SUBMERSIBLE VEHICLE SYSTEMS DESIGN

Written by
a Group of Authorities

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Dedication

Submersible Vehicle Systems Design is dedicated to the memory of Rear Admiral Andrew I. McKee, USN (Ret.), a member of this book's Control Committee who passed away before its publication. Admiral McKee enjoyed a distinguished career in the United States Navy and, upon retirement, as Vice President for Research and Design at the Electric Boat Division of the General Dynamics Corporation. Professionally, his life exemplified the highest traditions of naval and civilian service in the field of naval engineering with particular reference to submarine design and construction. Personally, he was the epitome of a gentleman—a distinct pleasure to know in both a professional and social context.

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Foreword

The use of manned submersibles reached a high plateau from the mid to late '70s when over 100 of these underwater vehicles were in an operational status. Early in this period, the MS-2 (Submersibles) Panel/Marine Systems Committee of the Society of Naval Architects and Marine Engineers resolved to record the major aspects of technology associated with their design by sponsoring a book to be entitled *Submersible Vehicle Systems Design*. The limited amount of material in *Principles of Naval Architecture* pertinent to this subject and the subsequent decision to delete the "Submersible Vehicles" chapter from the 1980 edition of *Ship Design and Construction* reinforced the rationale for undertaking this project. The rationale was further supported when, in June of 1986, with a review copy of the draft text available, the Society's Technical and Research Committee recommended that the book be published by the Society—stating that it would be a contribution to its record of engineering technology by filling a major gap in this record. In February of 1987, the Society's Executive Committee unanimously endorsed publishing *Submersible Vehicle Systems Design* and final publication procedures began.

At the initiation of its project, in 1975, the Panel established a Control Committee consisting of individuals with extensive experience in submarine/submersible design and construction and composed as follows:

Mr. Victor T. Boatwright—Electric Boat Division
of General Dynamics Corporation (Ret.)
Captain Robert L. Evans, USN (Ret.)
Captain Donald H. Kern, USN (Ret.)
Rear Admiral Andrew I. McKee, USN (Ret.)

Professor E. Eugene Allmendinger of the University of New Hampshire was invited to serve as the book's Editor and, in this capacity, also as the Control Committee's chairman.

Both Panel and Committee members were involved in the formulation and subsequent revising of the book's outline and in selecting authors for chapters or chapter sections—the knowledge and expertise of these individuals being reflected in their contributions to this publication. The Committee maintained liaison with the authors and reviewed their work throughout the lengthy process of bringing this book into being.

John A. Pritzlaff
Chairman SNAME, Marine Systems
Committee, Panel MS-2 Submersibles
(1975–1985)
Chairman SNAME, Offshore Committee
Panel, OC-6 Submersibles and
Diving Systems
(1986——)

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

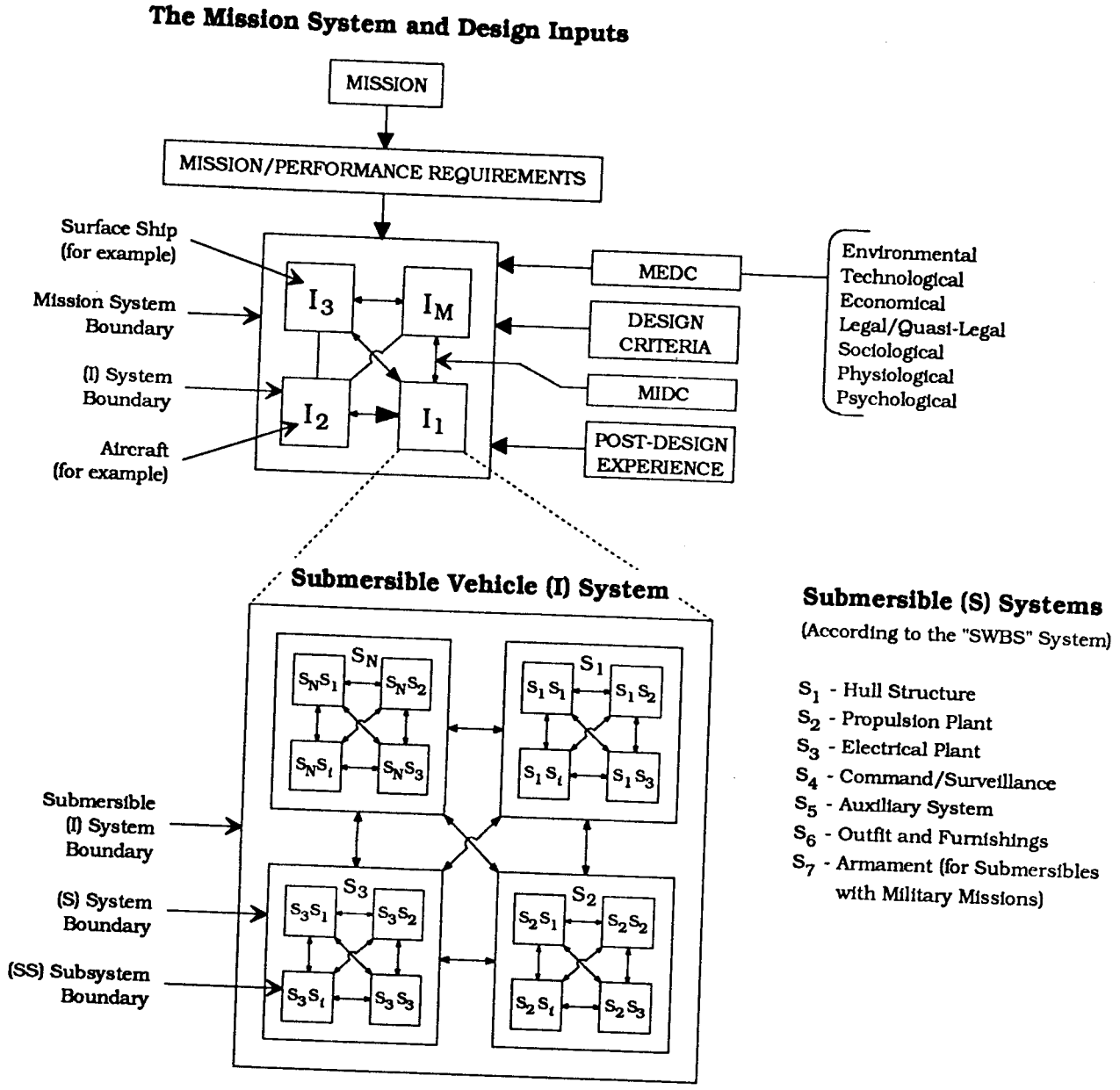


Fig. 1 The mission system and its subsystems

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

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

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

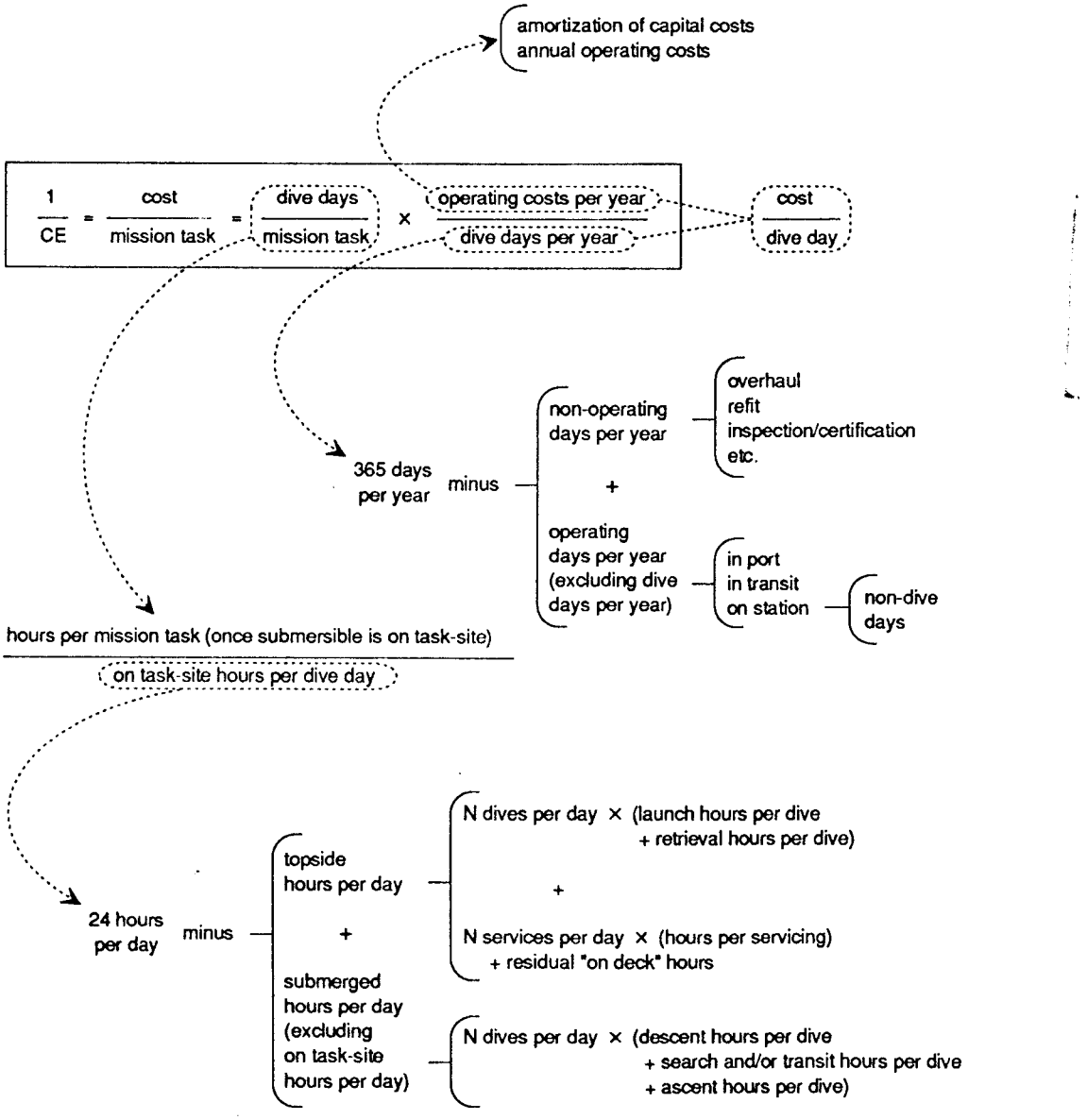


Fig. 2 Mission system cost-effectiveness

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

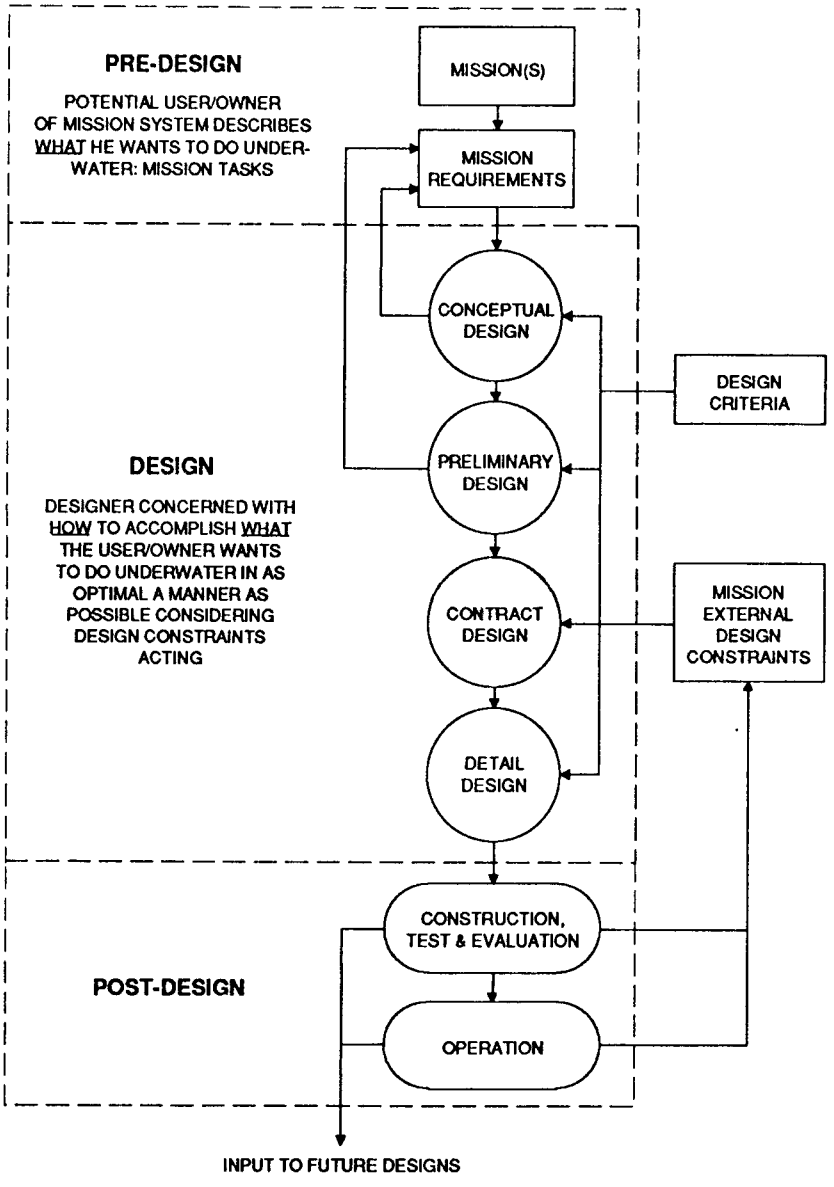


Fig. 3 Design and associated processes

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,

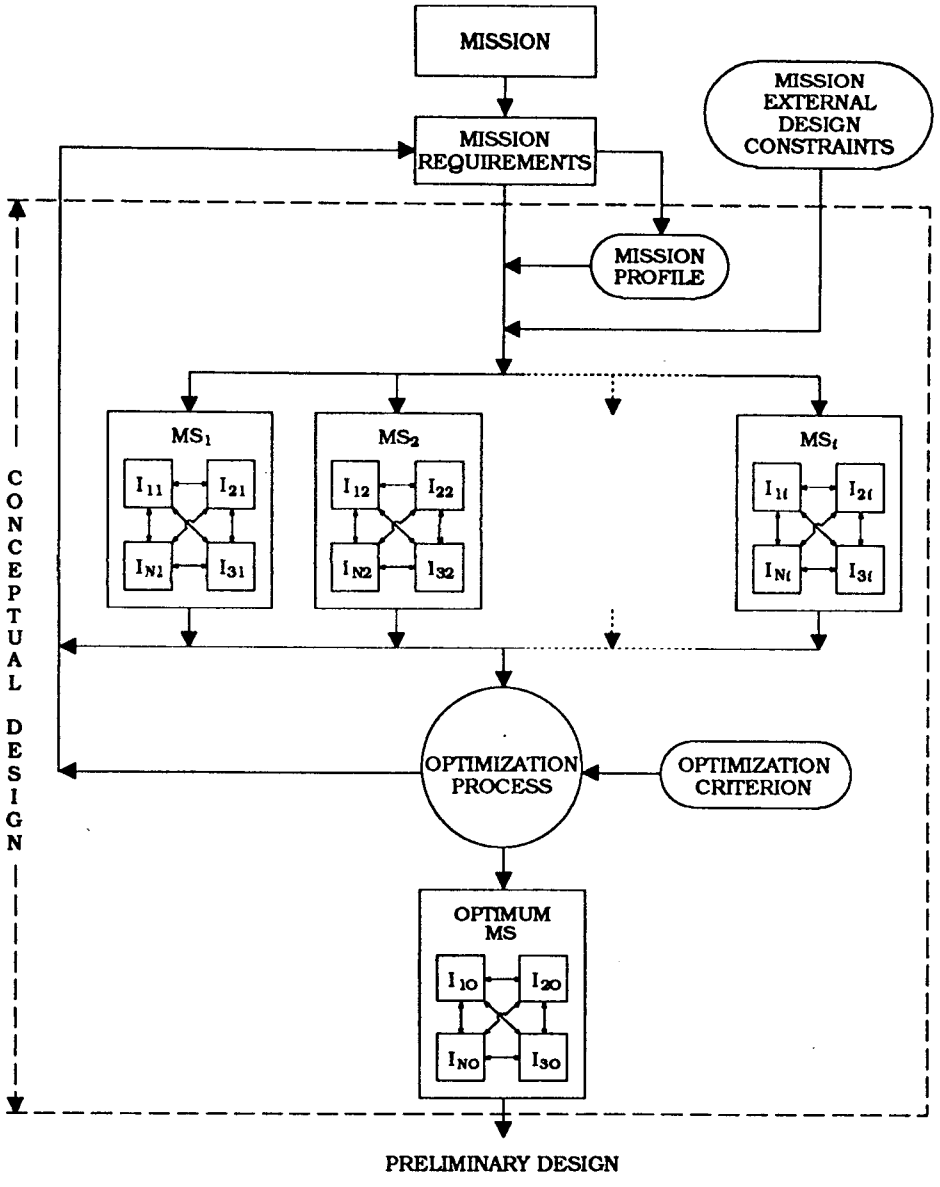


Fig. 4 Conceptual design of mission system

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 on-board 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

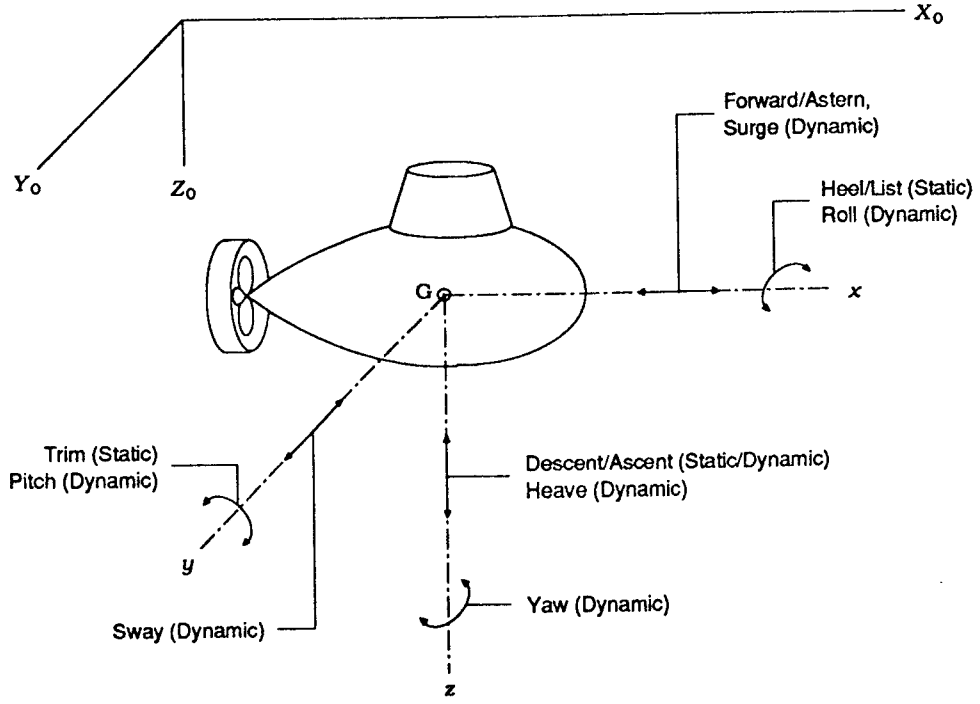


Fig. 5 Degrees of freedom—static/dynamic control systems

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

weight/space criteria, which will be discussed in Subsection 4.5 under "Technological Constraints." chapters of this publication and references noted provide details on these constraints.

4.5 Mission External Design Constraints

Following is a brief summary of the various sources of MEDCs which place constraints on the design of manned submersibles, given here to provide a general perspective on this subject. Other

Environmental constraints—Submersible operations, from launch to recovery, involve several or all of the following environments and their interfaces:

1. Atmosphere
2. Surface interface
   a. Air/water interface

Table 1 Environmental design considerations

Environment	Consideration
1. Atmosphere	
(1) Weather	navigation–communications
	structure (wind forces)
	stability (wind forces, icing)
(2) Energy Propagation	navigation–communications
2. Air/Sea Interface	
A. Open Water	
(1) Surface Waves	structure (wind forces, corrosion–splash zone)
	propulsion–maneuvering
	stability (wind forces)
	regular operations (launch/recovery, other)
	emergency operations (submersible lost, etc.)
(2) Surface Currents	structure (current forces)
	propulsion–maneuvering
	regular operations (launch/recovery, other)
(3) Air/Water Density Difference	propulsion (wavemaking resistance)
	communications (acoustic problems)
B. Ice Cover	
(1) Ice (partial)	structure (moving ice forces)
	navigation
(2) Ice (solid)	structure (frozen in, icebreaking)
	navigation–communications (submersible-to-surface)
	life support (submersible)
	energy (source/capacity)
3. Water Column	
A. Shallow	
(1) Surface Waves	regular operations (launch/recovery, other motion in shallow zone)
	emergency operations
	structure (wind forces)
(2) Currents	propulsion–maneuvering
	structure (current forces)
	regular operations (launch/recovery, other)
	emergency operations
(3) Proximity of Air/Sea Interface	propulsion (wavemaking resistance)
(4) Pressure	human systems (pressure on body in ambient pressure systems)
	considerations similar to those for deep water column
(5) Temperature, Salinity, Density (variable), Energy Propagation	navigation–communications (acoustic problems)
	control systems (ballasting)
	human systems (life support in ambient pressure systems)
	considerations similar to those for deep water column
(6) Natural Light	optical systems (visibility–maximum penetration of about 1000 feet deep under best conditions)
(7) Artificial Light Transmission	considerations similar to those for deep water column
(8) Conductivity	considerations similar to those for deep water column
(9) Flora/Fauna	considerations similar to those for deep water column

Table 1 ***Continued***

Environment		Consideration
3. Water Column *continued*		
B. Deep		
	(1) Currents (usually low—less than or equal to 0.5 knot)	propulsion–maneuvering
	(2) Pressure	structure (pressure-resisting shell structure) compensating systems electrical/mechanical systems (exposed to pressure) fixed buoyancy systems
	(3) Temperature	structure (notch sensitivity of materials at low temperature) life support system
	(4) Temperature, Salinity, Density (stable), Energy Propagation	structure (corrosion) electrical/mechanical systems (exposed to the environment) navigation–communications (acoustic)
	(5) Artificial Light Transmission (about 30 feet maximum)	optical systems–visibility electrical system (external lights) mission systems (still and movie cameras, TV, etc.) navigation/mission systems (low light transmission necessitates use of obstacle avoidance, side-scan and other sonars—acoustical visibility replaces optical visibility)
	(6) Conductivity	electrical systems (loss of power into water, connector problems, other) materials (galvanic action)
	(7) Turbidity and Turbidity Currents	optical systems–visibility propulsion–maneuvering protective systems (prevent fouling, accumulation of silt, etc.)
	(8) Flora/Fauna	protective systems (biological fouling, damage to exposed systems) propulsion (biological fouling)
	(9) Internal Waves (not much is known about these)	control systems (depth control)
4. Sea/Seafloor Interface and Sub-Seafloor		
	(1) Topography	navigation–communications maneuvering protective systems (against collision) siting (of cable/pipeline routes, seafloor structures)
	(2) Nature of Seafloor Surface and Sub-Seafloor Material	soil mechanics considerations (composition, shear strength, scouring, etc.) siting (of seafloor structures, etc.) optical systems (visibility—silt-clouds, etc.) navigation–communications (acoustic problems—sound reflections) protective systems (against accumulation of silt and mud—added weight)

b. Ice/water interface
c. Air-ice/water interface

3. Water column
   a. Shallow water column
   b. Deep water column
4. Water/bottom surface
5. Sub-bottom

Each segment of this vertical profile of environments influences a host of design considerations. An overview of these influences is given in Table 1. Details of these constraints are to be found in Chapter III, "The Environment," as well as in most of the other chapters of this publication, since environmental considerations permeate all aspects of design.

The shallow- and deep-water columns require brief explanations at this point; the other environmental segments are easily visualized. In this regard, the water column can be viewed from three perspectives: (1) the effect of surface waves, (2) the generation of wavemaking resistance and (3) the variability or constancy of environment parameters, primarily water temperature. In (1), the depth D_{sw}, separating the shallow and deep water columns is approximately

$$D_{sw} = \frac{\lambda}{2}$$

where λ is the significant surface-wave length. Surface waves are felt above this depth, causing motion

of the submersible. In perspective (2), the depth, D_{WR}, separating these columns is about

$$D_{WR} = Fd_M$$

where F is a factor that increases with the speed of the submersible to a maximum value of approximately 3.0 and d_M is the vehicle's maximum diameter. The forward motion of the vehicle generates surface waves, creating wavemaking resistance, when operating in depths above D_{WR}. Below this depth, no surface waves are generated and the wavemaking resistance is zero. Finally, in (3), the water temperature varies with depth in the shallow-water column in complex ways. These variations of the water column, called "thermoclines," change with latitude, season and, very near the surface, even with the time of day. In this regard, the depth, D_E, separating the two columns is about 10 000 ft (3048 m). In the deep water column, the water temperature remains reasonably constant at about 32°F (0°C) almost everywhere. As will be seen, the variation or constancy of temperature affects the design of the submersible's control, communications and navigation systems.

Technological constraints—These constraints are associated with technological areas pertinent to the design of manned submersibles, including

1. Materials
2. Structure
3. Machinery and equipment
4. Energy sources and conversion
5. Propulsion and maneuvering
6. Control
7. Navigation and positioning
8. Communications
9. Data processing
10. Life support and habitability
11. Instrumentation and displays
12. Handling and transport

Constraints related to items within these areas arise from a few or several technical considerations, including

1. Safety
2. Reliability
3. Producibility
4. Maintainability
5. First cost
6. Efficiency
7. Simplicity of operation
8. Characteristics—weight, space, power requirements, and the like

Priority ranking of these considerations as they are applied to items within the various technological areas depends on design criteria in effect, including

1. State-of-the-art status
2. Minimum cost
3. Minimum weight/space

State-of-the-art status may involve several technological considerations. However, it always conveys first priority to *safety* and *reliability*; that is, the item in question must first of all be judged safe and reliable before other considerations come into play. The minimum cost criterion, imposed in the interest of obtaining a relatively low-cost submersible, gives second priority to *first cost* and cost aspects related to *producibility, maintainability,* and *efficiency.* On the other hand, the minimum weight/space criterion, promoting the design of a relatively light and small submersible, gives second priority to *efficiency* and the items' *weight/space* characteristics. If the state-of-the-art status criterion should not obtain to a particular item, either of the other two criteria would hold sway. An illustration of technological constraints, elaborating on a previous example, follows.

A designer, in the early stages of a design, is required to select a battery based on the criterion that it have state-of-the-art status, thus giving safety and reliability considerations first priority. A survey of the "energy source and conversion" technological area reveals that this criterion constrains that choice to two battery types, A and B. Further, extensive knowledge of this area convinces the designer that no other battery type will have attained this status by the time the battery is to be installed. The other considerations now come into play—primarily, first cost, efficiency, weight, and space. In this instance, the last three considerations can be melded into efficiency in terms of energy density (watt hours per unit weight or watt hours per unit volume). In this example, battery A is much less expensive than battery B, but its energy density is significantly lower. Consequently, if the minimum cost criterion is operative, battery A will be selected and submersible performance depending on energy storage capacity will be constrained by A's low energy density. On the other hand, if the criterion of minimum weight/space is operative, battery B will be selected, which will reduce constraints on performance but will result in a more costly submersible. If performance related to energy capacity is held constant, use of A will result in a larger, heavier, more difficult and expensive to handle but less costly submersible than if B were used. The conflict in cost aspects of the battery choice is representative of the many questions raised in attempting to optimize the mission systems' design on the criterion of cost-effectiveness.

As has been indicated, the extent and nature of technological constraints, from the view of a few or all of the technical considerations, vary with the passage of time. Consequently, the designer must keep abreast of developments in all technological areas. An up-to-date reference file of technical literature on components in these areas is indispensible. Chapters IV, VI, and VIII, entitled "Materials," "Structural Principles," and "Submersible Design and Operating Safety Considerations" are concerned with the major source of MEDCs.

Economical constraints—These constraints are associated with the economic climate, in terms of supply and demand, and with the economic status of the user/owner. In a free society, they are always present in varying degrees—from severe enough to prevent the initiation of a design or to cause its termination, to mild enough to be minor factors in design decisions involving cost considerations.

The supply of and demand for submersible services are determined, of course, by market analysis procedures critical to the financial well-being of any organization designing/constructing or owning/operating submersibles. These analyses serve various purposes, including the following: providing bases for the development of mission requirements; indicating technological areas needing research and development if the organization expects to remain ahead of, or at least even with, competition; and placing constraints on profit margins. The third purpose is of interest here and is illustrated by the following example. Suppose that a certain organization's market analysis for a particular area of submersible service reveals that the supply/demand balance favors demand, indicating keen competition from other organizations to meet this demand. If it is assumed that all organizations possess about the same level of expertise, the "cost" side of the "cost-effectiveness" criterion becomes critical in determining the successful competitor, "effectiveness" side issues presumably being neutralized by the equality of the competitors' expertise. If reasonable profit margins are to be realized, this situation places severe constraints on the economic aspects of the technological considerations, including, of course, first costs. It may also cause profit margins to shrink to the point at which the organization in question decides to abandon that particular market or to increase its research-and-development efforts to improve its competitive position from the "effectiveness," as well as "cost," perspectives.

The economic status of the user/owner, quite obviously, places varying degrees of economical constraints on the design of the submersible as well as the design of the mission system. This status initially constrains mission/performance requirements, limiting them to prescribing what can be accomplished realistically in a safe and efficient manner with given funds for acquisition and operation of the overall system. For a particular set of these requirements, this status, of course, acts to constrain design through its impact on the economic aspects of the technological considerations.

Legal/quasi-legal, political, sociological constraints—These sources of MEDCs are interrelated. For example, environmentalists can express concerns over pollution of the marine environment to the point at which they become political issues and, eventually, legal acts. In this instance, these constraints impact on submersible design by requiring that no foreign material be injected into the sea—this requirement, for example, necessitating the use of human waste/waste-water holding tanks and preventing the trim/list system's mercury from being dumped overboard to gain positive buoyancy for emergency ascents.

Constraints from these sources may be classed as *compulsory* and *noncompulsory constraints*—only the legal constraints being compulsory. Sociological and political constraints may impact directly on design in a noncompulsory manner rather than through the "domino effect" cited in the example above. In this regard, user/owners and designers may elect to comply with the attitudes of groups of people or political persuasions to gain good will and all it implies for their organizations. For instance, with other considerations aside, these are noncompulsory constraints militating against the use of nuclear power by nonnaval submersibles.

Legal/quasi-legal constraints are the most important of this group to submersible design. Quasi-legal constraints, while noncompulsory in fact, often have the effect of legal constraints. Sources of these MEDCs include classification societies and certifying agencies. The American Bureau of Shipping (ABS), Lloyds Register of Shipping (LR), Det Norske Veritas (DNV) and Bureau Veritas (BV) are examples of classification societies which are scattered throughout the world. Their rules are contained in publications such as ABS's "Rules for Building and Classing Underwater Systems and Vehicles." The U.S. Navy is a certifying agency, certifying all submersibles to be used by Navy personnel. It should be stressed that a submersible does not have to be classified or certified, but failure to be so will adversely affect its useability and insurability. Chapter VIII, "Submersible Design and Operational Safety Considerations," deals with many aspects of this subject, particularly from the view of the certifying process.

Legal constraints are compulsory, the principal source of these constraints in the United States being the U.S. Coast Guard. In this regard, however, constraints on U.S. submersibles are meager. If they carry no more than six "passengers," they must comply with "Rules and Regulations for Uninspected Vessels," which requires that a minimum amount of safety-related equipment be on board: a fire extinguisher, life preservers for each person on board, and so forth. Larger submersibles which carry more than six "passengers" are subject to more strict and complex rules and regulations paralleling those for surface ships in similar size ranges.

4.6 Mission Internal Design Constraints

Figure 1 indicates that MIDCs come from two sources within the mission system's boundary: from the interfacing of the submersible with other (1) sys-

tems and from the interfacing of (S) and smaller systems within the submersible (I) systems' boundary. An untethered submersible interfaces with other (I) systems in handling, navigation, communications, and rescue/salvage activities. Handling activities place mild to severe constraints on weight and size of the submersible as well as on the design of its exostructure. Navigation and communication activities impose constraints merely to the extent of requiring compatibility between internal (on the submersible) and external (on the support ship and seafloor) components of these systems. Rescue/salvage activities also impose constraints in requiring compatibility between internal and external (on the rescue/salvage system) components. In the past, little compatibility has been achieved with these activities being accomplished on a more-or-less "jury-rigged" basis. Standardization of rescue/salvage systems for small and large submersibles is needed to overcome this critical shortcoming.

The tether of a tethered submersible, of course, places severe constraints on the vehicle's mobility and maneuverability; that is, it is no longer free to move in three dimensions at will. On the other hand, constraints on submersible systems using energy are removed by an "unlimited" supply of energy through the tether, and communications are improved by "hard-wire" transmission.

MIDCs at the interface of a submersible's (S) and smaller systems arise because of incompatibilities between characteristics of two or more systems—incompatibilities which have the potential for impairing the functions of these systems or causing their failure. This potential is removed by placing constraints on system location, protection, and operation. For example, consider once again the characteristics of the high-pressure and human systems and the potential the former system possesses for impairing or causing failure of the latter. The human system, then, imposes the constraint on the high-pressure system that its air flasks be located outside the pressure hull. As a second example, consider the characteristics of an ambient-pressure life-support system, using helium, and a system whose instrumentation is damaged by exposure to this gas. In this case, the life-support system imposes the constraint, or requirement, that the other system's instrumentation be housed in special, helium-proof containers to protect against the intense invasiveness of the gas. A third and obvious example considers the characteristics of a propulsion system, utilizing an open-cycle diesel-generator set, and the human system. The human system places the constraint on the diesel that it be operated only on the surface—that it be stopped immediately on submerging to avoid drawing a vacuum in the pressure hull and exhausting the oxygen in the hull prior to stopping of its own accord.

Many MIDCs exist involving even the smallest of systems; say, a nut-bolt and plate assembly of dissimilar metals which must be protected from each other by an insulating material to avoid galvanic action. It is obvious from the few examples given that the designer must be intimately acquainted with the characteristics of all systems. Again, many of these characteristics may change with the passage of time, and consequently, an up-to-date reference file on them must be maintained.

5. Basic Design of Manned Submersibles

5.1 Overview

Basic design, as considered herein, consists of the *conceptual* and *preliminary design stages*. Conceptual design develops two or more design alternatives from which a design is selected in accordance with specified criteria. Preliminary design advances the selected alternative's development to the point of entry into the contract design stage. This section is concerned primarily with conceptual design, background for the more detailed procedures of preliminary design being given in other chapters of this publication and in references provided.

This section discusses two approaches to conceptual design: the *empirical approach*, which is also useful in the earlier phases of preliminary design, and the *systematic parametric analysis approach*. The empirical approach is modeled by the well-known *design spiral*, which also serves as a basis for brief discussion of the entire design process. Guidelines associated with the design spiral concept are given. Details of the systematic parametric analysis approach are provided by an example of a design optimization program given in the Appendix.

The section is introduced with a discussion of the initial objective of basic design—the selection of a conceptual design alternative from among the number of alternatives developed. In this discussion, it is assumed that selection is based on the criteria of technical feasibility and cost-effectiveness. Associated procedures are often referred to as feasibility and optimization studies, the latter term being appropriate when the number of alternatives considered is large and the probability of identifying the optimum design is relatively high.

It is well to reiterate at this point that submersible design procedures, as well as those for mission-system design, vary widely according to practices of individual designers and the degree of complexity of the design under development. Consequently, material in this section constitutes suggested, rather than hard-and-fast, guidelines.

5.2 Conceptual Design—Selection of Alternatives

The conceptual design process for submersibles parallels this process for the mission system as dia-

grammed in Fig. 4: mission system alternatives and combinations of individual, or (I), systems being replaced by submersible system alternatives and combinations of various submersible (S) systems. In paralleling mission system design, the goal of feasibility and optimization studies in conceptual design is *to select the most cost-effective design(s) from among all of the viable alternatives considered. "Viable alternatives" are those which meet performance requirements and are technically feasible from the producibility/availability, operability, safety,* and *reliability points of view.* It should be observed that the "optimization," in this sense, is a relative term which must be qualified by the expression *to the extent feasible*—to the extent feasible, considering design constraints acting. As has been seen, these constraints form a framework for the design—those designs within the framework being viable alternatives, while those without are nonviable alternatives, which must be discarded. Returning to the statement's key words:

- *most cost-effective* brings economic and technological MEDCs into play;
- *performance requirements*, providing foci for the development of the submersible's characteristics, reflect legal/quasi-legal MEDCs imposed principally by classifying societies and certifying agencies, such as factors of safety used in arriving at the design depth;
- *technically feasible considerations* are based on a host of environmental and technological MEDCs as well as certain legal/quasi-legal MEDCs;
- *safety* and *reliability*, included in technically feasible considerations, invoke use of the design criterion requiring state-of-the-art status for critical, or "man-rated," systems and subsystems as well as for those among these systems which have a significant impact on cost-effectiveness.

An appreciation for the submersible design's role in mission system optimization was conveyed in Subsection 2.3 with details of the cost-effectiveness equation being given in Fig. 2. With reference to that figure, it was seen that mission system optimization involves minimizing the product

$$\frac{\text{dive days}}{\text{mission task}} \times \frac{\text{operating cost / year}}{\text{dive days / year}}$$

to the extent feasible, this objective often being achieved when neither term has a minimum value itself. Submersible design contributes to this process essentially in two ways: (1) by reducing the impact of MIDCs it exerts on other individual mission systems to the extent possible and (2) by minimizing this product for the submersible itself, again, to the extent possible. Design considerations for the first point include

- minimizing the submersible's size and weight in air to the extent feasible, thus reducing the impact of these MIDCs on the transportation and handling systems, and
- adopting a design philosophy stressing reliability, simplicity and ruggedness, thereby reducing the MIDC impact on shore- and sea-based maintenance and repair systems.

In general, these considerations also serve the submersible design's best interests. For example, minimizing size and weight also minimize drag and enhance maneuvering, while the design philosophy cited promotes the submersible's cost-effectiveness. In this regard, it is evident from studying Fig. 2 that the second consideration is the submersible's primary contribution tending to maximize dive days per year, while also tending to increase task site hours per dive day and to decrease hours per mission task and the submersible's annual operating cost.

Design considerations for the second point focus on the feasibility and optimization studies of conceptual design, which involve the development of

- *performance requirements* derived from mission requirements pertinent to the submersible system and which set forth the capabilities on which selection of the submersible's characteristics are based,
- *submersible characteristics* necessary to meet these requirements and on which the selection of systems and subsystems are based, and
- *cost estimates* including both capital cost, as amortized, and annual operating cost associated with the selection systems and subsystems.

In the designer's pursuit of these studies, the submersible may be decoupled from the rest of the mission system if certain assumptions are made. They are that (1) the characteristics and operating costs per year of other (I) systems of the mission system are not changed significantly by variations in size and weight of the submersible design alternatives considered and (2) all alternatives are composed of reliable systems. Then, the submersible's cost-effectiveness equation may be written as

$$\frac{1}{CE} = \frac{\text{sub cost}}{\text{mission task}} = \frac{\text{dive days}}{\text{mission task}} \times \frac{\text{sub operating cost / year}}{C} \quad (2)$$

where, for the assumptions made, it may be further assumed that the term *dive days per year* is a constant, C, for all alternatives developed. The most cost-effective alternative may be identified, leaving C as an unknown constant. Obtaining values of sub-cost per mission task requires that C be evaluated for the mission system alternatives considered. As previously observed, submersible design optimization involves the fine art of balancing antagonistic considerations in minimizing the products of the two terms of equation (2). Decreasing dive days per mis-

sion task, or increasing the submersible's productive time, tends to increase submersible operating costs per year, that is, decreasing the first-term necessitates higher performance requirements which require system components usually costing more money.

Equation (2) is applicable to many design studies and is the basis for continuing the discussion of design optimization in this chapter. However, for studies in which the assumptions noted are not valid, the submersible should not be decoupled from the rest of the mission system. For example, supposing submersible design alternatives developed involve a wide variation of vehicle size and weight—wide enough to require different sizes of support ships and launch/retrieval systems having different operating costs per year. In these cases, then, optimization studies should be based on the mission system as a whole and equation (1) used.

Figure 6 conveys an impression of the optimization process and the manner in which conceptual design alternatives may be generated. A given mission can be met by n sets of performance requirements, each set leading to a certain dive days per mission task achievable by the requirements of that set. In this illustration, only one of the n sets is shown in the interest of reducing the diagram's complexity. Associated with this one set are i design alternatives composed of various combinations of submersible systems and subsystems in which, for example, different pressure-hull materials, different energy-storage systems, different types of manipulators, and so forth are used. All alternatives meet the given set of performance requirements, but only those within the framework of design constraints acting are technically feasible and become viable design alternatives. Each of these alternatives becomes the basis for calculating the associated submersible operating cost per year/C, which, when multiplied by dive days per mission task, yields one value of submersible cost per mission task. It is this product which should have a minimum value for optimization considering design constraints acting.

One of the i design alternatives is shown in Fig. 6 lying outside the framework of design constraints acting to provide an example of a nonviable alternative. Although meeting performance requirements, it is not technically feasible. Suppose, for instance, that in the designer's effort to reduce vehicle size and weight, an alternative uses a battery with a very high energy density which is currently under development. Suppose further that, on investigation, the designer decides that prospects for the battery attaining state-of-the-art status by the time needed are

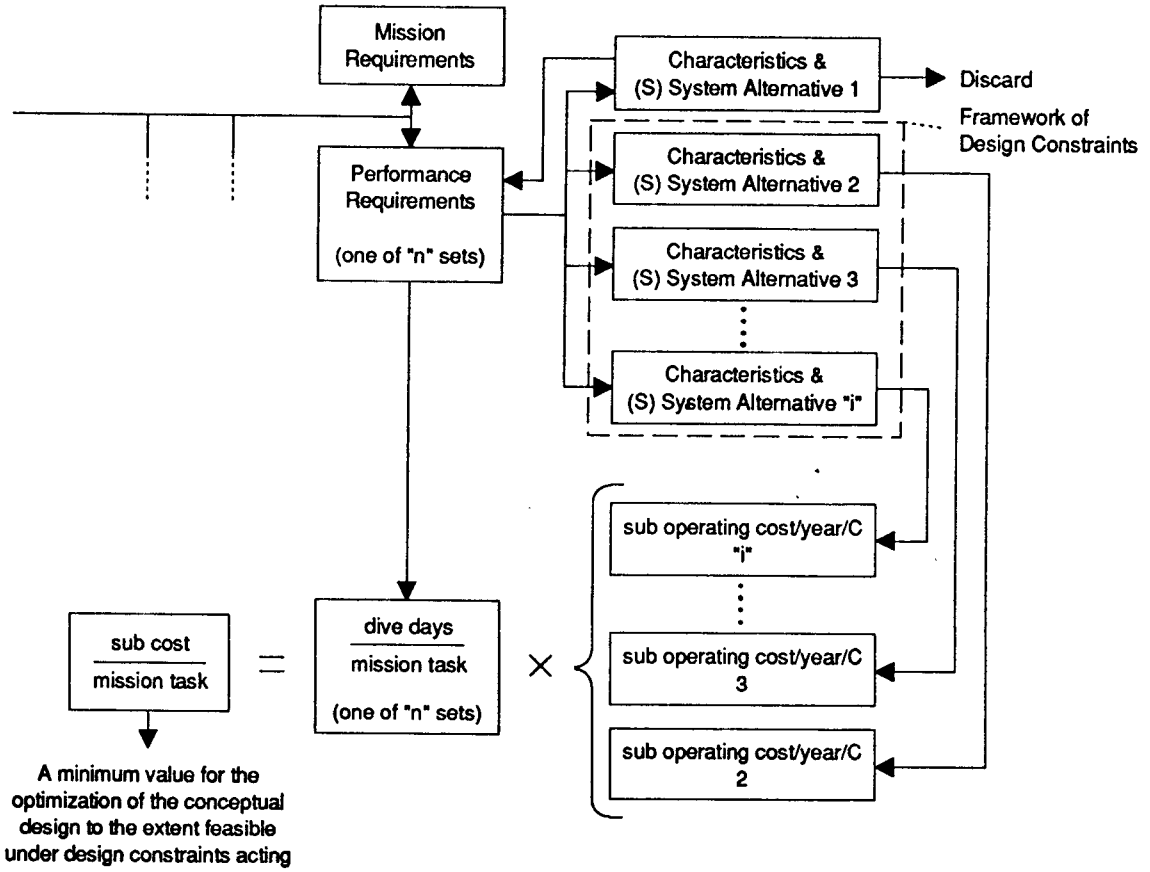


Fig. 6 Conceptual design alternatives and optimization

doubtful at best. In this case, the alternative is discarded as being nonviable. Though discarded, the alternative is useful in providing feedback to performance requirements, as indicated in Fig. 6—raising questions concerning the excessiveness of requirements involving large expenditures of energy. In most instances, these questions are relayed to related mission requirements as illustrated in the figure. It may be decided in owner/user-designer consultations that alterations in these requirements will facilitate the search for the most cost-effective, viable alternative.

The number of design alternatives investigated, N, is given by the expression

$$N = i_{\text{SET }1} + j_{\text{SET }2} + \cdots + m_{\text{SET }n} \qquad (3)$$

where i, as indicated in Fig. 6, is the number of alternatives associated with the first set of performance requirements, j is the number associated with the second set, and so forth. The magnitude of N varies with the approach taken to conceptual design, being relatively small for the empirical approach, larger for the systematic parametric approach using longhand methods and very large for this approach using computers. Consequently, the latter method is well suited for conducting optimization studies.

The empirical approach relies on experience and data accumulated from past designs and submersible operations, these data being organized so as to be useful design tools (graphs, tables, empirical formulations and so forth). Data for a new design are extracted by interpolation/extrapolation from information on submersibles having reasonably similar mission requirements. Further, since N is relatively small for this approach, these submersibles should have successful operating histories. It is reasonable to assume, then, that their designs are at least very satisfactory, if not optimum, and that they will serve as good "points of departure" for the development of new design alternatives. As many alternatives as possible should be developed to explore the results of both variations in mission requirements from those of similar submersibles, which should be relatively small, and changes in technological, economic, and other MEDCs that may have occurred since the similar submersibles were designed and constructed. Studies using the empirical approach do not result in design optimization but in identifying the most cost-effective, viable alternative from among the number of alternatives considered.

The systematic parametric analysis approach explicitly expresses relationships between dependent and independent design variables as a series of parametric equations to be solved simultaneously in arriving at a particular design alternative. For example, performance requirements having significant impact on dive days per mission task and submersible operating costs per year may be selected as independent variables, or parameters, and varied systematically in a search for the optimum combination of dependent variables such as submersible size, weight, cost, and effectiveness in performing mission tasks. As indicated above, this approach can be followed using longhand or computer methods—the latter is used predominantly today. It should be emphasized that the systematic parametric analysis approach is by no means free from empiricism: its parametric equations are based on empirical formulations and coefficients derived from existing data. The quality of these equations and associated computer programs, of course, can be no higher than the quality of the input. Consequently, a periodic review and adjustment of input data is necessary to ensure that this input reflects current technological, economic, and other pertinent information.

5.3 Empirical Approach Guidelines

These guidelines are associated with the empirical approach to basic design and utilize the design spiral concept shown in Fig. 7. They are intended to provide a perspective, rather than details, of the design process. Details and underlying principles are to be found in this and other chapters of this book, as well as in references given.

Numerous perspectives on the design process exist. Consequently, guidelines presented herein, including the design spiral and nomenclature associated with it, should be viewed as presenting but one approach.

Mission requirements—As indicated in Fig. 7, the basic inputs to the design process are those overall mission requirements that are pertinent to the submersible's design. While they may be stated in many ways, they should include the following information:

1. Task site
   a. Location
      (1) area of the world
      (2) seafloor or water column site
      (3) maximum operating depth
   b. Type
      (1) point or separated points—site at one "point" or more than one "point" separated by specified distances
      (2) line—site along line(s) of specified distance(s), as would be the case in a pipeline inspection mission
      (3) area—site over an area of specified dimensions, as would be the case in a seafloor area mapping mission
2. Task Work
   a. Category
      (1) inspection
      (2) maintenance/repair

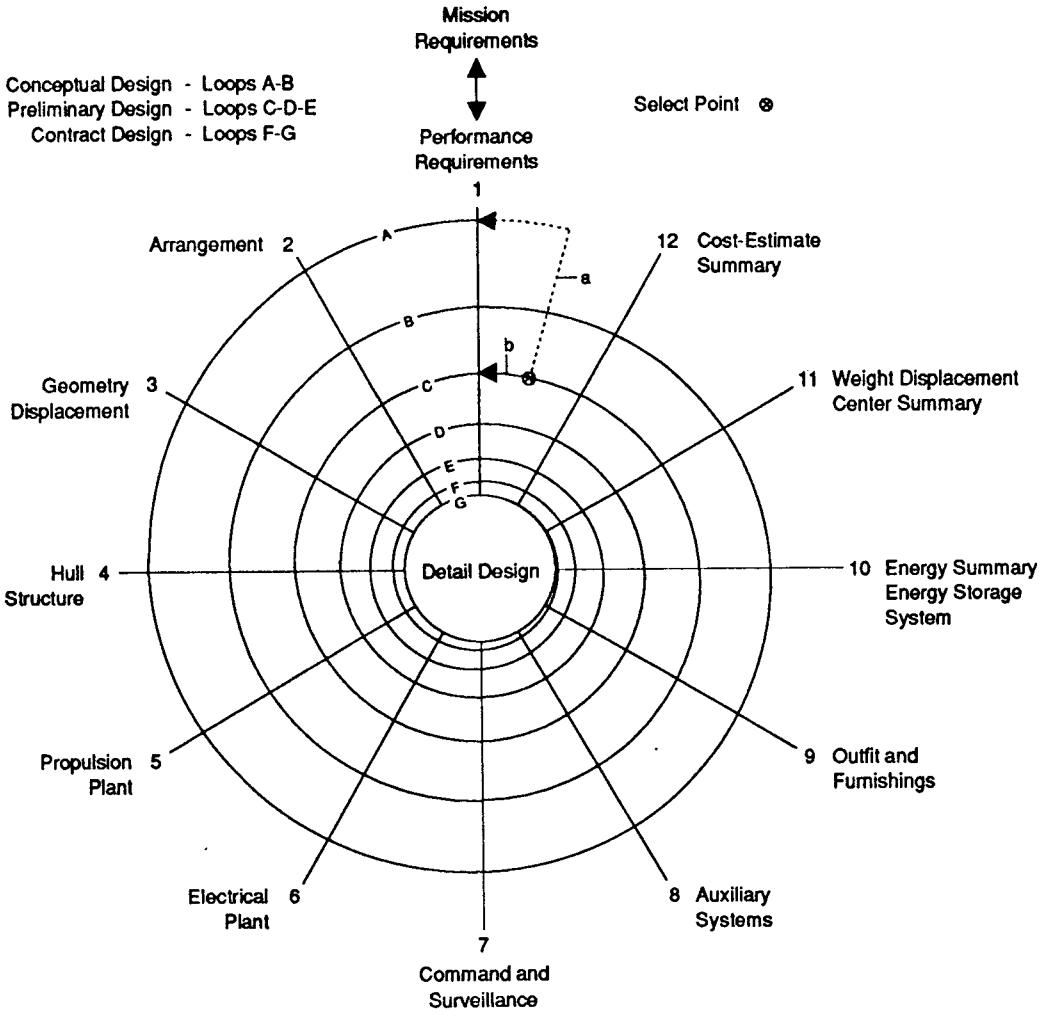


Fig. 7 Design spiral

(3) other

b. Work object(s)
   (1) general—a general description of the work object such as a buried pipe, an open-frame fixed platform or a wet-well head
   (2) specific[2]—as extensive information as possible needed to formulate adequate maneuvering and mission subsystems performance requirements facilitating proper vehicle positioning, vehicle/object orientation and mating of tools and other equipment—a critical mission requirement in affecting a submersible's productive time

c. Task description
   (1) work to be performed—data to be acquired, nondestructive inspection to be performed, values to be replaced and so forth
   (2) loads to be transported—from surface to site or site to surface or both. These loads are part of the overall payload which also includes weights of all items used exclusively in performing mission task work

d. On-board personnel—required to accomplish work
   (1) number
   (2) type—one-atmosphere or ambient pressure (diver) types

e. Response time—time allotted for the submersible to reach the work site, if critical in performing mission tasks

[2]This mission requirement highlights a "two-way-street" situation. The designers of systems requiring submersible services should design their system with submersible capabilities and mating requirements in mind as well as vice versa.

The design spiral—The iterative nature of the overall design process may be modeled by the design spiral shown in Fig. 7. The spiral consists of *spokes* and *loops*. The spokes represent *design considerations* including *performance requirements, submersible characteristics,* and *cost estimates*; submersible characteristics are embodied in its arrangement, geometry, six SWBS systems groups, and summaries of energy requirements and weight/displacement centers. The loops indicate *design iterations* with the design's refinement increasing as the loops spiral inward. Outer loop procedures use empirical formulations and estimates based on similar design data, whereas middle and inner loop procedures are based on progressively more detailed development of the design considerations utilizing increasingly precise formulations. The intersection of spokes and loops form "points" which are identified alphanumerically. Thus, for example, 3B is a point in the spiral at which the geometric characteristics of the submersible are being considered in the second iteration in a more detailed manner than they were initially at point 3A. At 3B, adjustments are made as the result of both the more detailed consideration of the geometric characteristics required and the impact on geometry of interrelated design considerations for which data were generated at points in the spiral preceding 3B. Adjustments are made at other points in the spiral for the same reasons. For the design to converge to a satisfactory solution of the design problem, these adjustments must become progressively smaller with each succeeding loop through conceptual, preliminary, and contract design. This fact is demonstrated on the spiral by the decreasing spacing between loops as they spiral inward. Convergence is illustrated by the innermost loop becoming a circle, indicating that all design considerations are fully developed and require no further adjustments. At this point in the spiral, detailed design can begin.

Conceptual design feasibility studies are somewhat arbitrarily confined to the A and B loops of the design spiral; it is recognized that other approaches to design may involve a different number of loops. One trace around these loops to the "select point," shown in Fig. 7, generates one conceptual design alternative or feasibility study. The "select point" may be thought of as a repository for completed alternatives—the point at which these alternatives are compared and the most cost-effective, viable one is selected in a manner indicated in Subsection 5.2. Paths (a) and (b) are shown issuing from this point. Path (a) leads back to 1A and the beginning of a new conceptual design alternative based either on changes in performance requirements, which call forth a new set of submersible characteristics and associated cost estimates, or on a new set of characteristics and cost estimates which satisfy the same performance requirements. Path (b) leads to the beginning of preliminary design and is followed by the conceptual design emerging from the "select point" as the optimum alternative from among those generated. Quite obviously, the likelihood of selecting the optimum alternative is limited by the relatively low number of total alternatives that can be considered by the empirical approach.

Preliminary design is also arbitrarily limited to loops C, D and E. Note in Fig. 7 that loops D and E are relatively close together, indicating that only very small adjustments in design considerations are being made at this point. Should preliminary design's more detailed procedures and exact formulations reveal the need for relatively large adjustments to be made in one or more of the considerations, the validity of the selected conceptual design alternative may well be questioned. The end result of preliminary design is a well-defined submersible system which meets performance requirements and for which a firm cost estimate exists. As indicated on the spiral, completion of this phase of design marks the beginning of contract design.

The A loop—A loop procedures are primarily concerned with the approximate *sizing of the submersible* and utilize empirical formulations and similar design data extensively. Sizing involves three steps: selecting the geometry and the major elements of the SWBS systems to be used for the particular conceptual design alternative under study; obtaining first estimates of sizing data for these systems; and the summarizing of these data to estimate the size parameters of total weight and displacement in the neutral buoyancy condition and the total cubic. Points on the A loop are now considered.

1A Performance Requirements: Derive performance requirements from the mission requirements pertinent to the submersible's design. These performance requirements should include

1. Depth
   a. Maximum operating depth (___ m or ft)
   b. Design or collapse depth (___ m or ft)
2. Range (___ nautical miles)
3. Speed
   a. Forward speed profile (___ hours at maximum speed of ___ knots, ___ hours at cruise speed of ___ knots, ___ hours at search speed of ___ knots)
   b. Descent/ascent speeds (___ m/s or ft/s)
4. Maneuvering
   a. Maneuvering emphasis—on position changing/keeping or path changing/keeping
   b. Dynamic (___ degrees of freedom in specified directions)
   c. Static (___ degrees of trim, ___ degrees of list)
5. Personnel on board

a. Crew (___ at one atmosphere)
b. Non-crew (___ at one atmosphere, ___ at ambient pressure)

6. Endurance
   a. Life support (___ hours or days normal, ___ hours or days emergency)
   b. Energy for propulsion and auxiliary or hotel power loads (___ hours or days normal, ___ hours or days emergency)
7. Payload
   a. Items fixed during dive—all items used exclusively in performing mission task and located inside or outside pressure hull (items listed together with weight/displacement data). Non-crew and effects are included in this item for weight-accounting purposes
   b. Items varying during the dive—variable loads (items listed together with weight/displacement data)
8. Systems requirements—such as may be specified by the owner/user or derived from his requirements
9. Safety features—such as may be specified
   a. Personnel—means of escape or rescue
   b. Submersible—means of emergency ascent or salvage or both
10. Limiting seastates—for submersible operations

2A Arrangement: Initially, the submersible may be considered as consisting of the pressure hull(s), envelope, and all other items comprising it. Select types and number of pressure hulls, type of envelope, and location of major items of SWBS systems either inside or outside the pressure hull.

1. Pressure hull(s)—type and number
   a. One-atmosphere type—submersible must have at least one pressure hull to house crew. It may have more than one hull to isolate crew from non-crew or to reduce diameter of hull if enclosed volume is large
   b. Ambient-pressure type—submersible must have one ambient pressure, in addition to one atmosphere hull, if ambient pressure personnel (divers) are carried
2. Envelope types
   a. Enclosed envelope—favored for relatively high forward and vertical speeds, reduction of potential for entanglement and for silt/mud entrapment and for protection of systems
   b. Non-enclosed envelopes
      (1) open-frame envelopes—favored for maneuvering, access to systems, ease of mounting systems and weight saving
      (2) envelope composed of individual shapes such as pressure hull, battery pods below pressure hull and conical stern section—this type has some of the advantages/disadvantages of the other two types
3. Other item location—guidelines provided by similar design arrangements
   a. Inside pressure hull(s)—personnel and all items of equipment to which they must have ready access
   b. Outside pressure hull(s)
      (1) items which must be outside to perform functions—such as propulsion units
      (2) items potentially dangerous to personnel—such as high pressure air bottles
      (3) items which would require excessive pressure hull volumes and add structural weight if located inside—such as main ballast tanks
      (4) items which would require mechanical penetration if located inside pressure hull thus creating seal problems—such as propulsion motors
   c. Either inside or outside
      (1) items tend to be located inside—for shallow-diving submersibles
      (2) items tend to be located outside—for deep-diving submersibles

3A Geometry—shape and dimensions: Obtain first estimates of pressure hull and envelope geometry and volumes and displacement of submersible.

1. Pressure hull
   a. Shape
      (1) shallow depths—shape governed by hydrodynamic and arrangement considerations. A variety of shapes can be used, cylinders, closed by hemiheads, being the most common
      (2) deep depths—shape governed by structural considerations. Shapes restricted to spheres or nested spheres
   b. Dimensions—Dimensions estimated from volume which pressure hull(s) must enclose. Relationships between volumes and dimensions for various shapes are shown in Fig. 8. One expression for volume, (V_{PH}), is

$$V_{PH} = \frac{nV_m}{(1-C)} \qquad (4)$$

where
n = number of persons in pressure hull,
V_m = cubic/man data from Fig. 9, and
C = fraction of V_{PH} occupied by items inside pressure hull other than persons. Use similar design data.

2. Envelope
   a. Shape

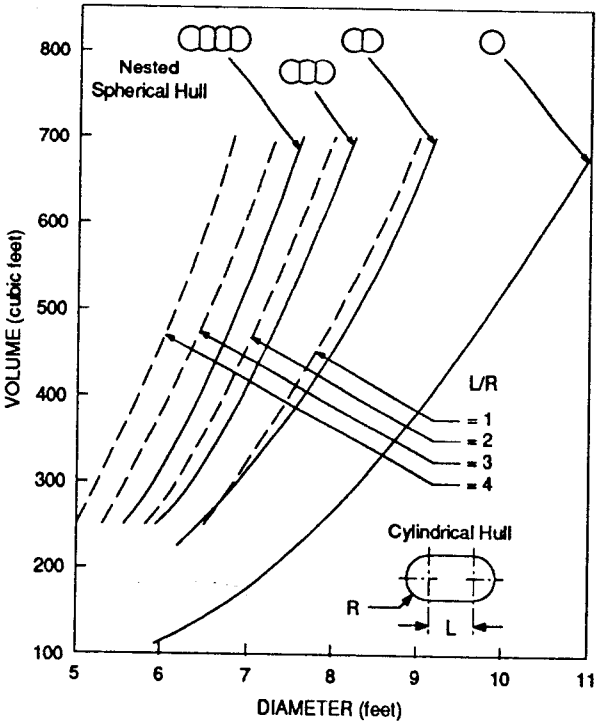


Fig. 8 Pressure hull volume versus diameter (from "Weight Considerations for Deep Submersibles")

(1) enclosed envelope—envelope streamlined in direction of motion for relatively high forward or vertical speeds or both. If not streamlined, it should be a fair form to reduce the potential for entanglement and fouling
(2) open-frame envelope—shape usually approximates a rectangular prism
(3) envelope composed of individual shapes—envelope has no particular overall shape as such

b. Dimensions—Envelopes have circumscribing, or "principal," dimensions of length L, breadth B, and depth D. L can initially be estimated from similar designs; B and D estimates can be based on the frontal areas of similar designs or on the diameter of the pressure hull, now estimated, making allowances for items outside its boundary such as tanks and syntactic foam

c. Envelope volumes, (V_E)
(1) enclosed envelopes—volumes may be estimated from the expression

$$V_E = C_B \times L \times B \times D \qquad (5)$$

where V_E is the volume of displacements of pressure hull and all exter-

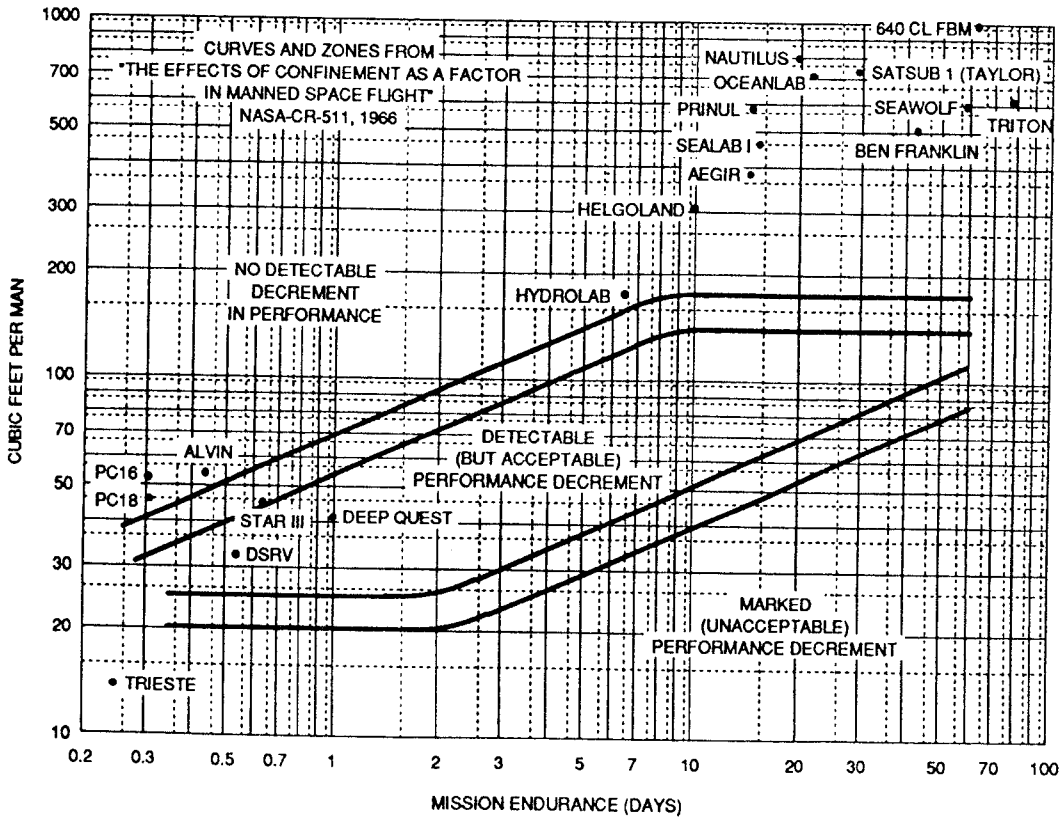


Fig. 9 Submersible habitability

nal items within the envelope plus volume of free flooding water within envelope and C_B is the block coefficient relating principal dimensions and shape which may be estimated from similar designs.

(2) open-frame envelopes—volumes may be estimated from the expression

$$V_E = L \times B \times D \quad (5a)$$

(3) envelope composed of individual shapes—volume may be estimated from the simple expression

$$V_E = \sum V_E \text{ of individual shapes} \quad (5b)$$

3. Displacement of submersible—A first estimate of displacement may be obtained using the expression

$$\Delta = PV_E\gamma \quad (6)$$

where P is the packing factor, derived from similar designs, and γ is the specific weight of seawater. P reflects the efficiency with which the pressure hull(s) and external items can be "packed" into an envelope of given type and shape. 1.0-P is a measure of the amount of free-flooding water filling the void spaces within the envelope.

4A–9A SWBS System Groups: Systems within these groups are described in Section 3 and are listed in Table 2 for ready reference. Two of the sizing procedures should be followed in this portion of the A loop—(1) select major components of systems to be used with the particular design alternative under study and (2) obtain first estimates of sizing data for the systems selected. These data include approximations of weight, displacement, dimensions, space, or cubic occupied, including space required for routine maintenance, and power requirements. Note that estimates of cost data can be deferred until the B loop, when the systems are developed in more detail.

Procedures involving system selection and sizing data estimating are many and varied; their detailed consideration is beyond the scope of this discussion. However, they all demand access to well-organized files on existing submersible design data and manufacturers' technical information. For purposes at hand, this information should include system

1. performance data
2. cost data
3. weight, displacement, center data
4. overall dimensions and space requirements
5. power requirements
6. location requirements
7. protection requirements
8. maintainability
9. reliability

A most useful way of organizing information establishes relationships between system sizing data and performance requirements which have a major impact on the system's design. These relationships are presented in graphical form with performance requirements being independent variables. Prime examples are curves of "$(W/\Delta)_{PH}$ versus Design Depth" for given hull shapes and materials, where $(W/\Delta)_{PH}$ represents the weight

Table 2 SWBS systems and associated variable loads/ballast

	Systems	Variable Loads/Ballast
1. Hull Structure	Pressure Hull	
	Exostructure	
	Foundations	
	Outer Hull	
	Structural Appendages	
	Special Purpose Systems—Fixed Ballast/Buoyancy	
2. Propulsion Plant	Open/Closed Cycle Energy Conversion Systems	Fuels/Oxidants
	Propulsion Batteries	
	Propulsion Thrusters	
	Propulsion Drives	
	Propulsion Support System	
	Special Purpose Systems—Operating Fluids, Spare Parts, Etc.	
3. Electrical Plant	Auxiliary Open/Closed Cycle Energy Conversion Systems	Fuels/Oxidants
	Auxiliary Batteries	
	Emergency Batteries	
	Power Distribution System	
	Lighting System	
	Special Purpose Systems—Operating Fluids, Spare Parts, Etc.	

Table 2 ***Continued***

Systems			Variable Loads/Ballast
4. Command and Surveillance Systems	Command and Control Systems Navigation System Communications Special Purpose Systems—Spare Parts, Etc.		
5. Auxiliary Systems	Human Systems		Crew/Non-Crew and Effects
	Life Support Systems		Food, F. W., Oxygen, Etc.
	Air, Gas, Miscellaneous Fluids		High-Pressure Air, Gas, Etc.
	Static Control:	Main Ballast	Seawater
		Descent/Ascent	Seawater/Weights
		Emergency Ascent	Seawater/Weights
		Trim/List	Seawater
		Variable Ballast/Buoyancy	Seawater/Weights
	Dynamic Control:	Thrust-Producing	
		Lift-Producing	
	Special Purpose:	Handling	
		Rescue/Salvage	
		Auxiliary System Fluids, Spare Parts, Etc.	
		Mission Systems—Internal/External	Variable Mission Loads
6. Outfit and Furnishings	Hull Fittings, Ladders, Gratings Painting/Insulation Furnishings		

of the empty pressure hull divided by its displacement.

The SWBS system points on the design spiral require some comment:

1. 4A Hull Structure: Fixed ballast and buoyancy are included in this group—these systems have the function of providing for neutral buoyancy, zero trim, adequate stability, and a weight margin in design. At this point, only the amount of fixed ballast for the last function is included; its weight is usually 10 percent or less of the total displacement. Fixed ballast or buoyancy serving other functions are considered at Point 11A, weight-displacement-center summary.

2. 5A Propulsion Plant: The propulsion power source—which consists almost always of batteries for untethered submersibles—is included in this group. However, for convenience, consideration of this system's sizing data is deferred until Point 10A, energy summary.

3. 6A Electrical Plant: The nonpropulsion power source for auxiliary and other systems' service loads is included in this group. Again, for convenience, consideration of this system's sizing data is deferred until Point 10A. The remaining systems of this group, the power distribution and lighting systems, constitute very small percentages of Condition A1 (light ship minus fixed ballast/buoyancy) weight and displacements. Consequently, first estimates of weight and displacement of this group, excluding nonpropulsion power source, may be obtained using percentages from similar design data.

4. 7A Command and Surveillance: The total weight and displacement of this group of systems are also very small percentages of Condition A1 weight and displacement. As at Point 6A, first estimates of this group's weight and displacement may be obtained using percentages derived from similar designs.

5. 8A Auxiliary Systems: Certain systems in this group require comment.

   a. Human system—This system is composed of crew and noncrew and personal effects brought on board; noncrew and effects are part of the payload. The system is considered to be a part of variable load even though its weight does not vary during the dive unless divers are involved.
   b. Air, gas, and miscellaneous fluids systems—As noted in Section 3, these systems are primarily those required to expel seawater from ballast tanks and to furnish hydraulic power. The first system involves the storage of pneumatic energy; consideration of its sizing data is deferred until Point 10A.
   c. Static control systems—Care must be taken that items associated with these systems which vary during the dive, such as seawater in the main ballast system, are accounted for as loads under load to submerge (see Table 2).

d. Dynamic control systems—Only thrust-producing systems used exclusively for maneuvering are included in this subgroup. If used for both propulsion and maneuvering, they are considered to be a part of the propulsion plant.
e. Mission system—This special-purpose system is a part of the payload and may be located internal and external to the pressure hull. Weight/displacement of external items may be fixed or vary during a dive (variable items are included in variable load). Performance requirements should convey sufficient information to permit estimating the system's sizing data with relative ease.

6. 9A Outfit and Furnishings: This system's weight and displacement are minute percentages of the Condition A1 weight and displacement. In both instances, 1 percent may be used for first estimate purposes. Space requirements are another matter for items inside the pressure hull, ladders, lockers, hull insulation, seats, and so forth. Even first estimates of these sizing data should be made with great care because they have considerable influence on pressure hull volume and, hence, have a cascading effect on other design considerations.

10A Energy Summary—Energy Storage Systems: At this point, the power-energy requirements for the SWBS systems should be summarized and the sizing data for the electrical- and pneumatic-energy storage systems estimated, the two forms of energy carried on board almost all untethered, manned submersibles. The summarization procedure for electrical-energy storage systems, assumed to be secondary batteries, involves use of a *power-energy profile*. Such a profile is shown in Fig. 10 for a hypothetical, seafloor-oriented mission having a

	Dive Profile								
		Endurance (Normal)							
			Bottom Time						
Power User	Pre-Dive Check 1	Descent 2	Search 3	Work 4	Relocate 5	Work 6	Ascent 7	Retrieval 8	Emergency
Propulsion	+		+		+			+	
Maneuvering	+	+	+	+	+	+	+	+	
Hotel Load	+	+	+	+	+	+	+	+	
Mission Systems	+			+		+			
Emergency Systems									+
Power (kW)	4.5	3.5	6.25	5.00	6.0	5.5	3.5	3.0	3.0
Time (Hours)	0.5	1.0	1.0	2.75	0.75	3.0	1.0	0.5	3.0
Energy (kWH)	2.25	3.50	6.25	13.75	4.50	16.50	3.50	1.50	9.0

Energy Requirement = 51.75 kWH; Energy Available (1.35 × 51.75) = 70 kWH

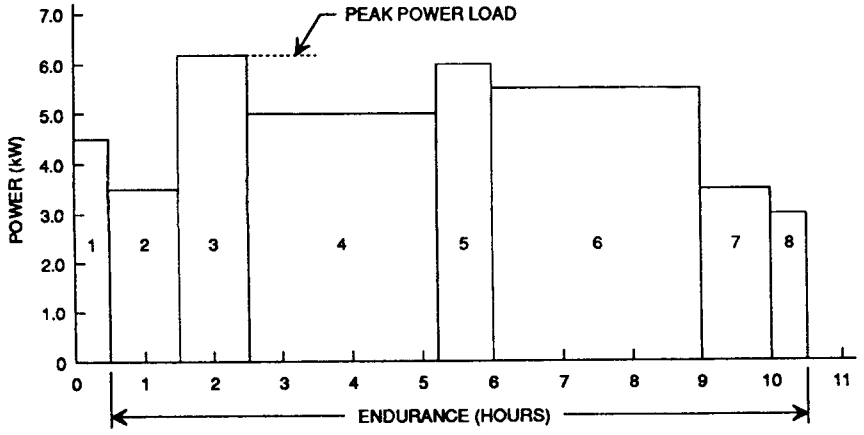


Fig. 10 Power energy profile

normal endurance of ten hours. The following comments refer to this figure:

1. Dive profile: This profile is that part of the overall mission profile which details the submersible's operations on a chronological basis; the mission profile also includes other events, such as shore-to-site in-transit operations of the support ship, which have no bearing on underwater operations of the submersible. The dive profile can be subdivided as indicated in Fig. 10. Note that launching requires a very short period of time, and, hence, is not included in this profile for power-energy summarizing purposes. On the other hand, note that retrieval, even under normal conditions, requires a considerable amount of time during which energy must be expended. Bottom-time events vary greatly depending on the nature of the mission tasks specified in the mission requirements (those shown in Fig. 10 have been chosen arbitrarily).

2. Bottom-time work: This is the only productive time in the entire mission profile and calls attention to cost-effectiveness considerations. In this regard, it is apparent that use of different SWBS systems, forming different conceptual design alternatives, will alter the length of this period and the amount of useful work accomplished during it. As suggested in Subsection 5.2 and by equation (2), the optimization objective is to find the best balance between system capability and cost to maximize cost-effectiveness of the submersible.

3. Endurance (normal): Endurance, as the term is used herein, includes not only the submerged endurance but also retrieval time spent on the surface, a period of continued energy use and taxing of human physiological and psychological characteristics.

