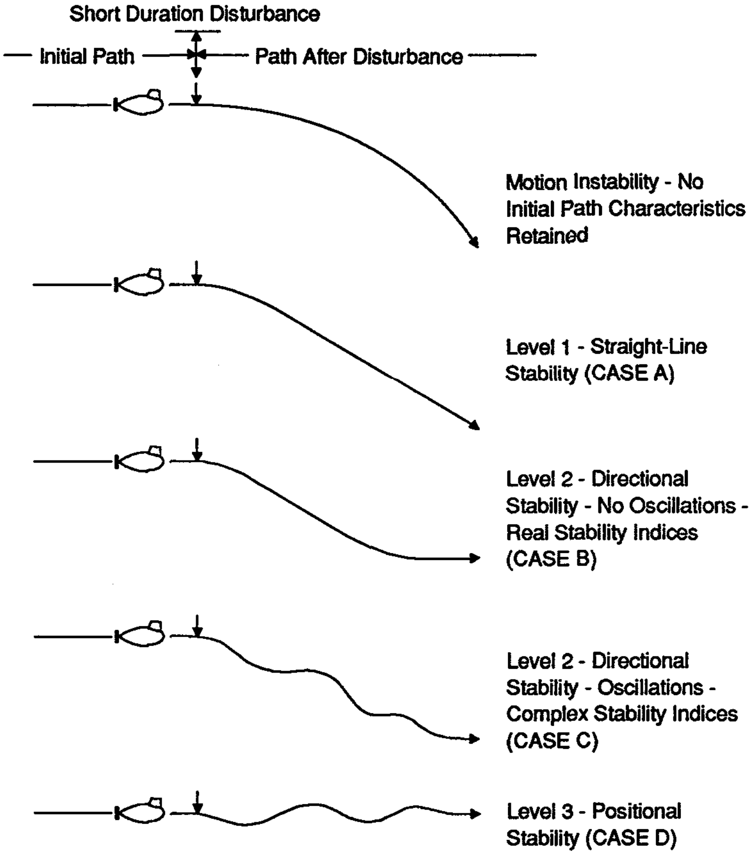

Fig. 9 Motion stability in the vertical plane

A submersible may also possess motion instability, as indicated in Fig. 9, in which case none of the initial motion characteristics are retained after the disturbance. In this instance, the vehicle is said to have *straight-line instability* or *directional instability*, if, in the latter case, the stability indices are complex and the amplitude of the transition path's oscillations increase, rather than decrease, with time.

The concept of motion stability is also applicable to the movement of a submersible in the horizontal plane, straight-line stability being the only level of controls-fixed stability attainable. The reason for this is that metacentric stability, or the weight-displacement righting moment, is absent in horizontal plane movement, whereas it is present in vertical plane movement. In the latter instance, it plays a vital role in the vehicle's ability to attain either straight-line or directional stability by resisting the efforts of induced hydrodynamic moments to cause motion instability. Its effectiveness in this role is a function of its magnitude and forward speed, this effectiveness increasing with its magnitude and decreasing with speed since induced moments increase with speed while the metacentric stability remains constant. In essence, then, attainment of directional stability in vertical plane motion means that, for a given initial forward speed, the metacentric righting moment has overpowered the net induced hydrodynamic moments, tending to cause instability and thereby causing the submersible to regain its level-flight path.

Motion stability is desirable in the horizontal plane and mandatory in the vertical plane due to the potentially serious consequences of loss of depth control. Design objectives regarding the level and type of motion stability for a submersible are set forth in its performance requirements, which emphasize either path-keeping or path-changing abilities, subjects to be discussed shortly.

Motion stability with respect to vertical speeds may be of concern in the design of a submersible because of emergency ascents, particularly if it has considerable x-y plane asymmetry due to the topside sail, or access hatch shield, and other appendages. Such asymmetry may result in oscillatory motions about the x, or roll, axis by induced hydrodynamic forces/moments as the vehicle rises in the water column. Motion stability in this sense, then, should be sufficient to limit the amplitudes of these oscillations to acceptable values avoiding synchronous conditions. As before, metacentric stability and speed play important roles in motion stability. In this

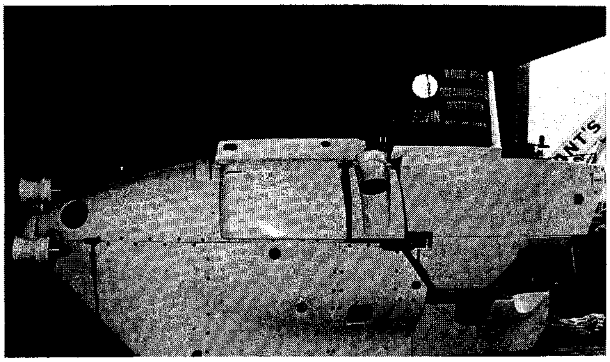

Fig. 10 Example of a neutral-buoyancy submersible. The manned, deep-submergence vehicle *Alvin* is the primary platform for conducting deep-sea science and is operated by the Woods Hole Oceanographic Institution. (Courtesy of WHOI)

regard, emergency ascent speeds are relatively high due to large positive buoyancies created by the release of heavy weights such as battery packs and droppable keels. These weights are low in the submersible, and their release usually results in a rise of the center of gravity, G, and a reduction in metacentric stability. In some designs, the pressure hull is released from the rest of the submersible in an emergency ascent. In any case, the metacentric and motion stabilities should be thoroughly investigated for this critical emergency operating condition.

2.3 Hydromechanical Categories of Submersibles

A submersible may be categorized in three ways from the perspective of hydromechanics: by *design condition, speeds,* and *maneuvering.* The following discussion of these categories covers a broad spectrum of both manned and unmanned submersibles.

2.3.1 *Design Condition Category*

Representative vehicles of the following categorization of submersibles based on the design conditions of neutral and non-neutral buoyancy are shown in Figs. 10 to 13.

1. Neutral-buoyancy submersibles
   a. Reserve displacement
   b. Zero reserve displacement
2. Non-neutral-buoyancy submersibles
   a. Statically supported
   b. Dynamically supported
      (1) Thrust-supported
      (2) Lift-supported

All categories except 2a, which is always restrained, may be further categorized as *unrestrained* or *restrained.* These designations indicate the degree to which submersibles are restrained by physical linkages,[4] or tethers, to other I systems[5] of the mission system such as a surface support ship. "Unrestrained" indicates complete freedom of movement, "restrained" limited freedom of movement.

Fig. 14 illustrates these categories of submersibles. The force systems are shown acting so as to place all of the submersibles, except 2b2, in submerged conditions of static equilibrium with zero trim and list. Category 2b2 is shown in a submerged condition of dynamic equilibrium with zero or non-zero trim because motion is necessary to generate lift and because trim, or an inclination about the y-axis, is sometimes used to create lift. These specific submerged conditions are herein called *design conditions* to indicate that they set forth the basic force systems acting on the submersible within these categories.

The identification of design conditions by no means implies that the submersible always operates in one of these conditions—either stationary in the water-column or, for Category 2b2, proceeding

[4]Linkages, or connections, between two or more *I* systems of a mission system may be visual, electronic, acoustical, and physical. Physical linkages, hereafter called tethers, are connections such as fiber-optic "threads," wires, cables, and drill-pipe strings.

[5]As defined in Chapter I, *I* systems are the major independent systems which, collectively, form the mission system.

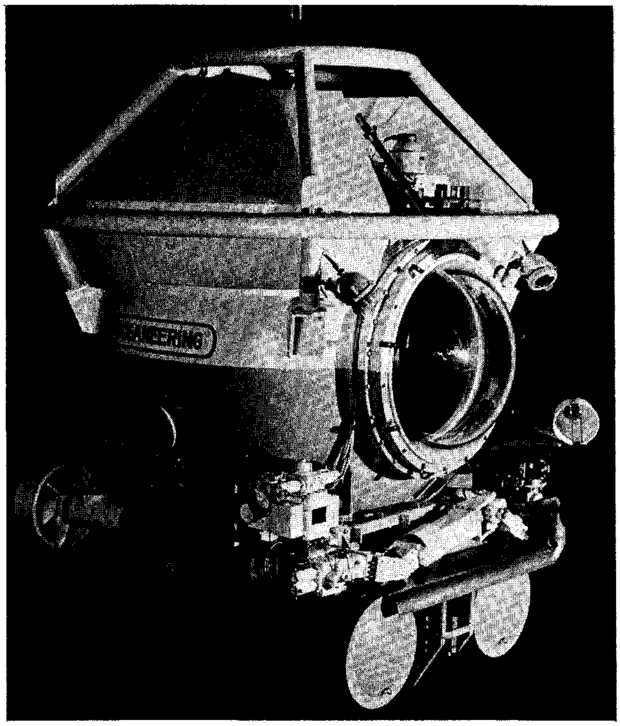

Fig. 11 Example of a statically-supported submersible. The manned Observation Manipulator Bell (OMB) was built by Perry Oceanographics and is used in support of undersea exploration and work missions. The bell can be made neutrally as well as negatively buoyant and is self-propelled. (Courtesy of Perry Offshore, Inc.)

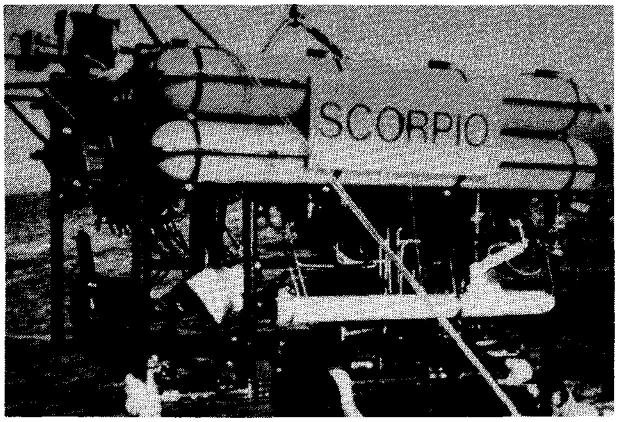

Fig. 12 Example of a thrust-supported submersible. The unmanned, remote-operated vehicle (ROV) *Scorpio* was built by Ametek-Straza for underwater work missions. It is controlled from the surface support ship through an umbilical cable to the vehicle. (Courtesy of Ametek-Straza)

through it at constant-depth with constant-speed, straight-line motion. The submersible, using its control systems, must be able to attain all operating conditions comprising its dive profile, including its design condition, in order to accomplish the underwater mission. Departure from the design condition to other operating conditions is caused by the imposition of static and dynamic forces/moments from various sources on the basic force system. Consider, for example, a submersible in the design condition of neutral buoyancy with zero trim and list. Other operating conditions, for instance, may require the vehicle to "sit hard" on the bottom for coring tasks or to depart from zero trim in facilitating manipulator access to a work object. In general, these departures from the design condition necessitate superimposing static and dynamic control forces/moments of weight, displacement, and thrust on the basic force system. Yet other operating conditions may involve mobility and maneuverability necessitating the imposition of control and induced forces/moments of thrust, lift, drag, and inertia on the basic force system.

Following is a discussion of the various design-condition categories of submersibles. The presentation is simplified by using the x,y,z body axes and by declaring, for this purpose, that the $+z$ direction is vertically downward in the $+z_0$ direction.

Neutral-buoyancy submersibles—This design condition, as indicated, is the "baseline condition" in the sense that principles discussed in this chapter relating to it can be adapted for use in the designs of submersibles with other design conditions. As shown in Fig. 14, the only forces acting on the neutral-buoyancy submersible are the total submerged weight W and displacement Δ. Equations (2) for the design condition become

$$X = Y = 0$$

$$Z = W - \Delta = 0$$

$$K = M = N = 0 \qquad (2a)$$

The first force equation (2a) merely states that the resultants of x and y components of hydrostatic pressure acting on the submersible are zero. The second force equation (2a) states that the resultant of W and Δ, buoyancy F_B, is zero. As noted, it is this zero value of F_B which gives rise to the term *neutral buoyancy* to identify this category. The moment equation (2a) indicates that W and Δ must be collinear forces. Further, the design condition requires the forces' common line of action be normal to the submersible's x–y plane for zero trim and list. In summary, then, *the basic force system consists of two equal and opposite, collinear forces,* W *and* Δ, *whose line of action is normal to the submersible's* x–y *plane.* As will be seen, the latter part of this statement, together with statical stability requirements, fix the relative locations of the centers of gravity, G, and displacement[6] B in the vehicle.

Subdivisions of neutral-buoyancy submersibles include *reserve-displacement* and *zero-reserve-displacement* vehicles, terms which refer to predive, surface conditions. The reserve-displacement submersible floats on the surface with a certain amount of potential, displacement-producing volume above the waterline, thus providing freeboard, which elevates the access hatch above this waterline

[6]The "center of displacement" and "reserve displacement" are more often called "center of buoyancy" and "reserve buoyancy." However, consistent with defining "buoyancy" as the resultant of W and Δ and not a term synonymous with displacement, terminology in the text will be used.

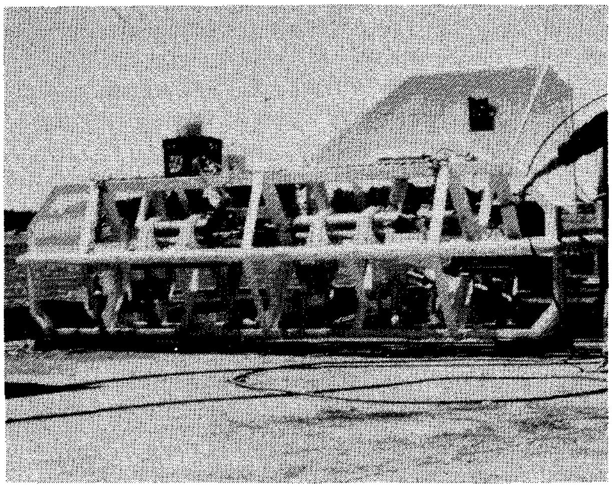

Fig. 13 Example of a lift-supported submersible. The unmanned, towed vehicle *Argo* was developed by the Woods Hole Oceanographic Institute for use in the visual and acoustical surveying of large areas of the seafloor. (Courtesy of WHOI)

and facilitates surface operations. This volume is called *volume of reserve displacement* $\overline{V}_{RD}$ and is usually given as a percentage of the normal surface displacement volume $\overline{V}_N$, a term to be discussed later. Its value for submersibles is about 5 to 12 percent of $\overline{V}_N$. As will be seen, designs of this type must provide for main ballast tanks (MBT) whose net volumes are equal to $\overline{V}_{RD}$. If so-called descent weights are not used, the MBT, when completely flooded, will cause the submersible to just submerge in an "awash" design condition.

Zero-reserve-displacement submersibles have no $\overline{V}_{RD}$ above the waterline. They float either "awash" at the waterline or with negative buoyancy, in which case they must be restrained from descent by external forces from Source 5, usually cable tension. Because main ballast tanks are not required, the size and weight of the vehicle are reduced, but surface operations, primarily launch and retrieval, are more complicated.

Neutral-buoyancy submersibles are particularly suitable for missions requiring a high degree of self-mobility in three dimensions with either unrestrained or restrained operating radii. As noted, unrestrained submersibles have complete freedom of movement: their operating radii are determined by design factors affecting submerged range and endurance such as on-board energy storage and life-support (for manned submersibles) capacities. Restrained submersibles retain their mobility in three dimensions, but their operating radii are limited by tethers with other *I* systems. For this category, it is assumed that the tethers are relatively light or buoyed in such a way that they exert negligible Source 5 forces of tension on the submersible in its design condition. Some design trade-offs between unrestrained and restrained submersibles are, respectively, a much greater operating radius and freedom of movement without danger of tether entanglement or breakage versus simplification of the design, increased endurance (if power and life support are transmitted by tether), and improved "hard-wire" communications.

Neutral-buoyancy submersibles are inherently more easily operated by on-board human control than by remote control. Consequently, this category is composed largely of "free-swimming," manned submersibles, and incidentally, all naval submarines.

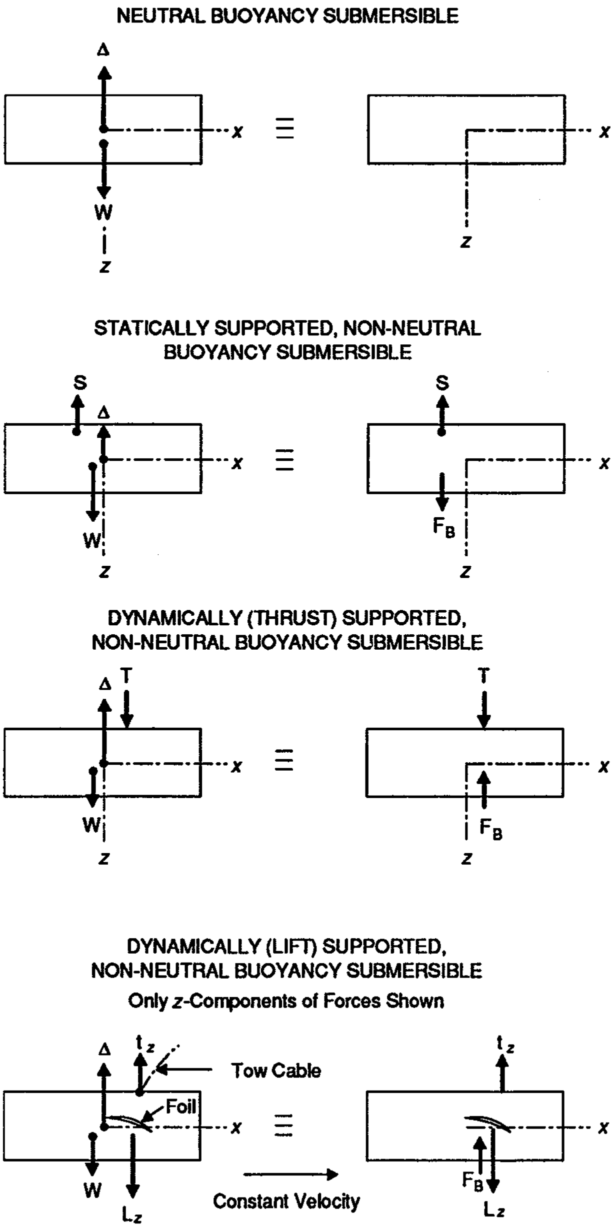

Fig. 14 Hydromechanical categories of submersibles based on design conditions

Non-neutral-buoyancy submersibles—Figure 14 shows the statically supported, non-neutral-buoyancy submersible being acted on by the total submerged weight W and displacement Δ as well as by a Source 5 force such as cable tension or seafloor reaction forces (cable tension is shown in the figure). Equations (2) for the design condition become:

$$X = Y = 0$$

$$Z = W - \Delta \pm S = \mp F_B \pm S = 0$$

$$K = M = N = 0 \qquad (2b)$$

where S is the vertically directed resultant of the static Source 5 forces and F_B is buoyancy. The first force equation (2*b*) states that (1) the horizontal plane components of hydrostatic pressure balance and (2) the resultant of any S forces in this plane must be zero. The second force equation (2*b*) indicates an inequality in W and Δ, leading to a non-zero value[7]

[7] As noted, popular usage has led to labelling upward-directed buoyancy F_B as positive, causing the submersible to rise, despite the fact that "upward" is in the negative z_0 direction as earth axes are customarily oriented. Thus, to be consistent, positive buoyancy should be $-F_B$, an anomaly which is confusing until the direction convention is understood.

of F_B in the positive or negative sense. This non-zero value leads to the use of the term *non-neutral buoyancy* to identify the second major design condition of this category. The alternate force equation (2*b*) shows that the three-force system can be reduced to a two-force system composed of F_B and *S*, the design condition requiring that *S* be equal and opposite to F_B. The moment equations (2*b*) indicate that F_B and *S* are collinear forces for the resultant moments about all axes to be zero. Again, the design condition dictates that the common line of action be normal to the submersible's *x–y* plane for zero trim/list. In summary, then, the *basic force system consists of two equal and opposite, collinear forces of* F_B *and* S, *whose line of action is normal to the submersible's* x–y *plane*. The latter part of this statement, together with stability requirements, fix the locations of the centers of gravity and displacement and the point of *S* force application relative to each other. As will be seen, stability considerations for this and the neutral-buoyancy submersibles differ significantly.

Submersibles of this category may be used for missions requiring high payload capacity, "unlimited" power, and good communications, while needing minimal three-dimensional mobility. Vehicles of this category are restrained by tethers, which connect them with other *I* systems and which exert *S* forces on them, such as cables[8] or pipe-strings,[9] which are capable of lowering and raising submersibles and their payloads. Diving bells, rescue chambers, cages, or "garages" for other vehicles and heavy-lift vehicles are examples of manned and unmanned submersibles statically supported by tension/forces. Seafloor crawlers, such as trenchers and earthmovers, are examples of these vehicles statically supported by seafloor reaction *S* forces.

Principles involved in the design of dynamically (thrust) supported, non-neutral-buoyancy submersibles are similar to those for statically supported, non-neutral-buoyancy submersibles except that support is provided by hydrodynamic, thrust forces *T* instead of static *S* forces. Figure 14 shows this category being acted on by the total submerged weight *W* and displacement Δ as well as by this *T* force. Equations (2) for the design condition become

$$X = Y = 0$$

$$Z = W - \Delta + T = 0 \quad \text{or} \quad -F_B + T = 0$$

$$K = M = N = 0 \qquad (2c)$$

where *T*, more specifically, is the resultant of vertically downward-directed thrust forces produced by one or more of the vehicle's thrusters. The first force equation (2*c*) again notes that the horizontal plane components of hydrostatic pressure and thrust forces must be zero for this condition. The second force equation (2*c*) indicates an inequality in *W* and Δ, again leading to a non-zero value of F_B or a non-neutral-buoyancy condition. In this instance, Δ is always greater than *W*, producing an upward F_B, which must be balanced by the downward-directed *T* force. This situation invokes the "fail-safe" design philosophy wherein positive F_B will cause the submersible to surface in the event of power failure or thruster malfunction. The alternate force equation (2*c*) again shows that the three-force system can be reduced to a two-force system composed of F_B and *T*, the design condition requiring that F_B and *T* be equal and opposite forces. The moment equation (2*c*) indicates that these forces are collnear forces and that, for zero trim/list, their common line of action must be normal to the submersible's *x–y* plane. In summary, *the basic force system consists of two equal and opposite, collinear forces of* F_B *and* T, *whose line of action is normal to the vehicle's* x–y *plane*. As before, the later part of this statement, together with stability requirements, fix the relative locations of the centers of gravity and displacement as well as those of the thrusters producing *T*.

Thrust-supported submersibles may be either restrained or unrestrained, with advantages and disadvantages paralleling those noted for neutral-buoyancy vehicles of these types. Restrained, thrust-supported vehicles have tethers which are assumed to be light or buoyed in such a manner so as to produce no *S* forces in the design condition. They are, in general, more easily controlled from remote stations than neutral-buoyancy submersibles because control is greatly facilitated by "hard-wire" communications via the tether. Thus, unmanned, tethered submersibles, which are generally called remote-operated or -controlled vehicles (ROV or RCV), make up the vast majority of this category.

The unrestrained thrust-supported category includes the latest development in underwater vehicle systems—unmanned, untethered submersibles. Severing the tether has made remote control much less feasible because of difficulties with through-water communications. Consequently, it is necessary to place a high degree of "self-control" on board exercised through sophisticated microprocessors/computers, sensors and associated software. These vehicles hold great promise for the future because the elimination of both man and tether gives them the potential for use in advanced ocean engineering and scientific systems.

A particular attribute of this category is the ability of these vehicles to maintain precise altitude control in near-bottom movement and to avoid creating "silt

[8] Cables used for lowering/raising submersibles must have strength elements built into them capable of handling the static loads F_B plus dynamic loads due to changes in motion. A single cable, or bundle of cables, may also furnish other services such as transmitting power and breathing gases and providing for "hard-wire" communications. The cost and difficulty of handling cables increase dramatically as the numbers of services provided are increased.

[9] A procedure resembling oil/gas drilling, except for pipe rotation, lowers/raises submersibles by connecting/disconnecting lengths of pipe with the vehicle being attached to the initial length. The overall assembly of pipe lengths is called a "pipe-string."

clouds" by using downward-directed T created by upward-directed water particles through the thrusters. In this regard, neutral-buoyancy submersibles often depart from the design condition to one of positive buoyancy using T forces to maintain fine altitude control.

Dynamically (lift) supported, non-neutral-buoyancy submersibles are the only ones for which the design condition is based on dynamic, rather than static, equilibrium (Fig. 14), one of "level flight" with straight-line, constant-speed motion. This motion, of course, is necessary to generate dynamic lift L forces created by waterflow over fixed fins, movable control surfaces, or the submersible's body itself. This category also contains both unrestrained and restrained types, that is, self-propelled and towed vehicles. Both types are acted on by the total submerged weight W and displacement Δ, the horizontal plane components of pressure p, resistance R, lift L, and lift-associated drag D. Additionally, self-propelled and towed vehicles are acted on by thrust T and towing-cable tension t, respectively. For simplicity, Fig. 14 shows only the z-axis-directed forces acting as they are sufficient to convey the concept of lift-supported vehicles as a hydromechanical category. Equations (2) for the design condition of this category become

$$X = Y = 0$$
$$Z = W - \Delta \pm L_z \mp D_z - t_z = \mp F_B \pm L_z \mp D_z - t_z = 0$$
$$K = M = N = 0 \tag{2d}$$

where L_z, D_z, and t_z are the z-direction components of L, D, and t in the case of towed vehicles. Once again, the first equation (2*d*) indicates that the sum of the horizontal components of all forces acting on the vehicle must be zero for dynamic equilibrium. The second equation (2*d*) indicates an inequality in W and Δ, resulting in a $\pm F_B$, which must be balanced by the z-components of the forces indicated. The design condition also dictates that the list be zero, but, unlike the other categories, here the trim may or may not be zero, depending on whether or not inclination of the vehicle itself is used to create lift. Thus, this category's *basic force system is coplanar with reference to the submersible's* x-z *plane; the* x- *and* z-*direction resultants of forces acting are zero, with individual forces being located so as to provide the desired zero or non-zero trim at specified forward speeds.*

Submersibles of this category are unmanned[10] because they are, for the most part, towed vehicles with performance requirements specifying level-flight at specified depths over large areas of the seafloor. Practically all of these vehicles possess positive $-F_B$ when they are held at the desired depths by downward-directed $+L$ developed at specific towing speeds, surfacing of their own accord when the towing speed approaches zero. Examples include a wide variety of towed sensor platforms often called *fish*. A prime example of the self-propelled vehicle in this category is a torpedo, which is designed with negative buoyancy $+F_B$ to cause it to sink at the end of its run, with speed and lift $-L$ reduced to zero, if it fails to strike its target or other object.

Category combinations—Submersible systems are sometimes composed of two vehicles of different design-condition types. A major reason for using these systems is to partially or totally overcome problems associated with heavy-tethered, unmanned submersibles, problems such as cable drag, restricted maneuverability, and, in some instances, transference of the support ship's motion to the submersible. An example of such a combination is a statically supported vehicle, suspended from the surface by a heavy cable, which serves as a "garage" for a smaller neutrally buoyant or thrust-supported vehicle radiating from it on a light tether. In another example, a towed lift-supported vehicle, usually called a *depressor*, serves as the "garage" for the smaller vehicle radiating from it on a light tether. A final example involves remote launching/retrieval of manned, neutral-buoyancy submersibles by unmanned, statically supported vehicles, the latter being lowered/raised by motion-compensated tether systems attaching it to the surface support ship. This system, as noted, permits these critical operations to take place at depths at which surface-wave action is negligible.

2.3.2 *Speed Categories*

Speed provides another way of hydromechanically categorizing submersibles since the performance requirements for both forward, x-direction speeds and vertical, z_0-direction speeds have major impacts on the hydromechanical aspects of design. These speed categories may be outlined, quite simply, as

1. Forward speed
   a. low
   b. high
2. Vertical speed
   a. low
   b. high

However, it is not as simple to state precisely at what speeds transitions occur. Actually, there is a range of intermediate speeds in which the impact of speed on design must be weighed very carefully. Perhaps the best criterion for distinguishing between the low- and high-speed ranges is based on the fact that the magnitudes of lift L and drag D or resistance R are functions of the square of the speed. Consequently, low- and high-speed ranges may be considered, respectively, as those

[10] In some early submarines (circa 1898) designed on the positive, rather than neutral, buoyancy philosophy, downward-directed lift was created by planes required to keep them submerged at ordered depths.

in which these Source 4 forces/moments have minor and major impacts on design.