4. Power users: Power users are shown in the left column of Fig. 10 as user categories rather than as individual SWBS systems requiring electrical power. Examples of components within these categories are given below.

   a. Propulsion: propulsion thruster motors
   b. Maneuvering: maneuvering thruster motors, control surface activators, and trim-list system pumps
   c. Hotel load: normal life-support systems, navigation and communication equipment, and lights
   d. Mission systems: manipulator systems, movie/still cameras and lights and inspection/testing equipment
   e. Emergency systems: emergency life-support, emergency communications, and jettisoning systems

5. Power requirements: These requirements are derived from manufacturer's specifications for SWBS systems chosen for a particular conceptual design alternative.

6. Power totals and time: The data given in Fig. 10 are arbitrarily chosen for this illustration.

7. Reserve energy: Reserve energy is stored to meet unexpected situations which may be encountered. Usually, a minimum of 25 percent of the total energy requirements is added to the energy-storage system for this purpose.

8. Emergency energy storage: A separate source of electrical energy may be carried inside the pressure hull to operate emergency systems in case of loss of power from the main source.

9. Energy storage capacity: The capacity of the main electrical-energy storage system equals the total energy requirement, the area of the power-energy profile diagram of Fig. 10, plus reserve energy.

Estimating sizing data for the electrical energy storage system involves knowing the energy storage capacity, selecting the type of battery to be used, and selecting the type of protection. Table 3 lists three types of batteries that are used in submersibles and the associated energy density data necessary for estimating sizing data. Regarding protection, batteries may be located either inside or outside the pressure hull, although the outside location is selected for most submersibles. In this case, the batteries must be protected from seawater in pressure-resisting or pressure-compensated containers. Weight estimates must include weights of container and compensating fluid systems, if used. Displacement and space estimates are based on container dimensions.

Summarizing pneumatic energy storage requirements often involves these requirements for both high and low pressure systems; high pressure systems are used in designs with maximum operating depths down to about 610 m (2000 ft). In both systems,

$$v_1 = \left(\frac{p_2 v_2}{p_1}\right) R \tag{7}$$

where

v_1 = total volume of air stored at a working pressure of p_1 in n high-pressure air flasks having a volume of v_F per flask; n flasks often are called an air bank,

p_1 = working pressure; representative values for high-pressure systems are 16.54×10^6 Pa (2400 psig) and 20.68×10^6 Pa (3000 psig)

Table 3 Battery energy densities

Battery	Theoretical	Actual	
	WH/lb	WH/lb	WH/in^3
Lead-Acid	80–115	10–20	0.8–1.2
Nickel-Cadmium	105	12–19	0.7–1.25
Silver-Zinc	205	40–60	2.4–3.8

and for low-pressure systems 9.8×10^6 Pa (1422 psig),

p_2 = pressure at maximum ballast-blowing depth for high-pressure systems and pressure at or near the surface for low-pressure systems,

v_2 = total volume of seawater ballast which can be expelled by a fully charged air bank against a pressure of p_2, and

R = a factor allowing for a reserve of storage volume and derived from similar design data.

Summarizing the pneumatic energy storage for the high pressure system to obtain v_1 necessitates a breakdown of v_2. Since these systems are often used to discharge variable as well as main ballast, the following expression may be used:

$$v_2 = N_C v_{VBC} + K v_{MB} \qquad (7a)$$

where

N_C = number of cycles of variable ballast per dive,

v_{VBC} = average volume of variable ballast expelled per cycle against a pressure of p_2,

v_{MB} = total volume of main ballast seawater, and

K = fraction of total volume of main ballast expelled against a pressure of p_2.

For high-pressure systems, the value of K depends on design decisions regarding initiation of the normal and emergency ascents from the maximum operating depth. Normal ascent may be initiated by dropping ascent weights or by injecting "bubbles" of high-pressure air into the main ballast tanks—the bubbles, of course, expanding with decreasing depth. If the bubble scheme is used, with or without dropping ascent weights, K may be 0.20 or less. Emergency ascent may be initiated by dropping large weights, such as the battery pack, or by blowing all of main ballast, in which case K is 1.00. The low-pressure system is used to complete blowing of the main ballast at or near the surface after a normal ascent, where v_2 of equation (7) is the volume of water remaining in the tanks. Note that for deep-depth submersibles, the use of compressed air for expelling ballast is not feasible and both normal and emergency ascents are initiated by dropping ascent or large weights. For these vehicles, only a low pressure air system is required and v_2 of equation (7) is the total volume of the main ballast tanks.

Sizing data for the high and low pressure pneumatic-energy storage systems may be obtained with estimates of v_1 for these systems in hand. In both systems, the weight of compressed air, a part of variable load, is v_1 times its specific weight, which is a function of p_1 and temperature. Sizing data for the systems' flasks may be found from manufacturer's data knowing v_F. Data for remaining components of the systems may be estimated from similar design data.

11A Weight-Displacement-Center Summary: At this point, summarize weights and displacements of SWBS systems and their loads and achieve the submerged design condition criterion of *neutral buoyancy*, in which total weight and displacement are equal. Note that the other two design condition criteria of zero trim and adequate positive stability are addressed at Point 11B when estimates of longitudinal and vertical centers of weight and displacement are available.

Figure 11 illustrates the summarizing procedure and methods of achieving neutral buoyancy for both *volume-limited* and *weight-limited* designs. As shown in the figure on the SUM lines, weight is less than displacement for volume-limited designs and greater than displacement for weight-limited designs. For volume-limited designs, fixed ballast is added in an amount necessary to establish equality between weight and displacement, its displacement being zero if located inside the pressure hull. In the case of weight-limited designs, there are three options: increase pressure-hull volume beyond the volume needed to just house personnel and equipment

Item	Weight	Displ.
1. Hull Structure		
2. Propulsion Plant		
3. Electrical Plant		
4. Command and Surveillance		
5. Auxiliary Systems		
6. Outfit and Furnishings		
CONDITION A1		
Normal Diving Trim Load[1]		
Main Ballast Water		
SUM	$W \neq \Delta$	

SUM: $W < \Delta$; VOLUME LIMITED		
Fixed Ballast		
SUB.—DESIGN CONDITION	$W = \Delta$	

SUM[2]: $W > \Delta$; WEIGHT LIMITED		
(1) Increase pressure hull volume		
(2) Add fixed buoyancy		
(3) Place items outside press. hull		
SUB.—DESIGN CONDITION	$W = \Delta$	

[1]Loads which may be on board in surface diving trim condition: variable load, variable ballast/buoyancy, descent/ascent ballast, and residual water.

[2]Use option (1) or (2). Option (3) may be used with (1) or (2).

Fig. 11 Weight–displacement summary

located inside; add fixed buoyancy in the form of solid material, hollow buoyancy spheres, or both; or locate items in the "inside/outside" pressure-hull category outside the hull so that their weights are at least partially compensated for by their displacements. The choice between the first two options depends on the relative values of the pressure hull and fixed buoyancy weight/displacement ratios, $(W/\Delta)_{PH}$ and $(W/\Delta)_{FBY}$. The first option is exercised if $(W/\Delta)_{PH}$ is less than $(W/\Delta)_{FBY}$ and the second option is exercised if the reverse if true.

12A Cost Estimate Summary: Summary of cost estimates is deferred until Point 12B.

The B loop—B loop procedures begin to develop the submersible and its SWBS systems in detail for the purpose of refining sizing data, obtaining cost estimates, and obtaining assurance that design condition criteria of neutral buoyancy, zero time, and adequate stability can be attained. Brief comments on B loop points follow.

1B Performance Requirements: Review performance requirements in the light of sizing data generated in the "A" loop. The requirements should remain fixed unless these data indicate that certain requirements are excessive and lead to unfeasible designs. In this case, the designer should consult with the owner/user concerning altering mission requirements from which the questionable performance requirements are derived.

2B–3B Arrangement and Geometry-Displacement: Obtain the first definitive view of the submersible and locate the major components of the SWBS systems. These procedures are facilitated by considering the two points together. Input includes estimates of system space or cubic data, generated at Points 4A through 10A, and volumes of fixed ballast/buoyancy and pressure hull(s) derived from the weight-displacement summary at Point 11A. Relatively close estimates of envelope volume and principal dimensions can now be made with these data in hand. An estimate of displacement is also now available from Point 11A.

Location of major components of SWBS systems requires drawing *inboard* and *outboard profiles* of the submersible as well as *cross-section views* at critical points along its length. The inboard profile and cross sections show the pressure hull, tankage, batteries, fixed ballast/buoyancy, and other major components of the SWBS systems in their relative locations. The outboard profile presents the overall appearance of the submersible and locates appendages not clearly shown in the other views. The profiles and sections are needed to check for space requirement conflicts and to obtain vertical and longitudinal centers of gravity and displacement.

4B–9B The SWBS System Groups: Develop systems in sufficient detail to permit refinement of sizing data and close estimates of cost data to be made. More precise formulations and direct calculations are used in this portion of the B loop. For example, A loop procedures might estimate the weight of the pressure hull from "weight/displacement ratio versus design depth" curves knowing the design depth from performance requirements, selecting a pressure-hull material and estimating its enclosed volume. In the B loop, a closer estimate of its weight may be found using pressure-hull strength equations given in Chapter IV and making allowances for the weights of insert plates compensating for large penetrations. In general, B loop procedures begin to analyze each system, breaking it down into its parts for study, a process which increases in detail as the design progresses through the inner loops of the spiral.

10B Energy Summary-Energy Storage Systems: Refine electrical power-energy profile and pneumatic storage system based on more precise B loop SWBS data.

11B Weight-Displacement-Center Summary: Complete this summary using the more refined sizing data and obtain (1) a more accurate estimate of fixed ballast or buoyancy, or both, required to achieve neutral buoyancy, (2) a check on the vertical distribution of weights and displacements necessary for adequate stability, and (3) a check on the longitudinal distribution of weights and displacements required for zero trim. At this point, the summary form shown in Fig. 11 is abandoned for the form shown in Fig. 12, which facilitates the computations of the submersible's vertical and longitudinal centers of weight and displacement. Background information concerning the details of this figure is given in Subsection 3.5 of Chapter V. Procedures (2) and (3) usually require returning to Point 2B to adjust the arrangement, and centers of the systems' major components including fixed ballast/buoyancy. It may be necessary to alter the amounts of fixed ballast or buoyancy, or both, as well as their positions, to achieve adequate stability and zero trim as well as neutral buoyancy in the revised weight-displacement summary.

12B Cost Estimate Summary: Summarize estimates of first and annual operating costs associated with the SWBS systems selected for a particular design alternative. This summary permits an estimate to be made of the submersible's operating cost per year, in which the first cost is amortized over a given number of years. This estimate, when divided by the number of dive-days per year, is the second term of equation (2).

Title	Weight					Displacement				
	WT	KG	VMW	LCG	LMW	DISP	KB	VMΔ	LCB	LMΔ
100—Hull Structure (excluding FBB)										
200—Propulsion Plant										
300—Electrical Plant										
400—Command and Surveillance										
500—Auxiliary Systems										
600—Outfit and Furnishings										
700—Armament (military subs. only)										
CONDITION A1										
Fixed Ballast/Buoyancy to Balance										
CONDITION A										
Normal Diving Trim Load										
CONDITION N										
Main Ballast										
SUBMERGED—DESIGN CONDITION	W	KG	ΣVMW	LCG	ΣLMW	Δ	KB	ΣVMΔ	LCB	ΣLMΔ

equal

$BG = KB - KG = \Sigma VM\Delta/\Delta - \Sigma VMW/\Delta$

equal

CONDITION A1										
Normal Diving Trim Load										
Main Ballast										
SUB. CONDITION (excluding FBB)	W′		ΣVMW′		ΣLMW′	Δ′		ΣVMΔ′		ΣLMΔ′
Fixed Ballast/Buoyancy to Balance	W_{FBB}		VMW_{FBB}		LMW_{FBB}	Δ_{FBB}		$VM\Delta_{FBB}$		$LM\Delta_{FBB}$
SUBMERGED—DESIGN CONDITION	W		ΣVMW		ΣLMW	Δ		ΣVMΔ		ΣLMΔ

Fig. 12 Weight-displacement-centers summary

The select point—"Deposit" the conceptual design alternative just completed at the *select point* and return to Point 1A via path (a), shown in Fig. 7, if more design alternatives are to be generated, or select the most cost-effective, viable design from among the number of design alternatives already generated and deposited at this point. Assuming the second option is followed, the procedure suggested in Subsection 5.2 may be followed. Equation (2) is solved for each of the alternatives; the one with the lowest $(1/CE)$ is selected. With the second term of this equation being determined at Point 12B for each alternative, consider the first term, dive days per mission task. This term can be expressed as

$$\frac{\text{dive days}}{\text{mission task}} = \frac{\text{hours}}{\text{mission task}} \Big/ \frac{\text{hours on task site}}{\text{dive day}} \qquad (8)$$

where, on the right side of the equation, the numerator is the *time required to accomplish the mission task* once the submersible is at the task site and the denominator is the *productive work time per dive day.* The numerator and denominator should be made as small and large as possible, respectively, bearing in mind that such procedures will most likely also increase system costs. The numerator's solution involves the mission systems and others, such as the maneuvering system, which are in use at the work site. Estimating this term requires performance data reflecting the expeditiousness of these systems in completing the work to be done. Examples of these data are the number of degrees of freedom a manipulator possesses and the rates at which it can execute various motions. The denominator's solution involves systems in use during the necessary but nonproductive activities of the dive profile shown in Fig. 10, including, for example, the descent/ascent, propulsion, maneuvering, navigation, and communication systems. For a given dive profile time, which nearly equals the endurance, estimating this term also requires performance data reflecting the expeditiousness of these systems in reducing the nonproductive, dive profile time. Descent/ascent rates, achievable with various types of descent/ascent sys-

tems for a given envelope geometry, are examples of the kind of data required. With the select point process completed, the selected conceptual design moves along path (b), shown in Fig. 7, into preliminary design.

The C, D, and E loops—The preliminary design procedures of these loops progressively refine all aspects of the submersible's design, including those aspects not considered in the conceptual design phase. Only small adjustments, if any, are required at the spiral's point at the end of the E loop. Some general comments on these points follow.

1 C, D, E Performance Requirements: Review these requirements in light of the design's more detailed development. C and D loop procedures may reveal that it is still necessary or desirable to make small changes in certain of these requirements. There should be no changes in them as the result of E loop procedures or, in other words, they should be fixed at entry into the contract design phase.

2 and 3 C, D, E Arrangement and Geometry-Displacement: Obtain a detailed view of the submersible with the location of SWBS systems being essentially fixed so that the systems can function properly, routine inspection and maintenance can be made with relative ease, there are no conflicts in space requirements, and the design condition criteria of adequate stability and zero trim are met. These procedures require detailed inboard/outboard profiles and cross-section views to be drawn. A *lines drawing* may also be drawn depicting the envelope's geometry; this drawing is necessary to obtain accurate data required for hydrostatic and hydrodynamic calculations.

4–9 C, D, E The SWBS System Groups: Develop these systems in sufficient detail to provide the assurance that, both individually and collectively, they will satisfy all performance requirements and very close estimates of sizing and cost data. Direct calculations and preliminary drawings, as contrasted with working drawings, are used extensively (the basis for these calculations is contained in Chapters V and VI of this publication and in references given). Procedures, for example, may involve the construction and testing of structural and appended envelope models to verify data derived from theoretical strength and hydrodynamic formulations.

10 C,D,E Energy Summary-Energy Storage Systems: Make progressive refinements of electrical power-energy profile and pneumatic storage system based on increasingly precise C, D, E loop SWBS data. Electrical and pneumatic storage systems should be very accurately defined at point 10E.

11 C, D, E Weight-Displacement-Center Summary: Complete this summary in more detail and investigate stability and trim in the design and other operating conditions as well as in the emergency ascent condition. These procedures will require a breakdown of SWBS systems into their components as described in Subsection 3.5 of Chapter V. Procedures at Point 11C and, to a lesser extent, at Point 11D usually indicate that some adjustments in arrangements or system details or both are necessary to attain design condition criteria of neutral buoyancy, adequate stability, and zero trim. They may also indicate that adjustments in envelope geometry and systems influenced by this characteristic are necessary to obtain adequate surface stability. These adjustments, if any are to be made, should be relatively small on entry into the contract design phase.

12 C, D, E Cost Estimate Summary: Summarize the progressively refined first and annual operating cost estimates obtained in these loops. A close estimate of the submersible's operating cost per year should be in hand at the beginning of contract design.

5.4 Systematic Parametric Analysis Approach—Optimization Programs

These programs are associated with the systematic parametric analysis approach to design discussed in Subsection 5.2. Their use in conjunction with computers permits a very large number of conceptual design alternatives to be investigated, thus greatly increasing the probability that the optimum conceptual design will be identified.

The development of these programs, as noted, still requires information used in the empirical approach to design. In this case, however, the behavior and interplay between the SWBS systems and their components and their influences on cost-effectiveness must be made explicit. Design procedures are formalized by establishing mathematical relationships between dependent design variables and independent design variables or parameters. These relationships, or parametric equations in the program, permit rapid evaluation of design parameter changes on the dependent variables. In making systematic parametric analyses, both SWBS system characteristics and performance requirements may be considered as parameters.

The explicitness of optimization programs is useful in facilitating discussions between the potential owner/user of a submersible and its designer regarding any aspect of the mission requirements or the design in question. For example, a program can quickly and clearly demonstrate that a certain mission requirement or owner/user bias regarding use of specific equipment excessively constrains the design. The details of design optimization are best

presented by an example. This example, presented in the Appendix of this chapter, was prepared in 1967 for the development of a deep-submergence search vehicle. While the values of coefficients in the program's equations are unique to specific mission requirements and technological state-of-the-art in 1967, the example does show the basic format of such a program and how it may be used to aid the submersible designer.

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Appendix[3]

Design Optimization Program

The initial step in designing an optimization program is to define the system to be optimized. In this case, the system consisted of a search vehicle, whose mission was to locate objects on the deep sea floor, and its operations at the search site, which included preparation for diving, descent, search, ascent, recovery, and preparations for the next dive.

It is necessary to vary design concepts in some organized manner in order to select the best alternative. This is done by varying a limited set of design variables that affect the performance, economics and survival of the vehicle. For optimization of this system, the variables chosen were

—Maximum design speed
—Endurance
—Number of crew
—Operating depth
—Safety factor

The four optimization models of this program, described in more detail below, are summarized as follows:

- Parametric design model: For specific values of such design variables as maximum speed, endurance time, number of crew, operating depth, and safety factor, this model calculates weights and volumes in functional groups such as pressure hull, batteries, exostructure, and total stored energy.
- Operational model: This model accounts for the time and energy used to prepare for dives and to descend and ascend when provided with values for the operational variables.
- Effectiveness model: When supplied with net time and energy remaining after exercising the operational model, this model simulates a bottom search for a given object and computes the probability of detecting that object.
- Cost model: After the design and operational alternatives have been ascertained, this model provides a relative measure of the acquisition and operating costs for the system.

There can be no guarantee that the alternative selected by such a program will be truly the optimum. Models of this type depend heavily on statistical data gathered from past design experience, and can therefore produce results no better than those data. The solutions are nevertheless a good starting point for detailed design, since they represent a technically feasible and viable baseline.

Parametric Design Model

The design analysis program accepts as input data the weights and volumes of specific equipment groups and, using those data, together with physical and empirical scaling laws, performs many of the

[3]The author of the Appendix is R. H. Hobert.

routine design calculations for weight and volume balances within the pressure hull and hydrodynamic hull. It also automatically adjusts the amount of syntactic foam or other buoyancy needed to make the vehicle and its subsystems neutrally buoyant.

The primary input data include fixed values for operational features such as speed, depth, endurance, auxiliary power requirements, and payload characteristics. Thus, it is possible to start from a reference design and then conduct systematic excursions from that design in order to assess the changes in vehicle characteristics resulting from changes in the operational features. A wide variety of operational characteristics may be evaluated and adjustments made in the reference design so that systems of identical operational capability can be compared directly.

Many different potential materials for the pressure hull, differing types of flotation material, types of batteries or other energy storage devices, and unusual configuration requirements substantially increase the number of possible solutions for any given set of design specifications. Components such as the propulsion motor can be held at some fixed power level, while allowing other components within the overall system to vary. Or, the propulsion motor can be varied in steps corresponding with commercially available equipment, or parametric extrapolations of system weight and volume can be made to assess the desirability of developing items especially tailored to specific needs.

The parametric equations are described below. The purpose of these equations is to provide a means for ascertaining the effect of a change in performance requirements on the overall design. To this end, the vehicle is divided into various systems. Each system has a group of performance variables that determine its weight in a reasonably simple way, once its general design has been chosen. This allows weight and volume equations to be written and solved for the new design. The expressions below are typical for a small research submersible, with a spherical pressure hull and batteries outside the pressure hull.

Weight Equations

Pressure hull:

$$W_{PH} = V_I(C_1 + C_2 D S_F)$$

where

V_I = volume inside the pressure hull,
D = operating depth, and
S_F = safety factor.

All C values are constants for a particular design "family." This equation results from the fact that the thickness of a metal sphere is proportional to the outer radius times the pressure. The constant C_1 accounts for the thickness required to support the structure even at zero depth.

Exostructure:

$$W_{EX} = C_3(\Delta - W_I - W_{PH}) + C_4(W_I + W_{PH})$$

where Δ is the envelope displacement and W_I is the weight inside the pressure hull. For this equation, it is assumed that the exostructure weight is proportional to the weight of the structure it is attached to, and that this proportionality is different for the living space and the rest of the vehicle. This, as well as other assumptions in this set of equations, is based on statistical studies of large submarines and on data that have been assembled so far on small research submarines.

Fairings:

$$W_F = C_5 \Delta^{2/3} + C_6$$

Here the weight is proportional to the area covered.

Control surfaces:

$$W_{CS} = C_{25}\Delta$$

This includes the actuators as well as the planes and rudder.

Compressed air system:

$$W_{CA} = (C_9 + C_{10}D)\Delta$$

As in the ballast system, this weight is assumed to be a fixed percentage of displacement, but a term to account for the pressure-dependent character of this system has been added.

Ballast system:

$$W_B = C_{21}\Delta$$

This includes trimming weights, hovering weights, fixed droppable weights, etc.

Main propulsion:

$$W_{MP} = C_{11} + C_{12}K_8\Delta^{2/3}V_M^2$$

where V_M is maximum speed. K_8 is a constant accounting for drag, propeller efficiency, and power plant efficiency. Main propulsion weight includes everything from the battery to the propellers.

Thrusters:

$$W_T = C_{13}\Delta^{2/3}$$

This assumes constant maximum turning rates.

Batteries:

$$W_{BB} = C_{22}(1 + C_{23}D)Q$$

where Q is total stored energy. Because this submersible's batteries are outside the pressure hull, they are pressure compensated and, therefore, depth-dependent.

Electrical power conditioning:

$$W_{EPC} = C_{24}L_M$$

where L_M is maximum nonpropulsive power demand. This includes cables, breakers, fuses, inverters, etc. and their housings.

Control, navigation, communication systems (internal):

$$W_{CI} = C_{14} + C_{15}D + C_{16}\Delta$$

This is equipment needed to operate the submersible, but not to perform its mission (search, etc.); the latter is internal payload.

Metabolic processing:

$$W_{MET} = C_{17}NT + C'_{18}$$

where N is the number in crew and T is habitation time limit. This includes oxygen and carbon dioxide, food, and waste-disposal systems.

Climate control:

$$W_{CLC} = C''_{18} + C'_{19}N + C_{20}(W_{PI} + W_{CI})$$

where W_{PI} is internal payload. Climate control includes humidity and temperature regulating equipment. The equation states that the equipment weight for heat rejection is proportional to the system weight producing the heat.

Furniture, crew, and effects:

$$W_{FCE} = C''_{19}N$$

Living system:

$$W_L = W_{MET} + W_{CLC} + W_{FCE}$$

$$= N(C_{17}T + C_{19}) + C_{18} + C_{20}(W_{PI} + W_{CI})$$

$$C_{18} = C'_{18} + C''_{18}$$

$$C_{19} = C'_{19} + C''_{19}$$

Internal payload is W_{PI}; external payload is W_{PE}. The payload weights are considered performance requirements. Flotation material is W_S and entrained water is W_W; these two variables are solved for, using the above equations.

Volume Equations

Most systems are assumed to have constant density, so that the volumes are simply weight divided by density. There are two exceptions: living system and battery. For mathematical convenience, the following volume scaling coefficients are defined for the living system:

$$K_2 = \frac{C_{17}}{\rho_{MET}}$$

$$K_3 = \frac{C'_{18}}{\rho_{MET}}$$

$$K_4 = \frac{C''_{18}}{\rho_{CLC}}$$

$$K_5 = \frac{C'_{19}}{\rho_{CLC}} + \frac{C''_{19}}{\rho_{FCE}}$$

$$K_6 = \frac{C_{20}}{\rho_{CLC}}$$

Then, the volume of the living system is

$$V_L = N(K_2T + K_5) + K_3 + K_4 + K_6(W_{PI} + W_{CI})$$

Because the battery system is pressure compensated, its volume is defined as

$$V_{BB} = K_9(1 + K_{10}D)Q$$

Auxiliary Equations

In addition to weight and volume equations, it has been found useful to define the following:

Stored energy:

$$Q = C_7E_4 + C_8K_8\Delta^{2/3}T_EV_E^3$$

where

E_4 = total auxiliary energy,
T_E = propulsive endurance time, and
V_E = endurance speed.

Each vehicle may be operated in many ways, each with its own load schedule. Therefore, it is felt more useful to specify the available energy then to scale according to a given load profile.

Envelope-to-interior volume ratio:

$$K_7 = \frac{\Delta}{\rho_W V_I}$$

where ρ_W is water density.

Operational Model

Once a vehicle has been designed, it is necessary to ascertain the best way to operate it. How fast should it dive? How long should a mission be? At what speeds should it search? The object of these questions is effectiveness. For the example of a search vehicle, effectiveness is determined by how well and how quickly an area is covered. The vehicle must be operated in such a manner as to make the most effective use of the day. To solve this problem, it is broken down into several parts: surfaced and preparatory operations; diving and ascent; and search or bottom operations.

Surfaced and preparatory operations cover replenishment of the submersible and the acts of putting it into and removing it from the water. They are dominated by battery charging, which would vary with the extent of discharge. However, for preliminary calculations, those times may be assumed to be constant.

Diving time and energy, as well as surface time, set the time and energy limits for bottom operation. The best strategy for bottom operation is selected as part of the operational optimization program discussed later. The time constraints are defined as

T_I = launching time, h,
T_O = recovery time,

T_D = descent time,
T_A = ascent time,
T_T = topside servicing time,
T_C = time for changing crew,
N_1 = number of dives per day,
N_2 = number of crew changes per day, and
N_3 = number of topside servicings per day.

On the premise that the boat makes repeated dives of identical pattern, the bottom time, T_B, in hours per day, is

$$T_B = 24 - \left[N_1(T_I + T_D + T_A + T_O) + N_2 T_C + N_3 T_T\right]$$

Bottom time is made up of actual searching time, T_S, plus time consumed in submerged transiting between sites, T_R, so that

$$T_B = T_S + T_R$$

The energy terms of interest are

E_S = energy consumed per search,
E_I = energy consumed in launching,
E_D = energy consumed in descent,
E_A = energy consumed in ascent,
E_O = energy consumed in retrieval of boat, and
E_C = energy consumed in changing crew.

Then the total energy that must be supplied in a day is at least

$$N_1 E_S + N_1(E_I + E_D + E_A + E_O) + N_2 E_C$$

Since the energy supplied in topside servicing is limited by design, compromises must be made to reduce both the loads and duration of activities when not searching.

Power and speed terms must be introduced in order to calculate energy consumption:

P_{HP} = nonpropulsive power demand—searching,
P_{HT} = nonpropulsive power demand—transiting,
V_S = searching speed, and
V_T = transiting speed.

With the foregoing, the energy consumed in bottom operations may be calculated from

$$E_S = T_R\left(P_{HT} + K_8 \Delta^{2/3} V_T^3\right) + T_S\left(P_{HP} + K_8 \Delta^{2/3} V_S^3\right)$$

Effectiveness Model

In general, when searching for an object, one begins looking where there is the greatest likelihood of finding it. If that is unsuccessful, one should continue searching in a systematic manner in the zones of highest remaining probability of detection considering searching time and consumed battery power.

The first problem, therefore, is to select the zone of highest probability at any given time in the search. Usually, before the search begins, the search location is described by coordinates with some uncertainty. The uncertainty in the x and y directions may be different, but based on those uncertainties a probability versus range function or surface can be described about a known point. The surface would provide a value for the probability of the object existing at some range and direction. The surface should reflect such effects on the probability of location as known or possible errors in locating the splash point, a distribution to account for the migration caused by the object in its descent, and a modification based on the topography of the bottom.

A simplified example of how such a probability curve might be constructed is shown in Fig. 13 (only one plane shown). The resulting probability curve represents the total probability of the object existing for any given range. If we were to consider, in addition, a similar curve perpendicular to the plane of the paper, we could construct a probability surface; it is this probability surface that is of interest. The type of search pattern that should be executed for the given area is precisely defined by the contours obtained by slicing the probability surface with horizontal planes. For some reasonable distribution function perpendicular to the surface of the paper, the horizontal planes might produce contours such as shown in Fig. 14.

The lines represent contours of constant probability, the greatest being at the center and decreasing outward. The general shape of these contours is highly dependent on the bottom terrain. The ellipticity depends on the differences between the distributions in the x and y directions. In the case of

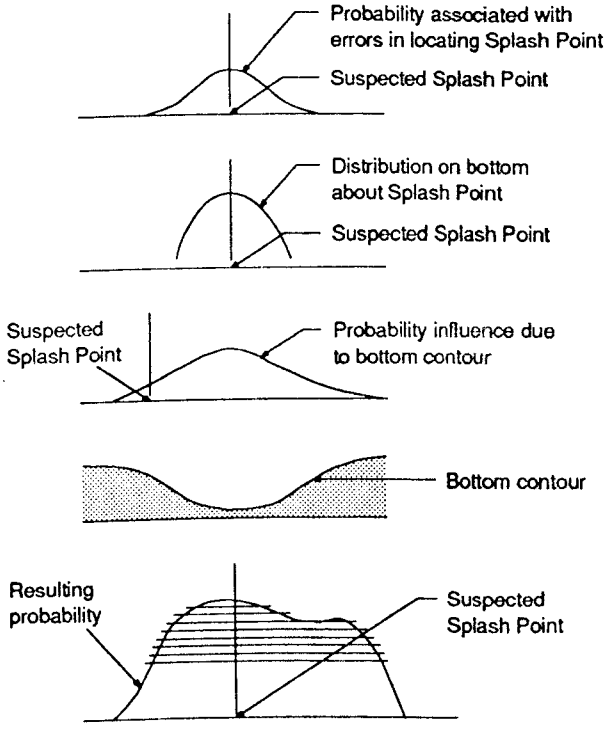


Fig. 13 Probability of finding an object (one plane)

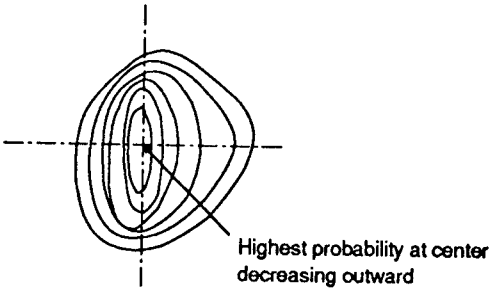


Fig. 14 Contours of constant probability for finding an object

near symmetry in the x and y directions, the search contours become circular. For lack of better information, the simple symmetrical case, leading to circular rings of constant probability, may be chosen for the effectiveness model.

Model Description

The program deploys the submersible in the ring of highest probability, and searches that area, consequently reducing the probability of target presence in that ring. The reduction modifies the height of the probability surface to correspond with the just completed search. The submersible then proceeds to the ring of next highest probability, probably at a speed different from the searching speed. Repeatedly, the submersible would be redeployed to the ring of highest remaining probability, until the total probability of the object remaining undetected falls below some level or the total bottom time exceeds some limit, or the total battery power expended at the bottom exceeds some limit. Figure 15 shows how the vessel might be deployed. In that figure, the vessel begins the search at location A and searches the circular area swept out. The submersible then transits radially to location B and searches that ring. The search proceeds to location C, and so forth.

In this example, searching at 2 knots was found to provide a greater probability of detection than speeds of 3 knots or faster. It was not ascertained whether or not 2 knots was the precisely optimal value, but it could be inferred that designing the submersible for speeds greater than 2 knots or so could not be justified.

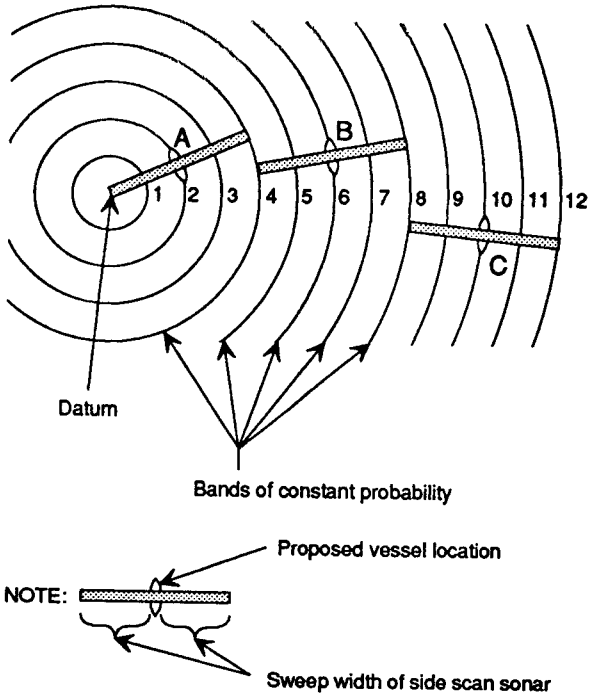


Fig. 15 Search deployment

Cost Model

Tradeoff decisions are reflected in life-cycle costs for the entire system. These include design development, procurement, maintenance, operation, and training costs.

Table 4 Vehicle cost equations

System	Cost
Pressure Hull	$D_{PH} = S_1 + S_2 W_{PH}$
Exostructure	$D_{EX} = S_3 + S_4 W_{EX}$
Fairings	$D_F = S_5 + S_6 W_F$
Compressed Air	$D_{CA} = S_7 + S_8 W_{CA}$
Main Propulsion	$D_{MP} = S_{11} + S_{12} W_{MP}$
Thrusters	$D_T = S_{13} + S_{14} W_T$
Control, Navigation, Communications (internal)	$D_{CI} = S_{15} + S_{16} W_{CI}$
Metabolic Processing	$D_{MET} = S_{17} + S_{18} W_{MET}$
Climate Control	$D_{CLC} = S_{19} + S_{20} W_{CLC}$
Furniture	$D_{FU} = S_{21} + S_{22}(W_{FCE} - 200N)$
Ballast	$D_B = S_{23} + S_{24} W_B$
Battery	$D_{BB} = S_{25} + S_{26} W_{BB}$
Electric Power Conditioning	$D_{EPC} = S_{27} + S_{28} W_{EPC}$
Control Surfaces	$D_{CE} = S_{29} + S_{30} W_{CE}$
Flotation	$D_S = S_{31} + S_{32} W_S$
Internal Payload	D_{PI}
External Payload	D_{PE}

Acquisition costs for specific subsystems, components, etc., should include all of the costs involved in producing them. The cost equations for each element are solved and the values are tabulated. The initial costs of elements not considered in the cost equations (for example, crew training and special development costs) are accumulated and tabulated with the results of the cost equation calculations. Cost factors such as mission profile, maintenance, and overhaul costs are used to establish the operational life-cycle cost estimate. The total life-cycle cost for each system will be the aggregate sum of all development, production, and operational costs.

Vehicle Cost Equations

The cost equations in Table 4 are those for manufacturing and installation. For a given set of design criteria, it may be assumed that the cost of a system is linear with its weight. The coefficients S therefore are descriptive of one design family for one set of numerical values. The total cost is then the sum of the items in Table 4.

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

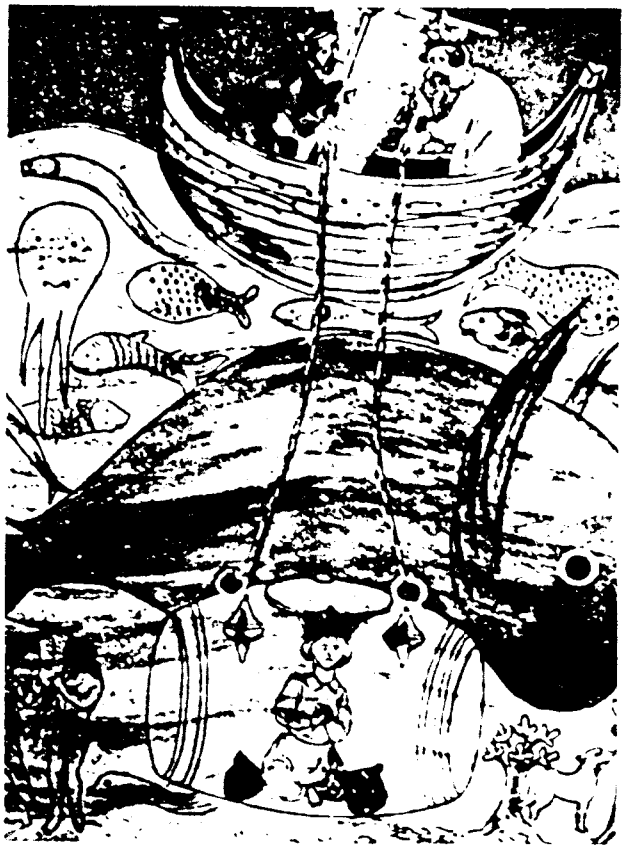


Fig. 1 Alexander the Great observing the wonders of the Aegean Sea from inside a glass diving bell (322 B.C.)

[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

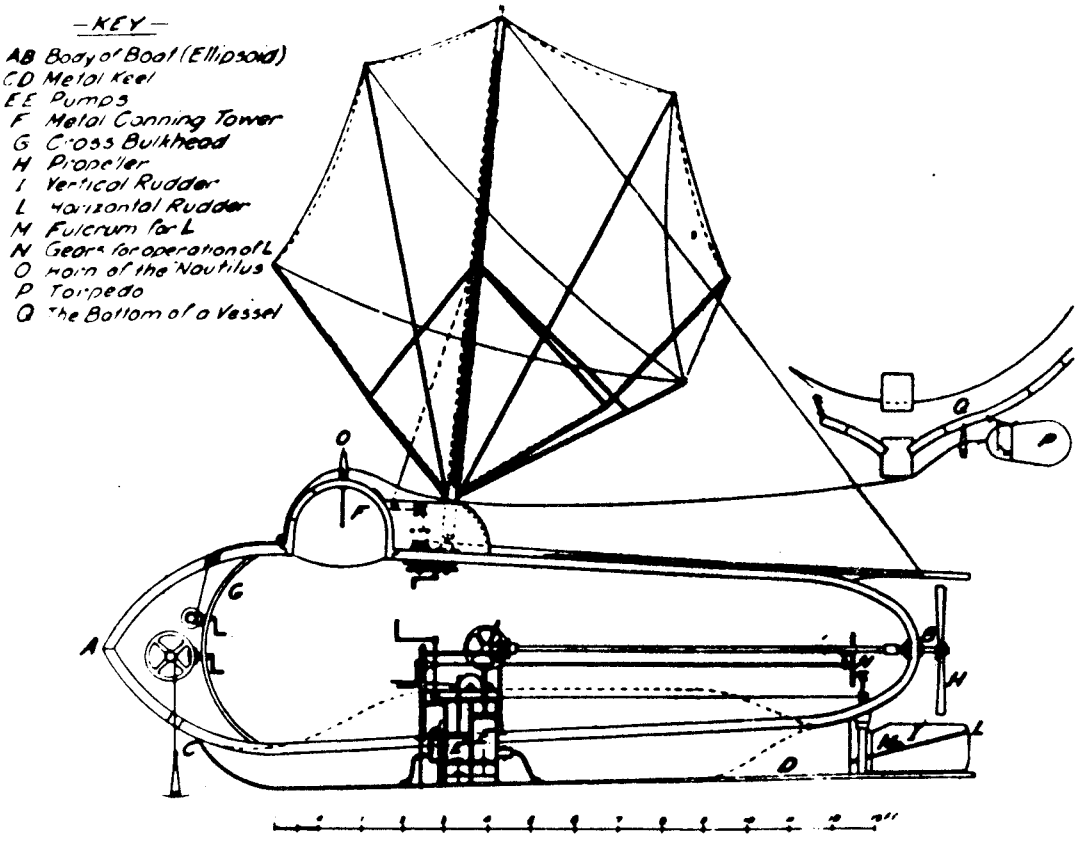


Fig. 3 The *Nautilus*, built by Robert Fulton, was demonstrated to Napoleon in 1800

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 *Abouktir*, *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

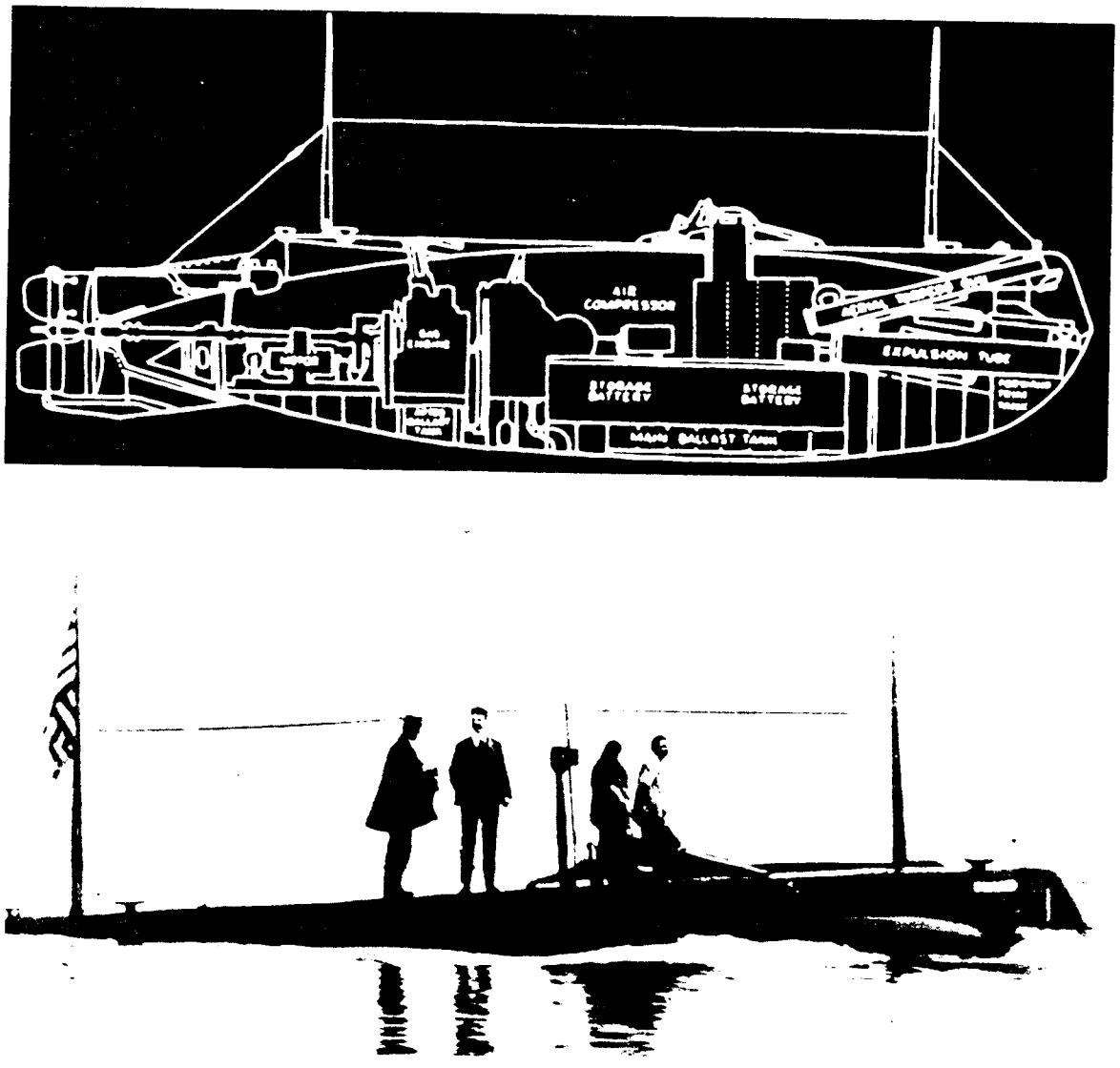


Fig. 4 *USS Holland* (SS1). The cross-sectional plan (*top*) shows that both gun and torpedo were housed within the hull. Conventional surface-ship design persisted in the form of masts for which a submarine had no use

"fleet boat"—a long, slender hull to reduce wavemaking 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 wavemaking 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.

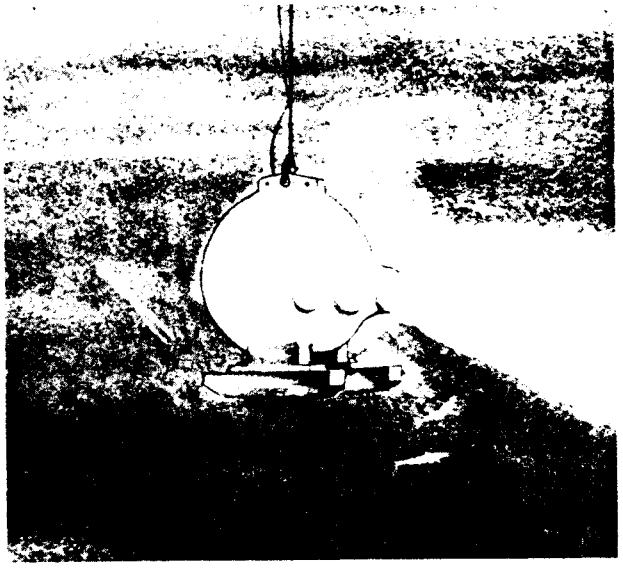


Fig. 5(*a*) Bathysphere (1930)

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 (¾ 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.

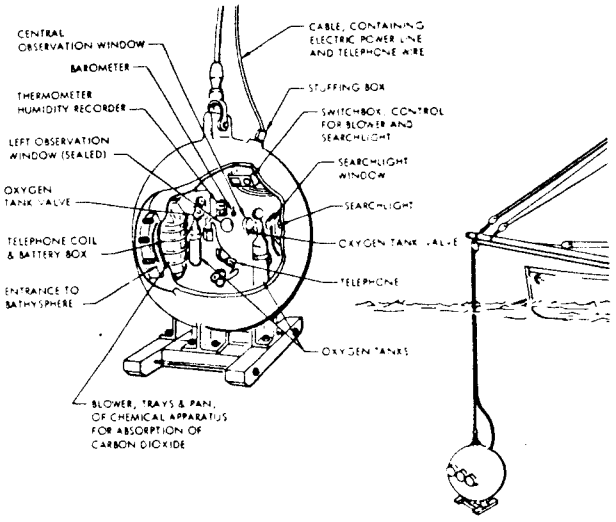


Fig. 5(*b*) Cutaway view of bathysphere used by William Beebe in his record dive of 1934, marking the beginning of modern submersible development (courtesy of National Geographic Society)

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).

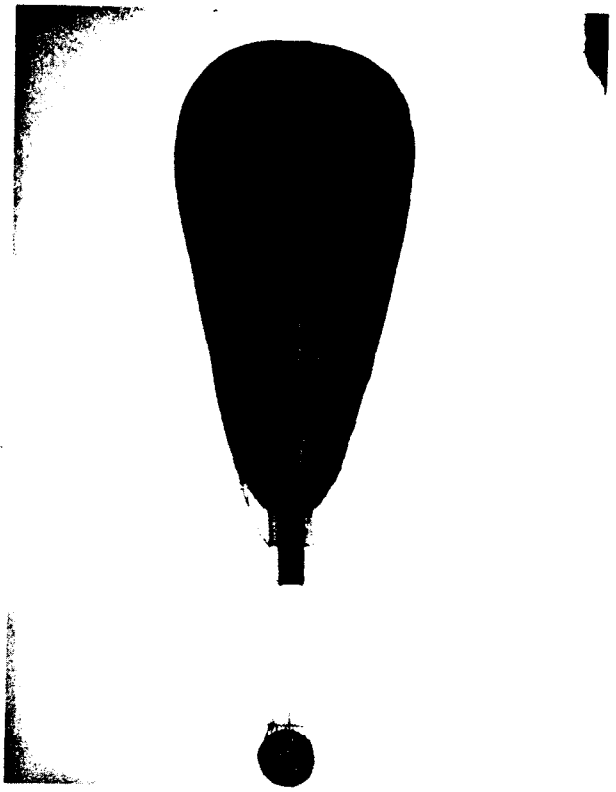


Fig. 6(*a*) Stratospheric balloon by Auguste Piccard

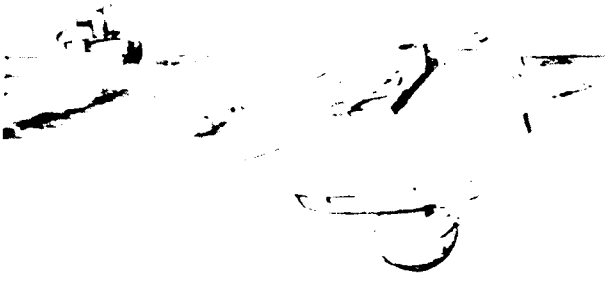


Fig. 6(*b*) Piccard's first bathyscaphe, *FNRS-2* (1948)

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.

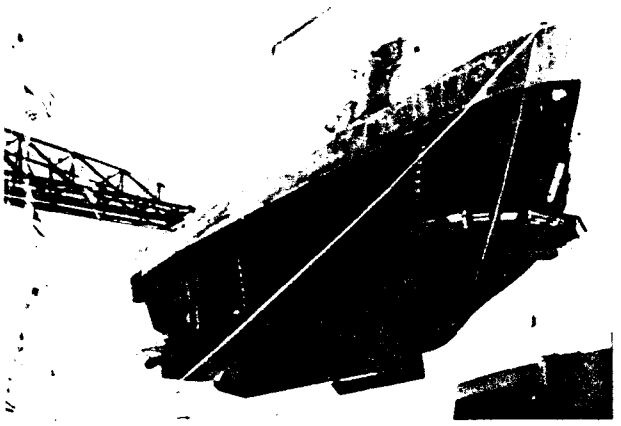


Fig. 8 The French bathyscaphe *Archimede*

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

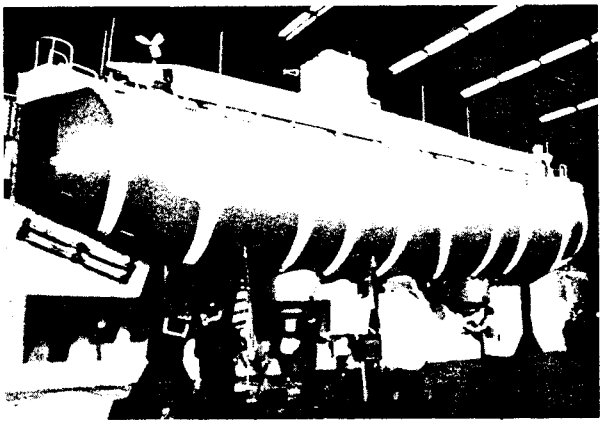


Fig. 7(*a*) The bathyscaphe *Trieste* (1960 configuration) on display at the Navy Yard Museum, Washington, D.C.

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

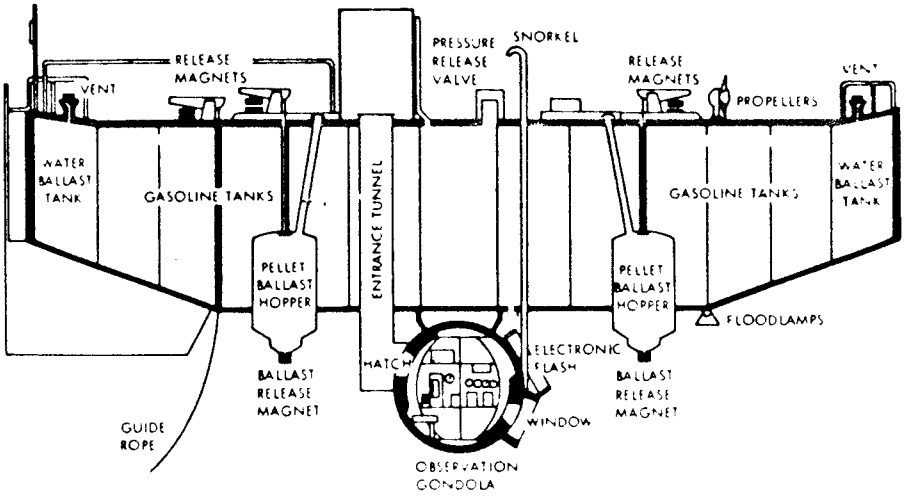


Fig. 7(*b*) Cutaway of the *Trieste*, which made the world's record dive to 10 915 m (35 800 ft) in the Mariana Trench off Guam on January 23, 1960 (from Terry, *The Deep Submersible*, 1966)

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.

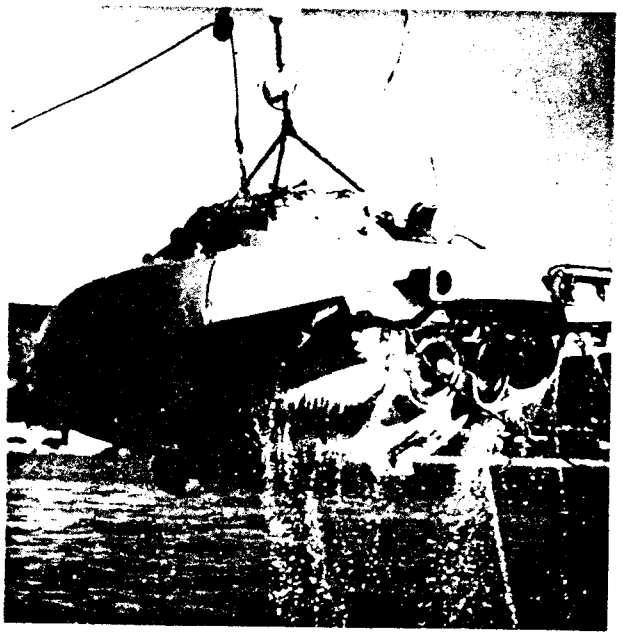


Fig. 9(*a*) Cousteau's diving saucer *Denice*

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"

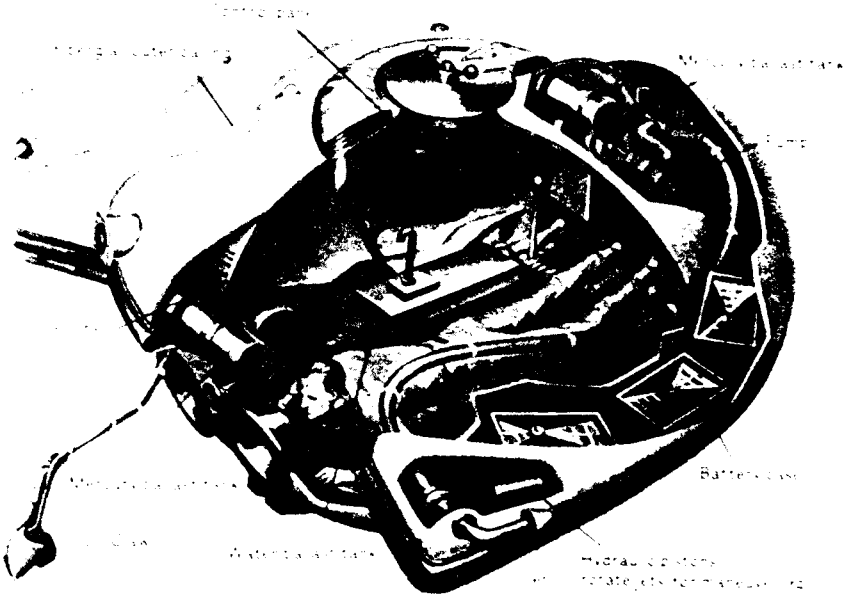


Fig. 9(*b*) Cutaway view of the *Denice*

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

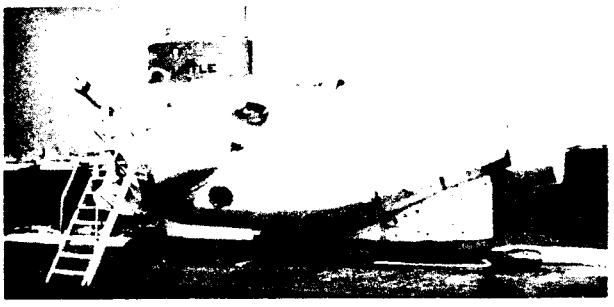


Fig. 11 The *Turtle/Seacliff* by Electric Boat Division, General Dynamics Corporation, Groton, Connecticut

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*).

The *Alvin* (Fig. 12), launched in 1963, had an HY100 pressure hull. Since it was the first time this material had been used for a manned personnel sphere for these depths (1829 m [6000 ft]), three hulls were built—one to test manufacturing techniques, one to test to destruction, and one to use for the *Alvin*. All three spheres were built with better than the expected tolerances for thickness, sphericity, and metallurgical flaw inclusion. When one was to be tested to destruction, it was the test tank that failed. Hence all three hulls were available: one for the *Alvin* and two for future use (*Sea Cliff* and *Turtle*).

Alvin is perhaps the most successful of any of the U.S. submersibles, having more than 2100 dives in her 25-year history. She was also the first manned submersible to be lost at sea. During a launch evolution in October 1968, off the Atlantic coast, in 1540 m (5052 ft) of water, a cable on the support ship's launch elevator broke and dropped *Alvin* into the water with its main hatch open. The crew escaped, but *Alvin* sank to the bottom. After about one year she was recovered through the use of another submersible (*Aluminaut*). *Alvin* was rebuilt and later a titanium pressure hull was added, giving her a depth capability of 4000 m (13 120 ft). During U.S. Navy overhauls, the *Turtle* was upgraded to 3048 m (10 000 ft), and *Sea Cliff* was extensively rebuilt with a titanium pressure hull for a 6097 m (20 000 ft) capability.

2.2 Manned and Unmanned Submersibles

The term *submersible* has been used to this point with few qualifying adjectives. However, continued discussion of second- and potential third-generation submersibles necessitates a categorization of these vehicles. Although there is no universally accepted terminology for this purpose, Table 1 outlines the categories on which the following discussion is based and lists examples for each category.

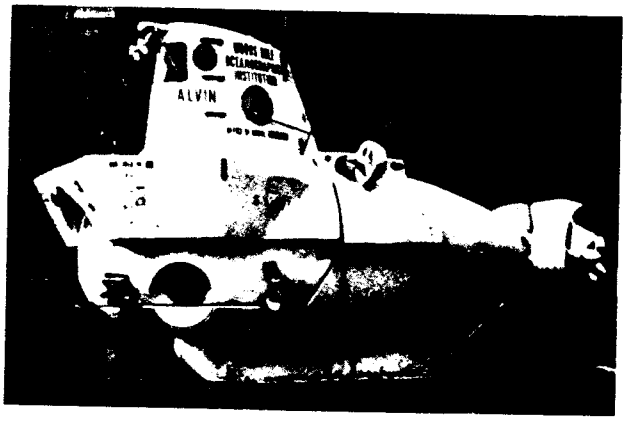


Fig. 12 The *Alvin* by General Mills/Litton Industries, Minneapolis, Minnesota

Table 1 Types of submersibles (see Fig. 13)

A. Manned submersibles
  1. One-atmosphere submersibles
    a. untethered or free-swimming
      - *Trieste*
      - *Alvin*
    b. tethered
      - observation/work bell
      - ADS-atmosphere diving suit, Jim
      - *Mantis*
  2. Ambient-pressure submersibles
    a. untethered: swimmer/diver delivery vehicles
    b. tethered: diving-bell or personnel transfer capsule
  3. Diver lockout submersibles—combination of 1 and 2
    a. untethered
      - *Perry PC 1801*
      - *Johnson Sea-Link I* and *II*
    b. tethered: mobile diving unit

B. Unmanned submersibles
  1. Untethered submersibles
    a. preprogrammed: torpedo
    b. autonomous
      - *Epaulard* (France)
      - *Eave-East*
  2. Tethered submersibles—remotely operated vehicles (ROV)
    a. towed: deep-tow
    b. self-propelled
      - 3DM (three-dimension mobility): CURV
      - 2DM (two-dimension mobility): "bottom crawlers"

The two major categories are manned and unmanned submersibles. Principal types of manned submersibles include one-atmosphere, ambient-pressure, and combination vehicles that may be either untethered or tethered to the support ship. Principal types of unmanned submersibles include untethered and tethered vehicles, either free from the support ship or connected to it or to an intermediate "vehicle garage" by a cable. Unmanned, untethered submersibles may be subtyped as either preprogrammed or autonomous—vehicle systems for which man may either be included or excluded from the "control loop." Unmanned, tethered submersibles may be subtyped as towed or self-propelled, but all of these include man in the "control loop." For this reason, these submersibles are almost always referred to as remotely operated vehicles (ROV).

In a manned, one-atmosphere vehicle, the man is encapsulated in a thick-walled, shell structure called a pressure-hull, which allows his body to remain essentially at the same pressure throughout the dive. These hulls, which resist external pressure, may be cylindrical for relatively shallow depths, but must be spherical for deeper depths, the sphere being the more structurally efficient shape. For a

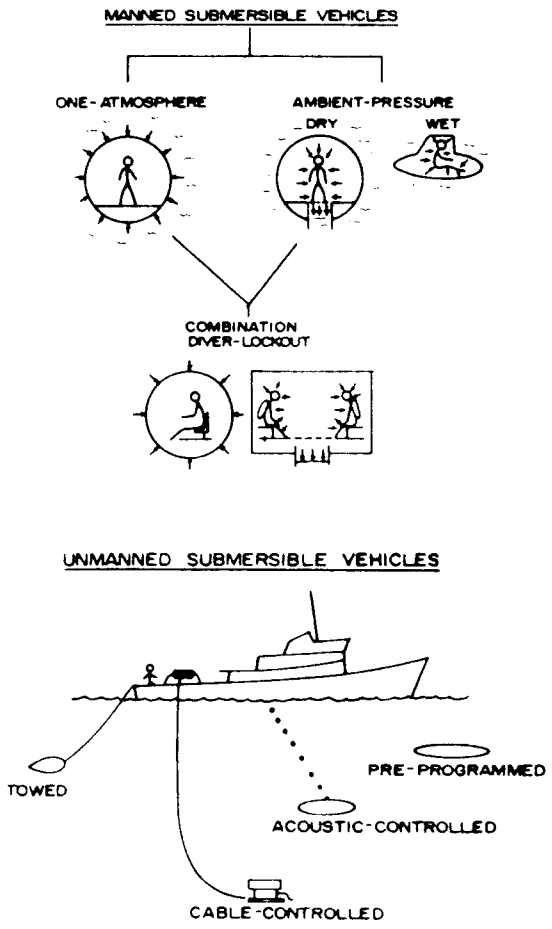


Fig. 13 Categories of submersibles

given weight/displacement ratio of a spherical pressure hull, only the efficiency of the material (strength/density) of the hull limits a one-atmosphere submersible in its depth capability (the world's two bathyscaphes, *Trieste* and *Archimede*, have descended to greater depths than *Alvin*'s 4000 m capability.

In an ambient-pressure vehicle, by contrast, man is still enclosed in a pressure-hull, but his body is subjected to the ambient pressure at the work site. Unlike that of the one-atmosphere vehicle, the hull of this vehicle may resist pressure from both within and without. Ambient-pressure submersibles are depth-limited by man's limitations in withstanding hydrostatic pressure. Diver-depth records in hyperbaric laboratories currently exceed 610 m, while the record at sea is somewhat more than 457 m, "saturation diving" techniques and equipment being used in both instances. Ambient-pressure submersibles also include "wet" vehicles, such as the Waterdinger, on which divers ride through the water. These are usually called diver transport vehicles.

Also, one-atmosphere and ambient-pressure submersible characteristics may be combined in what is known generally as a diver lock-out vehicle. The vehicle operator (the pilot) and the person directing diver activities occupy the one-atmosphere hull while divers occupy the ambient-pressure hull, as in the mobile diving unit. Untethered versions of this vehicle include the Perry *PC 1801* and the *Johnson Sea-Link*. Such a system conserves diver energy and time, especially when work sites are extended over a large area or long distance such as would be the case in pipeline survey and repair work.

Untethered and tethered manned submersibles embody design trade-offs, with the choice between them based on mission requirements and cost-effectiveness. Untethered submersibles, often called free-swimming vehicles, are completely free to move in three dimensions and descend uninhibited by the support ship's position or motion. *Trieste II*, an example of this type (Fig. 14), was retired from active service in May of 1984. Untethered submersibles, however, are obliged to carry their energy source (lead-acid or silver-zinc batteries) and life-support materials on board, thus adding very significantly to the vehicle's weight, size, complexity, and cost.

Use of a tether permits an essentially unlimited supply of power and breathing gases, a lowering/lifting capability, and a superior (to sonar) "hard-wire" communicating ability. The observation/work bell is an example of this type (Fig. 15). Consequently, tethered submersibles can be made lighter, smaller, and less complex than untethered submersibles, while submerged endurance and heavy-work capability are improved. On the other hand, mobility, maneuverability, and depth capability are now significantly more restricted with the vehicle being physically linked to the support ship. In addition, tether drag inhibits vehicle motion, and the dangers of tether entanglement and breakage are always present, particularly if the work site is an underwater structure or is cluttered with obstacles.

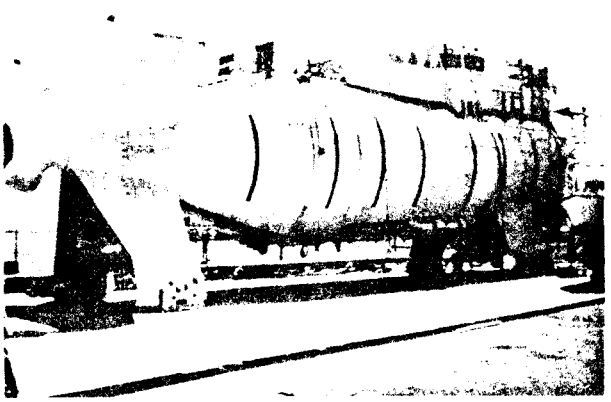


Fig. 14 *Trieste II*: untethered, manned submersible

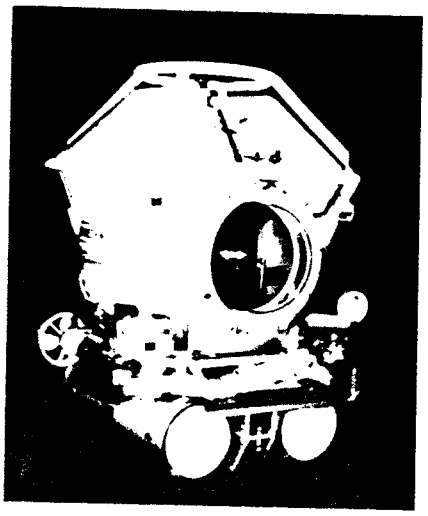


Fig. 15 Observation/work bell

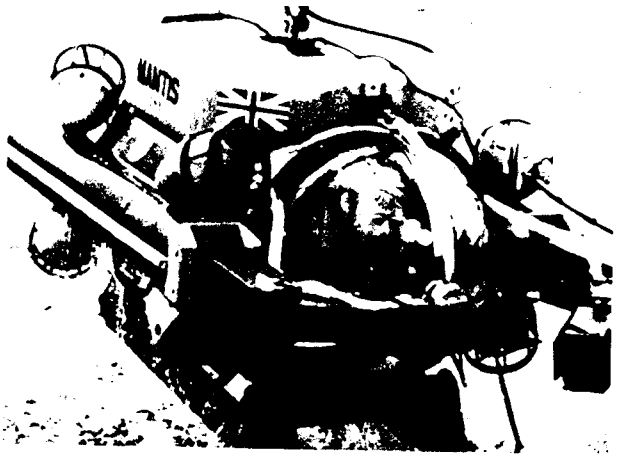


Fig. 16(*a*) *Mantis*—manned/unmanned-tethered

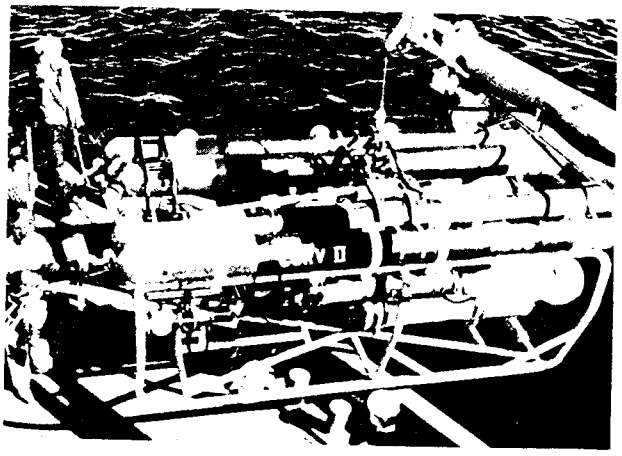


Fig. 16(*b*) *CURV*—unmanned-tethered

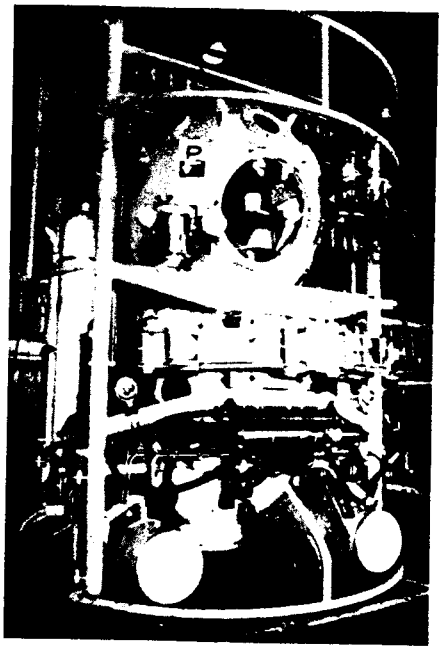


Fig. 16(*c*) Mobile Diving Unit—manned-tethered

Unmanned submersible types also may be designated as untethered or tethered vehicles. Design trade-offs for these types parallel those already noted for manned submersibles, with additional considerations created by man's absence from the vehicle: how to control its motion and work functions to accomplish mission tasks and how to transmit *in situ* information to a remote station. To date, these considerations have made the use of tethers mandatory for most vehicles. Tethers for these functions alone can be of small diameter and light weight.

Untethered submersibles of the autonomous type have been receiving increased attention in the past few years. Mostly in the experimental stage at present, they show potential for eventually removing man from the "control loop" for many applications.

The preprogrammed subtype of unmanned, untethered submersibles has existed since 1866, when Robert Whitehead's torpedo made its debut. Programmed prior to being released from man's control, this type of vehicle is capable of executing predetermined decisions, the simplest example being a torpedo following a prescribed course at a specified depth.

The essential difference between it and the autonomous submersible is that the latter can both make and execute decisions. This ability is the result of incredible developments in the fields of microprocessors/computers and associated software, robotics, and artificial intelligence. However, extremely difficult problems, including the inability of the prototypes to transmit large quantities of information through water, remain to be solved before the autonomous submersible can realize its full potential.

The category of unmanned, tethered submersibles, as noted, includes towed and self-propelled subtypes, with the latter being further subdivided into three- and two-dimensional mobility (3DM and 2DM) vehicles. The towed vehicle is a descendant of the first deep tow (*Fish*). It carries an instrument

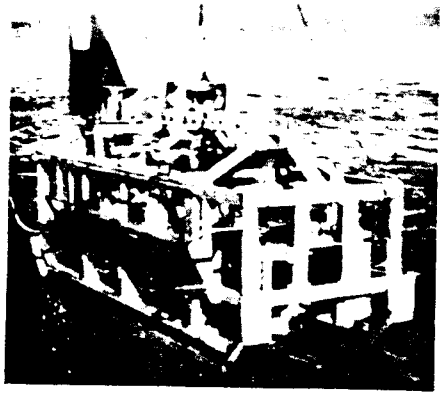


Fig. 16(*d*) *Angus*—unmanned-tethered

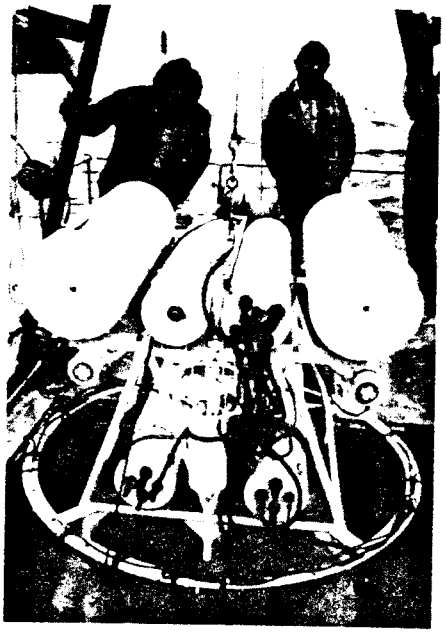


Fig. 16(*e*) *Eave-East*—unmanned-autonomouis

package and a television camera that is connected to a monitor at the operator's station on the towing ship. A typical mission might involve an acoustic survey of a large area of the seafloor, during which the vehicle is "flown" at a constant height above the bottom.

The towed vehicle discussed above must always be in constant motion. However, many needs also exist for vehicles that can hover or possess three or two dimensions of mobility. The 3DM submersible, commonly called a free-swimming vehicle, is equipped with thrusters that give it all of these capabilities. It is tethered to and remotely operated from a surface ship that is either stationary or moving slowly with the submersible in a coordinated manner. The vehicle's mobility and maneuverability can be improved by tethering it not directly to the ship but to a "garage" from which it travels to the work site. The garage, in turn, is suspended from the ship—a particularly effective system for deeper-depth work. This subtype is currently the most numerous of all submersibles, varying from a vehicle somewhat larger than a basketball, an underwater "eyeball" for observation only, to an automobile-sized 3DM used for work or recovery of objects from the seafloor or both.

Finally, a need exists for vehicles to move primarily in two dimensions. Examples of submersibles designed for specialized tasks are the so-called "bottom crawlers" for digging pipeline and cable trenches and the "ship-hull cleaners" that cling to the hull while working. These 2DM submersibles use tracks, wheels, or Archimedes' screws for movement and, in contrast with 3DM vehicles, can be negatively buoyant.

Examples of various types of tethered and untethered submersibles are shown in Fig. 16, including the *Mantis Duplus II* submersible built by Offshore Submersible Engineering, Ltd., of England. This vehicle represents a technology innovation that combines the direct intervention of a manned submersible with the versatility of an unmanned remotely operated vehicle (ROV). It is, in fact, a one-man submersible that can have full manned operation, or remote operation by a surface operator with a man acting as an observer or test operator, or it can be operated unmanned as an ROV.

Another vehicle of the future is the revived Argyonete project of 1968 (Fig. 17). Currently called *SAGA* (Sous-Marin D'Assistance DeGrande Autonomie) and sponsored by COMEX and CNEXO of France, this vehicle is an autonomous diver lockout workboat. With a crew of six and six divers it can remain submerged for ten days at depths of 200 m while diver lockout activity is accomplished.

2.3 Surface Support Ships

Almost all vehicles classified as submersibles require some form of surface support ship. Large submersibles (75 tons and up) were usually towed to and from a dive site (*Trieste, Aluminaut, Ben Franklin,* etc.). The *Auguste Piccard* was modified by Horton Maritime to contain diesel engines for surface propulsion and battery charging. To date, not too much demand has been evident for this first modern-day autonomous manned submersible.

All of the other vehicles, often classed as small manned and unmanned submersibles, are carried on

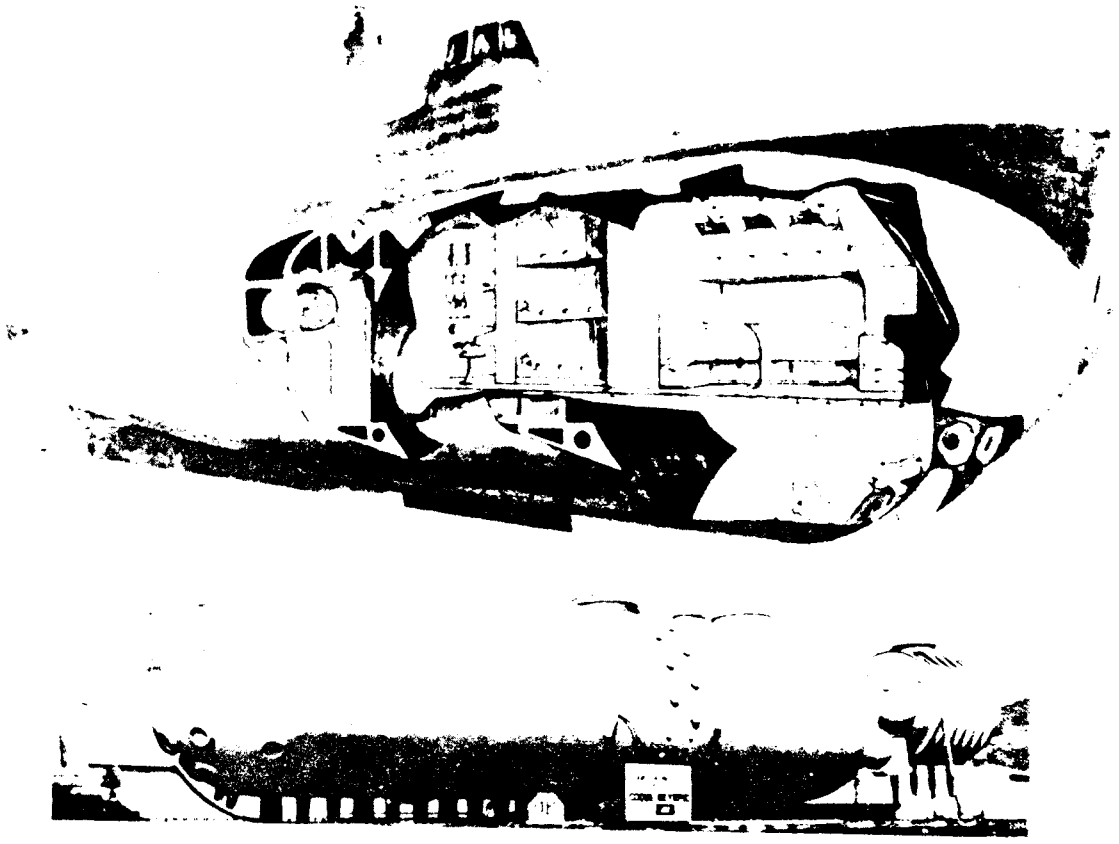


Fig. 17 Artist's rendition of the completed *Argyronete/SAGA*, by COMEX/CNEXO

board and launched from a surface support ship using a variety of handling systems. Typical of these handling systems are hydraulic articulated cranes (Fig. 18), straight leg booms, elevators (Fig. 19), and, most widely used, over-the-stern A-frames (Figs. 20 and 21). The success of offshore submersible operation depends on a total system approach in which the vehicle, the ship, and the handling system are all integrated into a simple, smoothly operating system. In bad weather or under emergency conditions, the degree of preplanning and the applicability of each part of the systems for the task at hand can make

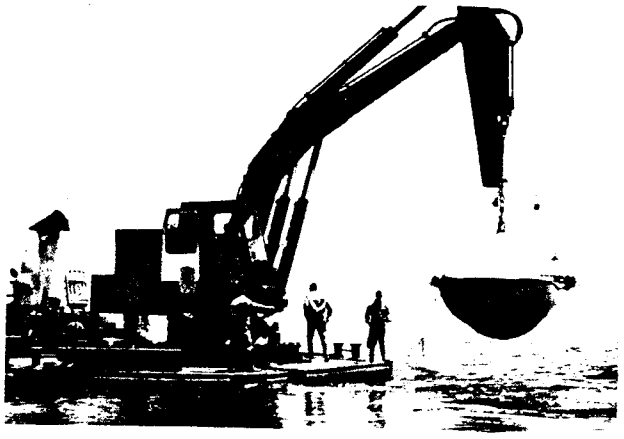


Fig. 18 Hydraulic articulated crane (backhoe type) on the Westinghouse support ship MV *Search Tide*

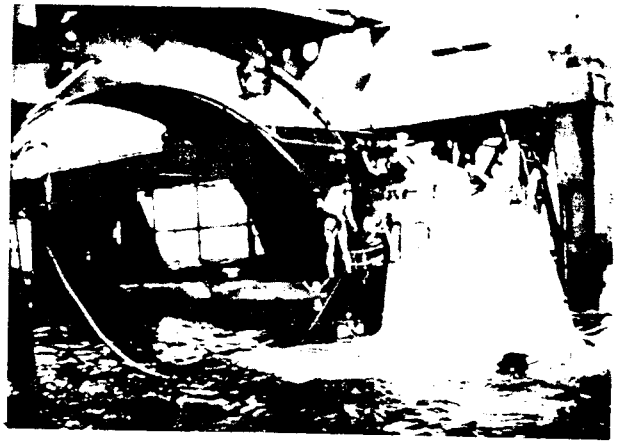


Fig. 19 The *Alvin* entering the elevator-equipped support ship of Woods Hole Oceanic Institution (WHOI)

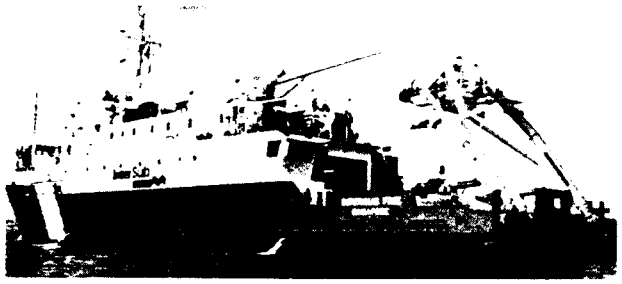


Fig. 20 The stern A-frame submersible handling system of *Intersub Five* (note the hoist system on the A-frame)

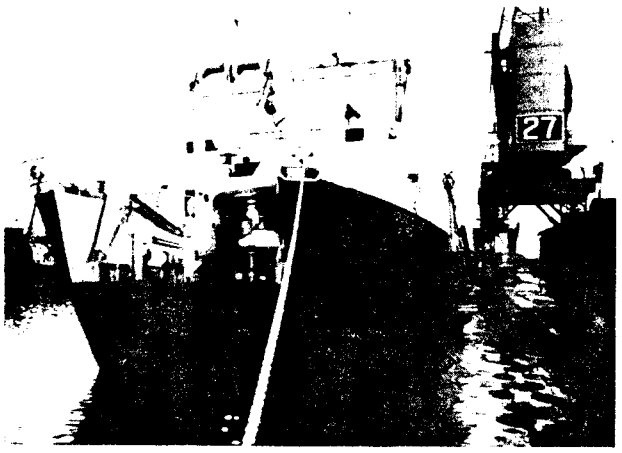


Fig. 21 Stern view of Vickers' *Voyager* (note the A-frame with its pivoting, inverted triangle-stop assembly)

the difference between success and failure. The measure of success must include operational finances, operational technical completion schedule timeliness, and no accidents to material or personnel.

2.4 Future Considerations

Third-generation submersibles? Attempting to predict the future of these vehicles is an audacious undertaking, but perhaps a few thoughts on the next ten years can be advanced based upon a perception of trends in underwater mission requirements and upon the current status of submersibles.

The offshore oil and gas industries are expected to continue to generate the vast majority of markets for submersible services with mission tasks being generally categorized as observation, survey, monitoring, maintenance and repair, operation of underwater equipment, delivery of payloads to the seafloor, underwater construction, and the underwater transportation and support of personnel. Three of several requirements associated with these tasks are singled out for brief comment: operating areas, depth capability, and limiting surface conditions.

It appears that most offshore tasks will continue to have today's requirements for operating area and depth: operating areas in open waters at depths from shallow to 610 m (Fig. 22). However, certain activities can be expected to occur in areas of broken ice, icebergs, and under solid ice, areas that demand a rethinking of traditional submersible-surface ship linkage systems. Also, depths can be expected to increase for some tasks, perhaps to 1830 m as the industry goes further offshore in quest of oil and gas. Limiting surface conditions of poor weather or rough seas or both are, of course, closely associated with operational cost-effectiveness. Consequently, pressure to design systems with greater surface operating capabilities has been building throughout the second-generation period and will continue to build in the future, resulting primarily in improved launch and retrieval systems and better support-ship seakeeping characteristics.

Limiting surface conditions also seems to have initiated a trend toward locating most or much of the oil/gas production and transportation systems on the seafloor. This trend has already begun with the help of today's submersibles in establishing undersea production complexes. If it continues, it will have a profound effect on the development of third-generation submersibles.

Future military and scientific missions will likely have many underwater tasks paralleling those of the offshore industry. Mission requirements will reflect the need to perform at least some of these tasks under the arctic ice cap and down to very deep depths. Already several manned submersibles from various countries, including the United States, France, Japan, and the U.S.S.R., have 6095-m (20 000 ft) depth capability.

Recreational missions are seen as reviving the need for large, manned submersibles. This need has lain dormant since Dr. Jacques Piccard built the *Auguste Piccard* in 1962–63 for carrying 40 passengers to depths of 91 m (300 ft) in Lake Geneva, Switzerland. Currently, there is an increasing number of tourist submersibles being built for service in the more exotic waters of the world. Most of these vehicles are designed to carry up to 50 passengers to depths of 30.5 to 76 m (100 to 250 ft).

The development of third-generation submersibles will be based, in large measure, on an extrapolation of experiences with second-generation vehicles. In this regard, the current status as well as the future potential of individual submersible types must be discussed.

Manned versus unmanned submersibles has been a subject of debate since the late 1960's. It is a fact, however, that manned submersibles have been surpassed by unmanned submersibles for reasons that include lower vehicle and logistic support costs, great improvement in underwater instruments and manipulative capability, and the removal of anxieties and legal/regulatory considerations associated with man in the submersible. Consequently, the third

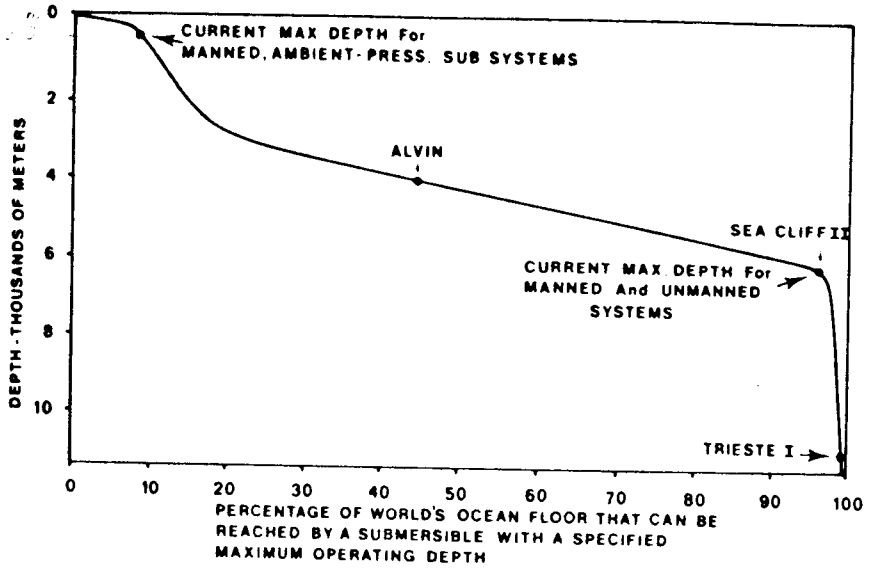


Fig. 22 Percentage of world's ocean floor that can be reached by a submersible with a specified maximum operating depth

generation will most likely see the continued ascendancy of unmanned over manned submersibles, but not to the exclusion of the latter.

Manned submersibles will be used wherever man's presence is deemed necessary. There probably will remain intricate tasks, such as those associated with the operation of a complete subsea production complex, which will require all of man's senses coordinated with his dexterity and mobility. If these complexes utilize one-atmosphere well-head enclosures and central collection centers, manned submersibles will be required for transportation of personnel to and between them. Furthermore, the trend to "break with the surface" may spark the development of nonmilitary submarines to replace surface ships in the support of diving and other underwater activities. Under-ice missions also may call for the use of these large, manned vehicles to conduct or support operations.

The development of the unmanned, autonomous submersible began in the late 1970's and may be expected to continue throughout the third generation and beyond. This vehicle holds promise for overcoming all of the disadvantages of tethers while retaining all the advantages of unmanned over manned submersibles. Of course, it also will be burdened by a limited on-board energy capacity and, at least initially, by limited two-way communications. Its use is seen for relatively low-energy missions, such as inspection and surveying, particularly under ice.

Two phases of development for through-water communications are foreseen: those of synoptic and those of real-time communications. Synoptic communications will permit the reception of limited *in situ* information—a broad overview of the site derived from various sensors on the submersible—but will be insufficient for vehicle control. Real-time communications, those in which there is no delay between the transmission of data from the submersible and its reception by the operator, will allow receipt of large amounts of detailed, perhaps continuously transmitted *in situ* information and remote control of the vehicle if desired. Consequently, the first phase will see limited, but useful, employment of the vehicle, whereas the second phase, which may be considerably longer in coming, will see the full potential of the autonomous vehicle realized.

The first generation of submersibles was characterized as one of "great expectations" and the second as "coming of age." The third generation may well see submersibles "reaching maturity."

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Appendix

This appendix is intended to identify the various submersibles that have been designed and built in the United States and in foreign countries. The following tables included in this appendix provide a brief summary of submersible characteristics and other pertinent data:

Table 2—HYCO submersibles
Table 3—Perry Oceanographic submersibles
Table 4—Selected "other country" submersibles
Table 5—Selected U.S. submersibles

In 1962, Westinghouse Electric Corporation entered into a technical data/hardware development agreement with Capt. Cousteau. In 1964–1965 it operated the diving saucer to gain field experience for the time that their joint venture submersible *Deepstar 4000* (Fig. 23) would be ready for use in 1966. It went on to build *Deepstar 2000* (Fig. 24) and operated both of these vehicles for a number of years. In the early 1970's both of them were sold to French interests.

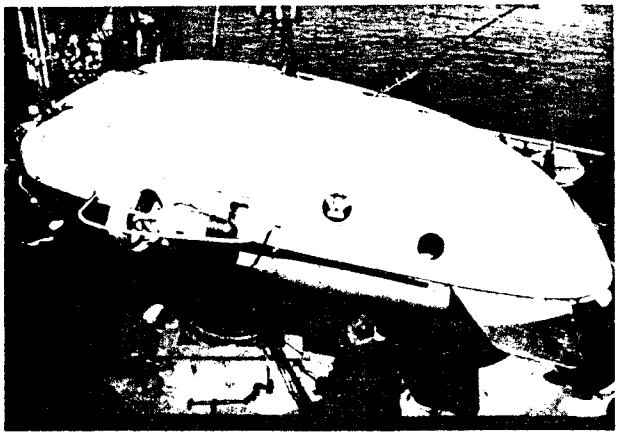


Fig. 23 *Deepstar 4000* by Westinghouse Electric Corp., Oceanic Division, Annapolis, Maryland

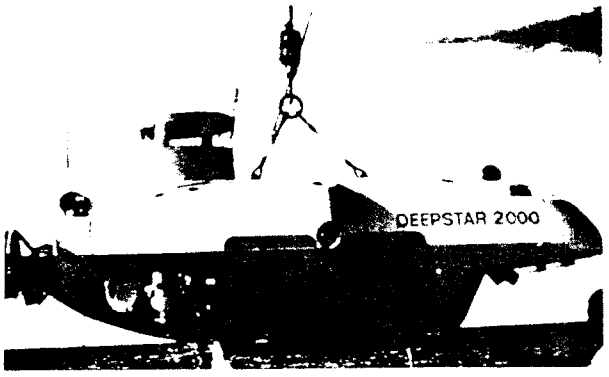


Fig. 24 *Deepstar 2000* by Westinghouse Electric Corp., Oceanic Division, Annapolis, Maryland

General Motors built the Deep Ocean Work Boat (DOWB), which had a unique 360-deg periscope vision system. Unfortunately, the pilot inside was unable to really comprehend what he was seeing when he looked at the whole world at once, so the optics were changed to give a normal (wide angle) view of the area in front of the submersible.

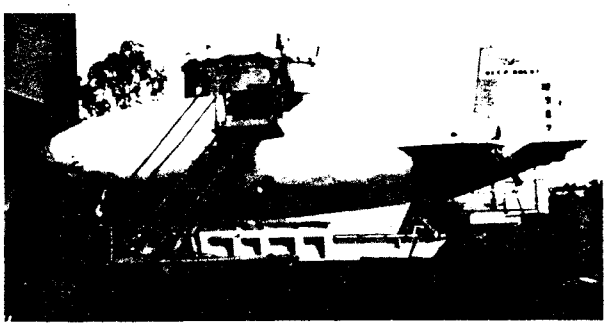


Fig. 25 *Deep Quest* by Lockheed Missiles and Space Corp., Sunnyvale, California

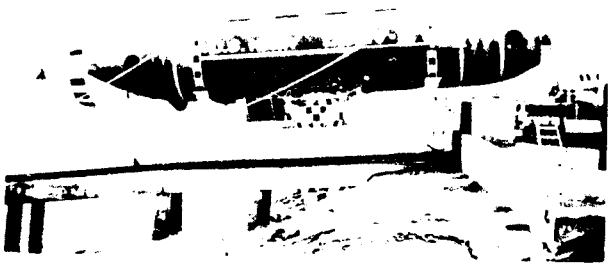


Fig. 26 *DSRV-1* and *-2 Mystic/Avalon* by Lockheed Missiles and Space Corp., Sunnyvale, California

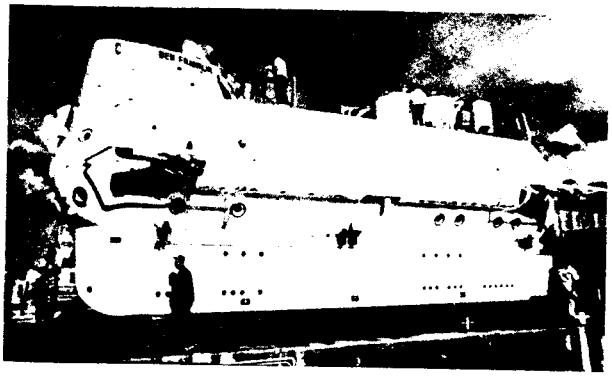


Fig. 27 *Ben Franklin* by Grumman Aerospace Corp., Bethpage, New York

Lockheed built *Deep Quest* (Fig. 25) and then went on to build the U.S. Navy's Deep Submergence Rescue Vehicles (DSRV) *Mystic* and *Avalon* (Fig. 26). These are operated by the Navy's Submarine Development Group 1 in San Diego, along with *Sea Cliff* and *Turtle*.

Grumman Aircraft entered into an agreement with Jacques Piccard, son of the bathyscaphe inventor,

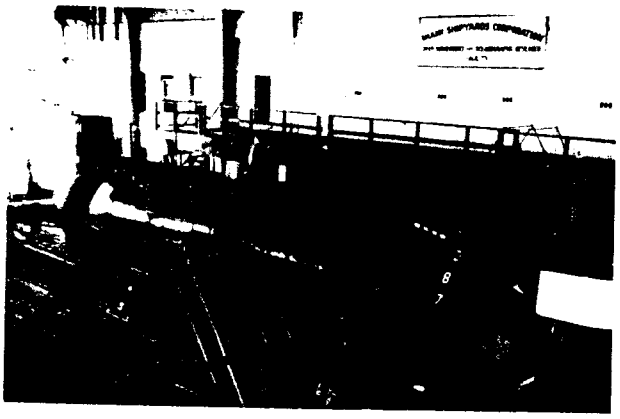


Fig. 29 *Aluminaut* by Electric Boat Division of General Dynamics Corp., Groton, Connecticut

Auguste Piccard. The *Ben Franklin* (Fig. 27) was built in 1968, and a 1444-mile-long, 30-day Gulf Stream drift mission was conducted in 1969. Jacques Piccard had also built a large mid-depth boat—the *Auguste Piccard*, with 185 tons displacement and a length of 28.5 m (93.5 ft)—for use as a tourist attraction at the Lake Lucerne Swiss National Fair. Both of these craft have been sold to Horton Maritime Explorations of Vancouver, British Columbia, Canada.

North American Aviation built the *Beaver* submersible (Fig. 28), which is now operated by International Underwater Contractors of City Island, New York. Reynolds Aluminum Company had the *Aluminaut* (Fig. 29) built in 1963. This was the submersible that assisted in the recovery of the *Alvin* in 1969.

During the 1970's, the offshore oil industry sparked the need for a smaller class of less expensive underwater work submersibles. Most of these were built by two companies: International Hydrodynamics



Fig. 28 *Beaver* by North American Rockwell Corp., Seal Beach, California

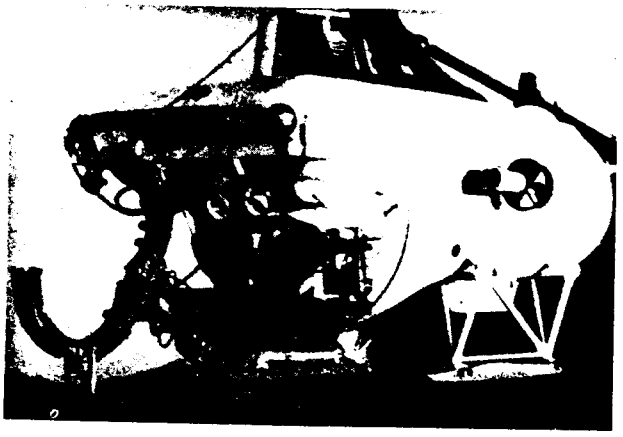


Fig. 30 *Pisces 8* by International Hydrodynamics (HYCO) of Vancouver, British Columbia

(HYCO) of Vancouver, British Columbia, with its *Pisces* series of submersibles (Fig. 30) and the Perry Oceanographics Company of Riveria Beach, Florida, with its *PC* classes of submersibles (Figs. 31 and 32).

Table 2 lists generalized data on HYCO submersibles; Table 3 provides similar information on the Perry submersibles.

Other countries besides the United States and Canada have participated in the development of submersibles. Selected examples of these submersibles are shown (Figs. 33, 34, 35, and 36) while Table 4 provides data on some of these countries' more widely known vehicles. Table 5 summarizes the characteristics of some additional U.S. submersibles while an example of one of these vehicles is shown (Fig. 37).

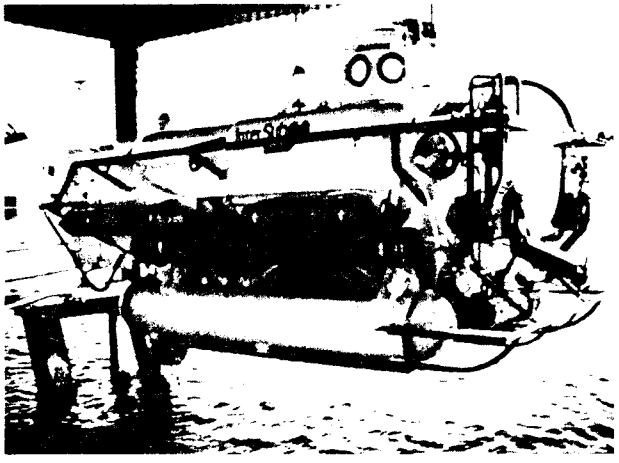


Fig. 31 Perry Oceanographic *PC 12*

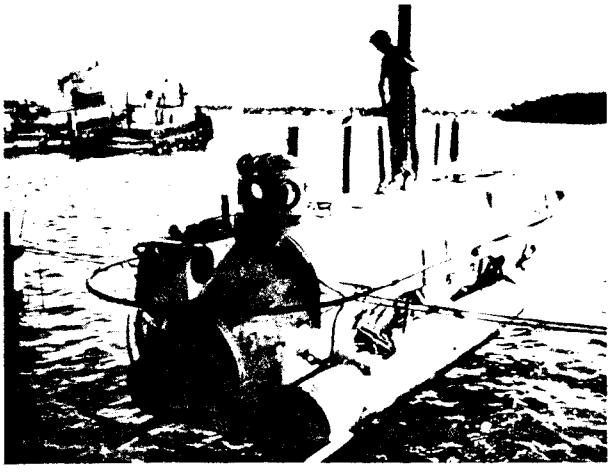


Fig. 32 Perry Oceanographic *PC 15*

Table 2 HYCO submersibles

Designation	Depth, m (ft)	Size,[a] m			Displace., metric tons	Crew	User/Owner[b]
		L	W	H			
Pisces 1	914 (3000)	6.4	3.3	3.3	7.5	2	HYCO
Pisces 2	732 (2400)	5.8	3.2	3.1	11.9	2–3	Vickers/British Oceanics
Pisces 3	914 (3000)	5.9	3.2	3.2	11.9	2–3	Vickers/British Oceanics
Pisces SDL	610 (2000)	6.1	3.0	3.7	11.8	2 + 2 divers	Canadian Armed Forces (Navy)
Pisces 4	2012 (6600)	6.1	3.0	3.7	10.3	2–3	Canadian Department of the Environment
Pisces 5	2012 (6600)	6.1	3.2	3.3	10.8	2–3	Intersub
Pisces 6	2012 (6600)	5.85	3.05	3.6	9.8	2–3	International Underwater
Pisces 7	2000 (6600)	6.88	3.2	3.65	10.7	2–3	USSR
Pisces 8 (Fig. 30)	1000 (3300)	6.2	3.3	3.2	10.7	2–3	Vickers/British Oceanics
Pisces 10	1000 (3300)	6.2	3.3	3.2	10.7	2–3	Vickers/Fred Olson

Table 2 ***continued***

Designation	Depth, m (ft)	Size,[a] m			Displace., metric tons	Crew	User/Owner[b]
		L	W	H			
Pisces 11	2000 (6500)	6.8	3.2	3.65	10.7	2–3	USSR
Aquarius	305 (1000)	4.2	1.9	2.0	5.4	2	HYCO
Leo	610 (2000)	6.1	3.1	3.7	10	2–3	P&O Subsea
Taurus 1	335 (1100)	10.1	3.3	3.3	24	3 + 3 divers	Vickers/British Oceanics
PC 15 (3) (Fig. 32)	366 (1200)	10.1	3.3	3.3	14.5	2 + 3 divers	Vickers/British Oceanics
PC 16 (2)	914 (3000)	7.6	2.4	3.0	13.6	4[c]	Intersub/Subsea Oil Services
PC 18 (5)	305[d] (1000)	6.7	2.4	2.6	9.1	2 + 2 divers	Intersub (3), Shell, Superpesa

[a]Size and displacement figures change from mission to mission depending on equipment carried and the job to be done.
[b]The user/owner changes as vehicles are sold—many are inactive awaiting sale.
[c]Dry transfer with bottom hatch.
[d]200 (640) lockout.

Table 3 Perry Oceanographic submersibles

Designation	Depth, m (ft)	Size,[a] m			Displace., metric tons	Crew	User/Owner[b]
		L	W	H			
PC 3 A (2)	91 (300)	5.5	1.06	1.75	2.15	2	U.S. Army/U.S. Air Force
PC 3 B (2)	182 (600)	6.7	1.06	1.75	2.75	2	Ocean Systems/International U/W Contractors
Deep Diver (PLC 4)	407 (1335)	6.7	1.5	2.6	8.3	2 + 2 divers	EO. Link
Shelf Diver (PLC 4B)	244 (800)	7.0	1.7	2.7	8.6	2 + 2 divers	Intersub
OPSUB (tethered)	610 (2000)	5.5	2.6	2.3	5.2	2	Ocean Systems
PC 5C	366 (1200)	6.7	1.4	2.65	10.0	2–3	Subsea Oil Services
PC 8 (2)	366 (1200)	5.8	2.1	1.8	5.0	2	Northern Offshore (1) (Intersub)
PC 9	412 (1350)	7.9	2.2	2.4	10.4	2–3	Brown & Root
PC 12 (4) (Fig. 31)	366 (1200)	6.7	2.4	2.4	7.2	2–3	Northern Offshore (Intersub)
PC 1202 (lockout)	305 (1000)	9.4	2.4	2.7	13.6	2 + 2 divers	Intersub
PC 14 (2)	183 (600)	6.0	1.8	2.4	5.4	2	U.S. Army/Texas A&M University

[a]Size and displacement figures change from mission to mission depending on equipment carried and the job to be done.
[b]The user/owner changes as vehicles are sold—many are inactive awaiting sale.

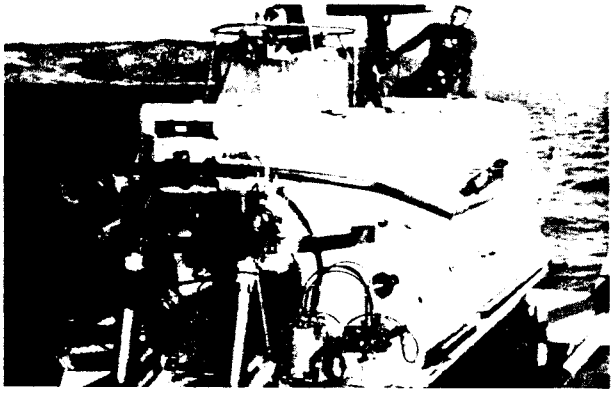


Fig. 33 *LR-4*, United Kingdom



Fig. 35 *Mermaid VI*, Germany

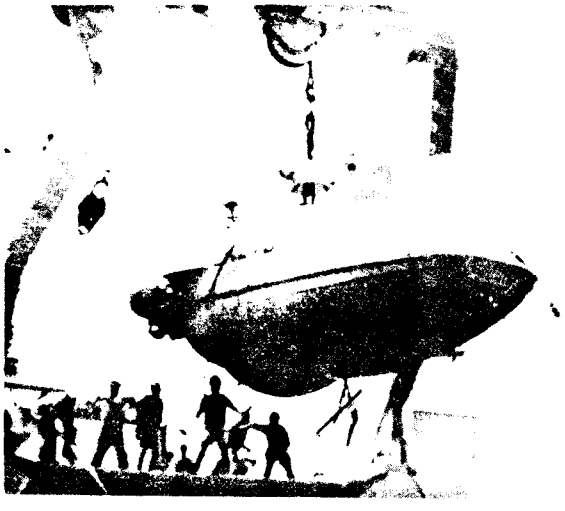


Fig. 34 *Cyana*, France

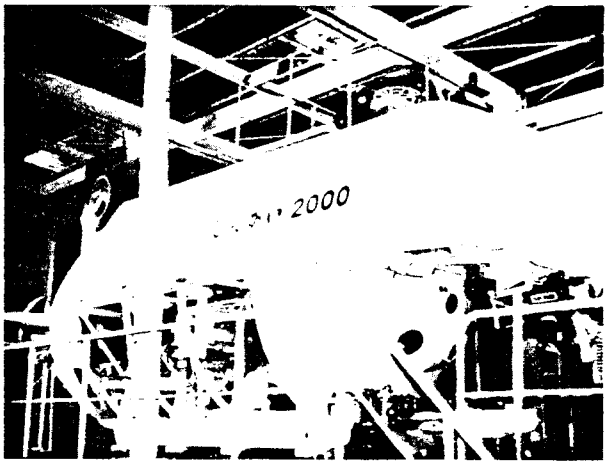


Fig. 36 *Shinkai 2000*, Japan

Table 4 Selected "other country" submersibles

Designation	Depth, m	Size, m			Displace.	Crew	Builder
		L	W	H			
England:							
LR-2	366	7.3	3.0	2.6	11.8 tons	2–3	Slingsby Engineering, Ltd.
LR-3	457	7.3	2.8	2.7	12.7 tons	2–3	Slingsby Engineering, Ltd.
LR-4 (lockout) (Fig. 33)	457	10.36	3.0	2.7	12.87 tons	2–3	Slingsby Engineering, Ltd.

Table 4 ***continued***

Designation	Depth, m	Size, m			Displace.	Crew	Builder
		L	W	H			
LR-5 (lockout)	457	9.6	3.0	2.7	15 tons	3 + 2 divers	Slingsby Engineering, Ltd.
Mantis (10)	610	2.05	1.37	1.07	1179 kg	1	Offshore Submersible Engineering, Ltd. (OSEL)
ROV Mantis (2)	700	2.7	1.37	1.07	1043 kg	0–1	OSEL
France:							
Cyana (Fig. 34)	3000	5.7	3.0	2.10	8.5 tons	3	Centre D' Etudes Marines Avances (CEMA)
Griffon	600	7.8	2.3	3.1	12.5 tons	3	CERTSM (Naval Res. Labs)
SM358 (2) (lockout)	300	7.1	2.5	2.9	12.8 tons	3 + 3 divers	COMEX
SM360 (lockout)	300	10.0	2.75	2.9	25 tons	3 + 3 divers	COMEX
Germany:							
Tours 64/66	300	7.28	3.8	3.2	10 tons	2	Maschinenbau Gabler GMBH
Mermaid II	305	5.5	1.7	2.6	6.2 tons	2	Brucker Meerestechnik GMEH
Mermaid III/IV (lockout)	260	7.2	2.5/1.8	2.7	15.5/13.0 tons	2 + 2 divers	Meerestechnik GMEH
Mermaid VI (4) (Fig. 35)	600	8.0	2.9	3.2	19.1 tons	2 + 3 divers	Meerestechnik GMEH
Japan:							
Kuroshio (retired)	200	11.8	2.2	3.2	12.5 tons	3–4	Japan Steel & Tube Co.
Shinkai (old) (retired)	600	15.3	5.5	5.0	100 tons	4	Kawasaki Heavy Industries
Shinkai 2000 (new) (Fig. 36)	2000	9.3	3.0	2.9	25 tons	3	Mitsubishi Heavy Industries
Italy:							
MSM-1	600	9.9	2.5	3.4	21.6 tons	3	Cantiere Navale Breada
Sweden:							
URF[a]	460	13.5	4.3	4.1	52 tons	5 + 25 rescuees	Kockums Shipyard for Swedish Navy

[a]Swedish Submarine Rescue Vehicle

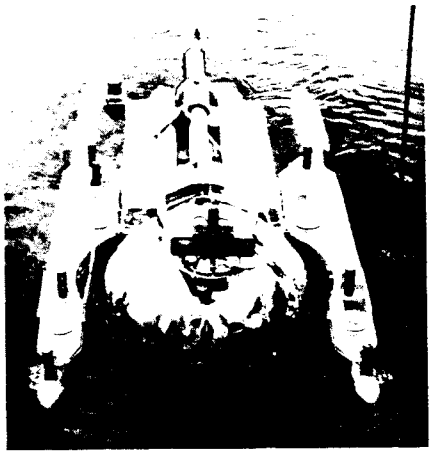


Fig. 37 *Johnson Sea Link I*, United States

Table 5 Selected U.S. submersibles

Designation	Depth, m	Size, m			Displ., tons	Crew	Builder
		L	W	H			
Nekton A/B/C	305	4.6	1.1	1.8	2.3	2	Nekton, Inc., San Diego, CA
Pioneer	366	5.1	2.4	2.7	5.4	2	Martch International, Houston TX
Beaver (Fig. 28)	610	7.9	3.5	3.1	17	2 + 2 divers	North American Rockwell, Seal Beach, CA
Ben Franklin (Fig. 27)	610	14.6	6.1	6.4	143	6	Grumman Aerospace Corp., Bethpage, NY
Deepstar 2000 (Fig. 24)	610	6.1	2.3	2.6	8.75	3	Westinghouse Electric Corp., Annapolis, MD
Star III (Fig. 10)	610	7.5	2.1	2.4	10.5	2	Electric Boat Division, General Dynamics Corp., Groton, CT
Johnson Sea Link I/II (lockout) (Fig. 37)	800	6.9	2.4	3.2	9.5	2 + 2 divers	Harbor Branch Foundation, Fort Pierce, FL
Deepstar 4000 (Fig. 23)	1220	5.5	3.3	2.1	9	3	Westinghouse Electric Corp., Annapolis, MD
SDRV Mystic/Avalon (Fig. 26)	1525	15.0	2.5	3.5	37.3	3 + 24 rescuees	Lockheed Missiles and Space Corp., Sunnyvale, CA
Aluminaut (Fig. 29)	1830[a]	15.5	4.7	5.0	76	6	Electric Boat Division, General Dynamics Corp., Groton, CT
Deepquest (Fig. 25)	2440	12.2	5.8	4.0	52	4	Lockheed Missiles and Space Corp., Sunnyvale, CA
Alvin (Fig. 12)	4000	7.6	2.4	3.9	16.7	3	General Mills/Litton Industries, Minneapolis, MN
Sea Cliff/Turtle (Fig. 11)	6000/3000	7.9	3.6	3.6	24	3	Electric Boat Division, General Dynamics Corp., Groton, CT

[a]The Aluminaut was designed for 4574 m, but has been tested at only 1830 m.

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

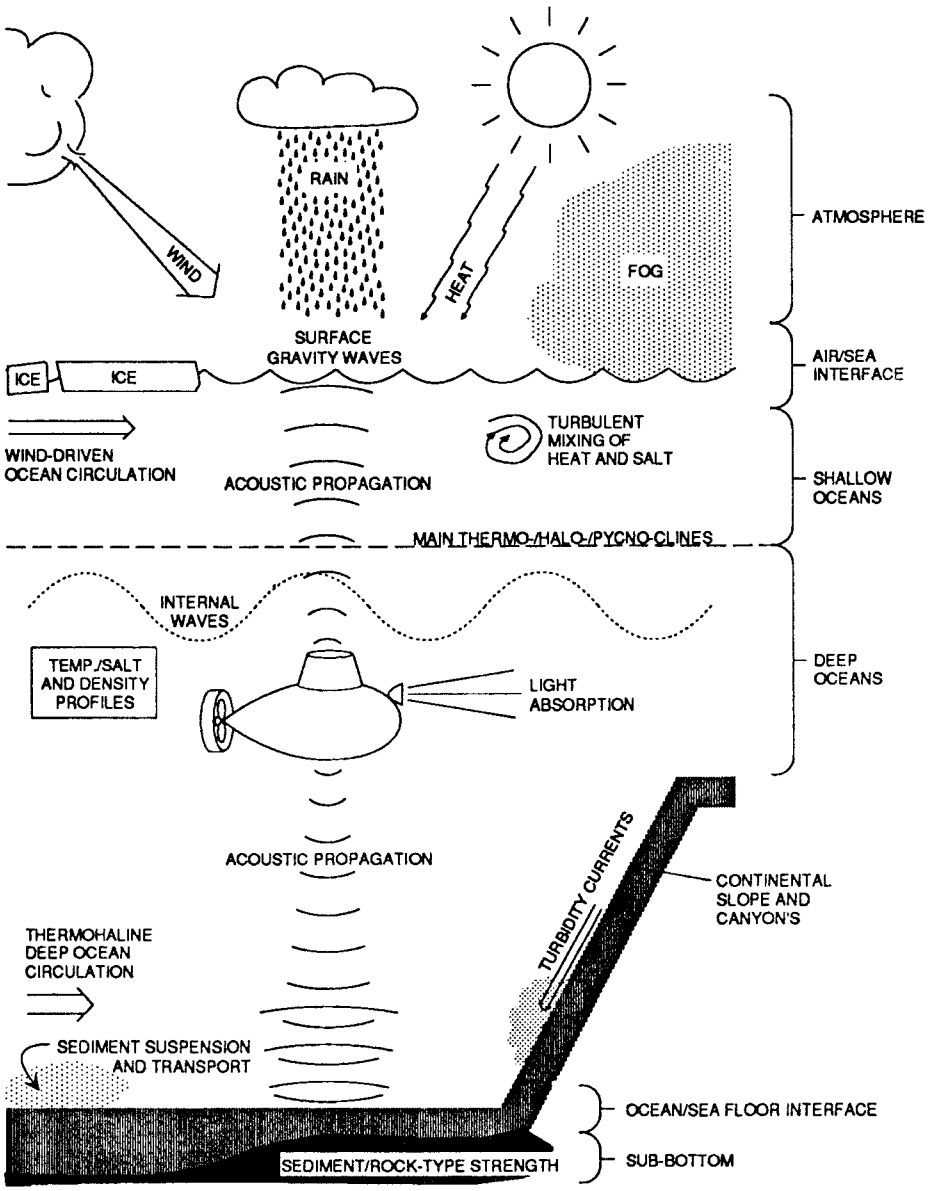


Fig. 1 Factors affecting submersible design for various regions of ocean profile as shown at right

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 consid-

ered, 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 $\frac{1}{2}$, 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.)

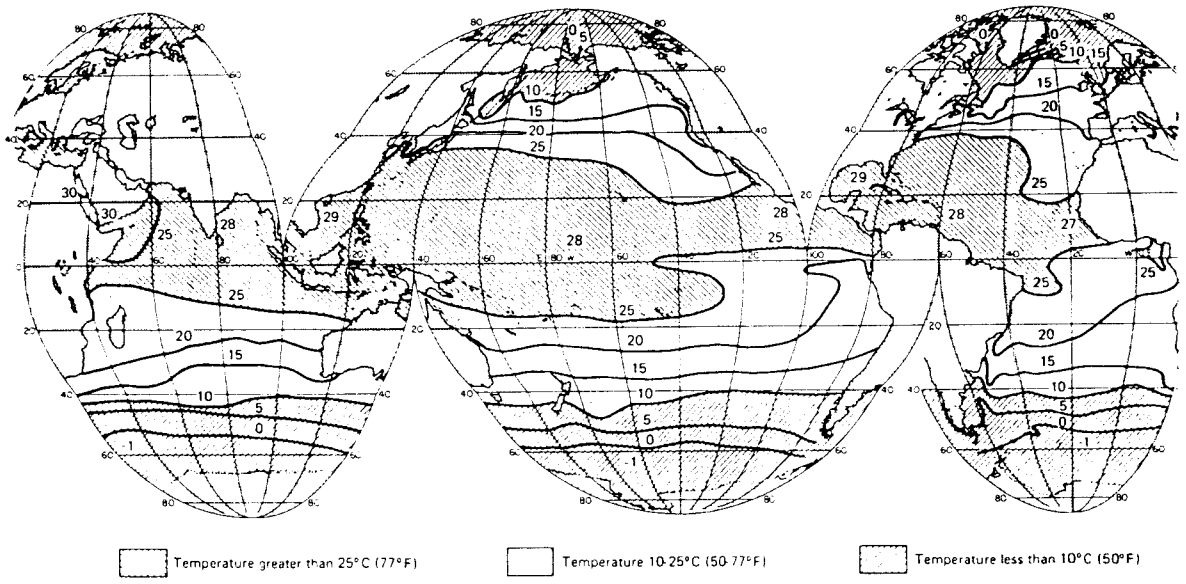


Fig. 2 Sea-surface temperatures in August. (From Sverdrup et al., 1942)

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

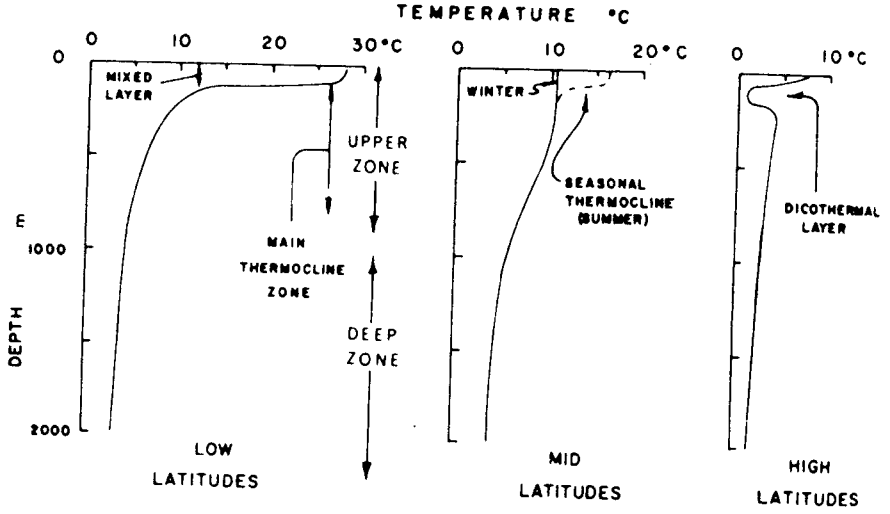


Fig. 3 Typical vertical distribution of oceanic temperature

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.

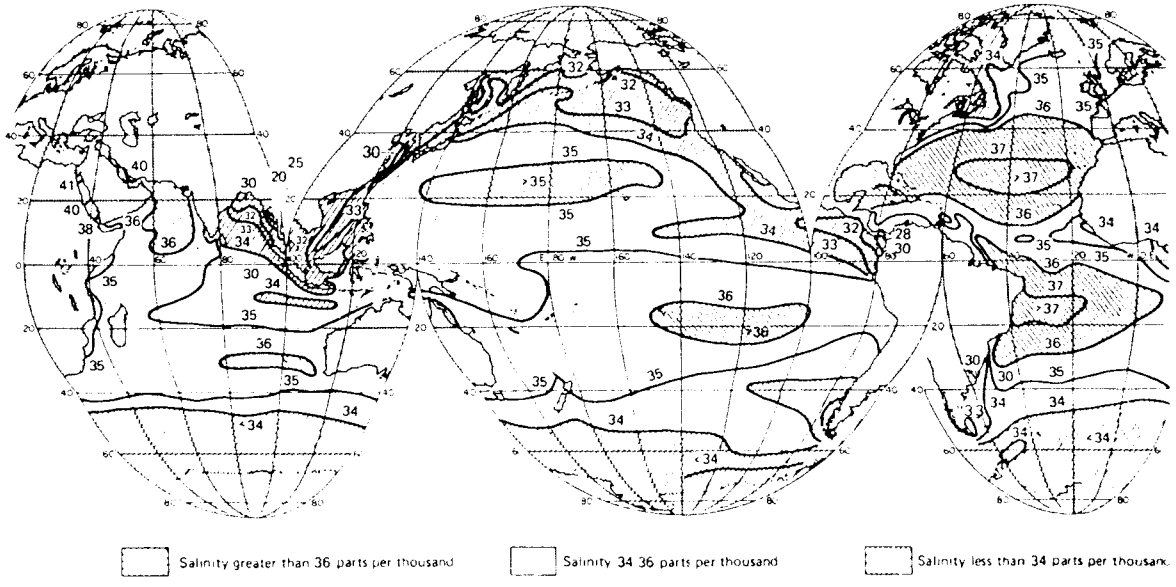


Fig. 4 Salinity of sea-surface water in northern summer. (From Sverdrup et al., 1942)

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

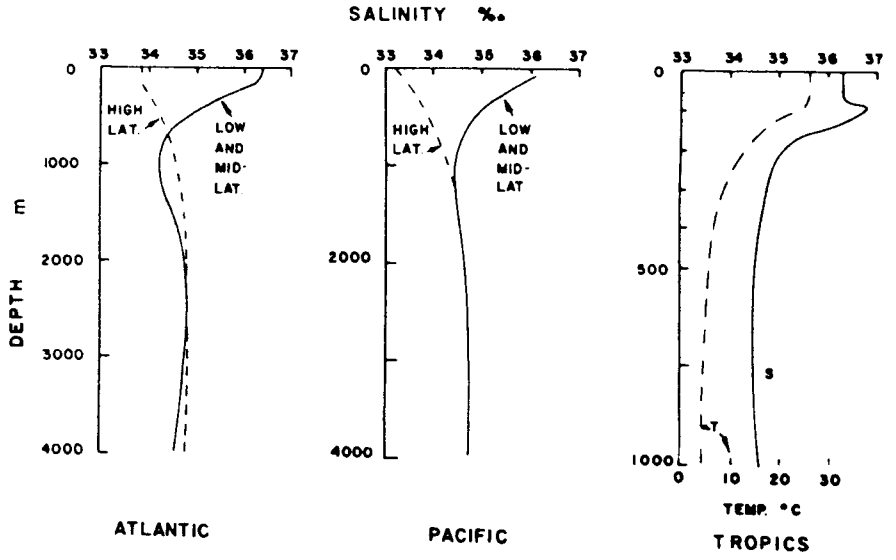


Fig. 5 Vertical variation of oceanic salinities

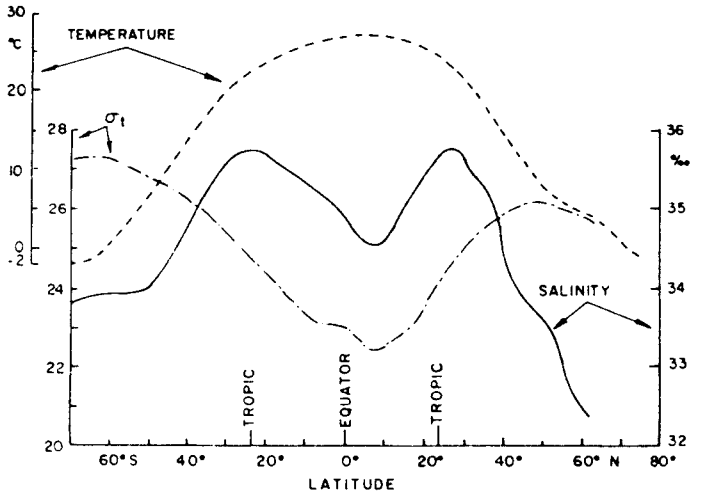


Fig. 6 Latitudinal variation of oceanic temperatures, salinities, and densities

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.

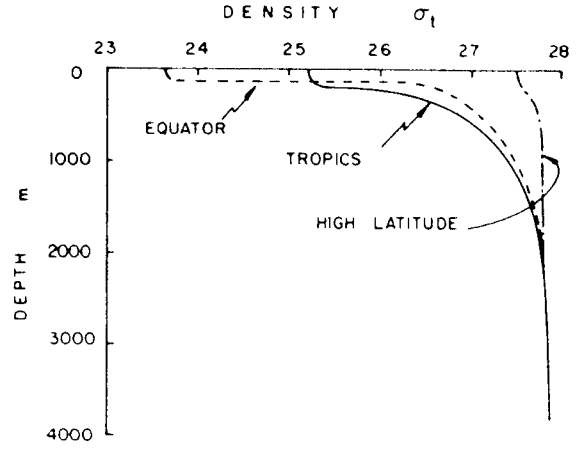


Fig. 7 Vertical profiles of densities at various locations

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-

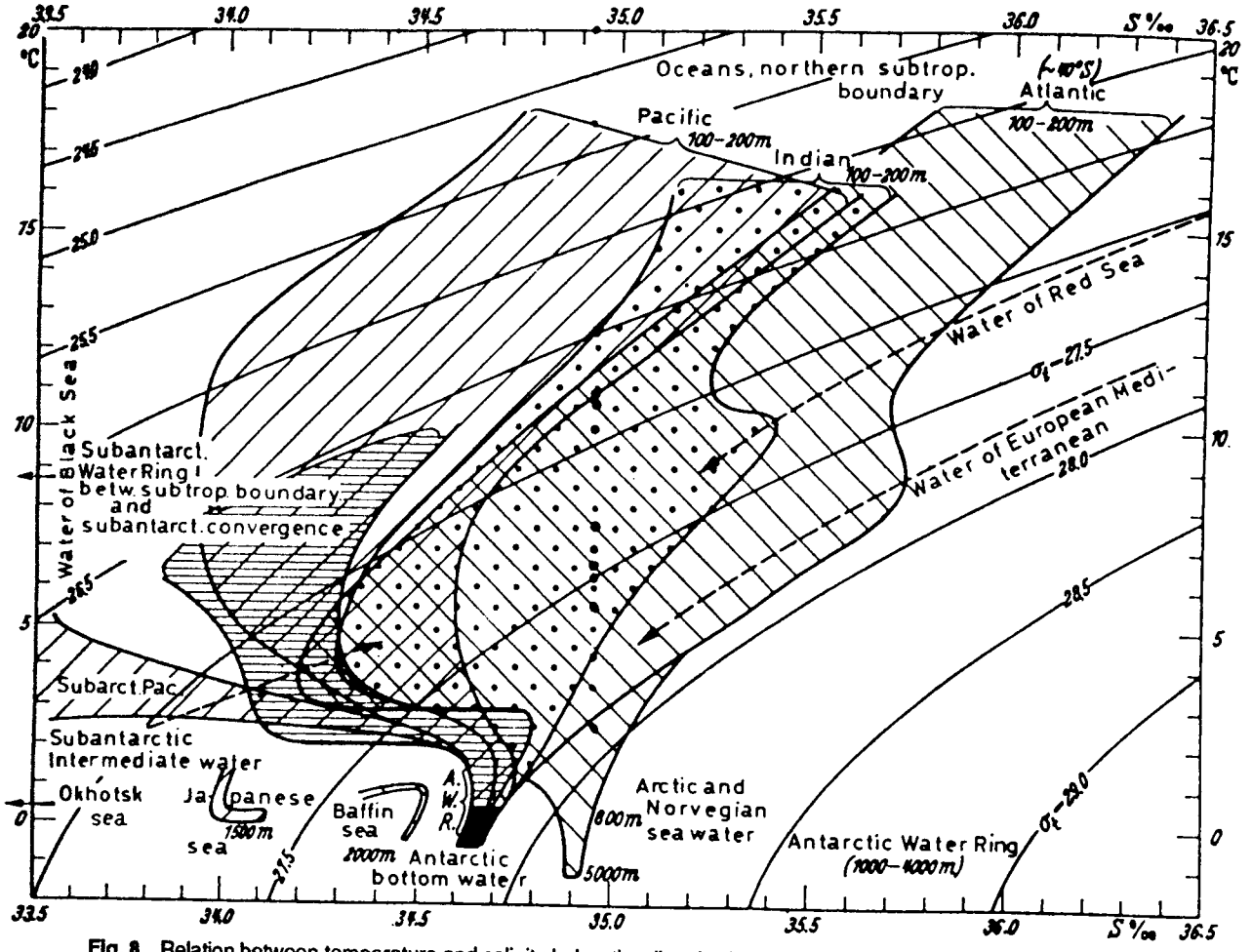


Fig. 8 Relation between temperature and salinity below the disturbed near-surface layer for the entire world ocean

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 Millero, 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

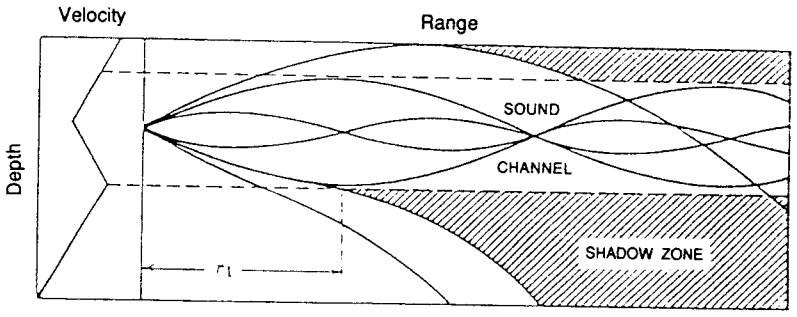


Fig. 9 Acoustic ray trace showing sound channel and shadow zone

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

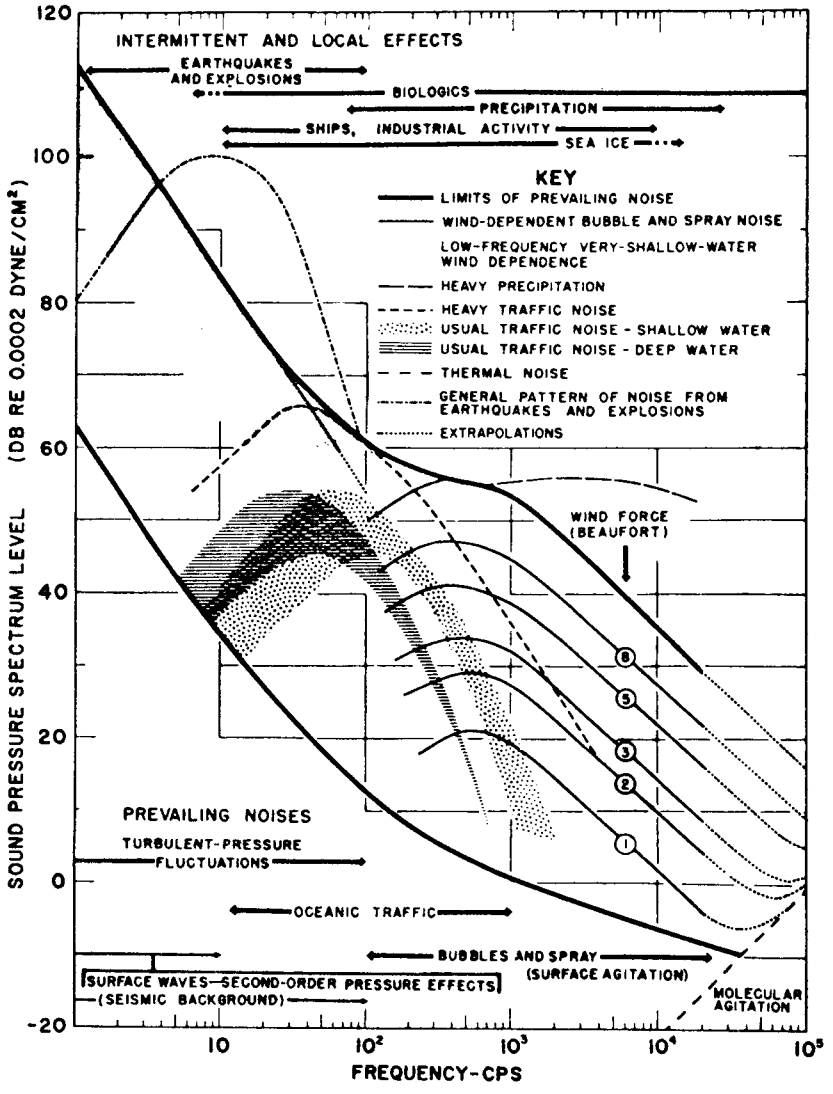


Fig. 10 Background spectrum of ambient noise level in the ocean

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.

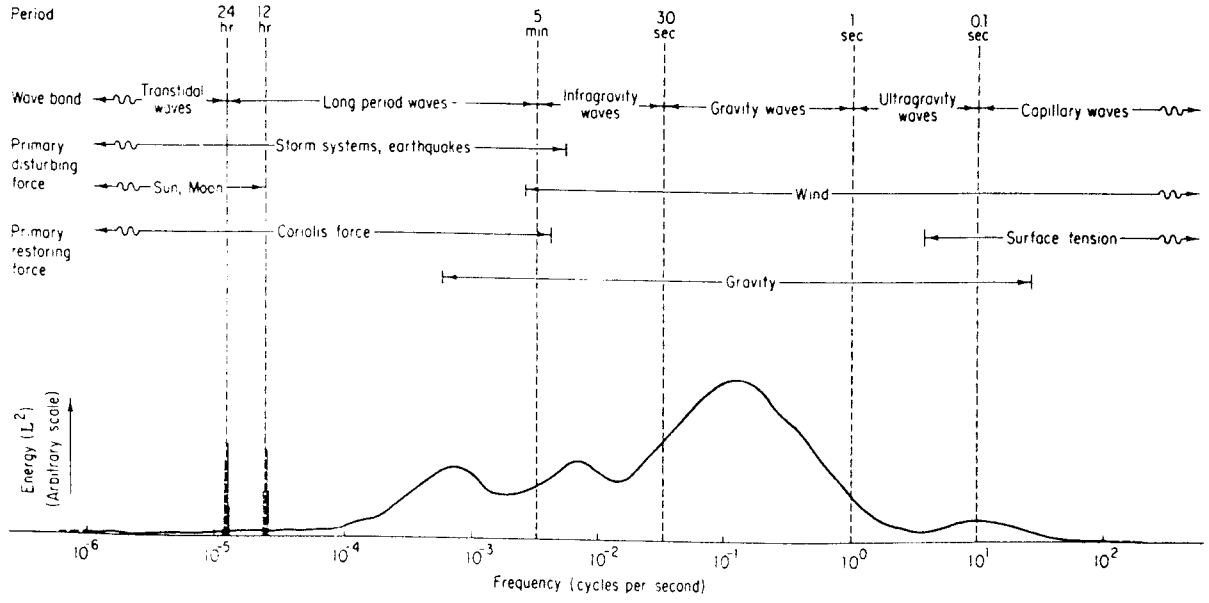


Fig. 11 Estimated schematic representation of the energy contained in the surface waves of the oceans

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 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,

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

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

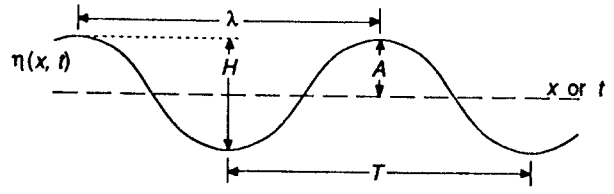
where g is the acceleration due to gravity and h is the water depth. The phase velocity can be determined from equation (10), but not simply, since we cannot get an explicit solution to ω/k.

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



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

Fig. 12 Characteristics of a wave

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

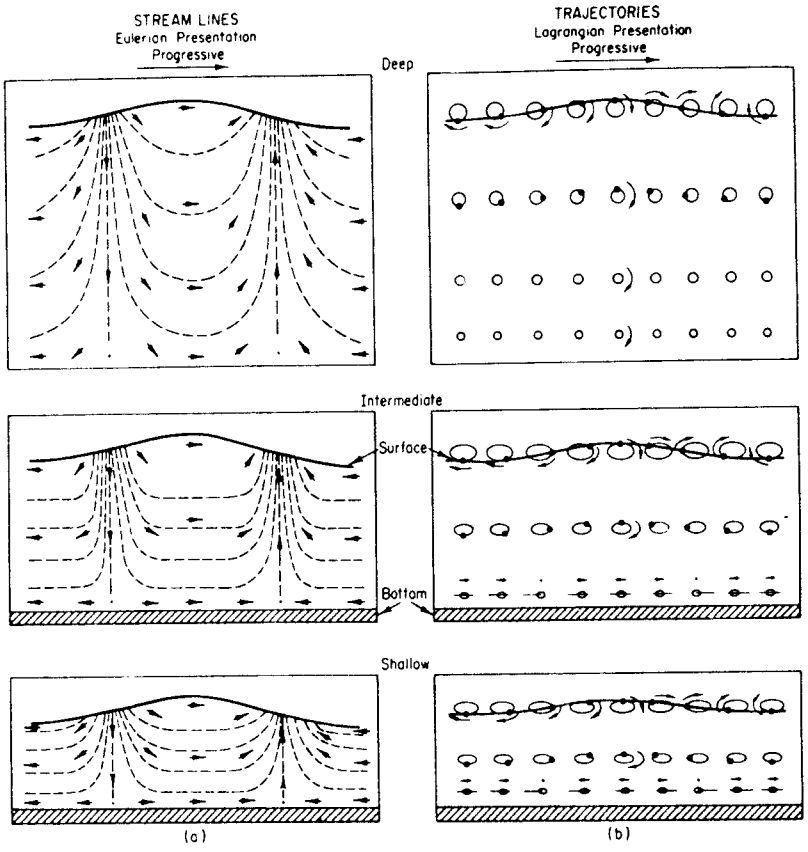


Fig. 13 Relationships of Airy solutions in deep, intermediate, and shallow waters

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.

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

Hence the total energy is $\frac{1}{2}(\rho g A^2)$ or, in terms of the wave height,

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

This could be thought of as the work required to lift a sheet of water of thickness H through a vertical distance of $\frac{1}{8}$ the wave height, or the energy in a layer of water of depth H moving at a velocity of the square root of $\frac{1}{4}(gH)$. 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

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

The flux of power in a given length of wave of wave crest s is about $10^4\ H^2 Ts$, 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

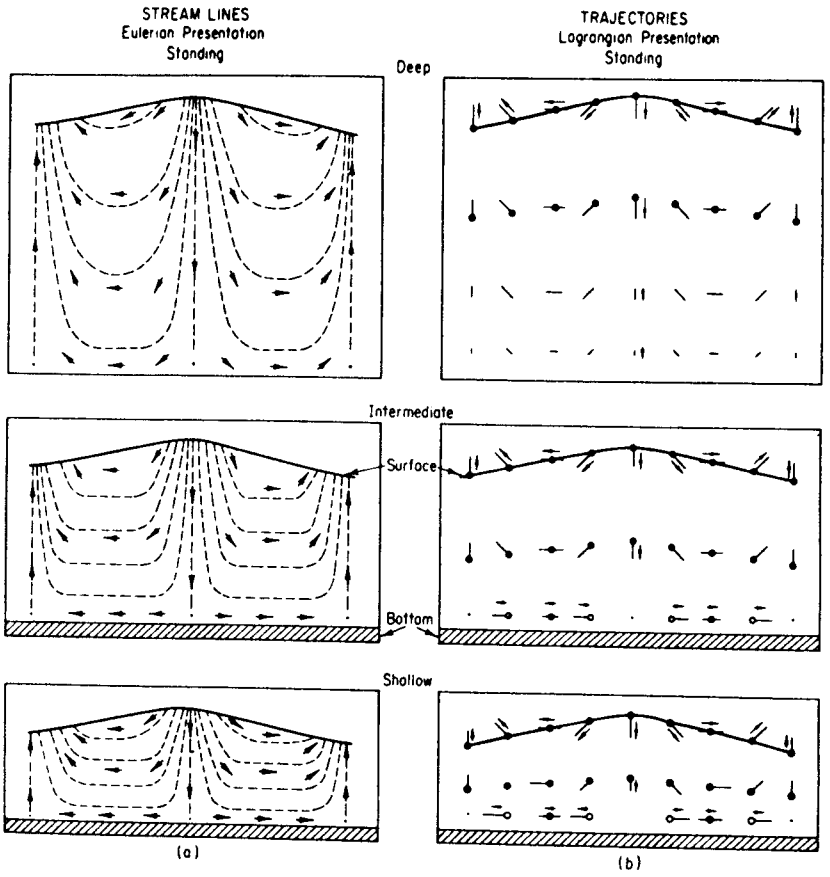


Fig. 14 Summary of orbital velocities beneath a standing wave

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

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

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

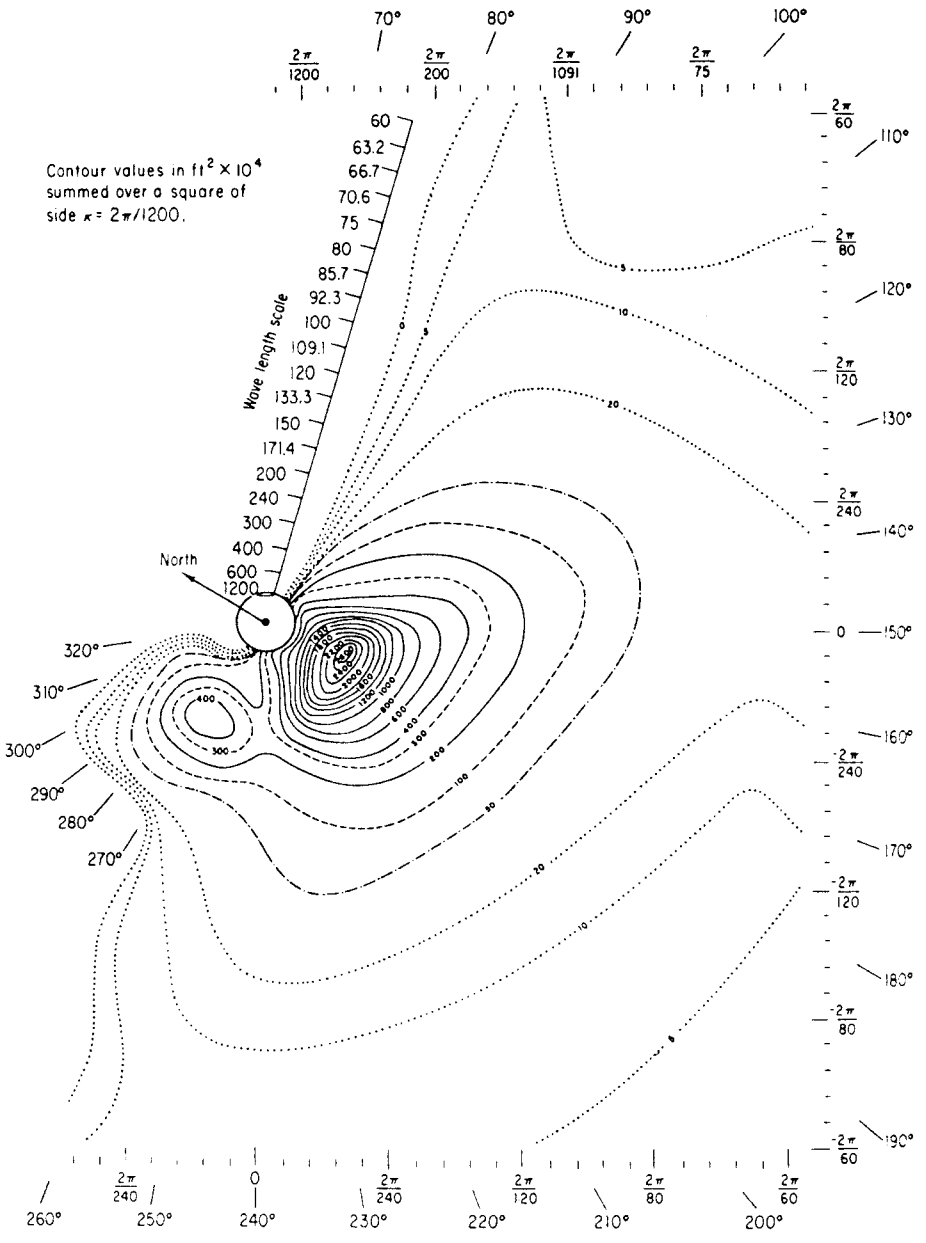


Fig. 15 Smoothed directional spectrum of wind-generated waves

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

$$\text{energy density} = \frac{Bg^2}{f^5} \qquad (16)$$

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

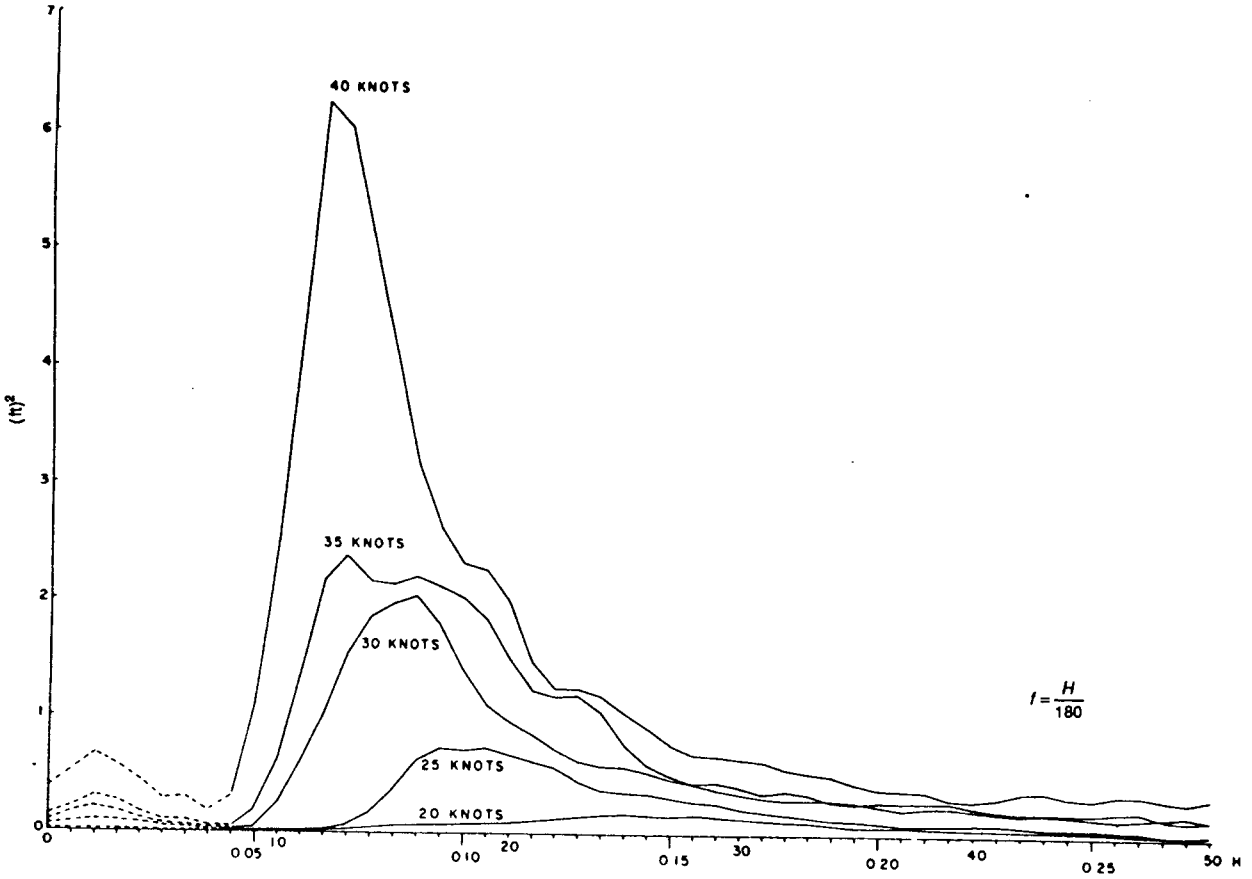


Fig. 16 Averages of selected spectra for winds from 20 to 40 knots

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

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

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

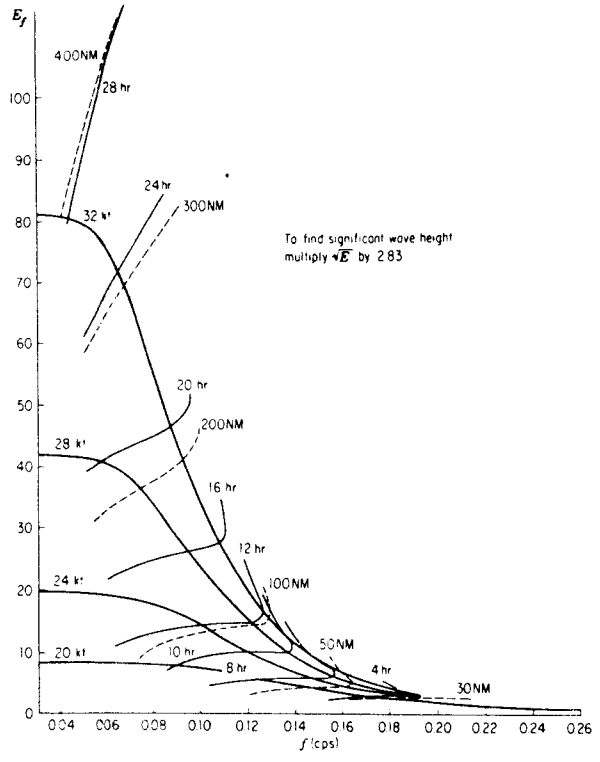


Fig. 17 Example of combined duration and fetch

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.

$$\text{slope} = \frac{\omega^2 - f^2}{N^2 - \omega^2} \tag{18}$$

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.