Forward speeds—The great majority of submersibles from all design condition groups, except the lift-supported vehicles, also belong in the *low-forward-speed* category, their maximum speeds being but a few knots. There are many and diverse reasons for this fact, including on-board energy limitations, speed limitations imposed by sensor capabilities, and navigational difficulties encountered at higher speeds. Consequently, most mission tasks are limited to "point" locations and relatively small areas.

Some major design considerations include geometry, envelope, appendages, and maneuvering systems. Since Source 4 forces/moments have minor impact, the geometry (shape and dimensions) is not required to be streamlined and the envelope may be either open-framed or enclosed. The latter, in this instance, provides a fairing surface to reduce possibilities of entanglement and to protect internal systems. Appendages, such as manipulators, sensors, and lights, may extend beyond the envelope without great concern for hydrodynamic considerations. Maneuvering systems must employ thrust rather than lift-producing devices.

There are some submersibles in the design-condition groups and of the unrestrained, or untethered, type which may also be placed at least in the lower part of the *high-forward-speed* category, with speeds exceeding three or so knots. Use of these vehicles may be expected to increase as previously noted limitations and difficulties are eased, thus enabling these submersibles to undertake mission tasks travelling long distances or across large areas of the seafloor.

Design considerations previously noted for low-speed submersibles are now influenced by the increasingly strong impact of Source 4 forces/moments as the speed requirement increases. The vehicle's geometry should begin to approach the streamlined form, and an enclosed envelope should be used. Great care should be given to appendage design and location, with provisions made for retracting these into the envelope where possible. Maneuvering systems must employ lift-producing devices. Thrust-producing devices should be included only if the submersible is expected to do low-speed maneuvering. In addition, as indicated, motion stability must be considered.

Vertical speeds—Vertical speed requirements, in general, are related to the maximum operating depth requirements. The need for higher vertical speeds increases with an increase in depth to reduce the necessary but unproductive transit time between surface and seafloor. These speeds are functions of descent/ascent paths and forces involved. Four paths may be followed: vertical with the x-axis remaining more or less horizontal; inclined with the x-axis inclined at some angle with the horizontal; inclined descent and vertical ascent; and spiral descent/ascent paths. Motions along these paths are caused using thrust, lift, descent/ascent weights, or combinations of these forces.

The hydromechanical category of *low-vertical-speed* submersibles is associated, in general, with the shallower maximum operating depth submersibles although many current vehicles with low vertical speeds go much deeper. Applying the criterion, then, this category has speeds in the direction of the descent/ascent path limited to one or two knots. A streamlined geometry in the direction of motion is not particularly important and relatively small motivating forces are involved. Numerous shallow and even deep-diving submersibles are in this category.

The *high-vertical-speed* category is associated with mid to great maximum operating depth submersibles. Ideally, these vehicles should have descent/ascent path velocities well in excess of two knots, with Source 4 forces/moments becoming significant. Consequently, streamlining the vehicle in the direction of motion becomes an important consideration, together with providing relatively large motivating forces. Some unmanned submersible designs, in both the high-forward and -vertical speed categories, orient the x-axis vertically for descent/ascent operations.

2.3.3 Maneuvering Categories

Maneuvering capabilities, as set forth in performance requirements, also have a major impact on the hydromechanical aspects of design and, hence, provide a third way of categorizing submersibles. In general, maneuvering may be defined as the controlled change or retention of a body's position or direction of motion and its speed in that direction. Thus, maneuvering categories can be identified in the following terms:

1. Position-changing
2. Position-keeping
3. Path-changing
4. Path-keeping
5. Speed-changing
6. Speed-keeping

If the body is a submersible, speed-changing capabilities, involving acceleration/deceleration rates, are usually not specified in its performance requirements, while speed-keeping capabilities are addressed in speed-power rather than maneuvering investigations per se. Consequently, only the first four categories will be considered.

Position-changing—Performance requirements for this category of submersible emphasize the need to change the vehicle's spatial orientation at zero or very low forward speeds, for example to trim, list, or move sidewise (sway) to facilitate accomplishing underwater tasks. These requirements may be stated in terms of the aforementioned degrees of freedom, or the degrees of controlled movement along or about the x,y,z axes. The many submersibles of this category usually have four to six degrees of freedom.

Design considerations for this category focus on characteristics enabling the submersible to attain a new position, or orientation, easily and rapidly. Briefly, these considerations alone tend to favor the use of relatively short vehicles with open-frame envelopes having low, but safe, metacentric stability and adequate means of achieving the desired changes by use of static or dynamic forces/moments or both. Dynamic forces/moments, as noted for zero and very low speeds, must be produced by thrusting devices, which are located and oriented so as to produce maximum control moments while minimizing or eliminating cross-coupling effects in which undesired motions are created in the process of generating the desired motions.

Position-keeping—Performance requirements for this category of submersible stress the need to maintain specified spatial orientations, usually in the horizontal plane, at zero and very low forward speeds and in the presence of external forces/moments such as those of current fields and reactive torques. These requirements are imposed for the purpose of providing a stable platform for mission tasks needing precise orientation in space. This category may also possess four to six degrees of freedom, in this case, to provide control forces/moments resisting the efforts of external forces moments to alter the vehicle's position.

Design considerations of this category are concerned with characteristics enabling the submersible to be held in position with relative ease. They are similar to those of "position changing" submersibles except that as high a metacentric stability as possible is desirable to resist inclinations about the x- and y-axes.

Path-changing—Performance requirements for this category of submersible concern the need for expeditiously altering the vehicle's path and often stipulate "maneuvering dimensions," for example the diameters of turning circles. These requirements are usually associated with submersibles having military missions, including naval submarines, but may also apply to those having nonmilitary missions.

Design considerations for this category focus on characteristics enabling the submersible to achieve a new path easily and rapidly and are those associated with controls-fixed, straight-line motion stability. They relate to the vehicle's bare-hull size and geometry, the geometries and locations of all appendages and metacentric stability, and they affect path-changing ability through their effects on hydrodynamic derivatives of the equations of motion. A discussion of these design characteristics requires familiarity with these derivatives and the equations of motion, which are presented in Section 4 of this chapter.

Path-keeping—Performance requirements for this category of submersible emphasize the need for the vehicle to maintain its path of motion in performing its underwater mission. Such missions, for example, might involve surveying large areas of the sea-floor, using straight-line grid patterns, or the transportation of items over relatively long distances.[11]

Design considerations for this category concern characteristics allowing the submersible to maintain a specified path with ease and with minimum course error. These are the characteristics also associated with controls-fixed, straight-line stability in the horizontal plane and controls-fixed, directional stability in the vertical plane. Comments regarding the characteristics of path-changing submersibles also apply here. One descriptive differentiation between path-changing and path-keeping vehicles may be made at this point. Both vehicles may be likened to arrows, with the path-keeping arrow having more "tail feathers" to increase its motion stability. Consequently, path-keeping submersibles have fixed-fin or "deadwood" areas or both at the stern, whereas path-changing submersibles do not have these surfaces. In addition, metacentric stability should be larger for path-keeping than for path-changing submersibles.

2.4 Operating Conditions

The design condition has been singled out as the specific operating condition which is the fundamental condition from the design perspective and which provides one means of identifying hydromechanical categories of submersibles. However, it was noted that it is but one of several static and dynamic conditions the submersible must achieve in successfully performing mission tasks. A complete listing of these conditions, excluding launch/recovery operations, forms a *dive profile*. This profile is outlined for a neutral-buoyancy, unrestrained submersible and many elements of it are also applicable to other design condition categories:

1. Surface, normal diving trim: ready to dive in all respects; a static condition
2. Transition, diving: through air/water interface on diving; a dynamic condition
3. Submerged
   a. Descent: a dynamic condition
   b. Design condition: a static condition
   c. Non-design conditions: static and dynamic conditions
   d. Ascent: a dynamic condition
      (1) Normal: by use of regular control systems

[11] Large submarine tankers have been designed to transport oil long distances under arctic ice and in open water.

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(2) Emergency: by use of emergency control systems
4. Transition, surfacing: through air/water interface on surfacing; a dynamic condition
5. Surface, post-dive: a static condition

2.5 Control Systems

Control systems must be incorporated into the designs of submersibles enabling them to *change* safely and effectively from one operating condition to another or to *retain* a particular operating condition in the presence of forces/moments attempting to change it.

One way of viewing an overall control system is shown in Fig. 15, its major components being briefly outlined below:

1. Pilot (operator)/autopilot gives control commands (by pulling a toggle switch, pushing a button, etc.) to the command transfer component of the system and also enters desired response of submersible into display-response comparison component.

2. Command transfer component transfers the commands mechanically, electrically, hydraulically, or pneumatically to the actuator.

3. Actuator component, such as thruster units, movable control surfaces, and variable ballast pumps, puts the commands into action, generating control forces/moments causing the submersible to respond in a certain manner (to descend, trim, yaw, etc.)

4. Display-response comparisons component compares information received on the actual response with the desired response and provides response error feed-back to the pilot/autopilot for corrective action. It also displays actuator information such as rudder and plane angles, propeller rpm, etc.

Control systems may be divided into *static* and *dynamic systems* from the perspective of hydromechanics. Examples of static control systems, which employ static forces/moments of weight and displacement in performing their functions, are main ballast water, variable displacement, and trim and list weights. Dynamic control systems, on the other hand, use hydrodynamic forces/moments of thrust and lift in accomplishing their functions, thrust being generated by the main propulsion and maneuvering systems' thrusters while lift is created by the latter system's movable control surfaces. With reference to the x,y body axes and the z_0 earth axis, static control systems are effective in three degrees of freedom, causing or resisting rotation about the x-axis (list) and y-axis (trim) and translation in the $\pm z_0$ direction (descent/ascent). Dynamic systems, however, are effective in all six degrees of freedom, causing or resisting rotation about and translation along the three body axes. The number of degrees of freedom incorporated into a submersible's design is dependent upon the mission's maneuvering requirements.

Static and dynamic control systems include the following individual systems. It must be emphasized that submersible designs may employ all, some, or combinations of these systems in a great variety of approaches to meeting the mission's maneuvering requirements.

1. Static control systems
   a. Variable ballast system

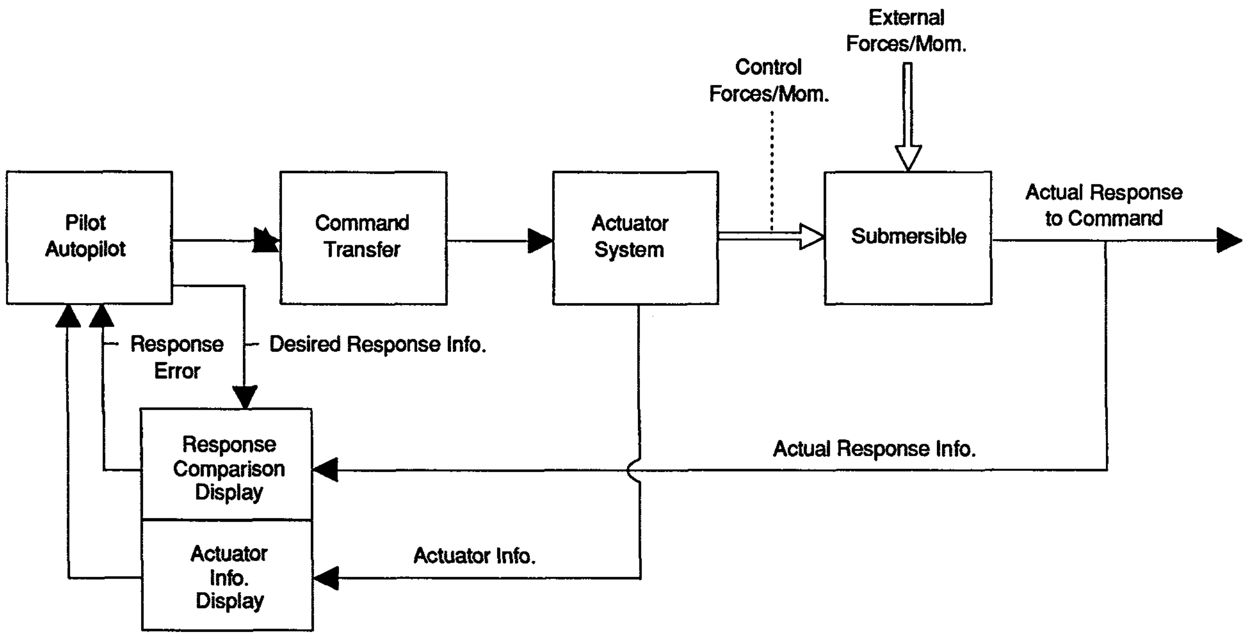

Fig. 15 Overall control system

Chapter VI
Structural Principles

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1. Overview

STRUCTURAL DESIGN must be considered early in a submersible vehicle development program soon after general mission characteristics have been outlined, such as operating depth, speed, manning, endurance, and payload. Size is obviously the predominant consideration in hull design, and is influenced by these variables. The structural design of the hull may be considered in two broad divisions: the pressure hull and the exostructure, the latter commonly called the nonpressure hull or outer hull. The design process generally differs for these because of their differing functions and design loads, but, of course, there is an interrelationship, and the design of each influences the other, the pressure hull being paramount. Various failure criteria including yielding, instability, fatigue, fracture, creep, and corrosion greatly affect the structural design.

Pressure hulls usually are fabricated from steel, but other metals (aluminum and titanium) and nonmetals, including glasses, acrylics, glass-reinforced plastics (GRP), and fiber-reinforced plastics (FRP) also have been used. Hydrodynamic form, internal and external arrangements, material availability, costs, and ease of fabrication play important roles in pressure hull design. The operating depth is a major factor since the structural weight is a relatively large percentage of the total displaced weight of deep-diving vehicles. For relatively shallow depths (commonly considered to be less than 610 m (2000 ft), and generally coinciding with the accepted norm for continental shelves), pressure hull weight is of lesser concern. Various pressure vessel configurations have been designed and built for submersibles, the predominant ones being stiffened cylinders with hemispherical heads and single spheres. Other shapes have been built, including multiple intersecting spheres and oblate spheroids. The common basis for evaluating the structural efficiencies of the various hulls is the buoyancy factor, defined as the hull weight to displaced water weight ratio. Buoyancy ratios for various hull designs can be found in many texts and reports [1,2].

The single unstiffened sphere frequently has been used for deep operating, small submersibles because of its attractively low buoyancy factor. For an ideal sphere with no stiffeners or reinforcements the buoyancy factor is

$$B = \frac{3}{2}\left(\frac{\rho_s}{\rho_w}\right)\left(\frac{p}{\sigma_{max}}\right) \tag{1}$$

where

ρ_s, ρ_w = densities of the shell and water, respectively,

p = hydrostatic pressure, and

σ_{max} = maximum stress in an ideal sphere.

By comparison, the buoyancy factor for an ideal cylinder with no stiffeners or reinforcements is

$$B = 2\left(\frac{\rho_s}{\rho_w}\right)\left(\frac{p}{\sigma_{max}}\right) \tag{2}$$

Thus, in the ideal case, the single sphere has a 33 percent advantage in structural efficiency. The weight advantage of the single sphere is even greater when compared with a cylinder with stiffeners and reinforcements. However, when total weights are compared, considering hydrodynamic form and internal and external arrangements, the net weight advantage of the single-sphere vessel declines. Another advantage of the single sphere is that it allows for a fairly short and acceptably light vehicle that can be handled by a single shipboard crane on a tender. Thus, many of the shallower depth submersibles are of single-sphere design, with a crew of just two or three. Stiffened spheres have been studied, but they afford no appreciable structural cost savings or practical advantages. However, multiple intersecting spheres stiffened at their intersections have very attractive features. The buoyancy factor is low, the shape is more attractive than a single sphere for hydrodynamic fairing, and the internal and external arrangements are more favorable. Structural details of this concept will be covered in this chapter, since multiple spheres are becoming more popular.

The stiffened cylinder generally permits superior hydrodynamic form, better internal arrangements, lighter exostructures, and lower fabrication costs. It is also less affected by initial geometric imperfections than shells with compound radii of curvature. The ring-stiffened cylinder is used less frequently for deeper operating depths, because of its less favorable buoyancy factor. However, there are situations when material selection and fabrication techniques can justify the use of ring-stiffened cylinders down to 6100 m (20 000 ft).

Other pressure hull shapes have been used, but for various reasons these are not as efficient as stiffened cylinders, spheres, and multiple spheres. Conical shapes have been used as transition structures between cylinders and hemispherical, elliptical, or flat end closures, but this leads to inefficient weight because of high transition stresses. Oblate spheroids, the so-called diving saucers used primarily for observation, are not structurally efficient, but are suitable for shallow depths. Relative to a sphere, this shape improves hydrodynamic performance in the horizontal plane. Prolate spheroids have attractive buoyancy factors and hydrodynamic form, but are difficult to fabricate. Penetrations, except at the poles, are especially costly and difficult to fabricate.

Penetrations are a major design consideration. There are local stress variations around the penetrations that affect the design depth limitations, as well as the cyclic life of the pressure vessel. Small penetrations affect only local stresses, but large penetrations could affect the design collapse depth. An efficient penetration design cannot be achieved simply by compensating for the shell material removed, as is the case for penetrations in less efficient pressure vessels. Detailed design and analysis techniques must be used to achieve a balanced design (if possible), avoiding both over-reinforcement and under-reinforcement.

For deep-depth submersibles, major penetrations are limited to access hatches, electrical connections, and viewports. Hydraulic penetrations, stuffing tubes, and shafts normally are avoided in order to minimize any possibility of leaking.

Access hatches can be categorized as being either of the seat type or plug type. The seat type of hatch normally is used in relatively shallow depth submersibles, since it is more easily fabricated and costs less; however, it is somewhat heavier than a plug hatch. The plug hatch generally is used for deep-diving submersibles so as to save weight. It is used normally in shells of uniform in-plane stresses, such as spheres, since the stiffening ring around the hatch in a sphere is of uniform thickness. The plug hatch supports the shell's in-plane loading. The reinforcement ring weight for this type of hatch is appreciably less than for the reinforcement of a seat hatch.

Viewports for submersibles are mostly of three designs: flat plates, conical frustums, and spherical domes. The flat plate designs are the least costly, but are restricted to shallow depths. Spherical domes offer the widest view, but also are normally restricted to shallow depths because of their relatively low strength. For deep depths, conical frustums made of acrylic plastics are normally used because of their strength.

Electrical penetrators must have a watertight juncture at the hull. They are normally of a double seal design, so that if a leak occurs in the outboard cabling or one of the seals fail, no water can penetrate the pressure hull boundary. If the outboard cable fails, the penetrator itself does not have to be repaired or replaced, and the hull is not affected.

The pressure hull, because of its thick scantlings, high-performance material, and tight fabrication tolerances, is relatively costly and constitutes a large part of the total vehicle weight. Large pressure-resistant structures become especially impractical for deep-diving submarine craft. For these reasons, the pressure hull size is minimized, sufficient only to accommodate personnel, any mission equipment that must be inside, system controls, and perhaps certain emergency life support equipment. All other components and subsystems are placed outside the pressure hull and, for the most part, within the supportive and protective confines of an exostructure. The exostructure normally forms the entire external boundary of the vehicle, with the pressure hull enclosed and supported within that structure. Less commonly, the pressure hull forms the vehicle's external boundary for some part of the length with the nonpressure hull attached as bow or stern appendages or both. The purposes of an exostructure are fivefold: to provide a faired hydrodynamic form for the vehicle; to minimize potential entanglement with underwater objects; to serve as integral support for the pressure hull and various systems and components; in some cases, to hold main ballast water or other ballasting liquid or solids; and, in some cases, to provide extra freeboard and a weather deck.

Exostructure design evolves from efforts that begin with the establishment of mission requirements and constraints, evaluation of system requirements, and development of pressure hull form and size. The equipment and components of the various vehicle systems (for example, propulsion motor, batteries, stability tanks, oceanographic equipment) are then arranged in such a manner as to distribute vehicle weight properly for surfaced and submerged stability and, of course, to fulfill functional requirements. Once the various system component locations and spatial requirements are established, the exostructure envelope begins to take form. The full diameter of the exostructure usually is determined by the diameter of the pressure hull, and the depth of structure required to support it. The exostructure length derives from arrangement and hydrostatic considerations, and from other constraints such as transport limitations. An enveloping faired surface is then developed with sufficient internal space allowed for support structure, external systems and equipment, construction access, and maintenance. Other considerations also can influence the size and shape of the exostructure. It might be desired, for instance, that the exostructure provide appreciable reserve buoyancy for enhanced safety when the submersible is surfaced in rough water.

Hull forms for submersibles have varied widely, primarily because speeds are so slow that form has little effect on drag, and hence other considerations govern. Vehicle form generally can be approximated by either a cylinder or series of shallow-angle truncated cones. These shapes have certain advantages over more complicated ones: they are efficient structural forms which can withstand the wide range of loading conditions encountered by a submersible vehicle; the axisymmetric form facilitates the arrangement, attachment, and servicing of enclosed components and equipments; adequate hydrodynamic and power/speed characteristics are readily achievable; and lastly, such axisymmetric structures in general require less material, are relatively easy to construct, and therefore are usually economically advantageous.

Achieving a least-cost structure is more difficult and complex than achieving a least-weight structure since this process introduces factors that vary from time to time and place to place. It should be recognized that the term "minimum cost structure" will often be interpreted differently depending on whose costs are involved. To the builder, it usually means structures that can be fabricated and installed at lowest cost, and that can mean different designs for different builders. To the design agent it may mean the same thing, but it also may mean structures that can be designed, analyzed, and have drawings produced for the least engineering and drafting costs. The owner/operator might prefer a third interpretation involving service life ownership and operating costs, which are not necessarily the same as least design and acquisition costs.

It is highly improbable that each of these objectives can be optimized collectively; consequently, development of a feasible structural system invariably will be based on compromises and tradeoffs of certain design features. For instance, structural designs optimized for minimal weight rarely lend themselves to such requirements as ease of fabrication and repair, accessibility, arrangements, or lowest cost. In some structural elements, stiffness is the critical factor that determines the thickness or material selected, and a high strength-to-density ratio becomes less important.

Despite the importance of minimum structural weight (which translates into maximum payload for a given displacement and form), the influence of cost and schedule constraints on the design are considerable and, in fact, cannot be overstated. The demands of tight design and construction schedules, with the attendant lead times and associated procurement costs for special materials and material shapes in small quantities, serve to drive the design to a simpler, lower-cost form, and militate against any lengthy investigative pursuit of the "optimum" structure. For these reasons, what may be called the traditional design process frequently is followed, in which alternate structural configurations in conjunction with various alternate materials are examined, basically using trial and error procedures, and comparatively evaluated. Stated differently, the process consists of three distinct steps: enumerate options, assess each, and finally, select the best. This process has the semblance of being somewhat "non-systematic," but when applied by experienced designers, it actually embodies many higher order selections and shortcuts by which means effective designs can be achieved economically. This traditional design method is well suited to the purpose and intent of this chapter, namely, to show the steps by which a submarine hull structure is designed, to list the many requirements that must be met, and to provide guidance and fundamental technical data that will be useful to those engaged in submersible design.

2. Pressure Hull Design

Pressure hulls for submersibles are stiffened, reinforced (around penetrations), or both. Thus, shell theory, considering normal forces, bending moments, and shearing forces, is required for their design. If the pressure hull is considered as a thin shell having a thickness-to-radius ratio t/R of less than 0.05, then the buoyancy factor B can be approximated simply from

$$B = \left(\frac{A_h t}{V_h}\right)\left(\frac{\rho_s}{\rho_w}\right)\left(1 + \frac{W_s}{A_h t \rho_s}\right) \tag{3}$$

where

A_h = outer surface area of the pressure hull shell,
t = shell thickness,
V_h = volume of the pressure hull shell,
ρ_s, ρ_w = densities of the pressure hull shell and water, respectively, and
W_s = weight of stiffeners and reinforcements.

In all cases, the actual buoyancy factor will be less than this approximate buoyancy factor [equation (3)]. Once the form has been selected, the area A_h and volume V_h are known. When the material has been selected, the material density ρ_s is known. This leaves only the shell thickness t and the weight of the stiffeners and reinforcements W_s as variables affecting the buoyancy factor. The shell thickness will be a function of hull radius, the stress at the design operating depth, and imperfections such as out-of-roundness or imperfect sphericity. The weight of stiffeners and reinforcements W_s will be a function of the collapse mode, frequently elastic yielding versus instability failure, and the number and weight of the penetrations.

For shells with a thickness-to-radius ratio t/R between 0.05 and 0.10, thick shell theory is required [3] for detailed analysis. However, to begin the design procedure, membrane theory, considering only the normal forces in the shell, can be used to approximate the design thickness. That thickness is subsequently adjusted to accommodate the required stiffeners and reinforcement.

The basic equation of equilibrium for a constant thickness membrane of revolution subjected to hydrostatic pressure in the radial direction [4] is

$$\frac{F_\theta}{R_\theta} + \frac{F_\phi}{R_\phi} = p \tag{4}$$

where

F_θ = circumferential force,
F_ϕ = meridional force,
R_θ, R_ϕ = circumferential and meridional radii, and
p = hydrostatic pressure.

For a sphere, $R_\theta = R_\phi = R$, so

$$F_\theta = F_\phi = \frac{pR}{2}$$

and

$$\sigma_\theta = \sigma_\phi = \frac{F}{t} = \frac{pR}{2t}$$

where σ_θ is the circumferential stress and σ_ϕ is the meridional stress. For a cylinder, $R_\phi = \infty$ and $R_\theta = R$, thus

$$F_\theta = pR \quad \text{and} \quad \sigma_\theta = \frac{pR}{t}$$

From equilibrium in the meridional (longitudinal) direction,

$$F_\phi = \frac{pR}{2} \quad \text{and} \quad \sigma_\phi = \frac{pR}{2t}$$

For an oblate spheroid, which is an ellipsoid rotated around its minor axis (Fig. 1), the normal forces are

$$F_\theta = \frac{pa^2}{2b^2}\left[\frac{b^2 - (a^2 - b^2)\sin^2\phi}{\sqrt{a^2\sin^2\phi + b^2\cos^2\phi}}\right]$$

$$F_\phi = \frac{pa^2}{2}\left[\frac{1}{\sqrt{a^2\sin^2\phi + b^2\cos^2\phi}}\right]$$

where a and b are equal to major and minor semiaxes, respectively, and ϕ is equal to meridional angle (Fig. 1).

Solving the equation of equilibrium for the oblate spheroid requires solutions in the meridional as well as the circumferential directions [4]. For the general problem, the third equilibrium equation (in the circumferential direction) must be considered.

An example of a two-to-one oblate spheroid is shown in Fig. 2. This geometry is not a good selection for a submersible pressure hull designed for any appreciable depth, since the peak stresses (σ_ϕ at ϕ = 90 deg and σ_θ at ϕ = 0 deg and 90 deg) correspond to that of a cylinder with a radius equal to the major semiaxis of the oblate spheroid, and whose buoyancy factor B is twice that of an equivalent cylinder and 8/3 that of an equivalent sphere. Moreover, consideration of elastic and elastic-plastic instability characteristics eliminates this shape as a good design selection for deep depth.

Membrane theory is valid only for the initial selection of geometries and thicknesses. Stiffeners and reinforcements introduce bending and shearing stresses in addition to the normal stresses, thus requiring the use of shell theory. Design techniques using thin shell theory for stiffened cylinders, spheres, and multispheres are the most important for submersibles.

2.1 Stiffened Cylindrical Shells

Axisymmetric ring-stiffened cylinders under external hydrostatic pressure can have three principal

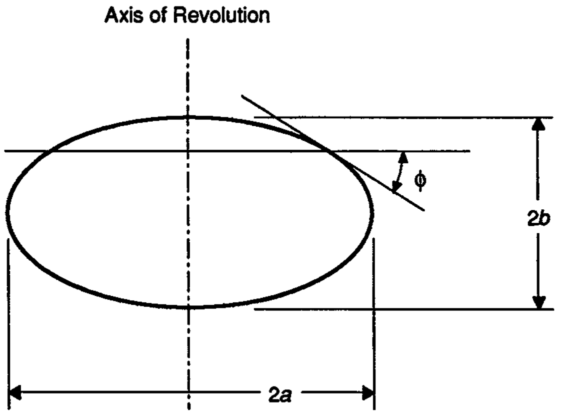


Fig. 1 Oblate spheroid

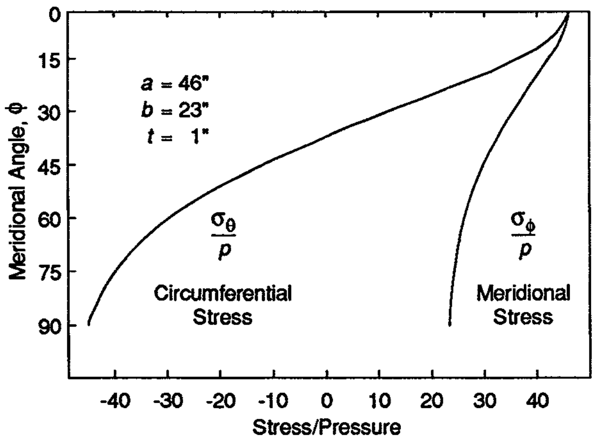


Fig. 2 Oblate spheroid-membrane stresses

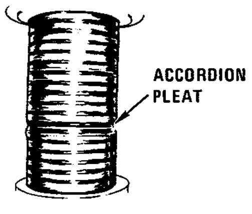


Fig. 3 Model failure in axisymmetric collapse of shell

modes of failure: axisymmetric shell yielding, lobar buckling of the shell, and general instability failure.