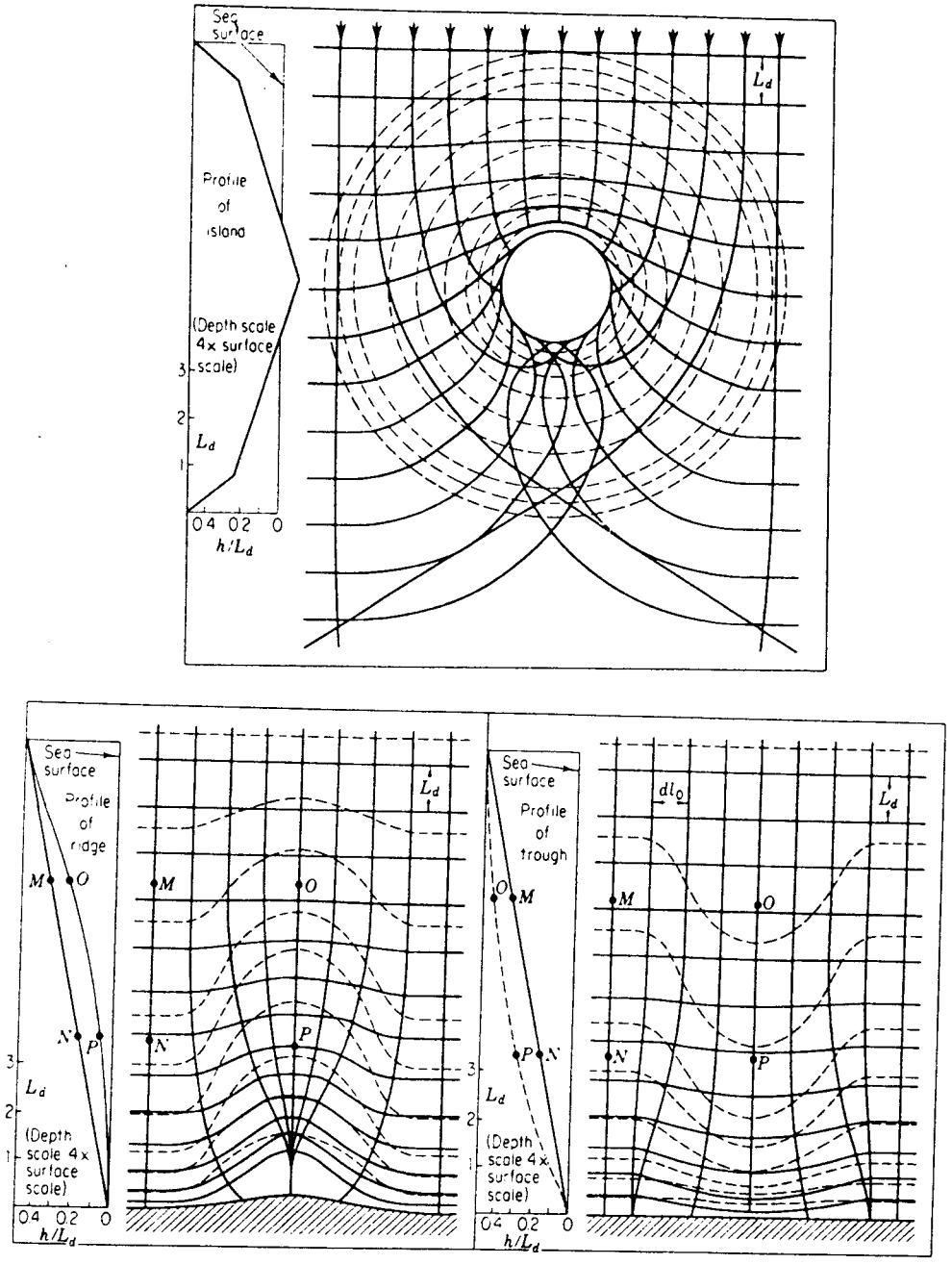


Fig. 18 Wave refraction around a circular island, over a ridge, and over a trough

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

less than the continental shelf slope, so there will be a region of increased motion associated with this beam. Note that the energy is now moving with tidal periods, that is, with 12-h periods, so even though the amplitude is large, the velocity is fairly small.

There are nonlinear effects which take an internal tidal oscillation generated over a bathymetric feature and convert it into a single or group of oscillations referred to as a soliton. These oscillations can have amplitudes of 50 m in 150-m water depths over shelf regions, with periods of 30 min. In the Strait of Gibraltar these amplitudes can exceed 150 m. Thus, when operating in these regions, care should be taken to realize that there are large oscillations occurring which may affect the submersible. Finally, these internal oscillations propagating from the open ocean into a submarine canyon can become concentrated—perhaps in resonance—and become fairly large and observable currents. The study of internal waves and their distribution in the ocean and deviation from the simple equilibrium theory is currently an active area of study. The recent interest in internal waves was triggered by the tragic loss of the submarine *USS Thresher* and the suggestion that the submarine, operating near its maximum depth, was carried by the "downdraft" from an internal wave off the coast on New England into water deep enough to cause a failure in the pressure hull.

3.5 Atmosphere and Ocean Interaction

The atmosphere is driven by the meridional variations of net heating of the earth. While some of the solar heat is absorbed directly, the air absorbs much more heat from (a) the long wave (or infrared) radiation from the ground and the sea, (b) latent heating associated with evaporation, and (c) sensible heating from the surface. Thus most of the energy which drives the atmosphere comes from below.

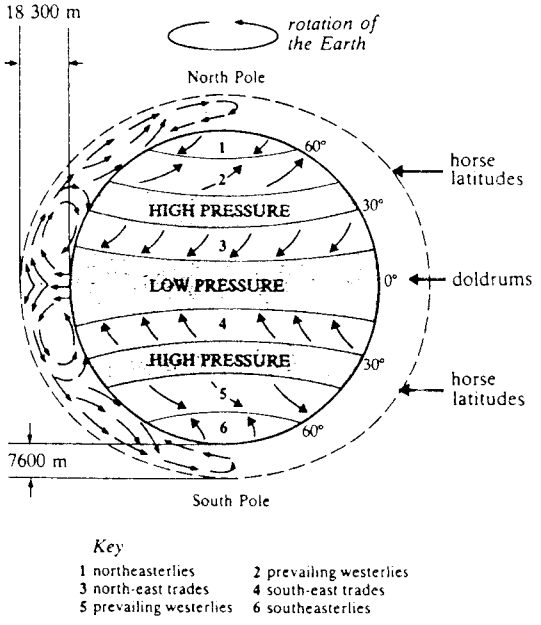


Fig. 19 Banded structure of average winds

The combination of convective heating, continuity, and earth rotation leads to the banded structure of the average winds (shown in Fig. 19). The trade wind regimes in the tropics are less variable than the other regimes. Much of the poleward heat transport in the atmosphere is accomplished through the Hadley Cell circulation, which consists of vertical convection in the equatorial region and sinking at about 30°N (or S) longitude. In the region of the westerlies (30° to 60°), the poleward transport of heat is accomplished to a large extent by the fluctuating winds associated with the high and low pressure cells which are seen to move from west to east on the weather maps. As these cells develop and grow, heat is transported poleward across latitude circles. As the cells diminish in intensity, the winds become more westerly (eastward moving) and heat is transported zonally.

3.5.1 Influence of Atmospheric Circulation on the Oceans

The atmospheric circulation patterns described above affect the ocean by forcing movement of surface currents through frictional effects and by changing surface temperature and salinity properties. The net effect of this interaction is a complex superposition of ocean currents which can be described in terms of their characteristic time and length scales as summarized in Table 3.

The wind stress, $\vec{\tau}_W$, exerted on the sea surface can be expressed in terms of the direction of the wind vector $\vec{W}$ and is proportional to the square of the wind speed $|\vec{W}|$ according to the relation

$$\vec{\tau}_W = \rho_a C_D |\vec{W}| \vec{W} \qquad (19)$$

where ρ_a is the air density and C_D is a drag coefficient.

Evaporation is relatively large in regions where dry warming air is descending in the region of 30°N (or S) (see Fig. 19). High evaporation regions correspond to regions of large loss of latent heat from the sea surface with a corresponding increase in salinity. Precipitation is high in the regions where wet cooling air is ascending in the region of the equator and to a lesser extent at 60°N (or S). Excess precipitation over evaporation tends to decrease sea surface salinity (see Fig. 4).

3.5.2 Frictional Effects: A Slight Digression

Frictional effects on ocean currents are usually confined to regions near the surface and bottom boundaries. Near the surface, the effect of the winds causes the movement of the surface layer through frictional effects; near the bottom, friction slows the

Table 3 Summary of the range of oceanic motions

Type	Time Scales (T)	Length Scales (L)	Amplitude (H)
Mid-Ocean "Mean" Flow	years	1000 km	cm/s
Mid-Ocean Eddy Currents	months	100 km	10 cm/s
Western Boundary Currents:			
Surface	—	100 km	200 cm/s
Deep	—	100 km	cm/s
Upwelling/Downwelling	days–seasonal	100 km	1 m/day
Continental Shelf Currents	days–months	100 km	10 cm/s
Estuarine Currents	days–seasonal	100 km	cm/s
Turbulence (Mixing)	s	cm–m	mm–cm/s
Capillary Waves	<0.1 s	<1 cm	<1 cm
Short Surface Gravity Waves	0.1–60 s	cm–km	m
Long Surface Gravity Waves	s–h	m–10's km	m
Internal Gravity Waves	min–h	m–10's km	m
Tides	day	1000 km	m
Internal Tides	day	10–100 km	10 m

ocean flow above it. In the former case, momentum is added to the ocean flow, and in the latter case momentum is extracted from it. Figure 20 shows a typical velocity profile in water (or air) flowing over a solid surface.

In general, the horizontal stress τ is proportional to the vertical gradient of the horizontal velocity through the relation

$$\tau = \mu \frac{\partial u}{\partial z} \tag{20}$$

where μ is the dynamic viscosity. Because the units of stress are force per unit area or momentum flux, stress can be thought of as a measure of the vertical transport of horizontal momentum.

The constant of proportionality between stress and velocity gradient depends upon the molecular properties of the fluid and is sometimes described in terms of the ratio μ/ρ; a quantity known as the molecular kinematic viscosity ν. Typical values for ν are about 0.02 cm^2/s. As we will see, however, molecular viscous effects are much less important than the so-called "eddy" viscous effects that arise in the presence of turbulent flow.

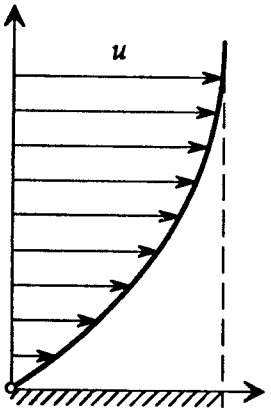


Fig. 20 Typical frictional "mean" velocity profile near a boundary

Eddy viscous effects can be explored by defining turbulent flow in terms of a mean and eddy-induced fluctuating part. A time series of the eastward current component is shown in Fig. 21, for which the time averaged mean value $\bar{u}$ is shown by the dashed line. The fluctuating part of u or u' remains after $\bar{u}$ is subtracted from u (u' has a zero mean by definition). Treating the northward (v) and upward (w) velocity components similarly permits us to define the fluctuating velocity components, v' and w' respectively.

The time-average of the many random turbulent eddy processes produces an effective eastward stress S_x on the fluid. This turbulence-induced stress is called the Reynolds stress and is related to the flow according to

$$S_x = -\rho \overline{u'w'} \tag{21}$$

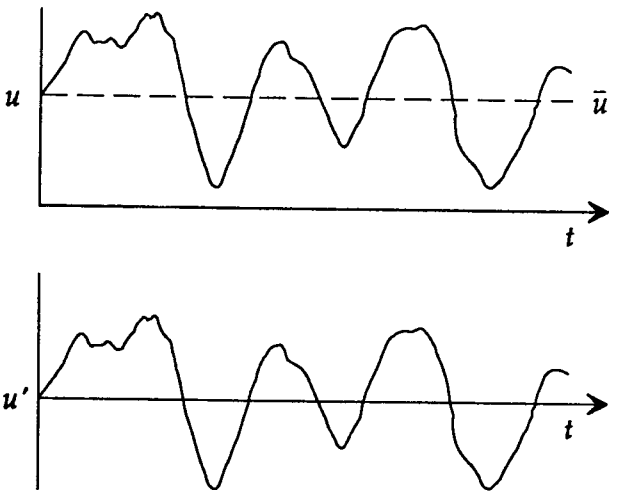


Fig. 21 Time series of the eastward current component

where the overbar signifies a time average. The Reynolds stress is also a measure of turbulent momentum transport. Therefore, by analogy to molecular stress, we can relate S_x to the vertical gradient of the mean velocity according to

$$S_x = A^e \frac{\partial \bar{u}}{\partial z} = -\rho \overline{u'w'} \tag{22}$$

where A^e is an "eddy" dynamic viscosity, which depends upon the generally highly variable flow in the ocean. The corresponding "eddy" kinematic viscosity is $\nu^e = A^e \rho^{-1}$. Similar relationships can be derived for stresses on other "faces" of the water column. Typical values of the vertical eddy kinematic viscosity range from 2 to 10^4 cm^2/s. Corresponding values of lateral kinematic eddy viscosity are generally larger (10 to 10^8 cm^2/s) because both the "thinness" and the stratification of the ocean inhibit vertical momentum transport relative to the horizontal momentum transport.

3.6 Wind Stress on the Sea Surface

We wish to express the stress on the sea surface as a function of the wind velocity at 15 m above the sea surface W_{15}. We can surmise from dimensional considerations that wind stress τ_W can be expressed as

$$\tau_W = \text{constant}\, \rho_a W_{15}^2 \tag{23}$$

But what is the constant? To answer that question we define friction velocity u^* such that $\tau = \rho_a u^{*2}$.

To determine u^* in terms of W_{15}, consider a turbulent surface layer of winds with a mean profile like that shown in Fig. 20 in which the time average of fluctuating winds decreases from a maximum at the top of the so-called "boundary layer" to near zero at the sea surface. It can be shown that the mean velocity profile $\bar{u}(z)$ in a constant stress turbulent boundary layer can be described in terms of a friction velocity u^* and Von Karmen's constant $k_0 = 0.4$ according to

$$\bar{u}(z) = \frac{u^*}{k_0} \ln \frac{z + z_0}{z_0} \tag{24}$$

where z_0 is a parameter related to ocean surface roughness. This form of the boundary layer mean velocity is the so-called Log Profile. This formulation permits us to compute u^* and z_0 from observations of $\bar{u}(z)$ so that the unknown constant in an expression for wind stress can be determined. Typically,

$$\tau_W = 2.6 \times 10^{-3} \rho_a W_{15}^2 \tag{25}$$

for density units g/cm^3 and wind units m/s.

The direct effects of wind stress on the sea surface are confined to a relatively thin (10 to 100 m) boundary layer in the upper ocean. We can explore these effects of wind stress by considering a simplified form of the horizontal equations of motion in which earth rotational and frictional forces are balanced. The flow that results from an imposed surface wind stress is called Ekman flow (graphically shown in Fig. 22). While the detailed form of Ekman flow has not been verified experimentally, the existence of the average flow to the right (northern hemisphere) of the wind stress is called the Ekman transport and has been observed. As we will see, large-scale Ekman transport in the relatively thin Ekman layer creates the conditions for large-scale deep ocean currents.

3.7 Wind-Driven Surface Currents

Most deep-ocean flow can be explained in terms of relatively simple physics. The pressure gradient forces (induced by sea level slopes) are balanced by the so-called Coriolis force (induced by earth rotation) in the direction perpendicular to the "geostrophic flow." The Coriolis force is to the right (left) of the geostrophic flow in the northern (southern) hemisphere.

The global wind stress field at mid latitudes creates an Ekman transport convergence in the vicinity of 30° in both the northern and southern Atlantic and Pacific Oceans. Thus there are associated north/south pressure differences. Because of the presence of continents, which block zonal ocean flow (more or less depending upon ocean basin), high pressure cells exist. The combined effects of the geostrophic flow around the pressure cells and the earth's rotation lead to clockwise (or counterclockwise) basin scale "wind-driven" gyres in the northern (or southern) hemisphere ocean basins (see Fig. 23). The gyroscopic interaction between the gyre motion

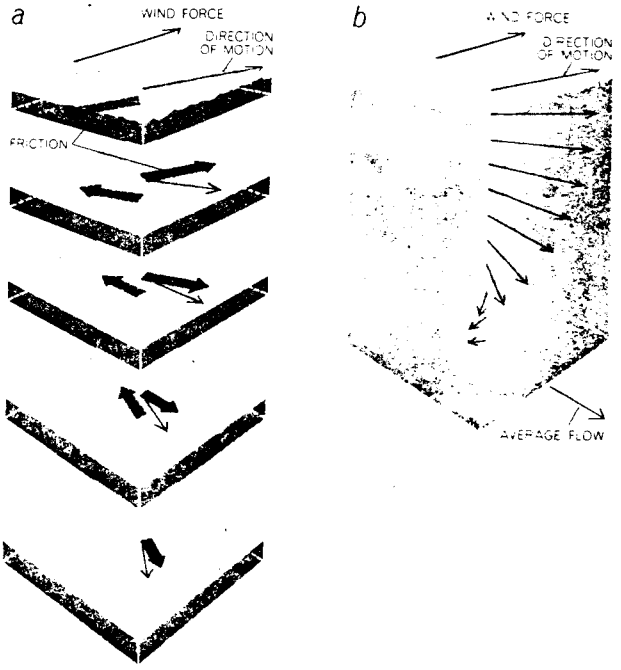


Fig. 22 Graphic description of the Ekman flow

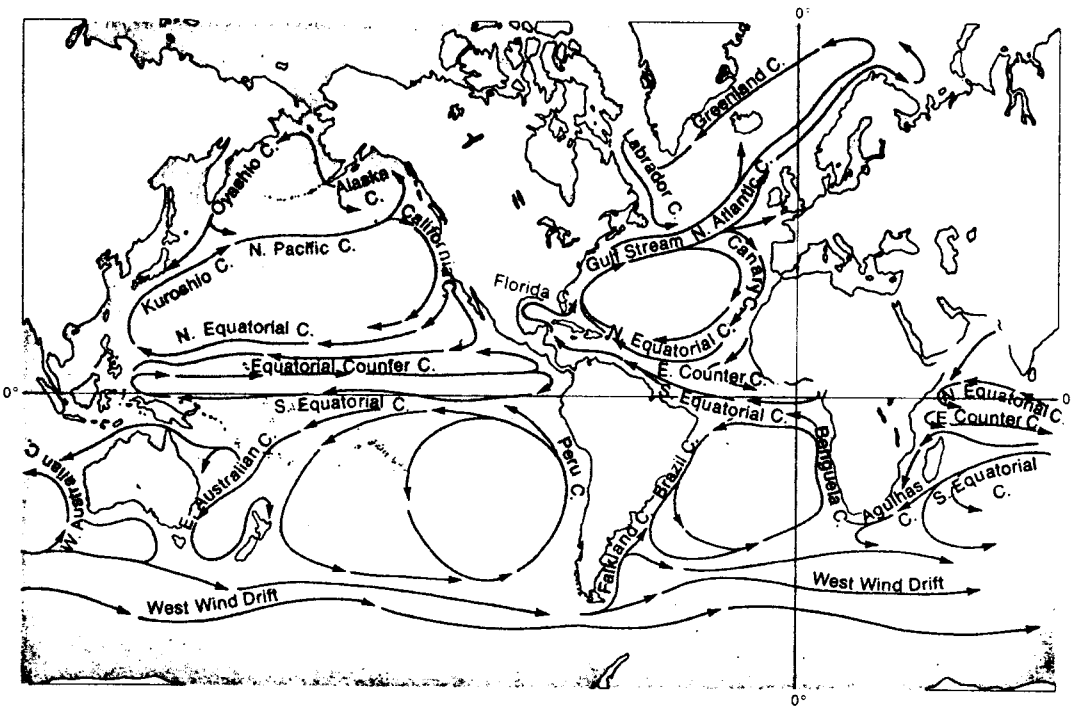


Fig. 23 Basin scale "wind-driven" gyres in the hemispherical ocean basins

and earth rotation leads to intensification of gyre currents along the western boundary of all the ocean basins, exceeding 200 cm/s in the Gulf Stream of North Atlantic and in the Kuroshio of the North Pacific. The total transport of the Gulf Stream has been estimated to be in the range of 50 to 90 × 10^6 m^3/s, with that of the Kuroshio being somewhat less.

Despite the greater intensity of winds, but for reasons which are poorly understood, the intensity of the surface western boundary currents in the southern hemisphere is less than that for either the Gulf Stream and Kuroshio. The Brazil current in the western South Atlantic has typical speeds of 50 to 100 cm/s and penetrates to depths of only about 400 m, in contrast to 800 m for the Gulf Stream. The reduced transport of the Brazil Current may be related to the presence of deeper western boundary undercurrents and their role in transporting heat poleward. The reduced intensity of the Australian current in the western Pacific and the Agulhas Current in the western Indian Ocean may be related to the lack of complete continental blockage of zonal flow.

Along the eastern boundaries, the equatorward flow area of the wind-driven gyre is generally less intense and less organized than the western boundary currents. The exception is the Benguela Current, which is an apparent extension of the Agulhas Current in the South Atlantic. Figure 24 shows an example of an eastern ocean boundary current—the California current—in terms of the dynamic topography of the sea surface (that is, the oceanic pressure field). This is a "snapshot" of that current regime, which is obviously more diffuse than its oceanic partner to the west, the Kuroshio.

In the equatorial regions, the significant trade winds can drive surface currents in the zonal circulation patterns shown in Fig. 23, patterns that exist in all major oceans. The wind-driven westward flow (Coriolis effects are negligible) in a zone bracketing the equator is blocked by the presence of land masses. A resulting eastward pressure gradient force drives a return equatorial undercurrent jet along the equator and a seasonal countercurrent in the doldrum region. These countercurrents, while primarily zonal, are also the very important connection between the major ocean basins.

The circulation at poleward latitudes greater than 50° differs greatly in the different oceans because of differences in the geography of the northern and southern hemisphere ocean basins. The North Atlantic Ocean effectively ends at the sills at about 60°N and is weakly associated with the shallower Arctic Ocean, which has a circulation of its own. In the South Atlantic there is a zonal ocean which circles the entire Antarctic continent. The winds are important in driving the broad Circumpolar Current (Fig. 25). Density adjustments in the interior produce pressure gradient forces which balance the Coriolis forces to produce a current profile which diminishes

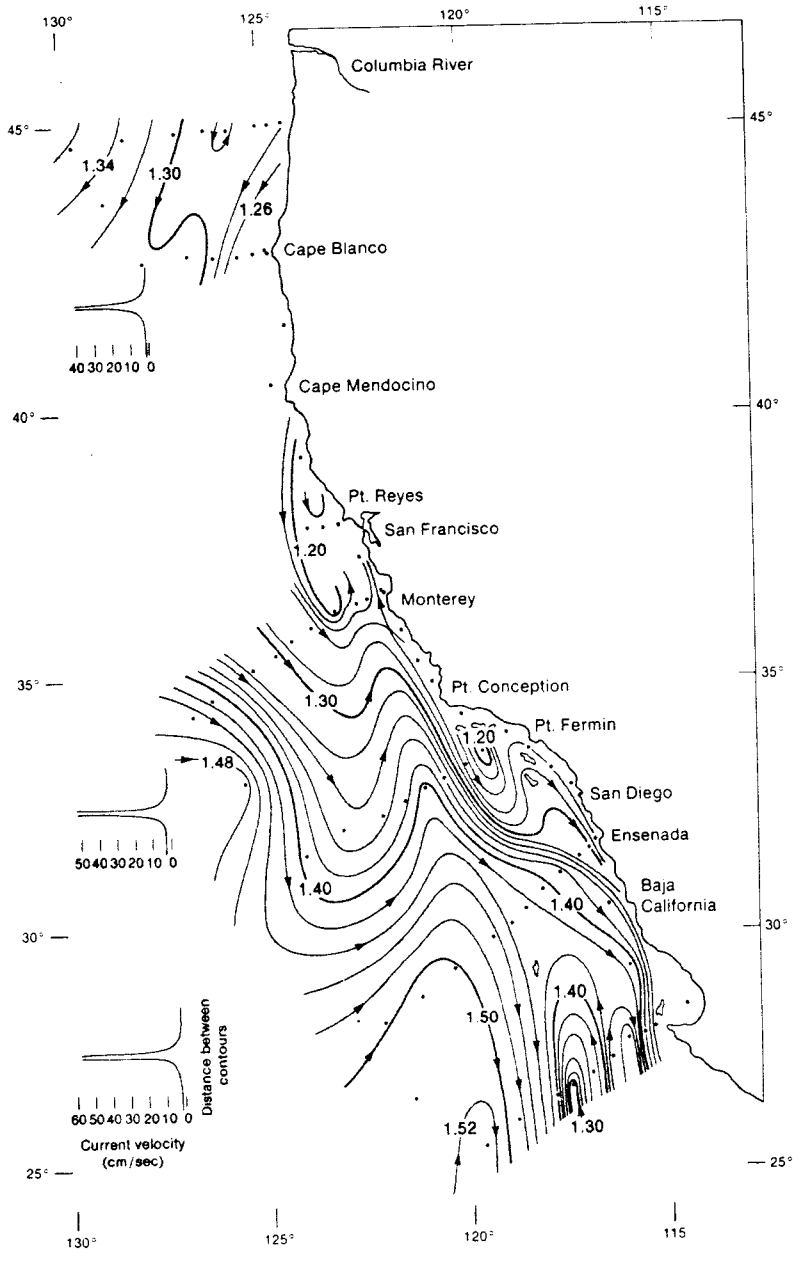


Fig. 24 Expression of the California current in terms of the dynamic topography of the sea surface. The actual regime is more diffuse

from about 15 cm/s at the surface to near zero at 3 km depth. So although Circumpolar Current speeds are considerably less than Gulf Stream speeds, the transports associated with the Circumpolar Current are 130 to 190 x 10^6 m^3/s and larger than Gulf Stream transports. As such, this current has the largest transport of any in the world's oceans and plays a crucial role in controlling the deep ocean circulation.

Before leaving the discussion of the wind-driven surface currents, it is important to point out that the currents discussed thus far actually change considerably over periods of months. For example, the Gulf Stream is known to become unstable, meander, and actually shed eddies (Fig. 26) containing water from both the warm Sargasso Sea and the cold North Atlantic. This process leads to an effective poleward heat transport. Perhaps more important to the reader is the fact that these eddy-like features, which have Gulf Stream–like ocean currents, intensify (Fig. 27*a*).

Warm-core rings with clockwise circulation are observed to propagate slowly southwestward along

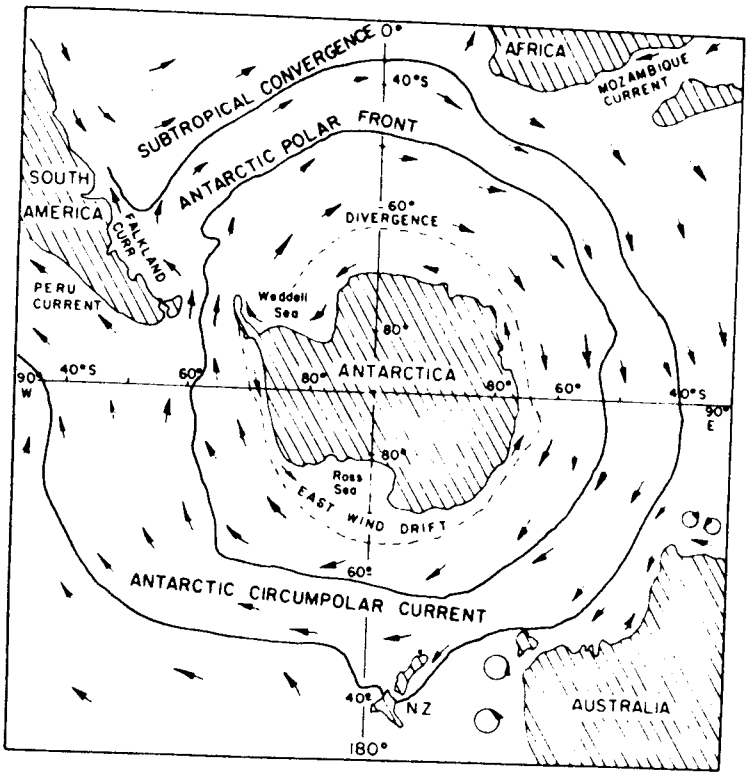


Fig. 25 Circumpolar current

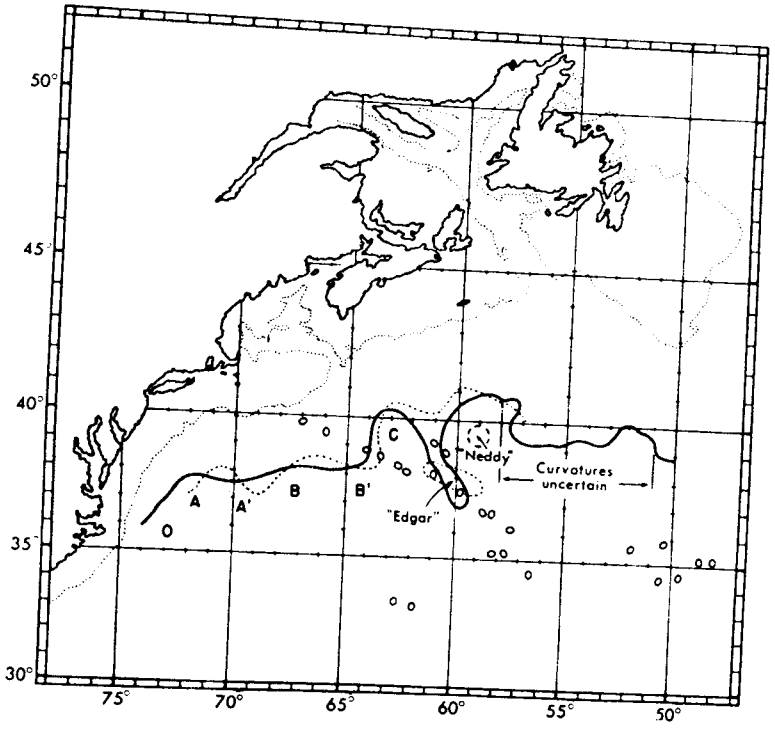


Fig. 26 Changes occurring in the path of the Gulf Stream over periods of months

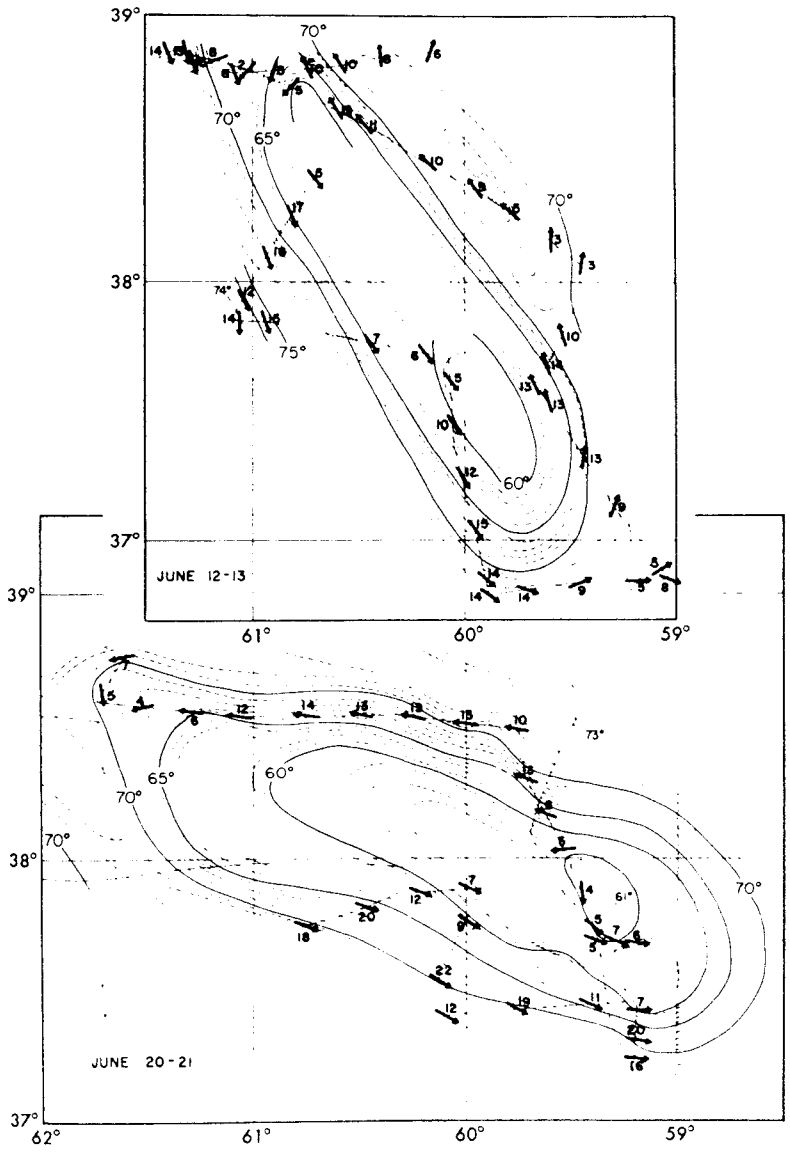


Fig. 27(*a*) Intensification of eddies shed by the Gulf Stream

the continental margin on the northern side of the Gulf Stream (Fig. 27*b*). Cold-core rings south of the Gulf Stream also propagate southwestward to be eventually entrained by the Gulf Stream in the region of Cape Hatteras. Satellite infrared imagery is very helpful in tracking these anomalously warm and cold eddies.

In reality, all of the upper oceans of the world are filled with eddy-like current structures embedded in the large-scale, wind-driven gyres described above. Typical currents in these eddies are about 10 to 20 cm/s, in contrast to the 1 cm/s gyre mean flow, and have lateral extents of about 100 km. Such eddies have been observed to propagate through a region in a period of months. The role of these eddies in the overall transport of heat and other constituents is only just beginning to be understood.

The circulation in the equatorial regions is likewise unsteady. The "El Niño" is now known to be part of the larger scale Southern Oscillation in the Pacific (the whole process is referred to as ENSO) in which the trade winds moderate, allowing the equatorial flow propagation from west to east. This anomalous flow of warm equatorial water has devastating effects on the fisheries along the western coast of South America.

3.8 Deep Ocean Currents

Superimposed on the wind-driven circulation of the upper ocean is the less intense deep flow in the

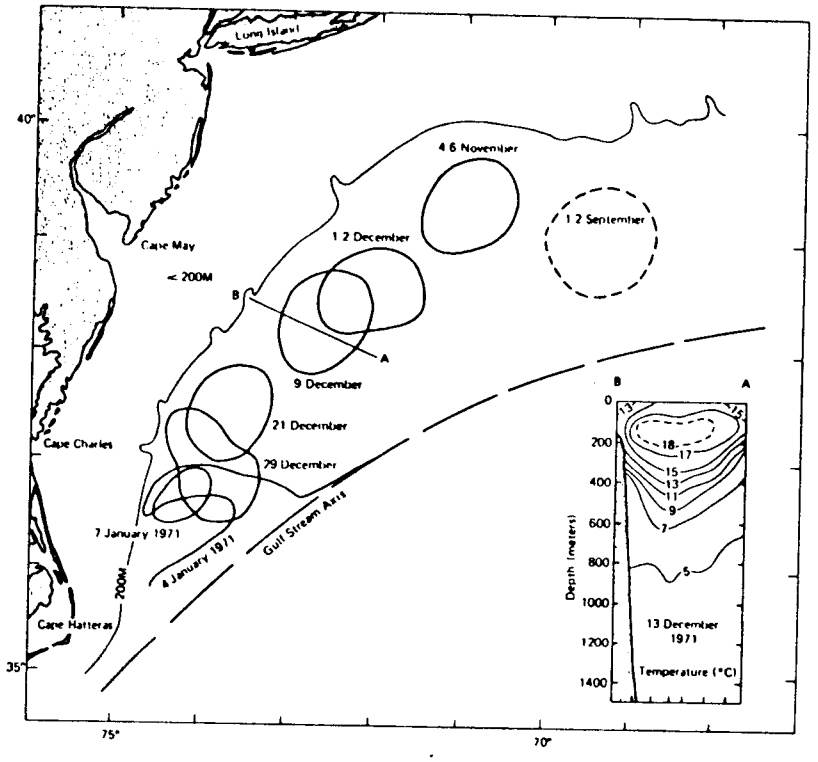


Fig. 27(*b*) Southwestward propagation of eddies from the Gulf Stream

ocean which is driven primarily by a sinking of cold water in the polar regions and thus referred to as thermohaline circulation. It is believed that the deep water in the oceans returns to the surface by a complicated and poorly understood vertical advection process. The net effect of this vertical return

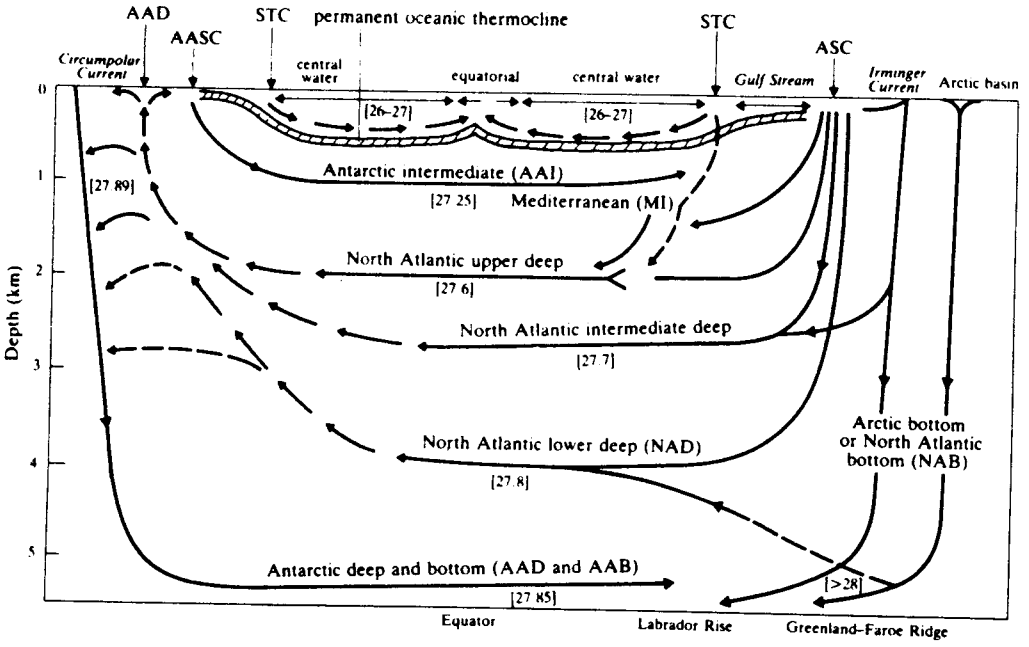


Fig. 28 Deep ocean currents. Vertical arrows at top designate wind-driven convergences and divergences—AAD is the Antarctic divergence, AASC is the Antarctic subpolar convergence, STC is the subtropical convergence, and ASC is the Arctic subpolar convergence

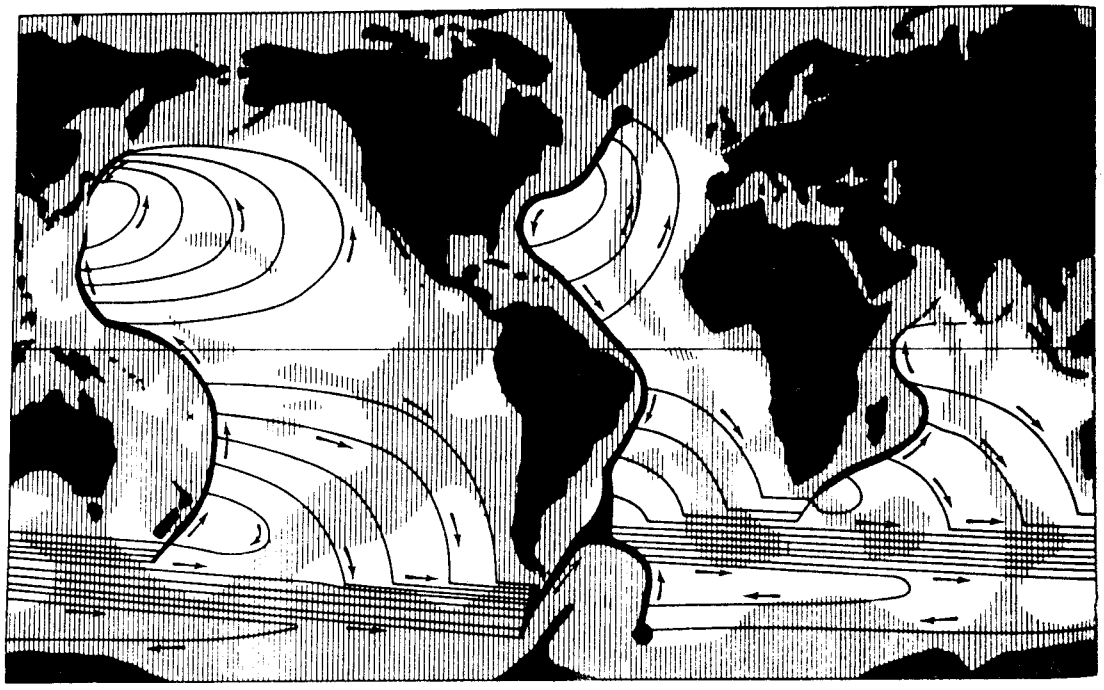


Fig. 29 Deep ocean circulation patterns

(estimated at centimeters per day!) is to maintain the depth of the main pycnocline against downward heat diffusion in the upper ocean (Fig. 28).

The first attempts at developing a theory of thermohaline circulation were motivated by an interest in explaining Wust's discovery of a much less intense wind-driven circulation in the subtropical oceans of the southern hemisphere. This observation seemed to be a paradox because the winds are less affected by continental friction and thus are stronger in the southern hemisphere. Stommel hypothesized that this observation could only be explained by a combined theory of thermohaline and wind-driven circulation.

The principal elements of a theory of thermohaline circulation developed by Stommel about 1957 included a sinking in the polar regions which is balanced by upward advection in the mid-ocean regions, which was connected to the vertical return flow through a set of "relatively" intense western boundary currents and weak flow of water in the interior deep ocean. Despite its weakness, Stommel's model leads to the deep circulation patterns shown in Fig. 29. The deep western boundary currents in the Atlantic Ocean is shown in Fig. 30.

Wust has used water property distributions or T-S relationships (Subsection 2.4) to infer the following transports of these deep water masses. He finds in the western basin of the South Atlantic:

Antarctic Bottom Water: 2×10^6 m^3/s (northward),
North Atlantic Deep Water: 27×10^6 m^3/s (southward), and
Antarctic Intermediate Water: 7×10^6 m^3/s (northward).

Thus, an overall picture of combined wind-driven and thermohaline flow can be described. In the northern Atlantic, the most intense Gulf Stream is opposed by the deep countercurrent of Atlantic deep and bottom water, while the less intense Kuroshio is aided by deep underwater currents. The Brazil current is aided by the relatively strong undercurrent of North Atlantic Deep Water.

3.9 Shelf Currents

The relative shallowness of the continental shelf and coastal regions of the world's oceans leads to a distinct current regime in which short term fluctuations are much larger than longer term mean flows. For example, tidal currents, which are relatively weak (5 cm/s) in the ocean, are relatively intensified and play a more important role in the mixing of shelf and coastal water masses. In general, the shallower water on the shelf responds more readily to wind forcing and exhibits important variability on the two- to ten-day time scales of the mid-latitude synoptic weather systems which pass through the region. Density fronts caused in part by the proximity of the

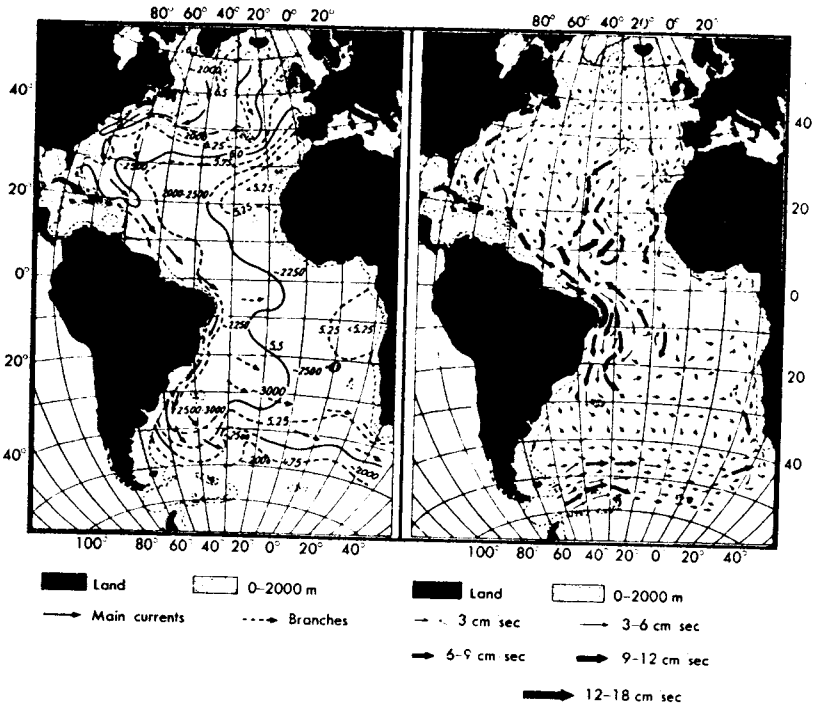


Fig. 30 (*Left*) Flow of mid-Atlantic deep water and (*right*) current flow at 2000-m depth based on dynamical calculations

coastal ocean to significant fresh water river sources are associated with significant intensification of shelf and coastal currents.

The Georges Bank region produces relatively intense tidal currents because of the shallowness of the Bank and the large volume of water required to fill the Gulf of Maine on each tidal cycle. In this region the clockwise rotary currents (Fig. 31) are largest in the on/off Bank direction leading to the elliptical current vector pattern. The intensity, shape, and rotation sense of astronomically forced tidal current ellipses are predictable at a particular location. Vertical profiles of tidal currents at this location (Fig. 31) are characterized by vertical shear which intensifies near the bottom, where frictional effects are predominant. Currents associated with internal tides (as described previously) with more complex vertical structures are occasionally superimposed on astronomically forced tidal currents.

Wind-forced shelf currents change over longer time scales than tidal currents and thus are usually more effective in transporting nutrients, biota, pollutants, and other constituents on the shelf. Wind-forced shelf fluctuations are usually most intense (typically 10 to 50 cm/s) in the along-shelf direction, with the maximum currents occurring at mid-shelf.

The presence of significant across-shelf density changes, such as those found along the East Coast of North America, leads to a modified current structure in which maximum along-shelf currents are concentrated in the density front along the outer shelf. These currents change over longer time scales in response to the more slowly changing density structures. Occasionally, deeper ocean currents in the form of Gulf Stream warm core rings penetrate onto the shelf and disturb the "normal" distribution of currents found there.

On many continental shelves, including the East Coast of North America, there is a more or less steady along-shelf flow superimposed on the fluctuating tidal, density-induced and wind-driven current field. Typical velocities of these steady flows are usually about 5 to 10 cm/s. Thus while astronomical wind forcing plays an important role in shelf currents, their superimposition leads to complex and, as yet, unpredictable patterns crudely described by the characteristics discussed above.

4. The Geography of the World's Ocean Basins

4.1 Introduction

Plate tectonic theory has been developed in recent years to explain the Earth's crustal motions of the past 250 million years which are responsible for the general form of the present day ocean basins. These crustal motions—with the major new crust being formed along mid-ocean ridges—and sedimentation

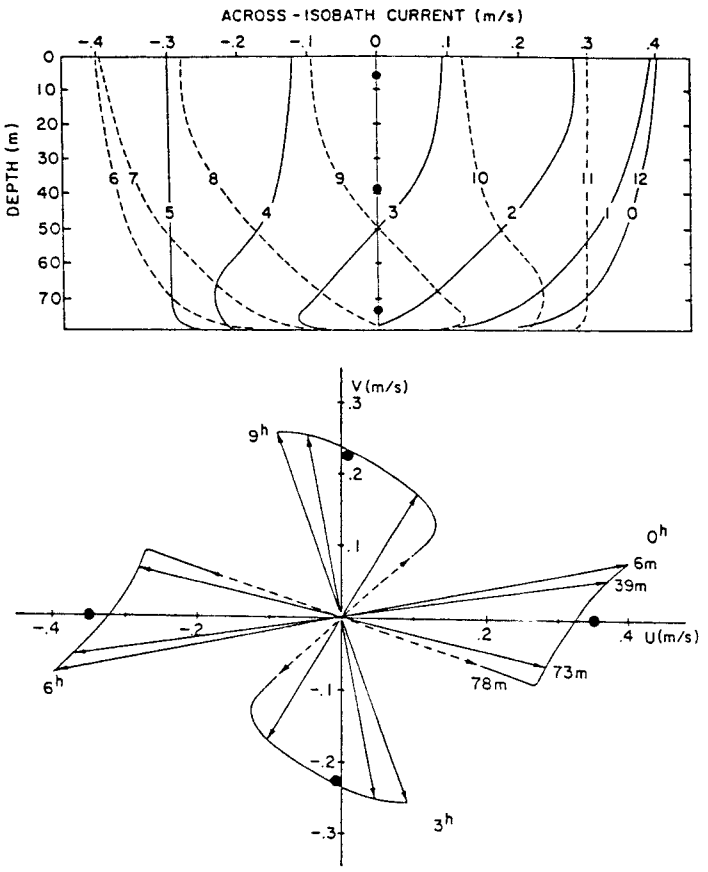


Fig. 31 (***Top***) Clockwise rotary currents in the Georges Bank region. (***Bottom***) Vertical profiles of tidal current in the region

processes have combined to form most of the seafloor features we know today. A planview of the principal seafloor provinces in the North Atlantic is shown in Fig. 32, which also identifies some of the important features in a vertically exaggerated view of the A–B transect. More detailed pictures of the continental margin region, where most of the world's sediments are found, and deep ocean trenches, where ocean crust is subducted and destroyed, are shown in Fig. 33. While the details differ from ocean basin to ocean basin, many of the typical features are identified. The distribution of these major geological features in the world's oceans are presented in Fig. 34.

A rain of sediment particles from the water column coats the seafloor very slowly (over millions of years) and in concert with the ocean currents modifies the ocean seafloor during its typical 100- to 200-million-year lifetime. This sediment rain is composed of four major components which are classified according to their origin and size. The four principal categories are (a) lithogenous sediments, which are derived from the weathering of continental silicate rocks; (b) biogenous sediments, which are the calcareous or siliceous skeletal remains of oceanic plankton (radiolaria and diatoms); (c) cosmogeneous sediments, which consist of the remains of meteorites and other forms of cosmic dust; and (d) hydrogenous sediments, which are formed from the precipitates of dissolved weathering products.

These sediments are transported laterally by ocean currents and, except for hydrogenous sediments, are deposited in a particular region of the seafloor according to the location of their source and settling speed. Because most sediments are derived from terrestrial weathering processes, 75 percent of the ocean sediments are found in the region of the continental margin near their river sources. Fine clay-size (1 μ) sediment particles have settling speeds of about 2.5×10^{-4} cm/s in contrast to course sand-size (100 μ) sediment particles which settle 10 000 times faster. Thus, the coarser sediments are found nearer the shore, whereas the finer sediments are found in the regions of the continental slope, rise, and deep ocean.

The sediments in the deep ocean basins are formed by mixtures of the four major sediment types

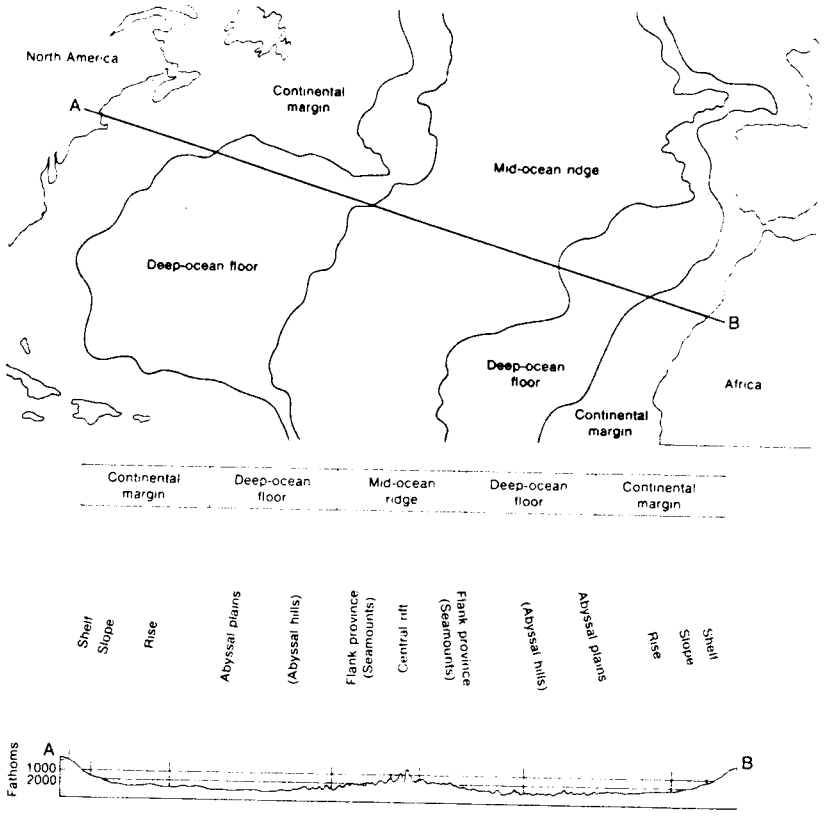


Fig. 32 Planview of the principal seafloor provinces in the North Atlantic Ocean

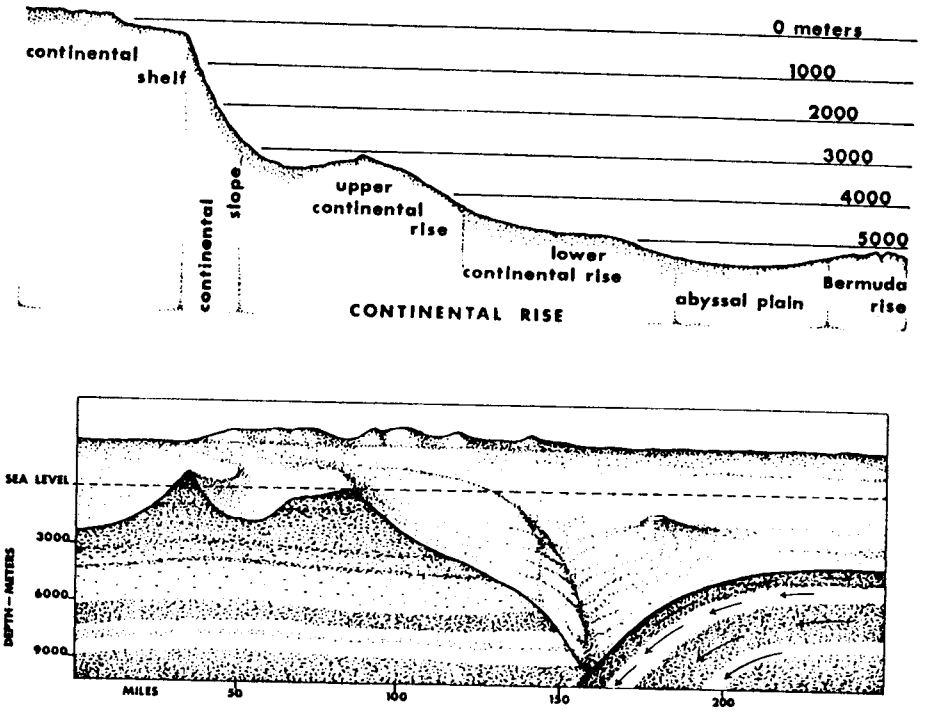


Fig. 33 (*Top*) Continental margin region and (*bottom*) deep ocean trenches of the North Atlantic Ocean seafloor

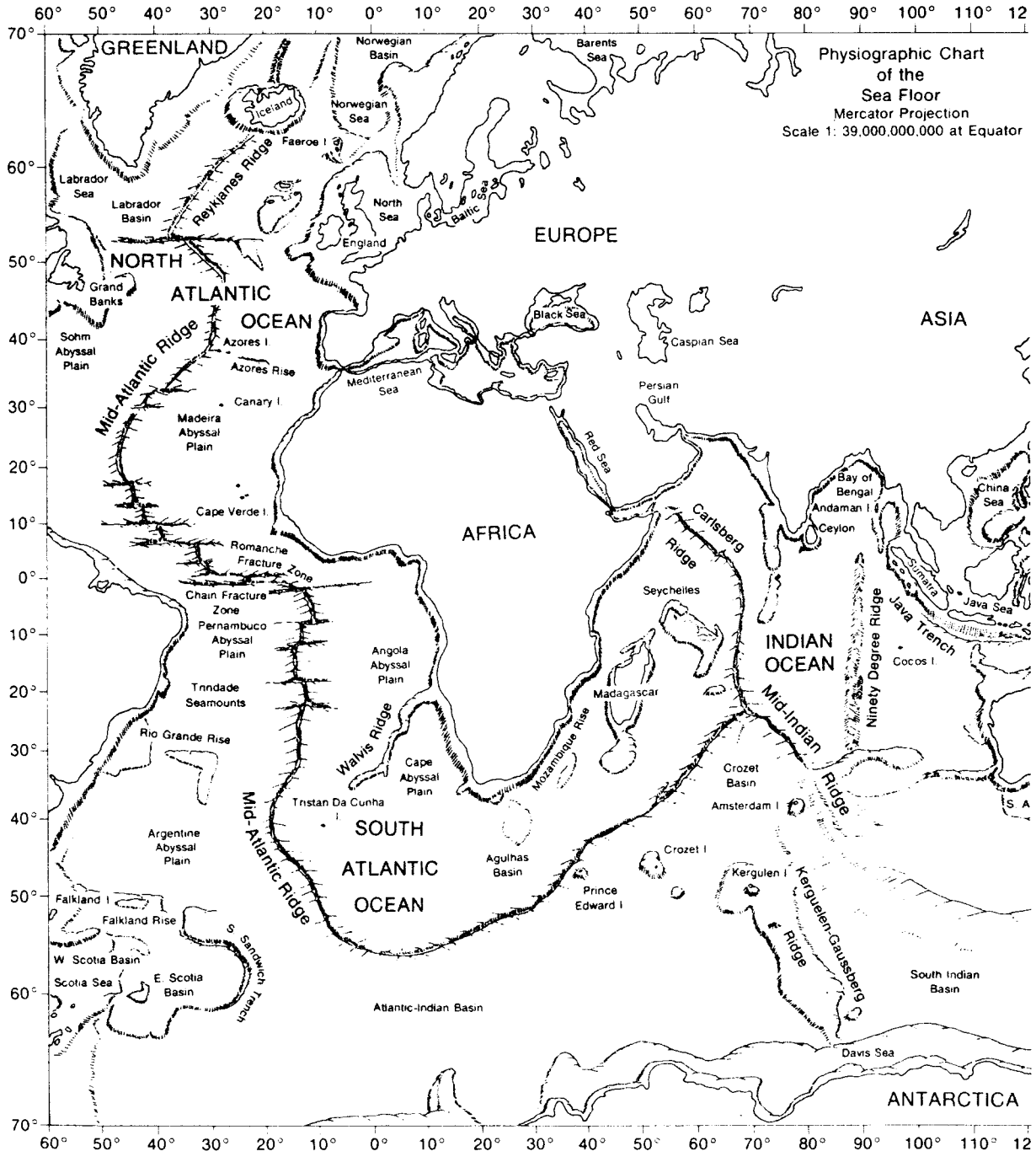


Fig. 34 Distribution of major geological features in the world's oceans

in different proportions according to many factors, including the proximity to the continental margin, the rates of biological productivity, the depth, and the age of the oceanic crust at a particular location. A crude map of the distribution of sediment types in different parts of the world's oceans is presented in Fig. 35.

4.2 Continental Margin

The continental shelves are the relatively shallow regions adjacent to the continents which constitute 18 percent of the land area and 7.5 percent of the ocean-covered surface. The continental shelf is an important transition region between the coast and

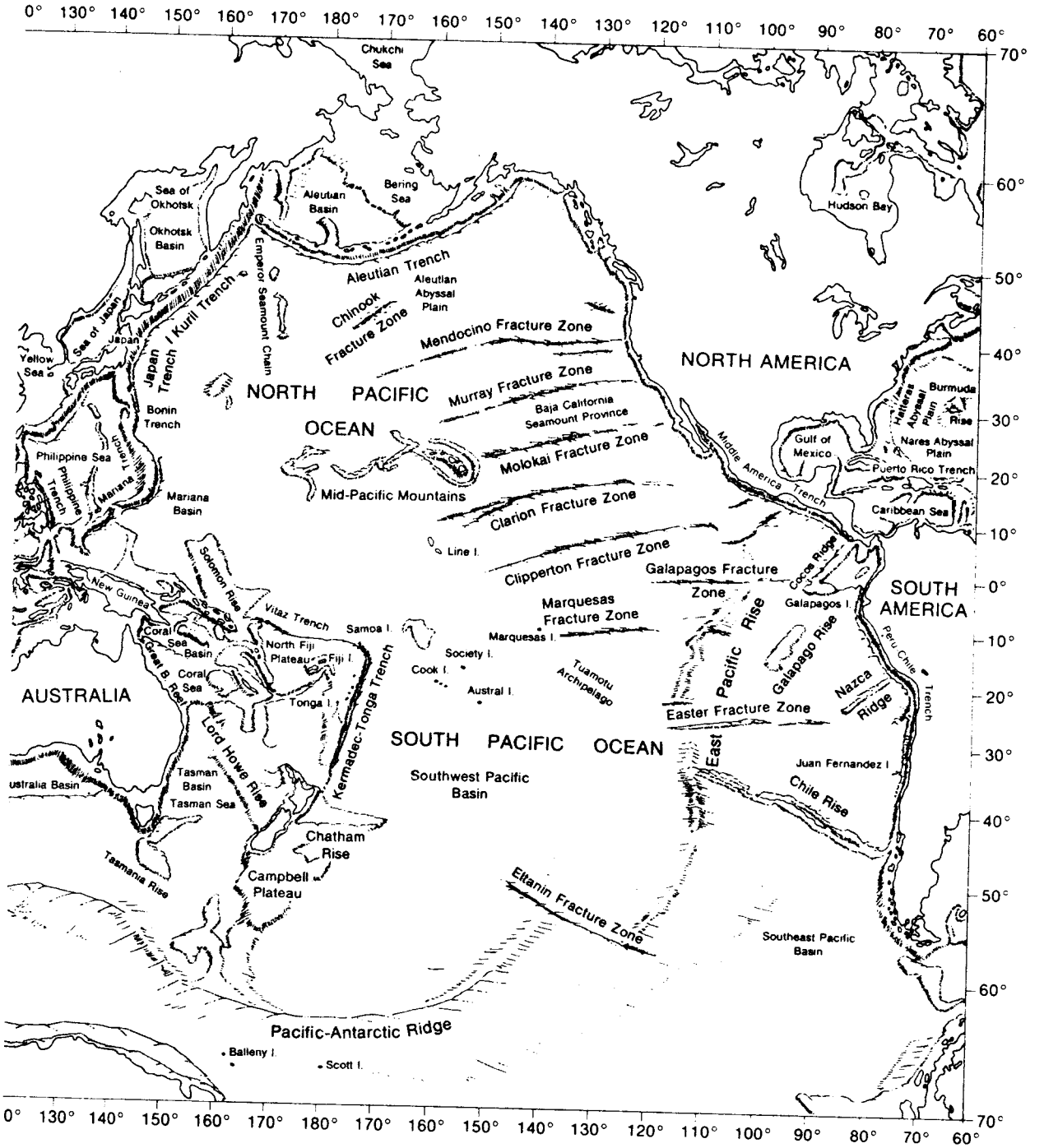


Fig. 34 ***Continued***

the relatively steeper continental slope and is a zone where many fisheries and mineral resources are accessible. Geologically speaking, the shelf regions should really be considered part of the land mass because of the thickness of the crust in this region. The definitions of the shelf vary, but are related to the slope of the bottom, which is about 1/500. For typical continental shelves, the depth range is from 20 to 500 m, with the average for the world's oceans being about 130 m. The width range of the shelves is from 0 to 1500 km, with an average of 78 km.

Continental shelves are classified according to the rock type that underlies the sediments that form the seafloor on the shelf. The principal rock types include

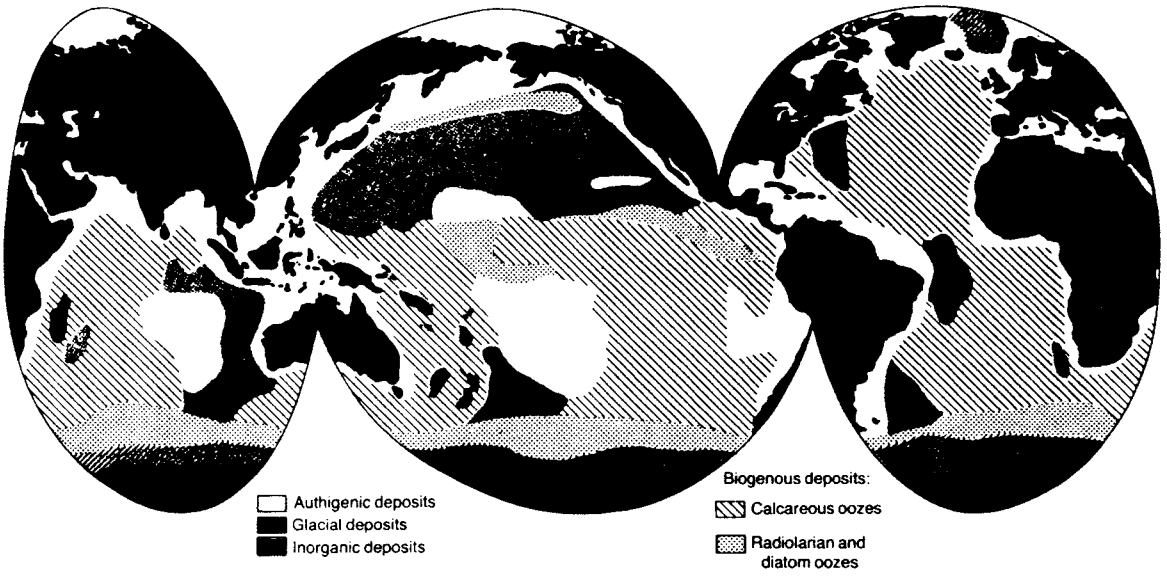


Fig. 35 Distribution of sediment types in different parts of the world's oceans. Hydrogenous sediments comprise autogenic deposits; lithogenous and cosmogeneous sediments comprise inorganic deposits

(a) sedimentary rocks, which are consolidated forms of layers of the unconsolidated sediment which is composed of particulate organic and inorganic matter; (b) igneous rocks, which are the solidified form of magma—the mobile form of rock material generated within the earth and capable of intrusion and extrusion; and (c) metamorphic rocks, which are modified forms of rocks that are transformed in response to changes in the temperature, pressure, or chemical conditions below the earth's surface.

The seafloors of a large majority of the continental shelves mark the surface of long thick prisms of sedimentary strata. Many of these sedimentary prisms are held in place against the continent by long narrow fault rocks called tectonic dams. The continental shelves which rim much of the Pacific, are of this type. A dam of this type existed off the East Coast of the United States between 270 and 60 million years ago. In time, sediments from the land filled the basin shoreward of the dam and eventually overflowed to form a continental slope which is held in place only by the angle of repose (or rest) of the sediments. Diapir dams (uplifted salt domes) and coral reef dams also help to form continental shelves in other places in the world—the Gulf of Mexico, for example. Shelf regions which are underlain by igneous or metamorphic rock are found in some areas of the world. The Gulf of Maine is one of those regions where glacial action, which followed the formation of the shelf, has swept away the sedimentary layers. The detritus—rock debris left after the melting of the latest glaciation about 20 000 years ago—forms what we know today as Cape Cod and Georges Bank.

Carbon-14 dating methods applied to shelf sediments indicate that dramatic changes occurred on the continental shelves during the one million years of the Pleistocene, when sea level fluctuated by as much as 150 m in response to four major glaciations. The surficial sediments found today were laid down after the latest of these glaciations. The only regions which exhibit a graduated distribution of sediment size are those less than about 20 m deep. Here sediment transport is dominated by the effects of waves. The parts of the shelf just seaward of 20 m do not receive constant supplies of new sand. These deeper parts of the shelf are bypassed by the contemporary silts and clays which remain in suspension en route to the deeper, quieter waters off the edge of the shelf. The sediments in the deeper regions of wider shelves are primarily composed of relic sediments. That is, sediments which were laid down during the times that sea level was much lower than its present level. Imagine a 75-km-wide lowland area covered by wildlife and forests and crossed by rivers. The coarse sands found here were probably laid down by the rivers which flowed over the then coastal plain.

The fine-grained sediments were probably carried to the sea which lay beyond the edge of the continental shelf or what we now call the continental shelf break. This marks the transition in slope from the relatively flat continental shelf to the much steeper continental slope (1/50). The continental slope is the effective boundary for the circulation in the deep ocean. Its seafloor is marked by numerous V-shaped canyons leading from the shelf down the slope to the deep ocean. These submarine canyons have only slightly modified the shelf regions along the East

Coast of North America. However, the heads of these canyons are situated very close to the shore region on the very narrow shelf along the West Coast. These West Coast canyons apparently play an important role in diverting the sand from the beaches into the deep ocean.

There is accumulating evidence that sediment-laden water forms underwater avalanches called turbidity currents. These currents are responsible for many of the submarine canyons, some of which have dimensions which are comparable to those of the Grand Canyon. There is a well-documented case of a turbidity current occurrence, on November 19, 1929, when a severe earthquake shook the Grand Banks in the Atlantic south of Newfoundland. It so happened that this area had the highest concentration of submarine cables in the world and, as expected, many of the cables within 100 km of the epicenter of the quake broke immediately after the earthquake struck. However, it was also found that during the following 13 hours, other cables were broken, with each successive break being downslope from the preceding break, until the last break occurred in the deep ocean basin more than 460 km from the earthquake center. Apparently the quake triggered a submarine sediment avalanche. The precise time of the break of each of the cables was recorded automatically so that the speed of the suspected turbidity current could be calculated. The speed ranged from 80 km/h on the steep slopes to 24 km/h on the plains. Engineers reported difficulty in finding the cable ends, which were presumably buried. The conclusive evidence was the fact that the thickness of the silt layer at the deepest cable break was consistent with what had been predicted.

The continental rise is composed of overlapping delta-like regions of sediments at the base of the numerous submarine canyons. The sediment deposits called turbidites are characterized by layers of graduated sediment sizes. Apparently, mixtures of the different sediments, which form the turbidity currents, settle out in layers according to their particle settling speed.

4.3 The Deep Ocean Floor

The deep-sea bottom is the most extensive area of the ocean seafloor. Depths from 3000 to 6000 m are found over 76 percent of the ocean basins, with about 1 percent deeper than that. There is a large variation in bottom topography in the ocean bottom regions which were once regarded as uniformly smooth. With the advent of submarine telegraph cable laying and the invention of acoustic depth measuring methods, it became clear that much of the seafloor was very rugged. In fact, there are actually regions of both smooth abyssal plains and extremely rugged mountainous regions.

The large abyssal plains of the seafloor are in many cases extremely smooth—possibly more so than the plains areas on land. Stretches of these plains are smooth (within 2 m) over distances of 100 km. On the continental shelf and slope, most of the bottom material comes directly from the land and tends to be relatively coarse. The material on the seafloor tends to be more finely divided than that on the continental shelf and slope. Much of the material is pelagic in character, that is, it has been formed in the sea itself. Two major deep-sea sediments are the inorganic "red" clays and the organic oozes. The former has less than 30 percent organic material and is mainly mineral in content. It contains fine material from land which may have travelled long distances before settling out on the bottom, volcanic material, and other inorganic remains. The oozes are over 30 percent organic in origin and were originally living organisms in the sea above. Calcareous oozes have a high percentage of calcium carbonate from the shells of animal plankton (*Globigerina*), while siliceous oozes have a high proportion of silica from the shells of planktonic plants (radiolaria and diatoms) and animals. The actual distribution of these sediment types plus hydrogenous types (authigenic, for example) in the world's deep oceans are presented in Fig. 35.

Sedimentation rates have slowed from about 60 cm/1000 yr (20 000 years ago) to about 1 cm/1000 yr in the last few thousand years. Much information about the history of the oceans and the earth itself is stored in the strata of sediments which are deposited on the seafloor. A "core" of sediment may represent 10 000 to 10 million years per meter of length. The different layers may indicate periods in the earth's history when volcanic activity was particularly high. Other layers may have entombed the remains of warm water animals, while other layers are composed of the remains of cold water animals—thus indicating that the water which flowed above the seafloor had changed between the two periods.

Moving from west to east in the North Atlantic, the smooth abyssal plains slowly become more hilly approaching the Mid-Atlantic Ridge. Ocean basement crust pushes a few peaks above the sediments in the west. Eventually, the only sediments found are concentrated in valleys between the peaks. No sediments appear to exist near the Mid-Atlantic Ridge. Mid-ocean ridges form the most extensive topographic feature on the Earth (Fig. 34). These ridges rise from 1 to 3 km above the seafloor and appear to be more or less continuous. However, more complete surveys have shown that the ridge systems themselves form a zig-zag line consisting of many short segments. These segments can be identified from the rifts, which stop abruptly. Fracture zones called transform faults connect these different ridge segments and run nearly perpendicular to the main ridge system.

The ocean ridge system is an area of volcanic activity and new mountain formation. New mountain formation produces the rugged terrain which is in sharp contrast to that of the abyssal plains. This active earth-forming process can be clearly seen in Iceland, a part of the Mid-Atlantic Ridge which has grown so quickly it has emerged above sea level. A 45-km-wide central rift valley cuts the island parallel to the ridge. The sides of the rift contain active step-like faults. There are other step faults on the floor of the rift floor, which is otherwise dominated by a large number of longitudinal fissures. Fluid lava wells up in these fissures and either buries the surrounding mountains, valleys, and faults or forms volcanoes. Hundreds of volcanoes dot the floor of the rift; 30 are currently active.

The lava flow in the regions of the ocean ridges lays down a layer of volcanic crust on top of the oceanic crust. This layer extends from the ridge area out across the ocean basin. The volcanoes extend above the sediments and form what are known as sea mounts which are found throughout the ocean floor. The existence of sea mounts has been known for some time; however, their origin and distribution have not been well understood until recently.

4.4 Ocean Trenches

Another outstanding feature of the ocean floor is the series of very deep trenches (Figs. 33, 34) which mark some of the boundaries between the oceans and the continental margins. These regions constitute the 1 percent of the ocean which lies below 6000 m. These trenches can be as long as 3000 km and as deep as 11 km. Considering that the average depth of the ocean is less than 4000 m, they are a very prominent oceanic feature. The characteristic V-shape has the steeper slope towards the islands which are of volcanic origin. Most of the deeper trenches exhibit this V-cross section, some being slightly flatter at the bottom. Other shallow trenches are more U-shaped with large flat bottoms, as if they have been partially filled with sediments.