Axisymmetric shell yielding (Fig. 3) is initiated by elastic yielding at the extreme fibers at both the outer surface of the shell midway between stiffeners and at the inner surface of the shell at the stiffeners. This yielding leads to elastic-plastic collapse that is characterized by an accordion type of pleat extending around the periphery of the cylinder. This failure may occur in one or more spaces between stiffeners (frames).

Shell, or lobar buckling (Fig. 4), is an elastic instability of the shell between stiffeners and is characterized by inward and outward lobes, which may or may not develop around the entire periphery of the cylinder. The failure may occur in one or more frame spaces. This mode of failure indicates that the frames have greater resistance to buckling than the shell between frames.

General instability of the shell-and-stiffener combination (Fig. 5) can occur between structural bulkheads or deep frames. The instability is caused by the elastic buckling of the frame-shell combination. It is also characterized by inward and outward lobes, but the lobes are fewer (usually just two or three) than the number of lobes in shell or lobar buckling.

To select the shell thickness, frame size, and frame spacing, an iterative design process is used to ascertain the best design to resist the three failure modes described. Using the geometry and shell thickness derived from simple membrane theory, and knowing the material to be used, a refined thickness of shell t can be estimated by

$$t \approx \beta_1 \frac{p_c R}{\sigma_y} \tag{5}$$

where

p_c = design collapse pressure,
R = mean radius of the cylinder,
σ_y = yield strength of the material, and
β_1 = reduction factor for frame size and spacing.

Equation (5) is a modification of the well-known "hoop stress" equation $\sigma_y = (p_c R)/t$, in which circumferential yielding in an unstiffened cylinder is based on the Rankine principal stress failure theory. Frame reduction factors β_1 usually range from 0.7 to 0.9. For the lower values of β_1, the pressure hull is lighter but generally more costly.

With an estimated thickness established, the frame spacing L_f for a relatively shallow-depth, stiffened cylinder can be estimated from

$$L_f \approx \frac{R}{6} \tag{6}$$

For very deep depth hulls, for which elastic buckling is not a major factor, the frame spacing can be greater. The area of the frames can be estimated by

$$A_f \approx \beta_2 L_f t \tag{7}$$

The value of β_2 can range from 0.3 to 0.6.

Most frames for stiffened cylinders are T-sections consisting of a web and a flange, the dimensions of which can be estimated in terms of cylinder radius and shell thickness. The depth of the frame should range from 5 to 10 percent of the cylinder radius. The aspect ratio, defined as the ratio of web depth to web thickness, should range from 15 to 20. The flange width should be about three-quarters of the total frame depth, and the flange thickness should be about equal to the shell thickness. These values will vary for minimum weight at differing operating depths.

Three collapse pressures are calculated. For axisymmetric yielding (Fig. 3) the collapse pressure p_y is estimated by [1]

$$p_y = \frac{\sigma_y \dfrac{t}{R}}{1 + H\left(\dfrac{0.85 - B}{1 + \beta}\right)} \tag{8}$$

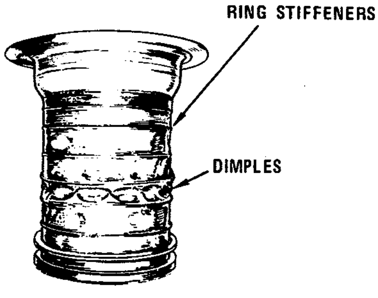


Fig. 4 Model failure in shell buckling

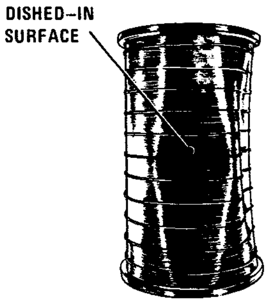


Fig. 5 Model failure in general instability

where

$B = \dfrac{bt}{A_f + bt}$

b = width of the frame web

$\beta = \dfrac{11N}{\sqrt{50t/R}}\left(\dfrac{t^2}{A_f + bt}\right)$

$N = \dfrac{\cosh\theta - \cos\theta}{\sinh\theta + \sin\theta}$

$\theta = 10\left[12(1-\mu^2)\right]^{1/4}\left(\dfrac{L_f}{2R}\right)\left(\dfrac{50t}{R}\right)^{-1/2}$

$H \approx -\dfrac{3\sinh\left(\frac{\theta}{2}\right)\cos\left(\frac{\theta}{2}\right) + \cosh\left(\frac{\theta}{2}\right)\sin\left(\frac{\theta}{2}\right)}{\sinh\theta + \sin\theta}$

For lobar buckling (Fig. 4), the collapse pressure p_b is estimated by [1]

$$p_b = \frac{2.42E}{(1-\mu^2)^{3/4}}\,\frac{\left(\dfrac{t}{2R}\right)^{5/2}}{\left[\dfrac{L_f}{2R} - 0.45\left(\dfrac{t}{2R}\right)^{1/2}\right]} \qquad (9)$$

For general instability (Fig. 5), the collapse pressure [1] is

$$p_{cr} = \frac{Et}{R}\left[\frac{m^4}{\left(n^2 + \dfrac{m^2}{2} - 1\right)(n^2 + m^2)^2}\right] + \frac{(n^2-1)EI}{R^3 L_f} \qquad (10)$$

where

$m = \dfrac{\pi R}{L}$,

L = bulkhead or deep frame spacing, and

I = moment of inertia for the frame-shell combination, using one frame spacing as the effective length of shell.

The value of n is set by trial, with the critical value of n (usually 2, 3, or 4) resulting in the lowest value of p_{cr}.

The design values for the factor of safety will usually vary for these three primary modes of failure. Normally, a lower factor of safety is applied for yielding failure than for instability failure, since yielding failure normally is less influenced by fabrication tolerances in stiffened cylinders.

2.2 Unstiffened Spheres

The unstiffened spherical shell has long been used for storing fluids and gases under pressure because it has the least weight per unit volume of any shape. Since similarly efficient weight-to-buoyancy ratios for spheres under external hydrostatic pressure are to be expected, numerous studies have been done to establish design theories that correlate with experimental results. The classical small deflection theory for elastic buckling of thin spherical shells was first developed by Zoelly in 1915 and published by Timoshenko in 1936 [5]. The collapse pressure predicted by Zoelly is

$$p_e = \frac{2E\left(\dfrac{t}{R}\right)^2}{\sqrt{3(1-\mu^2)}}$$

where

p_e = elastic buckling pressure,

E = modulus of elasticity in compression,

μ = Poisson's ratio,

t = shell thickness, and

R = midsurface radius.

For a Poisson's ratio of 0.3, the classical equation reduces to

$$p_e = 1.21E\left(\frac{t}{R}\right)^2$$

Results from model tests, however, have departed considerably from this equation, with collapse pressures as low as one-fourth of the classical prediction. Departures from sphericity and uniform shell thickness, as well as variations in material properties, were undoubtedly major contributors to these discrepancies. Subsequent tests of more accurately machined models have indicated that the collapse pressure predicted by classical theory is approachable. These tests have demonstrated that even small differences from the ideal dimensions and material properties will cause the collapse pressure to fall to about 70 percent of the classical prediction for it.

Based on many test results, the David Taylor Model Basin (now the David Taylor Naval Ship Research and Development Center) proposed [6] the following empirical equation for near-perfect elastic buckling of spheres:

$$p_e = 0.84E\left(\frac{t}{R_o}\right)^2 \quad \text{for } \mu = 0.3 \qquad (11)$$

where R_o is the outer radius.

Employing a plasticity factor, $\sqrt{E_{sec}E_{tan}}/E$, the following equation for inelastic (or elastic-plastic) buckling of a near-perfect sphere [6] is

$$p_p = 0.84\sqrt{E_{sec}E_{tan}}\left(\frac{t}{R_o}\right)^2 \quad \text{for } \mu = 0.3 \qquad (12)$$

where E_{sec} is the secant modulus in compression and E_{tan} is the tangent modulus in compression. To account for local imperfections in the shell, Krenzke and Kiernan [6] extended the empirical equation by demonstrating that the collapse of a spherical shell can be calculated from the shell radius and thickness dimensions over a critical arc length

$$L_c = 2.42\sqrt{R_l t_a} \quad \text{for } \mu = 0.3 \qquad (13)$$

where

L_c = critical arc length,

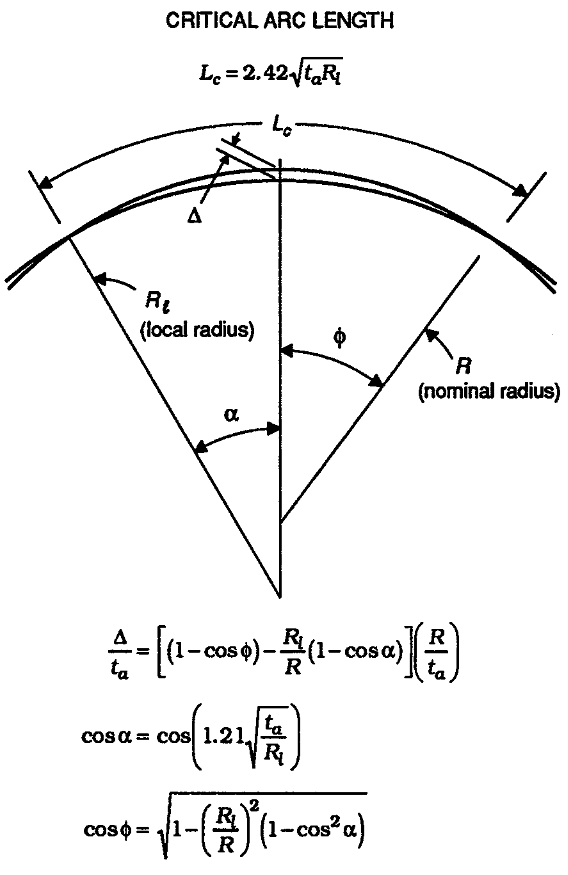


Fig. 6 Departure from sphericity (Δ)

R_l = local midsurface radius over the critical arc length, and

t_a = average thickness over the critical arc length.

Figure 6 shows the relationship between critical arc length and out-of-sphericity Δ. The elastic buckling collapse pressure becomes

$$p'_e = 0.84E\left(\frac{t_a}{R_{lo}}\right)^2 \quad (14)$$

where R_{lo} is the local outer radius over the critical arc length. The inelastic buckling collapse pressure becomes

$$p'_p = 0.84\sqrt{E_{sec}E_{tan}}\left(\frac{t_a}{R_{lo}}\right)^2 \quad (15)$$

The average stress in the area of the critical arc length is

$$\sigma_a = \frac{p(R_{lo})^2}{2R_l t_a} \quad (16)$$

For a machined, near-perfect sphere, a 10 percent increase in the radius R_{lo} is a reasonable design value, corresponding to an out-of-sphericity of about 7.4 percent of the shell thickness.

The type of manufacturing process employed also affects the collapse pressure. Tests [7] have shown that the collapse pressure will be reduced by a manufacturing process factor k. For HY-80 (yield strength of 80 000 psi) steel, the k factor is shown in Fig. 7.

For a spherical shell, an iterative procedure is used to obtain a design thickness that matches the design collapse pressure p_c. Figure 8 shows the plot

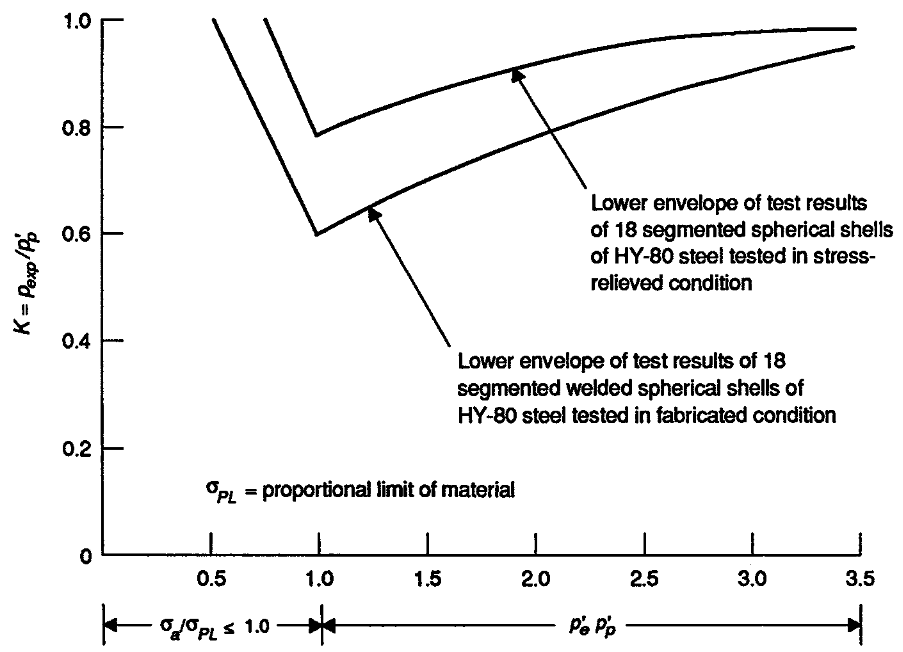


Fig. 7 Summary of DTMB data on unstiffened spherical shells

Stress, psi	$\sqrt{E_{sec}E_{tan}}/E$	p'_p, psi	p_e, psi
102 000	1.0	47 800	10 610
108 000	0.846	40 500	11 300
115 000	0.768	36 700	11 900
121 000	0.623	29 800	12 600
127 500	0.541	25 800	13 300
134 000	0.394	18 700	13 900
140 000	0.301	14 400	14 600
146 500	0.202	9 650	15 250

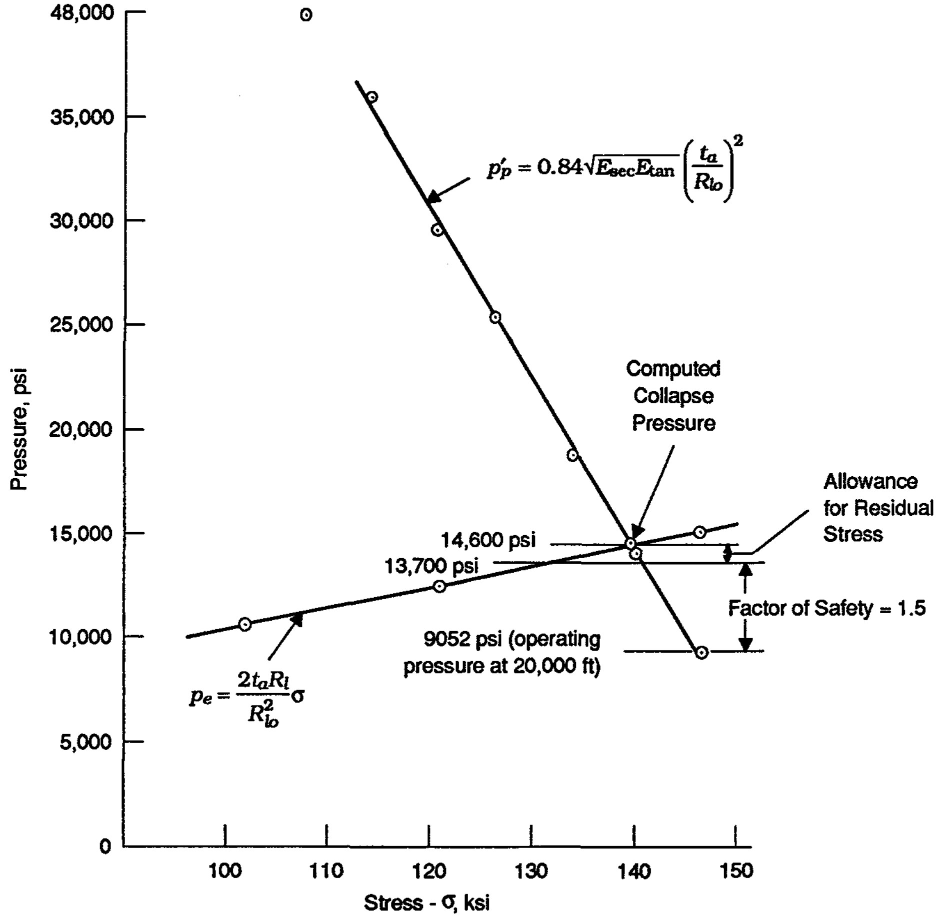


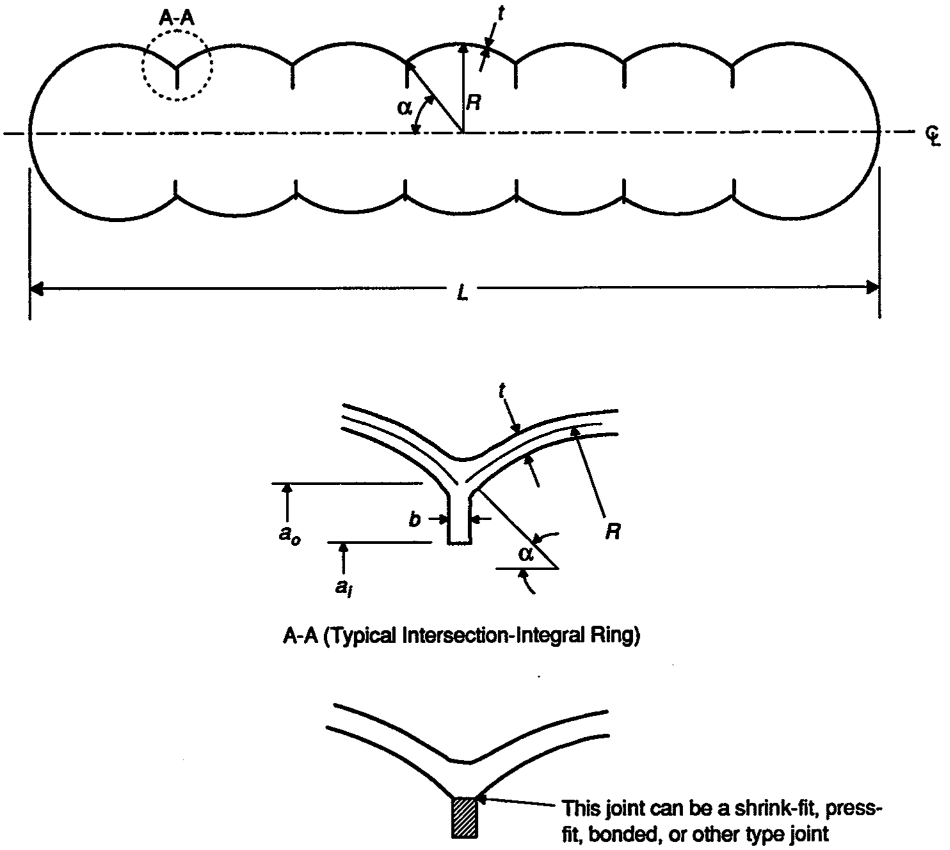
Fig. 8 Collapse pressure for a titanium sphere

of p'_p versus the pressure corresponding to the average stress σ_a over the critical arc length for a titanium sphere. As a first step, a thickness is approximated from equation (16) by

$$t_a \approx \frac{p_c(R_{lo})^2}{2\sigma_y R_l}$$

where σ_y is the yield stress and p_c is the design collapse pressure. With this thickness as a reference point, the true collapse pressures can be calculated, using the actual stress-strain plot of the material to compute E, E_{sec}, and E_{tan}. The collapse pressure can be found from the intersection of the p'_p curve with the p_e yield curve (see Fig. 8). These values will be reduced by the manufacturing process factor k, such as by the values shown in Fig. 7 for HY-80 steel.

With this information, all the data for the basic spherical shell are known. That is, the nominal and local values of the radius and thickness are known, as well as the material properties. Calculations for reinforcements and stiffeners must follow. The only stiffener in single spheres normally is at the equator, where the two hemispheres are joined.



A-A (Optional Case of Composite Construction)
n = Number of Stiffened Rings (6 shown above)

Fig. 9 Multiple intersecting spheres

2.3 Multiple Intersecting Spheres

A logical derivative of the unstiffened single sphere design is the multiple intersecting spheres design. This configuration has the structural advantages of the single sphere and, with fairings between spheres, the hydrodynamic advantage of a cylinder having a suitable length-to-diameter ratio. It also has internal arrangement advantages as compared to a single sphere.

The multiple spheres concept involves a series of spheres which are reinforced at their intersections by reinforcing rings (see Fig. 9). These reinforcing rings can be made either of the same material as the spherical shell, or of a material that is stronger and that has a greater elastic modulus. The latter alternative results in a considerably better buoyancy factor B, since the substantial weight of the reinforcing ring is reduced. The annular area of the reinforcing ring is also smaller, thus allowing for a larger access path between spheres. Costs may be attractive too, since the rings are a small part of the total hull weight.

The buoyancy factor for intersecting spheres is [8]

$$B = \frac{3\left[nW_R + 4\pi R^2 t\rho_s(n\cos\alpha + 1)\right]}{4\pi R^3\rho_w\left[(n\cos\alpha)(1 + \tfrac{1}{2}\sin^2\alpha) + 1\right]} \tag{17}$$

where

n = number of rings,
W_R = ring weight, and
α = angle of shell intersection.

The component of the shell's membrane displacement normal to the axis of revolution is

$$\delta_s = \frac{pR^2 \sin\alpha}{2tE_s}(1 - \mu_s) \tag{18}$$

where E_s is Young's modulus of the sphere and μ_s is Poisson's ratio of the sphere. The deflection of the ring at the intersection of the shell is

$$\delta_R = \frac{pR^2 \sin\alpha \cos\alpha}{E_R b}\left(\frac{1 + C^2}{1 - C^2} - \mu_R\right) \tag{19}$$

where

E_R = Young's modulus of the ring,
μ_R = Poisson's ratio of the ring,

b = width of the ring, and

C = ratio of inner/outer radii of the ring (a_i/a_o).

By setting the shell deflection equal to the ring deflection at their intersection,

$$E_R(1-\mu_s)\left(\frac{1+C^2}{1-C^2}-\mu_R\right)=K(1-C^2)$$

$$\frac{1+C^2}{1-C^2}-\mu_R=\frac{2E_R(1-\mu_s)}{K(1-C^2)} \tag{20}$$

where K is the ratio of maximum stress to membrane stress in the shell. The stress in the ring at its inner radius is

$$\sigma_R=\frac{2pR\cos\alpha}{b(1-C^2)} \tag{21}$$

If the maximum stress in the ring is K times the membrane stress in the shell, $\sigma = K(pR/2t)$, then the ring's width is

$$b=\frac{4t\cos\alpha}{K(1-C^2)}$$

$$K(1+C^2)-K\mu_R(1-C^2)=2E_R(1-\mu_s)$$

$$C^2(K+K\mu_R)+K-K\mu_R=2E_R(1-\mu_s)$$

$$C^2(1+\mu_R)+(1-\mu_R)=\frac{2E_R}{K}(1-\mu_s)$$

$$C^2(1+\mu_R)=\frac{2E_R}{K}(1-\mu_s)-(1-\mu_R)$$

$$C=\left[\frac{2E_R}{K}\frac{(1-\mu_s)}{(1+\mu_R)}-\frac{(1-\mu_R)}{(1+\mu_R)}\right]^{1/2} \tag{22}$$

From equations (20) and (22),

$$C=\left|\frac{2E_R}{KE_s}\frac{(1-\mu_s)}{(1+\mu_R)}-\frac{(1-\mu_R)}{(1+\mu_R)}\right|^{1/2}\qquad 0\le C\le 1 \tag{23}$$

and

$$B=\left(\frac{6t}{KR}\right)\left\{\frac{n\rho_R\cos\alpha\sin^2\alpha+K\rho_s(1+n\cos\alpha)}{\rho_w\left[2+(n\cos\alpha)(2+\sin^2\alpha)\right]}\right\} \tag{24}$$

For single material construction, the buoyancy factor is

$$B=\left(\frac{6t}{KR}\right)\left(\frac{\rho_s}{\rho_w}\right)\left[\frac{n\cos\alpha\sin^2\alpha+K(1+n\cos\alpha)}{2+(n\cos\alpha)(2+\sin^2\alpha)}\right] \tag{25}$$

and

$$C=\left[\frac{(2-K)(1-\mu_s)}{K(1+\mu_s)}\right]^{1/2}\qquad 0\le C\le 1 \tag{26}$$

Therefore, for homogeneous material the maximum value of K is 2, which means that $C = 0$ and $B = 3t\rho_s/R\rho_w$, indicating the case of a solid ring reinforcement of width

$$b=2t\cos\alpha$$

The buoyancy factor is independent of the number of spheres, or the same as for a single sphere of equivalent characteristics. For the case of single material construction, and a value of K of 1 (ring stresses equal to membrane shell stresses), $C = 0.73$ for $\mu_s = 0.3$. The angle α can be selected to provide suitable access from one compartment to another.

For composite construction, considerable weight can be saved by using a ring having a high strength-to-weight ratio σ_y/ρ_R and a high specific stiffness E_R/ρ_R. Most metals have a specific stiffness of about 10^8 lb/in. Fiber-reinforced plastic (FRP), glasses, and ceramics have higher values of both strength-to-weight ratios and specific stiffnesses [8]. For example, a multiple sphere ($n = 10$) design for a titanium shell and ring results in a buoyancy factor B of 0.765.

$$E_s = 18\times10^6\ \text{lb/in.}^2$$
$$\rho_s = \rho_R = 0.16\ \text{lb/in.}^3$$
$$\alpha = 45\ \text{deg}$$
$$R/t = 20$$

For a similar titanium multiple sphere ($n = 10$) design having a ceramic (alumina) ring $\rho_s \approx \rho_R$ and $K = 3$, B becomes 0.609 for total weight savings of 20.4 percent. Such weight savings are very important for a deep-diving submersible, because the effective payload is a very small percentage of the total displaced weight. Savings of 20 percent in the total displaced weight might increase the payload by a factor of five of more.

Since composite ring construction for multiple spheres can yield buoyancy factors better than those for single spheres, this technique offers much promise for future deep-diving hydrospace vehicles. In some cases, the ring material might be less ductile than the shell, so the strength of the ring would be used to attain a suitable factor of safety only, and not be required to obtain a safe operating depth in the case of a ring failure. In this way, safe return to the surface in the event of a ring failure would be assured. The hydrodynamic and arrangement benefits of multiple intersecting spheres, combined with the structural advantages of composite spheres and rings, make this concept very attractive.

2.4 Special Structural Considerations

The primary considerations governing the selection of the design and materials for submersible pressure hulls must be augmented by detailed consideration of low cycle fatigue, dynamic effects, fracture toughness, and environmental effects. These affect the selection of the material, particularly with regard to connections and penetrations.