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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-

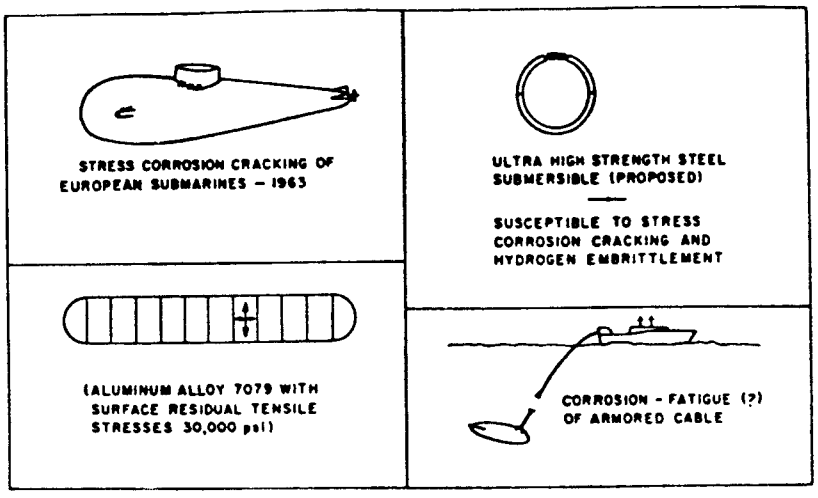


Fig. 1 Types of current or possible future corrosion control problems [3]

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,

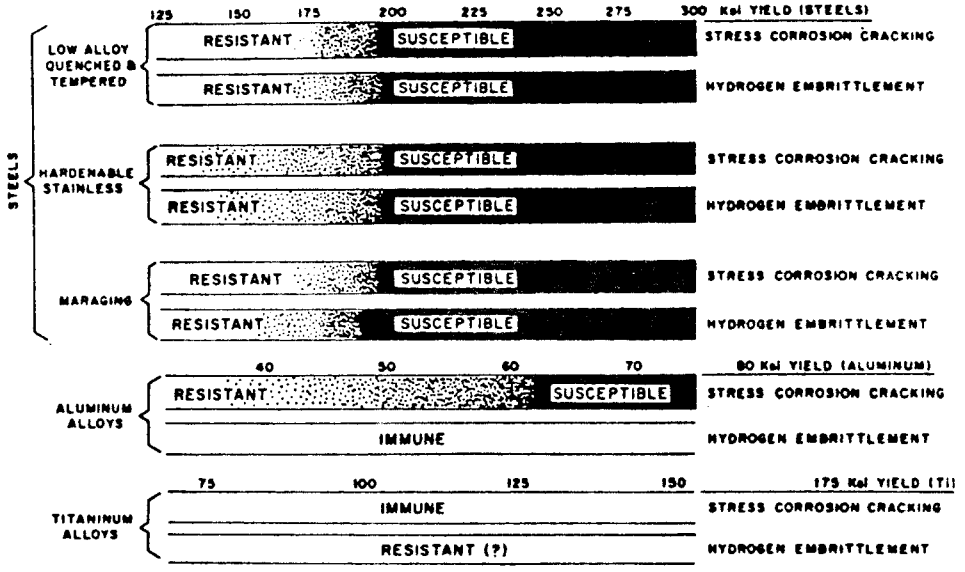


Fig. 2 Estimated strength ranges for susceptibility to cracking in sea water, fresh water, or humid atmospheres [3]. Note: Readers are cautioned that titanium alloys were regarded to be immune to both stress corrosion cracking and hydrogen embrittlement when this figure was originally drawn; however,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

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.

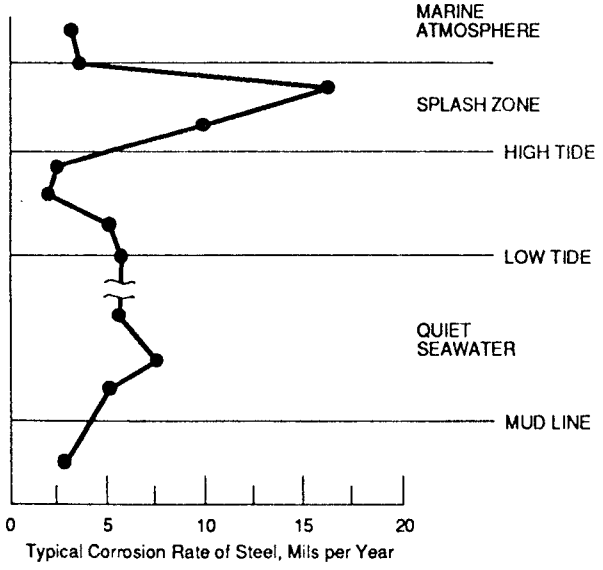


Fig. 5 Corrosion of steel piling [6]

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 c stress and corrosion than would be expected fror the sum of the individual effects of stress and corrc sion acting alone. Stress corrosion cracking involve a brittle-type fracture in a material that is otherwis ductile. The surface direction of the cracks is pei pendicular to the direction of the load. Stress corrc sion cracking should not be confused with othe types of localized attack, such as pitting, galvani attack, intergranular corrosion, impingement, c cavitation.

Causes of stress corrosion cracking—There ar many variables affecting the instigation of stres corrosion cracking. Among these variables are allo, 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, deadweight 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

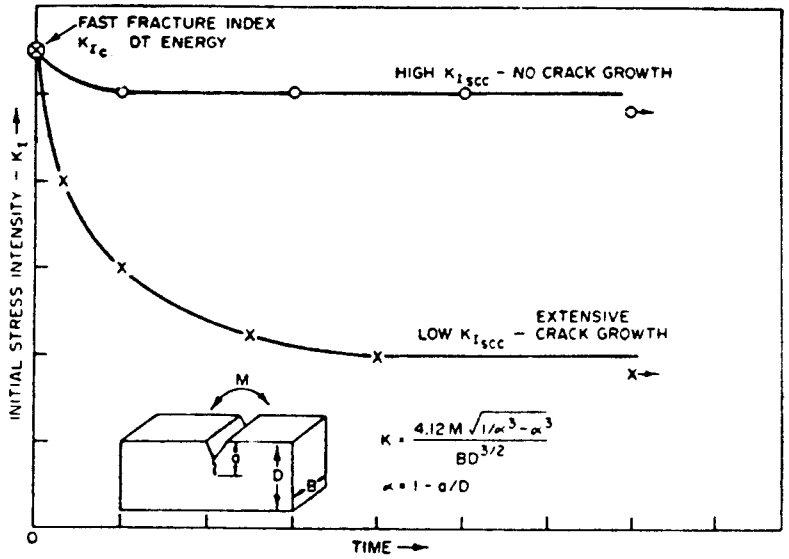


Fig. 6 Graph of K_{Iscc} determined by cantilever test method. Purpose of test is to determine threshold level for crack propagation [8]

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

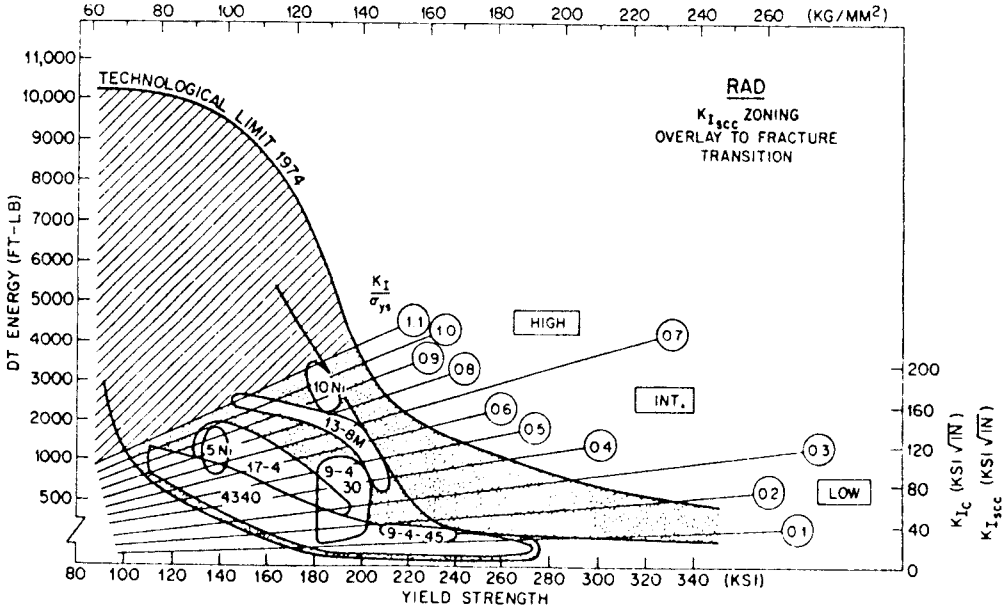


Fig. 7 Ratio analysis for SCC of various types of steel [8]

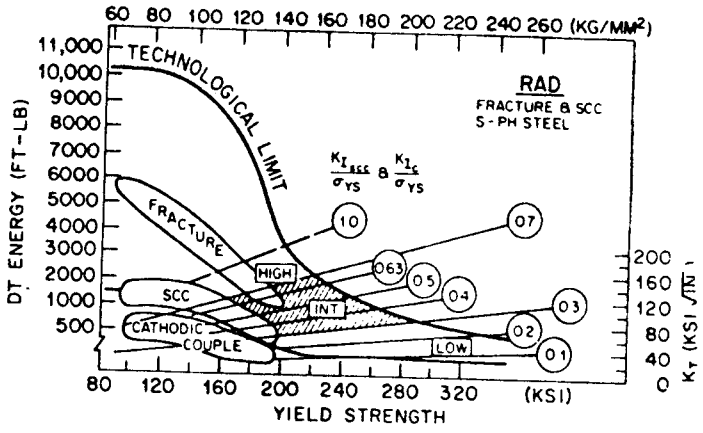


Fig. 8 Ratio analysis showing fracture and SCC properties for S-PH, a high-strength stainless steel [8]

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.

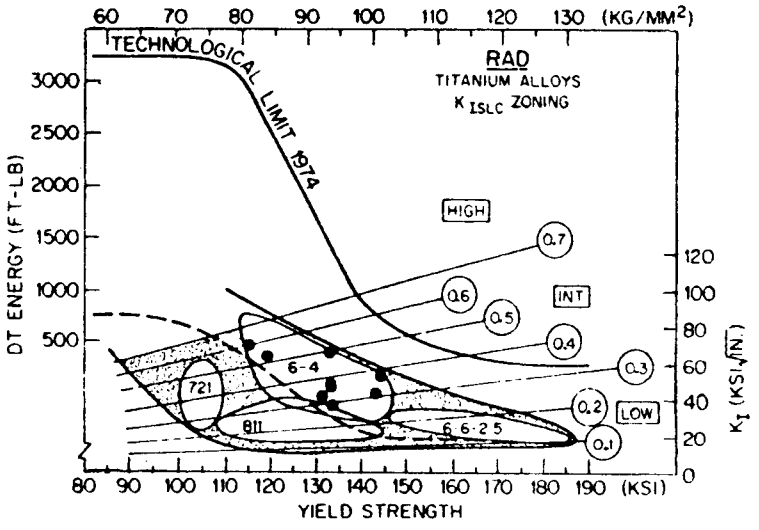


Fig. 9 RAD comparison of SLC properties of Ti-6Al-4V alloys with the data zone for salt water SCC [8]

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-

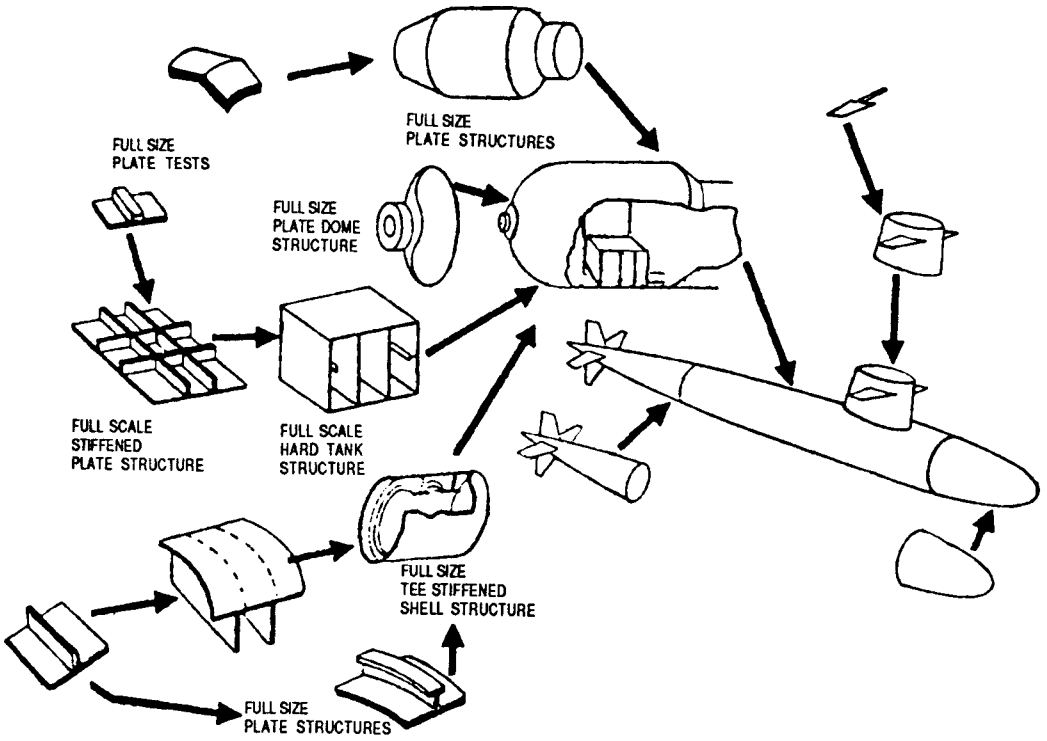


Fig. 10 Structure development and certification process for advanced submarines [13]

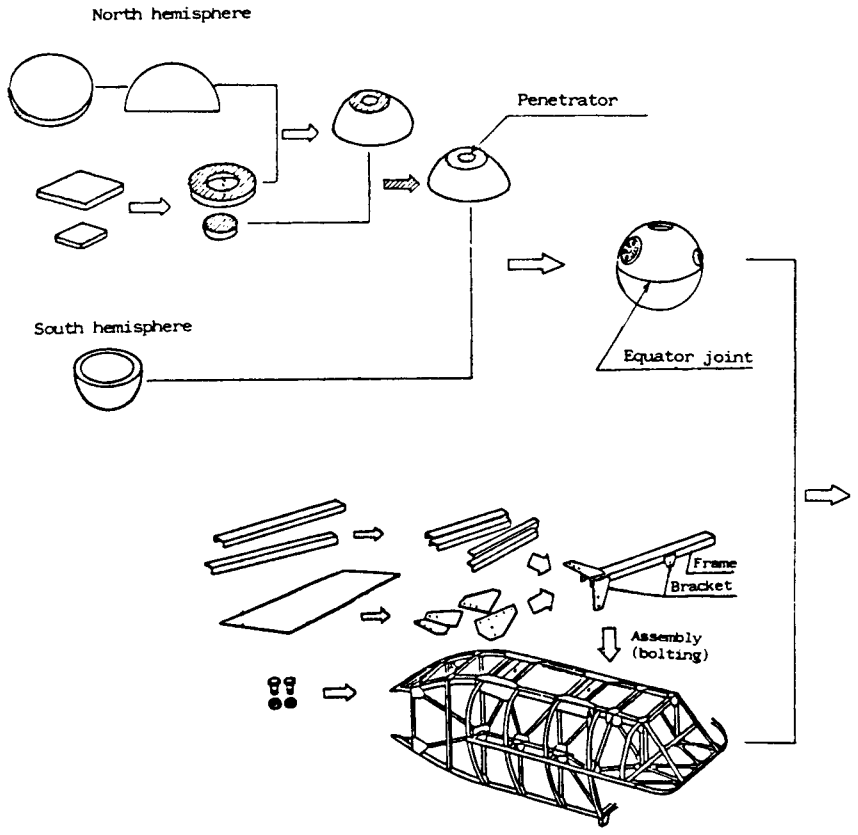


Fig. 11 Structure development for deep-diving submersible [14]

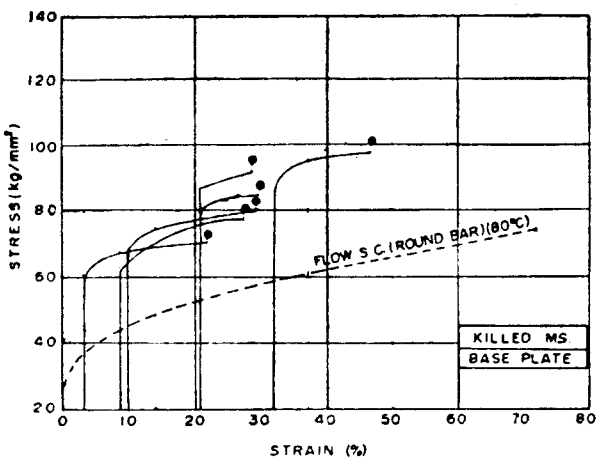


Fig. 12 Fracture stress curves of base plate [15]

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

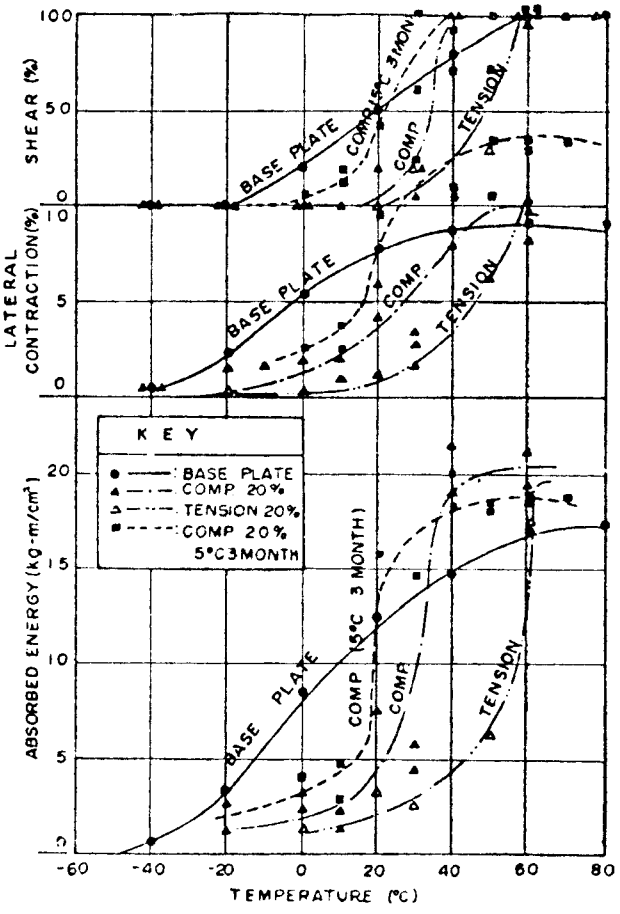


Fig. 13 Transition curves of prestrained steel (Charpy V-notch test) [16]

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 semisphere. 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

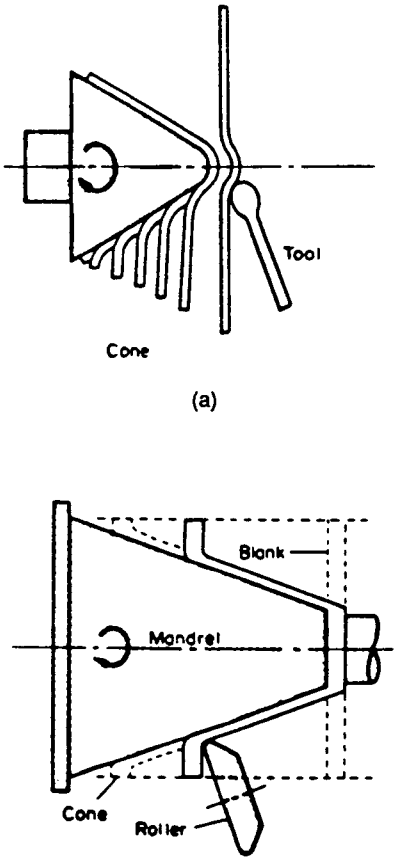


Fig. 14 Spinning processes [17]: (*a*) conventional spinning of sheet metal; (*b*) power spinning of conical part

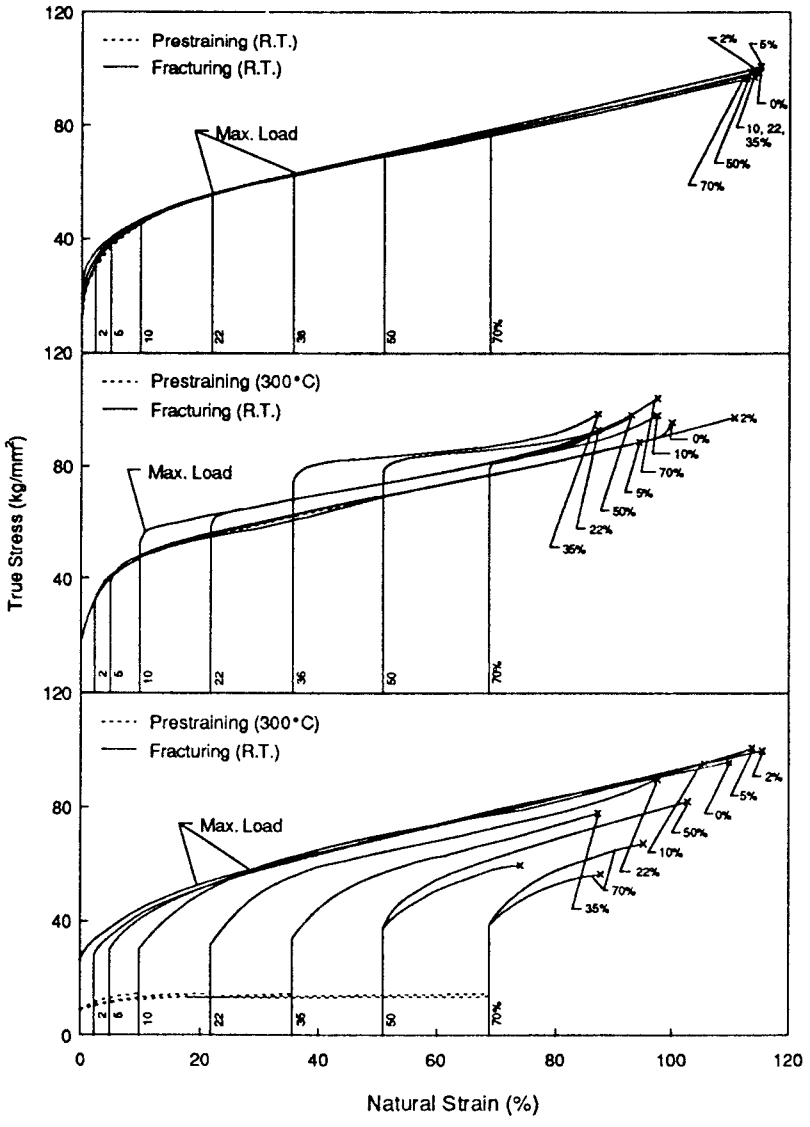


Fig. 15 Effect of prestrain on flow stress curve (tensile prestraining) [18]

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

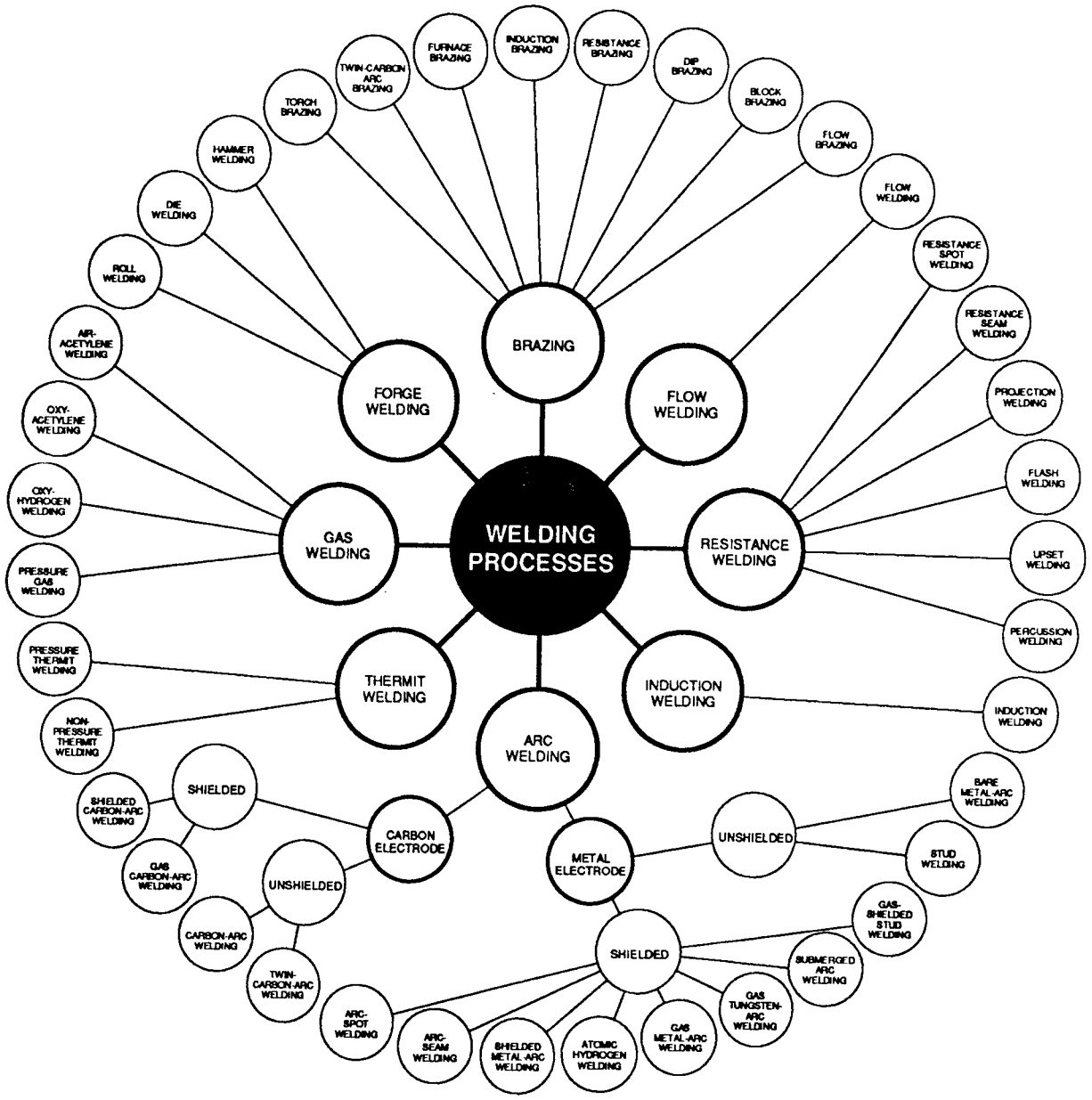


Fig. 16 Master chart of welding processes [1,19]

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.

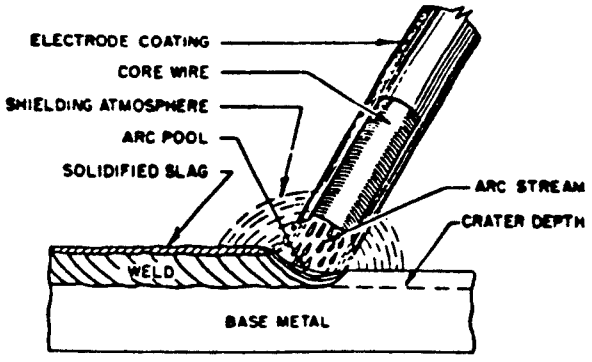


Fig. 17 Schematic representation of shielded metal-arc process [1]

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.

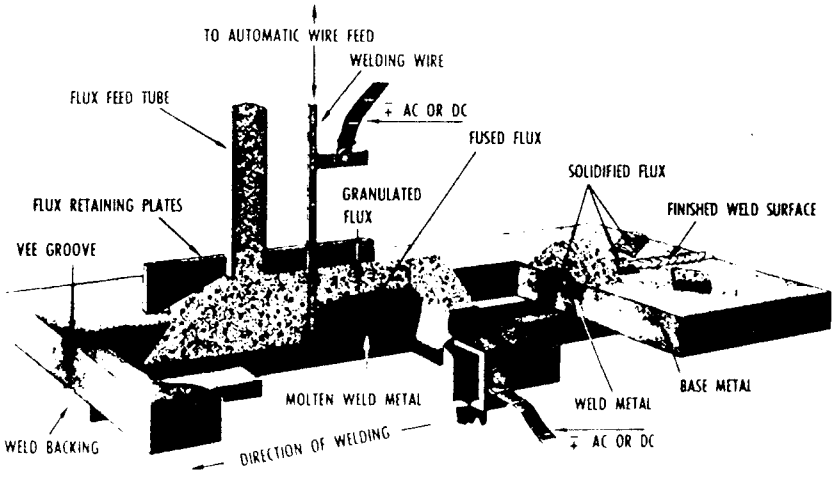


Fig. 18 Processes of a submerged-arc groove weld

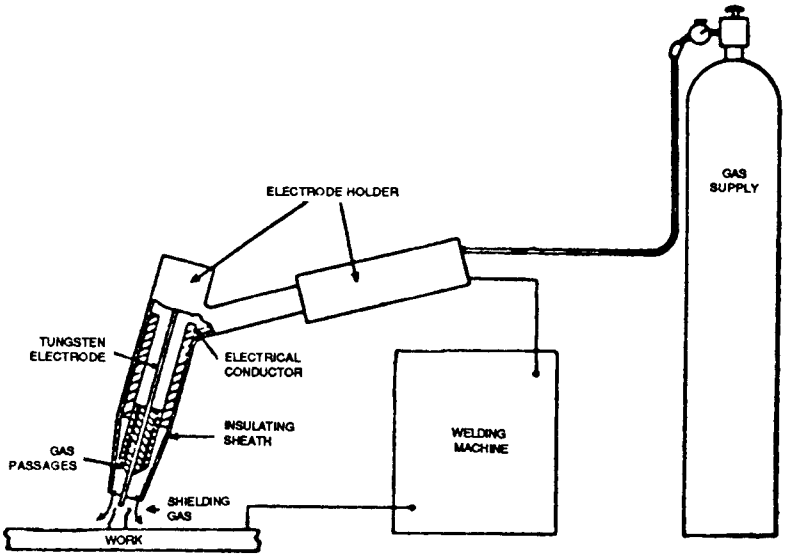


Fig. 19 Schematic diagram of gas tungsten-arc welding

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.

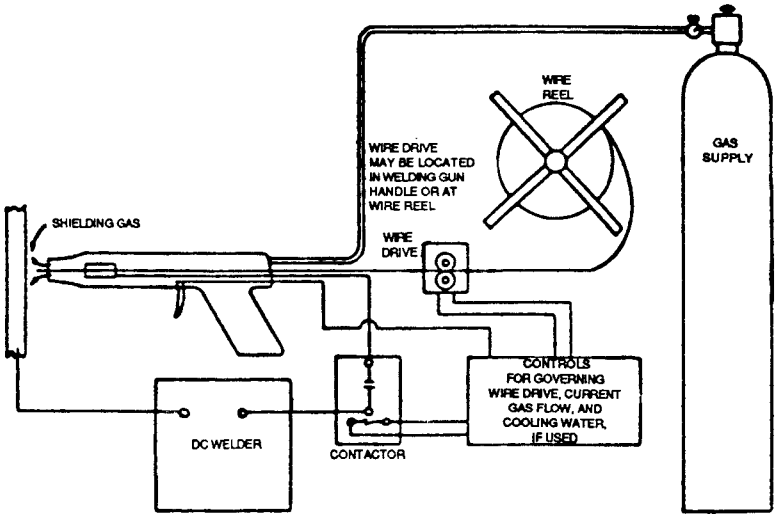


Fig. 20 Schematic diagram of gas metal-arc welding

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.)

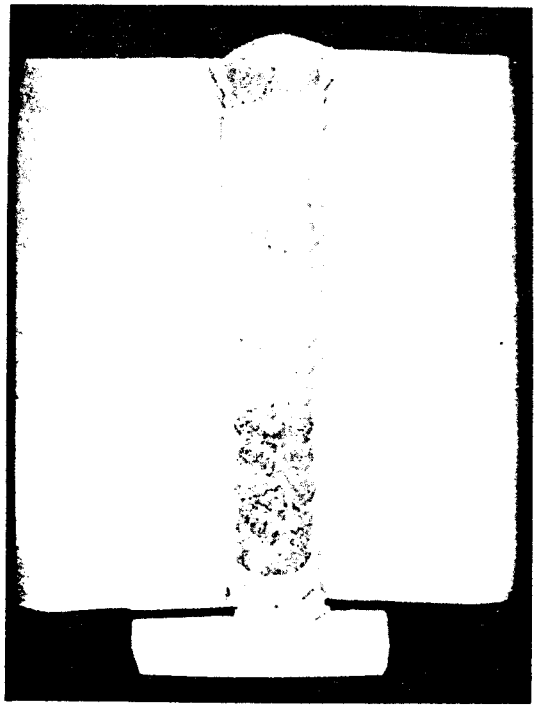


Fig. 21 A weld deposited by the narrow gap process

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

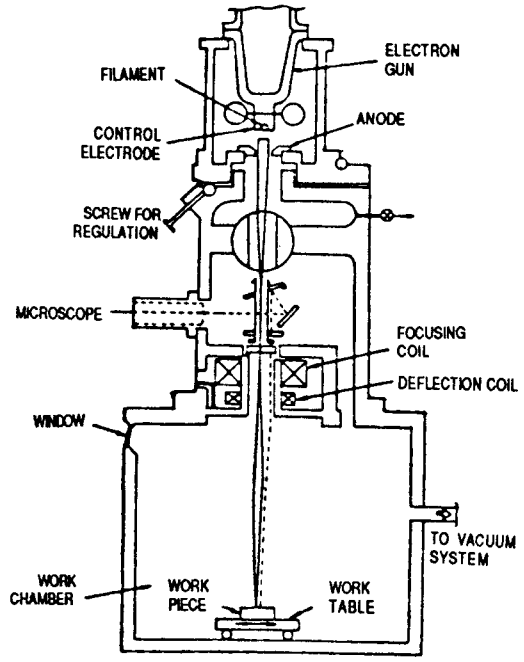


Fig. 22 Schematic representation of EB welding machine [23]

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.

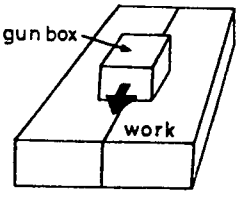


Fig. 23 Schematic drawing of movable chamber type EB welding machine [23]

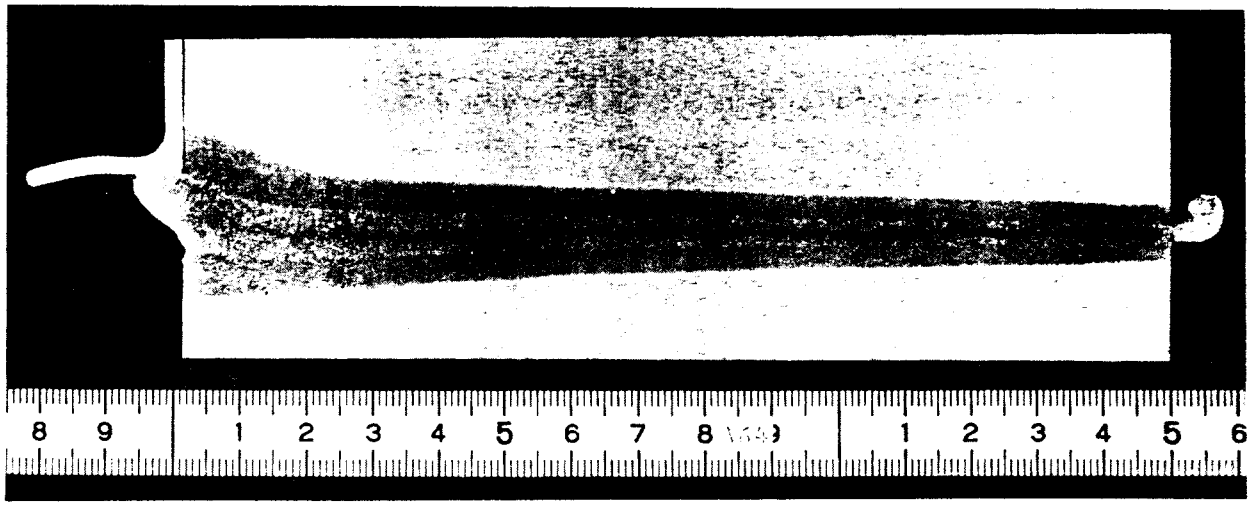


Fig. 24 EB welded 150-mm-thick (6-in.) steel plate

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

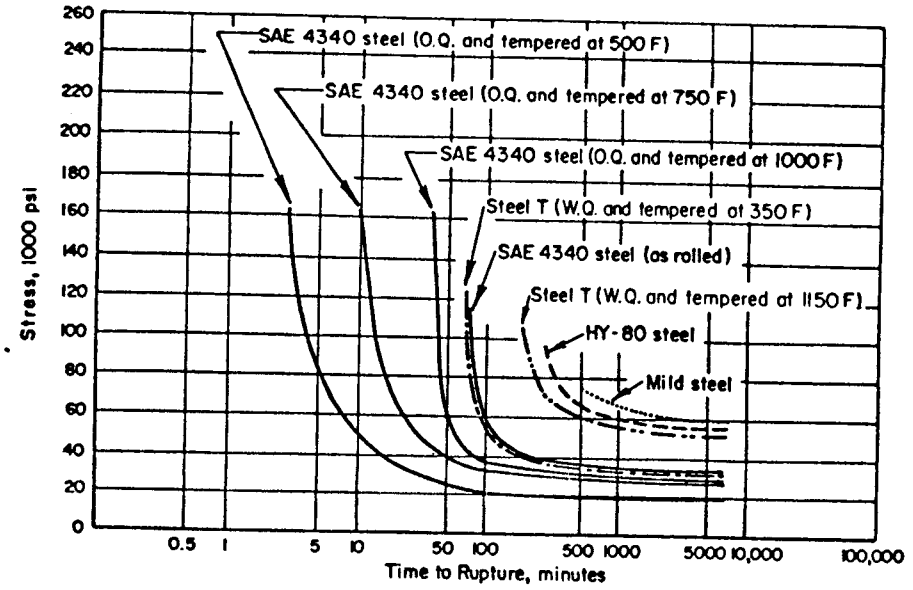


Fig. 25 Hydrogen-induced delayed fracture characteristics of various steels [25]. Note: Curves are estimated from results obtained by Simcoe *et al.* [26]. The commercial high-strength structural steel is identified above as Steel T. Table 2 contains approximate tensile strengths of these steels

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

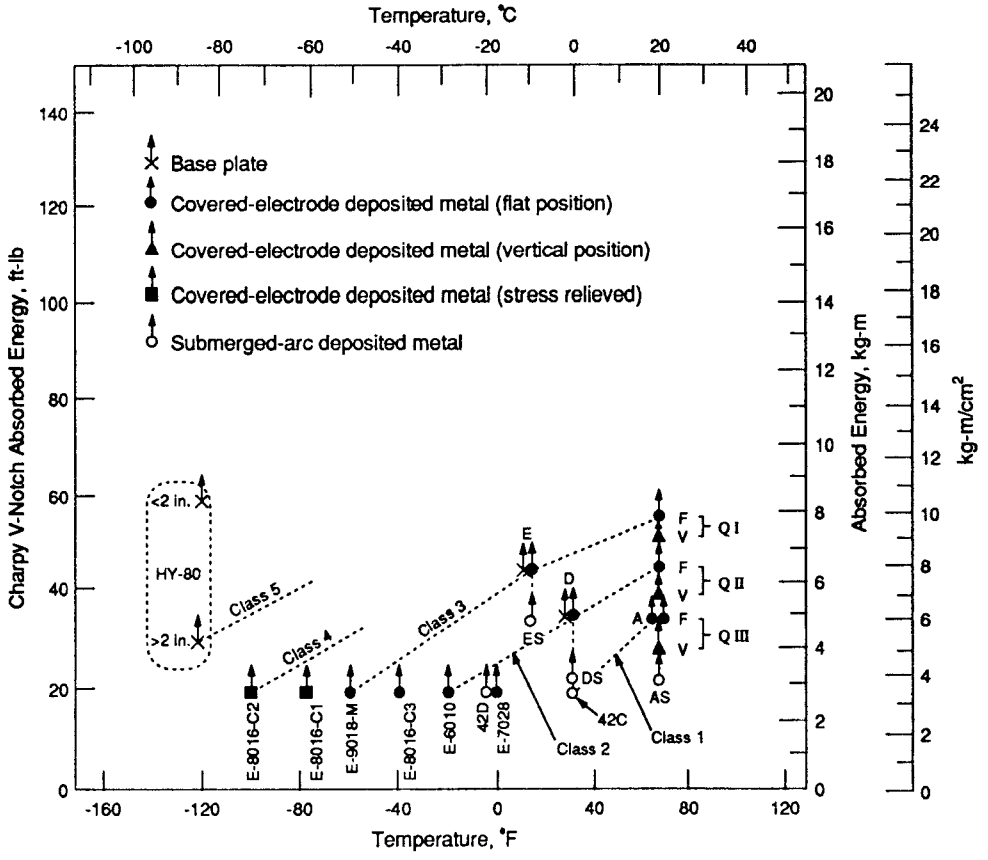


Fig. 26 Notch-toughness required by various specifications: 1. International specification for ship steel—A, D, E (AS, DS, ES are 70% of A, D, and E, respectively; submerged-arc welding). 2. AWS: E-7028, E-6010, E-8016-C3, E-9018-M, E-8016-C1, E-8016-C2. 3. IIW, Commission II: QI, QII, QIII for flat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

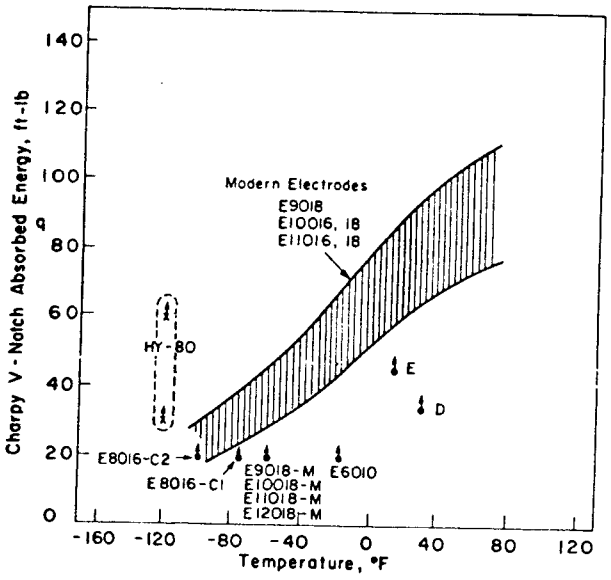


Fig. 27 Notch-toughness of weld metals of low-hydrogen high-strength steel electrodes

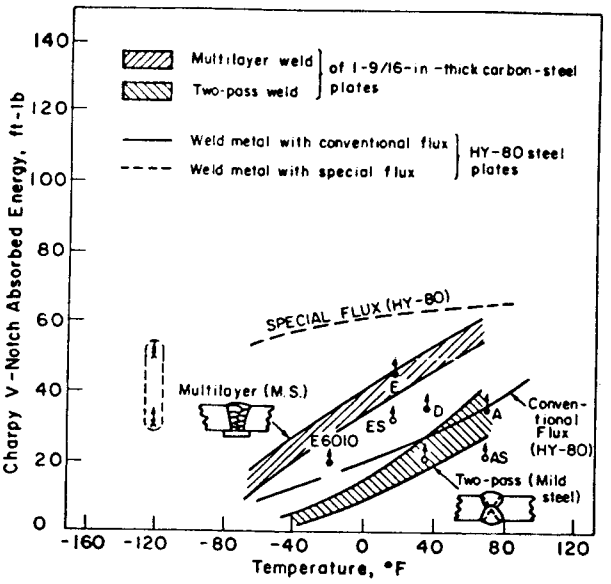


Fig. 28 Notch-toughness of submerged-arc deposited metals in mild steel and high-strength steels

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.

nique with heat inputs of 45 000 J per inch of weld bead. The improvement of the experimental weld metal was over 90 000 psi.

The oxygen content of the experimental weld metal was less than one half that of the conventional weld metal. Microscopic investigations revealed that the experimental weld metal was significantly cleaner (with fewer inclusions) than the conventional weld metal. The investigators believed that the improvement in notch toughness was a result of the lower oxygen content and fewer inclusions in the experimental weld metal.

The development of a filler metal matching the base plate in both strength and notch toughness, then, is very difficult. In the case of higher-strength materials, the difficulty is even greater. The state-of-the-art of the development of base metal and filler metal for individual higher strength materials will be discussed in Section 3.

2.3 Other Characteristics

From the materials point of view, the most important consideration is the avoidance of any kinds of fracture, thereby keeping the integrity of structures. There are several types of fracture, including brittle fracture, fatigue fracture, corrosion, creep, and buckling, brittle fracture and fatigue being the most important.

2.3.1 Brittle Fracture

Because a submersible vehicle is a welded structure, brittle fracture is a major consideration in its design. This section begins with a presentation of the theories of brittle fracture as a background for a later discussion of the methods of evaluating notch toughness and the fracture of welded structures in terms of fracture mechanics.

Theories of brittle fracture—Since a number of books have been written recently on theories of fracture, the coverage in this section is limited to the important information which would help readers understand the contents included in this book.

It is recognized that much research has been done recently on microscopic and atomic mechanisms of brittle fracture. However, the discussion in this section covers primarily macroscopic mechanisms of fracture, because they have been developed to improve the understanding of macroscopic fractures of engineering structures.

Theoretical strength and actual strength of materials [29]: The common denominator of all thinking on the fracture of solids is the ideal elastic solid, or one which would exhibit elastic response to a load until atomic separation took place on a plane by overcoming the interatomic forces. Such behavior is expected in the case of an ideal crystalline solid which contains no defects of any kind.

The calculation of the theoretical strength of an ideal elastic solid is based on the proposition that all the energy of separation is available for the creation of two new surfaces; the only energy expenditure in creating these two new surfaces is assumed to be the surface energy. Then the theoretical strength, σ_{th}, is

$$\sigma_{th} = \sqrt{\frac{ES}{a}} \tag{1}$$

where

E = Young's modulus,
S = Surface energy per unit area, and
a = Lattice parameter of crystal.

In the case of steels, for example, σ_{th} is approximately 3×10^6 psi, using the typical values for E, S, and a. However, the strength of a real material is much less than the theoretical strength; even most modern high-strength steels have ultimate strengths less than 3×10^5 psi, which is only one tenth of the theoretical value.

The major reason for this great difference between the theoretical strength and the actual strength is that real materials contain many flaws which include atomic irregularities, cracks on microscopic or even macroscopic scales, and metallic and nonmetallic inclusions. This is known as the structure-sensitive property of the material. For the study of notch toughness, this structure-sensitivity against a crack or other defect should be understood.

The Griffith Theory and its modification: Griffith, in the 1920s, developed one of the earliest attempts to rationalize the observed strength of real brittle solids (elastic behavior up to fracture) [30]. He proposed that these materials have an array of preexisting flaws. This model is the basis of many modern fracture theories, especially the fracture-mechanics theory.

Griffith calculated the strain energy (per unit plate thickness) resulting from a crack of length $l = 2c$ in a thin plate under normal stress, σ, as

$$\text{Strain energy} = -\frac{\pi c^2 \sigma^2}{E} \tag{2}$$

where E is Young's modulus. The sign is negative because this is the energy which will be released in propagation of the flaw. The surface energy associated with the crack is given by

$$\text{Surface energy} = 4cS \tag{3}$$

where S is the surface energy per unit area.

The equilibrium crack size can be calculated for the condition of net change of potential energy equal to zero, namely,

$$\frac{d}{dc}\left(4cS - \frac{\pi c^2 \sigma^2}{E}\right) = 0 \tag{4}$$

or

$$c = \frac{2SE}{\pi\sigma^2} \tag{5}$$

The critical stress (fracture stress) for a crack of size $2c$ is given by

$$\sigma_c = \sqrt{\frac{2SE}{\pi c}} \tag{6}$$

As the crack extends or as c increases, the stress necessary to propagate it decreases.

The theory predicts strengths reasonably well for bodies which behave in a brittle fashion, such as glass and ceramic materials. The fracture of metals is accompanied by dissipation of energy by plastic flow adjacent to the fracture surface, and this energy absorption is often at least ten times the energy corresponding to surface energy. Orowan [31,32] accordingly proposed that the S term in Equations (5) and (6) should be replaced by p, a term which includes both the surface energy and the energy of plastic deformation, yielding

$$c_c = \frac{2pE}{\pi\sigma^2} \tag{7}$$

$$\sigma_c = \sqrt{\frac{2pE}{\pi c}} \tag{8}$$

Fracture-mechanics theory [1]: Irwin and associates [33,34] have expanded the Griffith theory to the commonly called *fracture-mechanics theory*. The fracture-mechanics theory has been used extensively to explain fracture of brittle materials, especially ultra-high-strength materials for aerospace applications.

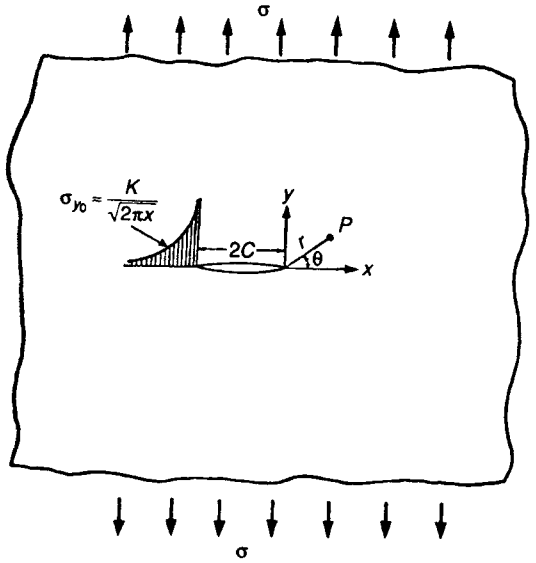


Fig. 29 Infinite plate containing a sharp crack of length $2c$ under uniform tensile stress σ

When a crack of length $l = 2c$ exists in a plate with uniform thickness under uniform tensile stress, σ, as shown in Fig. 29, the stress components (σ_x, σ_y) at a point, $P(x,y)$, near the crack tip are expressed approximately by

$$\sigma_y = \frac{K}{\sqrt{2\pi r}}\left(\cos\frac{\theta}{2}\right)\left(1+\sin\frac{\theta}{2}\sin\frac{3\theta}{2}\right)$$

$$\sigma_x = \frac{K}{\sqrt{2\pi r}}\left(\cos\frac{\theta}{2}\right)\left(1-\sin\frac{\theta}{2}\sin\frac{3\theta}{2}\right) \tag{9}$$

where K is the stress-intensity factor, which is expressed as

$$K = \sigma\sqrt{\pi c} \tag{10}$$

where c is one-half the crack length.

Then the rate of elastic strain energy release due to the crack extension, $G = \partial U/\partial c$, is expressed as

$$\frac{dU}{dc} = G = \frac{K^2}{E} \tag{11}$$

Figure 30 shows schematically the typical results when a series of fracture tests are made using specimens with different crack lengths in a certain high-strength material [1]. When the length of the initial crack is long enough, and the fracture occurs before general yielding, the relationship between the initial crack length, $2c$, and the average fracture stress, σ, is expressed by

$$K = \sigma\sqrt{\pi c} = \text{const.}$$

Consequently, fracture occurs when the K value reaches a certain critical value K_c for that material. K_c is the *critical stress-intensity fracture*, or it is often simply called *fracture toughness*.

When the fracture stress increases and exceeds approximately 80 percent of the yield stress, experimental fracture stresses deviate from the K = constant curve, and fracture occurs in a ductile manner. As illustrated in Fig. 30, the fracture-mechanics theory should be used only when fracture occurs before general yielding.

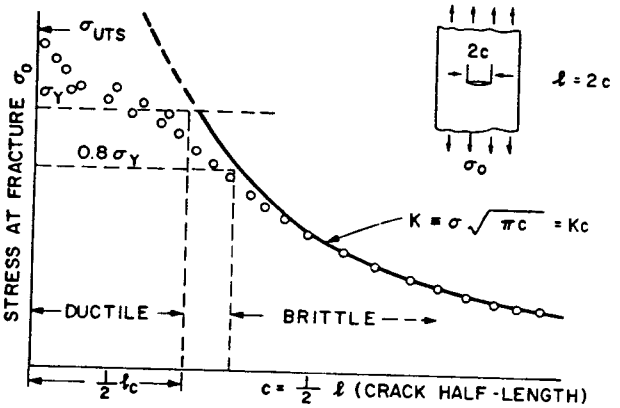


Fig. 30 Effect of crack length on fracture strength

Fracture toughness, K_c, has the following physical meanings:

- Relative to the locally elevated stress field at the leading edge of a crack, K_c represents the intensity of local tensile stress necessary for unstable crack propagation.
- For a crack of length 2c in a large sheet, the K_c value permits estimation of the membrane tensile stress σ necessary for unstable crack propagation through the relationship

$$\sigma = \frac{K_c}{\sqrt{\pi c}} \tag{12}$$

- Relative to the modified Griffith theory, the strain-energy-release rate G_c for unstable crack propagation may be directly expressed in terms of K_c by the relationship

$$K_c^2 = EG_c \tag{13}$$

When further tests are conducted with specimens in different thicknesses, the K_c changes as shown in Fig. 31. As thickness increases, the K required to produce failure decreases and approaches a lower limit which is the critical stress intensity under plane-strain condition, K_{Ic}. From the engineering viewpoint, plate-strain fracture toughness, K_{Ic}, is considered as a material constant.

The ASTM Committee on Fracture Testing of High-Strength Sheet Materials [35] has described techniques of determining fracture toughness of high-strength sheet materials. Their interest has been limited to materials, both ferrous and nonferrous, having a strength-to-density ratio of more than 700 000. In order to obtain a valid K_{Ic}, ASTM specifies the following requirement on the thickness of the specimen:

$$B \geq 2.5\left(\frac{K_c}{\sigma_y}\right)^2 \tag{14}$$

where B is the specimen thickness and σ_y is the yield strength. When the requirement of Equation (14) is met, K_c in Equation (14) can be considered a valid K_{Ic}.

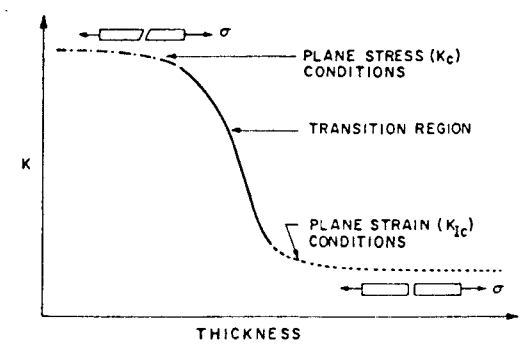


Fig. 31 Behavior of stress intensity required to produce failure as a function of material thickness

Table 4 Typical values of K_c for various materials [35,36]

Material	Yield Strength, 1000 psi	K_c Value, 100 psi$\sqrt{\text{in.}}$
AISI 4340 Steel (air melt)		
Tempered at 350°F	208.3	192
Tempered at 425°F	203.9	204
Tempered at 500°F	197.9	174
Tempered at 700°F	181.6	204.5
6434 (vacuum melt)		
Longitudinal	190	205
Transverse	190	195.5
Tricent (Vacuum melt)		
Longitudinal	190	197
Transverse	190	204.5
Ti-6Al-4V		
Longitudinal	159	114
Transverse	164	113

Fracture toughness of materials: Table 4 presents the typical values of K_c for various sheet materials [1,36]. Figure 32 shows the general trend indicating how K_{Ic} values of different steels decrease with increasing yield strength [1,37]. For steels with over about 180-ksi yield strength, the K_{Ic} value is usually low enough that the fracture-mechanics approach is applicable. However, for steels with less than 180-ksi yield strength, the K_{Ic} value increases drastically; consequently, the fracture-mechanics approach is generally not applicable.

The above discussions show that

- A crack will grow under stress when its size is larger than a critical size determined by the material properties for a given stress and
- The critical crack length of steel decreases greatly as the yield strength increases.

However, there is evidence that fractures have occurred in actual structures from flaws smaller than the critical size. Many fractures in welded ships originated in flaws of various kinds, most of which were much smaller than critical crack lengths estimated from fracture tests. Additional factors are needed to cause the growth of a subcritical crack to a critical crack. Boyd [38] has discussed the effects of additional factors on initiation of brittle fracture. The effect of various factors, mechanical or metallurgical, may be classified according to the following two.

The first is the additional energy supply. If additional energy is supplied either by residual stress, by stress concentration due to structural discontinuities, or by impact loading, a crack of subcritical size may grow to critical size. According to the Griffith-Irwin theory of brittle fracture, the relationship between crack length and the rate of release of strain energy G is as shown by Line OA_1 in Fig. 33 [1,36,39]. When the G_c value of the material is $2S_1$, an initial

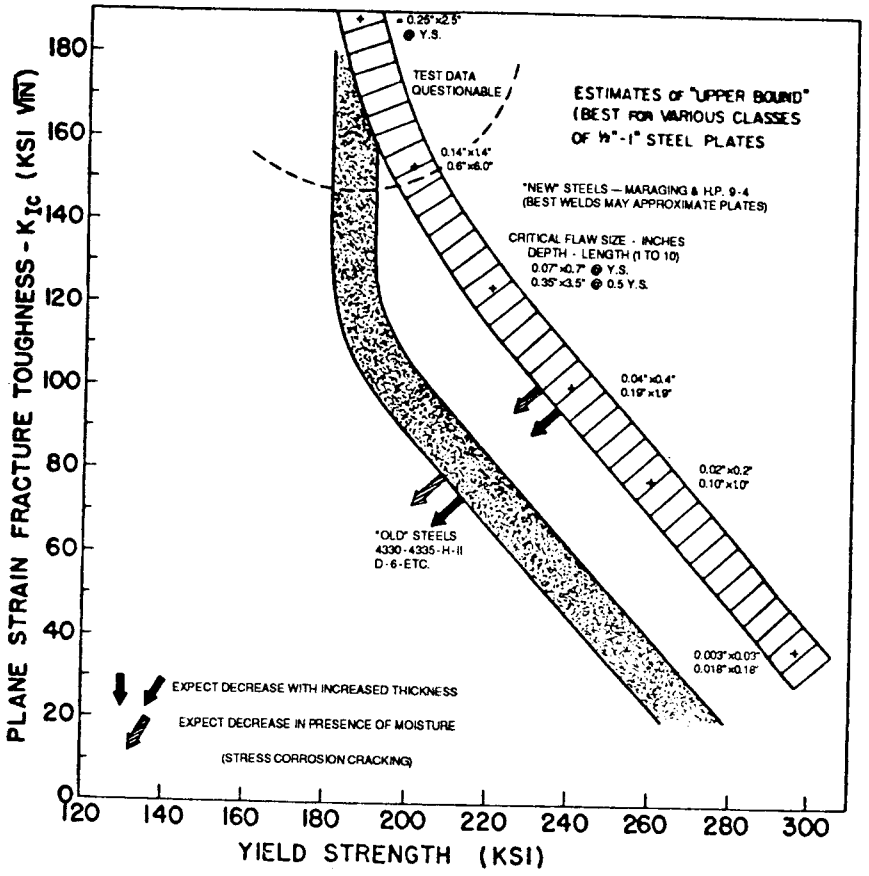


Fig. 32 Upper bound, best value, limits of plane stress fracture toughness (K_{Ic}) for ½- to 1-in. plates of high-strength steels, as reported by various qualified laboratories [37]

crack has to be longer than l_{c1} in order that unstable fracture can occur. However, if a crack is located in an area of high tensile residual stresses, the relationship between the crack length and the rate of release of strain energy may be shown schematically by Curve OR_1B_1 of Fig. 33. Then, for a G_c value of $2S_1$, the crack can grow in length l_{r1}, which is much shorter than l_{c1} for a uniform stress. For this reason, a structure causes a catastrophic fracture under unexpectedly low working stress.

The second factor effecting the initiation of brittle fracture is material embrittlement. The material resistance to fracture can be reduced by welding heat. Embrittlement by such mechanisms as strain aging, precipitation hardening, and hydrogen embrittlement can change the properties of a material so that a subcritical crack may grow to a critical crack or may become of critical size without growth. Figure 33 shows that the critical crack length for uniform stress σ_1 decreases from l_{c1} to $\frac{1}{2}l_{c1}$, when the G_c value of a material changes from $2S_1$ to S_1. Mylonas [40] proposed that loss of ductility due to plastic straining is an important factor which causes low-applied-stress fracture of a steel weldment.

Materials may be locally embrittled by hydrogen or stress corrosion cracking, as discussed in Sections 2.1 and 2.2, respectively. Masubuchi and Martin [41] applied the fracture-mechanics theory to analyze crack patterns produced in weldment in SAE 4340 steel.

2.3.2 *Fatigue Fracture*

When a material is subjected to repeated loading, fracture takes place after a certain number of cycles. The lower the applied stress, the larger the number of cycles before fracture takes place.

So far as fatigue fractures of engineering structures are concerned, the following two types of fractures are important:

- high-cycle, low-stress fatigue and
- low-cycle, high-stress fatigue.

In high-cycle fatigue, the endurance limit of a material after several million cycles or more is usually considered. In low-cycle fatigue, on the other hand, fracture after repeated loading of less than 10^5 cycles is usually discussed.

The above discussion presupposes that there is no defect, for example, a crack, in the structure. This

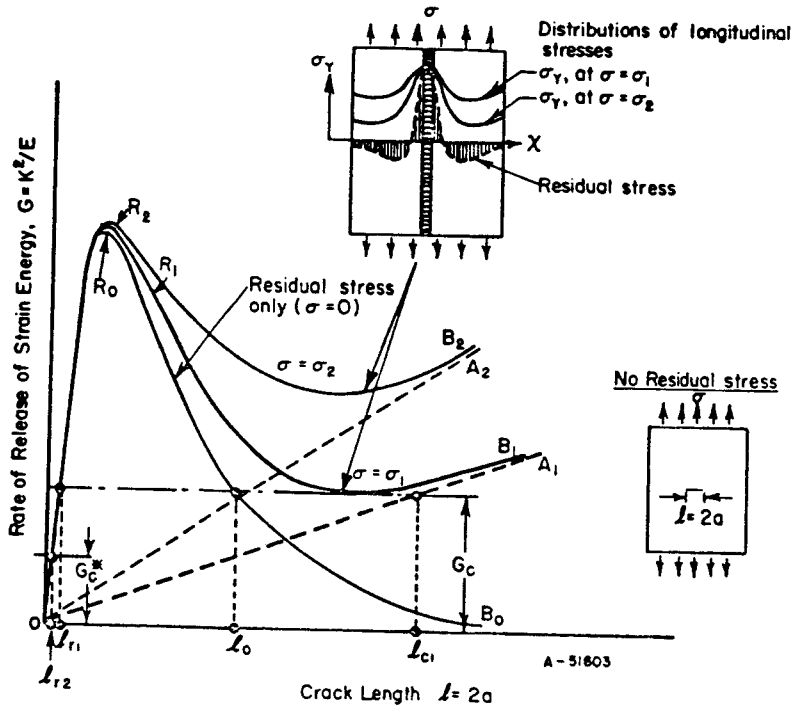


Fig. 33 Schematic diagram showing mechanisms of the low-applied-stress fracture of weldments [1,36,39]

supposition does not apply, however, in the case of a welded structure. Though careful inspections are conducted and repairs made on major defects, many defects remain in the welds and adjacent HAZ. Although these defects may not be serious enough to cause a catastrophic failure, they often are stress raisers, from which fatigue cracks start.

In considering the foregoing situation, it is safer to assume the existence of such defects. Because of these defects, a fatigue crack begins to grow shortly after the application of an external repeating load. Fatigue-crack propagation has thus come to be the main concern of fatigue-crack researchers today. For this reason, it will be discussed along with the two types of fatigue fracture mentioned above.

However, for ordinary members of ships and ocean engineering structures in which major loads are induced by waves or pressure changes, low-cycle fatigue is more of a problem. And, when structural members are exposed to sea water, corrosion fatigue often becomes a serious problem.

The trend toward a broader application of high- and ultrahigh-strength materials has increased the potential for failure by low-cycle fatigue. The principal reason for this is that the resistance to low-cycle fatigue, either to crack initiation or to crack propagation, is almost independent of the yield-strength and tensile strength of the materials [42]. Nevertheless, the basic property on which designers rely is the yield strength. As a result, structures made of high-strength materials become very susceptible to low-fatigue fracture because the high-strength material must sustain greater design loads, even though the low-fatigue strength has no gain. In this discussion, then, more emphasis will be placed on low-cycle fatigue, although some basic characteristics of high-cycle fatigue will be mentioned.

The significance of fatigue is explained as follows. When the material is sufficiently ductile, final fracture does not occur until the crack has reduced the cross-sectional area to such an extent that the nominal stress approaches the ultimate strength of the material. The monitoring of fatigue crack growth and subsequent repair are then enough. For relatively brittle materials, for example, higher-strength materials, fatigue problems are more serious. Although an initial fatigue crack can grow very slowly (years may elapse before it becomes long) it causes a catastrophic failure once it reaches the critical length. In fact, catastrophic brittle fractures in a number of structures actually started from fatigue cracks.

High-cycle fatigue—High-cycle fatigue is often discussed in connection with the Stress-Number of cycles to failure (*S-N*) curve.

Fatigue testing—stress cycles: Most of our knowledge about the fatigue behavior of materials has resulted from laboratory tests of relatively small specimens.

There are four possible basic parameters which can be used in the definition of the stress cycles to which a fatigue specimen is subjected. They are [43]

The minimum stress in the cycle: S_{min}

The maximum stress in the cycle: S_{max}

The mean stress: $S_m = \frac{1}{2}(S_{min} + S_{max})$

The stress range: $S_r = S_{max} - S_{min}$

The stress cycle is fully defined when any two of the above four quantities are given.

The *S–N* curve and fatigue strength: In order to determine the fatigue strength of a particular material under a given load condition, it is necessary to test several similar specimens. To do this, each of the three specimens is subjected to a different level of cyclic stress, and the number of loading cycles required to produce failure in each specimen is recorded. A relationship is then established between the applied stress, *S*, and the number of cycles to failure, *N*.

Figure 34 shows a relationship between the logarithms of both stress and the number of cycles, $\log S - \log N$ [44]. Because $\log S - \log N$ relationships for many materials are approximately linear, most fatigue data are presented on the basis of log–log relationships.

For plain specimens of ferrous metals it has been found that, after about 2 to 5 million cycles, the curve is almost parallel to the *N*-axis, indicating that at a very slightly smaller stress, the specimen would have an infinite life. This limiting stress is called the endurance limit of the material. Most other materials do not exhibit an endurance limit, although the *S–N* curve becomes substantially horizontal as *N* increases.

The sloping line of Fig. 34 can be expressed by the relationship

$$F_n = S\left(\frac{N}{n}\right)^k \qquad (15)$$

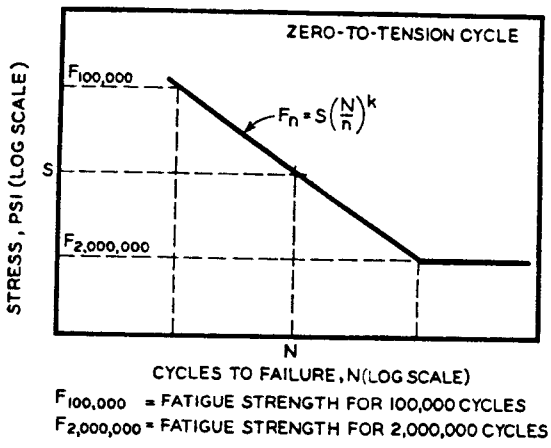


Fig. 34 *S-N* curve presented on a log-log scale

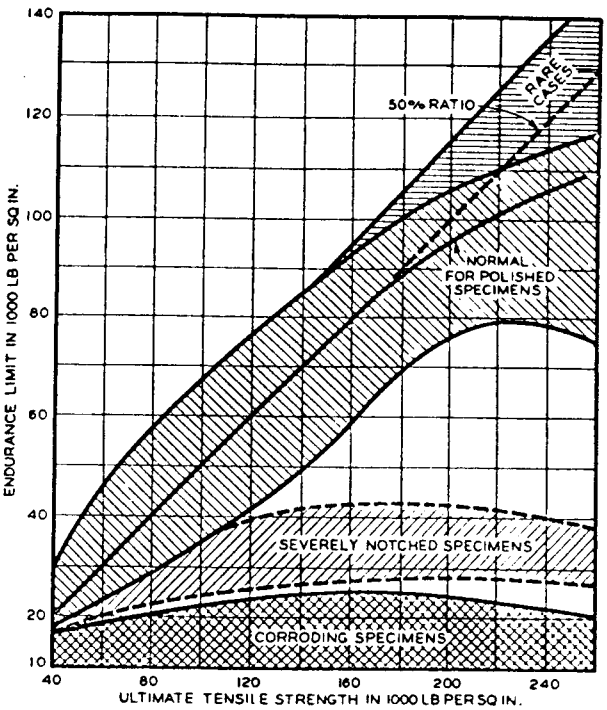


Fig. 35 Relationship between the endurance limit and ultimate tensile strength of various steels [44]

where

F_n = the fatigue strength computed for failure at *n* cycles,

S = the stress which produced failure in *N* cycles, and

k = the slope of the best-fit straight line representing the data [44].

In such a relationship, the fatigue strength can be computed over the range covered by the sloping line for any selected number of cycles of the same types of stress cycle, if the slope of the line and one point on the line are known [44].

Only one type of stress cycle is represented on an individual *S–N* curve. Thereafter, for a general understanding of the fatigue behavior of a material or joint, it is necessary to have a number of *S–N* curves, one for each different type of stress cycle.

Factors influencing fatigue limit: Figure 35 shows the relationship between the endurance limit and the ultimate tensile strength of various steels [44]. As Fig. 35 illustrates, the fatigue strength of materials increases as the strength of the material itself increases at a ratio of about 50 percent. This relationship is true, however, only when the specimen is polished and the surface is very smooth.

The most deleterious factor affecting the fatigue life of metals is the localized concentration of stress by geometric discontinuities, including cracks, notches, fillets, holes, and surface imperfections. The shape and size of the discontinuity determine its

ability to concentrate stress. The highest stress-concentrating effects are caused by cracks, while generous fillets with a smooth, polished surface have the lowest [45].

The stress-concentrating effects of discontinuities may be calculated by using methods based on geometry, dimensions, and assumed elastic behavior. The effect arrived at is called the theoretical stress concentration factor (K_t). The actual effect of a given stress concentration, however, may vary both within and among materials. It is possible, through tests, to establish the reduction in fatigue strength caused by a particular stress-concentration factor for a particular material at a particular strength level. By comparing these data with unnotched (smooth) test data, we can calculate the so-called fatigue strength reduction factor (K_f). From K_t and K_f, the notch sensitivity index (q) of the material can be calculated as

$$q = \frac{K_f - 1}{K_t - 1} \tag{16}$$

In most materials, the notch sensitivity tends to increase with increasing strength. Therefore, in the presence of sharp notches, it is not unusual to find little or no advantage in fatigue strength for high-strength materials [45].

As shown in Fig. 35, a material subjected to a corrosive environment is drastically affected in its fatigue strength. Corrosion, in fact, is one of the most detrimental factors resulting in low fatigue strength. Fatigue under a corrosive environment is called *corrosion fatigue.*

Many stressed structures in the sea fail because the sea water accelerates the fatigue process. A common solution to the problem is to design with a low-strength alloy at a low working stress, say below 5000 psi, and make a big, bulky marine structure. In recent years, with the need for better performance, attempts have been made to use higher-strength materials in the ocean environment. However, it has been found that if the designer also designs for the effect of the marine environment, the useful increase in strength is more than cancelled out by the material's reaction with its environment and an acceleration of the rate of crack propagation across the component. In fact, the advances made with AISI 4340 steels and other high-strength materials in the aerospace structures are often lost when these materials are submerged in sea water.

Figure 36 shows the effects of notch and sea water on flexural fatigue strengths of HY-130/150 steel and special grade Ti-6Al-4V [1,45]. The figure shows the fatigue strength of HY-130/150 to be more severely damaged by notch and sea water than that of Ti-6Al-4V.

Low-cycle fatigue—So that high-cycle and low-cycle fatigue can be distinguished, 10^5 cycles of failure will be used to represent the dividing line between the two types of fatigue.

Characteristics of low-cycle fatigue: Many metals exhibit low-cycle fatigue failure during the loading and unloading cycles which involve plastic deformation. Under such conditions, true stresses are not calculable by elastic theory, and plastic theory is not developed enough to handle such problems. The control and analysis of low-cycle fatigue, therefore, is easier on the basis of strain rather than stress.

Figure 37 illustrates the stress-strain relationships likely to develop under cyclic loading conditions. The relationship shown in Fig. 37*a* happens when the applied force or moment is completely reversed; but within the elastic region, σ_t is the total stress range and ε_t is the total strain range. Figure 37*b* shows the relationship that develops under reversed loading into the plastic region. The stress-strain relationship is no longer linear, but follows the hysteresis loop BCDEB during each cycle. σ_t is the total stress range, but in this instance may be nominal or true. ε_t, the total stress range, involves two

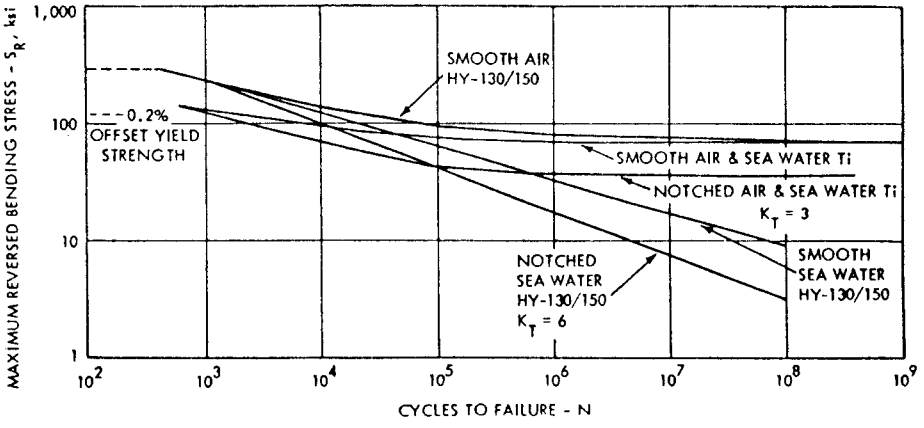


Fig. 36 Flexural fatigue curves for special grade Ti-6Al-4V [1,45]

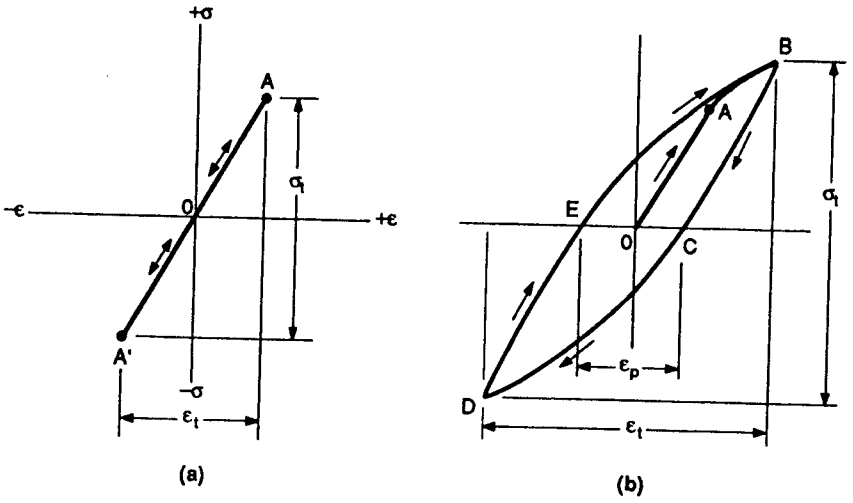


Fig. 37 Stress-strain relationships under cyclic loading [1,42]: (*a*) elastic range; (*b*) straining into plastic range

parts: the elastic strain range ε_e, and the plastic strain range ε_p, where $\varepsilon_t = \varepsilon_p + \varepsilon_e$.