If the design of a submerged vehicle were ideal, stress reversals would not occur and fatigue problems would be greatly alleviated. However, because of imperfections introduced in hull fabrication and because of the existence of penetrations and "hard

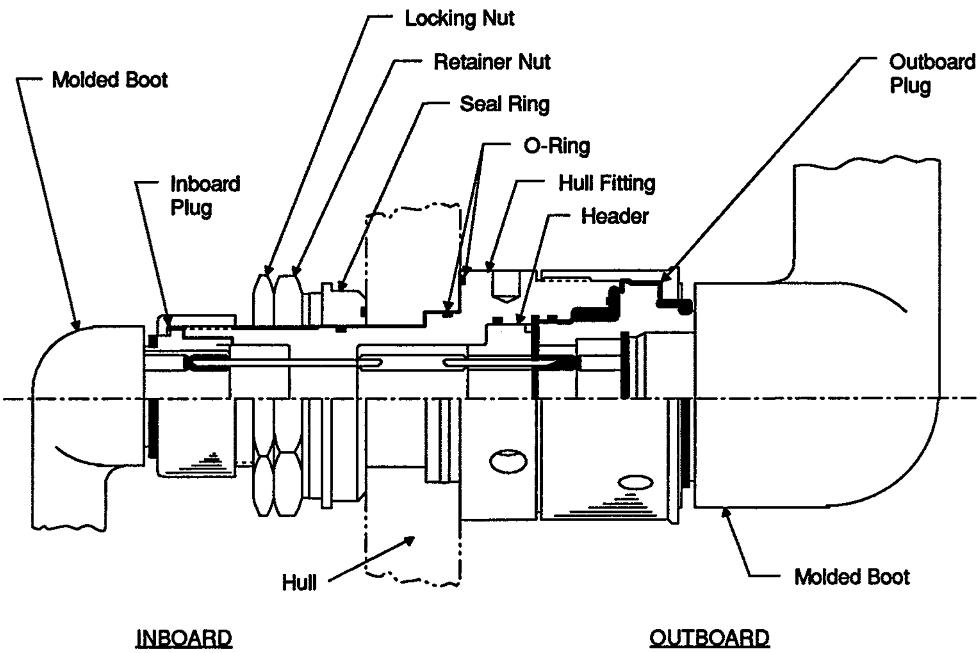


Fig. 10 Electrical hull penetrator

spots" at connections, zones of both residual and applied tensile stresses can exist. Thus, low-cycle fatigue must always be considered in the design, particularly in the design of pressure hull penetrations. Typical penetrations in deep submersibles include electrical penetrations, viewports, and hatches.

A typical electrical penetration is shown in Fig. 10. An electrical hull penetration usually consists of a penetrator body with outboard and inboard plug assemblies. The penetrator body houses a primary and a secondary conductor in the form of glass-sealed contact header assemblies. Electrical cables are wired and molded to the inboard and outboard plug assemblies. A typical viewport is shown in Fig. 11. Most viewports to date have consisted of 45-deg conical frustums made of acrylic, the design of which has proven to be structurally sound and optically efficient. Hatches can be of two basic designs, either seat or plug type. Seat hatches (Fig. 12) are used in relatively shallow depth submersibles because they are economical and simple to operate; however, they do incur a weight penalty. Plug hatches (Fig. 13) are used in deep-diving vehicles because they save weight by directly reinforcing the pressure hull. Details of the designs of penetrations are presented in reference [9].

3. Exostructure Design

The primary objective in the development of an exostructure design is to achieve the least-weight structure at the lowest cost that satisfies operational and structural requirements. Ideally, the quest for an optimum exostructure design would begin with efforts to achieve a least-weight design for a given vehicle envelope—the material and structural form

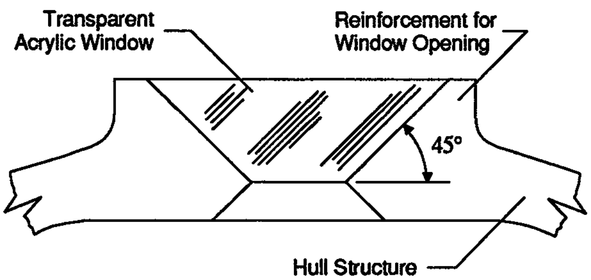


Fig. 11 Viewing port for oceanographic vehicle

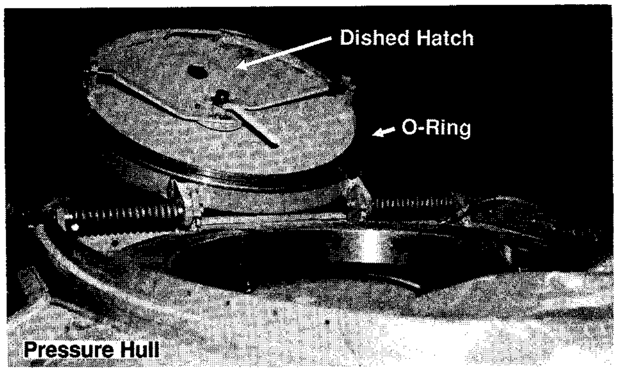


Fig. 12 Seat-type hatch

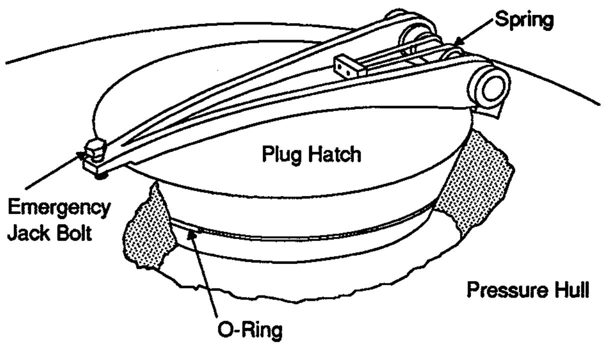


Fig. 13 Plug-type hatch

that safely would carry the prescribed loads with the minimum structural weight. This then would be followed by tradeoff studies with non-strength-related variables such as cost and serviceability. The benefit to be gained from this procedure is that it enables the designer to evaluate quantitatively the weight penalties incurred in exchange for other important attributes such as lower cost. There are systematic procedures for selecting minimum weight designs, commonly referred to in the literature as "principles of optimum structural design." These methods can be subdivided into two categories: analytical and numerical. Such methods are addressed in Chapter II, Section 4, and will not be discussed further here. The interested reader may consult references [10] through [14] for further information.

3.1 Material Selection

Since material selection is an integral part of exostructure design, the designer requires some rational basis for assessing the structural efficiency of the various potential materials. The designer's initial efforts will turn to that eternal quest for the ideal material—one with infinite strength and zero weight. In considering these important properties, it is worth noting that we seek not as much strength as possible for a given weight, but rather a given strength for the intended service with the least possible weight. To aid in that selection, a quantitative means is presented below for making useful weight-strength comparisons of potential materials, based on the two dominant modes of structural behavior—flexure and buckling—that govern the design of an exostructure.

For the flexural or bending modes, the ratio of weights of any two materials resisting the same bending moment M can be expressed as

$$\frac{W_1}{W_2} = \frac{\rho_1}{\rho_2}\left(\frac{\sigma_{a_2}}{\sigma_{a_1}}\right)^{1/2} \tag{27}$$

where
W = weight (lb),
ρ = material density (lb/in.^{3}), and
σ_a = allowable material stress.

For the buckling of plate elements, the ratio of weights of any two materials resisting the same compressive force is

$$\frac{W_1}{W_2} = \frac{\rho_1}{\rho_2}\left(\frac{E_2}{E_1}\right)^{1/3} \tag{28}$$

where E is the modulus of elasticity (lb/in.^{2}). Equations (27) and (28) provide a numerical tool for comparing weight-strength properties of various materials, specifically their efficiency in resisting flexure and compressive buckling. It is now possible to construct a table of data, using only three material properties, which will permit a weight-strength comparison of alternate materials. Using aluminum as the reference material and some representative handbook values for the various material properties, Table 1 presents such a comparison. Restating equations (27) and (28):

Flexure:

$$\frac{\text{weight of material } n}{\text{weight of reference material}} = \frac{\rho_n}{\rho_1}\left(\frac{\sigma_1}{\sigma_n}\right)^{1/2} \tag{29}$$

Compressive buckling of plates:

$$\frac{\text{weight of material } n}{\text{weight of reference material}} = \frac{\rho_n}{\rho_1}\left(\frac{E_1}{E_n}\right)^{1/3} \tag{30}$$

As a practical matter, such comparisons must be combined with cost considerations. None of the materials in Table 1, for instance, would be a better choice than aluminum 5456 for either flexure or compressive buckling properties if the alternative costs several times as much to procure and fabricate.

Table 1 Weight/strength comparison of certain materials

Material	$\sigma_a \times 10^3$ psi	ρ, lb/in.^{3}	$E \times 10^6$ psi	$\frac{\sigma_a}{\rho}$	$\frac{E}{\rho}$	$\frac{\rho_n}{\rho_1}\left(\frac{\sigma_1}{\sigma_n}\right)^{1/2}$	$\frac{\rho_n}{\rho_1}\left(\frac{E_1}{E_n}\right)^{1/3}$
Aluminum 5456	21.0	0.096	10.3	218.8	107.3	1.00	1.00
Titanium Ti-6Al-4V	100.0	0.160	16.0	625.0	100.0	0.76	1.44
CRES 316	3.0	0.286	28.0	104.9	97.9	2.49	2.14
Glass-reinforced plastic	24.0	0.070	3.2	342.9	45.7	0.68	1.08
Steel HY-100	100.0	0.286	29.0	349.7	101.4	1.37	2.11

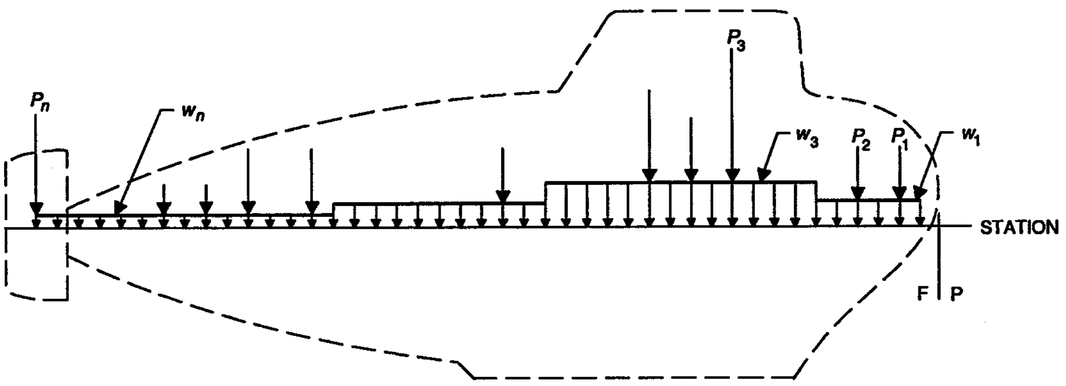


Fig. 14 Typical vehicle weight distribution diagram

3.2 Exostructure Loads

Proper definition of design loads is another important facet of the exostructure design process, since underpredicted loads can result in structural failure and overpredicted loads in excessive vehicle weight. If structural weight becomes excessive as a result of designing for a particular load, one of the tradeoffs that might be made is to impose specific limits on, say, vehicle operations or constraints for transportability, so as to limit the total weight. For these reasons, the structural designer must have complete knowledge of the various mission operating, handling, and transport conditions, and of the resultant loads. The conditions and loadings listed below are considered representative of what can be expected during the life of many submersible vehicles.

A. Design Conditions:
1. Surfaced
2. Submerged
3. Launching and retrieval at sea
4. Cradle or transport
5. Towing
6. Impact (submerged)
7. Bottom sitting

B. Design Loads:
1. Static weight of vehicle (gravity loads)
2. Buoyancy forces
3. Hydrodynamic forces
4. Sea slap (wave forces) 500 psf
5. Dynamic load factor of 2 for condition (A3)
6. Kinetic energy associated with impact velocity v_1
7. Propulsion and thruster forces
8. Equipment operating forces
9. Towing forces

Once the design conditions and associated loads are defined, load diagrams consisting of weight distribution, applied external forces, and reactions are developed for each such condition. Figure 14 is an example of a typical weight distribution diagram.

The load distribution diagram for the surfaced condition (A1) consists of vehicle gravity loads and buoyancy force distributions as shown, for example, in Fig. 15. Similarly, the load distribution diagram for condition (A2) would be composed of gravity and buoyancy forces in the submerged mode.

For design condition (A3), launching and retrieval at sea, a dynamic load factor of 2 customarily is applied to the weight of the vehicle suspended in air to represent dynamic conditions encountered at sea; the vehicle's estimated weight is doubled to define the force used to design the structure through which the lift forces would be transmitted. The line of action of this static equivalent of an inertial force is taken to be vertical, consistent with the manner in which the vehicle normally would be suspended.

Design condition (A4), cradle or transport, must be evaluated for all anticipated modes of transport, that is, aircraft, ground vehicle, and, of course, by sea. For the transport-by-sea mode, it is important to know the type of vessel or vessels considered for use in transporting the submersible vehicle to its operating site so that ship's motion in a seaway can be assessed. Once the roll, pitch, heave, etc., characteristics of the transport vessel are known, a means of supporting the vehicle can be devised by considering the various inertia forces described below.

The general forms of the equations used to calculate the magnitude of these dynamic forces are presented below. A detailed treatise on their derivation may be found in reference [15].

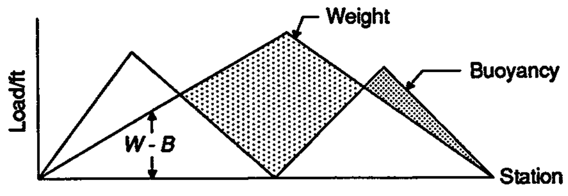


Fig. 15 Vehicle gravity load and buoyancy force distribution

$$R_x = m(\pm g \sin\theta \pm x\dot{\theta}^2 \pm y\ddot{\theta}) \quad (31)$$

$$R_y = m(-g \cos\phi \cos\theta \pm y\dot{\phi}^2 \pm z\ddot{\phi} \pm \dot{\theta}^2 \pm x\ddot{\theta}) \quad (32)$$

$$R_z = m(\pm g \sin\phi \pm z\dot{\phi}^2 \pm y\ddot{\phi}) \quad (33)$$

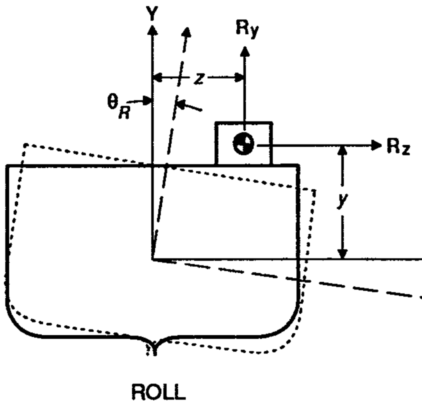
where

g = acceleration due to gravity (32.2 ft/s^2),
m = mass of supported object (lb·s^2/ft),
$R_{x,y,z}$ = forces acting at supported object's center of gravity that result from gravity and rolling and pitching motions of the transport vessel. Subscripts indicate the reference axis to which each is parallel (lb),
x, y, z = distances parallel to X, Y and Z reference axes from ship's (assumed) center of motion to the supported object's center of gravity (ft),
θ = ship's pitch angle (rad),
$\dot{\theta}$ = ship's pitching velocity = $d\theta/dt$ (rad/s),
$\ddot{\theta}$ = ship's pitching acceleration = $d^2\theta/dt^2$ (rad/s^2),
ϕ = ship's roll angle (rad),
$\dot{\phi}$ = ship's rolling velocity = $d\phi/dt$ (rad/s),
$\ddot{\phi}$ = ship's rolling acceleration = $d^2\phi/dt^2$ (rad/s^2).

These equations represent the resultant components of force acting through the center of gravity of the supported mass, and parallel to the ship's reference axis (see Fig. 16). Here, the origin of the ship's reference axis is at the center of motion, which for convenience can be taken at the vessel's center of gravity without introducing appreciable error. Each resultant force R is composed of terms representing contributions from the force of gravity and the dynamic centrifugal and tangential forces produced by ship rolling and pitching. The magnitude of these terms, and their additive or subtractive effect, depends on the supported object's location relative to the ship's center of motion; the direction, speed, and acceleration of roll and pitch; and the degree of angular deviation from level keel.

The complexity of the computations is simplified greatly when the following conditions are realized:

- Pitch and roll velocities are greatest as the ship passes through the point of zero pitch or roll angle, and are zero at maximum pitch or roll angle.
- Pitch and roll accelerations are greatest at maximum pitch or roll angles, and zero as the ship passes through the point of zero pitch or roll angle.



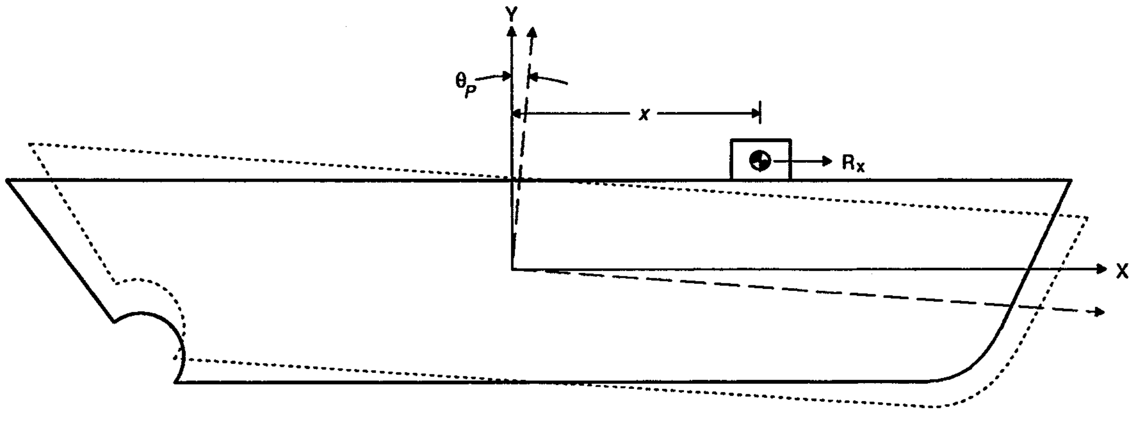


Fig. 16 Ship inertia force reference system

Table 2 Characteristics of pitch and roll velocity and acceleration for the four definitive combinations of ship pitch and roll angles

	Ship Angles		Velocities		Accelerations	
Case	θ	ϕ	$\dot{\theta}$	$\dot{\phi}$	$\ddot{\theta}$	$\ddot{\phi}$
1	0	0	max	max	0	0
2	max	0	0	max	max	0
3	0	max	max	0	0	max
4	max	max	0	0	max	max

- Submersibles being shipped almost invariably will be supported on the ship's deck, that is, above the ship's center of motion.

Given these conditions and assumptions, it is necessary to consider only four definitive cases of pitch and roll angles, which may combine to impose forces on the supported object. The combinations and their corresponding values of velocity and acceleration are given in Table 2.

For each of these combinations, the general equations for inertial forces reduce to more manageable versions, as certain terms reach zero or maximum values. Each case may have three nonzero force components, corresponding to the vehicle's principal axes, acting in different proportions on the supported object. R_x and R_z may have either positive or negative directions, depending on the direction and position of the pitch and roll angles.

The maximum velocities and accelerations are obtained from:

$$\dot{\theta}_{max} = \frac{2\pi\theta_{max}}{T_\theta} \qquad \ddot{\theta}_{max} = \left(\frac{2\pi}{T_\theta}\right)^2 \sin\theta_{max}$$

$$\dot{\phi}_{max} = \frac{2\pi\phi_{max}}{T_\phi} \qquad \ddot{\phi}_{max} = \left(\frac{2\pi}{T_\phi}\right)^2 \sin\phi_{max}$$

where T_θ and T_ϕ are the periods of one complete pitch and one complete roll, respectively.

The simplified versions of equations (31) and (33) are listed in Table 3 for the four conditions of interest. R_y always will be negative (downward) except where very rapid pitch and roll velocities occur. The designer must ascertain which combination of force directions causes the most severe loading condition in each of the four cases outlined. The corresponding equations for vertical forces from equation (32) are given in Table 4.

The design towing force, condition (A5), is of concern, of course, only if the vehicle will be towed. The magnitude of the towing force to be used for exostructure design or stress analysis purposes or both is determined by consideration of the following factors: vehicle hydrodynamic drag (which is a function of vehicle form and surface area); sea state; towing line length and elasticity; and towing system design (that is, bridle or single point).

Design condition (A6), impact, is the most difficult to predict with any degree of certainty since it encompasses numerous possibilities, both inevitable and potential, that the vehicle might encounter in service. Aside from surface encounters with support vessels in various sea states, the main concern is for vehicle impact with rigid submerged objects. Collisions with ice in very low air temperatures would be of special concern in a vehicle intended for arctic research. One means of designing for impact is to postulate that the vehicle might be carried uncontrolled in a lateral direction by underwater current into some undefined rigid object. The point of impact is postulated conservatively to be on a horizontal line through the vehicle's center of gravity. The maximum velocity of underwater currents in the regions of planned or anticipated vehicle operation would be examined and used to predict the kinetic energy developed by the vehicle at the moment of impact. The resulting value of kinetic energy is

$$KE = \frac{mv^2}{2g} \tag{34}$$

where

KE = kinetic energy,
m = virtual mass, or mass of the vehicle including entrained water plus an added mass of water. As an approximation, use m equals twice the mass of the vehicle, including entrained water,
v = current velocity,
g = gravity constant.

Table 3 Formulae for fore-and-aft and athwartship inertial forces for the four definitive combinations of ship pitch and roll angles

Ship Angles			
ϕ	θ	Fore-and-Aft, R_x =	Athwartship, R_z =
0	0	$\pm m_x(\dot{\theta}_{max})^2$	$\pm m_z(\dot{\phi}_{max})^2$
max	0	$\pm m[g \sin\theta_{max} + y\ddot{\theta}_{max}]$	$\pm m_z(\dot{\phi}_{max})^2$
0	max	$\pm m_x(\dot{\theta}_{max})^2$	$\pm m[g \sin\phi_{max} + y\ddot{\phi}_{max}]$
max	max	$\pm m[g \sin\theta_{max} + y\ddot{\theta}_{max}]$	$\pm m[g \sin\phi_{max} + y\ddot{\phi}_{max}]$

Table 4 Formulae for vertical inertial forces for the four definitive combinations of ship roll and pitch angles

Ship Angles		
θ	ϕ	Vertical Force, R_y =
0	0	$-mg$ for the static condition, or $-m[g - y(\dot{\phi}_{max})^2]$
max	0	$m[\cos\theta_{max} - y(\dot{\phi}_{max})^2 + x\ddot{\theta}_{max}]$
0	max	$m[g \cos\phi_{max} \cos(\dot{\theta}_{max})^2 + z\ddot{\theta}_{max}]$
max	max	$-m[g \cos\phi_{max} \cos\theta_{max} + z\ddot{\theta}_{max} + \ddot{\theta}_{max}]$

This kinetic energy must be absorbed by the strain energy of deformation (elastic and inelastic) of the exostructure, which can be represented by

$$KE = U \tag{35}$$

where U is the internal strain energy of deformation of the structure; it is the sum of the strain energies of its members. See references [16], [17], or [18], or any standard advanced text on strength of materials for a treatment of energy methods in structural analysis.

An alternate procedure, often used in treating this issue of impact resistance, is to design the exostructure so as to satisfy all other design load conditions, and then to analyze the resulting structure by the kinetic energy method described above for the degrees of exostructure deformation resulting from various impact velocities. While in the first method, a known or anticipated impact velocity is used for purposes of designing sufficient vehicle strength to absorb the resultant kinetic energy at impact, in the alternate method an exostructure design developed for other than impact conditions is used to calculate impact velocities that cause varying degrees of structural deformation. The latter procedure is useful in gaining some quantitative measure of the vehicle's vulnerability to damage by impact.

Design condition (A7), bottom sitting, is treated in a manner similar to condition (A6), with three important exceptions: first, the vehicle descent velocity at touchdown is controlled and assumed to be unaffected by underwater currents; second, landing skids or bottom supports are used to distribute bottoming impact forces into the exostructure in a fairly uniform manner, in contrast with the concentrated forces induced by lateral impact; and last, for the condition where repeated bottoming excursions are planned or might reasonably be anticipated, it is desirable to keep structure deflections (strains) within elastic limits, otherwise the cost and delays required for repair of damage would become prohibitive. This means that the kinetic energy of the vehicle in bottoming must be dissipated through the elastic strain energy of deformation U of the exostructure, whereas for uncontrolled lateral impact, which is essentially a casualty situation (repeated incidents are not expected), permanent deformation of the shell and framing may be permitted in order to avoid an appreciable weight penalty in designing the exostructure to behave elastically. Stated differently, since most materials that are likely to be used in a submersible exostructure are fairly ductile, considerably more strain energy can be developed inelastically than elastically. This can be seen in a typical stress-strain curve for, say, 5456 aluminum (Fig. 17), where the triangular area under the linear portion of the curve represents elastic strain energy, and the area under the remainder of the curve (which is essentially rectangular) represents inelastic strain energy. Of course, the entire inelastic region of the stress-strain curve would not be used for design purposes, since this would result in fracture; rather, some fracture-safe limiting value of inelastic strain would be chosen for design.

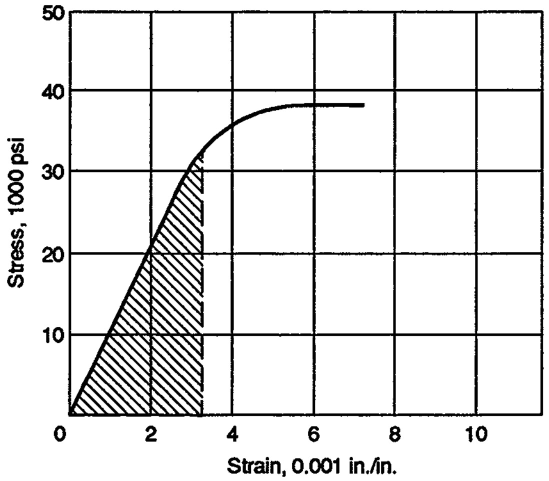


Fig. 17 Stress-strain curve—aluminum 5456-H321

The design loads specified under list B have been addressed, or are self-evident, with perhaps the exception of (B4), sea slap forces. The 500 psf loading, like the design load factor of 2 for design condition (A3), is largely empirical and has its origin in satisfactory, time-proven usage. These wave slap forces generally are considered to be multidirectional (including vertical upward to represent slamming of the vehicle when pitching in severe seas), and to act on the projected area of circular cross-sections as a cosine distribution, as shown in Fig. 18, and on the projected area of flat surfaces such as a rudder, as a uniform load.

Once the loading conditions described above are translated into load or force diagrams such as those shown in Fig. 19, the design of the exostructure can proceed. It is worth noting that structural members having different functions are apt to have differing load conditions as their principal design constraint.

3.3 Types of Framing Systems

For classification purposes, exostructure framing systems can be divided into two major categories: framed structures and stiffened shells. Examples of each type are shown in Fig. 20 and Fig. 21. In framed structures, an assemblage of interconnected structural members constitutes a skeletal system that is self-sufficient in strength for the vehicle, without benefit of contribution from the shell. In stiffened shells, the shell is for the most part continuous and serves as the main load-carrying member, with frames and stiffeners serving to strengthen and stiffen it. Three-dimensional framed structures may be further categorized as space trusses or space

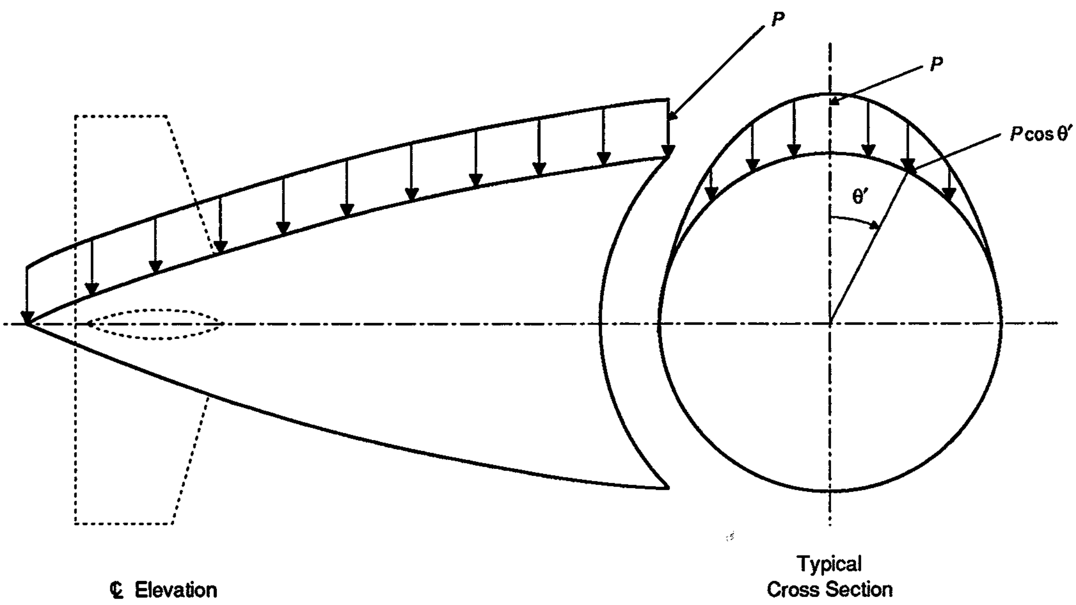


Fig. 18 Sea slap depicted as a cosine distribution

frames. The sole distinction between the two lies in the manner in which the members are connected at the joints.