High-cycle fatigue results usually are in the form of *S–N* diagrams, such as Fig. 34. Since the cyclic stresses needed for failure in the high-cycle region are well within the elastic region (see Fig. 37*a*), it is not important whether we use stress or strain as the independent variable.

With low-cycle fatigue, however, this is not the case. Due to the great preponderance of evidence collected over the last ten years, it has become clear that the strain becomes the dominant factor with decreasing fatigue life [1,42].

Figure 38 shows the low-cycle fatigue results obtained for unnotched (smooth) specimens by the U.S. Navy Marine Engineering Laboratory [42]. The yield strength of the steels ranges from 41 000 to 230 000 psi, and the plot is on the basis of total strain range versus cycles to failure. The similarity of behavior irrespective of yield strength is important to note. Perhaps even more significant is the fact that the behavior appears to be independent of material, as illustrated in Fig. 39. It seems that if strain is the controlling factor in low-cycle fatigue, the strength or type of material used makes little difference. Recent tests on pressure vessels tend to support this view for steels with yield strengths from 30 000 to 90 000 psi. However, high-cycle fatigue strength of materials varying in strength level are markedly different, as shown in Fig. 35. Consequently, the relationships shown seem to be valid only in the restricted range of 100 to 10 000 cycles. Divergence from the rela-

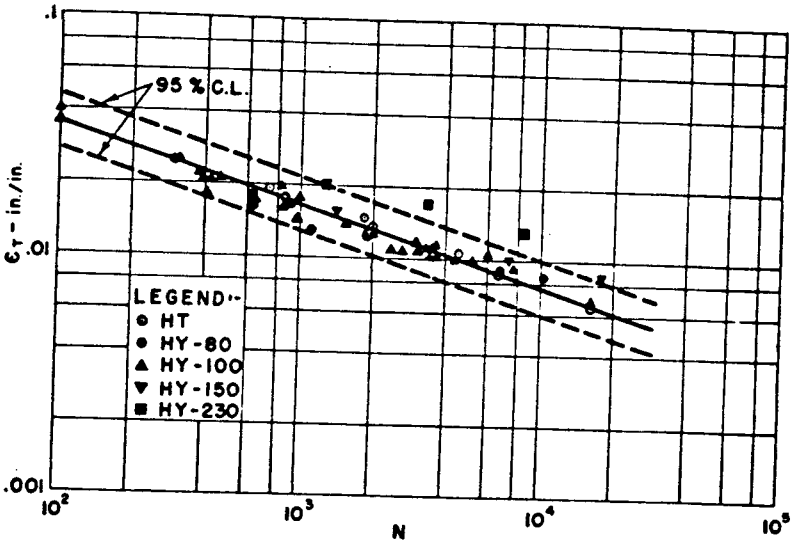


Fig. 38 Total strain range versus cycles to failure for steels in air [42]

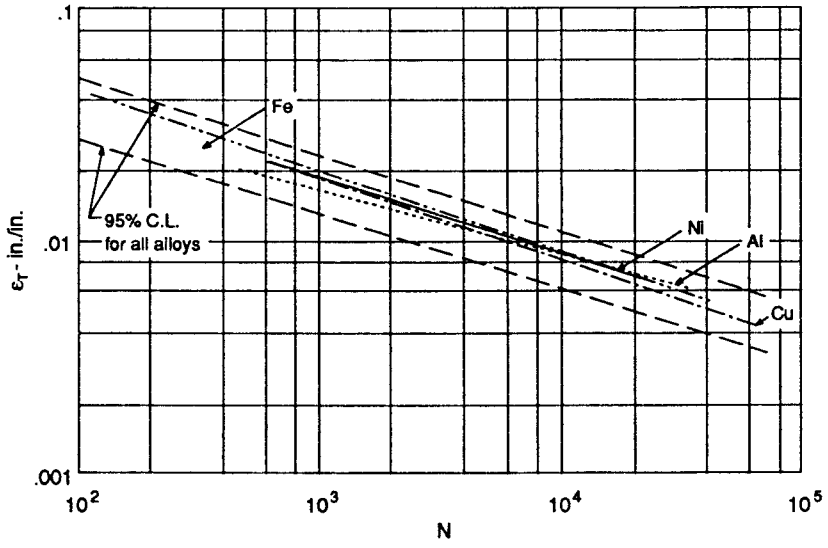


Fig. 39 Total strain range versus cycles to failure for alloys in air [42]

tionship is apparent for the HY-230 steel shown in Fig. 38.

In Figure 40, three steels with markedly different yield strengths are shown. Above 1000 cycles, advantages for materials at high yield strength are apparent. However, for a given cycle life greater than 1000, the relationship shows that the high-yield-strength materials can suffer low-cycle fatigue failure while apparently being cycled elastically. This actually has happened in tests of HY-230 steel and certain aluminum alloys [45].

Fatigue failure of welded structures: As mentioned in Section 2.2.3, welding poses many problems in structures. Fatigue, among other structural characteristics, can suffer the effects of welding. For example, welding often causes small defects (acting as the stress-raisers like notches) from which fatigue cracks emanate; further, welding tensile residual stresses reduce the fatigue life, for even a low-working stress produces high stress near the weld in combination with tensile residual stress [46].

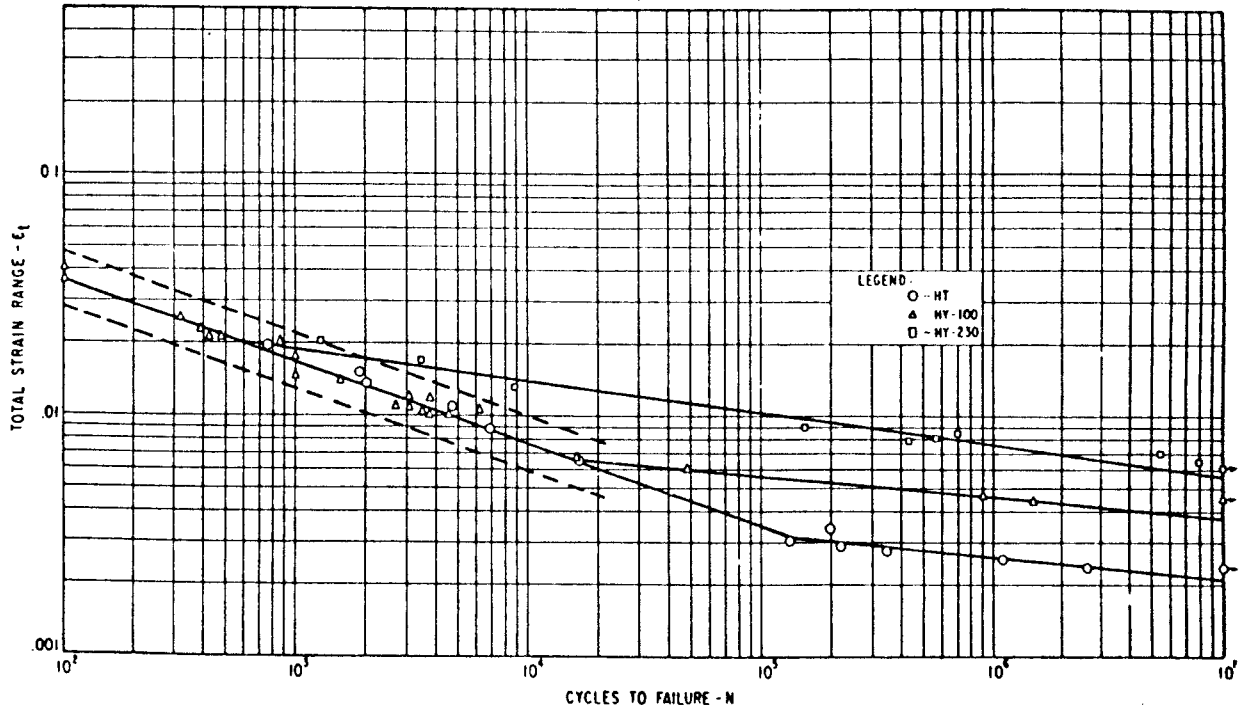


Fig. 40 Fatigue relationships for steels (completely reversed strain) [42]

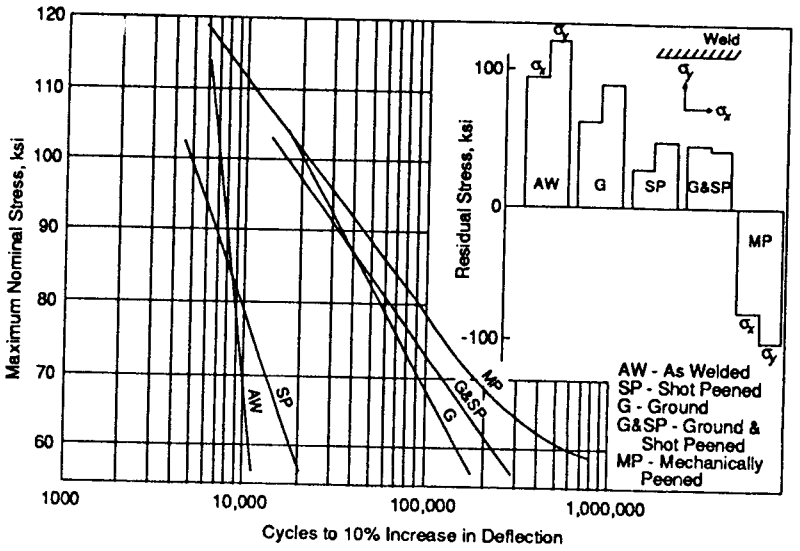


Fig. 41 Summary of fatigue curves and residual stress measurements obtained on plate-type specimens with tee-fillet welds finished by various fabrication improvement techniques [47]

In contrast to the extensive methods available for calculating static strength of structures, theories of fatigue fracture rely heavily on test results from large structural fatigue specimens. A fatigue test was conducted to investigate the effect of residual stress on the fatigue strength; the results are presented in Fig. 41 [47]. A plate type fatigue specimen with a tee-fillet weld was fatigue-tested in various conditions; welded, shot-peened, ground, ground and shot-peened, and mechanically peened. The residual stresses were measured in each case. From Fig. 41, it is obvious that tensile residual stress at and near the weld greatly reduces fatigue strength.

Fatigue crack growth—As mentioned in the beginning of this discussion, fatigue crack propagation has become one of the major areas of fatigue research. The successful application of fracture mechanics to this problem makes possible the prediction of the growth of fatigue cracks. In this process, information was obtained by periodically measuring the length of the crack which had been artificially initiated from a sharp saw cut. Once the crack length, $2a$, and the stress range, $\Delta\sigma = \sigma_{max} - \sigma_{min}$ (σ_{max}, σ_{min} are the maximum and minimum stress, respectively), were determined, the stress-intensity range was computed as

$$\Delta K = \Delta\sigma\alpha\sqrt{\pi a} \qquad (17)$$

where α is the correction factor for specimen geometry. (For example, $\alpha = [(2b/\pi a)\tan(\pi a/2b)]^{1/2}$ for finite panel of $2b$ wide with a crack length $2a$ at the center.)

The crack growth rate, da/dN, was obtained by dividing the difference in the crack length reading, da, by the cyclic number change, dN, for each measurement. The relationship between the crack growth rate and the stress intensity range was established from the test results [48].

$$\frac{da}{dN} = C\,(\Delta K)^m \qquad (18)$$

where C and m are constants.

Equation (18) was then modified in various formulas, taking the stress ratio, $R(\sigma_{min}/\sigma_{max})$, into account; Table 5 presents several of these. The data of high-strength steels, known as HY steels, are found in Fig. 42.

The growth of a crack can be computed from Equation (18) if a preexisting defect size is known and if the stress spectrum of the structural component containing the crack is given. Since the stress-intensity range is a function of crack length and the applied repeating stress range, the crack size after a certain number of stress cycles is calculated by integrating Equation (18):

$$N - N_0 = \int_{a_0}^{a} \frac{da}{C(\Delta K)^m} \qquad (19)$$

where

N = number of stress cycles,
N_0 = number of stress cycles at the initial crack location,
a = crack length, and
a_0 = initial crack length.

Together with brittle-fracture theory, Equation (19) enables one to evaluate the significance of preexisting defects in the welded structure. From this procedure, the design criteria known as *damage-tolerance* design has developed.

Table 5 Various fatigue crack growth laws

	Laws	Reference
Paris	$\frac{da}{dN} = C(\Delta K)^4$	[48]
McClintock	$\frac{da}{dN} = \frac{7.5}{16} \frac{(\Delta K)^4}{\varepsilon_f E^2 \sigma_{ys}^2 \rho^2}$	[49]
Krafft	$\frac{da}{dN} = \frac{A \cdot f(\Delta K/K_{max})}{E^3 K_{Ic}^2 N_w} \cdot K_{max}^4$	[50]
McEvily/ Johnston	$\frac{da}{dN} = A \frac{(\Delta K)^4}{\left(\frac{\sigma_{ys} + \sigma_{uts}}{2}\right) E \sigma_{uts}^2 \varepsilon_f}$	[51]
Pearson	$\frac{da}{dN} = \frac{3.34 \times 10^7}{E} (K)^{3.6}$	[52]
Forman	$\frac{da}{dN} = \frac{A}{\left(1 - \frac{\sigma_{min}}{\sigma_{max}}\right) K_c - \Delta K} (K)$	[53]
Throop/ Miller	$\frac{da}{dN} = \frac{A'}{E \cdot \sigma_{ys} \cdot K_{Ic}} (K_{max})^m$ $m = 3.5$ S.D. of $m = 0.65$	[54]

$K = K_{max} - K_{min}$	stress intensity range
ε_f	fracture strain
ρ	density
σ_{ys}	yield strength
σ_{uts}	ultimate tensile strength
σ_{max}	maximum stress
σ_{min}	minimum stress
E	Young's modulus
K_c, K_{Ic}	fracture toughness
C, m, A, A'	constants

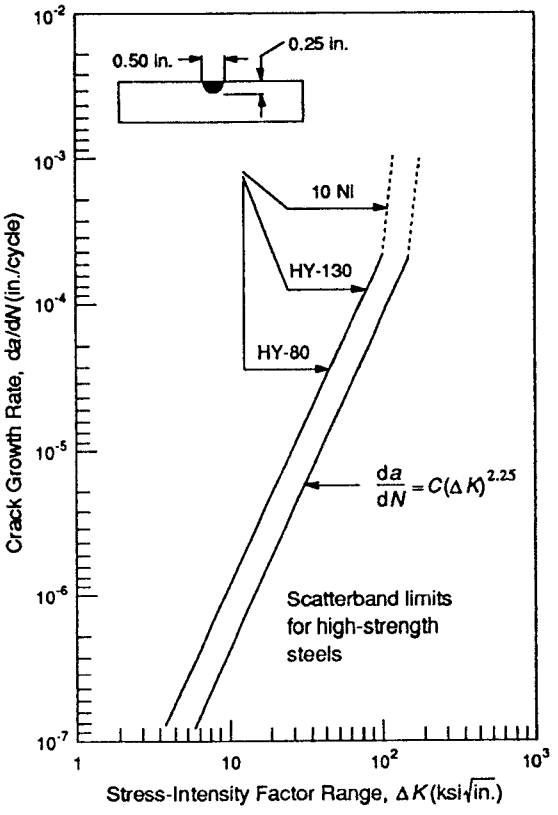


Fig. 42 Log-log plot of fatigue crack growth rate (da/dN) versus stress-intensity factor range (ΔK) showing the scatterband limits for a broad sample of high-strength steels. The bold arrows indicate the respective rates of crack propagation that would occur in each of three steels (HY-80, HY-130, and 10Ni) for a stress level/flaw size condition of a 0.25-in.-deep semicircular embedded surface flaw cycled at zero to 75 percent of the yield strength stress [55]

2.3.3 *Damage-Tolerance Design Concept*

Development of damage-tolerance design concept—As the advanced structures (such as atomic reactors, liquefied natural gas [LNG] carriers, airplanes, and submarines) increase, a design philosophy relying on more definite theoretical bases becomes necessary. Failure of any of the above-mentioned structures would cause a disaster if they were to fail.

Conventional design criteria have relied heavily on both the past performances of structures and their experiences in different situations. Many disastrous failures, however, have taken place in structures such as ships, bridges, and storage tanks since they sometimes undergo more severe conditions than expected. Hence, new design concepts are needed to meet these requirements.

As mentioned in Section 2.2, a welded structure contains various types of defects, which often become the origins of structural failure. If there are sufficient information about the size and location of defects and sufficiently accurate mathematical formulations for calculating crack growth (resulting from defects) under a given condition, the cracks can be repaired before they reach a size large enough to cause catastrophic failure. The development of nondestructive testing (NDT) techniques makes it possible to determine the size and location of a crack, though NDT techniques still have a limited ability to detect very small defects. Fracture-mechanics theories enable one to predict fatigue crack growth and stress corrosion cracking growth and to calculate the size of the critical crack initiating the brittle fracture. Through the probabilistic approach, one can define the load spectra experienced by a structure throughout its life and predict the probability of the failure occurrence.

The combination of these theories can minimize the occurrence of catastrophic failure of structures. Figure 43 is a flow chart of this integration of theories. Some aspects of this procedure are weak, but they can be strengthened by the further development of theories and the accumulation of experimental data. Good examples of the application of the damage-tolerance design include the design of airplanes and LNG carriers.

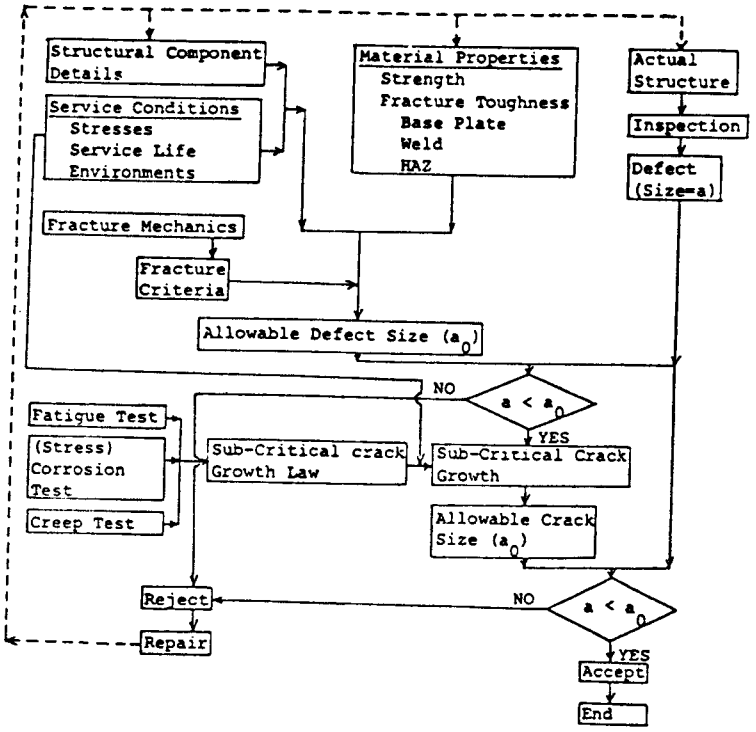


Fig. 43 Outline of damage tolerance design procedure

Fracture-analysis diagram: As investigators seek more accurate descriptions of the fracture phenomena in terms of mathematical formulations, their theories become more complex. To avoid this growing complexity, Pellini [56] proposed a fail-safe design concept for preventing brittle fracture of structures which is more engineering-oriented [1].

In Pellini's fail-safe design concept, the *fracture-analysis diagram* (FAD), as shown in Fig. 44, plays a key role. The FAD provides a generalized definition of

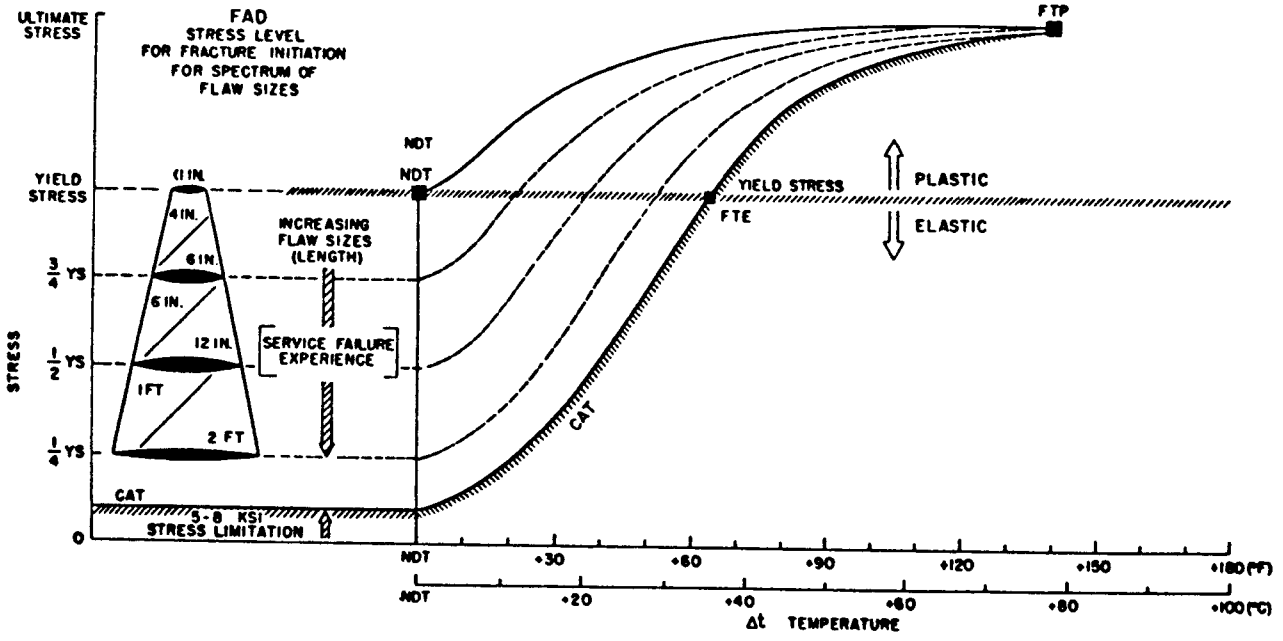


Fig. 44 Fracture-analysis diagram (FAD). Note that the stress level for plastic (over yield) fracture is not indexed because of the lack of analytical procedures for its definition. Ultimate stress signifies only that maximum load and strain tolerance is attained at FTP for the specific flaw size cited. It obviously does not indicate the equivalent of the tensile test specimen maximum load or maximum strain limits [56]

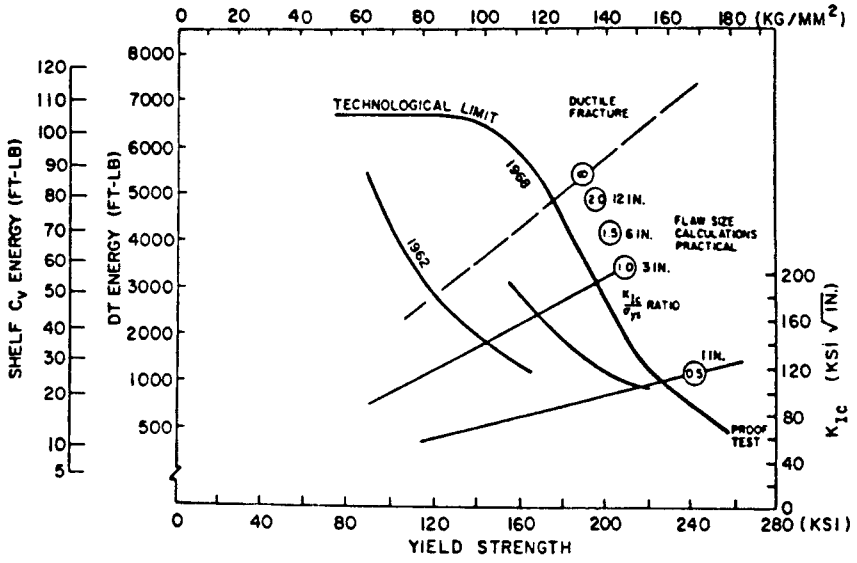


Fig. 45 Ratio-analysis program (RAD). The grid of K_{Ic}/ys ratio lines indexes the region of the diagram which relates to the plain strain fracture [56]

the flaw size, relative stress, and temperature relationships by a *t* or "temperature increment" reference to the nil-ductility temperature (NDT), below which a flawed material fractures without any noticeable plastic deformation. The localizing of the generalized diagram to specific positions in the temperature scale requires the experimental determination of a single parameter—the NDT. The NDT can be determined by the dynamic tear (DT) test.

Definitions of important terms used in the FAD are (see Fig. 44)

CAT: Robertson **C**rack **A**rrest **T**emperature transition curve

FTE: **F**racture **T**ransition **E**lastic, or the highest possible temperature for unstable fracture propagation through elastic stress fields

FTP: **F**racture **T**ransition **P**lastic, or the temperature point of fully ductile tearing

Pellini has used the FAD to explain many fracture problems, including analyses of failures in actual structures [56].

Ratio-analysis diagram (RAD): Pellini has also developed the ratio-analysis diagram (RAD), as illustrated in Fig. 45 [56]. The RAD has been developed to establish a relationship between the fracture-mechanics theory and the FAD concept, or stated differently, the relationship between the plane-strain critical stress-intensity factor, K_{Ic}, and results obtained by the dynamic tear test. As shown in the figure, the ratio of K_{Ic} to the yield stress, σ_{ys}, plays an important role in defining the fracture toughness of a material, for K_{Ic}/σ_{ys} is closely connected to the ductility or plastic zone size developed at the tip of a crack before the onset of unstable fracture.

The RAD may be separated into three general regions. The top region, above the ∞ ratio line, relates to ductile fracture. The bottom region, below the 0.5 ratio line, relates to low levels of plane-strain fracture toughness, involving flaw sizes too small for reliable inspection. These are generally in the order of tenths of inches for stubby flaws decreasing to hundredths of inches for long, thin flaws subjected to high elastic stress levels. Fracture mechanics applies well in principle for this region, but poorly in practice due to the flaw detection problem. This region requires proof-test procedures for ascertaining pre-service structural integrity.

The width of the remaining "slice" of the RAD, the central region for which plane-strain fracture-mechanics calculations appear feasible, depends on the section size. For very thick sections (up to over 10 in.), the applicable region available for possible plane-strain calculations is between the 0.5 and ∞ ratio lines. For section sizes in the order of 1- to 3-in. thickness, this region is bounded by the 0.5 and (approximately) the 1.0 ratio lines. This range represents a very thin portion of the population of steels represented in the diagram.

RAD is very useful in selecting structural materials since the plot of data for one material clearly shows the comparative quality among various materials. This type of representation will be presented for individual materials in Section 3.

3. Structural Materials

3.1 Steels

This section discusses various steels that have been used or may be used for marine structures,

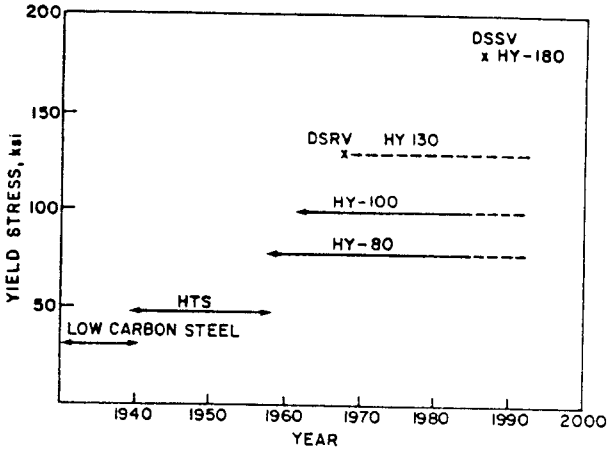


Fig. 46 Use of high-strength steels for U.S. Navy submarines and submersibles

especially deep submersibles. These steels include low-carbon steels, high-strength, low-alloy (HSLA) steels, and quenched and tempered (Q&T) steels. Detailed discussions on these steels may be found in Technical and Research Bulletin 2-19 "Higher-Strength Steels in Hull Structures" [57], published by SNAME, and several other documents [1,58,59].

Discussions in this chapter are presented in three sections:

- historical perspective and overview,
- low-carbon steels and high-strength steels with less than 80-ksi (552 MPa) yield strengths, and
- high-strength steels with 80- to 120-ksi (552 to 827 MPa) yield strengths.

The first part of the first section discusses how the yield strengths of materials used for fabricating marine structures, especially U.S. Navy submarines, have progressed over the years. We are facing considerable difficulties in increasing the level of yield strength over 120 ksi. The second part of the first section discusses some of the major difficulties including how to maintain good fracture toughness of the base metal while increasing its yield strength and how to develop technologies for welding such a steel. Discussions in this section cover a wide range of steels, including those with yield strengths over 120 ksi and some recently developed steels.

The second and third sections cover basic information on some steels that are widely used today: the second section covers steels with yield strengths less than 80 ksi, while the third section covers steels with 80- to 120-ksi yield strengths.

3.1.1 Historical Perspective and Overview

Uses of high-strength steels for submarines and deep submersibles—Figure 46 shows how the yield strengths of materials used for pressure hulls of U.S. Navy submarines and deep submersibles have increased over the years [1,58,59]. Prior to the early 1940's, combat submarines were fabricated mainly with low-carbon steel, a material with a tensile yield strength of approximately 32 000 psi (221 MPa). Between 1940 and 1958, high-tensile steel (HTS) with a 50 000 psi (345 MPa) yield strength was used in most submarine structures. In 1958, HY-80 steel, a quenched and tempered (Q&T) steel with a minimum specified yield strength of 80 000 psi (552 MPa), was first introduced to submarine hulls [60]. Several years later, HY-100, a Q&T steel with a minimum specified yield strength of 100 000 psi (689 MPa) and similar to HY-80, was introduced. Today, HY-80 and HY-100 are the basic steels for submarine hulls.

The next steel in line is HY-130, a Q&T steel with a minimum specified yield strength of 130 ksi (896 MPa). In 1969, the first Deep Submergence Rescue Vehicle (DSRV) was fabricated by Lockheed Missile and Space Company using HY-130. DSRV is capable of diving to a depth of 6000 ft (1830 m). The U.S. Navy has been interested in using HY-130 for future submarines. The U.S. Navy also has a plan to build a Deep Submergence Search Vehicle (DSSV) with a depth capability of 20 000 ft (6100 m); the material being considered is HY-180 steel.

Applications of high-strength steels to commercial structures, including ships, offshore oil drilling rigs, bridges, and pressure vessels, occurred several years later; and most applications have been limited to steels up to 100 000 psi. Besides the Navy's HY-80 and HY-100 steels, there are a number of commercial Q&T steels such as ASTM A 514/517. These steels have excellent fracture toughness at low temperatures, and they have been extensively used for various structures.

As stated above and shown in Fig. 46, the use of high-strength steels with yield strengths exceeding 120 000 psi (827 MPa) has been very limited. Stated differently, most marine structures today are still manufactured from steels with yield strengths of less than 120 000 psi. The primary reasons for this are twofold:

- For any type steel, fracture toughness decreases as the yield strength increases. Therefore, we must develop higher quality steels in order to increase the strength level and yet maintain good fracture toughness.
- As the strength level increases, it becomes increasingly difficult to successfully fabricate welded structures using these steels.

An overview of these problems is presented in the following pages.

In addition, as the strength level of a steel increases, the following problems also become increasingly serious:

- When higher strength steels are used, design stresses are often increased. Then, structures fabricated with these steels will become increasingly vulnerable to fatigue fractures.

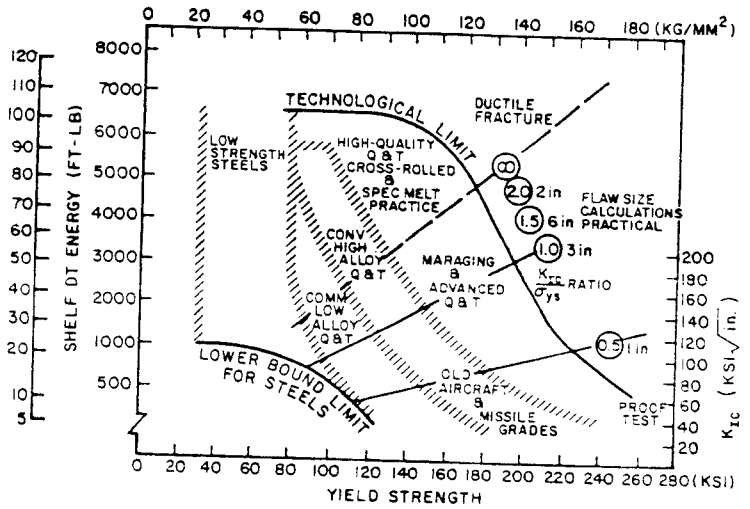


Fig. 47 Metallurgical zoning of the ratio analysis diagram which defines the general effects of melting and processing factors on the strength transition. The three corridors of strength transition relate to metallurgical quality (void site density) which controls micro-fracture processes and, thereby, the macroscopic fracture toughness of the metal. The location of generic alloy steel types are indicated by the notations

- As the strength level increases, steels become increasingly sensitive to stress corrosion cracking and hydrogen-induced delayed cracking.

Some discussions on these subjects are presented in the first pages of this chapter.

Fracture toughness of a base material—The most fundamental problem in the development of a high-strength material is that there is a general tendency for the fracture toughness, or notch toughness, of a material to decrease as its strength level increases. In other words, a material tends to become less ductile, or more brittle, as one tries to increase its strength.

Figure 47 shows relationships between the yield strength and fracture toughness for various types of steel [56]. For any type of steel, fracture toughness decreases as the strength level increases. Therefore, when one tries to develop a steel with higher strength without losing fracture toughness, one must develop a higher-quality steel by doing one or more of the following:

- increase certain alloying elements,
- reduce undesired impurities, or
- use certain rolling techniques and heat treatments.

An element most commonly used for increasing fracture toughness is nickel. Although carbon is a very effective element for increasing strength, it also causes a significant reduction in fracture toughness. In addition, as the carbon content increases, it becomes increasingly difficult for the steel to be welded without cracking. As a result, as the strength level of a steel is increased, its carbon content is generally decreased.

In developing a high-strength steel with good fracture toughness, it is important to reduce impurities, including phosphorus and sulfur. Reduction of phosphorus and sulfur is also important for preventing weld cracking. Special steelmaking techniques such as vacuum melting are often used.

Table 6 compares some high-strength steels, including ABS AH36, HY-80, HY-130, and HY-180 steel. ASTM A710 (Grade A, Class 1) is added as an example of the newly developed steels. As the strength level increases, the nickel content is increased, while maximum allowable amounts of carbon, phosphorus, and sulfur are decreased. ASTM A710 steel has a rather low amount of carbon, as shown in Table 6. There is a new trend in making high-strength steels that contain very low amounts of carbon, while required strengths are obtained through increasing certain alloying elements, using controlled-rolling techniques, or using special heat treatments, or some combination of these. These steels are often referred to HSLA steels [61].

The controlled-rolling technique is a recently developed technique in which steel plates are rolled at relatively low temperatures by carefully controlling rolling operations [62,63]. This technique is also called "thermomechanical rolling."

A number of heat treatments have been used to improve fracture toughness. They include normalizing, quench and temper, precipitation treatments, and maraging treatments. Quenching followed by tempering is commonly used for producing steels with high strength and excellent fracture toughness. Quenching a steel first produces a material with very high strength but rather poor ductility. Through

Table 6 Comparison of some high-strength steels

Type	Minimum Yield Strength, ksi	Contents of Some Element, percentage				Heat Treatment
		C max.	P max.	S max.	Ni approx.	
ABS AH36	51	0.18	0.04	0.04	—	—
HY-80	80	0.18	0.25[a]	0.025[a]	3	Q&T
HY-130	130	0.12	0.010	0.015	5	Q&T
HY-180	180	0.03	NA[b]	NA[b]	9	Q&T
ASTM A710 Grade A, Class 1	80	0.07	0.025	0.025	0.9	Precipitation

[a]The total amount of phosphorus and sulfur must not exceed 0.045 percent.
[b]Not available.

tempering, the strength decreases slightly but the ductility is increased significantly.

Welding considerations—Important considerations in developing high-strength steels which must be fabricated by welding are

- development of a steel which is easy to weld and
- development of (a) welding materials and (b) welding processes and procedures most suitable for the steel being developed.

The emphasis of the discussions presented here is placed on subjects related to welding metallurgy rather than welding processes and procedures. There are three important considerations in the welding of high-strength steels:

- how to develop welding materials (and processes) which can produce a weld metal that can match the base metal in both strength and fracture toughness,
- how to develop a technique for welding the material without cracking by either controlling welding procedures or developing proper materials, and
- how to minimize losses of strength and fracture toughness of the heat-affected base metal.

Fracture toughness of the weld metal: In welding any material, the goal is to produce a weld metal that has the same metallurgical and mechanical properties as those of the base material. In welding an ordinary carbon steel, it is not difficult to obtain a weld metal that can match the base metal in both strength and fracture toughness. However, as the strength level of the base metal increases, it becomes increasingly difficult to obtain a weld metal that can match the base metal in both strength and fracture toughness. Some discussions on notch toughness of weld metals are presented in the first pages of this chapter. So far it has not yet been possible to develop welding electrodes which produce weld metals as notch-tough as the HY-80 steel base metal. There are basically two reasons why it is difficult to obtain a weld metal that has a fracture toughness comparable to that of a quenched and tempered steel:

- The weld metal is as-welded, while excellent properties of the base metal are obtained through carefully controlled heat treatments, and
- Since a weld metal is produced through solidification of metal particles transferred in the welding arc, the weld metal tends to contain higher amounts of gases, including nitrogen, oxygen, and hydrogen, compared with those contained in the base metal. These impurities reduce fracture toughness.

Figure 48 shows relationships between the yield strength and fracture toughness of weld metals produced by different welding processes [56]. Fracture toughness of weld metals produced by the shielded metal arc (SMA) process decreases rapidly as the strength level increases. When using covered electrodes, it is not possible to produce weld metals with excellent fracture toughness when the strength level is around 120 000 psi (827 MPa) or higher. This causes a serious problem in fabricating structures in HY-130 steel. One way to solve the problem is to use gas metal-arc (GMA) welding, or even gas tungsten-arc (GTA) welding. However, the problem is that welding of heavy plates with the GTA process can be very time consuming.

Control of materials and welding procedures for preventing weld cracking and minimizing degradation of the heat-affected base metal: Among the various necessary considerations for successful welding fabrication of welded steel structures, two most important are how to avoid weld cracking (both hot cracking and cold cracking) and how to minimize degradation of the HAZ, as stated earlier. These subjects are discussed in detail in a number of publications [1,57–61,64–67].

Hot Cracking: Hot cracking occurs at temperatures near the melting temperature, and cracks are usually intergranular, or cracks occur between grains. Hot cracks are found in the weld metal and the HAZ adjacent to the weld fusion line. Hot cracking is strongly dependent on the chemical composition of the base metal. Inagaki [66] has suggested a formula for evaluating the hot cracking sensitivity [HCS] of an HSLA steel:

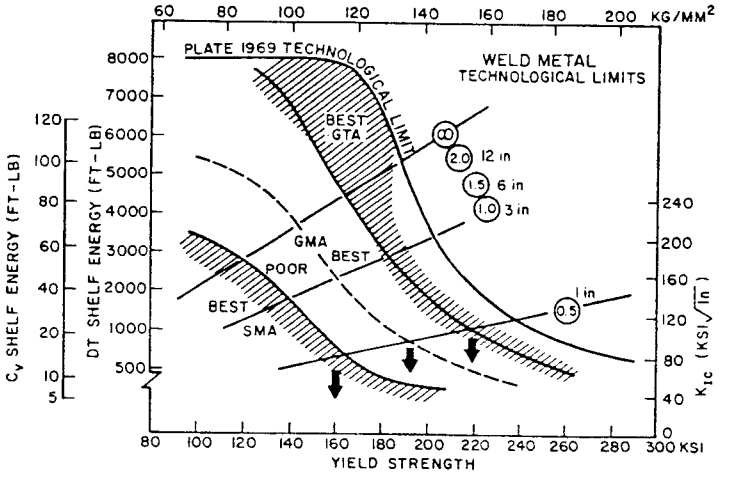


Fig. 48 Zoning of the ratio analysis diagram which defines weld metal factors of first order importance to strength transition features. The strength transition corridors relate to weld metal deposited by metal arc (SMA), inert gas shielded metal arc (GMA), and inert gas shielded tungsten arc (GTA). The bold arrows indicate the strength level of transition to the 0.5 ratio value

$$[HCS] = \frac{C\left[S + P + \frac{Si}{25} + \frac{Ni}{100}\right]}{3Mn + Cr + Mo + V} \times 10^3$$

Chemical compositions shown above are given in percent. Those elements shown in the numerator (carbon, sulfur, phosphorus, silicon, and nickel) are considered to promote hot cracking, while those elements shown in the denominator (manganese, chromium, molybdenum, and vanadium) are considered to retard hot cracking. When the [HCS] value is less than 4, the steel is believed to have good resistance against hot cracking.

Cold Cracking: Cold cracking occurs at temperatures below 400°F (204°C), and cracks are intergranular, that is, the cracks occur through grains. Cold cracks are found in the weld metal and HAZ. Cold cracks even extend into the unaffected base metal. Cold cracks in steel are hydrogen-induced delayed cracks, and they occur as late as one week after welding. Methods for preventing cold cracking include

- proper selection of the chemical composition of the base metal,
- control of the hydrogen content in the welding atmosphere, and
- preheating.

The concept of carbon equivalent (CE) of the base metal has been used for preventing cold cracking. A formula given in the *Welding Handbook* [64] is

$$CE = C + \frac{Mn}{4} + \frac{Ni}{20} + \frac{Cr}{10} + \frac{Cu}{40} - \frac{Mo}{50} - \frac{V}{10}$$

Chemical compositions shown above are given in percent. Those elements preceded by plus signs (carbon, manganese, nickel, chromium, and copper) promote cold cracking, while those elements preceded by minus signs (molybdenum and vanadium) retard cold cracking. When the CE value is less than 0.40, the steel is believed to have good resistance against cold cracking.

The probability of causing cold cracking can be reduced by the use of welding processes or welding electrodes containing a lesser amount of hydrogen. The hydrogen content of the welding atmosphere can be held at a very low level by the proper uses of GMA and GTA processes. Low-hydrogen-type electrodes have been developed to control the hydrogen content in shielded-metal arc (SMA) welding. The term "low-hydrogen" has been developed because these electrodes are produced with a maximum limit of moisture in the electrode coating. The specified maximum moisture content of low-hydrogen electrodes with different strength levels are

0.6% for E7015, E7016, E7018
0.4% for E80XX
0.2% for E100XX and E110XX

It must be stressed that low-hydrogen electrodes must be stored properly in order to maintain these low amounts of hydrogen.

An important operational problem, especially in a shipyard, is that the electrodes will start absorbing moisture as soon as they are exposed to a humid atmosphere. Figure 49 shows the moisture pickup by E11018G electrodes exposed to an atmosphere of 90% relative humidity at 75°F (24°C) [65]. According to a study by Mishler [67], if the welder does not have a heated electrode container, the electrode should be used within the following time limits:

E70XX	4 h
E80XX	2 h
E90XX	1 h
E110XX	½ h

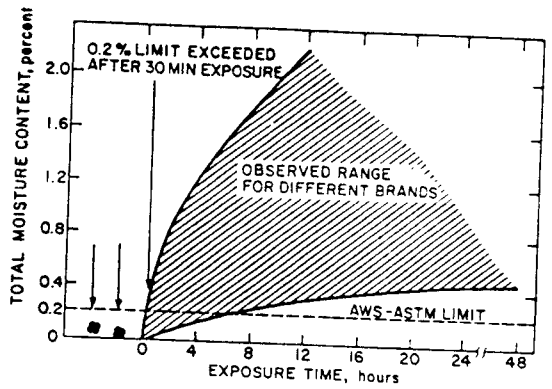


Fig. 49 Moisture pickup by E11018-G electrodes exposed to an atmosphere of 90% relative humidity at 75°F

If there is any doubt about the moisture level of electrodes, they can be rebaked at approximately 800°F (427°C) for a total time at that temperature of 30 min to 1 h before use.

Control of HAZ Microstructures: There are basically two important things in controlling metallurgical structures of the HAZ:

- to prevent too rapid cooling which results in very hard (and brittle) structures, and
- to prevent too slow cooling which could destroy the excellent properties of the base metal, especially those of heat-treated materials (such as a Q&T steel).

In order for both to be accomplished, the heat input used in welding a high-strength steel is controlled within a limit, and this limit becomes smaller as the strength level of the material increases. Figure 50 shows control schemes developed for some Japanese steels; their chemical compositions and some mechanical properties are shown in Table 7 [59]. These steels are similar to the HY-steels developed in the United States. Therefore, NS30, NJ46, NJ53, NS80, and NS90 steels may be regarded as equivalent to HY-43, HY-65, HY-90, HY-100, and HY-130, respectively, although some of the above designations do not really exist.

Table 7 Specifications for Japanese high-strength steels for submarine application[a]

Steel	Chemical Composition (weight percentage)										Yield Strength, kg/mm^2
	C	Si	Mn	P	S	Cu	Ni	Cr	Mo	V	
NS30	≤0.16	≤0.55	≤1.35	≤0.040	≤0.040	—	—	—	—	—	>30
NS46	≤0.18	≤0.55	≤1.45	≤0.035	≤0.040	≤0.30	≤0.60	≤0.15	≤0.15	≤0.15	>46
NS63	≤0.16	0.15 to 0.40	≤0.80	≤0.025	≤0.025	≤0.25	2.00 to 3.50	0.30 to 1.20	0.20 to 0.60	≤0.03	>63
NS80	≤0.12	0.15 to 0.40	0.35 to 1.00	≤0.015	≤0.015	≤0.15	3.50 to 4.50	0.30 to 1.00	0.20 to 0.60	0.03 to 0.10	>80
NS90	≤0.10	0.15 to 0.40	0.35 to 0.80	≤0.010	≤0.010	≤0.10	5.00 to 5.50	0.30 to 0.80	0.20 to 0.60	0.03 to 0.10	>90

[a]Specification requirements on some items are not included in this table.

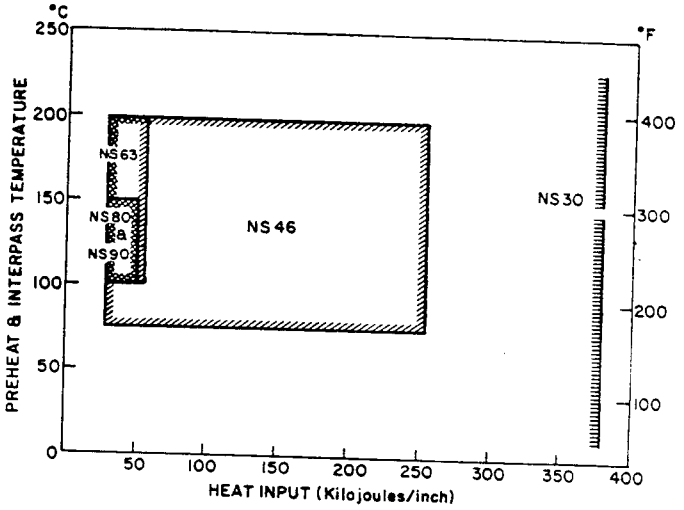


Fig. 50 Practical welding conditions for some Japanese high-strength steels (see Table 7 for details of the steels)

Shown in the abscissa of Fig. 50 are upper and lower limits of welding heat input for preventing too slow cooling and too rapid cooling, respectively, of the HAZ. Shown in the ordinate are upper and lower limits of preheating and interpass temperatures. Preheating of a welded joint to a certain temperature is necessary for preventing cold cracking. At the same time, preheating and interpass temperatures must be kept below a certain limit to avoid excessively slow cooling for minimizing degradation of the HAZ. Figure 50 shows that, as the strength level of the material increases, increasingly precise controls of both welding heat input and preheating and interpass temperatures are needed.

3.1.2 Low-Carbon Steels and High-Strength Steels with Less Than 80-ksi Yield Strengths

Base material—Low-carbon steels and high-strength steels with less than 80-ksi (552 MPa) yield strengths are among the most widely used materials for marine structures.

Ship steels: In this section, discussions are made, mainly in reference to ship-hull steels, because most ship hull steels are applicable to many marine structures including deep submersibles. U.S. merchant vessels are constructed in accordance with requirements established by the U.S. Coast Guard and the American Bureau of Shipping (ABS). Naval combatant vessels and many noncombatant naval vessels are constructed in accordance with U.S. Navy specifications.

Requirements by the ABS can be found in its *Rules for Building and Classing Steel Vessels*, which is revised annually. The 1984 edition of the Rules includes details of the requirements for steels to be used for ship hull structures in Section 43, Materials for Hull Construction and Equipment. Presented below are some of the tables included in Section 43:

Table 3: Requirements for ordinary-strength hull structural steel, 51 mm (2 in.) and under—Grades A, B, D, E, DS, CS.
Table 4: Requirements for higher-strength hull structural steel, 51 mm (2in.) and under—Grades AH32, DH32, EH32, AH36, DH36, and EH36.
Table 5: Requirements for low temperature materials, Grades V-039, V-051, V-060, VH-039, VH-051, and VH-060 and min imum service temperature.

When the plate thickness exceeds 51 mm (2 in.), a steel must satisfy more stringent requirements than those imposed on the same grade steel 51 mm or less in thickness (refer to the ABS Rules for further details). ABS also specifies how these steels should be used for various structural members (refer to the ABS Rules).

As can be seen in Tables 8 through 10 of this text, the present ABS steels are classified into three groups:

- ordinary-strength steels with the minimum yield strength of 24 kg/mm^2 (34 000 psi),
- higher-strength steels in two levels of yield strength: 32 kg/mm^2 (45 500 psi) and 36 kg/mm^2 (51 000 psi), and
- low-temperature materials with two levels of yield strength: 24 kg/mm^2 (34 000 psi) and 36 kg/mm^2 (51 000 psi).

Ordinary-Strength Steels: Ordinary-strength steels are further classified into six grades (A, B, D, E, DS, and CS), depending upon the deoxidation process used during the steelmaking heat treatment, as requirements for fracture toughness (see Table 8). For the deoxidation process, any method is accepted for Grade A steel plate less than 12.5 mm (½ in.) thick, while any method except rimmed steel is accepted for Grade A steel 12.5 mm and over and for Grade B steel in any thickness. Fully killed fine-grain practice is required for other grades, including D, E, DS, and CS. Normalized heat treatments are required for Grades D and DS steels over 35 mm (1⅜ in.) thick and Grades E and CS steels for any thickness. For plates 51 mm (2 in.) and under, fracture toughness is required only for Grades B, D, and E. Fracture toughness is evaluated by Charpy V-notch impact specimens performed at different temperatures:

0°C (32°F) for Grade B steel
−10°C (14°F) for Grade D steel
−40°C (−40°F) for Grade E steel

Charpy V-notch impact tests are performed on three specimens, and the average value of energy absorption must not be less than

- 2.8 kg/m (20 ft/lb) for longitudinal specimens of which longitudinal axis is parallel to the rolling direction of the plate and
- 2.0 kg/m (14 ft/lb) for transverse specimens of which longitudinal axis is transverse to the rolling direction.

Further details of performing tests are described in the ABS Rules. When plates are thicker than 51 mm (2 in.), fracture toughness tests are required for all grades, including A, B, D, DS, CS, and E.

Higher-Strength Steels: Higher-strength steels are further classified into six grades (AH32, DH32, EH32, AH36, DH36, and EH36), depending upon the strength and fracture toughness. AH32, DH32, and EH32 steels have a minimum specified yield strength of 32 kg/mm^2 (45 500 psi), while AH36, DH36, and EH36 steels have a minimum specified yield strength of 36 kg/mm^2 (51 000 psi). Charpy V-notch impact tests for fracture toughness are performed at the following temperatures:

0°C (32°F) for AH32 and AH36 steels
−20°C (−4°F) for DH32 and DH36 steels
−40°C (−40°F) for EH32 and EH36 steels.

Table 8 Requirements for ordinary-strength hull structural steel Grades A, B, D, E, DS, and CS 51 mm (2 in.) and under

Grades	A	B	D	E	DS	CS
			Deoxidation			
	Any method except rimmed steel for material 12.5 mm (0.5 in.)	Any method except rimmed steel	Fully killed fine-grain practice[a]	Fully killed fine-grain practice	Fully killed fine-grain practice	Fully killed fine-grain practice
			Chemical Composition[b] (Ladle Analysis)			
Carbon %	0.23 max.[c]	0.21 max.	0.21 max.	0.18 max.	0.16 max.	0.16 max.
Manganese %	2.5 × carbon min. for plates over 12.4 mm (0.5 in.)	0.80–1.10 (0.60 min. for fully killed or cold flanging)	0.70–1.35 (0.60 min. for 25 mm (1.0 in.) and under thickness)	0.70–1.35	1.00–1.35	1.00–1.35
Phosphorus %	0.04 max.	0.04 max.	0.04 max.	0.04 max.	0.04 max.	0.04 max.
Sulphur %	0.04 max.	0.04 max.	0.04 max.	0.04 max.	0.04 max.	0.04 max.
Silicon %	—	0.35 max.	0.10–0.35	0.10–0.35	0.10–0.35	0.10–0.35
			Tensile Test			
Tensile Strength	For all Grades: 41–50 kg/mm^2 (58 000–71 000 psi); for Grade A shapes and bars 41–56 kg/mm^2 (58 000–80 000 psi). For cold flanging quality: 39–46 kg/mm^2 (55 000–65 000 psi)					
Yield Point, min.	For all Grades: 24 kg/mm^2 (34 000 psi); for Grade A over 25.0 mm (1.0 in.) in thickness 23 kg/mm^2 (32 000 psi). For cold flanging quality: 21 kg/mm^2 (30 000 psi)					
Elongation, min.	For all Grades: 21% in 200 mm (8 in.) or 24% in 50 mm (2 in.) or 22% in 5.65 $\sqrt{A}$ (A equals cross-sectional area of test specimen). For cold flanging quality: 23% min. in 200 mm (8 in.)					
			Impact Test—Charpy V-Notch			
Temperature		0°C (32°F) Over 25 mm (1.0 in.)	–10°C (14°F)	–40°C (–40°F)		
Energy Avg. Min.						
Longitudinal Specimens		2.8 kg·m (20 ft·lb)	2.8 kg·m (20 ft·lb)	2.8 kg·m (20 ft·lb)		
Transverse Specimens		2.0 kg·m (14 ft·lb)	2.0 kg·m (14 ft·lb)[d]	2.0 kg·m (14 ft·lb)		
			Heat Treatment			
			Normalized over 35 mm (1.375 in.) thick[e]	Normalized[f]	Normalized over 35 mm (1.375 in.) thick and mark as CS	Normalized
			Marking			
	AB/A	AB/B	AB/D[g]	AB/E	AB/DS	AB/CS

[a]Grade D may be furnished semi-killed in thickness up to 35 mm (1.375 in.) provided steel above 25.0 mm (1.00 in.) in thickness is normalized. In this case the requirements relative to minimum Si and Al contents and fine grain practice do not apply.

[b]For all grades exclusive of Grade A shapes and bars, the carbon content plus one-sixth of the manganese content is not to exceed 0.40 percent. The upper limit of manganese may be exceeded up to a maximum of 1.65 percent provided that this condition is satisfied.

[c]A maximum carbon content of 0.26 percent is acceptable for Grade A plates equal to or less than 12.5 mm (0.5 in.) and all thicknesses of Grade A shapes and bars.

[d]Impact tests are not required for normalized Grade D steel when furnished fully killed fine-grain practice.

[e]Controlled rolling or thermo-mechanical controlled rolling of Grade D steel may be specially considered as a substitute for normalizing.

[f]Controlled rolling or thermo-mechanical controlled rolling of Grade E shapes may be specially considered as a substitute for normalizing.

[g]Grade D hull steel which is normalized, thermo-mechanical controlled rolled, or controlled rolled in accordance with Note (*e*) is to be marked AB/DN.

Charpy impact tests are performed on three specimens, and the average value of energy absorption must not be less than

- 3.5 kg/m (25 ft/lb) for longitudinal specimens and
- 2.4 kg/m (17 ft/lb) for transverse specimens.

Low-Temperature Materials: Chapter 6 of the IMCO (Inter-Governmental Maritime Consultative Organization) "Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk" (IMCO Gas Code) is the governing document for material selection. This part of the ABS Rules covers requirements for specific approved materials for cargo tanks and secondary barriers carrying liquefied, low-temperature cargoes. Low-temperature materials currently included in the ABS Rules are classified into six grades (V-039, VH-039, V-051,

Table 9 Requirements for higher-strength hull structural steel Grades AH32, DH32, EH32, AH36, DH36, and EH36 51 mm (2 in.) and under

Grades[a]	AH32	DH32	EH32	AH36	DH36	EH36
Process of Manufacture						
	For all grades: Open hearth, basic oxygen, or electric furnace					
Deoxidation						
	Semi-killed or killed[b]	Fully killed fine-grain practice[c]	Fully killed fine-grain practice[c]	Semi-killed or killed[b]	Fully killed fine-grain practice[c]	Fully killed fine-grain practice[c]
Chemical Composition for All Grades (Ladle Analysis)						
Carbon, %	0.18 max.					
Manganese, %[d]	0.90–1.60					
Phosphorus, %	0.04 max.					
Sulphur, %	0.04 max.					
Silicon, %	0.10–0.50					
Nickel, %	0.40 max.	These elements need not be reported on the mill sheet unless intentionally added.				
Chromium, %	0.25 max.					
Molybdenum, %	0.08 max.					
Copper, %	0.35 max.					
Columbium (Niobium), %	0.05 max.					
Vanadium, %	0.10 max.					
Tensile Test						
Tensile Strength	48–60 kg/mm^2; (68 000–85 000 psi)			50–63 kg/mm^2; (71 000–90 000 psi)		
Yield Point, min.	32 kg/mm^2; (45 500 psi)			36 kg/mm^2; (51 000 psi)		
Elongation, min.	For all Grades: 19% in 200 mm (8 in.) or 22% in 50 mm (2 in.) or 20% in 5.65 $\sqrt{A}$ (A equals cross-sectional area of test specimen).					
Impact Test—Charpy V-Notch						
Temperature	0°C (32°F)	–20°C (–4°F)	–40°C (–40°F)	0°C (32°F)	–20°C (–4°F)	–40°C (–40°F)
Energy avg. min.						
Longitudinal Specimens	3.5 kg·m (25 ft·lb)[e]	3.5 kg·m (25 ft·lb)[e]	3.5 kg·m (25 ft·lb)	3.5 kg·m (25 ft·lb)[e]	3.5 kg·m (25 ft·lb)[e]	3.5 kg·m (25 ft·lb)
Transverse Specimens	2.4 kg·m (17 ft·lb)[e]	2.4 kg·m (17 ft·lb)[e]	2.4 kg·m (17 ft·lb)	2.4 kg·m (17 ft·lb)[e]	2.4 kg·m (17 ft·lb)[e]	2.4 kg·m (17 ft·lb)
Heat Treatment						
	For all grades: Refer to ABS Rules					
Marking						
	AB/AH32	AB/DH32[f]	AB/EH32	AB/AH36	AB/DH36[f]	AB/EH36

[a]The numbers following the Grade designation indicate the yield point or yield strength to which the steel is ordered and produced in kg/mm^2.
[b]Grade AH plates to 12.5 mm (0.50 in.) inclusive may be semi-killed, in which case the 0.10 percent minimum silicon does not apply. Unless otherwise especially approved, Grade AH plates over 12.5 mm (0.50 in.) are to be killed with 0.10 to 0.50 percent silicon.
[c]Grades DH and EH are to contain at least one of the grain refining elements in sufficient amount to meet the fine grain practice requirement.
[d]Grade AH 12.5 mm (0.50 in.) and under in thickness may have a minimum manganese content of 0.70 percent.
[e]Impact tests are not required for AH, 12.5 mm (0.50 in.) and less in thickness, and aluminum treated Grade AH, 35 mm (1.375 in.) and less in thickness. Impact tests are not required for any fully killed, fine grain normalized Grade AH, or DH, 51 mm (2 in.) and less in thickness.
[f]The marking AB/DHN is to be used to denote Grade DH plates which have either been normalized thermo-mechanical controlled rolled or control rolled in accordance with an approved procedure.

VH-051, V-060, and VH-060), depending upon their yield strengths and fracture toughnesses. V-039, V-051, and V-060 steels have a minimum specified yield strength of 24 kg/mm^2 (34 000 psi), while VH-039, VH-051, and VH-060 steels have a minimum specified yield strength of 36 kg/mm^2 (51 000 psi). The Charpy V-notch impact tests for fracture toughness are performed at the following temperatures:

–39°C (–38°F) for V-039 and VH-039 steels
–51°C (–60°F) for V-051 and VH-051 steels
–60°C (–76°F) for V-060 and VH-060 steels.

Minimum service temperatures for these steels are

–34°C (–30°F) for V-039 and VH-039 steels
–46°C (–50°F) for V-051 and VH-051 steels
–55°C (–67°F) for V-060 and VH-060 steels.

Table 10 Requirements for low-temperature materials, Grades V-039, V-051, V-060, VH-039, VH-051, and VH-060 and minimum service temperature

Grades	V-039	V-051	V-060	VH-039	VH-051	VH-060
Chemical Composition for All Grades (Ladle Analysis)						
Carbon, max., %	0.20	0.16	0.12	0.20	0.16	0.12
Manganese, %	0.90–1.35	1.15–1.50	1.30–1.65	0.90–1.35	1.15–1.50	1.30–1.65
Phosphorus, max., %	0.04	0.04	0.04	0.04	0.04	0.04
Sulphur, max., %	0.04	0.04	0.04	0.04	0.04	0.04
Silicon, %	0.10–0.50	0.10–0.50	0.10–0.50	0.10–0.50	0.10–0.50	0.10–0.50
Nickel, max., %	0.80	0.80	0.80	0.80	0.80	0.80
Chromium, max., %	0.25	0.25	0.25	0.25	0.25	0.25
Molybdenum, max., %	0.08	0.08	0.08	0.08	0.08	0.08
Copper, max., %	0.35	0.35	0.35	0.35	0.35	0.35
Aluminum (acid soluble), max., %	0.060	0.060	0.060	0.060	0.060	0.060
Aluminum (total), max., %	0.065	0.065	0.065	0.065	0.065	0.065
Columbium (Niobium), max., %	0.05	0.05	0.05	0.05	0.05	0.05
Vanadium, max., %	0.10	0.10	0.10	0.10	0.10	0.10
Tensile Test						
Tensile Strength	41–50 kg/mm^2; (58 000–71 000 psi)			50–63 kg/mm^2; (71 000–90 000 psi)		
Yield Strength, min.	24 kg/mm^2; (34 000 psi)			36 kg/mm^2; (51 000 psi)		
Yield Strength, max.	80% of tensile strength			80% of tensile strength		
Elongation, min.	21% in 200 mm (8 in.) or 24% in 50 mm (2 in.) or 22% in 5.65 $\sqrt{A}$ (A equals cross-sectional area of test specimen)			19% in 200 mm (8 in.) or 22% in 50 mm (2 in.) or 20% in 5.65 $\sqrt{A}$ (A equals cross-sectional area of test specimen)		
Charpy V-Notch Impact Test Temperature						
	–39°C (–38°F)	–51°C (–60°F)	–60°C (–76°F)	–39°C (–38°F)	–51°C (–60°F)	–60°C (–76°F)
Minimum Service Temperature[a]						
	–34°C (–30°F)	–46°C (–50°F)	–55°C (–67°F)	–34°C (–30°F)	–46°C (–50°F)	–55°C (–67°F)
Marking						
	AB/V-039	AB/V-051	AB/V-060	AB/VH-039	AB/VH-051	AB/VH-060

[a]At service temperatures intermediate between those listed above, intermediate amounts of carbon and manganese will be allowed, provided all other chemical, tensile, and impact requirements are met. The grade of these steels would be the letter "V" or "VH" followed by the test temperature to the nearest degree Celsius. A prefix "0" to the test temperature is to indicate a temperature lower than 0°C. The specified upper limit of the manganese may be exceeded up to a maximum of 1.65 percent provided that for each 0.06 percent increase of manganese the specified maximum carbon content is to be decreased by 0.01 percent. Any steel acceptable for a particular minimum service temperature is also acceptable for higher service temperatures.

Various structural steels: In addition to the above steels, which have been developed for ship hull applications, there are many other steels which can be used or have been used for some marine applications. Tables 11 and 12 provide lists of various steels which may be used for marine applications [1,68].

Steels plates and shapes listed in these tables are for ship structures requiring an increase of strength not originally available with standard ship steels. The values tabulated are for steels of 1-in. thickness and may vary with mill practice. Further details of these steels are available in the ASTM or manufacturers' specifications. Steels with less than 60 000 psi (414 MPa) minimum yield strength are usually available in the as-rolled or normalized condition, whereas steels with minimum yield strength over 60 000 psi are usually available in the quenched-and-tempered condition. This trend is true for steels with minimum yield strength over 80 000 psi (552 MPa), as discussed later (see Table 14).

Welding—Some discussions on welding processes are presented in the beginning of this chapter. This section discusses briefly the welding processes used for fabricating structures in low-carbon steels and high-strength steels with less than 80-ksi yield strengths. Table 13 lists welding electrodes and processes that can be used for joining carbon steel and high-tensile steel, as specified by NAVSHIPS 0900-060-4010 [69]. The electrodes and processes listed here can be regarded as typical electrodes and processes which can be used for low-carbon steels and high-strength steels with yield strengths of up to 80 ksi.

Welding of low-carbon steels: Shielded-metal arc (SMA) welding continues to be the major welding process for fabricating ships and other large struc-

Table 11 Steels, 30 000 to 49 000 psi, yield strength minimum [68]

Type	For Reference ABS Class "B"	ASTM A-255 Grade A Flange Quality	ASTM A-242	ASTM A-441	MIL-S-16113C Grade HT
Mechanical Properties					
Yield Strength, psi, min.	not specified (≈32 000)	40 000 (pressure vessels)	46 000	46 000	47 000
Tensile Strength, psi	58 000 min.	70 000 min.	67 000 min.	67 000 min.	88 000 min.
Elongation in 8 in., percent	28	17	19	19	20
Approx. NDT Range, °F	–20 to +40	+20 to +90	–20 to +40	0 to +70	–60 to +20
Available Thickness Range, in.	½ to 1	3/16 to 4	3/16 to 4	3/16 to 4	no limit
Relative Cost,[a] Ref. ABS Class "B"	1	1.77	1.20	1.20	1.67
Heat Treatment	as rolled	as rolled[b]	as rolled[b]	as rolled[b]	normalized
Weldability	good	good	good	good	good
Chemistry: Ladle[c]					
C max.	0.21	0.18	0.22	0.22	0.18
Mn	0.80–1.10	1.45 max.	1.25 max.	1.25 max.	1.30 max.
P max.	0.05	0.035	—	0.04	0.04
S max.	0.05	0.04	0.05	0.05	0.05
Si	—	0.15–0.30	—[d]	0.030 max.	0.015–0.35
Cr	—	—	—[d]	—	0.15 max.
Ni	—	—	—[d]	—	0.25 max.
Mo	—	—	—[d]	—	0.06 max.
V	—	0.09–0.14	—[d]	0.02 max.	0.02 max.
Cu	—	—	—[d]	0.02 max.	0.35 max.
Ti	—	—	—[d]	—	0.005 max.

[a]Includes chemistry and mechanical property extras only, except Grade HT includes heat treatment extra.
[b]Plates ¾ in. and above should be normalized for ship structural applications.
[c]Where no figure is given, that particular element will occur only in residual amounts—no planned addition is made.
[d]Can vary based on material source.

Table 12 Steels, 50 000 to 79 000 psi, yield strength minimum [68]

Type	CA C-Mn	ASTM A-302 Grade B	C-Mn-V (or Cb)	C-Mn-V (or Cb)	CB C-Mn	(MIL-S-13326) 70 000 Class
Mechanical Properties						
Yield Strength, psi, min.	50 000	50 000	50 000	55 000	60 000	70 000
Tensile Strength, psi, min.	70 000	80 000	65 000	70 000	80 000	90 000
Elongation in 8 in., percent	19	15	18	15	19	17
Approx. NDT Range, °F	–10 to +40	–20 to +50	–60 to +30	–40 to +40	–75 to –40	–65 to –30
Available Thickness Range, in.	3/16 to 1¼	¼ min.[a]	3/16 to 1½	3/16 to 1¼	3/16 to 2	3/16 to 2
Relative Cost, Ref. ABS Class "B"	1.12	1.77	1.12	1.15	1.43	1.80
Heat Treatment	as rolled[b]	as rolled	as rolled	as rolled	. Q&T	Q&T
Weldability	good	special	special	special	good	special
Chemistry: Ladle[c]						
C max.	0.22	0.23	0.24	0.24	0.20[c]	—
Mn	1.35 max.	1.15–1.50	1.40 max.	1.40 max.	1.40 max.	—
P max.	0.040	0.035	0.040	0.040	0.040	0.040
S max.	0.050	0.040	0.050	0.0505	0.050	0.040
Si	0.15–0.30	0.15–0.30	0.15–0.30	0.15–0.30	0.15–0.30	compositions vary with manufacturer
Cr	—	—	—	—	—	
Ni	—	—	—	—	—	
Mo	—	0.45–0.60	—	—	—	—
V	—	—	0.02 min. (or Cb)[d]	0.02 min. (or Cb)[d]	—	—
Cu	—	—	—	—	—	—
Ti	—	—	—	—	—	—

[a]May be heat-treated by normalizing subject to purchase agreement. Plates over 2 in. in thickness require heat treatment.
[b]Plates 3/4 in. and above should be normalized for ship structural applications.
[c]Aluminum deoxidized fine grain practice.
[d]Plates containing columbium above ½ in. should be normalized for ship structural applications.

Table 13 Welding electrodes and processes for carbon and high-tensile steels as specified by NAVSHIPS 0900-060-4010 [69]

Electrode (MIL-Type Specification)[a,h]		Welding Process[b]	Welding Position[b]	Joint Type	Application	Base Material Controls			
						Thickness	Grade	Type[c]	Specification
6010	MIL-E-15599	SMA	all	all	carbon steel plates and shapes	all	M	all	MIL-S-20166
6011								all	MIL-S-22698
6010G	MIL-E-15599	SMA	all	all[d]	carbon steel plates and shapes	all	M	P, Z	MIL-S-20166
6011G								II, III	MIL-S-22698
6020	MIL-E-15599	SMA	F, H	all	carbon steel and high-tensile steel plates and shapes	all	M	U	MIL-S-20166
6027								I	MIL-S-22698
6030							HT[e]	I, II[f]	MIL-S-16113
									MIL-S-24113[g]
7018	MIL-E-22200/1	SMA	all	all	carbon steel plates and shapes	all	M	all	MIL-S-20166
8018									
7015-16	MIL-E-22200/6	SMA	all	all	carbon steel plates and shapes	all	M	all	MIL-S-22698
8015-16									
6010	MIL-E-15599	SMA	all	(5)	high-tensile steel plates and shapes	all	HT	U	MIL-S-20166
6011								I, II[f]	MIL-S-16113
									MIL-S-24113[g]
7015-16	MIL-E-22200/6	SMA	all	all	high-tensile steel plates and shapes	all	HT	U	MIL-S-20166
8015-16									
7018	MIL-E-22200/1							I, II[f]	MIL-S-16113
8018									MIL-S-24113
6010G	MIL-E-15599	SMA	all	(5)	high-tensile steel plates and shapes	all	HT	P, Z	MIL-S-20166
6011G								I, II	MIL-S-16113
									MIL-S-24113[g]
Electrode	MIL-E-18193	submerged arc	F, H	all	carbon steel and high-tensile steel plates and shapes	all	M and HT	U	MIL-S-20166
Flux	MIL-F-18251							I	MIL-S-22698
	MIL-F-19922							I, II[d]	MIL-S-16113
									MIL-S-24113[g]
MIL-E-23765/1		GMA	all	all	carbon steel plates and shapes	all	M	I	MIL-S-22698
70S-1								U	MIL-S-20166
70S-2									
70S-3									
70S-4					high-tensile steel plates and shapes	all	HT	I, II[d]	MIL-S-16113
70S-5									MIL-S-24113[g]
70S-6								U	MIL-S-20166

[a] QQ-E-450 when issued will supercede MIL-E-15599.
[b] Codes—F: Flat. H: Horizontal fillets. SMA: Shielded metal arc.
[c] See Sec. 13 for limitations or special qualification requirements for welding over galvanized or primer-coated surfaces.
[d] Suitable for all joints. Primarily intended for horizontal fillets and welds over galvanizing.
[e] Fillet welds other than in or on to primary hull structure.
[f] Class P or Z.
[g] Normalized condition only.
[h] Or equivalent ASTM or AWS specification.

tures in low-carbon steels. Electrodes for use in welding ship hulls should meet the requirements of the AWS-ASTM Tentative Specifications for Mild Steel–Covered–Arc Welding Electrodes, or the ABS Rules for the approval of Electrodes for Manual Arc Welding. The 60XX series is used for all hull con-

structions, except for E6012 and E6013, which are not approved for any joints in shell plating, strength decks, tank tops, bulkheads, and longitudinal members of large vessels, or in galvanized material because of their slightly lower ductility.

Electrodes of the E7015, E7016, and E7018 classes are often used where improved mechanical properties in the weld are desirable. The trend is toward the use of electrodes having iron-powder coating because they offer high deposition rates.

Various automatic and semiautomatic processes are often used in the fabrication of ships and other ocean engineering structures. Such processes include submerged arc, gas metal-arc, electroslag, and electrogas processes. Welding processes with high deposition rates, such as submerged arc, electroslag, and electrogas, often provide weld metals with rather low notch toughness.

Welding of high-strength steels with less than 80-ksi yield strengths: Some problems with welding high strength steels have been discussed in an earlier part of this chapter (*Welding considerations*, pp. 145–148).

In the welding of high-strength steels, two factors are emphasized. First, select the proper electrodes to meet the strength requirements for the welded joint. Second, select the adequate preheat to assure that a sound weld will be produced. In the welding of a joint under high restraint, it is recommended that higher preheating temperatures be used. Better results are obtained when low-hydrogen electrodes are used for high-strength steels. Satisfactory welds can be obtained using these electrodes at lower preheats than are necessary for conventional electrodes. Recently developed high-strength steels with very low carbon contents can be welded successfully with no preheat [63].

Hydrogen has been associated with both underbead cracks and toe cracks. The source of hydrogen can be either the electrode coating or the surface of the plates to be welded. So it is important to keep the plate surface dry, as well as to use an electrode with a low hydrogen content. For effective use, the low-hydrogen electrodes must be properly stored and kept dry. Porosity at the start of a weld is one indicator that the electrode coating has picked up moisture.

High-strength steels can also be welded with various processes other than SMA welding. However, further discussions of welding high-strength steels are not presented here because the subject is considered to be outside the scope of this book. Further information may be obtained in the *Welding Handbook* [20,64] and other publications [1,57–61,65–67].

3.1.3 Steels with 80- to 120-ksi Yield Strengths

Various steels with yield strengths ranging from 80- to 120-ksi—Table 14 lists several steels with 80 000 to 120 000 psi (552 to 827 MPa) minimum specified yield strength. All steels listed here are quenched and tempered to obtain high strength, good notch toughness, and good weldability.

Steel plates and shapes at the strength level given in Table 14 for industrial applications have evolved in the United States since World War II. This has made possible new and improved design concepts. Many of the steels here are grades proprietary to the major steel manufacturers. The exceptions are HY-80 and HY-100 steels made to MIL-S-16216 which are used principally for naval ships. Specification MIL-S-13326 covers four strength levels, including 90 000 and 120 000 psi (620 and 827 MPa, respectively). Steel plates at higher strengths are available from several steel manufacturers. Some of these steels are not covered yet by ASTM designations.

Among the steels listed, HY-80 is the most commonly used for submarine hulls. ASTM A517-67 is also commonly used for commercial applications including pressure vessels, storage tanks, and merchant ships. For example, "T-1" steel is an ASTM A514/A517 steel produced by the U.S. Steel Corporation. Compared with HY-80, ASTM A517-67 contains less nickel and is less costly, but about twice as expensive as ABS B steel, although HY-80 is about 3.5 times as expensive as ABS B steel. Since HY-80 is the most well known among the steels listed in Table 14, the following discussions primarily cover HY-80 steel.

Manufacture of HY-80 steel and its mechanical properties—HY-80 is a low-carbon steel which derives its strength and notch toughness from quenching and tempering. In other words, HY-80 is a "hardenable" steel which obtains its properties from heat treatment [60].

Chemical composition: Table 14 shows the specification limits of HY-80 chemical composition. The maximum limits of HY-80 steel are

- 0.18 percent for carbon;
- 0.15 percent to 0.35 percent for silicon, which is used for deoxidation;
- 0.025 percent each for sulfur and phosphorus, not to exceed 0.045 percent combined. This strict control of sulfur and phosphorus requires that more care than usual be taken during the steelmaking process;
- 0.1 percent to 0.4 percent for manganese which is used for sulfur control, rather than strength. An amount over 1.0 percent of manganese would cause embrittled steel during heat treatments;
- 2.00 percent to 3.25 percent nickel for obtaining good notch toughness; and
- molybdenum, slight quantities for lowering the temper embrittlement.

Method of manufacture [60]: HY-80 steel is a fully killed and fine-grained steel. The final heat treatment of quenching and tempering is performed for obtain-

Table 14 Steels, 80 000 to 120 000 psi, yield strength minimum [68]

Type	Proprietary Grades[a]	MIL-S-16216 Gr. HY-80	ASTM A543 C1.1	ASTM A543 C1.2
Mechanical Properties				
Yield Strength, psi, min.	80 000	80 000	85 000	100 000
Tensile Strength, psi, min.	—[b]	—[b]	105 000	115 000
Elongation in 2 in., percent	18	20	16	16
Approx. NDT Range, °F	–40 or lower	–130 or lower	–120 or lower	–90 or lower
Available Thickness Range, in.	3/16 to 2½[c]	3/16 to 8	3/16 to 4	over 4
Relative Cost, Ref. ABS Class "B"	2.0	3.5	3.0	3.0
Heat Treatment	Q&T	Q&T	Q&T	Q&T
Weldability	special	special	special below 0.21 C	special below 0.21 C
Chemistry: Ladle				
C	0.21 max.	0.18	0.23 max.	0.23 max.
Mn	—	0.10–0.40	0.40 max.	0.40 max.
P max.₁	0.040	0.025	0.035	0.035
S max.₁	0.040	0.025	0.040	0.040
Si	—	0.15–0.35	0.20–0.35	0.20–0.35
Cr	—	1.00–1.80	1.50–2.00	1.50–2.00
Ni	—	2.00–3.25	2.60–3.25	3.00–4.00
Mo	—	0.20–0.60	0.45–0.60	0.45–0.60
V	—	—	0.03 max.	0.03 max.
Cu	—	—	—	—
Ti	—	—	—	—
Zr	—	—	—	—

Type	ASTM A517-67 a	MIL-S-13326 Class 90	MIL-S-16216 Gr. HY-100	MIL-S-13326 Class 120
Mechanical Properties				
Yield Strength, psi, min.	100 000	90 000	100 000	120 000
Tensile Strength, psi, min.	115 000	—[b]	—[b]	—[b]
Elongation in 2 in., percent	16	no limit specified	18	no limit specified
Approx. NDT Range, °F	–50 or lower	–40 or lower	–100 or lower	–20 or lower
Available Thickness Range, in.	3/16 to 2½	no limit specified	3/16 to 3	no limit specified
Relative Cost, Ref. ABS Class "B"	2.0	2.0	3.5	3.5
Heat Treatment	Q&T	Q&T	Q&T	Q&T
Weldability	special	special	special	special below 0.21 C
Chemistry: Ladle				
C	0.21 max.	no chemistry specified	0.20	no chemistry specified
Mn	—		0.10–0.40	
P max.₁	0.035		0.025	
S max.₁	0.040		0.025	
Si	—		0.15–0.35	
Cr	—		1.00–1.80	
Ni	—		2.25–3.50	
Mo	—		0.20–0.60	
V	—		—	
Cu	—		—	
Ti	—		—	
Zr	—		—	

[a]This column includes proprietary grades individually identified by chemistry, mechanical properties, and thickness range.
[b]Approximately 10 000 to 15 000 psi above yield strength.
[c]Normal range available. Plates to 8-in. thick are available in some proprietary grades.

ing both strength and notch toughness. Two specified limitations for the heat treatment are

- the final tempering temperature shall not be less than 1100°F (593°C) and
- the microstructure at mid-thickness of the plate must contain not less than 80 percent martensite.

In actual practice, the heat treatment consists of austenization in the range of 1550 to 1650°F (843 to

Table 15 Specification limits of HY-80 mechanical properties

Property	Plate thickness	
	Less than 5/8 in.	5/8 in. and Over
Ultimate Strength, psi	for information	for information
Yield Strength at 0.22 Percent Offset, psi	80 000 to 100 000	80 000 to 95 000
Elongation in 2 in., percent, min.	19	20
Reduction in area, percent		
Longitudinal	—	55
Transverse	—	50

Property	Charpy V-Notch Energy Requirements		
	1/4 in. to 1/2 in. excl.	1/2 in. to 2 in. excl.	over 2 in.
Specimen Size, mm	10 × 5	10 × 10	10 × 10
Foot-Pounds, min.	for information	50	30
Test Temperature, °F	−120	−120	−120

899°C) followed by water quench, and tempering in the range of 1150 to 1250°F (621 to 677°C) followed by water quench.

Mechanical properties: Table 15 shows specification limits for mechanical properties of HY-80 steel [1,60]. Values of specified yield strengths are 80 000 to 100 000 psi (552 to 689 MPa) for plates less than 5/8 in. (16 mm) thick and 80 000 to 95 000 psi (552 to 655 MPa) for plates 5/8 in. and over. Minimum specified Charpy V-notch impact values at −120°F (−84°C) are

- 50 ft/lb (6.9 kg/m) for plates 1/2 to 2 in. (12.5 to 51 mm), inclusive
- 30 ft/lb (4.1 kg/m) for plates over 2 in. (51 mm) thick.

Welding of HY-80 steel—Table 16 lists electrodes currently used for welding HY-80 steel and their applications in submarine construction [1,60]. In addition to the electrode type designation, the appropriate Military Specifications, welding processes, welding positions, and applications are included in the table. For the electrodes for the shielded metal-arc process, low-hydrogen iron powder type EXX18 is used most widely. Table 16 also includes electrodes for gas metal-arc (semiautomatic) and submerged arc processes.

Table 17 shows specification limits of mechanical properties of as-deposited weld metals. A family of EXX18-type electrodes has been developed which can provide a range of yield strengths from 60 000 to 110 000 psi (414 to 689 MPa). It is possible, therefore, to select from Table 17 an electrode which will undermatch or overmatch the strength of HY-80 steel base metal, as desired or required by the design.

As has been discussed earlier (*Fracture toughness of the weld metal*, p. 145), welds that have properties the same as the base metal are desired. Since welds may contain some flaws which escape detection, it has been considered desirable as a practice to have a weld metal slightly stronger than the base metal, or the weld metal overmatch the base metal. This will offset the possible loss in strength in the weld metal, and provide superior performance of a weldment as a whole. For this reason, the basic philosophy taken

Table 16 HY-80 welding electrodes and application [60]

Electrode Type	Specification	Process	Position	Application
Mil-11018	MIL-E-22200/1	shielded metal arc	all	all
Mil-10018	MIL-E-22200/1	shielded metal arc	all	fillet, fillet groove, or groove joints
Mil-9018	MIL-E-22200/1	shielded metal arc	all	limited use
Mil-B88	MIL-E-19822	semi-automatic or automatic metal inert-gas arc	flat or horizontal	all
Mil-EB82[a]	MIL-E-22749	submerged arc	flat	all
Mil-MI88[b]	MIL-E-22749n	submerged arc	flat	all
Mil-8218Y QT	MIL-E-22200/5	shielded metal arc	all	limited to procedure approval

[a]Granular flux particle size 10 by 50.
[b]Granular flux particle size 12 by 150.

Table 17 Specification limits of deposited weld metal mechanical properties[a] [60]

Property	For HTS		For HY-80						
Electrode Type	Mil-7018	Mil-8018	Mil-9018	Mil-10018	Mil-11018	Mil-B88	Mil-EB82	Mil-MI88	Mil-8218
Ultimate Strength, psi	70 000	80 000	90 000	100 000	110 000	—	—	110 000	—
Yield Strength at 0.2% Offset, psi	60 000–75 000	70 000–82 000	78 000–90 000	90 000–102 000	95 000–107 000	88 000	82 000	88 000	82 000
Elongation in 2 in., percent	24	24	24	20	20	14	16	20	18
Charpy V-Notch Impact Energy, ft/lb									
At –10°F	20	20							
At –60°F			20	20	20	20		30	20
At –80°F							20		

[a]Single values are minimums.

by the U.S. Navy has been to use electrodes which slightly overmatch the base metal in strength. As pointed out earlier, however, it has not yet been possible to develop an electrode which provides a weld metal as notch tough as an HY-80 base metal (see Tables 15 and 17). In other words, all weld metals produced today are undermatched in terms of notch toughness.

3.2 Aluminum Alloys

Aluminum alloys have been used extensively in a number of submersible vehicles. Among specific components in which they have been used are: structural members, hard containers (such as housings for electrical components), hydraulic components, propellers, and (in some cases) the personnel hull. These alloys are attractive because of their availability, low cost, fabricability, and high strength/density ratios. The materials have the disadvantage of being anodic to most other structural alloys so that they are vulnerable to corrosion when used in mixed structures. However, with the exception of the severe problem caused by proximity to copper-base alloys, most of the problems can be avoided by special design modifications. Aluminum alloys also have an additional vulnerability which is not present in most other materials: they can be corroded by stray electrical currents of either polarity.

There are three main types of wrought aluminum alloys which can be used for submersible vehicle construction: 6061 alloy, certain of the 7000 series alloys, and the 5000 series alloys. The two former types are heat-treatable, the latter is nonheat-treatable.

The 7000 series alloys are often stated to be unsuitable for marine applications because of susceptibility to corrosion and stress-corrosion. However, there are several examples of their successful use in submersible vehicle components where stresses are predominantly compressive such as hard containers. The most notable example of their use is the submersible *Aluminaut*, in which the personnel hull is fabricated in 7079 alloy.

Table 18 shows typical base-metal mechanical properties for several alloys.

Cast alloy A356 has been successfully used for submersible-vehicle components, notably for propeller blades. Its corrosion characteristics are similar to 6061 alloy. Cast alloy A357, which has a yield strength about 5 ksi greater than that of A356, also appears suitable for submersible components.

3.2.1 Joining Aluminum Alloys

A general design problem with aluminum alloys is that in most alloys it is difficult or impossible to obtain matching strength in weld metal and base metal. Another problem is the difficulty or impossibility of stress-relieving aluminum alloy weldments by a post-weld heat treatment.

Welds in the 5000 series alloys typically have yield strengths of 50 percent to 60 percent of the base metal value so that it is necessary for welds to be thicker than surrounding base metal or for welds to be located in lightly stressed areas. These alloys are easy to weld (including welding by the electron-beam process), a factor which has promoted their extensive use in marine construction.

In the welding of 6061 alloy, several combinations of base metal condition, filler alloy, and post-weld heat treatment are possible: the optimum combination for a given application depending on thickness,

Table 18 Typical mechanical properties of some aluminum alloys

Properties	5086-H116	6061-T6	7075-T6	7075-T73	A356-T6
Tensile Strength, ksi	42	45	83	63	32
Yield Strength at 0.2%, ksi	30	40	73	58	27
Elongation in 2 in., percent	12	17	11	15	5

permissible distortion, properties required, and type of joint. The latter factor influences constraint and susceptibility to hot-cracking.

6061 alloy welds (made with 4043 filler metal) can be post-weld heat-treated to give full T6 strength properties. However, the tensile ductility in fully heat-treated welds is low, and the consequences of the quench-induced residual stresses must be anticipated. The alloy can be electron-beam welded in the T6 condition to give essentially matching weld properties in the as-welded condition. Weld filler metal selection is important for the 5000 series and 6061 alloys when used in submersible applications. In some cases, the optimum selection for weld strength is not the same as that for optimum corrosion resistance (see reference [20], "Aluminum and Aluminum Alloys").

The 7000 series alloys have poor weldability and should not be welded in marine applications.

Weld-joint design is important for submersible applications. Filler welds or partial penetration welds are undesirable if they permit seawater penetration into a crevice.

Dip-brazed construction has been successfully used on several 6061 alloy submersible components (with 4343 alloy filler) and the brazed joints appear to have an adequate corrosion resistance. In brazed joints for marine applications, the filler should penetrate the joint fully to avoid sites for crevice corrosion and salt entrapment.

3.2.2 *Riveted Structures*

Riveted 6061 alloy structures have been used successfully for underwater structures involving intermittent immersions. Rivet material used is generally 2117 alloy. This is in defiance of the general guideline which prohibits the use of 2000 series alloys in sea water, but other available rivet alloys present more difficult problems. The riveting procedure is important: (wet) wash-primer should be applied to the rivet hole immediately before riveting. Such riveted structures are expected to crevice-corrode eventually along the faying surface and should be regarded as replaceable components having limited life.

3.2.3 *Protection Against General Corrosion*

As isolated materials, 6061 alloy, the 5000 series alloys, and the A356 casting alloy have good corrosion-resistance in sea water. For continuously immersed periods of about two months, they will remain free of pits and crevice corrosion. After a one-year immersed period, pitting is likely to occur, with maximum pit depth about 0.005 in. Isolated 7000 series will pit more rapidly; pitting will occur during two-month immersed periods.

A particular precaution is required in using 5456 and 5086 alloys. In the H32 temper, these alloys can be susceptible to a form of rapid grain boundary corrosion when immersed in sea water. For marine use, it is advisable to perform special tests on coupons from the batch to be used, or to use material in the sensitization-resistant H116 or H117 tempers.

Aluminum alloys will pit slightly if they are exposed to a damp marine atmosphere such as submersibles will generally experience between missions. For this reason, aluminum alloy components are often painted. (For optimum painting and paint preparation procedure, see reference [70].)

During immersion, aluminum alloys can be cathodically protected very effectively by attachment of zinc anodes (magnesium anodes should not be used). If aluminum alloy components are metallically connected to more noble materials, these latter materials must be painted or otherwise electrochemically inactivated (as by coating with a grease) in order to minimize the aluminum alloy corrosion. In these situations, coating of the more noble material has the highest priority. Coating of the aluminum alloy component alone will usually cause disastrous corrosion at points of local paint breakdown.

Because of the severe corrosion they induce, copper-base alloys should be excluded from structures containing aluminum alloy components.

3.2.4 *Stress-Corrosion Cracking Susceptibility*

Of the alloys shown in Table 18, 7075 alloy is the only one having a serious susceptibility to stress-corrosion. The alloy will pit quite readily in sea water (it can also "exfoliate" along the flow lines resulting from metal-working operations), so that compared with most materials used in marine construction it is relatively easy for a crack-like defect to form which will grow in the stress-corrosion mode. For this reason, the alloy is used only where the applied stresses are compressive. Even in these applications, stress-corrosion cannot be ignored because residual tensile stresses can also be a cause of stress-corrosion cracking.

There are two approaches for avoiding this problem. If the component involved is to be fabricated from an extrusion or plate, the stock may be procured in the -T651 condition (mechanically stress-relieved) which will be free from residual stresses. Or, if the component involves a forging (which is not amenable to mechanical stress-relief), the material can be specified in the -T73 condition. This is a stress-corrosion resistant temper having a lower strength than -T6 temper. An approximate indication of the resistance to stress-corrosion of the two tempers is given by the following K_{Iscc} values:

7075-T651: 5 ksi $\sqrt{\text{in.}}$
7075-T73: 24 ksi $\sqrt{\text{in.}}$

These values were determined under dripping 3 percent NaCl solution so that they may not be accurate for the immersed condition (from reference [71]).

3.3 Titanium Alloys

In many ways, titanium alloys are ideal materials for submersible-vehicle components because of their near-immunity to corrosion and their high strength/density ratio. They are leading candidate materials for personnel hull fabrication and other components where long life and the highest reliability are needed. Their high cost and to some extent limited availability are the principal constraints on their use.

The chief technical problem area for the designer with these materials is ensurance of fracture safety for components fabricated in the high-strength alloys and which are loaded in tension. Other potential problems of which the designer should be aware are the susceptibility of titanium to biological fouling in sea water and to surface galling (where sliding contact occurs).

There are three main compositions of wrought titanium alloys which are suitable for marine applications: titanium–6% aluminum–4% vanadium (6-4 alloy); titanium–6% aluminum–2% columbium–1% tantalum–1% molybdenum (6-2-1-1 alloy); and Commercially Pure (C.P.) titanium. Typical properties are given in Table 19.

The 6-4 alloy is likely to be more available than the 6-2-1-1 alloy. It can be obtained in several compositional and metallurgical variants; for marine applications, special attention should be given to selection of the optimum composition and metallurgical condition (see below). The slightly lower strength 6-2-1-1 alloy will be the best choice for applications where high tensile stresses are present and where optimum fracture resistance is needed.

The C.P. titanium Grades 1 and 2 should be used for components in which their lower strength is acceptable. Compared with the alloys, they have the advantages of immunity to stress corrosion, easier fabricability, and greater availability.

Titanium alloy castings are available and may be used for components in which tensile stresses will be low. Centrifugally cast 6-4 alloy components have been used successfully for marine components requiring very high reliability [74] but where applied stresses were compressive.

3.3.1 Low-Stress Fracture

The 6-4 alloy is susceptible to stress corrosion cracking in sea water especially in the higher strength conditions. (For marine applications, the alloy is used in the annealed condition rather than in the highest-strength, aged condition.) In applications where tensile stresses are present, it therefore will be necessary to include a fracture-mechanics analysis as part of the design. This analysis will dictate the nondestructive test criteria to be used for flaw size and orientation. One of the problems in designing against stress corrosion cracking in this alloy is that its K_{Iscc} value cannot be accurately predicted from more readily available data such as tensile properties and composition. Thus for critical uses it is often necessary to make a fracture-mechanics determination of the K_{Iscc} value for the particular batch of material to be used. K_{Iscc} in this alloy has a fairly complicated dependence on composition (especially the interstitial elements oxygen and nitrogen) and on certain microstructural features. Attempts have been made to predict K_{Iscc} by an algebraic formula in terms of these factors [75], but should only be regarded as approximate indicators of the relative weight of these factors. There is one minimum guideline which should be observed in selecting material for components which will be exposed to tensile stresses: use only the ELI (extra-low interstitial) Grade as specified, for instance, under MIL-T-9046F, Type III, Composition D (for plate and sheet) and MIL-T-9047D, Type III, Composition B (for bars and forgings). Within this grade, lower strength levels and interstitial contents are likely to yield higher K_{Iscc} values.

The 6-2-1-1 alloy has been developed solely for marine applications and has a substantially greater resistance to stress corrosion than 6-4 alloy.

Nonenvironmental (brittle) low-strength fracture can also occur in these alloys, but since $K_{Ic} > K_{Iscc}$, safe-design against brittle fracture will generally be more easily achieved than against stress corrosion.

3.3.2 Welding

High-quality arc welds can be made in each of the materials discussed above. Approximately matching

Table 19 Typical mechanical properties of titanium alloys

Material	UTS[a], ksi	Y.S.[a], ksi	Elong., %	R.A., %	K_{Ic}, ksi $\sqrt{in.}$	K_{Isc}, ksi $\sqrt{in.}$
6-4 Alloy (annealed)[b]	130	120	12	22	40–70	30–50
6-4 Alloy (annealed), ELI Grade	120	110	13	25	90–100	35–80
6-2-1-1 Alloy	126	105	14	30	—	90
C.P., Grade 1	35	25	25	35	—	—
C.P., Grade 2	50	40	22	30	—	—

[a]UTS and YS values shown are typical minimum values which may be exceeded by a considerable margin. References [72] and [73] contain detailed mechanical and physical property data.
[b]Typical for high interstitial material containing 0.15–0.20 percent oxygen.

strength, toughness, and resistance to stress corrosion can be obtained in the as-welded condition. Stringent control of welding atmosphere (argon or helium) and cleanliness are necessary to obtain high quality welds. Welding-wire chemistry (especially with respect to the interstitial elements) is very important for obtaining weldments with high toughness and resistance to stress corrosion cracking.

Six-four alloy weldments can be stress-relieved by heat treatment in the range 1000 to 1200°F after welding. This will eliminate residual stresses which might otherwise contribute to stress corrosion crack propagation.

3.3.3 Galvanic Interactions with Other Materials

Titanium is cathodic with respect to almost all other structural materials, so it may induce accelerated corrosion of other materials to which it is connected. However, because of its polarization characteristics, the severity of the induced corrosion is relatively mild. For instance, the galvanic corrosion of aluminum caused by a titanium cathode is usually somewhat less than that caused by stainless steels (which have less noble galvanic potentials than titanium). The galvanic corrosion can be prevented by isolation mounting of the titanium component or by painting of the titanium surface. It is noted here that paint adhesion on titanium can be a problem, requiring special surface preparation procedures. A common practice is to sand-blast the surface immediately prior to wash-priming; if the application is delayed beyond 30 min, good adhesion will not be obtained.

The K_{Iscc} values for titanium alloys are almost insensitive to cathodic polarization (at least to the degree of polarization corresponding to coupling with zinc [76]) so that coupling to the common structural materials will not degrade the resistance to stress corrosion cracking.

3.4 Glass-Reinforced Plastics

3.4.1 Scope

This section on glass-reinforced plastics (GRP) covers some history of their use in submersibles and then reviews some basic principles that should be followed in the application of this material in the design of manned submersibles.

3.4.2 General Background

The recognition of GRP as a structural material came about in the 1940s with the introduction of filamentous glass woven in various fabric patterns. The modulus of a resin was found to be raised almost an order of magnitude with a cloth, woven reinforcement. Some of the earliest development work occurred with radomes in the aerospace industry. The advantages of corrosion resistance and light weight were recognized by the marine industry immediately, and the boats have used this material for more than 30 years.

With the exception of the *Aluminaut*, the secondary structures of practically all manned research submersibles have been built in GRP. This includes all the *STAR* boats, the *Autec* boats, and *NR-1* built by EBDiv; it also includes the *Alvin* and the *Perry* submarines.

A number of the sails and superstructures of conventionally powered combatant boats of the U.S. Navy have GRP structures. There are numerous other GRP components in present-day combatant submarines of the U.S. Navy. Details on these, however, will not be presented here.

3.4.3 General Design Parameters

A glass-reinforced plastic as presently used consists of a glass fiber dispersed uniformly in a rigid, thermosetting resin. The glass can be in the form of a nonwoven mat buildup of chopped fibers or continuous filaments woven into a roving or cloth pattern. Glass percentages range from 25 percent for the mat to about 65 percent for woven cloth and 75 percent for continuous filaments. The resin of largest commercial use is the polyester. The epoxy is used for the more demanding structural applications. The major specifications relating to GRP include those listed in Table 20.

3.4.4 Production Processes

The major production processes used to make structural GRP components include

- hand lay-up,
- vacuum-bag laminating,

Table 20 Specifications relating to GRP

Glass Cloth	MIL-C-9084C cloth, glass, finished, for resin laminates
Random Glass Mat	Equal to Owens Corning M-720
Polyester Resin	MIL-R-7575C, resin low pressure or MIL-R-21607D, resin, low pressure laminating fire resistant
Epoxy Resin	MIL-R-9300B resin, epoxy, low pressure laminating
Marine Polyester Glass Laminates	MIL-P-17549C (SHIPS) plastic laminates, fibrous glass reinforced, marine structures
Epoxy Glass Laminates	MIL-P-25421B, plastic material, glass fiber base–epoxy resin, low pressure laminated
Process Specification	MIL-P-9400 plastic laminate and sandwich construction parts, aircraft structural, process specification requirements

- autoclave pressure lay-ups,
- chopped strand spray-ups,
- compression molded mat buildups,
- pressure impregnation of dry lay-ups, and
- filament wound.

The first three processes comprise the bulk of structures designed for manned submersibles, and will be discussed here. The next two are serving more applications in automotive, trailer, housing applications, etc. The last processes are highly specialized, done by only a few companies. The following discussion will concentrate on the first three processes since they comprise the processes of primary interest in submersibles.

Hand lay-up—The hand lay-up process is very labor intensive, requiring simple plastic GRP or wooden molds. Resulting air-void levels are often high and physical properties are the poorest of the three processes; the costs are the lowest with this process. Resin levels range from 40 percent to 70 percent by weight, depending upon the labor content put into the part. Physical strengths vary directly with the resin level and void level. Cloth, roving, or mat can be used. If a simple functional part with no severe structural requirements is to be constructed this process is used. Oftentimes, large-sized parts are made with the hand lay-up process because of timing and mold requirements of the other two processes.

Vacuum bag—Vacuum-bag molding uses an atmospheric pressure difference by the drawing of a vacuum under a flexible plastic film. The wet, impregnated glass cloth has the trapped air forced out of the body by squeegee and the pressure difference. Vacuum-bag molding is capable of producing high-quality laminates using inexpensive tooling, but generally mold costs and part costs are higher than with hand lay-ups. Mold materials range from flat sheets to resin-sealed sculptured polyurethane foam. These molds must be capable of resisting atmospheric pressure differentials and must be sealed against air permeability.

Polyester laminates are generally produced by the wet vacuum-bag process employing room-temperature curing cycles. The amount of material is limited to the amount which can be handled in about 6 h prior to the gelling of the resin. The resin must be dispersed into the glass cloth and air-screened out after the base is in place.

Small epoxy components are also produced by the room-temperature wet vacuum-bag process but are limited in their size because of the difficulty of removing the air from the higher-viscosity epoxy resins. The use of epoxy or polyester pre-impregnated glass cloth (prepreg) facilitates the fabrication of larger parts, but requires both heat and pressure to complete the cure.

Autoclave molding—The application of both heat and pressure upon the lay-up by the use of autoclave aids in producing a high-glass-content laminate. Large monolithic parts are generally produced by this method. Some submarine structures may be molded in specially designed molds which by nature of their shape comprise the outer portion of the autoclave itself, thus reducing the difficulty of locating suitable sizes of autoclaves for the molding of large structural components. This process produces the highest-quality part and is used where structural service is most severe. Most of the composite structures appearing in aircraft are made by this process.

3.4.5 Woven Cloth Laminates

Laminates produced from woven cloth are generally tested in the warp direction (parallel to the length of the roll) unless otherwise noted. Woven cloth laminates are classified as bidirectional when the fiber orientation is equal in both the warp and fill directions. A general description of the most widely used industrial weaves are as follows:

1. Plain: The plain weave consists of yarns in an alternating fashion, one over and under every other yarn. This weave is very stable but is the least pliable.
2. Basket: The basket weave consists of two or more warp yarns over and under two or more alternate fill yarns. This weave is more pliable, flatter, and stronger than the plain weave, but it is not as stable.
3. Four-Harness Satin: The weave pattern is a three-by-one interlacing whereby a filling yarn floats over three warp yarns and under one. This weave is more pliable than the plain weave and is used on curves typically in reinforced composites.
4. Eight-Harness Satin: The eight-harness satin is similar to the four-harness satin except that one fill yarn floats over seven warp yarns and under one. This is the most pliable weave, specially suitable for forming on small radius curves.
5. Twill Weave: The twill weave, more pliable than the plain weave, has better drapability while maintaining more stability over the satin weaves. The weave is formed by one warp yarn floating over two or more fill yarns.

In addition to the variation in weaves, the yarn weight and fabric construction determine the weight and thickness of the fabric. Subsequent finishing treatments of glass yarns are necessary for the chemical bonding of the glass to resin matrix. The tensile strength of fabric is generally stated by the weaver prior to the finishing treatment and is not indicative of the laminate strengths. The proper surface finish on the glass is particularly important in laminates exposed to high moisture levels. The A-1100 finish is generally used in structures for submersibles.

Specific strengths of resin laminates are obtained from cloths that have been suitably finished for resin impregnation and are covered in MIL-C-9084C.

3.4.6 *Control of Thickness*

The control of laminate thickness and glass to resin content is more automatically accomplished by the use of prepreg than by the wet lay-up method. Per ply thickness of a 181 cloth laminate may be about 10 mils for a prepreg and could vary from 15 to 20 mils for a hand lay-up.

3.4.7 *Engineering Properties and Design of Laminates*

The designer or analyst who normally works with metals or organic polymers or both must become familiar with the nature of these composite laminates. Depending upon the quantity and direction of the glass fibers laid up in the laminate, strengths can be isotropic or directional in the plane of the laminate. Many laminates are laid up so that strengths are highly oriented. Whatever the strengths in the plane of the laminate, the limitations of strength and stiffness across the plane of the laminate must be understood. Here the bare resin limits stresses and deflections imposed on the laminate. The laminate's poor interlaminar shear strengths of less than 10 000 psi must not be allowed to limit laminate use. Design must avoid allowing stresses to be built up so that significant interlaminar shear occurs. Table 21 lists the minimum properties of composite laminates as required by references [77] and [78].

Fiberglass laminates are complex structurally and do not always follow Hooke's Law, but in a majority of cases this assumption can be made without appreciable error. The designer has control of the number of variations afforded by cloth weaves, resin content, and laminate thickness in addition to form and molding method. A range in mechanical properties can be obtained by varying the relative proportions of resin and glass and controlling the positioning of the reinforcing fibers. The strength of a laminate in any direction is proportional to the amount of fiber by volume present in that direction. The modulus of a laminate in any direction is also proportional to the amount of fiber by volume present in that direction. Across the plane of the laminate, stiffness and strength of the laminate are close to those of the resin itself.

Poisson's ratio varies as the direction of imposed stress and can range from 0.22 to 0.35, reflecting the influence of glass and resin respectively on the performance of the laminate [79].

3.4.8 *Stress-Strain Relationship*

The stress-strain curves of laminates do not have yield points and are similar to those of wood and nonferrous metals. The deformations between the proportional limit and ultimate strengths are less than for ferrous metals. If the arbitrary strain of 0.2 percent is applied to laminates, the yield strength will intersect the curve near or well above the ultimate strength. Accordingly, only the ultimate strength of the laminate is considered in the selection of a factor of safety. If the factor of safety is 4 or more, then the laminate can be assumed to follow Hooke's Law without appreciable error.

In some instances, stress-strain curves for laminates exhibit two separate proportional limits and moduli of elasticity. The behavior is predominant for

Table 21 E Glass composite material design values

Physical Property	MIL-P-17459C Grade[a] (Polyester Resin)						MIL-P-25421B (Epoxy Resin)			
	1	2	3	4	5	W	181	Crossplied Filament	Isotropic Filament	Unidirectional Filament
Strength, ksi										
Flexural, Dry and 23°C[b]	50	37	31	23	18	32	75	100	65	125
Flexural, Wet	45	33	27	20	15	29	65	85	58	110
Modulus, ($\times 10^{-6}$ psi)										
Flexural, Dry and 23°C[b]	2.5	2.0	1.45	1.1	0.85	1.65	3.2	3.3	2.5	5.0
Flexural, Wet	2.3	1.8	1.25	0.99	0.77	1.50	3.2	3.3	2.5	5.0
Strength, ksi										
Tensile, Dry and 23°C[b]	37	28	20	14	9	35	48	60	40	100
Tensile, Wet	—	—	—	—	—	—	45	55	35	90
Strength, ksi										
Compressive, Dry and 23°C[b]	33	25	21	17	16	18	50	55	45	70
Compressive, Wet	28	23	19	15	14	17	45	50	40	62
Void Content, %, max.	1.5	2	3	4	5	4	—	—	—	—
Resin Content, %	35–43	42–52	49–59	55–65	65–75	45–56	—	—	—	—

[a]Lengthwise in the direction of maximum strength (warp direction for woven cloth).
[b]Standard conditions.

tension, less for flexure, and nonexistent for compression. Unidirectional laminates do not exhibit this phenomenon when stress is applied in the warp direction.

Preloading above the initial proportional limit alters the straight line portion. Repeated preloadings to the same stress levels cause no additional changes. It is believed that the initial break in the curve occurs at the failure of the resin which transfers the load to the reinforcing fibers. The effect of preloading beyond the initial break may then simply be the behavior of the fibers with cracking or failure of the resin as the stress level is increased. Fiberglass-reinforced pipes and pressure bottles exhibit leaks well before the burst pressures are reached, which are considered to be caused by the dual proportional limit phenomenon.

An excellent reference for the design of complex laminates is the *Marine Design Manual* [80]. The treatment of a composite laminate of different lamina, such as 1 oz. mat, 10 oz. cloth, and woven roving is analyzed and exemplified in the manual.

The treatment for the design and analysis of composite laminates is also theoretically treated in MIL-HDBK-17A Plastics for Aerospace Vehicle [81], which covers a greater variety of reinforcements and resins commonly used in aircraft designs. The Navy handbook [82] is also a worthwhile general reference for design and analysis of composites.

Techniques to improve the strength between successive lay-ups of lamina such as "B staging" or step-curing of epoxy resins or partial curing of polyester resins by ultraviolet radiation, or both, have been employed to minimize the effect of joint strength upon post bonded laminates and secondary bonds. Although secondary bonds may possess up to 90 percent of the original mechanical properties, they should be minimized or strategically placed in low-stress locations whenever they are necessary or unavoidable.

3.4.9 Flexure Limitations

The major structural criterion used in the design of submersible structures for operation in seawater is waveslap. Similarly, docking and ship impact can be factored in with much more difficulty. Mounting techniques must be used since the heaviest GRP structure can be destroyed by impact.

Useful fatigue data based upon the low-cycle fatigue controlling in undersea structures are very scanty. Scatter in fatigue data is inherent with this physical property.

Effects of notch sensitivity, environmental factors, wide production variables, and the absence of endurance limit further increases the uncertainty of predicting the fatigue characteristics. Extended far enough, every laminate will eventually fail in fatigue.

3.4.10 Environment Effects

The effects of water on GRP are significant. Water permeates through fiber edges attacking adhesion of the resin. Surface finish used on the glass is a significant variable, with A-1100 finish the most often used GRP destined for submersibles.

Most laminates require some thickness of gel coat to reduce rates of water permeation. Thicknesses of 5 to 10 mils in good-quality laminates are common. Too thick an unreinforced resin layer leaves the laminate susceptible to crazing and cracking in service.

Any edge exposed by machining or grinding is normally top-coated by a suitable epoxy resin. Outwater and Kellog [83] more than 20 years ago ran tests on water movement through edges versus across the laminate and found a factor of 10 in water movement along fiber edges versus those across the plane of the laminate.

If a standard anticorrosive epoxy, polyurethane, or vinyl paint is used over the laminate, water permeation can be reduced as with a gel coat.

Heat-cured, epoxy resin systems are to be preferred for long-term water resistance to that of the polyester resin, and long-term degradations of 10 percent to 20 percent are common in many laminates exposed to sea water service. Tests to determine such degradations are based on water boil or elevated temperature exposures of laminates with cut edges. The correlation between actual service and such test data can vary significantly since the degradation mechanisms are so complicated.

3.4.11 Factor of Safety

The magnitude of the design factor of safety is of course commensurate with the amount of risk. The following factors of safety appear in the Gibbs and Cox *Marine Design Manual* [80]:

Condition	Safety Factor
a. Static short-term loads	2 minimum
b. Static long-term loads	4 minimum
c. Variable or changing loads	4 minimum
d. Repeated loads	4 minimum
e. Fatigue or load reversal	6 minimum
f. Impact loads—repeated	10 minimum

The selection of factor of safety rests with the designer unless established by code or authority. The extent of the number and type of allowable defects on or allowable repairs to a structural composite, or both, will also have an impact upon the magnitude of the factor of safety. The final choice is determined by balancing the initial cost versus replacement or repair cost of the component.

Equally important or more restrictive, or both, is the factor of safety on deflections. DDS 9110-9 [84] places limits on allowable deflection. Deflection cri-

teria would be the controlling factor for walking platforms and decks.

3.4.12 Quality Control

A degree of control is necessary because of the variations of material properties from one lot to another. The acquisition of qualified materials and the use of acceptable lots are only the beginning of quality control. A degree of control of the molding process is also necessary because of the variations of molding techniques between fabricators.

The control of quality is predicated upon measurable characteristics. The controls required by MIL-P-17549D and MIL-P-24521B are adequate for the initial and final screening of materials and products by sampling techniques. Although a process specification in accordance with MIL-P-9400 is required as an acceptance criterion prior to manufacture, neither specification addresses the quantitative or measurable characteristics of the components fabrication.

The requirements of MIL-P-9400 are detailed process descriptions of the following operations:

1. Acceptable Materials
2. Materials Storage
3. Production Equipment
4. Mold Preparation
5. Reinforcement Preparation
6. Resin Preparation
7. Lay-up
8. Curing and Part Removal
9. Trimming
10. Defects, Classification
11. Repair of Defects
12. Mechanical and Physical Properties
13. Inspection Equipment and Testing

Although the fabrication process description is complete, Operations 2 through 11 have been scrutinized by inspectors reporting affirmatively or negatively to questions concerning the process description. The interpretation of such answers is generally determined qualitatively by responsible staff personnel.

3.4.13 Statistical Quality Control

It is suggested that in addition to the quality control invoked by the material specifications MIL-P-17549D or MIL-P-25421B, a statistical method be adopted to measure the average quality level (AQL) of the in-process molding technique. A meaningful measure of AQL may be derived from the number of negative answers to the generalized questions concerning the process technique and from the number of observations.

The following ten questions are excerpted from NAVSHIPS Form 4647, In-Process-Inspection Report [85], which questions the subtle details of a typical hand lay-up process:

1. Are the mold surfaces thoroughly clean and free of old parting agents, grease, and other foreign material?
2. Were tests on raw materials made in accordance with the specification requirements?
3. Was the glass reinforcement tailored properly?
4. Does the resin formulation provide an adequate pot-life?
5. Was the resin weighed and mixed properly?
6. Did gel coat, prior to lay-up, show signs of defects?
7. Was the glass reinforcement uniformly impregnated and wet-out?
8. Were successive plies of reinforcements overlapped in proper distance and at a different location each time?
9. Was care exercised to work out the trapped air?
10. Were surfaces properly prepared for lay-up of succeeding shots of a multi-stage lay-up?

The interpretations of such answers are statistically determined by considering them to be attributes.

Specific examples of AQL versus sample size are found in Table IIA in Single Sampling Plans for Normal Inspection of MIL-STD-105D [86]. A value of AQL of 99 percent is equal to 1 negative answer in a sample of 50 observations from Table IIA, while an AQL of 90 percent allows 10 negative answers for the same sample size. Obviously, larger sample sizes or observations yield more statistical precision.

The amount of inspection sampling is derived from the reduction in cost which can be realized from the inspection process. A molder with an AQL of 99 percent in technique will rarely require large sampling plans, while a molder with an AQL of 90 percent will always need surveillance to eliminate rework cost.

3.4.14 Summary

Glass-reinforced plastics have design flexibility and the superior advantages of being lightweight and corrosion resistant for the secondary structural components of submersibles.

Structural components are fabricated by a variety of processes and from a seemingly unlimited number of reinforcement variations. The choice of process and reinforcement type can be specifically tailored to meet stringent stress levels.

The design of such complex composites is largely affected by the magnitude of the design factor of safety. The quality of the laminated composite structure is a function of both the selection of materials and the control of the materials during the fabrication process.

It is postulated that the quality should be planned by the designer, built into the laminate by the fabricator using his experience and control, and ensured by process evaluation.

3.5 Acrylic Plastics

3.5.1 *Introduction*

Acrylic plastic or methyl methacrylate polymer is by no means a workhorse material in the fabrication of submersible vehicles. In fact, structurally, acrylic plastic is unimportant since steels and other metals provide much better physical properties and since data for design are well established. But acrylic plastic has one characteristic that no metal can match—it provides sight, a visual experience for those people occupying the submersible vehicle and a human experience that no machine can create, with the exception of video techniques, which produce an image somewhat lacking in the dimension of depth.

Yet acrylic plastic is a relative newcomer when compared to the only replacement for this material with "sight." Glass was used in 1929 for the submersible vehicle *Bathysphere*, which was built and designed by Otis Barton. But glass is a poor substitute for acrylic plastic because it is difficult to machine, install, and repair. Also, reproducibility of physical properties is poor, and catastrophic failure is hard to predict.

So acrylic plastic becomes the prime choice for a transparent material because of the reproducibility of physical properties, ease of part fabrication, and predictability of parts failure.

Being a relatively new material has its drawbacks. While volumes of material are published by the manufacturers of acrylic plastic for land-based use, these publications lack data for design of subsea applications. This is probably not due to lack of interest by producer companies but rather because of the limited interest in this end use when compared to the total number of uses.

However, there is one rather large source of information of submersible-vehicle design spanning years of experience from first-hand work with acrylic plastic for this specific purpose: Dr. J. D. Stachiw and colleagues have well over 20 publications to their credit dealing with windows and submersible vehicles, two topics which will be discussed in a summary of Dr. Stachiw's work.

3.5.2 *Acrylic Windows for Submersible Vehicles*

Three basic shapes prevail in window design—conical, flat, and spherical, and each shape has an important characteristic. Conical windows are already widely used, even in earlier submersibles, and depending on the included conical angle can sustain higher critical pressures (ultimate strength) than flat windows. Conical windows, however, are more difficult to make than flat windows. Flat windows have the desirable features of ease of fabrication from readily available sheet stock and simplistic mounting in the submersible. Spherical sector windows are the most difficult type of window to manufacture, but exhibit the best resistance to compressive loads and the advantageous visual properties of larger field of view for a given flange diameter and an undistorted view when the observer is in the center of the window's curvature.

3.5.3 *Pressurization of Windows*

Test data for conical windows represent the largest bank of information available for application to empirical relations. Tests conducted by Stachiw include scale-model and full-scale windows with included conical angles varying from 30 to 150 deg in 30-deg increments. Critical pressure is a function of many parameters, such as the plate thickness over the minor diameter, t/D_i, which for any given included conical angle has an exponential relation with critical pressure. Increasing the included conical angle in windows with $t/D_i > 0.125$ yields the greatest increase in critical pressure between the angles of 30 to 90 deg; for angles greater than 90 deg, the increase in critical pressure is slight.

Axial displacement for conical windows is inversely proportional to the t/D_i ratio and included conical angle. For a given pressure load and t/D_i ratio, the smallest displacement occurs with an included conical angle of 150 deg, while at the other end of the spectrum, a 30-deg included conical angle will produce the largest displacement. Windows with a 90-deg included conical angle offer a good compromise, with good values of critical pressure and average displacement values. Temperature changes will affect the critical pressure and displacement values for a given window. Generally, a decrease in temperature below room temperature will increase the critical pressure and decrease the magnitude of displacement of a conical acrylic window.

Duration of pressurization also plays a role in the critical pressure value—a decrease in the time that a window is under pressure load will result in an increase in the critical pressure. Also, for a given value of the t/D_i ratio and pressure, the 30-deg included conical angle windows failed before the 150-deg included conical angle windows did. In order for safe and optically reliable service to be achieved, the following t/D_i ratios are recommended for the given pressures: $t/D_i \geq 2$ for 20 000 psi (including conical angle between 90 and 150 deg); $t/D_i \geq 1$ for 10 000 psi; and $t/D_i \geq 0.625$ for 5000 psi. For the proper t/D_i ratio at each given depth, the acrylic window should be able to endure at least several hundred cycles at 25 percent loading, and any catastrophic failure will be preceded by cracking sufficiently in advance to warn of the need for window replacement.

For all t/D_i ratios, there is no major problem in scaling up from a tested scale model: critical pressure data are applicable to any size window, and

displacement may be applied using a linear ratio of model diameter to full scale diameter.

For flat acrylic windows, the relationships of critical pressure versus t/D_t ratio and temperature are basically the same as a conical window with an included conical angle of 90 deg and the same t/D_t ratio. As temperature is lowered, the flat windows can sustain higher critical pressures. Diameter, t/D_t ratio, and temperature also affect the axial displacement of the flat window. An increase in the t/D_t ratio or a decrease in ambient temperature and diameter will serve to decrease the magnitude of axial window displacement at a given hydrostatic pressure. In general, displacement data for the flat window are equal to or less than displacement data of a conical window of 30-deg included conical angle for the same t/D_t ratio. Scaling factors need only be applied when predicting displacement—the ratio of scale model diameter to full-scale diameter is satisfactory. Critical pressure values can be applied to any size model without the need for scaling factors.

Spherical shell windows, although the most difficult to make, yield the highest critical pressure for a given t/D_t ratio. Spherical shell windows are 300 percent to 1000 percent stronger than either conical or flat plate windows. Critical pressure varies with the t/D_t ratio, spherical sector angle, and temperature of the pressurizing medium. Holding all the previous variables constant except the spherical sector angle, critical pressure increases to a maximum at 180 deg, that is, a hemisphere. Between 180 and 120 deg with $t/D_t \leq 0.1$, the change in critical pressure is slight. As was the case with the conical and flat windows, spherical windows exhibit an increase in critical pressure as temperature of the pressurizing environment decreases. Also, displacement of the spherical window varies directly with pressure load and temperature while an inverse relation exists with the spherical sector angle. Change in displacement is slight between the spherical sector angles of 120 and 180 deg.

3.5.4 *Flange Design*

Restraint in the radial and axial directions by the flange in which a conical window is mounted can increase or decrease the serviceable critical pressure of a conical window. Maximum restraint provides for maximum critical pressure, and the key is to make the acrylic plastic extrude through the flange. Examples of flange design yielding various degrees of restraint are shown in Figure 51.

In Fig. 51*a*, the window is elevated and when pressure is applied, the window must extrude against a converging constriction that offers increasing resistance. In the other two cases, one extrudes into free space and the second extrudes against a constriction of constant resistance. Obviously, then, flange 51*a* offers maximum restraint and should be utilized before the other two designs. Ideally, D_t should be equal to 1.14 D_f. For improved visibility, the interior side of the viewport can be countersunk to the dotted lines in Figs. 51*a* and 51*c*.

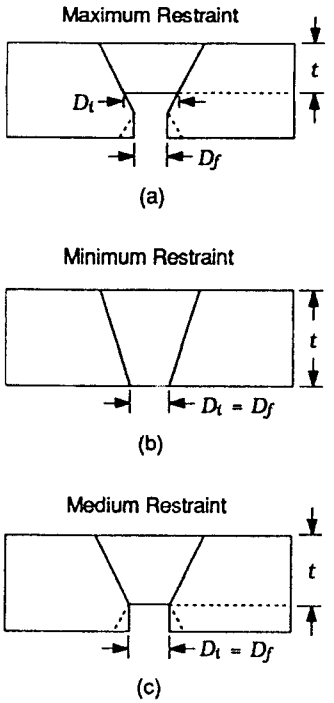


Fig. 51 Typical flange designs for conical frustrum windows [87]: (*a*) maximum restraint, (*b*) minimum restraint, (*c*) medium restraint

Although flat windows would seem to be simple to design a flange for, there are two important aspects to be considered. The ratio of the outside window diameter (D_o) to the flange diameter (D_t) is related to the critical pressure that a flat window can achieve. If the D_o/D_t ratio is below 1.5, the window will experience loss in critical pressure; conversely, a D_o/D_t ratio above 1.5 will enhance window performance. Clearance between the outer radius of the flat acrylic window and the flange wall is also important and should be considered. A clearance of 10 mils or less will improve the critical pressure properties because as the load on the window increases, interference between the window edge and flange wall will place the window in lateral compression. Low-pressure sealing of the window is also enhanced by a tight radial fit.

Spherical window flanges should be designed in a manner opposite that of conical windows. For a spherical window flange, restraint should be kept at a minimum and the flange area should be large enough to allow for some motion of the plastic. If the plastic is restrained mechanically or structurally, new stresses will be introduced and premature failure will result.

The surface finish on both the window seat cavity and window contact surfaces is important. The

ANSI/ASME standard specified surface finishes of 63 $\sqrt{}$ or better and conical angular mismatch of ±15 deg or less between the window and seat cavity. A surface finish of 32 $\sqrt{}$ or better is recommended to reduce restraint on the sliding window edge surface, thus improving the life and performance of the windows. Values of 125 $\sqrt{}$ or rougher can induce cracks on the acrylic bearing surfaces. An elastomeric seal on the pressure side should be provided to preclude low-pressure sealing problems at the window/seat cavity interface.

3.5.5 *Effect of Impact*

Impact-generated cracks should not pose any special problems if the t/D_i is on the order of 1 or greater, conical windows may use a value on the order of ½ or greater, and a safe ascent from operational depth is possible. Fracture of the acrylic plastic will occur on the side opposite the pressure face and will propagate to the compressive zone in the window. Cracks cannot migrate further into the window because compressive stresses will prevent further crack propagation. Also, as the operational depth increases and the resulting pressure load increases, the acrylic window becomes less susceptible to impact generated cracks.

3.5.6 *Window Design*

A simple design procedure for acrylic plastic windows was outlined by Snoey and Stachiw and is as follows:

1. Select the shape of window desired and multiply the submersible's operational pressure (depth) by factors of 4, 8, and 12.
2. Utilizing the appropriate short-term data curve (Figs. 52 to 54), find the t/D_i ratios associated with the converted pressures (depths).
3. The resulting three window designs should then be tested in an environment (pressure and temperature) simulating as nearly as possible the one anticipated.
4. The tests should include pressure cycling, thermal cycling, and long-term loading, representing operational conditions.
5. At the completion of the tests, the windows should be inspected for cracks and permanent deformation. Based on inspection results, one of the t/D_i ratios is chosen.

There is a simplified procedure, based on the above, which will be revised from time to time as more data become available, for a window that is of 90-deg conical shape, monolithic, less than 4 in. in thick-

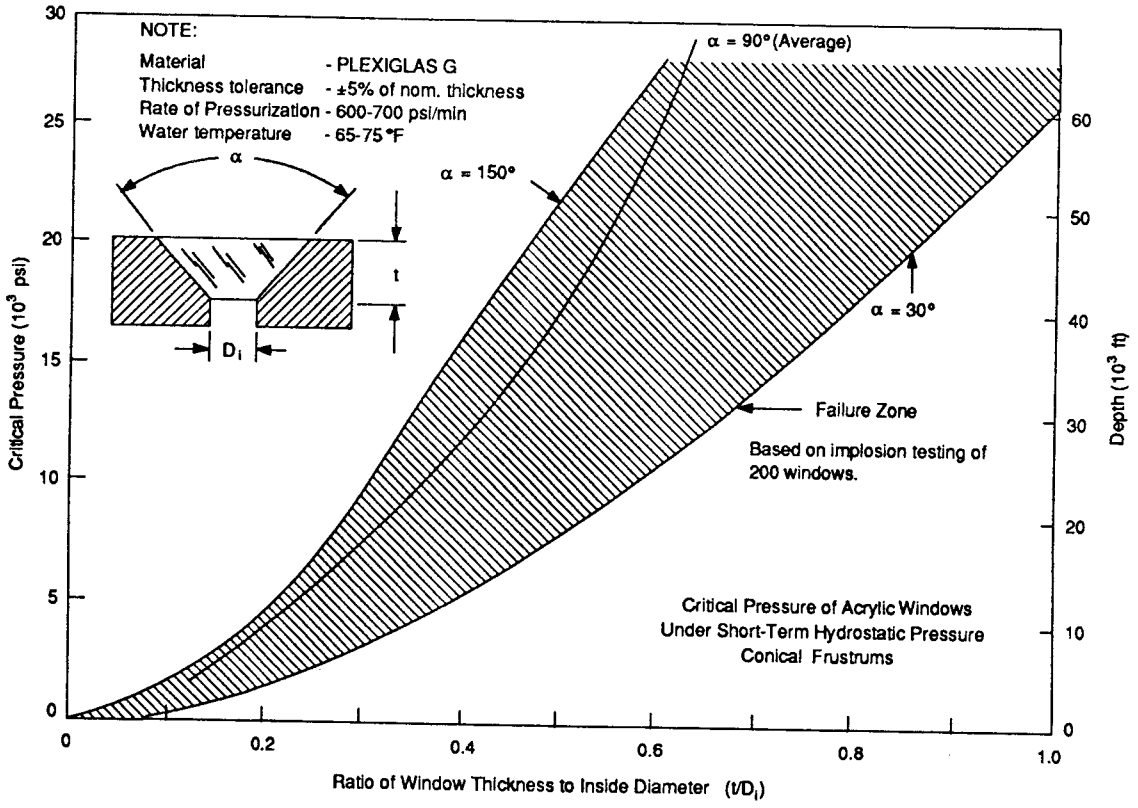


Fig. 52 Implosion test results for acrylic conical frustum windows under short-term pressurization [87]

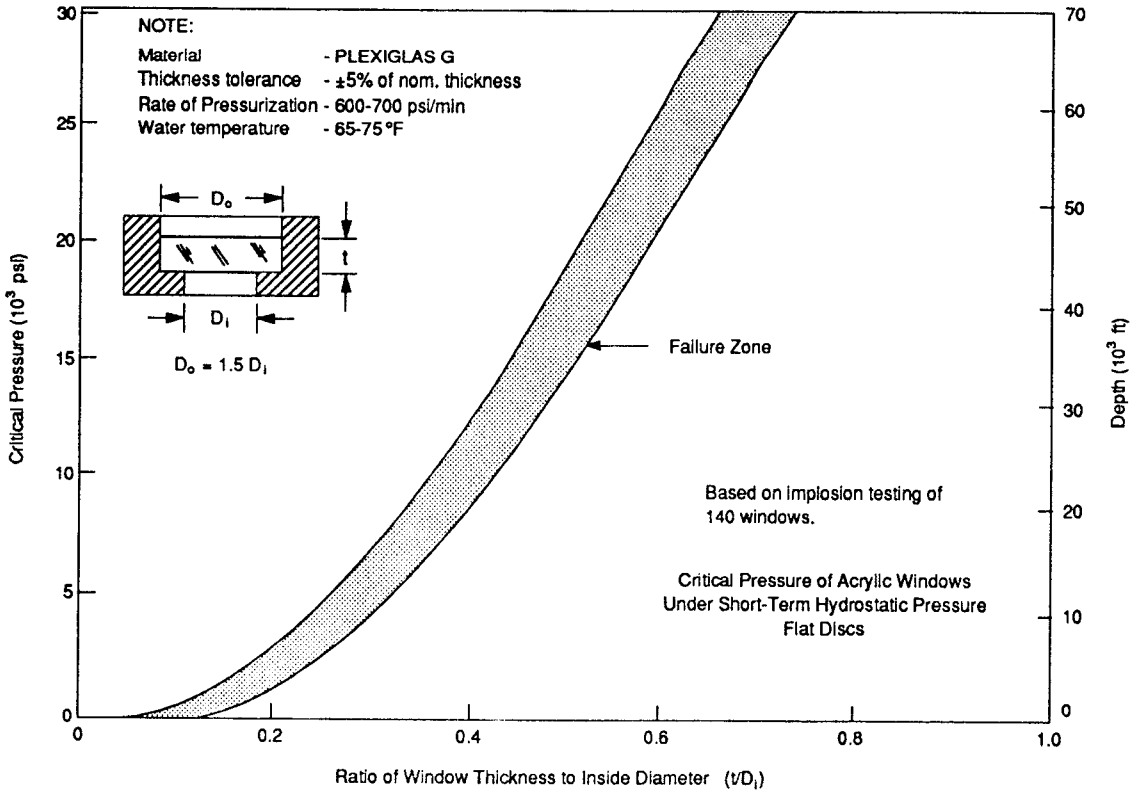


Fig. 53 Implosion test results for acrylic flat disc or cylindrical windows under short-term pressurization [87]

ness, and is fabricated by machining a commercially available cast acrylic sheet with mechanical properties equal to those in reference [88]. The conversion factor to use is 12 for any service condition, and the only test necessary is a proof test of the hull with the window installed. This implies, of course, the necessary use of a design in accordance with the section on conical acrylic windows. Windows should always be inspected visually for cracks, without removing them from the flange, prior to each diving operation.

3.5.7 *Submersible Vehicles Primary Structure*

Acrylic plastic submersible primary structures, unlike viewports, are limited to one basic design because of the material's mechanical properties and the nature of the subsea environment. Acrylic works best in compression, and a spherical craft represents the best configuration for a uniform hydrostatic load. Although the vehicle may be limited with respect to shape, there is no such problem when fabrication techniques are considered. Stachiw's work encompasses three such techniques which were all implemented successfully: casting hemispheres in a precision mold, free-forming hemispheres, and pentagon-shaped structural modules that were later joined with adhesive to form a sphere. Each method of fabrication has its advantages and drawbacks, whether they be economic or otherwise.

3.5.8 *Free-Forming Acrylic Hemispheres*

Free-forming is the least expensive method of forming a hemisphere, but the process produces the largest deviation in dimensions. Wall thickness decreases by as much as 55 percent at the pole of the hemisphere, and deviation from sphericity is of the order of 0.5 to 1.0 in. for a sphere 4 to 6 ft in diameter.

Free-forming involves clamping a sheet of acrylic plastic between two sheets of plywood, one of which has a circular cutout whose diameter is equal to that desired for the finished hemisphere. The assembly is brought to 320°F in an air-heated oven, and the space between the sheet of acrylic and the bottom panel is pressurized with compressed air, whereby the softened plastic expands like a bubble. When the bubble is hemispherical in shape, the pressure is reduced, the assembly removed from the heat, and the plastic allowed to cool and hardened. This hemisphere represents an almost finished product—trimming the acrylic sheet to form an external equatorial flange and fabrication of a suitable clamp to join the hemispheres represent the most sizeable tasks left for completion of a diving sphere. Surface

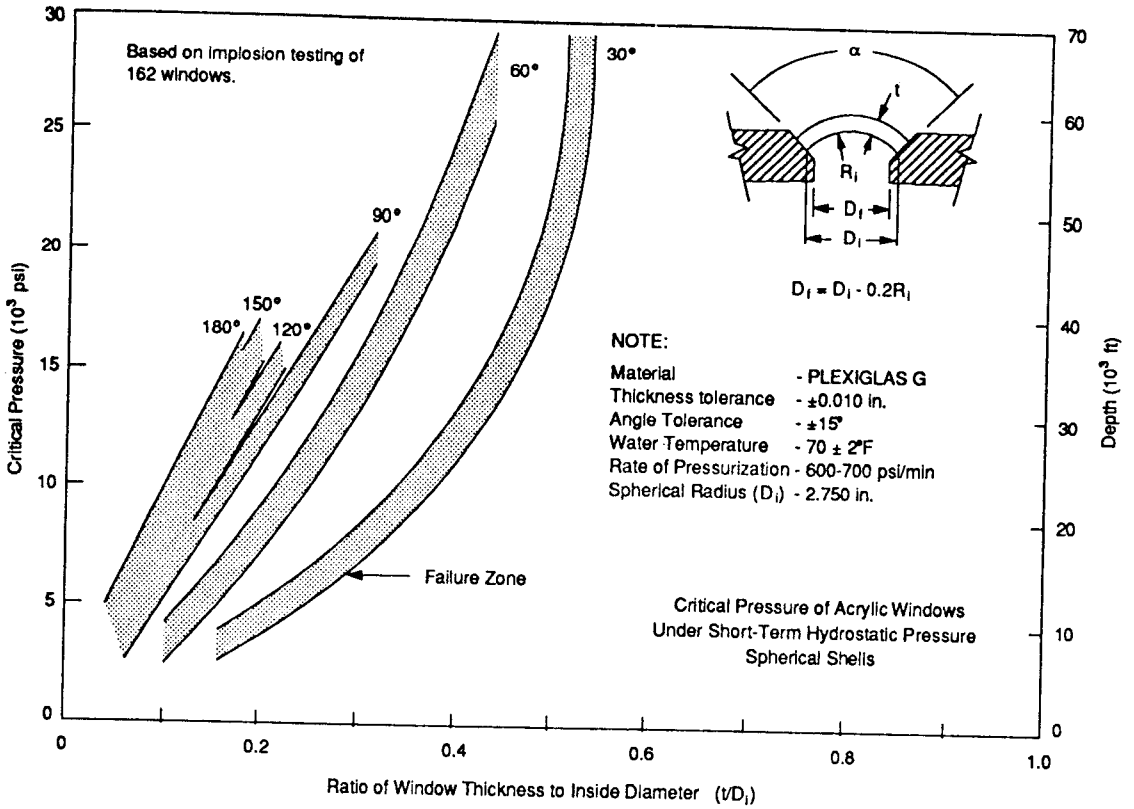


Fig. 54 Implosion test results for acrylic spherical shell windows under short-term pressurization [87]

finish will in most all cases be satisfactory as produced in the forming step; polishing will not be required.

3.5.9 Casting Acrylic Hemispheres

Casting acrylic plastic hemispheres represents an intermediate cost of fabrication but by far a superior finished product when the three fabrication techniques presented here are considered. Also, subsequent castings will help defray the initial mold cost and result in a savings if enough pieces are made. The equipment needed for this fabrication procedure includes a mold composed of a female and male form with the male form being able to be centered in the female mold; hydraulic jacks mounted on extensions of the male mold along with a pressurized air fitting in the bottom of the male mold to facilitate removal of the male form from the casting; a fitting to allow water to be pumped between the casting and the female form to facilitate removal of the casting; an autoclave and autoclave cart; a strongback; a forklift; and mixing equipment.

Fabrication of the hemispheres follows a five-step sequence, the first of which is to mix the resin and additives under atmospheric pressure and temperature according to the manufacturer's specifications. After mixing in a rotary mixer, degassing is essential. The second step is to pour the resin mix into the mold, which has been thoroughly cleaned prior to the fabrication process. The next step is simply to allow the resin mix in the mold to gel—several hours should be sufficient. Polymerization of the gelled resin in the autoclave is fourth. The best time-temperature-pressure relationships for polymerization can usually be supplied by the resin manufacturer. The fifth and most important step is the removal of the casting from the mold. The male form should be separated and raised slightly above the casting while the whole assembly is still hot. By the separation of the mold from the casting before cooling begins, tensile hoop stress is eliminated. The whole assembly should be put back in the autoclave so that the cooling process can be controlled. When the cooling process is complete, the male form is removed with a forklift and water is injected between the casting and the female form to achieve separation between those two parts and to float the casting up high enough to facilitate removal from the female form.

The only work remaining to complete the sphere is to repair any voids that occurred during fabrication and to machine the equatorial edge of each hemisphere in preparation for bonding two hemispheres together to form the diving sphere and final polishing.

Voids are repaired by gouging the affected area out, filling with resin mix, and polymerizing the mix

by placing the hemisphere back in the autoclave. Bonding two hemispheres together involves matching up the equatorial edge of two hemispheres and setting the joint dimensions by placing acrylic spacers between the hemispheres. The joint is then covered with aluminum foil tape, resin mix is poured in, and the sphere's equatorial joint is polymerized in the autoclave. Polishing completes the fabrication process.

3.5.10 Manufacturing Pentagonal Spherical Sector Module

Pentagonal spherical sector modules, the most expensive to produce, yield a sphere-quality intermediate to casting and free-forming both dimensionally and optically. The expenses associated with this fabrication method result from both the need to use a vacuum mold (female only), which must be specially made, and the amount of machining and other labor-intensive work.

The process starts by rough-cutting a circular piece of acrylic on a bandsaw, followed by milling the edge to a 200 rms finish and then chamfering one edge of the blank. The blank is then placed in the female mold (chamfered edge down), and this assembly is heated in a walk-in oven to 165°F. After approximately 16 h, the temperature is increased to 310°F. Eight hours after the temperature was set at 310°F, vacuum is applied to the mold; 15 min later the heat is turned off and the assembly allowed to cool while a vacuum between the mold and acrylic part is maintained. After cooling, the spherical sector is rough-cut to a pentagonal shape with a bandsaw, followed by exact machining with a miller. After being machined to the final dimensions, the parts should be annealed to relieve stress. This is accomplished by heating the parts to 160°F and maintaining the temperature for 24 h.

The bonding operation completes the fabrication of a sphere and proceeds by bonding six pentagonal spherical sectors into a hemisphere and subsequently bonding the two hemispheres together into a sphere. Bonding is accomplished by setting up six pentagonal spherical sectors into a jig and placing ⅛-in. spacers between each plate. A duct is formed by placing tape inboard and outboard of the space. Acrylic cement is then injected into the joint at a point just next to the spacer and allowed to cure. After the cement has cured, the spacers are removed and the process is repeated to completion of the sphere. Tests should be conducted to determine the best cement, and all cement should be degassed before using.

3.5.11 Dimensional Recommendations

Based on the experimental work of Stachiw, a thickness-to-diameter (t/D) ratio of 0.033 is safe for acrylic plastic capsules up to a depth of 1000 ft and manned operation can be certified to 600 ft. For depths greater or less than 600 ft, the t/D ratios on Fig. 55 are recommended for manned acrylic plastic capsules. Because of the time-dependent nature of

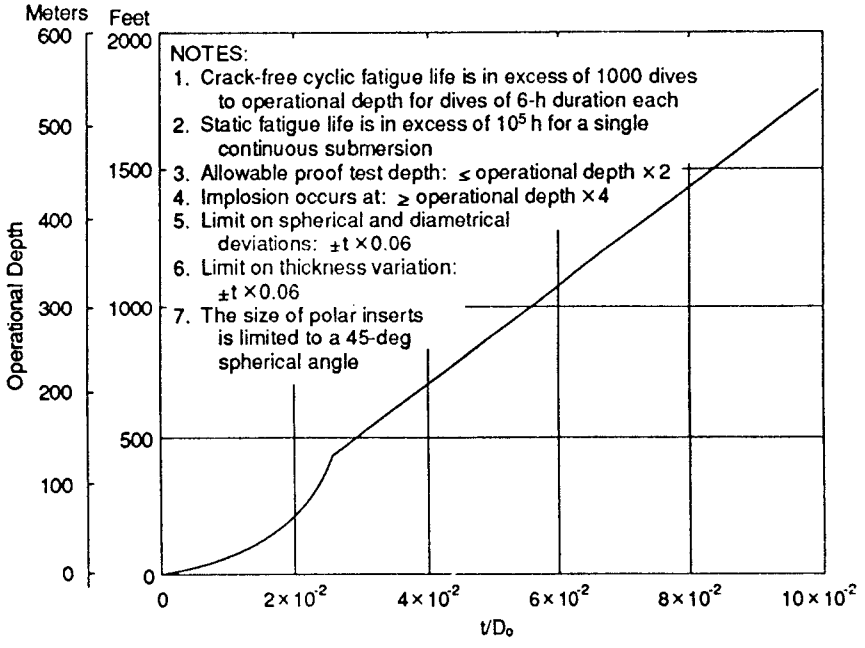


Fig. 55 Recommended dimensions of spherical acrylic capsules for certifiable manned undersea operations [89]

implosion, Fig. 56, which was derived by Stachiw from his testing on 15-in. outside-diameter scale model capsules, has been included to provide an awareness of this phenomenon.

It should be noted that Fig. 55 describes the design of acrylic spheres for the worst operational conditions, that is, high water temperature at design depth (submarine hot springs) and long submersion (ocean bottom habitats). For the case of less demanding situations, that is water temperatures less than 40°F at design depth and submersions of less than 100-h duration, the hulls can be made significantly thinner. ANSI/ASME Pressure Vessels for Human Occupancy (PVHO-1-1981) discusses the effect of temperature and should be referenced.

3.5.12 *Secondary Structures*

There might be instances in which the use of acrylic plastic in secondary structures of submersibles is desirable. Here, the structural considerations described above are not of primary concern. Rather, the characteristics are controlling. Normal fabrication techniques of vacuum forming and machining used in their plates would predominate in design and construction. Fastening would be through the flanged section using nuts and bolts.

3.5.13 *Acrylic Suppliers*

There are two basic manufacturers of acrylic plastics: E. I. DuPont, Inc., manufactures Lucite materials, and Rohm & Haas, Inc., manufactures Plexiglas. There are other manufacturers of finished acrylic plastic, but the above two supply the raw materials for such fabricators. The design information supplied by these two suppliers is extensive and should be obtained during the design process. Each produces a variety of grades, with Plexiglas G and Lucite 130 being the general-purpose grades that have been largely used in submersible design.

3.5.14 *Miscellaneous Considerations*

Acrylics have a tendency, over a period of years, to surface craze in a moist environment; and the material is subject to scratching under abrasive conditions. These problems can be reduced by proper surface polishing operations, and E. I. DuPont also offers Lucite AR as an improved abrasion-resistant grade. The Lucite A has a harder fluorinated coating, which imparts this improved abrasion performance. There are a large number of specifications relating to cast, molded, extruded acrylic materials as well as the resins used for these operations. ASTM 702 deals with cast methacrylic sheets, rods, tubes, and shapes. ASTM 788 defines the resins for molding and extrusion compounds. Relevant MIL-specifications include MIL-P-8184, MIL-P-21105, MIL-P-5425, and MIL-P-24191. The last specification applies to acrylic sheets used for illumination and signal lighting aboard ship. Both major manufacturers use ASTM test methods to describe the properties of their materials. A relevant federal specification for molding and extrusion compounds is L-P-380.

Acrylics may be cemented as detailed in reference [88]. Generally, solutions of the methyl methacrylate monomer are used, as well as several of the chlorinated hydrocarbon solvents by themselves.

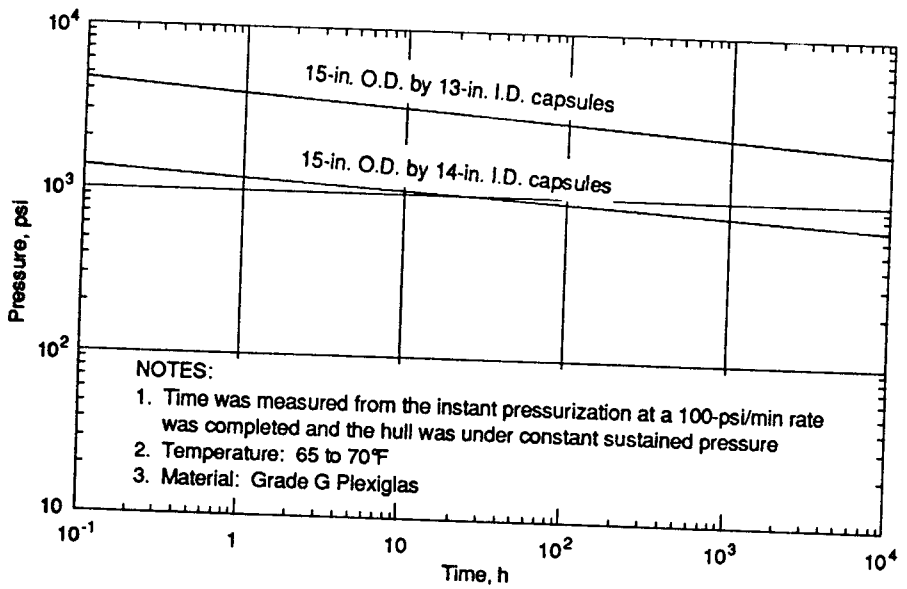


Fig. 56 Comparison of time-dependent implosion pressures for 15-in. O.D. capsules with different wall thicknesses [89]

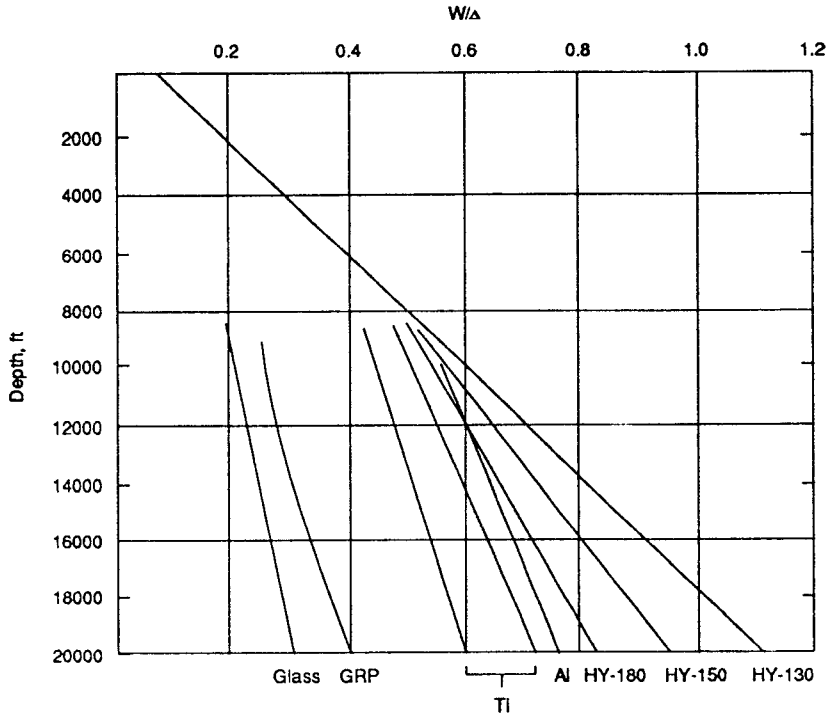


Fig. 57 Hull fraction versus weight of water displaced for various materials

3.5.15 Conclusion

Acrylic plastic is shown to be a suitable material for viewport and vehicle construction with sufficient engineering data to allow for safe design. Much more literature on the subject of acrylic plastic for submersible vehicles exists, but necessarily this section has summarized only a part of it. The literature by Stachiw referenced here is itself a wealth of information. The most current reference by Stachiw is *Acrylic Plastic Viewports, Ocean Engineering and Other Hyperbaric Applications* [90], which is probably the most in-depth summary of Stachiw's work. In addition, a national standard was published in 1977 by the American Society of Mechanical Engineers which standardizes design from a safety aspect (Safety Standard for Pressure Vessels for Human Occupancy [ANSI/ASME PVHO-1-1977]). It is left to the reader's curiosity in this subject to review these articles for a deeper understanding of acrylic plastic as used in hydrospace.

3.6 Glass

In the early 1960s, attention began to be focused on the use of glass as a structural material for deep-diving submersibles. Its advantages include high intrinsic strength, a favorable compressive strength-to-weight ratio, and, obviously, transparency, which permits the use of advanced optical inspection techniques. Glass also promises longevity for hulls, because of its chemical stability and creep resistance, although it must be noted that this promise has not been fulfilled in actual service. On the debit side, glass lacks fracture toughness. It has low tensile strength (in larger cross sections), high sensitivity to surface imperfections on surfaces under tension, and low resistance to mechanical shock. In practical designs for submersibles, glass has seemed to be most useful, and most used, as a component of a composite primary structure, for instance (as in the submersible *Deep View*), a structure composed of a glass hemisphere, forward, bonded to a steel cylinder (in that case, HY-100).