3.3.1 *Framed Structures*

A space truss is an assemblage of structural members of arbitrary direction in space, in which, under theoretically ideal conditions, the members are joined by means of frictionless pins. Consequently, the members are free to rotate at their joints and do not exert any torque on each other. This differs from the space frame, in which the members are connected rigidly at their intersection, resulting in the transfer of bending moment across the joints. The significance of this differing behavior at the joints can be seen in the way in which loads are carried by the members comprising a framed structure (see Fig. 22).

Ideally, it is desirable that there be no restriction on the manner in which loads are introduced into a framed structure, either as concentrated forces applied at the joints, or as concentrated or distributed loads, or both, at any arbitrary location along the length of the member. For without any location constraints, components could be arranged optimally and attached to the framing by foundations (see Section 3.6) via the most efficient path. However, there are certain limitations on the load capacity of the structural members as a consequence of the manner in which they are joined (that is, either free to rotate or rigidly interconnected). In a space truss, loads normally are introduced into the structure as concentrated forces applied at the joints. By loading the structure in this way, it is intended that only axial tension or compression forces exist in any member, except for the small amount of bending stress produced by the deadweight of horizontal and inclined members. If concentrated or distributed lateral loads are applied to a truss member at any location along its length, and combined with axial compressive forces, a nonlinear amplification of deflection and stress arises, commonly referred to in the structural literature as the beam-column effect. Depending on the magnitudes and relationship between the axial and lateral forces (dependent or independent of each other), it might become necessary to make certain truss members inordinately heavy in order to sustain such forces safely. The practical consequence of this is that large (heavy) equipment must be attached to the space truss only at or near joints, which can be a severe restriction both on the arrangement of equipment and on the weight of foundations supporting such equipment.

In the space frame, members theoretically are fastened rigidly together at the joints in such a manner that no change is possible in the angles between the members meeting there. Laterally loaded members of a space frame result in moments at the joints that are shared by common members in proportion to their relative stiffnesses. Because of the rigid connections at the joints, space frames are less vulnerable to catastrophic failure than space trusses, in which the loss of a single member may precipitate the collapse of the whole framework. Hence, space frames are more stable than space trusses. Since space frames do not have the same load attachment limitations as space trusses, there

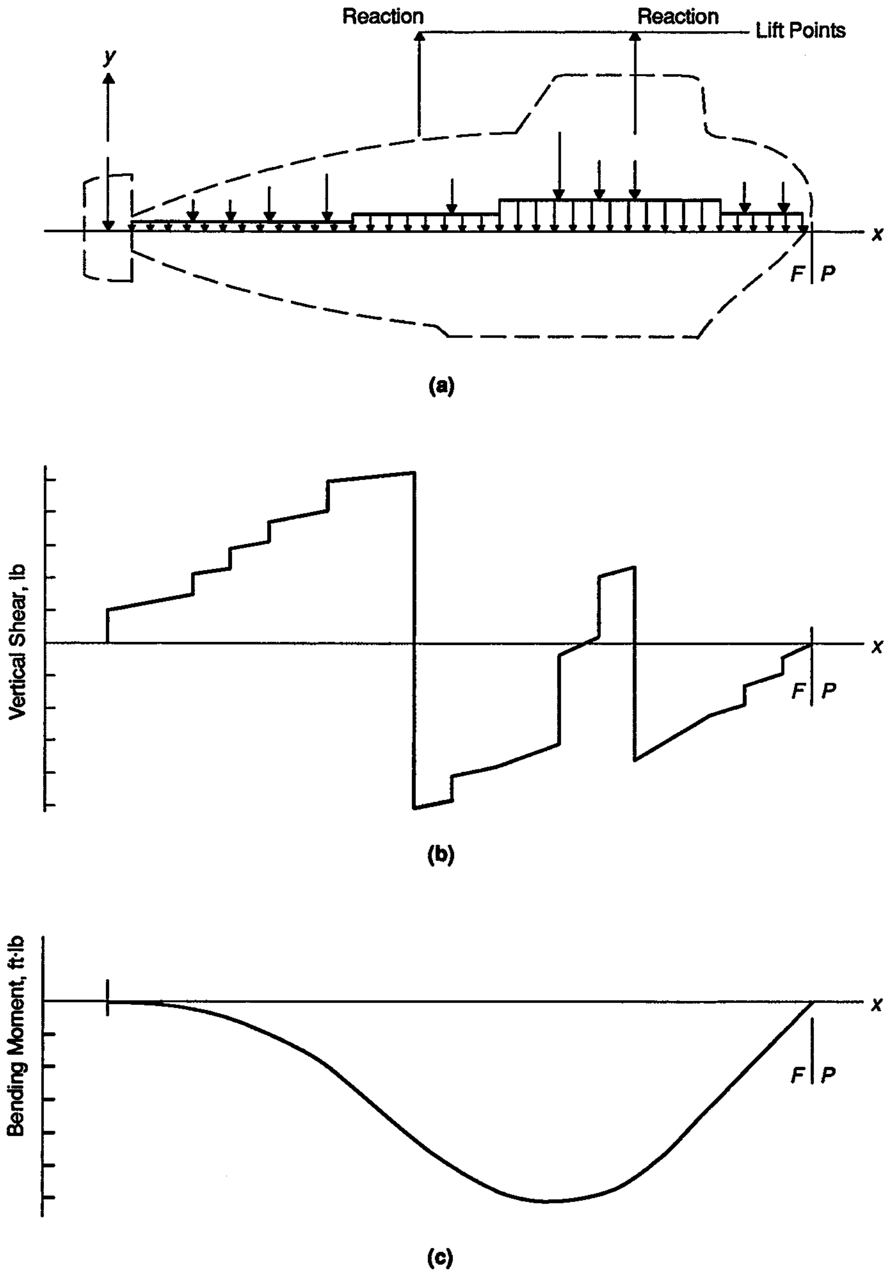


Fig. 19 Typical (*a*) load-lift, (*b*) shear, and (*c*) bending moment diagrams

is greater freedom for arranging equipment and, in general, smaller, lighter foundations result.

A distinctive advantage of the framed structure is that it permits removal of any or all portions of the shell for access to enclosed equipment, without affecting the overall strength of the exostructure. The shell serves only to provide a protective fairing for the vehicle, and to transmit hydrodynamic, wave-slap and impact forces to the framework. For these reasons, the shell usually is composed of numerous segments or panels, which are attached to the stiffeners by mechanical fasteners for ease of removal.

Since the shell does not contribute to the overall strength of the exostructure, as it does in the stiffened shell type of construction, there is little reason to construct the shell and framework of the same materials and, in fact, the two seldom are. Frequently, the shell is made of a glass-reinforced plastic, while the framing is made of a suitable metal, for example, a marine grade of aluminum. While both materials possess comparable weight/strength characteristics, as well as other requisite properties, there are two distinguishing attributes that dictate their selection, or that of similar combinations (that is,

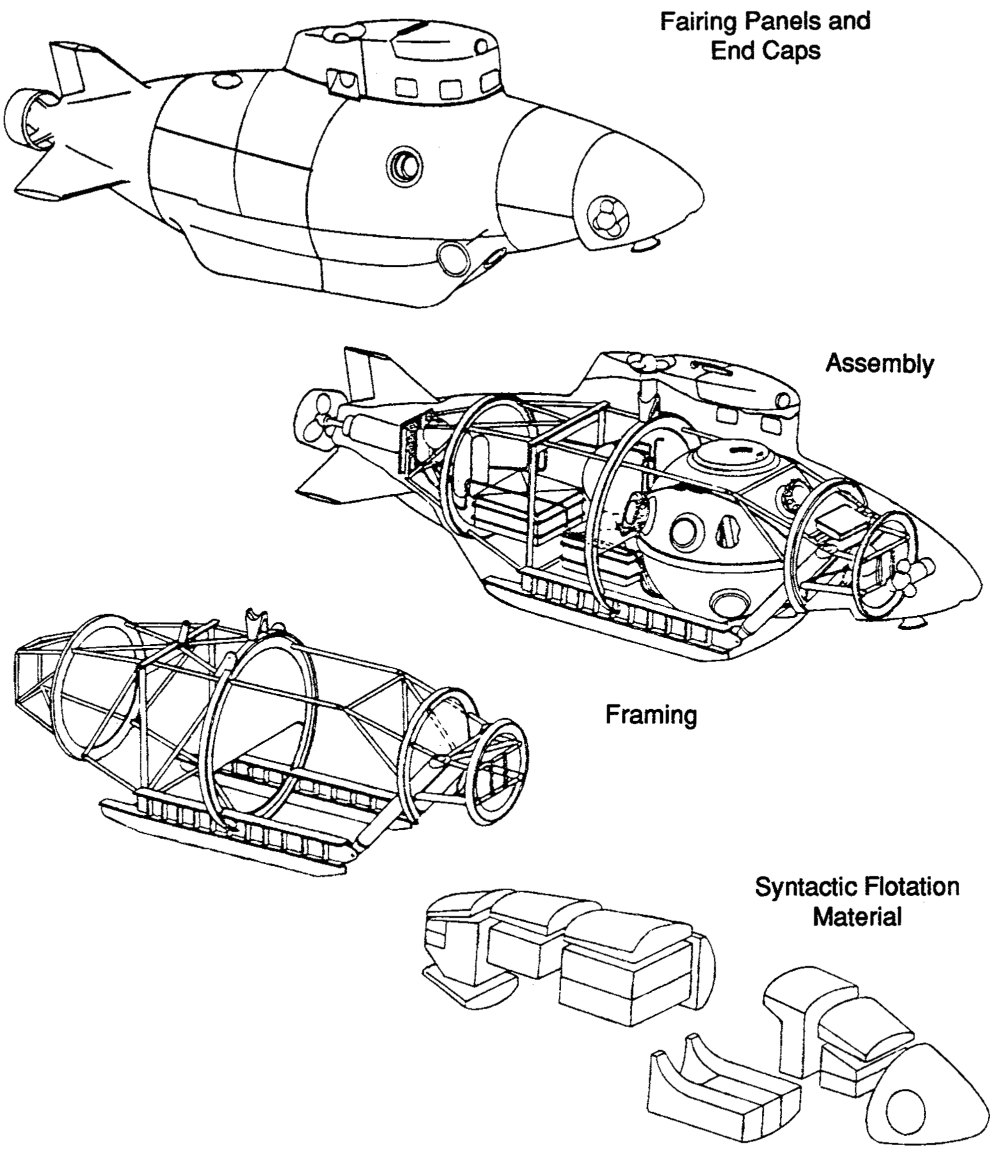


Fig. 20 Framed exostructure

nonmetallic shell with metal framing) for exostructures: ease of fabrication and ease of attachment.

The process by which glass-reinforced plastic (GRP) structural elements normally are produced for a single structure (that is, laying up successive plies of resin-impregnated glass cloth on low-cost molds formed to the finished shape, and then curing the laminate) makes this material particularly well-suited to the forming of panels of compound curvature and other shapes that are difficult to build from metal. Such laminates also offer the service life advantages of being impervious to corrosion and being fairly easy to repair. Metals, on the other hand, offer economic advantages for the fabrication of framing because of the availability of preformed, standard rolled or extruded shapes (such as tubes, angles, and I-beams) in a wide range of sizes, and the relative ease of joining (usually by welding) the intersecting structural members. Metal stiffeners also have certain advantages over nonmetallic materials for attaching foundations, namely, the ability to make connections by welding (which is of greater reliability and unit strength than the adhesive bonding of nonmetallics) and the greater local strength of metals (which permits mechanical connections with the fewest fasteners). Another characteristic of the structural metals is their superior ductility and fracture resistance. In view of the largely unpredictable (to any great degree of accuracy) nature and severity of loading conditions that might be encountered by a submersible vehicle, the plastic reserve strength afforded by a metal framing system may prove attractive.

Chapter VII
Submersible Vehicle Support Systems

B. Murphy and S. Hannuksela

1. Overview

THIS CHAPTER describes the principal support systems required for a submersible vehicle operation. In general, support systems can be defined as all systems, exclusive of the submersible itself, which are required to perform the mission. A submersible vehicle cannot operate independently; it must be transported, launched, recovered, refueled, guided, positioned, maintained, and repaired. The following sections of this chapter will concentrate on transportation, navigation, and maintenance/repair support systems.

The design of a submersible is the main concern of "Submersible Vehicle Systems Design." However, the vehicle is just one element, albeit the key element, in the overall system. The capabilities and limitation of all elements must be reviewed in the process of developing a safe, economic, and effective submersible operation. At the outset of a submersible-system design, the designer must identify which support elements will have to be designed, which will be adapted by modifying existing ones, and which will be utilized as is. Based on economics, the last two approaches are generally preferred. The mission internal design constraints imposed on the submersible design by the support systems are very dependent on the course chosen.

Section 2 of this chapter discusses land, sea, and air transportation systems. The information given on land and air transportation is in part derived from work performed on the U.S. Navy's Deep Submergence Rescue Vehicle program. Section 3 outlines the characteristics to be found in a maintenance-and-repair facility. It is based on typical capabilities required to support a manned-submersible operation.

Section 4 describes the principles of operation and gives the characteristics of the major aids available for navigation and positioning. Until the NAVSTAR/GPS worldwide, high-accuracy, navigation system reaches full capability, the designer responsible for the choice of submersible support system navigation must choose from the numerous navigational aids available. Three hyperbolic radio navigation aids, Decca, Loran C, and Omega are covered in Section 4.1 and Electromagnetic Distance Measuring systems used in offshore locations in Section 4.2. Descriptions of the two satellite navigation aids, Transit and GPS, are found in Section 4.3. A number of underwater acoustic navigation and positioning systems are presented in Section 4.4. Beacon navigation systems and velocity sonars which utilize acoustic energy measure range and velocity, respectively.

Section 4 ends with a discussion on inertial navigation. An introductory description of inertial components, gimbals, and the fundamental principles on which inertial navigation is based is contained in this section.

2. Transportation Systems

2.1 Landborne Transportation Systems[1]

Implicit in the overall requirements for any system utilizing a submersible is the requirement for a ground transportation system. This is evident from the fact that the vehicle must be moved from a maintenance or home-port facility, where it is normally serviced, to the dock side of a support ship or to the loading terminal of some other intermediate transportation system.

In the development of a new submersible system, the mission requirements may specify that the vehicle will be ground transportable and may even identify the ground-transport vehicle. Alternatively, the ground-transport vehicle may have to be specifically designed to meet the overall mission require-

[1]The material on the DSRV land transportation system was drawn from studies performed by Lockheed Missiles and Space Company and from personal discussions with Mr. R. Korte of that company.

ments. In each of these cases, the ground-transportation equipment will generate mission internal design constraints on the submersible design, which will now be discussed.

We will first look at transportation methods for a small submersible which must be moved from its home base which is a short distance from dockside. For such an operation, consideration should be given to the simplest, safest means of transporting the vehicle, while minimizing personnel and equipment hazards. Each of the following transportation methods imposes different constraints on the design of the submersible's internal system:

- towing the submersible on a maintenance cradle that is equipped with wheels,
- moving it with a mobile crane, and
- moving it on a tractor trailer.

The first method requires that the vehicle exostructure be designed to interface with a cradle, and to distribute any loads that result during transportation on the cradle. The second requires that the vehicle exostructure be designed with single or multiple lift points. (Lift points may be part of the submersible design requirements for at-sea launch and recovery operations.) As an alternative to lift points, the vehicle may be designed to interface with a lifting sling or frame which places a different design constraint on the exostructure. The last method requires either that the submersible vehicle have the ability to sit on a flat surface if, for example, it has skids, or that a cradle be provided on which the vehicle can sit when it is placed on a trailer. Depending on the distance it must be transported in this last case, tie-down points may be required between the vehicle, cradle, and trailer. Figure 1 shows the *Alvin* submersible, operated by the Woods Hole Oceanographic Institution, being transported from the maintenance area to dockside. For transportation of such a short distance, no special tie-downs are required.

In analyzing the overall home base transportation requirements, the designer should also review the requirements for maintenance, handling, and lifting fixtures to determine commonality of functions. By doing this, it may be possible to identify fixtures that can also be used as part of the home base transportation equipment.

For larger submersibles up to the medium class size of approximately 25 tons, transportation over roads can be accomplished by using a low-boy or flatbed trailer pulled by a tractor truck (henceforth, tractor). Standard, heavy-duty tandem-axle flatbeds are available which come in lengths from 35 to 45 ft and are rated for loads to 30 tons. Single, tandem, and triple-axle low-boys have capacity ratings in the 15- to 60-ton range. Looking at a worldwide directory of manned submersibles in use or under development, one finds that of the 91 submersibles with listed weights, 16 are over 25 tons; in this latter group only 9 are over 60 tons. For most of the

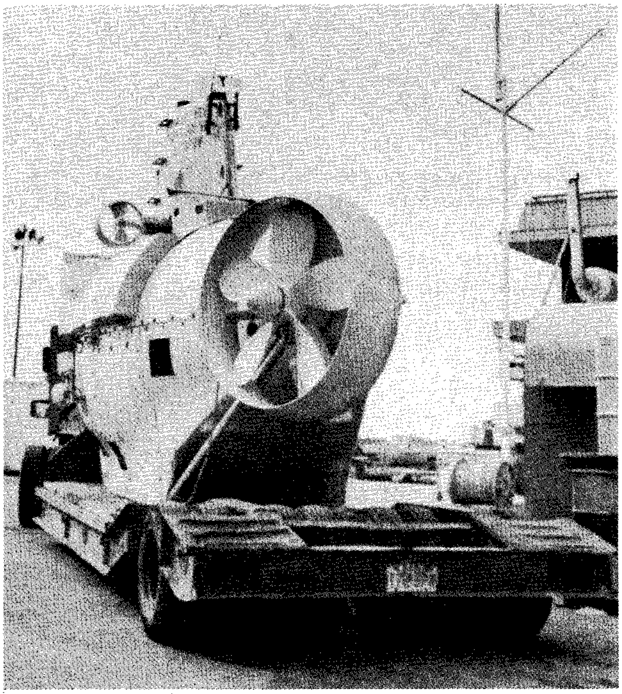


Fig. 1 Ground transportation of the submersible *Alvin*. (Courtesy of WHOI)

submersible community, then, the tractor trailer can fulfill the ground-transportation requirement with available equipment. Again, the design constraints on the submersibles are for the exostructure to be designed with enough strength and fittings to interface with any required cradles and tie-downs while mounted on a trailer. The designer must consider the shock, vibration, and acceleration environment induced while the submersible is being transported on a trailer. For a guide, the design values for the Deep-Submergence Rescue Vehicles (DSRVs) are given in Table 1.

The exostructure designers will use the transportation environment parameters during the stress and vibration analysis of the submersible's framework.

Table 1 Truck transportation shock, vibration, and acceleration environment

Load	Direction[a]	Amplitude, g	Duration, s	Frequency, Hz
Shock	Vertical[b]	+2, −2	0.03	—
	Transverse	+0.5, −0.5	0.03	—
	Longitudinal	+3, −1.5	0.03	—
Vibration		±2.5	—	2 to 200
Acceleration	Vertical[b]	+3, −1	—	—
	Transverse	+0.5, −0.5	—	—
	Longitudinal	+3, −1	—	—

[a]Vertical forces are positive (+) in the downward direction, transverse forces are positive (+) to starboard, and longitudinal forces are positive (+) in the forward direction.
[b]Includes acceleration due to gravity.

They will consider these in the design of foundations used for equipment mounting and will use them to generate the requirements for shock and vibration isolation mounts to protect sensitive equipment.

The transportation environment parameters are also used in the design of a tie-down system that may be required for ground transportation.

Each state or nation where a submersible may be transported has its own regulations governing permissible weights and dimensions of vehicles that traverse its roads and bridges, all of which must be considered in the design of the submersible's transportation system.

Table 2 lists the values for typical parameters. When the transportation of the medium-size submersibles requires any of these parameters being exceeded, special permits can be obtained from the local authorities. Oversize loads or overweight loads, or both, place restrictions on travel such as

- Speed not to exceed a certain value
- Travel allowed only during daylight hours and during weekdays
- Travel allowed only on certain roads and bridges
- A requirement for vehicle escorts front or rear or both.

These restrictions are more the concern of a submersible owner, but the system designer should be aware of them. If the owner rents the tractor trailer from a common carrier, the carrier takes care of all these arrangements. However, when the gross combined weight exceeds approximately 60 tons, the local authorities will require load analysis before granting a permit.

In large-submersible design, the considerations of permissible sizes and weights may become external design constraints. For example, if the submersible is required to be transported by truck, its dimensions will have to be such that it can pass under all overpasses and over all bridges en route. Large submersibles heavier than 25 tons also generate the requirements for special trailer rigs. Here, the system designer must make trade-off studies on the trailer complexity and costs versus the impact on the submersible design.

A special trailer must be designed to conform to all applicable regulations governing such items as bumpers, clearance, and marker, stop, and turn-signal lights. If a trailer must interface with a standard tractor, then it must have a correctly positioned king-pin which will connect it to the fifth wheel of the tractor. In addition, typical hardware such as glad-hand (air brakes) and electrical connectors may be required to interface with the mating connectors on the tractor. A special trailer can be configured as either a semi- or a full trailer. The latter configuration with a forward axle set is used to distribute the trailer load. Again, in order for existing tractors to be used, the combined load of the trailer should not exceed the maximum gross vehicle weight (GVW) and gross combined weight (GCW) ratings of the truck.

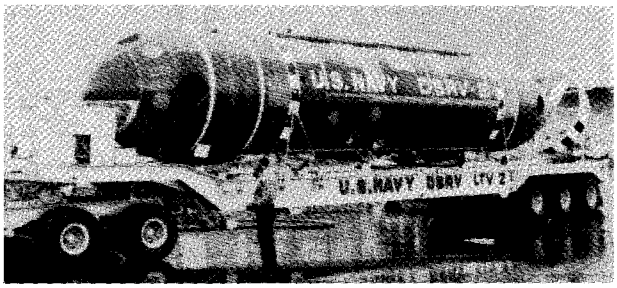


Fig. 2 DSRV land transportation vehicle. (Courtesy of Lockheed Missile and Space Co.)

The trailer developed for the DSRV will be described as an example of a special trailer design. This vehicle, shown in Fig. 2, is called the Land Transport Vehicle (LTV). Pertinent information on the LTV is given in Table 3. This vehicle is made up of a semi-trailer and a jeep dolly which provides a means of load distribution and of attaching the trailer to tractors having various heights of fifth wheels. The LTV has the following features:

- can carry DSRV (70 000 lb), 49 ft long,
- can use primary and secondary paved highways,
- is compatible with C-141 aircraft cargo deck system,
- has height and position control and support system capability for DSRV aircraft on/off loading,
- is equipped with loading winch,
- has power unit to provide compressed air for brake system when required, and hydraulic power for the winch,

Table 2 Permissible truck dimensions and weights

Parameter	Value
Overall length	55 ft (16.8 m)
Overall width	8 ft (2.4 m)
Overall height	13.5 ft (4.1 m)
Gross combined weight	80 000 lb (36 290 kg)
Single axle weight	22 400 lb (10 161 kg)
Tandem axle weight[a]	18 000 lb (8165 kg)

[a]Spacing less than 6 ft.

Table 3 Land transport vehicle parameters[a]

Parameter	Value
Weight	44 500 lb (20 185 kg)
Length (overall adjustable)	71 to 73 ft (21.6 to 22.3 m)
Width (deck)	112 in. (284 cm)
Width (tire to tire)	120 in. (305 cm)
Height (clearance)	100 in. (254 cm)
Ladened height (to top of deck rolls)	55 in. (140 cm)
Kingpin ground height (adjustable)	50 to 60 in. (127 to 152 cm)
Highway payload	72 000 lb (32 659 kg)

[a]Courtesy of Lockheed Missile and Space Company

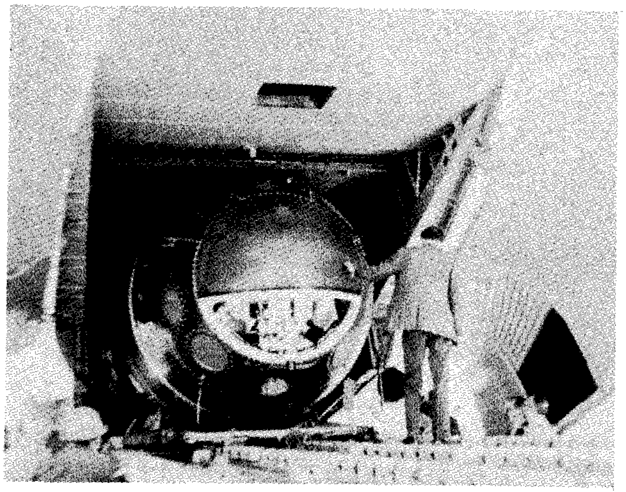


Fig. 3 Land transportation vehicle roller system. (Courtesy of Lockheed Missile and Space Co.)

- can interface with different tractors, and
- has air suspension system for road environment isolation.

The roller system on the LTV (Fig. 3) is compatible with the rollers on the cargo deck of the C-141.

Figure 4 depicts a 16-ton submersible mounted on a low-boy rated for a 25-ton load. Calculations are given in the figure to show the method of finding the axle loads for the tractor and the trailer.

The submersible designer and system engineer must be aware of the constraints imposed by the requirements for land transportation of the vehicle. These parameters form inputs to the design process in arriving at a good system configuration. Adherence to the design considerations of transportation will result in full compatibility between the submersible and its ground transportation system.

2.2 Seaborne Transportation and Handling Systems[2]

2.2.1 Introduction

This subsection describes the types and requirements of support vessels for manned submersibles and methods of launching and retrieving them at sea.

As a general rule, all submersibles require a support platform when operating in the open ocean. These platforms can be either surface or subsurface craft; both are in use today. The support vessel provides many functions for submersible operations. The major support functions provided are to [1]

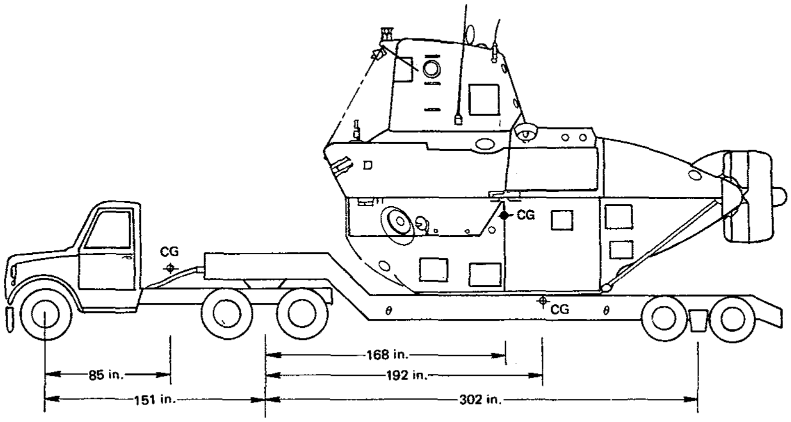
- transport (aboard or tow) to dive site
- launch/retrieve at dive site
- accommodate support personnel and diving party
- carry maintenance and repair equipment and provide sheltered work areas
- communicate with, track, and direct the submersible during submergence

[2]See author's note, page 379.

	Weight (lb)
Tractor	– 9,849
Submersible	– 32,000
Low Boy	– 10,200
Total	– 52,049

Tractor Rating – 43,000 GVW
60,000 GCW

Low Boy Rating – 25 ton



$$\text{Low Boy Rear Axle Load} = \frac{168\text{ in.} \times 32{,}000\text{ lb} + 192\text{ in.} \times 10{,}200\text{ lb}}{302\text{ in.}} = 24{,}286\text{ lb}$$

$$\text{Tractor Rear Axle Load} = \frac{85\text{ in.} \times 9{,}849\text{ lb}}{151\text{ in.}} + 32{,}000\text{ lb} + 10{,}200\text{ lb} - 24{,}286\text{ lb} = 23{,}428\text{ lb}$$

$$\text{Tractor Front Axle Load} = 9{,}849 - \frac{85\text{ in.} \times 9{,}848\text{ lb}}{151\text{ in.}} = 4{,}335\text{ lb}$$

Total = 52,049 lb

Fig. 4 Example of trailer loads

- monitor weather and clear traffic for surfacing
- provide means of crew transport to and from the submersible
- carry scientific/engineering instrumentation and provide storage area for samples and work area for data reduction
- conduct supplementary studies for functions in concert with the submersible before, during, or after the dive
- provide a safe haven in the event of emergency
- provide rescue/recovery facilities

It is obvious from the above functions that the support vessel is an integral part of the manned submersible system.

2.2.2 *Support Vessel Types and Configurations*

Ships-of-opportunity versus dedicated support ships—Support platforms are either ships-of-opportunity or dedicated support ships. When ships-of-opportunity are used, the submersible system is temporarily installed on an available ship. On dedicated support ships the submersible support equipment is permanently installed. Costs and mission duration are important elements in deciding which to use [2].

Overall costs may be considerably less with a ship-of-opportunity, though the efficiency of the total system may be reduced. Navigation and control systems for the submersible's underwater operations can be difficult to integrate into a ship-of-opportunity. Likewise, the time and costs of outfitting the ship with the submersible's equipment before each mission and then dismantling it when the mission is completed must be considered. In addition, the use of some arbitrary launch and retrieval system can lead to dangerous situations and may reduce the operable sea state limitations.

With a dedicated support vessel, all considerations peculiar to a submersible can be accommodated within the ship. This includes adequate room

Table 4 Characteristics of some dedicated, manned-submersible support ships

Ship	Submersible	Operator	Length Overall, ft	Beam, ft	Draft, ft	Gross Tonnage, tons	Cruise Speed, knots	Cruise Range	Launch/ Recovery Method	Ship's Crew	Support Accomodations
Aloha	*Mermaid II*	International Underwater Contractors	143	32	8.3	165	12	6000 nmi 30 days	A-frame	6	16
Atlantis II	*Alvin*	WHOI[a]	210	44	16	1529 (Displ.- 2300)	12	9000 nmi 30 days	A-frame	25	25
Calypso	*Divng saucer*	Cousteau Society	139	25	9.8	394 (Displ.)	10	2640 nmi 11 days	Articulated Crane	26[b]	26[b]
Edwin Link (ex-*R. V. Johnson*)	*Johnson-Sea-Link I* or *II*	HBOI[c]	125	27	12	239	12	3000 nmi 14 days	Articulated Crane	11	15
Nadir	*Nautile*	IFREMER[d]	183	39	15.5	2008	12	7500 nmi	A-frame	19	15
Natsushima	*Shinkai 2000*	JAMSTEC[e]	220	42.8	12.5	1523	12	8000 nmi 30 days	A-frame	30	25
Pandora II	*Pisces IV*	C. R. Ward & Associates	197	45	18	1500	10.5	25 200 nmi 100 days	A-frame	18	18
Pigeon and *Ortolan* (ASR-21 Class)	DSRVs *Mystic* or *Avalon*	U.S. Navy	251	86	21.3	4200 (Displ.)	15	INA[f]	Gantry Crane/- Elevator	188	38
Seward Johnson	*Johnson-Sea-Link I* or *II*	HBOI[c]	175.75	36	12	295	14	8000 nmi	A-frame	10	20
Transquest	*Sea Cliff, Turtle, Mystic, Avalon*	Lockheed/ U.S. Navy	100	40	7	392	6	1100 nmi 10 days	Elevator	7	23
DSVSS[g] *Laney Chouest*	*Sea Cliff, Turtle*	Edison Chouest Offshore/USN	240	62	13.5	499	12	8000 nmi 90 days	A-frame	12	40

[a]Woods Hole Oceanographic Institute.
[b]Total accomodations.
[c]Harbor Branch Oceanographic Institution.
[d]French Institute of Research for Exploration of the Sea.
[e]Japan Marine Science and Technology Center.
[f]Information not available.
[g]Deep Submergence Vehicle Support Ship.

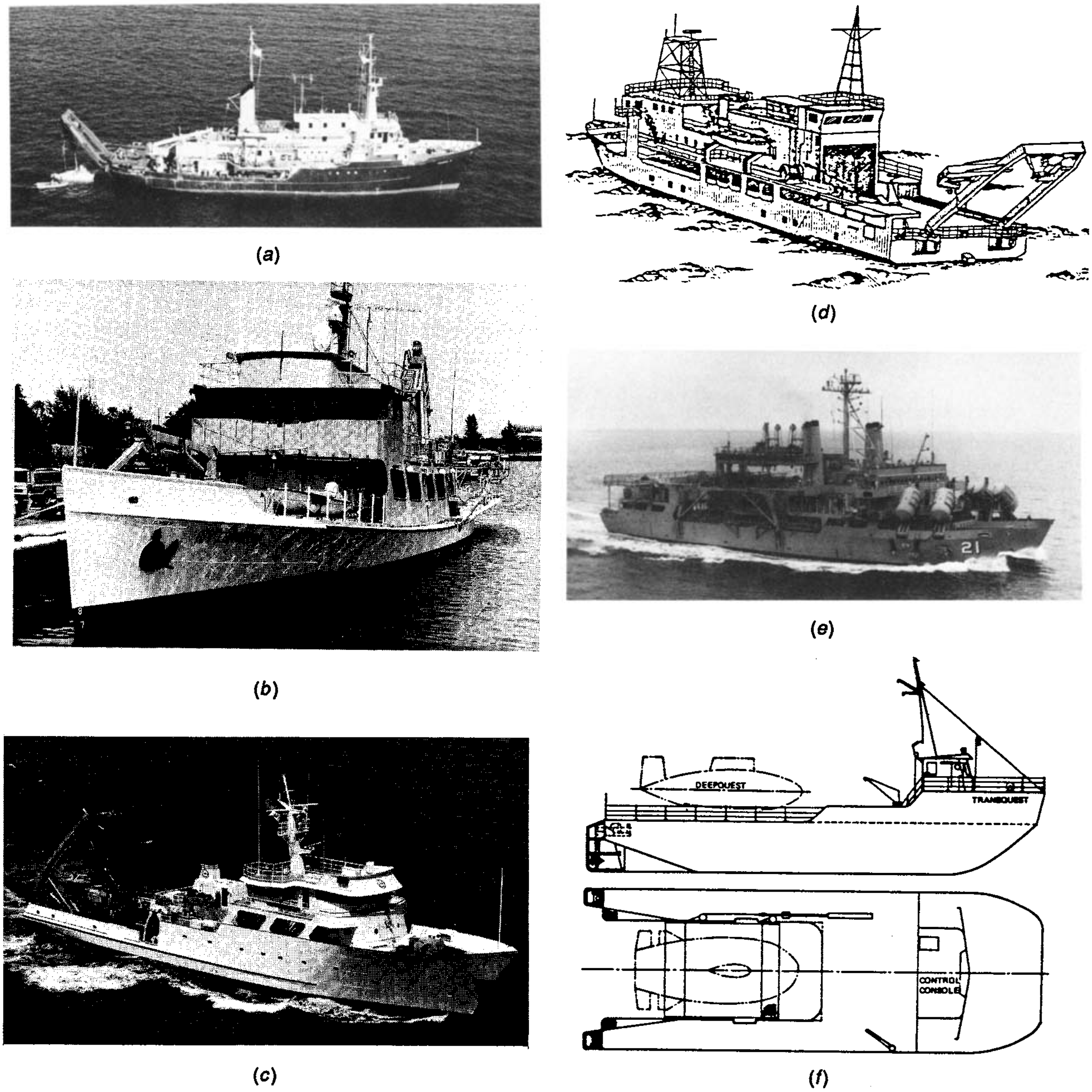
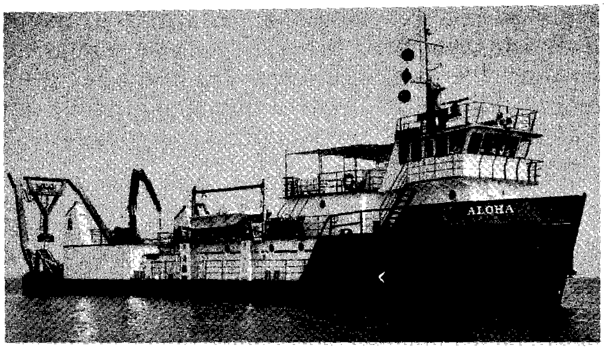


Fig. 5 Support ships: (***a***) R/V *Atlantis II*; (***b***) R/V *Edwin Link*; (***c***) R/V *Seward Johnson*; (***d***) R/V *Natsushima*; (***e***) USS *Pigeon*; (***f***) schematic of R/V *Transquest*; (***g***) R/V *Aloha*; (***h***) R/V *Pandora II*

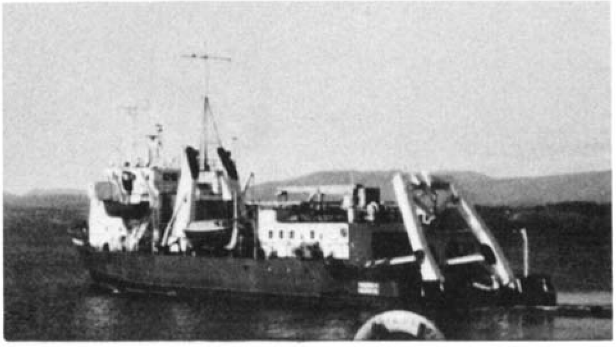
for the submersible, its crew, and the associated equipment and supplies. This is the greatest advantage of dedicated support ships.

Most manned submersibles in use today operate from a dedicated support ship. Table 4 gives some characteristics of selected ships. The same ships are shown in Fig. 5. (For a more complete listing of support ships, see [3].) Of the ships listed in Table 4, the Research Vessel (R/V) *Natsushima*, USS *Pigeon* (ASR-21) and USS *Ortolan* (ASR-22), R/V *Seward Johnson*, and R/V *Transquest* were specifically designed and built to support a submersible. The *Pigeon* and *Ortolan* are U.S. Navy submarine rescue ships constructed to carry the DSRVs and to support deep-ocean diving operations. The *Transquest* was originally built to transport the *Deep Quest*, but now primarily supports the U.S. Navy's Deep-Submergence Vehicles (DSVs) and DSRVs for close-to-home operations.

Few manned submersibles in use today operate from ships-of-opportunity. However, the *Deep Rover* (Fig. 6), developed by Deep Ocean Engineering, Inc. was specifically designed to be readily accommodated on a ship-of-opportunity. It is a one-man sub-



(*g*)



(*h*)

Fig. 5 ***continued***

mersible that weighs 6500 lb in air and can be transported by air, sea, or land in a standard 8 by 8 by 10 ft container. *Deep Rover* has been successfully launched and recovered using the submersible Launch, Retrieval, and Transport (LRT-12A) platform (Fig. 7), an A-frame on the R/V *Egabrag III* (Fig. 8), a crane on the R/V *Wecoma* (Fig. 9), and through a moon pool on the Research Platform *Orb* (Fig. 10).

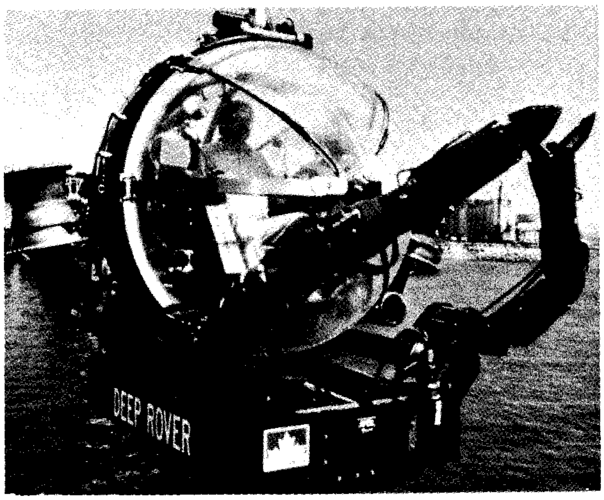


Fig. 6 *Deep Rover* submersible

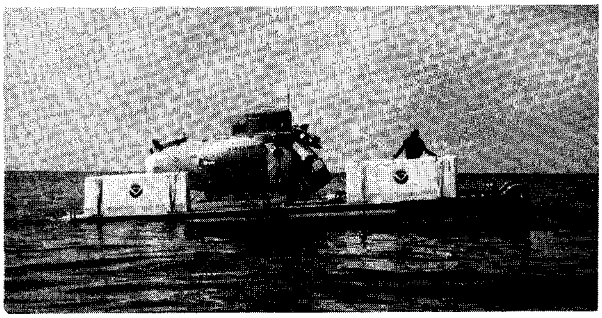


Fig. 7 Launch, recovery, and transport—12-ton aluminum (LRT-12A). (Courtesy of NOAA's Office of Undersea Research)

Hullforms—The following is provided as an overview of the advantages and disadvantages of existing hullforms.

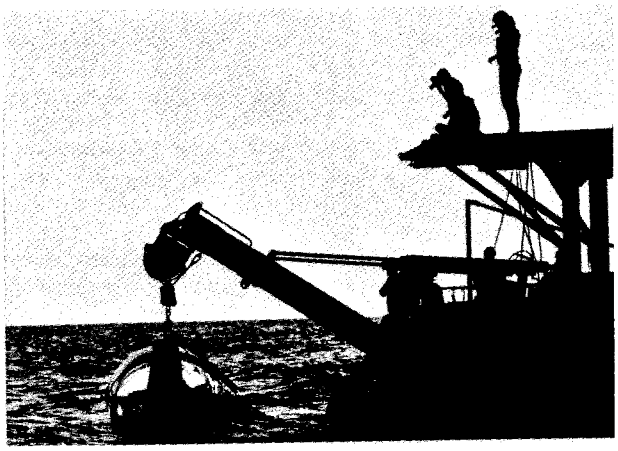


Fig. 8 Launching the one-man submersible *Deep Rover* from the support ship *Egabrag III*

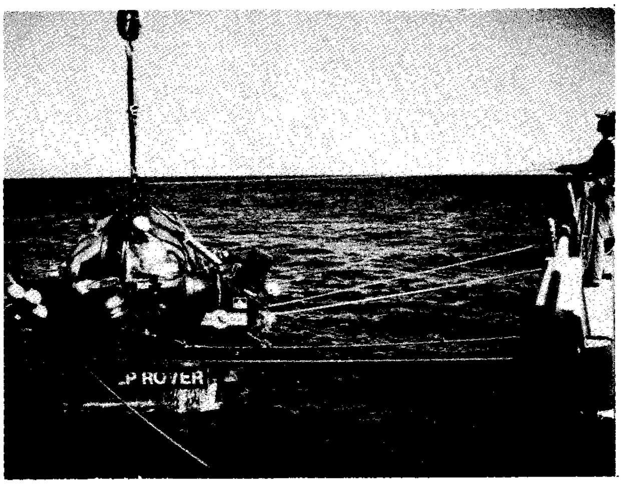


Fig. 9 Deployment of *Deep Rover* using crane from the vessel R/V *Wecoma*

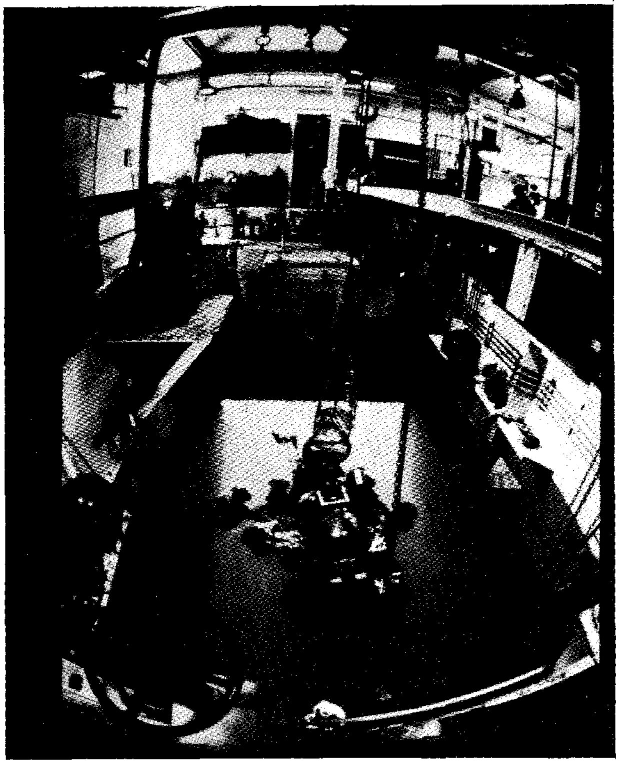


Fig. 10 Deployment of *Deep Rover* using moon pool on the R/V *Orb*

Monohull: Monohull ships have been, and continue to be, the most common type of ship used for manned submersible support. The main reason for this is the availability of existing ships with acceptable configurations or configurations that require reasonable modifications [4]. Their main advantages are the lower costs and the availability of technical expertise required to design, construct, or modify them and their ability to handle considerably greater payloads than multihull ships. The disadvantages of monohulls include their seakeeping performance and the lack of flexibility in arranging topside work spaces.

Catamaran: The ASR-21 class ships, the *Pigeon* and *Ortolan*, are prime examples of this type of hullform. The DSRVs are launched and retrieved between the hulls.

The ASR-21 class ships use an elevator handling system (on which the DSRV is capable of landing while submerged) with a gantry crane lift system. While the ASR-21 class ships were designed specifically for supporting the DSRVs, they have not resulted in significant advances in support ship or handling system design for manned submersibles [4]. The advantages of this type of ship are that they have larger deck areas, and, in average-to-lower sea states, they tend to be more stable and sea-kindly. Their disadvantages include the limited amount of payload that can be carried and limited available space below deck in the hulls for working and living. These space limitations are inherent characteristics since the beam of each catamaran hull is a good deal less than one-half the maximum beam of the ship. Other disadvantages include their seakeeping abilities in high sea states, excess drag caused by the two hulls, and the potential for structural problems associated with the cross structure between the hulls.

Small Waterplane Area Twin Hull (SWATH): There are two SWATH type vessels currently utilized as manned submersible support ships: the *Twin Drill* (Fig. 11), and the *Kaiyo* (Fig. 12). Neither is a dedicated support ship, but each was designed as a

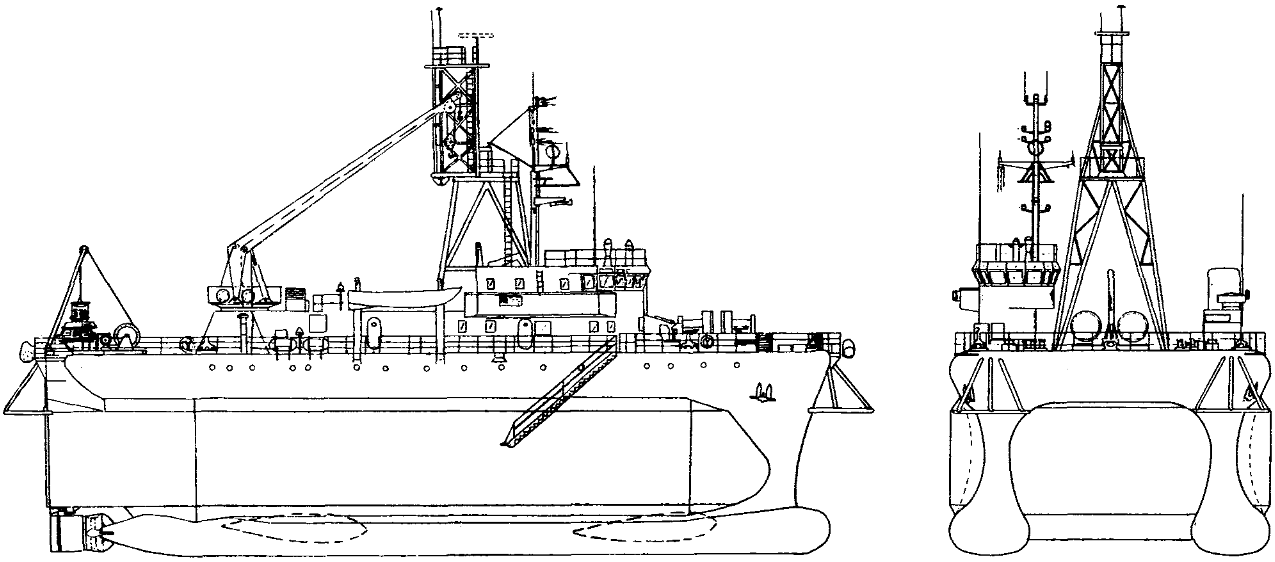


Fig. 11 Manned-submersible support ship *Twin Drill*

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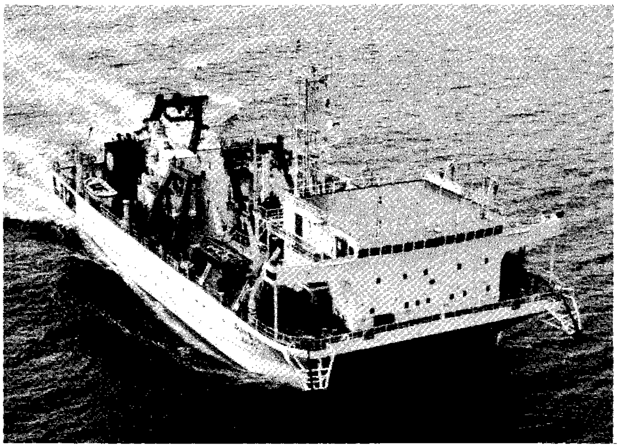

Fig. 12 Manned-submersible support ship *Kaiyo*

multipurpose ship capable of performing all facets of underwater work. The *Twin Drill*, which is owned and operated by International Underwater Contractors, can be outfitted with the *Pisces VI* or the *Beaver Mark IV* submersibles. The *Kaiyo*, owned and operated by the Japan Marine Science and Technology Center (JAMSTEC), can accommodate the *Shinkai 2000* submersible, and also a next-generation submersible currently being developed, the *Shinkai 6500*. The greatest assets of SWATH ships are their seakeeping abilities in all levels of sea states, and the ample on-deck working areas for their size. Their disadvantages include increased drag due to the underwater hulls, complex structural problems, and a sensitivity to the addition and removal of weights. Other drawbacks to SWATH ships are that they may be somewhat more costly than their monohull counterparts and that their freeboard is relatively large [5].

SeaTrailer™: A novel design for a dedicated submersible support platform for use with a ship-of-opportunity is the SeaTrailer (Fig. 13). SeaTrailer is a non-propelled partial column stabilized barge that is towed by a ship-of-opportunity. The towing hitch can be rapidly mounted on any tow vessel of convenience with structural clearance and sufficient stability. Close coupling to the tow vessel allows direct personnel access at all times.

SeaTrailer-type support platform dimensions can be varied to accommodate the desired payload; and displacement, column size, and spacing can be chosen so that the natural periods of motion will exceed normal wave periods. The heave period selected is about 12 s for SeaTrailer to ride and clear larger waves.

SeaTrailer's transit draft is about the same as that of the towing vessel to reduce towing resistance and permit usage in shallow water ports. The trailer is ballasted onto the columns upon arrival at the work site.

Submarine: Launching and recovering submersibles from a submerged submarine is extremely attractive because it enables both operations to be conducted independent of the effects of surface weather conditions and sea states. However, the very specialized crew that is required, the maneuvering capabilities necessary for the submersible to take-off and land on a submerged submarine, the lack of work and storage spaces, and the operational costs make a submarine impractical for all but unique submersible mission requirements. The only manned submersibles capable of working from a submarine are submarine rescue vehicles. The U.S. Navy's *Mystic* (DSRV1) and *Avalon* (DSRV2) are shown in Fig. 14. A civilian submarine support vessel design was developed in the 1970s, but was never constructed. It is discussed in detail later in this section when handling systems are presented.

2.2.3 Requirements

A submersible support vessel must carry certain standard equipment to allow it to operate in the open ocean without benefit of shore support. These are in addition to the normal complement of navigation (see Section 4 of this chapter), radar, ship-to-shore communications, and other equipment necessary for ships and boats to operate at sea. The following is a list of components specifically carried for submersible support [1]:

- battery chargers
- air compressors
- scuba equipment
- submersible navigation/tracking system
- small boat or rubber raft
- two-way radio
- launch/retrieval apparatus
- underwater telephone
- submersible system spares and supplies
- life-support replenishment
- standard electronic test equipment

Submersible system spares and supplies deserve special attention. Typically, submersible operations are conducted in a remote offshore location. This requires the support vessel to carry enough spare parts to cover almost every problem. Spare parts for the manipulators and tools, electronic/acoustic in-

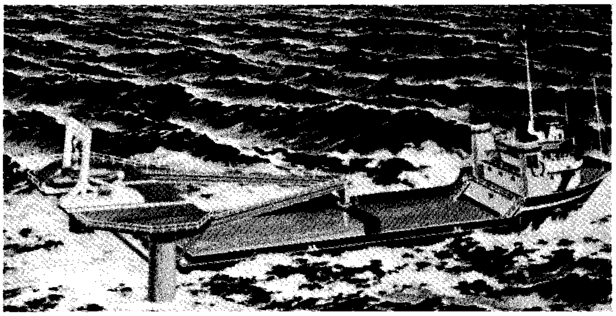

Fig. 13 SeaTrailer™

Chapter VIII
Design and Operating Safety

R. J. Dzikowski and J. H. Purcell

1. Overview

All submersible vehicle designs must embody operational safety considerations. From the selection of the hull structural material to the publication of an operational manual, design process decisions must consider the impact of all choices on vehicle reliability and particularly on the safety of personnel who will operate and occupy the submersibles.

Safety considerations touch all elements of a submersible's design. A submersible meeting key mission requirements relating to speed and diving and surfacing times must consider stability while on the surface, while submerged, and while passing through the sea-air interface at various sea states. A particular mission may demand minimizing time on station, thus requiring rapid handling of the submersible by the mothership. Submersibles which are launched from support platforms must be able to be lowered onto and lifted from the sea surface safely as well as expeditiously.

This chapter, Sections 1 through 3, provides a general discussion of safety considerations applicable to the submersible design process and to at-sea operation of these vehicles, recognizing that very specific classification agency requirements exist and need to be followed if a submersible is to be certified classified for operation at sea. For certification classification of a submersible, its design, fabrication, and testing must be in compliance with one of the sets of classification agency rules, such as Lloyd's Register of Shipping, Det Norske Veritas, or the American Bureau of Shipping. In addition, governmental agencies such as the U.S. Coast Guard have rules which govern the operation of equipment, the requirements for installed equipment, and the qualifications for operating personnel. Submersible characteristics relating to safety imposed by classification agencies and a detailed discussion of the U.S. Navy's role in establishing submersible safety programs are discussed in Sections 4 and 5 of this chapter.

Safety considerations also apply to the submerged endurance of the submersible. The amount of breathable gas on board must be sufficient for the duration of the designed mission time, plus an excess for emergencies. If the submersible is designed for carrying lock-out divers, safety considerations for design of hyperbaric chambers must be considered. Divers may be operating at ambient pressure, requiring additional gases which are of the proper mix for the intended diving depth. Also, the diver-occupied spaces must have air purification and heating facilities to ensure divers' safety and well-being.

The entire design must have a common goal of safety of personnel and equipment. With the constraint of size, weight, and costs, the challenge is great for the designer to create a submersible meeting the stated mission requirements and at the same time ensure that the end product will be a safe, operational vehicle.

2. General Discussion of Submersible Safety Considerations

In previous chapters the principles relating to the concept design approach have been given. The designer has selected the parameters of the hull structure, the propulsion system, the exostructure configuration, and all the materials to satisfy the mission requirements of the owner/operator of the submersible. All these must now be amalgamated into an efficient operating submersible which will provide inherent safety to itself as well as to the operators and users.

The pressure-resisting structure must perform with adequate safety in repeated cycling to the operating depth. Fatigue analysis of this structure, using finite element analysis programs, may be necessary.

Selected locations where high-stress levels are anticipated must be carefully analyzed. Penetrations, lift points, and other discontinuities and representative troublesome points should receive special attention and analysis. Verification of such analysis by strain gaging during the hydrostatic test should be specified. Where size permits, pressure testing of the instrumented pressure hull and other pressure-resisting structures in a suitable test tank should be considered, particularly if unique structural features are incorporated into the pressure hull design.

The safety of the occupants of the submersible must be of primary interest to the designer/operator/owner of the submersible. Safety can be considered in several ways:

- inherent safety incorporated in the basic design of the submersible,
- special safety systems and components incorporated in the design, and
- operator training.

An example of a submersible's inherent safety would be the choice of factor of safety used in the design and selection of material for the pressure-resistant structure. The ability to release mechanical equipment in case of entanglement is an example of a special safety feature incorporated into the submersible design. And, of course, operators must be trained in safety operations.

The adequacy of the life-support system to provide an atmosphere habitable for humans for long periods of time in case of an emergency must be considered in submersible design. A sufficient emergency onboard oxygen supply and carbon-dioxide-absorbent supply must be carried to provide life support during a rescue effort for a length of time specified by regulating agencies. In addition, emergency equipment, such as recirculating breathing apparatus and special exposure clothing, must be available. These prolong the time the occupants can be sustained while rescue efforts continue.

Reliability of the installed equipment, especially the propulsion, electrical, and sensor equipment, must be given special consideration by the designer. Operator confidence in the reliability of installed equipment is of paramount importance. This confidence can be gained by operator participation in specified equipment/system test programs and well-planned, step-by-step at-sea testing.

Quality-control programs are essential in establishing the inherent safety of a submersible. For example, most submersibles have very little sea water piping and valving inside the pressure boundary; however, all such piping and valving must be subjected to quality-control procedures from initial selection through fabrication, installation, and testing to ensure its integrity. A small, high-pressure leak in a submersible presents a very serious hazard to the occupants. In addition to strength and tightness integrity, the breathing gas piping and valving must have additional quality-control requirements to make certain that the system is clean and oil-free. The use of internal high-pressure gas piping and valves should be minimized in order to reduce the possibility of failure; this can be achieved by providing gas-reducing stations near the hull entry or gas-storage source. Minimizing the extent of piping serving critical systems can improve the inherent safety of the submersible.

Installed safety features should include a means of constantly monitoring the atmosphere inside the submersible for oxygen and carbon dioxide. A high percentage of oxygen can be hazardous in that the combustibility of commonly installed materials increases rapidly in above-normal oxygen mixtures. Similarly, high carbon dioxide content can induce minor but perceptive physical and mental changes in occupants if they are exposed to it for several hours.

There is a great number of electrical circuits and sensors present in a submersible, all of which must be protected from high-humidity conditions since these conditions pose a threat of an electrical short and thus an electrical fire. It is extremely important for the designer to minimize the potential for these incidents because the fire-fighting materials themselves may contaminate the atmosphere, in addition to the gas and smoke from the fire itself. Electrical-kill switches should be considered which remove all power, but maintain emergency lighting. Fire-extinguishing equipment must be selected which will extinguish the fire but not severely contaminate the atmosphere or prevent the occupants from restoring power to the unaffected circuits in order to return the submersible to the surface safely.

The designer must give careful consideration to the preparation of a comprehensive test program that tests systems and components dockside in preparation for the submersible's sea trials. A comprehensive dive plan and a rescue plan is required. Crew training in operation and emergency procedures is a prerequisite so that the operational testing at sea can be done efficiently with minimum risk to the safety of the embarked personnel and the submersibles. Any data which are obtained during the initial trials should be analyzed for unpredicted anomalies which could impair safety in any future at-sea operations.

At an early date, the owner/operators should select the classification agency that will certify the vehicle for safe operation at sea. The designer should then direct his or her effort during the design process and in preparing specifications for construction, outfitting, and testing of the vehicle to ensure that the end product will be certified for unrestricted operations within the mission envelope. The certificate issued by the regulatory agency indicates that the minimum requirements of the established rules have been met and assures that the submersible performance is adequate to assure safety of the occupants during normal operations.

3. Safety Considerations for Submersible Systems

3.1 Hull

The pressure-resisting elements of the submersible are normally referred to as pressure hull. This includes the hull penetrations required for electrical and piping services, personnel access hatches, viewports, penetrations for propulsion shafts, and mechanical features required for activation of external devices. Although they are not required for water tightness or pressure-hull strength, the hull may have various attachments such as pads for support of the exostructure, propulsion devices, lift points, drop weights, and other outfitting requirements. These may, however, generate "hard-spots" or stress concentrations that must be considered when analyzing the strength and fatigue resistance of the pressure hull.

3.1.1 Pressure Boundary Structural Integrity

The pressure boundary is the single most critical unit of the entire manned submersible. The weight of the hull contributes between 25 percent to 50 percent of the entire weight of the submersible. For this reason, trade-off studies are made evaluating the weight versus configuration, cost, and material. In each trade-off, the safety aspects of the hull design must be a foremost consideration.

In addition, the trade-offs must consider the ability of the material to be formed, joined, and reinforced without undue degradation of the properties of the selected hull material. Not only must the pressure structure withstand sea pressure, but imposed on it may be point loads caused by the propulsion weight, exostructure, lifting pads, and other attachments. Consideration must be given to the pressure structure being subjected to an impact with the bottom or other solid-type object, which in addition could impose high local loads at design depth. If there is a high probability of hard contact to the pressure-resistant structure, then hull-protection methods should be considered.

The penetrations, such as viewports and hatches, require special protection consideration. Penetrations are discontinuities in the pressure boundary and are considered hard spots. Detailed investigation by finite-element analysis may be required, especially if the overall weight of the hull is critical or if the stresses are approaching the allowable yield strength levels.

Generally, hatches and viewports are designed to seat with external pressure. However, the hatch mechanism and viewports must withstand a slight internal pressure above ambient when the submersible is at or near the surface due to the overpressure that a life-support system may generate or due to a drop in barometric pressure since the hatch was closed at the beginning of the dive.

Those submersibles which have a diver lock-out chamber require additional structural integrity considerations. The lock-out chamber may be at the pressure of the dive lock-out depth when the submersible is brought to the surface. The design must consider a complete reversal of stresses since the lock-out chamber is cycled from surface ambient to deep-depth working pressure and then brought to the surface at this pressure, at which point the lock-out chamber is reduced to surface ambient pressure. The lock-out chamber penetrations must be able to withstand this cycle of pressure, and the viewports and the hatch arrangements therefore must be double-acting, that is, capable of retaining their seating/sealing integrity if exposed to either an external or internal overpressure.

3.1.2 Material Specifications, Inspections, Nondestructive Testing (NDT), and Traceability

The selection of materials and the structural principles pertaining to submersible design have been discussed in Chapters IV and VI. To ensure the proper procurement of pressure-hull material, the designer must provide a specific material specification which should state the chemical and physical properties desired. Unless the material is an alloy or has a chemistry not usually specified in recognized publications, such as those issued by the American Society for Testing and Materials (ASTM), the American National Standards Institute (ANSI), the American Bureau of Standards (ABS), or others, then specifying the material by the standard published designation is usually sufficient. However, additional supplementary requirements should be identified, such as Charpy tests at specified lower temperatures, ultrasonic tests on plates using a specified grid pattern to check for internal defects such as laminations, and X-rays of castings to ensure that they are free of defects; these are all aimed at achieving greater assurance that the material is satisfactory for use in the pressure hull boundary. The documentation required to accompany the material, verifying compliance with the specification in all its requirements, should be clearly identified by the designer. Often a third-party inspection service is specified for use at the plate mill or foundry to verify that the specified inspection and testing has been satisfactorily completed.

The designer should also consider the need for receipt inspection of the material using some random-sample testing for such attributes as chemical and physical properties of the material. The plate or casting identification or both should be verified. This is to ascertain that the material received has been properly marked and is in fact suitable for further fabrication. The cost of such random additional sampling is small compared to the cost of redoing pressure hull structural work as a result of discovering that improper or substandard material has been incorporated.

Receipt inspection of the material is appropriate to see that the surface condition is satisfactory and that shipping damage such as stress-raising scars and gouges are not cause for rejection. Sandblasting and preservation of the structural steel is normally done as soon as the material is received and prior to the inspection for surface roughness.

Traceability of the structural material is extremely important. The plate identification must be transferred to all the cutoff plates and appropriately mapped for future reference and possible use as hull structure. Should it happen that a plate or casting is found to be defective for any reason, the faulted item can be readily identified, removed, and replaced if conditions so warrant. Also, such identification may direct attention to other plates or castings that were produced from the same billet roll or casting lot and thus may require reexamination. If no traceability is available to pinpoint the defective element, then the entire structural assembly is suspect.

3.1.3 Welding Process and Welding Certification

The steel plates and castings specified by the designer should be of weldable grade. The American Welding Society and other similar organizations have published detailed specifications for the qualification of the welding process and welder qualifications. The designer working on hull fabrication should take care to ensure that the fabricator can follow the welding procedures and will have welders properly qualified in accordance with the regulating agencies' requirements to weld the hull materials the designer specifies. This will assure that welders qualified through a test program can produce the minimum strength welds of acceptable quality. Nondestructive testing (NDT) of welds and welder traceability control must be established to allow identification of defective welds to pinpoint the use of improper weld techniques, preheat, etc.

3.1.4 Penetrations, Attachments, and Mechanical Fasteners

Penetrations for piping, electrical, and mechanical shafting must be designed and installed so that the structural integrity is not compromised. Penetrations which are grouped so that they are installed in a compensated plate is one method of accomplishing this. The insert weld is readily X-rayable and film can be easily interpreted and a permanent, auditable record is achieved. Electrical penetrators are specifically designed so that the electrical penetrator does not extrude into the hull. Shafting penetrators are similar to piping penetrators with provisions in the penetrations to prevent the shaft from being forced into the hull or, in the case of a lock-out chamber, forced in or out of the hull.

Welded attachments to the hull structure are considered stress raisers because they inhibit the shell from acting as a continuous membrane. An analysis for local stresses due to external loadings should be conducted on welded attachments especially those which can be subjected to vibration, impact, shock, lifting loads, or some combination of these. Where dynamic loads are encountered, an additional 2 *g* loading factor (minimum) should be superimposed on the hydrostatic loading [1].

Mechanical fasteners should be of prime concern to the designer. Specifying the proper fastener material to minimize the possibility of such phenomena as corrosion, stress corrosion, dezincification, and galvanic action requires diligent attention to design details. The purchase specification preparation should include marking, and traceability for all hull fasteners crucial to the operational safety of the submersible.

3.1.5 Viewports, Hatches, and Closures

The configuration of the viewport must be determined early in the design concept. The viewport insert must be designed to take the low and high pressure differentials and must have a sealing system that will accommodate the modulus of elasticity of the hull material. The structural hull insert used to accommodate the viewport is usually a ring forging machined for welding into the spherical or cylindrical hull. Extensive testing has been done on a variety of viewport shapes. Chapter VI contains detailed guidance for the design of viewports. For the most common viewports used in submersibles, reference should be made to the latest revision of ANSI/ASME PVHO-1, Safety Standard for Pressure Vessels for Human Occupancy.

Hatches on one-atmosphere submersibles are designed to seat with pressure; the reinforcement ring on which the hatch is seated is a ring machined for welding into the hull. Because the hatches are quite heavy, they should be designed using counterbalance or spring-assisted mechanisms to permit operation by one person from the inside. Locking and latching devices must be provided so that the hatch does not "pop" open when being undogged due to a build up of internal pressure higher than the external atmospheric pressure during a dive. Some have small viewports installed so that the operator can determine if the water above the hatch has drained, allowing the hatch to be safely opened.

Submersible pressure-tight closures may also be required external to the pressure hull. Enclosures to house batteries, and electrical or hydraulic components with one-atmosphere pressure internal and ambient pressure external are examples of this. The closure must be designed so that it is easily accessible for servicing the contained equipment, yet the design must use the same criteria as are applied to the pressure hull. External enclosures are fitted with leak detectors to alert the operator that water is present and corrective action is required.

3.1.6 Ballast and Trim Systems

When the desired hull configuration of the submersible is stated, the desired volume and weight of the buoyancy material and the fixed and variable ballast are assigned by the designer. The fixed or static ballast is usually placed as low on the submersible as possible. This will enhance its surface stability. Often the fixed ballast is releasable so that the submersible can achieve positive buoyancy and come to the surface in an emergency. The release of the fixed ballast and other droppable weights must not result in instability while the submersible is submerged, while rising to the surface, or while on the surface. Consideration must be given to the flight path of the submersible when rapidly rising to the surface after dropping emergency ballast. A high upward velocity can impart cross flow, causing severe oscillations of the submersible.

During the entire design process and during fabrication, weight control is an absolute necessity. A margin for construction growth must be provided as well as future growth margin while the submersible is in service. The variable ballast system must be designed to compensate for minor temporary changes in overall weight and for changes in the density of the sea water with change in depth and sea water temperature. The variable ballast system must be designed so that the power expenditure to change ballast is minimal. The variable ballast system may be a hard or soft tank system from which water may be pumped out to sea or flooded in from sea. In a hard system, the tank is designed to be able to withstand submergence pressure. A soft system has the tankage designed for low pressure and the pump, piping, and valves designed for high pressure, with suitable protective devices to avoid overpressurizing the tanks. Keeping the high-pressure piping runs to a minimum is prudent and enhances safety.

Trim system design must also consider the need to adjust weight longitudinally for correction of internal weight shifts, to accommodate the longitudinal position of added equipment, or to compensate for objects lifted by a mechanical arm. The trim system can be one in which liquid is shifted to compensate for the longitudinal weight shift, or a mechanical system can be devised where a ballast weight is shifted to compensate. The requirements for a trim system must be established early in the design concept because it affects many submersible systems and arrangements. It should be noted that many submersibles do not employ a dedicated trim system, but trim adjustments are made by shifting fixed ballast or buoyancy modules as required.

3.1.7 Circularity and Sphericity Measurements

Circularity or sphericity or both measurements of the pressure-resisting structure are required by the various regulatory agencies. Their purpose is to record the deviation of the structure from a theoretically perfect circle or sphere and any local departures that the designer may have specified. Limits have been established which, if exceeded, reduce the ability of the hull to resist the hydrostatic pressure at maximum operating depth. Out-of-roundness or -sphericity limitations are established based on the factor of safety that is specified for the overall pressure hull structure.

Upon completion of a set of circularity or sphericity measurements, the data must be analyzed by the designer to determine and revise the theoretical collapse depth if the specified limits are exceeded. An estimate of the theoretical collapse depth is useful in that it provides the operator with a guideline as to the hull's factor of safety. This will be useful in case of an emergency. It is also useful when conducting proof test of the hull when specified by regulatory agencies. It is important that complete and certified records of circularity or sphericity be retained for submersible safety records and for comparison purposes if modifications to pressure hull structure are made at some future date.

3.1.8 Cyclic Fatigue Analysis

Regulatory agencies require that a fatigue analysis be submitted for review if the lifetime full-pressure range cycles exceed the calculated maximum allowable cyclic life of the pressure hull structure. To arrive at the equivalent number of cycles a submersible may attain, a hypothetical dive profile that might be encountered during the operational life of the submersible should be developed. The dive profile must then be converted into equivalent full-depth cycles, which are compared to the calculated cycles to determine if the calculated maximum allowable cycles are exceeded. If it is exceeded, a fatigue analysis must be performed.

The various penetrations and the attachments to the pressure-resisting structure give rise to discontinuities and stress-raisers which require that the designer select and use an appropriate stress-intensity factor when conducting fatigue analyses. Such discontinuities may decrease the structure's allowable fatigue cycle life and require more frequent surveillance inspections during the operating life of the submersible.

3.1.9 General Hull Items

The superstructure or exostructure and protective rings and bumpers are usually attached to and derive their support from the pressure-resisting structures. Similarly, attachments for handling and lifting the vehicle in or out of the water are usually terminated on the pressure structure. The purpose of these attachments is to resist the forces which are imposed on the exostructure during operation and on protective rings during launch and recovery. Adequate consideration must be given to mitigate the point loading, which can be imposed on the pressure hull structure due to impact loads of heavy sea-state conditions on approach to the launch platform or during lowering or hoisting. Forces of up to 2 *g* may be experienced.

Some buoyancy materials have a tendency to absorb water when subjected to deep submergence pressures, and consideration of loss of buoyancy due to this phenomenon must be taken, or suitable coverings applied to minimize absorption. The alternative is to sacrifice some buoyancy and specify higher-density materials that have low water absorption characteristics at the desired operating depth.

Even though the time in water in the lifetime of the submersible may be short, the amount of dissimilar material employed and its contributing galvanic action when submerged can create severe electrolytic corrosion problems. It is important, therefore, that care be exercised in the selection of materials for systems that are in contact with or in close proximity to the pressure-resisting structure.

During normal operations, a submersible may be working near platforms, submerged objects, uneven and rocky bottoms, and in near vertical escarpments. These environments pose a danger in that entanglement or snagging of the vehicle becomes a real possibility. The superstructure and appurtenances, such as propulsion and thruster motors, rudders, and diving planes, should be faired or shrouded to reduce the risk of entanglement. Obviously, some risk of entanglement will always be present, but fairings will help conduct operations successfully in entanglement-prone situations.

3.1.10 Hydrostatic Tests

Assuming that the pressure hull has been designed to provide a factor of safety of 1.5 (collapse to operating depth), then external hydrostatic proof tests are required by all the regulatory agencies to between 1.10 and 1.25 times the design operating pressure: no permanent deformation should occur. Strain gaging in the vicinity of hull discontinuities or so-called hard spots is usually specified for the first unit. For those submersibles with a pressure structure that will be required to accommodate both internal and external pressure, an internal hydrostatic test must also be conducted. Such hydrostatic tests may indicate that local yielding of structure occurred, in which case previously obtained circularity or sphericity measurements should be rechecked, accompanied by an inspection of all the welds in the vicinity of the critical structural details.

External hydrostatic tests can be conducted using hydrostatic test tanks if size permits; alternatively, the pressure-resisting structure can be lowered to the test depth on a suitable tether. All hydrostatic tests are usually conducted in the presence of the regulatory agency representative. The results of the strain-gage monitoring of these tests should be analyzed and cyclic fatigue calculations verified.

3.2 Machinery

Proven high reliability must be the primary consideration when selecting and specifying mechanical equipment and system components for all critical submersible systems. Failure of any machinery or component that has been identified as vital for the operation of the submersible usually means, minimally, that the operation will be degraded or, as a worst case, that the vehicle operators can be in jeopardy. Repairs to the mechanical equipment inside the pressure structure while the submersible is submerged, even though the equipment may be accessible to the operators, is difficult and should not be planned for during any operations. Lack of room and the additional weight of spares preclude anything but minor adjustments. For missions that require relatively long periods of submerged operation, redundancy in vital components should be considered and additional weight and space allocated for this purpose.

3.2.1 Propulsion Integrity

Integrity of the propulsion system is paramount to mission reliability and safety in considering the safety aspects of a submersible design. Thrusters, if used as a part of this system, should be designed to augment propulsion power in case of an emergency. Reliability of the propulsion system becomes extremely critical when the submersible is required to operate under ice, overhangs, or other similar obstructions to emergency ascent procedures. Malfunctions usually result in aborting the ongoing operation until repairs are made.

Some designers have placed the main propulsion motors inside the pressure hull or in a separate propulsion pod in a dry environment to achieve greater propulsion system integrity. Others have placed the propulsion motors in an external, oil-compensated pod or used a canned stator-type motor. Some have used more than a single external propulsion pod. In this way, loss of one propulsion pod reduces the capability during the operation, but complete loss of propulsion is avoided. The designer must select the arrangement and method of propulsion to minimize the potential for loss of main propulsion with resulting hazardous situations that such a loss would generate.

3.2.2 *Power Sources*

The power source or energy storage system required for propulsion and support of other auxiliary services must have a high degree of reliability. There are many varieties of power-source systems available to the designer. However, the reliability, serviceability, recharging time, efficiency, cost (initial and life cycle), etc. can vary widely, requiring detailed and careful analysis to ensure that the system selected provides the best fit for the mission and at the same time provides an adequate level of safety.

Lead plate/acid storage batteries, the common submersible power source, are very bulky. They give off hydrogen and oxygen when charged, requiring special ventilation precautions. A large-capacity battery significantly increases the weight of the submersible, resulting in the need for additional buoyancy. This increases the size and launch weight of the submersible. Silver-zinc batteries have been shown to be the most attractive for use in submersibles. Silver-zinc cells achieve the highest energy density (that is, watt-hours per pound) of all cells. Initial costs of these batteries are higher than lead-acid batteries, but they have a significant salvage value due to their silver content. Fuel cells similar to those used in the space program have been adapted to a submersible with some success, but they are not safe or cost-effective at the present time. Closed-cycle engines are being used with success. Normally, the use of fuel cells or closed-cycle engines is considered when there is a demand for relatively long transitory periods of submerged operations. Care must be exercised in the selection of these systems, however, since their use can necessitate many safety precautionary measures. For example, as power sources they are more complex, the fuel used either is explosive or constitutes a fire hazard, or both, and in the closed-cycle engine the products of combustion must be carefully handled to avoid their becoming a danger to the operators.

The placement of battery power sources can be either internal in an ambient atmosphere or in a dry dedicated pod, externally located. Internal storage requires the pressure structure to be larger in order to contain the power source. Use of dry storage external to the pressure hull requires that the structure withstand the submergence pressure, and accessibility for maintenance must be carefully considered.

In this dry environment, the battery assembly can be handled, charged, and discharged somewhat like a standard golf-cart lead-acid battery. The potential for grounds is reduced and maintenance is minimal. The greatest danger occurs during the charging cycle when the batteries generate hydrogen and oxygen and must be safely ventilated out of the submersible or pod.

External placement of the battery power source in a nonpressure-resistant container requires that the battery container be oil-filled and compensated to allow for changes in ambient pressure. Oil-compensated batteries have been used in many submersible installations, but in the oil environment maintenance is high and requires a long turn-around time for recharging and battery replacement. The reliability of this type of power source is somewhat less than that of a dry battery but is acceptable when properly designed and maintained.

3.2.3 *High-Pressure Gas Storage*

For emergency and normal use in submersibles, oxygen for life support must be carried in some form. It is convenient to carry it as a compressed gas at high pressure to minimize the space required for this life-support gas. Storage of oxygen inside the pressure boundary must be carefully considered since an oxygen system leak can generate serious consequences. Escaping high-pressure oxygen can increase the submersible's internal pressure and oxygen concentration, which will increase the flammability potential of many materials commonly found in a submersible, with the possibility of a small spark causing a lethal fire or explosion. Additionally, the increased oxygen concentration can exceed safe breathing limits. Internal storage of high-pressure air, if used for ballast blowing, etc., should also be carefully considered because it introduces similar overpressuring and oxygen concentration dangers. As a general rule, therefore, for reasons of submersible safety, it is preferable to install high-pressure gas containers external to pressure hull and to minimize the extent of system piping installed within the pressure hull.

External storage of high-pressure gases is done on many submersibles with a high degree of reliability. The weight penalty of carrying gases externally is quite small compared to that of carrying the containers inside the pressure boundary. Submersibles fitted with diver lock-out capabilities must also have storage for the mixed breathing gases which are required for the divers.

3.2.4 *Droppable Ballast*

One of the rapid ways of gaining buoyancy in an emergency recovery situation is to reduce the weight of the submersible by dropping ballast. There are a number of satisfactory alternatives to achieve a rapid reduction in weight. Many submersibles have a system whereby a single large weight can be released mechanically from inside the pressure vessel. The release of lead or steel shot stored in free-flood containers or tanks provides a more controllable technique for gaining positive buoyancy when needed. Some submersibles have the ability to drop external stores, such as instrumentation packages, mechanical arms, and the external battery, in an emergency. Release of the buoyant pressure sphere or personnel compartment from the submersible frame, which then makes a free ascent to the surface, has been incorporated in submersibles. Notwithstanding the manner of gaining positive buoyancy, the submersible or section of the submersible housing the personnel must come to the surface and stay on the surface in a relatively stable condition so that it can be recovered or towed to safety. The mechanism, whether it is mechanical, electromagnetic, hydraulic, or other, must be such that a single accidental and inadvertent action does not result in the release of the emergency weight. Regulatory agencies usually require two distinct actions to activate the release mechanism. In addition, systems using releasable weights or buoyant personnel compartments must be capable of functioning at large angles of inclination and also when the submersible is on the bottom.

3.2.5 *Testing*

The submersible designer need be constantly aware of vehicle safety while producing a cost-effective design meeting all mission requirements. Assurance of vehicle safety and reliability can be gained in several ways. Selection of components and systems which have demonstrated high in-service reliability in similar applications is obviously an approach that can provide an acceptable, cost-effective design solution, but may not be the most desirable. Often, new materials, components, and systems can be selected which show great promise for reducing weight or space or both, while at the same time offering a significant improvement in performance. Because systems which have not been used in the sea water environment, particularly in the deep ocean, have no history of performance, such systems must undergo extensive testing before any confidence can be developed that the system or component will perform reliably. Prior to incorporating new systems or concepts, the designer must give serious consideration to the extent of testing required to gain the needed assurance of safe and reliable operation.

3.3 Piping Systems and Valves

As a general rule, all piping and valves that must confine and control sea water to the designed operating depth are considered part of a critical system demanding the most careful scrutiny by the submersible designer. The designer should attempt to minimize or eliminate sea water piping inside the pressure hull boundary. Any design effort that reduces the amount of high-pressure sea water piping inside the hull increases the inherent safety of the submersible. The inexhaustible supply of high-pressure sea water surrounding deep-ocean-operating submersibles must be treated by the designer with the greatest possible respect.

3.3.1 *Piping System Integrity*

Piping systems inside the pressure hull should be designed with a minimum of mechanical joints to reduce the potential sources for leaks. In such a confined space, a measurable leak of any fluid or gas can seriously jeopardize both the safety of the occupants and the success of an operation. Protection by means of shields and covers should be considered for electrical and electronic equipment which must be colocated with high-pressure sea water piping. High-pressure pinhole leaks in sea water or hydraulic system piping can produce a fog-like effect which limits visibility and the ability to locate a leak quickly.

Externally, a leak in a piping system will result in loss of the working fluid or, if the external sea pressure is greater, will flood the system, contaminating it and most likely rendering it inoperative. Flooding a system can also result in a decrease in buoyancy.

A piping-flexibility analysis should be considered for critical piping systems—sea water, oxygen, high-pressure air, etc.—and hydraulic piping, to ensure that stress levels generated by internal pressure and hull compression at depth are within allowable limits.

Because piping system design is extremely critical to the safety of the submersible, steps must be taken during the design process to ensure the integrity of piping systems and appropriate specifications must be prepared covering fabrication, installation, and testing. Internal cleanliness of piping systems is also an important consideration. The designer should specify cleaning processes to be followed to remove oil in oxygen systems and foreign material that can be present due to construction in all systems. Systems with sophisticated and close-tolerance valves, such as the hydraulic system, are prone to malfunction when the system is not adequately cleaned.

3.3.2 *Sea Water Piping and Valves*

Sea water systems on a submersible are usually confined to the variable ballast, trim, and depth indication systems and to where cooling is required by sea water heat exchangers. The volume of sea water handled should be kept to a minimum, resulting in piping sizes that are relatively small. All sea water piping systems, specifically these which must function at design depth pressure, must be selected with care to ensure service reliability. Attention should be focused on

- factor of safety at maximum operating depth,
- corrosion in sea water and under stress,
- erosion where high-velocity flow may be required, and
- hydraulic water hammer where quick closing valves are used.

Sea water piping systems that enter the pressure hull should have quick-acting valves that are readily accessible to the operators. The valves should be selected for ease of operation and reliability and should be clearly labeled to show open and closed positions. If remotely operated valves are selected, provisions for local operation of the valves must also be made.

3.3.3 Air Piping and Valves

An air system is used for blowing ballast water from tanks generally to gain freeboard when the submersible is surfaced. The high pressures required and the shock of changing velocities of air flow as valves are opened and closed must be considered in the design. The stop valves should be designed for slow opening to preclude a diesel effect when high-pressure air is directed into a dead-ended system. The air used in the system must have a low dewpoint to prevent ice formation at the control valves due to adiabatic expansion, which could cause a malfunction of the system. Air filters, if installed, are particularly susceptible to blockage by ice formation.

Piping located external to the pressure hull must be designed not only to withstand the internal pressure but also must resist collapse due to external pressure. Valves in external systems operating at a low internal pressure may require a special valve stem packing to prevent sea water at a higher pressure from leaking into the system. Most valve stem packing is designed to withstand internal pressure.

3.3.4 Hydraulic Piping and Valves

The hydraulic system in most cases powers control surfaces, valves, and other components critical to safe operation of the submersible. In addition to ensuring a reliable design and selection of components, the designer should take great care to avoid the possibility of salt water contamination in those portions of the system that are external to the pressure hull. It is prudent to select proven components wherever possible; if new concepts must be introduced, then they should be adequately tested under pressure and at temperature before they are incorporated into a design.

Any hydraulic piping and valves located inside the pressure boundary should have fabrication and installation requirements similar to those applicable to specified sea water or air piping. A pinhole leak in a hydraulic system can atomize hydraulic fluid as it escapes into the atmosphere. The resulting mist may be toxic to personnel; it can also generate an explosive hazard.

3.3.5 Life-Support System

Specifications should be prepared clearly detailing oxygen-system cleaning procedures. The oxygen, piping, valving, and the storage must be specified and certified as being oxygen clean prior to filling the system and placing it in operation. The valves specified should be slow-acting to avoid the danger of rapid compression resulting from the rapid release of high-pressure oxygen into a dead-ended system. Regulators must be specified as fail-safe to close with a manual bypass for use in an emergency. The introduction of the metabolic oxygen into the manned spaces of a submersible should be designed with outlet located in or close by the discharge of the carbon dioxide scrubber or other air-circulating ventilation component so that the oxygen is rapidly disbursed throughout the internal volume of the submersible. Oxygen has a tendency to collect in pockets, which creates a hazard by increasing the flammability of the material in an oxygen-rich locality or may result in an explosive mixture.

3.3.6 Material Selection

The material selection for the piping and valves used in sea water or exposed to sea water due to an external location must take cognizance of the effect of the salt-water environment on the material. The piping should be seamless and corrosion resistant; it should be designed to a recognized code or standard. The valves similarly should be corrosion resistant and be designed to a recognized code or standard.

Piping connectors for critical systems (sea water, high-pressure air, oxygen, hydraulic, or other high-pressure systems) should be kept to a minimum and the type of joint—welded, silbraze, O-ring, etc.—should be justified by including information such as service experience, recognized standards, specifications, etc. The connectors and piping systems must be of compatible materials as recommended by the connector manufacturer. (See Sections 5.4.2 and 5.6.1 for mechanical-type fittings in use.) External piping and valves may have a need for cathodic protection if the various piping and valve components are made of dissimilar materials.

3.3.7 Quality Assurance

A comprehensive quality assurance and control system must be in place at the start of the design program. It should be initiated to support a review of the purchase specification to ascertain that the materials are satisfactory for their intended use or service. A program that provides for the traceability of materials and components from receipt to final installation and testing should be established. Records maintained in support of the quality-assurance program must be complete, accurate, and available for survey by the regulatory agencies throughout the entire construction and testing process. Records should include all the results of the NDT conducted to establish material pedigree, weld, and silbraze pipe joint integrity, pressure hull weld integrity, hull circularity, etc.

3.3.8 Hydrostatic Testing

The designer should specify hydrostatic test requirements for all piping systems to demonstrate their strength and integrity prior to system operating tests. Piping systems are normally subjected to an internal hydrostatic pressure of 1.5 times the maximum allowable system working pressure or at test pressures as required by regulatory agencies. Precautions must be taken so that clean systems are not contaminated during the hydrostatic testing. Piping systems outside the pressure hull should be hydrostatically tested for inside and outside pressure influence.

3.4 Environmental Systems

In the closed environment of a submersible, the quality of the breathing air must be maintained within certain prescribed limits. Conditions in a one-atmosphere ambient are usually maintained between 0.18 and 0.21 atm partial pressure for oxygen, and the carbon dioxide partial pressure should be held below 0.015 atm. For optimum personnel comfort, the ideal humidity is between 50 percent and 70 percent, and the temperature should be near 21.1°C (70°F). Within a submersible, however, the ideal conditions of relative humidity and temperature are rarely achieved. Humidity and temperature control equipment are power-consuming components and are often not specified for installation.

In a submersible, carbon dioxide is considered a contaminant and can be removed using some form of carbon dioxide scrubbing equipment. Baralyme, Soda Sorb, or lithium hydroxide can be used as an absorbent for carbon dioxide scrubbing. Each has somewhat different characteristics and differs in efficiency depending upon the temperature, the humidity level, and the quantity of carbon dioxide in the atmosphere. The designer has to determine the best-suited absorbent under the conditions of use and for the anticipated duration that the occupants will have to remain within the closed submersible.

Many new materials, such as paints, that may be installed in a submersible off-gas during normal operations, and off-gassing is accelerated when the material is exposed to the high ambient temperature that can occur in the confined space of these vehicles. Some off-gas products can be detrimental to the occupants in a closed atmosphere after several hours. Aging new materials is one way to reduce off-gassing. To minimize the impact of off-gassing, operators should run equipment at elevated temperatures prior to initial sea trials, thereby reducing the quantity of off-gassing during normal operating modes.

3.4.1 Environmental Control Systems

The most important is the oxygen monitoring system. Alarm set points at the low and high allowable range of oxygen will alert the operator of an abnormal deviation so action may be taken. The carbon dioxide is measured by portable detectors such as Draeger detection tubes. Although carbon dioxide generation cannot be controlled because it is a product of respiration, carbon dioxide must be absorbed using a carbon dioxide scrubber when measurements indicate it has reached maximum allowable limits.

Trace gases such as methane and carbon monoxide are usually not monitored since the evolution of these gases is extremely slow and would not rise to harmful partial pressures during a dive duration. Humidity can become quite high, approaching 100 percent and causing hull condensation to occur. Electrical equipment must be protected from this condensation.

In a lock-out compartment, the breathing mixture is a helium-oxygen mix which is provided the occupants in accordance with the diving gas schedule. In a helium-oxygen mixture, temperature control of an atmosphere is recommended and provisions for reducing the loss of diver body heat is required. Carbon dioxide scrubbers also must be provided in the lock-out chamber.

3.4.2 Emergency Breathing System

It is essential that submersibles be fitted with an emergency breathing system in the event the submersible atmosphere becomes contaminated. A closed-loop system for use in these conditions is required by regulatory agencies. This equipment provides emergency breathing for up to 24 h and in addition will conserve body heat if the occupants wear exposure suits, which should be provided as part of the emergency equipment.

3.4.3 Fire Fighting

The most probable origin of a fire in a submersible would be electrical in nature. Insulation, when overheated, could smoulder and generate smoke, contaminating the atmosphere and requiring the use of the emergency breathing system. Because the use of any type of fire suppressor may further contaminate the atmosphere, the selection of a means of suppressing a fire requires close examination of alternatives. Prevention of electrical fires by use of circuit-overload protection devices and shielding of electrical components from salt-water spray due to a piping failure must be given careful consideration.

Prudent action in case of a fire in a submersible dictates the immediate use of emergency breathing equipment and a return to the surface to allow ventilation of the interior and to make possible fire suppression action by support ship personnel. The designer should review the latest available fire-fighting and suppression equipment to determine the possibility of their safe use in the closed atmosphere of a submersible.

3.5 Electrical and Electronic Systems

Control and communication systems are critical to the safe and efficient operation of submersibles. The signals and power must pass through the hull and must be distributed outside the pressure hull boundary in the very harsh environment of sea water. High humidity inside the submersible can also present a difficult environment for these systems. Highly reliable components and wiring must be selected if these systems are to operate effectively in the face of these conditions. Past failure reports from submersibles reveal that electrical and electronic failures have been responsible for most of the delays in planned operations and that such failures can hinder the safe operation of the vehicle.

3.5.1 Electrical Systems Integrity

In Section 3.4, internal and external power sources were briefly discussed. The integrity and reliability of the electrical power sources must be high. Failure of a submersible's power source constitutes a serious casualty that will in almost all cases terminate a dive. Quick-acting electrical-protection devices designed to operate in the submersible environment are essential to minimize damage and to switch to alternate standby power sources. External electrical cables should be selected for their ability to prevent sea water migration through an open or cut end and where connections to equipment junction boxes are required. Internally, the wiring should be selected having high-temperature-resistant insulation. Control switches and controls should be positioned so as to provide the crew access for rapid operation in case of emergency. Consideration should be given to the incorporation of a kill switch to cut off all power in case of an electrical malfunction which may threaten the safety of the occupants. A ground-detection system with an alarm should be installed as a means to monitor any deterioration of the electrical system integrity, especially the main power supply.

Dry power-source systems are less susceptible to sea water grounds, but the power system overall is heavier. The power penetrators are large in order to accommodate the current required by the propulsion and thruster systems. Dry systems are more easily serviced and can be completely removed when discharged and replaced by a fully charged system. This can shorten significantly the turn-around time because the charging time of the battery is not a controlling factor.

Oil-compensated external power sources have been used on the majority of submersibles. Various devices to reduce the electrolyte contamination during charging are available and provide reliable service. Compensating oil can become contaminated with electrolyte carry-over or sea water; when this situation occurs, grounds develop quickly. Reducing the ground to an acceptable level to conduct further operations may require a long period of time. Compensated systems should be fitted with a sea water indicator capable of detecting very small quantities of water in order to permit early corrective action.

Emergency power supplies, usually dry cells or rechargeable batteries, are provided for emergency communication equipment, carbon dioxide scrubber blower motor, and emergency lighting. Regulatory agencies require a minimum duration of the emergency power supplies to assist rescue efforts.

3.5.2 Electrical/Electronic Penetrators

A variety of components are available for bringing power and signals through the hull. Each manufacturer has a particular technique to prevent sea water from coming in contact with the conductors as they go through a hull penetrator. Some are more reliable than others insofar as electrical or electronic performance is concerned, but there is a universal requirement that if a cable is cut or severed the sea water will not enter through the penetrator to the interior of the hull. The selection of the penetrator type should be based on the operational characteristics of the submersible and the degree of inherent safety to be provided to the operators and submersible. The penetrators must be of a type approved by regulatory agencies and must be tested in accordance with the agencies' requirements. Unless there are very special requirements which cannot be satisfied by one of the available types of penetrators, the prudent course of action, from the point of view of both safety and reliability, is to select a penetrator type that has been proven in at-sea service.

3.5.3 Sensors

The field of vision from a submersible is very limited when the vehicle is operating submerged. For nearly all missions, it is essential that this limitation be overcome by the use of various devices, such as sonar, lighting, and underwater television, to provide the operators with information on the immediate surroundings. Other sensors are required for safety, such as depth indicators, obstacle avoidance sonars, compasses, and speed and distance logs. These sensors enable the operator to fix his position in hydrospace and anticipate dangerous situations. Regulatory agencies require that surface and underwater locating devices be fitted to allow surface support components to continuously track the location of the submersible. Reliability of these sensors is obviously essential for safety of operations and plays a key role in completing mission tasks in a timely and efficient manner. Again, equipment with a demonstrated history of reliable at-sea performance should be considered.

Hydrostatic testing to design operating depth in test tanks for the external components prior to initial sea trials is highly recommended.

3.6 Outfitting for Emergencies at Sea

All sea-going craft are required to be suitably outfitted to assist rescue efforts during emergencies. Regulatory agencies specify many elements of the emergency outfit, listing the systems and equipment that must be installed and operational on a submersible, including underwater telephones, emergency sonars for vehicle localization, exterior paint schemes to assist in positioning rescue gear, and individual survival outfits. Where feasible, individual survival outfits should be provided which preserve occupant body heat and absorb carbon dioxide in a closed-loop breathing mask. Such personnel survival outfits are usually designed to be effective for at least 24 h.

3.6.1 Emergency Communication Systems

Emergency communication while the submersible is submerged is a mandatory requirement of the regulatory agencies. The communication equipment selected to satisfy these requirements must transmit and receive at specified frequencies and effective range and must be supported by a power source that can provide power for a specified time period. On the surface, the submersible must be fitted with an emergency radio capable of transmitting at a designated distress frequency.

3.6.2 Locators

Both the launching platform or mother ship and the submersible should be equipped with a sonar system that permits tracking the submersible's position while submerged. The submersible should have an installed transponder which will enable the locating device to receive the signal and track the submersible while submerged. A pinger transponder with its own battery supply, designed to have an operating life of several days, should be fitted on the submersible, enabling the mother ship's sonar locating system to receive and track the pinger's signal. In addition, high-intensity, strobe-type flashers can be used for locating the submersible underwater. A flashing light is useful also for locating and tracking a surfaced submersible, especially in poor visibility.

3.6.3 High-Visibility Painting Schemes

In high sea states when the submersible is on the surface, a paint scheme using international orange on the superstructure is highly desirable and recommended by regulatory agencies. Paint schemes are also very useful in emergency recovery situations for identification of special equipment and fittings that are a part of the submersible's emergency outfit and to establish the overall orientation of the submersible.

3.7 Testing and Documentation

From the beginning of the design of a submersible, the sponsor, designer, fabricator, and test organization must be cognizant of the fact that the submersible and the documentation associated with its design, construction, and testing will be subject to inspection and review to assess the ability of the vehicle to operate safely. The assessment may be made by the user or a third party, usually the regulatory agency. This assessment is an in-depth review of the traceability of the material, the records of the NDT, the qualifications of the welders and welding procedures, the qualifications of NDT personnel, and many other materials and workmanship. Guidelines have been prepared by regulatory agencies, the U.S. Navy, and U.S. Coast Guard for use by the designers, fabricators, users, and others specifying the areas in which plan approvals, documentation, operator procedures, testing, and manuals are required. Section 4 contains the titles of the various regulatory agency publications which can be used during all the phases of design, construction, and testing of any newly built submersible.

3.7.1 Regulatory Requirements

The sponsor of the submersible, the designer, the fabricator, and other management personnel must be familiar with the certification requirements of the applicable regulatory agencies. The collection and review of all the documentation for the construction material, its fabrication, components, testing, assembly, and NDT must be established at the start of the construction. The documentation is extensive and is subject to an in-depth survey prior to initial sea trials.

3.7.2 Design Considerations

Starting with a requirement for a submersible to perform a particular set of mission tasks, the design evolves from concepts to feasibility studies to preliminary designs to fabrication drawings. During the numerous iterations involved in this design process, the design team must be concerned that the end product will be capable of performing safely and reliably all mission tasks. Compromises must always consider the safety aspects involved during all phases of the submersible's operating life—from launching from a mother ship to operation at depth to retrieval. It must be clearly understood that new and novel designs of system or subsystems will require more extensive design review and analysis and testing of hardware. The use of new or uncommon materials will require proof testing that it can be formed, joined, has predictable and consistent physical properties, and will have an acceptable operating lifetime.

3.7.3 Materials

Selection of materials for the hull structure boundary, including castings, fasteners, electrical and piping penetrators, the piping systems, buoyancy systems, viewports, and other vital components of the submersible, must be done with great care. The ability to join, form, and install material and components must be known and understood at the outset of the design process, or in-depth testing must be undertaken to establish their suitability. The various pieces and parts that comprise the vital or critical systems of the submersible must have their source and pedigree documented and NDT tests and inspections documented; these records must be traceable and verifiable at any point in the construction process and during the life of the submersible. In this way, the entire assemblage can then be certified by users or regulatory agencies. Emphasis must be placed on extensive testing of new and novel materials which have little service experience for compatibility in sea water. Reference should be made to Section 5.2 for additional information on selection of materials.

3.7.4 Fabrication

Fabrication processes and methods to be employed involving vital systems and components should be carefully specified where there is the possibility of modifying the characteristics of materials, reducing the thickness of materials below allowable limits, introducing significant residual stresses, etc. Tolerance limits for deviations from nominal specified geometry must be established in the early stages of the final design.

3.7.5 Testing

Comprehensive testing of materials, piping, and electrical components and systems, and all other elements that will be a part of the submersible's structure or systems is imperative. Documentation in a format that ensures traceability of such tests and inspections is essential. Testing is verification that the material is as specified by the designer, that the fabrication is within the specified tolerance, and that the system as assembled will perform in accordance with the specification. Testing must demonstrate not only its functional ability but also that it will perform safely and efficiently in the sea water medium at depth and at the anticipated temperatures. Documented testing establishes a confidence level which is very important to the operator, occupants, and users of the submersible.

3.7.6 Pre-Sea Trials Testing

Testing should be done both at the component level, such as testing hull penetrators, and at the system level, such as testing a completed steering-control system. Dry tanks for batteries can be hydrostatically tested prior to initial sea trials. A submersible of such a size that it can be accommodated in a hydrostatic test tank offers the opportunity to conduct the equivalent of at-sea deep dives under controlled conditions with a low level of risk, yet achieve a high level of confidence that the submersible will be safe at test depth. The test depth, when using such a tank, should be between 1.10 and 1.25 times the design operating depth (assuming the pressure hull was designed to a factor of safety of 1.5 collapse to operating depth), and the use of strain gages at strategic locations on the submersible's pressure hull allows documentation that stress levels remained within allowable limits.

When all the components, subsystems, and systems amenable to hydrostatic testing and functional testing are installed on the submersible integrated system, testing can be conducted and results documented. These results are reviewed by designated personnel and surveyed by representatives of the user or regulatory agencies. Whenever possible it is prudent to request that the integrated testing be done by the operators, since this provides some of the training required before the submersible conducts its initial submergence trials. Obviously, any discrepancy in any installed system must be corrected if it would impair in any way the safety of the submersible's operation.

3.7.7 Initial Sea Trials and Deep Dive

Safety considerations for manned initial sea trials suggest a conservative approach with a sea trial agenda that specifies the submersible undergo trials necessary to prove operational reliability of the various systems prior to making an excursion beyond a shallow diving depth. Preparation for conducting an initial dive to a specified shallow depth should include a review of documentation of the material used, qualification of the welders and welding procedures, results of out-of-roundness checks, NDT records of welds, brazing of pipe, etc., results of system testing, results of hydrostatic testing, insulation testing, and many other tests and outfitting checks to verify proper fabrication of the submersible. This review may be conducted by the construction activity in the presence of regulating agency representatives or by the regulatory agency itself. When the review is satisfactorily completed, the submersible and operators are permitted to make the initial excursions to shallow depth under strict surveillance by the accompanying mother ship or launch platform. Usually a representative of the regulatory agency will be an observer of the trial.

Following satisfactory completion of a shallow-depth dive, a deep dive to the design depth can be undertaken. Normally, deep-dive agenda require that the depth be increased in increments and that at each specified depth the submersible pause to allow all the systems to be operated to check the functional response for each system at the new depth. If the submersible has not been tested in a hydrostatic test tank to 1.25 design operating depth, it is possible that a submersible, unmanned with appropriate strain gages, can be lowered on a tether to between 1.10 and 1.25 times the design operating depth. This test can generate the need for some special handling and remote test readout facilities involving considerable time and added expense. This test must be witnessed by the applicable regulatory agency. After hydrostatic testing or at-sea deep dive testing, all welds in the pressure-resistant structure should be examined for cracks and results recorded. Additional checks may be required by regulatory agencies.

3.7.8 Certification/Classification

This milestone is reached when analysis of the results of at-sea trials, including the deep dive, is complete and found to be satisfactory. The documentation of satisfactory completion of sea trials combined with the pre-sea trial survey of construction documentation should lead to the assurance of a certificate by the regulatory agency. The certificate will show the depth for which the submersible is certified and the conditions both for keeping the certificate valid and under which it will remain valid.

3.7.9 Periodic Surveys

Regulatory agencies have specified the periodicity for conducting recertification of submersibles. The rules of the various agencies specify the systems, subsystems, and components to be surveyed and the maintenance records requiring review. A regulatory agency also may require that a special survey be conducted which can be more encompassing and which must be conducted in the presence of the surveyor to keep the submersible's certification current.

Records which have been prepared during the fabrication, construction, and testing of the submersible should be retained for a length of time commensurate with the operating life of the submersible. Records of the material tests, NDT results, and system test results are important should failures or casualties occur on the submersible. With these records, causes of malfunctions may be identified.

4. Background of U.S. Navy and Classification Societies' Involvement in Submersible Safety

The U.S. Navy, with its fleet of combatant submarines, has a strategic interest in extending its knowledge of the world's oceans. Until the mid and late fifties, most of the Navy's oceanographic information was gathered by surface ships probing the ocean from above. In the early fifties, with technical assistance from the world's most renowned balloonist, Auguste Piccard, and his son, Jacques, the Navy obtained its first deep-diving manned submersible. Christened the *Bathyscaphe Trieste I*, on January 8, 1960, with Dr. Jacques Piccard and Lt. Don Walsh of the U.S. Navy on board, the submersible dove to the deepest part of the ocean [10 912 m (35 800 ft) in the Marianas Trench off Guam]. Adventurists/scientists like these influenced the Navy's scientific community to direct more attention to the use of the submersible for probing the oceans.

Industry was quick to recognize the market potential for the noncombatant submersible. The oceanographer, geologist, and marine biologist were being offered the chance to study and observe the ocean environment at first hand. In a relatively short period, a number of submersibles of various configurations were made available to the Navy, for sale or lease. The sixties was being heralded as the decade for exploring and developing "inner space," our new Golconda. It was in this early sixties period when the Navy suffered a disastrous casualty to one of its nuclear submarines.

On April 10, 1963, while engaged in a deep test dive approximately 370 km (230 miles) off the northeast coast of the United States, the nuclear submarine SSN 593 (USS *Thresher*) was lost at sea with all persons aboard. The Navy immediately put a restriction on all of its submarines, a court of inquiry was convened and the following actions resulted: The Hon. Fred Korth, then Secretary of the Navy, directed the Chief of Naval Operations to prepare a plan for establishment of a submarine safety organization. In addition, Admiral E. C. Stephan, then head of the Deep Submergence Systems Review Group, initiated four actions: he assembled a cadre of experts on location, recovery, search, and operation of deeply submerged objects; he initiated and had completed a documentary review of all existing literature on this highly involved subject; he had his group solicit ideas from more than 200 industrial concerns, the scientific community, and other government agencies; and he instituted a study of the ocean environment itself as it pertains to the practicality and feasibility of rescue or recovery or both of deeply submerged objects.

The foregoing is a thumbnail sketch of the Navy's involvement with the use of the submersible and its concern with their safe application up to the early sixties. It was sparked by successful adventures with the submersible *Trieste* and then driven by fate into a frenzied pace to develop rules of safety and to profit from lessons learned as a result of the *Thresher* disaster, while reducing the possibility of any reoccurrences of such misfortunes that might slow our future advancements in the use of the submersible.

By December of 1963, the Navy had firmed its program for reestablishing maximum reasonable assurance that new and existing combatant submarines could again operate unrestricted.

Increased emphasis on the use of the submersible for oceanographic research in the early 1960s had significantly expanded the number of Navy leased or owned noncombatant submersibles. With the *Thresher* disaster still fresh in mind, there was a growing concern in the Navy over the safety of people using these submersibles.

4.1 Formal Noncombatant Submersible Safety Program

To alleviate the Navy's concern for the safety of its people, the Secretary of the Navy in mid June 1966 instructed the Chief of Naval Operations to establish policy for assuring the safety of Navy people engaged in the use of the noncombatant submersible. The major points to be considered were

- the competence of the submersible pilots;
- the operational control of the submersible; and
- the material and procedural adequacy of the submersible, that is, safety certification, typically
  - —the design and materials chosen must be safe,
  - —the craft must be properly built and tested to the design, and
  - —the craft must be operated and maintained safely.

Thus began the first action to assemble procedures and criteria into a guidance document that would provide acceptable levels of personnel safety throughout the operating ranges of a submersible.

Note that, with the exception of specific regulations imposed by the Canadian Ministry on the submersible *Auguste Piccard* and the U.K. Department of Trade, Merchant Shipping Regulations of 1981 (Submersible Craft Construction and Survey), there are few laws governing the design and construction of the noncombatant submersible. Despite the fact that several bills have been introduced to the U.S. Congress to grant regulation of submersibles to the U.S. Coast Guard, none of these has been passed.

The first formal document covering submersible safety procedures was issued by the U.S. Navy in 1967, *Material Certification Procedures and Criteria Manual for Manned Noncombatant Submarines* [2]. Since this document applied only to submersibles used by the U.S. Government, the Navy worked with the American Bureau of Shipping to see that submersible safety criteria and guidelines were available to the civilian sector of the industry. In parallel with this work, the Marine Technology Society's Undersea Vehicle Committee was generating a set of general design guidelines relating to the safe operation of submersibles.

In 1968, the following two documents were published:

Marine Technology Society, "Safety and Operational Guidelines for Undersea Vehicles."

American Bureau of Shipping, "Guide for the Classification of Manned Submersibles."

During the 1970s, the ship classification societies of many European countries issued documents covering their particular requirements for classification of submersibles.

1971: Germanischer Lloyd, "Regulations for the Classification and Construction of Submersibles."

1973: Lloyd's Register of Shipping, "Rules, Regulations and Guidance Notes for the Construction, Classification and Planned Inspection of Submersibles."

1974: Det Norske Veritas, "Tentative Rules for Construction and Classification of Submersibles."

1975: Det Norske Veritas, "Rules for the Construction and Classification of Diving Systems."

1976: Bureau Veritas, "Rules and Regulations for the Construction and Classification of Submersibles."

1973: U.S. Navy, "System Certification Procedures and Criteria Manual for Deep Submergence Systems."

1974: Marine Technology Society, "Safety and Operational Guidelines for Undersea Vehicles—Book II."

1976: U.S. Navy, "System Certification Procedures and Criteria Manual for Deep Submergence Systems."

Continued interest in submersible safety led to the publication in 1979 of three documents:

American Bureau of Shipping, "Rules for Building and Classing Underwater System and Vehicles."

Marine Technology Society, "International Safety of Standard Guidelines for the Operation of Undersea Vehicles."

Lloyd's Register of Shipping, "Rules and Regulations for the Construction, Classification and Planned Inspection of Submersibles and Diving Systems."

The early documents released for use in the submersible safety field were directed toward the submersible itself and were usually in the form of guidelines or guidance notes or were tentative or preliminary in nature. In later editions, these documents took the form of specific rules and regulations for the construction and inspection of submersibles to be "classed" by the various ship-classification societies. In addition, as these later revised rules were published, many of them were expanded to include other forms of undersea equipment, such as tethered submersibles, diving bells, transfer chambers, and habitat/work chambers. While not all subsystems are similar between diving systems and submersibles, there is little difference in the safety requirements or systems of a lock-out submersible and roving diving bells.

5. U.S. Navy and Classification Society Requirements for Submersible Safety

The following is intended to provide a detailed summary of safety requirements that have been established through the joint efforts of the U.S. Navy and various classification societies which, if followed, can lead to certification of a submersible for at-sea operations to its specified mission capabilities.

5.1 Design Parameters for Submersible Safety

A number of design parameters must be considered initially by the naval architect and builder in order to provide maximum reasonable assurance for the safety of the submersible occupants. In all cases, material selection, design, and fabrication techniques should be justified in accordance with recognized and accepted practices. Proper consideration should be given to complex configuration and intersection, cyclic fatigue, and low-temperature requirements.

Prudent application of various available codes, instructions, and standards to the submersible will provide a valuable base-line for the designer and builder. Such design practices include American Society of Mechanical Engineers (ASME) Code Section VIII, Division 1 and 2, ASME Code Section III, ANSI/ASME PVHO-1, and NASA Instruction KMI 8610.6, Attachment A: "Minimum Criteria for Operation Involving Personnel in a Vacuum/Oxygen Environment," in addition to a myriad of Military Standards and Specifications.

In addition to those documents mentioned above, the following design criteria have been specifically developed to satisfy submerged safety design parameters.[1]

5.2 Matching of Materials and Components

Experience has shown that submersible designers must be constantly aware of the need to match materials and components carefully to the expected service environments. Such things as relative location, compatibility in salt water, galvanic potential of dissimilar metals, service life, previous history of service, etc., are extremely important for a good design. Above all, designers must always be prepared to justify the materials and components that are specified for the submersible. It should be recognized that designers must be given latitude in their design of submersibles and that new materials/components and new applications of time-tested materials/components, along with new configurations, may be employed. Conversely, it is also recognized that the less experience available with materials, components, or application, the greater the burden upon the designer to justify the adequacy of the design. To aid the designer in this regard, it is suggested that materials and components be placed into three groupings:

Group 1: Materials and components with which there is both considerable fabrication experience and considerable operating experience in the intended environment and application.

Group 2: Materials and components which have *not* been used extensively in similar applications but are classed as conventional due to identification by Military Specifications, Federal Specifications, or recognized American Commercial Standards such as those published by ASME or ASTM. Materials or components available as standard stock items built to a recognized commercial or Federal standard will be considered in this category. Examples of materials and components that are currently considered to be in this category are certain types of aluminum, titanium, and some high- and low-strength steels. The determination of acceptable properties and allowable operating stress values will be based on the recommendations and supporting information provided by the designer.

Group 3: Materials and components for which definitive information and experience are not available. The basis for testing and the criteria for acceptance of new materials and components will not automatically be the same as for those currently in

[1]The information presented hereinafter was extracted for the most part from material that has been assembled into the unclassified *Naval Ship Systems Command Procedures and Criteria Manual for Non-Combatant Submersibles*, NAVSHIPS 0900-028-2010 of July 1967, which was updated in 1973 and later replaced by the *Procedures and Criteria Manual for Deep Submergence Systems*, NAVMAT P-9290 of June 1976 [2].

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