As a graphic example of the comparative strength of glass as a hull material, Fig. 57 shows the relationship of the "hull fraction," the ratio of the weight of a pressure hull (for a representative design, in this case a spherical hull submersible) to the weight of the water it displaces, to depth, for various materials used, or proposed for use in deep-diving submersibles.

When glass is used with other materials in pressure hulls, the design of the joints is critical. These must be designed in such a way as to allow practically no tension in the glass hull. Normally, the steel-to-glass joint is designed so that it will have the same radial deformation as the glass. Analysis, experiment, and extensive testing have shown that a round edge on the glass provides better performance than a flat edge or a "V" edge, and the round-edge design has been successfully used. Proper joint design is required not only in the mating of glass and steel (or

Table 22 Range of properties of glass

Specific Gravity	2.18 and up
Thermal Conductivity, (cal/s·cm·°C)	0.0025
Coefficient of Expansion (10^{-7}/°C)	8 to 127
Photoelastic Fringe Constant (5461 Å)	1150 lb/in. order
Light Transmission	up to 90%
Dielectric Constant	3.8 and up
Poisson's Ratio	0.18 to 0.24
Young's Modulus of Elasticity (10^6 psi)	8 to 17
Compressive Yield Stress (10^6 psi)	0.8 to 1.25
Flexural Strength, treated (psi)	
experimental	120 000
guarenteed minimum	35 000
Creep after 10 years at 200 000	zero

other metal) hull components, but also for access hatches and hull penetrations.

Table 22 gives ranges of selected properties of glass of interest to the submersible designer.

Early work on glass as a structural material for submersibles, and on glass-and-steel submersibles, was facilitated by the development and provision of cast spheres and hemispheres of up to 44-in. diameter (the largest diameter available at the time) by glass manufacturers such as Corning Glass Works and Pittsburgh Plate Glass Corp. Good castings are mandatory. Also, in order to eliminate as far as possible crack propagation, which can occur with cyclic loading, it is necessary that the surface of the glass be fire-polished or mechanically polished.

In addition to its use as a hull component, glass has also been used successfully as a viewport material, most notably by the Japanese in their submersibles *Kurushio* and *Yomiuri*. In this case, also, proper attention must be paid to the glass-to-steel joint design.

The designer who elects to use glass in a submersible structure should be cautious: there are comparatively little experimental data on its use, and the only submersible which embodied it extensively (*Deep View*) was retired some time ago and did not undergo an extended service life. However, the conclusions of early scientists and engineers who investigated the use of glass in submersibles [91] are valid:

1. Shell stability formulas developed and tested for metal shells predict accurately the implosion by elastic instability of glass shells.

2. Stress raisers, such as sharp radii at the root of stiffeners and deep scratches in shell surfaces, do not decrease the stability or compressive strength of a glass shell, if they are located in areas of the structure where only resultant compressive stresses occur.

3. The resistance of glass shells to underwater shock, when tested at static external pressures of 5 to 10 psi, is several orders of magnitude lower than that of 7075-T6 aluminum shells of identical dimensions; however, the resistance to underwater shock of glass shells increases or remains constant with the increase of external static pressure, while that of aluminum shells markedly decreases.

4. Glass shells tested for 3000 cycles under fluctuating pressure appear to have resistance to compressive fatigue stresses several magnitudes larger than that of fiberglass-epoxy laminates.

5. The elastic stability of cylindrical and spherical glass shells fabricated from unbonded beveled staves or unbonded beveled pentagons, respectively, is identical to the elastic stability of monolithic shells of identical dimensions and material.

4. Buoyancy: Ballast Materials

4.1 Overview

The payload capacity and the general performance of submersible vehicles can often be improved by providing the submarine with positive buoyancy, either permanently or temporarily, through the use of buoyancy materials. In the design process, this can sometimes be achieved, in part, by weight control, using hollow instrument cases, sealed tubular frame members, and other lightweight dual-purpose structures. Also, some components, such as "wet" motors, can be located outside the pressure hull. However, the utility and net weight savings of these approaches diminish at greater depths, because heavier construction is required to withstand the correspondingly greater hydrostatic pressures encountered. Submersibles designed for deep operation therefore have come to be provided with special buoyancy materials. These materials are most frequently solid materials of the types that have come to be called "foam composites" or "syntactic foam," which consist of small hollow spheres of glass or phenolic embedded in a matrix of plastic, or other material. Usually, but not always, structures with spheres larger than 200 to 300 μm are called "foam composites," and those with microspheres, "syntactic foam," but the latter term is also used for all. Liquids are also used as buoyancy materials, particularly in very deep diving vehicles, and under certain conditions gases (compressed air, or a "JATO" bottle) are used to provide emergency buoyancy. Finally, additional buoyancy can be provided by the use of pressure proof spheres, preferably fabricated from the same material as the pressure hull. For example, *Alvin* uses a combination of syntactic foam and titanium spheres to provide buoyancy augmentation.

The use of solid materials to provide buoyancy and at the same time a measure of strength is not new in submarine design. For many years, control surfaces such as rudders and diving planes were filled with a mixture called "pitch and pine," which has low weight but still provides a void filler with enough strength

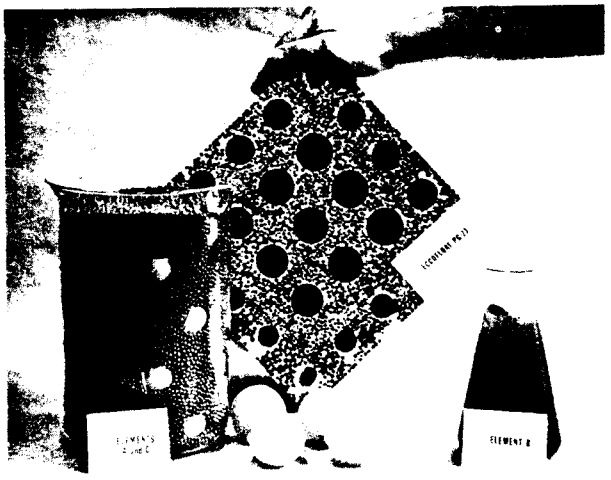


Fig. 58 Syntactic foam ingredients

to minimize crushing of the surfaces by the pressure (the so-called "hungry-horse" effect).

In the 1960's, foam composites and syntactic foams were developed which had several significant advantages over "pitch and pine." These materials, as described in detail in the following section, are characterized by low density, high hydrostatic strength, low water absorption, a bulk modulus equal to or slightly higher than that of sea water, fabricability to irregular shapes, the ability to be used as filler, when "foamed-in-place," and immunity to catastrophic failure. Syntactic foams with weight between 42 and 46 pcf that have successfully withstood a hydrostatic test pressure of 10 000 psi were developed and put in use by the end of the decade; since that time foams with even lower densities have been developed. Manufacturers should be consulted for costs and characteristics of available syntactic foams.

4.2 Solid Buoyancy Materials: Syntactic Foam

There are very few solid materials strong enough for deep-sea use, yet light enough to provide substantial buoyancy. Solid polyethylene plastic, for example, with a specific gravity of 0.91, is extruded around marine power cable for both electrical insulation and buoyancy [92]. Such a heavy material is not practical for submersible vehicles. Much more useful are light, but strong, plastic foams, especially the plastic/glass composite material known as *syntactic foam.*

4.2.1 *Definition of Syntactic Foam*

What is syntactic foam? The average dictionary is not much help: *syntactic* is defined as "arranged in accordance with an orderly or mathematical system." For submersibles application, however, we have developed our own definition: "Syntactic foam is any materials system comprising hollow *spherical* particles dispersed in some form of binder" [93]. Note that the only requirements of the filler elements are that they be spherical and hollow; nothing is said about size or composition. Likewise, the binder element is not specified and thus may be plastic or ceramic or concrete, rigid or flexible, etc.

Typical syntactic foams used in hydrospace applications contain fillers in the form of tiny glass bubbles, or microballoons. Because of their small size—a diameter range of 40 to 80 mm is common—microballoons appear as fine white powder to the naked eye. Their beautiful symmetry can only be appreciated under a powerful microscope. The microballoons are mixed into a liquid thermosetting plastic, such as epoxy or polyester resin. The resulting mixture may have a viscosity varying from that of heavy cream to a semisolid paste. Once catalyzed, the resin binder hardens so that the foam becomes a rigid mass with remarkable properties. It is much lighter than the plastic resin it is made from, yet much more resistant to hydrostatic pressure than are the microballoons by themselves. For the quality of lightness to be further enhanced, plastic macrospheres up to several inches in diameter may be interspersed in the microballoons-resin mix. Figure 58 shows some of the ingredients of syntactic foam, and Fig. 59 is a photomicrograph of glass microballoons.

4.2.2 *Behavior under Pressure*

A conventional hollow metal or glass float functions only as long as structural integrity is maintained. If the walls of the pressure vessel are damaged, the interior rapidly floods and all buoyancy is lost. If this failure occurs at great depth, the resulting

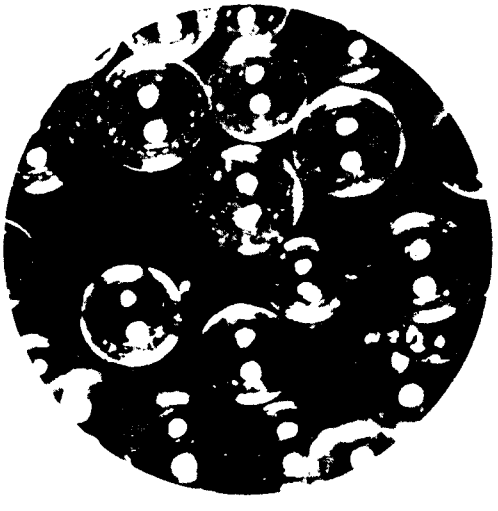


Fig. 59 Photomicrograph of glass microballoons® (×200 magnification)

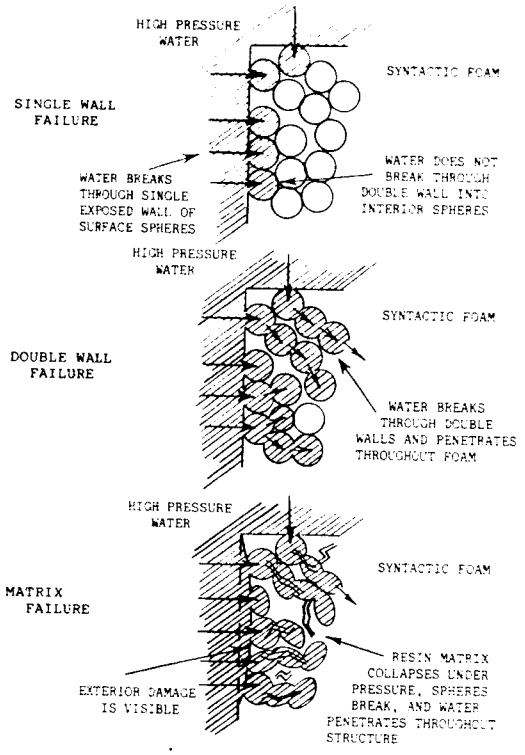


Fig. 60 Failure mechanisms of syntactic foam

implosion can be very violent. In contrast, syntactic foam, consisting of a multitude of tiny pressure vessels embedded in a tough plastic binder, is highly resistant to damage and relatively gradual in its mode of failure, even when pressurized far beyond its service limit.

Typically, loss of buoyancy in syntactic foam is the result of two mechanisms: absorption and compression [94]. Loss due to compression can be further subdivided into temporary (elastic deformation) and permanent (inelastic deformation). Most syntactic foams are quite rigid with bulk moduli approaching or even exceeding that of water; therefore, buoyancy loss due to compression is usually small and any deformation under pressure is largely elastic.

For the moment, we shall concentrate on absorption losses, which can be subdivided into three distinct kinds of failure: "single-wall" failure, "double-wall" failure, and "matrix" failure.

In describing these kinds of failure, we shall assume the syntactic foam to be in direct contact with water, and we shall use the term "sphere" to designate any of the aggregate particles, regardless of size. Figure 60 illustrates the concepts. Single-wall failure is the least serious of the three; in fact, it is unavoidable to a certain extent. The majority of the spheres are encapsulated on all sides by the plastic binder and thereby gain a considerable measure of strength. The outermost layer of spheres, however, is tangent to the surface of the casting and not so well reinforced. In addition, the spheres are not of exactly equal strength: some are exceptionally strong, some are of less-than-average strength. The weaker surface spheres break, and buoyancy is diminished. As long as breakage is confined to the surface, however, the buoyancy loss is small. Eventually, only strong spheres remain on the surface, and any pathways into the interior develop sufficient resistance to halt or greatly retard the entrance of additional water. Failure does not propagate to the interior because to do so the water must break through a *double thickness* of sphere walls—hence the term "single-wall" failure.

Assuming that pressure is held constant at some value equal to or less than the service limit, a plot of absorption versus time becomes essentially linear; the line will be either horizontal or of small positive slope, depending upon the pressure. Figure 61 illustrates a typical syntactic foam under these conditions with the slope of the curve somewhat exaggerated.

What happens if the pressure is increased? Obviously, more spheres are broken and the rate of water intrusion is accelerated. This has the effect of increasing the slope of the curve in Fig. 61. The curve remains more or less linear and predictable as long as the pressure is not great enough to break through two sphere walls. However, if pressure is increased to the "double-wall" failure point, water will infiltrate the entire syntactic structure and cause a serious loss of buoyancy. This type of drastic failure is illustrated in Fig. 62.

Greatly excessive pressure may have another effect on syntactic foam; at some point, the resin binder or matrix will be unable to support the mechanical loads imposed by the pressure. Matrix failure is collapse of the syntactic structure by compressive (or shear) buckling of the resin binder. Unlike the other two modes, matrix failure results in visible external damage; it also occurs much more rapidly. The hydrostatic pressure that results in matrix failure is called the *crush pressure* of the foam. Due to the

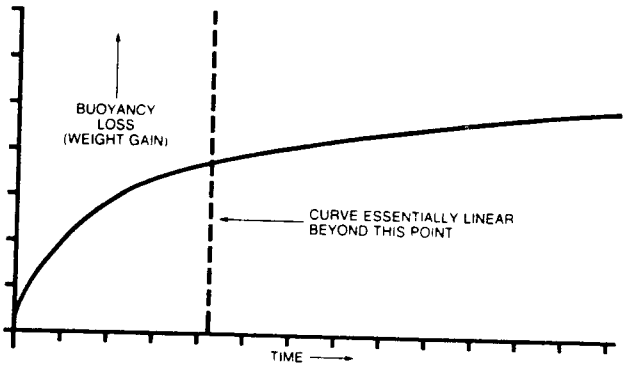


Fig. 61 Performance of syntactic foam at constant hydrostatic pressure

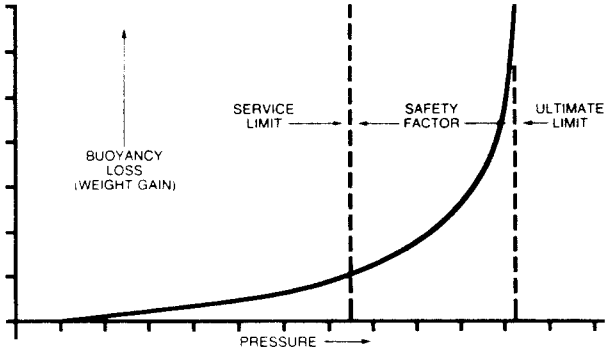


Fig. 62 Performance of syntactic foam at increasing hydrostatic pressure

overall rigid structure of syntactic foam, the failure at crush pressure is never as violent or dangerous as the implosion of other types of buoyancy, for example, metal air cans or spheres.

An important consideration in achieving the optimum hydrostatic performance of a syntactic foam is to keep the spherical particles as small as possible. Smaller hollow spheres are more likely to have uniform walls with less imperfections than larger spheres. Single-wall failure of smaller spheres also has less effect on buoyancy because smaller air voids are being filled with water. So-called "solid" syntactic foams (those containing only glass microballoons) are the strongest type of foam for a given weight or density. However, their higher cost and their need for special manufacturing conditions restricts their use. Specific applications for different types of syntactic foam will be reviewed in a later section.

4.2.3 Depth versus Density

The most important characteristics of a syntactic foam are density, service depth, and crush depth.

Table 23 Safety factors generally used for syntactic foams

Use Depth as % of Crush Depth	Nature of Application
50–65%	Critical: Manned submersibles, high-performance and/or long-service buoys
65–80%	General Purpose: Unmanned submersibles, buoys in moderate-duration service
80–90%	Non-Critical: Short-term or expendable buoys

The service depth, which is usually about 60 percent of the crush depth, corresponds to the pressure at which the foam is expected to lose no more than 1 percent to 2 percent of its buoyancy after 30 days of exposure [95]. Water absorption over longer exposure periods will continue at a progressively slower rate because of the inverse exponential nature of the absorption mechanism. Small floats may show higher absorption rates than larger floats, due to their higher surface area-to-volume ratio.

Service depth is the determining factor in material selection. The object, of course, is to choose a material that is both safe and economical. Syntactic foam products can be safely specified for most applications in accordance with published data with little or no additional safety factor, and routine quality-control procedures can be accepted as sufficient proof of reliability. However, those applications involving very valuable equipment or the possibility of injury to personnel may require more elaborate specifications or higher safety factors to minimize the risk of error or accident. Test methods and standards are discussed in greater detail below. Table 23 illustrates the generally accepted practice for rating syntactic foam.

The choice of service depth also establishes the allowable range of densities. Although it is to the

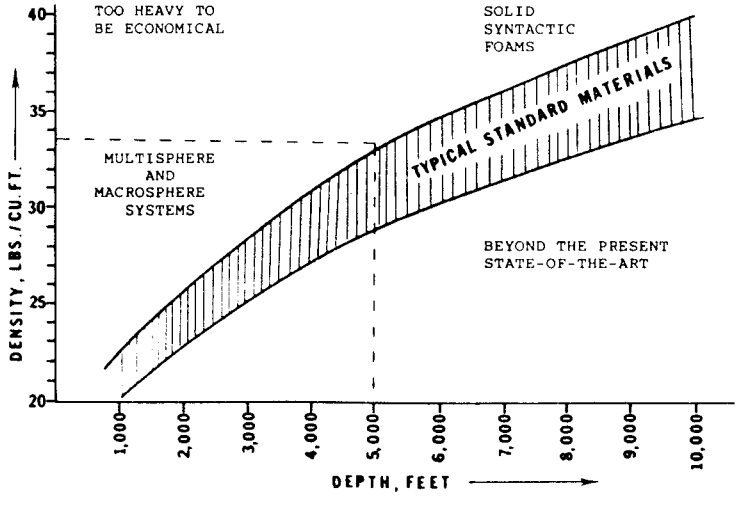
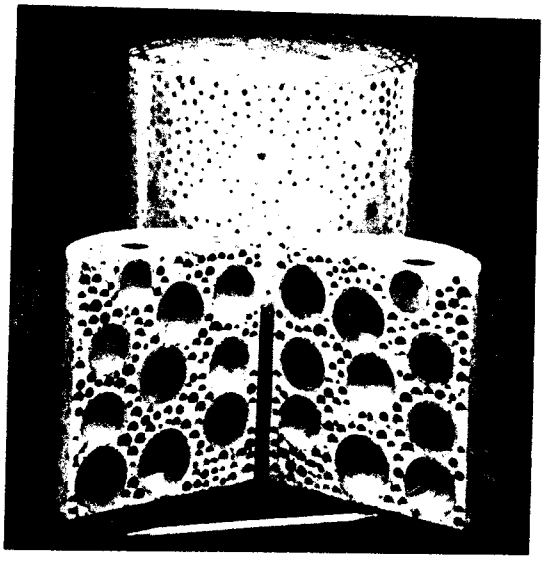
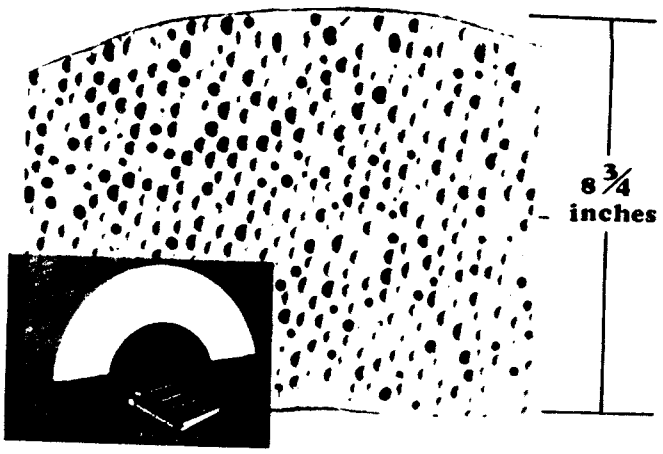


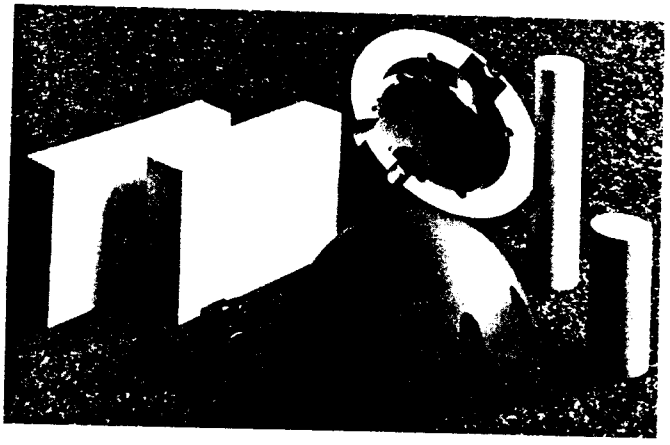
Fig. 63 Density versus depth rating of syntactic foams



(a)



(b)



(c)

Fig. 64 Types of syntactic foam: (*a*) Multisphere syntactic foam with three sphere types: 3- and 0.5-in.-diameter plastic macrospheres, glass microballoons. (*b*) Macrosphere syntactic foam with two sphere types: 0.5-in.-diameter plastic macrospheres, glass microballoons. (*c*) Solid syntactic foam with one sphere type: glass microballoons

advantage of both the manufacturer and the user to employ the lightest possible material, and research is always underway to lower the density of flotation materials, there is only a limited range of densities available to meet any particular requirement with reasonable economy. Varying densities are achieved primarily by using hollow spheres of varying density. Heavier spheres generally have thicker walls so they are stronger under pressure; they are used to make heavier foams that are stronger and can be used at deeper service depths. It is also possible, although less common, to vary the density of a foam by using macrospheres of varying sizes to increase their packing efficiency or by using resins of varying density or viscosity.

Figure 63 illustrates the densities which can be expected for a representative line of products that are currently available. This information is only approximate; exact densities depend on the size and shape of the float, and some materials may be excluded by other requirements.

4.2.4 *Types of Syntactic Foams*

The principal classes of syntactic foams are distinguished by the relative size of their spherical components. Each class has its own typical service depth range, as shown in Fig. 63, plus other advantages or disadvantages that affect its usefulness. Figure 64 compares the appearance of the three classes.

1. *Multisphere systems* employ macrospheres of varying diameters, in combination with glass microballoons, to achieve very low density and cost. The relative size of the macrospheres is chosen to obtain better sphere packing than if only one size were used. The basic idea is to get the smaller macrospheres to fit into the interstices between the larger macrospheres. Macrospheres used in syntac-

tic foams are commonly made from plastic (either thermoset or thermoplastic) and they may contain a reinforcing filler, such as fiberglass, in the shell for added compressive and shear strength. Multisphere systems are best suited to the molding of large floats; in smaller shapes, it may not be possible to obtain high packing due to the size of the large macrospheres. The disadvantages associated with larger spheres discussed earlier are most evident in multisphere foams where large spheres are needed.

2. *Macrosphere systems*, also called "aggregate" syntactic foams, use macrospheres of one size in combination with glass microballoons. Typical macrosphere diameters range from 0.125 in. (3 mm) to 3 in. (76 mm). The densities of this type of foam are usually higher than those of multisphere systems but lower than those of solid syntactic foams, which employ only one size of sphere. Macrosphere foams are moldable in large shapes, up to 60 ft^3 and they provide good long-term performance to depths as low as 5000 ft (1500 m). This varied and versatile class of material is probably used in greater quantity than any other class of syntactic foam.

3. *Microsphere systems*, also known as "solid" syntactic foams, contain only glass microballoons, which typically do not exceed 300 μm in diameter. Because the air cells are so small, this type of foam appears to be a homogeneous solid. The absence of large air cells permits solid syntactic foam to be cut, drilled, and machined without any effect on performance. The newly exposed surface is as hydrostatically sound as the original as-cast surface. Solid syntactics are the strongest foams available, with some grades rated for service as deep as 40 000 ft (12 000 m). Their superior strength is due to the rigid glass walls that enclose each microscopic air cell. Because these foams contain only one size sphere and because the cost of glass microballoons is considerably more than that of plastic microspheres, solid syntactics tend to be heavier and more expensive than systems containing macrospheres. They offer outstanding reliability and safety, however, for critical applications, such as manned submersibles. One important consideration for this type of foam is that it is more difficult to mold than other syntactics, so it is usually available in only certain sizes and shapes. Cubic foot (0.028 m^3) blocks, 6 by 12 by 24 in. (150 by 300 by 600 mm), are the most common commercial shape. It is possible to make large, complex solid syntactic foams by bonding together blocks that have been machined to the appropriate shape.

4.2.5 *Standards and Testing*

Since syntactic foam technology is relatively new, very few standards have been established for its construction and testing. Applicable ASTM standards are listed below; note that these standards are primarily designed for solid-type foam.

ASTM	D	638	Tensile Strength
ASTM	D	1623	Tensile Strength
ASTM	D	732	Shear Strength
ASTM	D	2926	Bulk Modulus
ASTM	D	1622	Density
ASTM	D	695	Compressive Strength
ASTM	D	2736	Hydrostatic Strength (discontinued in 1984)
ASTM	D	2735	Cyclic Immersion (discontinued in 1984)

The principal U.S. military specification on the subject is MIL-S-24154A (SHIPS), administered by the U.S. Navy Ship Engineering Center. MIL-S-24154A covers only solid syntactic foam and is intended for highly critical, deep-diving manned systems. Less frequently encountered is MIL-S-24167A (SHIPS), a specification for low-performance void-filler material. The lack of widely accepted standards for the many other grades of syntactic foam has resulted in a diversity of product descriptions and some confusion between suppliers and users as to what constitutes acceptable performance.

The basic function of a syntactic foam float is to provide buoyancy at depth; with this in mind, the three major objectives of the quality-control program must be to insure that [96]

1. The float is of the correct size and shape.
2. The density of the float is low enough to create sufficient buoyancy.
3. The material is strong enough to resist crushing and water absorption so that buoyancy does not decline below some minimum acceptable level over the life of the float.

The first two considerations may be checked by simple measurements. The third—and perhaps most important—is more difficult since it involves hydrostatic testing at high pressures. Large pressure vessels, capable of generating the required high pressures,

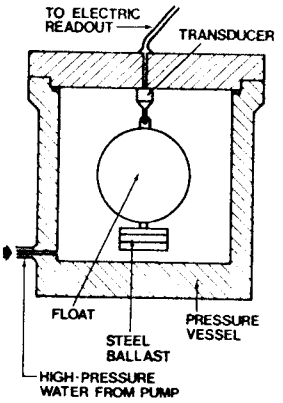


Fig. 65 Schematic of pressure vessel instrumented for continuous buoyancy measurement

are expensive to build and operate and, thus, are rare. For this reason, much of the testing of syntactic foam is done in small vessels on samples or "coupons" taken from or molded concurrently with the float.

The sampling approach relies on statistical inference to judge foam quality and thus must be applied with caution. More reliable is weight-gain information from a full-size test, since it simulates the actual performance of the float. Best of all is full-size instrumented data in which true buoyancy under pressure is measured by one or more strain gages. Such data, in conjunction with measurements of float weight before and after test, permit analysis of both elastic and inelastic losses as well as water absorption. Figure 65 is a schematic view of a pressure vessel instrumented for buoyancy measurement, and Fig. 66 is the sort of data typically generated by such equipment.

4.2.6 *Special Features*

The two most common special features added to syntactic foam floats are

- an outer skin or coating to protect the foam against damage or to improve appearance and
- straps, brackets, or inserts for attachment of the float to the vehicle or for mounting of accessories.

Because such features add to the weight and cost of the system, they should be used only when necessary.

Fiberglass is the preferred covering material; elastomerics are attractive but may not stand up to the rough handling experienced by most marine equipment. Fiberglass thickness may vary from 1/16 to 1/4 in. (1.5 to 6 mm), depending on the desired tradeoff between ruggedness and weight penalty. The fiberglass may be applied after molding or even after installation of the syntactic foam. In some molding processes, fiberglass can be integrally molded on the outer surfaces of the foam to provide a covering of superior durability. To improve appearance and visibility, the fiberglass may be pigmented or painted.

The preferred method of attachment of a foam float to a submersible or other device is by strap. Straps may be of metal or a synthetic material, such as Kevlar (DuPont). Strapping is preferred because it is simple, flexible, inexpensive and does not require modification of the float. Another common approach is to mold or drill holes all the way through the float for bolts. When these methods are not suitable, metal attachment points, either brackets for external bolting or internally threaded inserts, can be designed into the float. Since it is difficult to locate such brackets or inserts accurately, the attachment system should be designed to accommodate the largest possible dimensional tolerances. Figure 67 illustrates a successful insert design.

4.2.7 *Do-It-Yourself versus Factory-Made*

Some manufacturers offer syntactic foam kits that permit the user to make his own float. The depth capability of such kits is generally limited to 12 000 ft (3660 m) or less, and the resulting foam is generally inferior in strength/weight efficiency to factory-made material. Very few users have the equipment and skill necessary to make first-class castings, particularly when the product is large or complex and the rated service depth is more than a few hundred feet. There are cases, such as the foam filling of hull sections, when the use of kit materials is mandatory. In most instances, however, the user is well-advised to purchase syntactic foam in factory-finished form.

4.2.8 *Other Characteristics*

In addition to its buoyancy function, syntactic foam has other characteristics that can be useful to

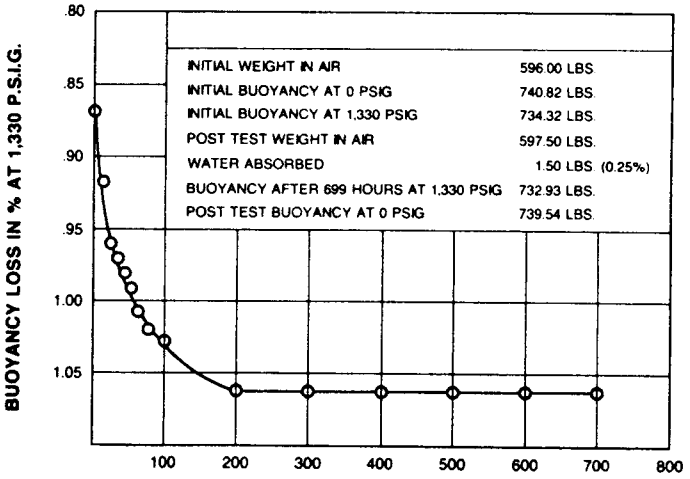


Fig. 66 Percent buoyancy loss versus time for the 700-h hydrostatic creep test at 1330 psig (3000 ft)

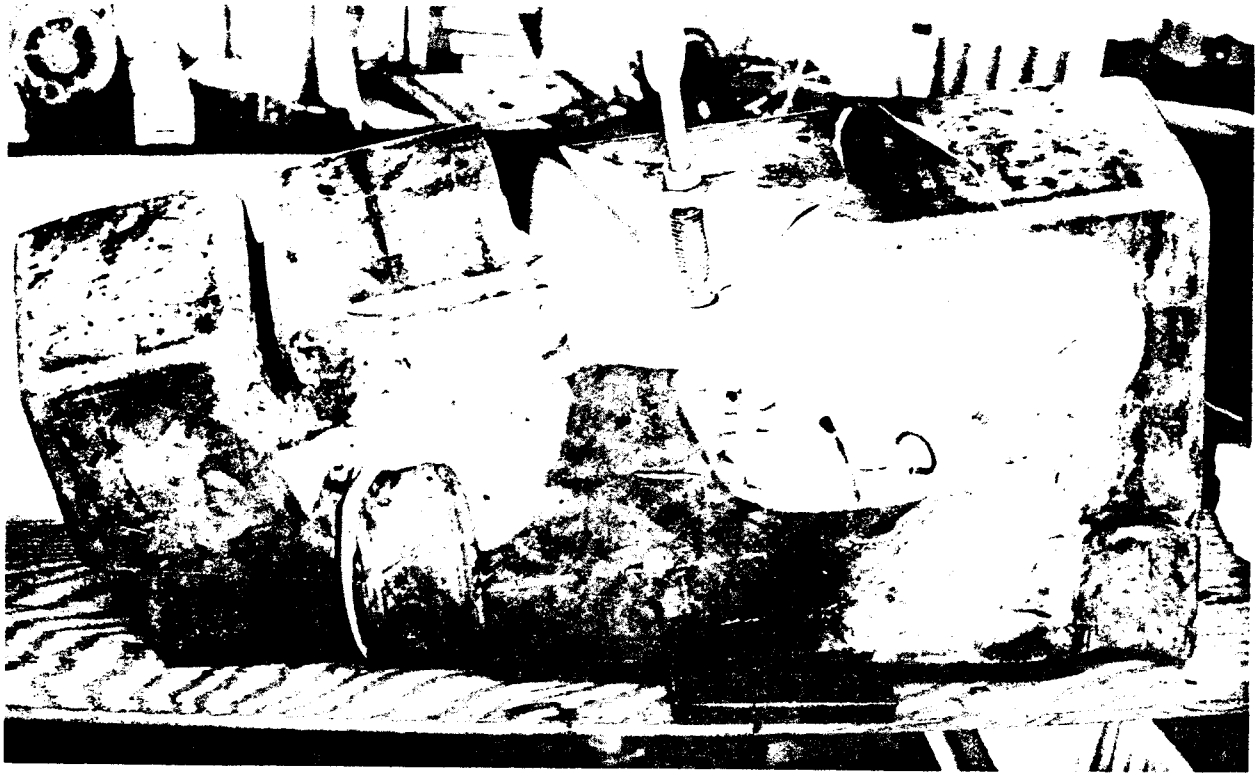


Fig. 67 Stainless steel insert after testing to failure

the submersible designer. Rigid foam has considerable structural strength and can bear some, if not all, of the vehicular loads. Another characteristic of rigid syntactic foam is its high specific stiffness, which results in unique acoustic properties. Syntactic foam can be made flexible by substituting an elastomeric resin, such as polyurethane, for the rigid plastic normally used as a matrix binder. And all syntactic foams, because of their high air content, are good thermal insulators.

4.2.9 Application Notes

Figure 68 shows an example of standard blocks of high-performance solid syntactic foam applied to a deep-diving work system—in this case, on the U.S. Navy's CURV III vehicle. The blocks have been cut to shape, mounted inside a metal frame, and lashed down with straps. Note that the distribution of blocks has been carefully arranged to keep the center of buoyancy coincident with the center of gravity.

Figure 69 demonstrates another approach to submersible buoyancy on BAC's *CONSUB 2*. Custom-molded segments form the upper portion of the vehicle and are secured with straps. Most submersibles have the flotation mounted as high as possible to maximize stability.

The body of the retrieval vehicle shown in Fig. 70 is made from cylindrical moldings of syntactic foam. The vehicle was developed by the Naval Ocean Systems Center for NASA's use in recovering solid-fuel rocket boosters dropped by the space shuttle.

In Figs. 71 and 72 we see that even tethered diving bells employ syntactic foam. Perry Submarine's Observation Manipulator Bell is fitted with a number of flotation segments bolted inside a steel frame. The segments have stainless-steel attachment brackets, bonded in place after molding.

Very different is Hydro Products' diminutive RCV-125 system, shown in Fig. 73. Its hull is made from solid syntactic foam, hollowed out to accept instru-

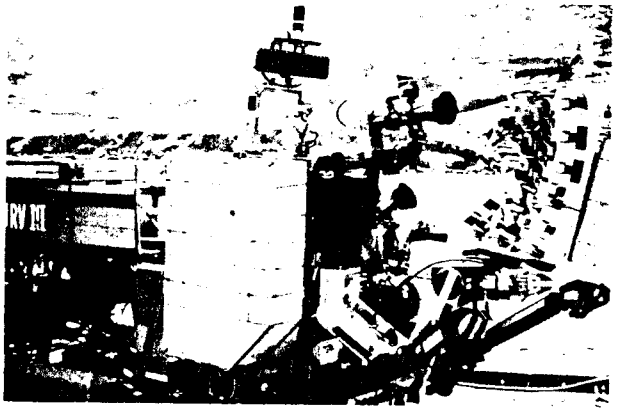


Fig. 68 Solid syntactic foam blocks on *CURV III*

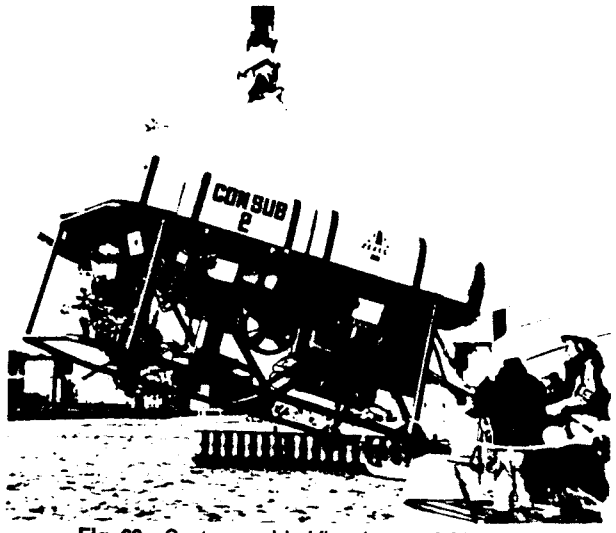


Fig. 69 Custom-molded flotation on *CONSUB 2*

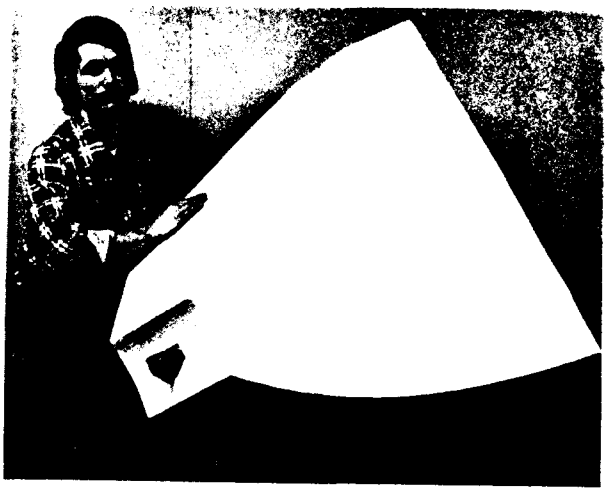


Fig. 71 Molded segment for a diving bell float (see also Fig. 72)

ments and equipment. The result is an extremely compact and efficient vehicle.

The syntactic foam float shown in Fig. 74 provides buoyancy for the *RECON III* (Perry Offshore). Molded-in wells in the syntactic foam float make room for the vertical thruster and sonar equipment. A number of stainless-steel studs securely anchored in the float bottom provide attachment points.

The well-known *Alvin* submarine (Fig. 75), operated by the Woods Hole Oceanographic Institute, uses syntactic foam along with hollow buoyancy spheres and variable ballast spheres to control its lift. The shaded areas in the figure indicate syntactic foam.

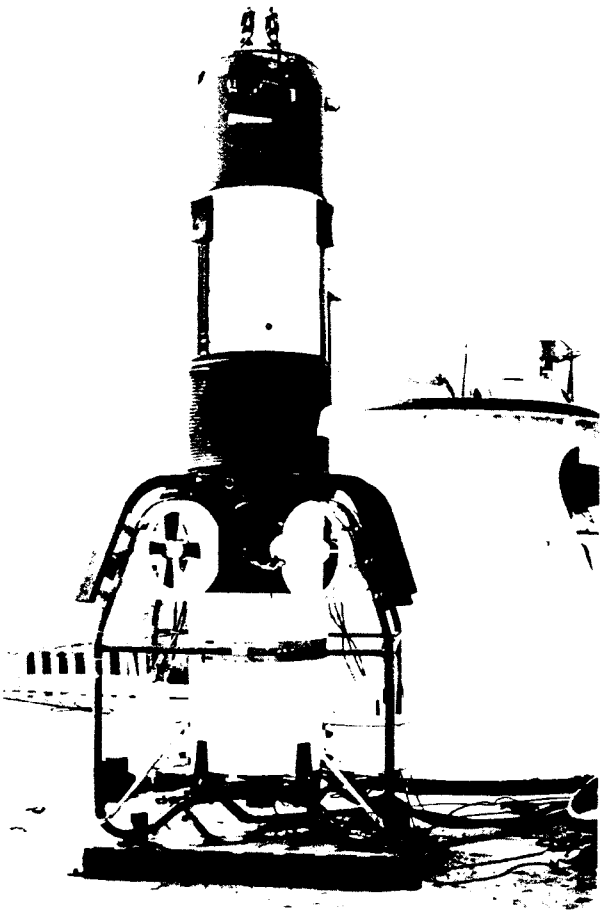


Fig. 70 Syntactic foam cylinders form the body of this NASA retrieval vehicle

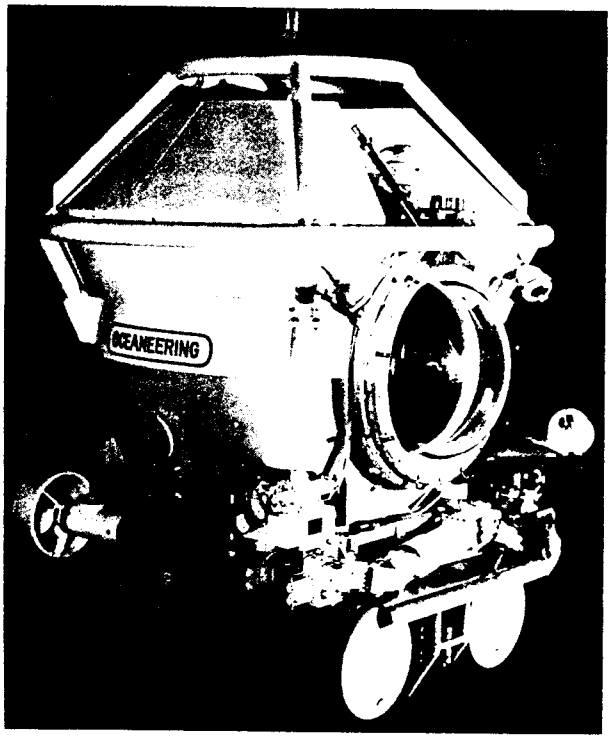


Fig. 72 Diving bell with syntactic foam float (upper portion)

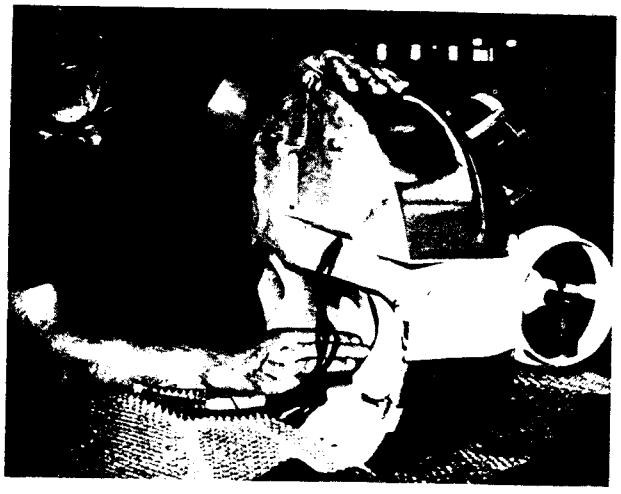


Fig. 73 Syntactic foam forms the hull of this RCV

4.3 Liquid Buoyancy Materials

The very deep-diving bathyscaphes *Trieste*, *Archimede*, and *FRNS-3* obtained positive buoyancy by the use of gasoline, which was carried in tanks located amidships. The gasoline tanks were fully compensated so that the pressure inside them was always equal to the pressure outside. As the submersibles descended, gasoline was vented off, reducing the overall buoyancy of the craft, and making them "heavier."

Gasoline is about twice as compressible as sea water, which, in these submersibles at least, makes it necessary to adjust buoyancy by dropping solid ballast, in addition to venting, during descent. Also, gasoline has a relatively high coefficient of thermal expansion, which can affect buoyancy calculations of the operator if the submersible should encounter thermal currents. It weighs 43.5 lb/ft^3 (68 percent of the weight of sea water). Because of its flammability, it does present a hazard in surface operations, such as loading or unloading the gasoline. However, it is relatively inexpensive and easily handled. The fact that it can be readily pumped out of the gasoline tanks on surfacing, thus lightening the vessel, aids the handling qualities of a submersible using it.

The amount of liquid buoyancy material required for a given design is a function of the weight of the vessel, the design depth, and the weight of other fixed and variable ballast and buoyancy materials used. As a point of comparison, *Trieste II* is loaded with up to 75 000 lb of gasoline at the beginning of a dive; its nominal submerged displacement is 220 tons.

In addition to gasoline, kerosene, aviation gasoline, and hexane have been used, or proposed for use, as buoyancy materials for submersibles. Their characteristics differ in some degree from those of gasoline, but not significantly.

4.4 Ballast Materials

In the design of submarines and submersibles, it is usually desirable, and frequently necessary, to use ballast materials. Ballast provides (1) the weight necessary to produce negative buoyancy (that is, to submerge the submarine), (2) stability, in some cases, and (3) a margin of weight so that at later periods in the designed life of the vessel additional equipment may be installed (necessitating the removal of a corresponding weight of the "marginal" ballast).

Ballast for submarines is classed as fixed or variable. Fixed ballast consists of material permanently or in the case of "margin," semipermanently designed and fabricated as part of the vessel. Variable ballast consists of weights taken on by the submersible, or discharged (or released) by it during the course of dives and other operations.

For submarines, the most common material used for variable ballast is sea water. Sea water is taken into the submarine's ballast tanks for a dive, and is either pumped out of these tanks or is blown out by compressed air when the submarine surfaces. Although the principle is simple, in essence, the systems required are relatively complex, and the pressures needed to blow tanks at deep depths (greater than 2000 ft) are high. In consequence, few submersibles are designed to use sea water ballast systems.

Many submersibles, particularly those intended for deep diving, have been designed to use droppable ballast as a means of ensuring required buoyancy. This ballast, which is classified as variable ballast, may also be considered to include weights (for example, a keel, or an external battery) which can be jettisoned if the submarine is in extreme peril, to provide extra positive buoyancy, but these are not considered ballast per se.

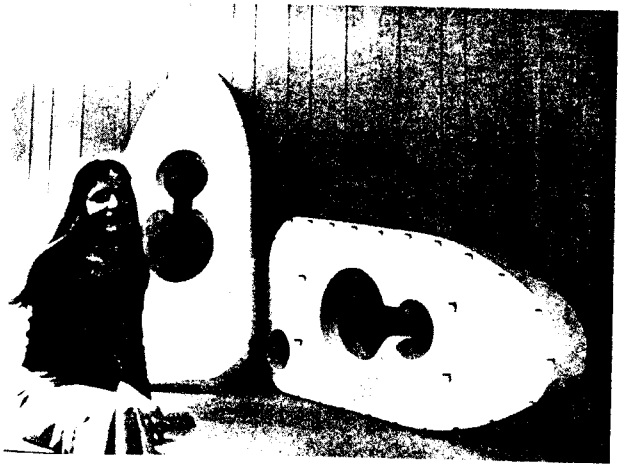


Fig. 74 Syntactic foam float containing wells for accessory equipment; stainless-steel bolts for attachment

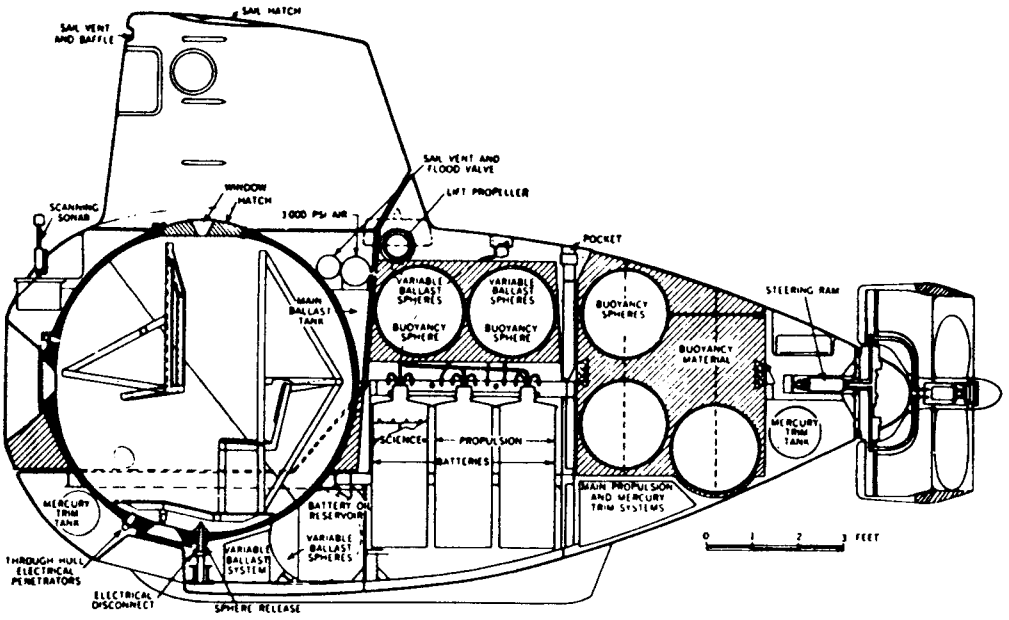


Fig. 75 Shaded areas indicate syntactic foam in *ALVIN*

The material most commonly used for fixed ballast in submersibles is lead. Lead has a high specific gravity (app. 11.34), and can easily be worked, cast into requisite shapes (usually, pigs), or if desired, melted, and poured into voids or irregular shape. It is also relatively inexpensive. Spent uranium, which has an even higher specific gravity (app. 19.3) has also been proposed as a ballast material, but it is not nearly as easily worked.

Lead is also used as droppable ballast on some submersibles. For this purpose, it is usually in the form of lead pellets, lead shot, or lead balls. Depending on the system design, droppable lead ballast is contained in recesses in the hull and dumped by opening a valve or aperture, or it may even be placed in bags attached to the exostructure and released when desired.

Iron and steel weights are also used as droppable ballast in some designs. One system uses an electromagnetic release mechanism to eject, successively, iron pellets, or balls. Another uses 4-lb steel plates, and another soft iron shot in 25-lb bags. Thousands of pounds may be jettisoned after a deep dive, which puts a constraint on preparation of the submarine for operations.

Still other designs simply use cast iron billets or shapes, which are attached to the submersible, and dropped as desired.

Cast iron has a specific gravity of about 7.2; steel has a specific gravity of about 7.8. Iron and steel are therefore not quite as dense as lead and, hence, not as effective for ballast. They are, however, inexpensive, readily available, and easily handled and for these reasons are attractive.

5. Other Material Considerations

5.1 Overview

The submersible operates in a hostile environment, at times under extreme pressure, and may for reasons of buoyancy or mission requirements have systems and components exterior to its pressure hull. Because of this, there are certain special requirements for materials in systems designed for use in it beyond the requirements for structural materials discussed in previous sections of this chapter. These include the effects of corrosion on systems exposed to sea water, in particular those systems which contain components fabricated from dissimilar materials, or materials dissimilar to the pressure hull material; means of avoiding or minimizing corrosion in components; prevention of low-stress fracture in components of submersible systems; lubrication of mechanisms exposed to sea water; maintenance of optical surfaces (for example, viewports) subject to fouling; and avoidance of the use of mercury.

5.2 Component Corrosion and Related Problems

Designers and materials specialists must be given a clear idea of a vehicle's intended mission profile, including the relative times the vehicle will be immersed and in air. Also, the overhaul/maintenance plan and field facilities for maintenance must be set forth. These factors may strongly influence the extent to which cheap, readily replaceable materials and components can be used. More likely than not, the

vehicle will be immersed for much less than half of its life and may never be immersed continuously for more than a few days. This does not necessarily simplify the problems of corrosion control, some of which become more severe because of this circumstance. For instance, it becomes necessary to protect the components against corrosion in air as well as in water (optimum procedures for the two environments may differ); evaporative salt concentration of residual sea water (in air) may cause a very aggressive environment to form in undrained crevices.

Most of the available writings on marine corrosion prevention refer to continuously immersed service, and the recommended practice for such service may have to be modified for submersible vehicles. For instance, paint protection may be used more extensively than in permanently immersed applications for the protection of carbon steel and aluminum alloy components which corrode readily in a damp marine atmosphere (whereas they might be left unpainted and cathodically protected for permanently immersed service). Painting will, of course, also be used to protect the more noble materials in mixed metal structures in accordance with the general advice to give priority to painting the cathodic member of a couple.

Engineers will generally be familiar with galvanic corrosion between dissimilar materials and the use of the galvanic series for sea water for determining the vulnerable member of a couple. However, there is one widespread misconception about the use of this information: it is commonly believed that the relative severity of damage to various couples is proportional to the difference in galvanic potential of the metals or alloys involved. This is only partially true. Among the common structural materials there are some examples where this idea would lead to erroneous conclusions. For instance, considering three bimetallic couples in which aluminum is connected to copper, passive stainless steels, and titanium, it might be thought that the galvanic corrosion rate of the aluminum would increase in that order, since the respective galvanic potential differences are typically 0.6, 0.8, and 0.9 V. Actually, the behavior follows the opposite trend: copper causes the most intense attack on aluminum and titanium the least intense. There are two useful, concise references on this subject: "Guidelines for Selection of Marine Materials" [5] and MIL-STD-889A, Dissimilar Metals.

The special case of galvanic corrosion between aluminum and copper calls for comment. This couple (in which the aluminum is attacked) is especially dangerous because the aluminum corrosion is strongly accelerated and also because this accelerated corrosion can be caused by the mere proximity of copper to aluminum in sea water, even though the two materials may be electrically isolated from each other. Aluminum and copper base alloys should not, therefore, be used in the same structure in sea water.

Galvanic interactions between dissimilar metals can be easily prevented in many other instances by electrical isolation. This may be effected by plastic spacers at attachment points. Such spacers have been used to isolate high-strength fasteners (usually stainless steel, titanium, or nickel-base alloys) from aluminum structural members. In some cases involving aluminum, advantage may be taken of the insulating properties of hard-coat aluminum anodize to ensure electrical isolation along a mating interface.

Cathodic protection can be used as supplementary corrosion protection, but generally it will not be practical to assign it a major role. The complex geometry and restricted flooded spaces between components makes it virtually impossible to get an adequate protection current density over the surfaces requiring protection. Also, there is the problem that the various alloys used will have different ideal protection requirements. However, sacrificial anode protection of particular components is often desirable. For this purpose, zinc ribbon-type anodes (with steel core wires) have been found useful. This type anode has features desirable for use in a submersible: it is flexible, so that it can be adapted to intricate geometrical situations; it occupies little space and is thus ideal for improvised use where cathodic protection is not "designed in" to equipment initially; and its large specific surface provides a high current capacity/unit weight.

Stainless steel is used extensively for submersible components, but it is important that its limitations be understood. Under unfavorable circumstances, the readily available grades of stainless steel can crevice-corrode severely after a few weeks in sea water. While some grades are recognized to be slightly more resistant to crevice corrosion than others (for instance, AISI Grade 316 is superior to Grade 304 and the precipitation-hardening grades), a substantial level of crevice-corrosion resistance is only obtained in some of the high chromium and molybdenum grades (such as the 24Ni-20Cr-6½Mo and 29Cr-4Mo grades). It should be noted that Grade 303 stainless steel has especially poor corrosion resistance and should not be used in sea water.

On the other hand, crevice-corrosion in these materials can be prevented by application of a corrosion-inhibiting grease at the faying surfaces of the crevice or by mild cathodic coupling (such as may be supplied by adjacent structural steel). Nevertheless, if stainless steel is used where its surface integrity must be maintained in a crevice-like situation (for example, seals, mechanisms), it will be advisable to inspect these areas frequently and to avoid uninterrupted development of stagnant crevice conditions over a long period. Periodic activation of a mechanism may accomplish this.

All faying surfaces (whether between similar or dissimilar metals) should be coated with a protective grease or paste such as that specified under MIL-C-

19459. This should include threads. In blind fasteners it is good practice to drill an axial hole the length of the fastener so that an excess of corrosion-inhibiting paste can be applied before torquing the fastener. This will ensure the exclusion of water along the threaded interface.

5.3 Low-Stress Fracture

Design and optimum materials selection for preventing low-stress fracture in components should start with appreciation of the fact that failure involves two general steps: (1) initiation of a flaw by some process which may or may not involve stress and (2) flaw growth under a tensile stress. Initial flaws may originate as manufacturing defects, hydrogen cracks, fatigue cracks, and corrosion pits. It is important to understand the likely causes of initial flaws in individual components in order to specify rational NDT procedures and inspection intervals.

Consideration of low-stress fracture will be confined to those components which experience applied or residual tensile stresses. Typically, the number of types of component of a submersible vehicle which experience appreciable tensile stress is small. The most critical component, the pressure hull, will not generally experience applied tensile stresses (local bending will occur, but this is usually negated by the superimposed compression). The pressure hull might contain residual tensile stresses from fabrication, and if the pressure hull is fabricated of a material susceptible to environmental cracking it will be desirable to stress-relieve it, so that cracking does not occur when the submersible is at, or near, the surface (when the tensile stress will be maximum).

There are two types of components in which high tensile stresses are likely to occur: hoisting components and fasteners. Low-stress fracture in these can be avoided by using low-strength materials which are immune to these failure modes, but this will often incur an unacceptable weight or space penalty.

Hoisting components will usually be accessible for frequent inspection so that (if fabricated of ultrahigh-strength materials) initial flaws can be detected by NDT before they reach the initial size. The NDT inspection criteria will be set by fracture-mechanics considerations and with a knowledge of the material's K_{Iscc} or K_{Ic} value. Candidate ultrahigh-strength materials would be MP-35N alloy and titanium-6-4 alloy (ELI grade). Generation of a defect by corrosion in either of these materials would be extremely unlikely. (It should be pointed out that initiation of a defect by fatigue is also very unlikely in typical submersible components since they only experience small numbers of load cycles.) Inconel 625 and Inconel 718 alloys are also suitable materials for this type of application but would have a lower strength/density ratio than the former two alloys. These two Inconel alloys have near immunity to both low-stress fracture and sea water corrosion. They are thus good choices for critical fasteners which are not easily inspected. At a much lower strength/density ratio level, commercially pure titanium fasteners are also a good choice, because of the material's immunity to sea water corrosion and low stress fracture.

5.4 Mechanisms Exposed to Seawater

On many mechanism components involving sliding contact which are exposed to relative water movement it will be impossible to retain a conventional lubricant, so that design must be based on the friction and wear characteristics of unlubricated surfaces. The materials selected will also need to have adequate sea water corrosion resistance, with the result that the number of suitable materials or combinations of materials may be small.

Available information on this topic is sketchy and often applies to very specialized problems. Based on experience and tests, two general statements can be made:

1. Coefficients of friction below 0.2 for seawater-lubricated conditions are achievable with only a few corrosion resistant material combinations and conditions. With normal loads less than 1000 psi, Waukesha-88 alloy versus an austenitic stainless steel will give a friction coefficient in this range. If the normal load is higher than this it may be necessary to utilize Stellite versus Stellite, or the nonmetallic DuPont Vespel (which can be used against a number of metals and alloys).
2. If normal loads are in the range of 5000 to 10 000 psi, coefficients of friction of 0.6 and higher are to be expected for couples involving most corrosion-resistant alloys (for example, Monel, stainless steel, nickel-aluminum bronze). Furthermore, the coefficient will increase with cycling and the surfaces will eventually gall.

The friction and wear characteristics of a couple will depend on initial surface finish and speed of relative motion as well as on normal load. The above specific observations were made on surfaces having a 63 rms finish (approximately) and with speeds up to 3 fps.

There are some further observations relevant to sea water–lubricated mechanisms:

- Ship propeller shafts are often run in rubber or lignum vitae sleeves, with copper-nickel journals. An ultra-high molecular weight polyethylene (Hercules 1900) is also said to perform well in such applications (that is, where a rigid material is not required).
- Seals in submarine periscopes and snorkel masts use Stellite versus a graphite-type rubbing surface.
- Solid-film lubricants may eliminate some frictional problems and simplify materials selection.

One problem which sometimes occurs in the baked-on Teflon film lubricants is that after cycling to great depth, the films may lift because of water penetration.

- The Armco "Nitronic" class of austenitic stainless steels has a much higher resistance to galling than other common stainless steels. One of these (22-13-5) also has better sea water corrosion resistance and higher strength than Grade 316 stainless steel.

5.5 Maintenance of Optical Surfaces

Exterior surfaces of viewport and camera port windows must be protected against loss of transparency which might be caused by fouling. This may become a problem if the submersible remains immersed for periods of several days. Fouling of these surfaces can be prevented by coating them with one of the commercially available clear transparent coatings which contain an anti-fouling addition.

Specular light reflectors which are exposed to sea water must be fabricated from materials having good reflectivity and corrosion resistance. Common choices for such reflectors are highly polished Grade 316 stainless steel and aluminum alloys. Typical reflectances for these materials are 55 percent and 80 percent, respectively. The latter value for aluminum alloy reflecting surfaces may be difficult to maintain in practice since it is typical of an unanodized surface (which is susceptible to mechanical and corrosion damage). The surface may be anodized to improve its durability, but this will reduce its reflectivity. Bright chromium and electrodeposited nickel surfaces also perform well, with reflectivities typically in the 60 percent to 65 percent range.

A specular reflectivity in the range 70 percent to 75 percent and good durability in sea water have been obtained with rhodium-plated reflectors. In this case, the basic reflector shape is electro-formed nickel over which ½ to 1 mil of dense gold is plated. A thin coating of bright rhodium (15×10^{-6} in. thick) is applied over the gold. The latter material is used to ensure that the underlying nickel is completely and reliably protected from contact with sea water (so that it will not suffer galvanic corrosion). Obviously, this type of reflector must be electrically isolated from surrounding structural materials because of the intensely cathodic nature of the noble metal coating.

5.6 Mercury

Because of its relatively high specific gravity (13.55), and the fact that it is liquid at temperatures ordinarily encountered within a submersible (it freezes at -38°C and boils at 359°C), mercury has been used as the active component of a trim system in some submersibles. In such systems, the mercury is pumped in requisite quantity to a position within a closed tube which will redistribute its weight and provide the desired trim for the vessel.

However, if a decision is made that a mercury trim system would in fact be useful as a means of providing proper trim, extreme care must be taken to ensure that the system is completely sealed and that no mercury, or mercury vapor, can leak into the living spaces of the submersible or into any of its systems or components.

Mercury and mercury vapor are toxic. Free mercury will give off minute quantities of mercury vapor, even at room temperature. Mercury vapor is harmful to people, particularly in a closed environment. Furthermore, mercury will form amalgams with other metals and can cause failure, particularly when the metal is subjected to high pressure, as, for example, tubing in a hydraulic system might be.

For these reasons, the use of mercury in any form has long been banned on military submarines. Submersible designers would be well advised to avoid its use as well.

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Chapter V
Hydromechanical Principles

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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

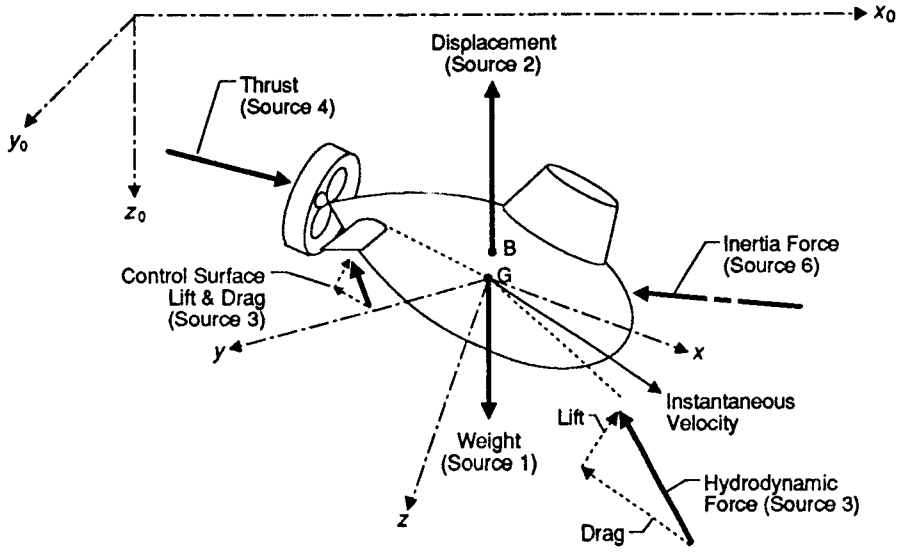
because they and related submersible characteristics differ substantially between the low- and high-speed regimes. For example, minimizing resistance is not critical for designs whose performance requirements include low forward or vertical speeds or both. Consequently, these designs are not obliged 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

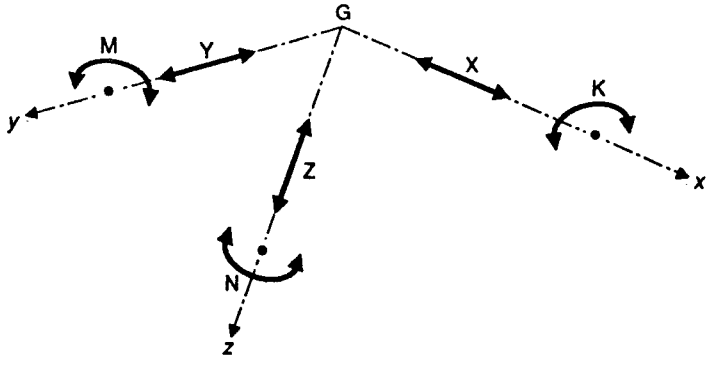
different masses and shapes represented by the submersible and its support ship. In most instances, little can be done by the submersible's designer to overcome the severe dynamic mismatch between these bodies, the burden for safe and rapid handling of the submersible falling essentially on the design 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.



FORCE-MOMENT RESULTANTS



X, Y, Z = Resultants of Static and Dynamic Forces along Body Axes *x, y, z*
K, M, N = Resultants of Static and Dynamic Force Moments about Body Axes *x, y, z*

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.

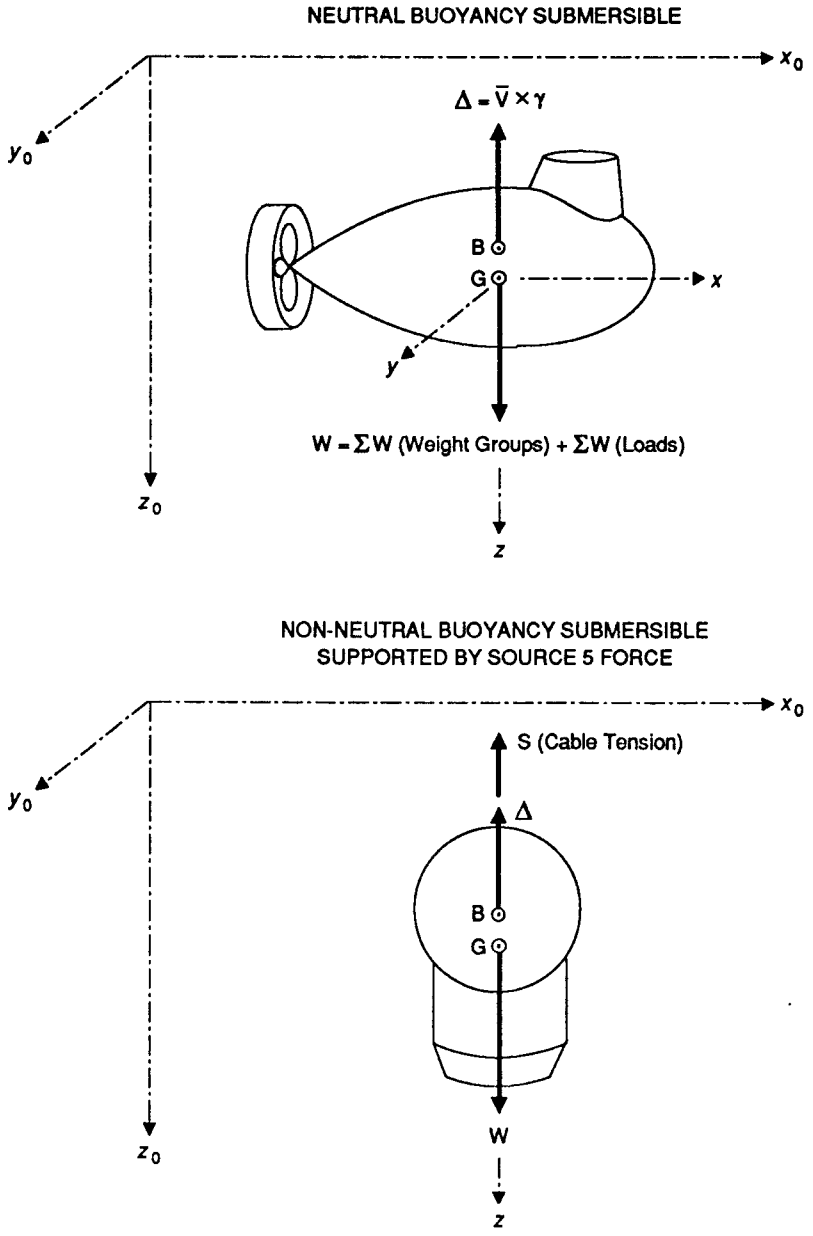


Fig. 2 Examples of static forces

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

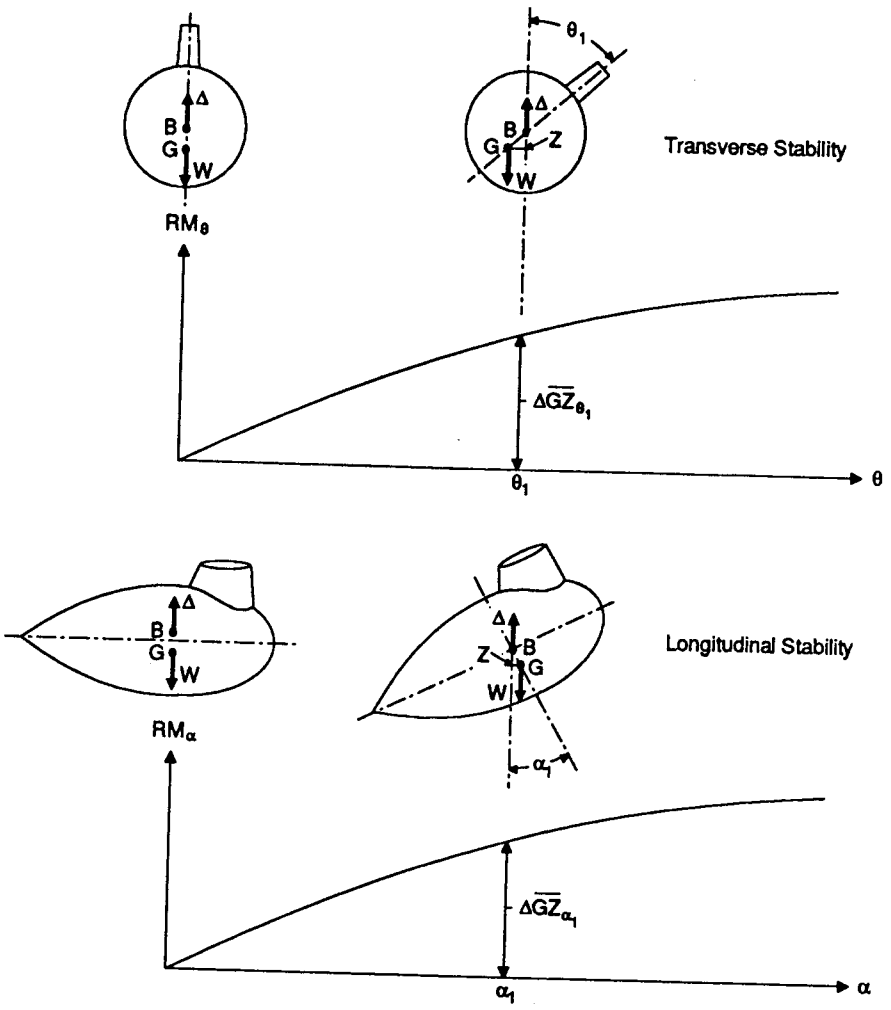


Fig. 3 Statical stability curves of righting moments

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

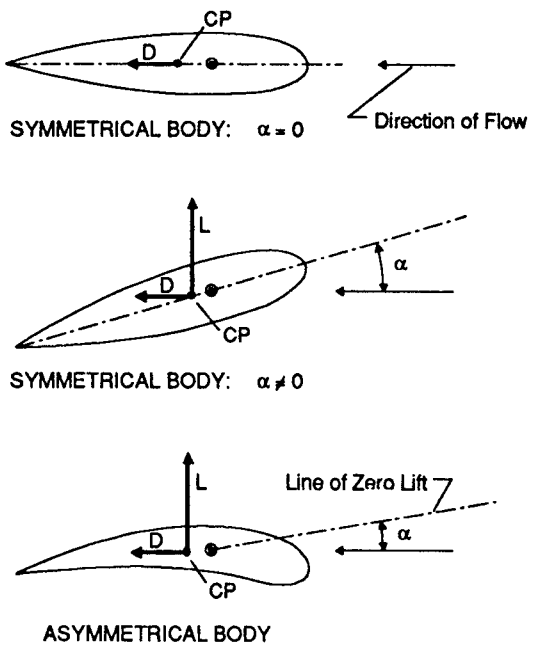


Fig. 4 Examples of lift and drag forces illustrated by all-movable control surfaces

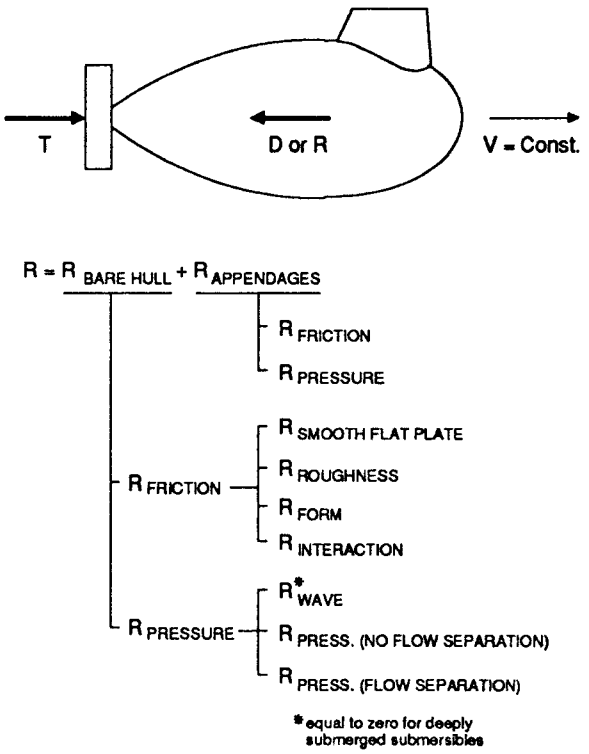


Fig. 5 The nature of resistance

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.

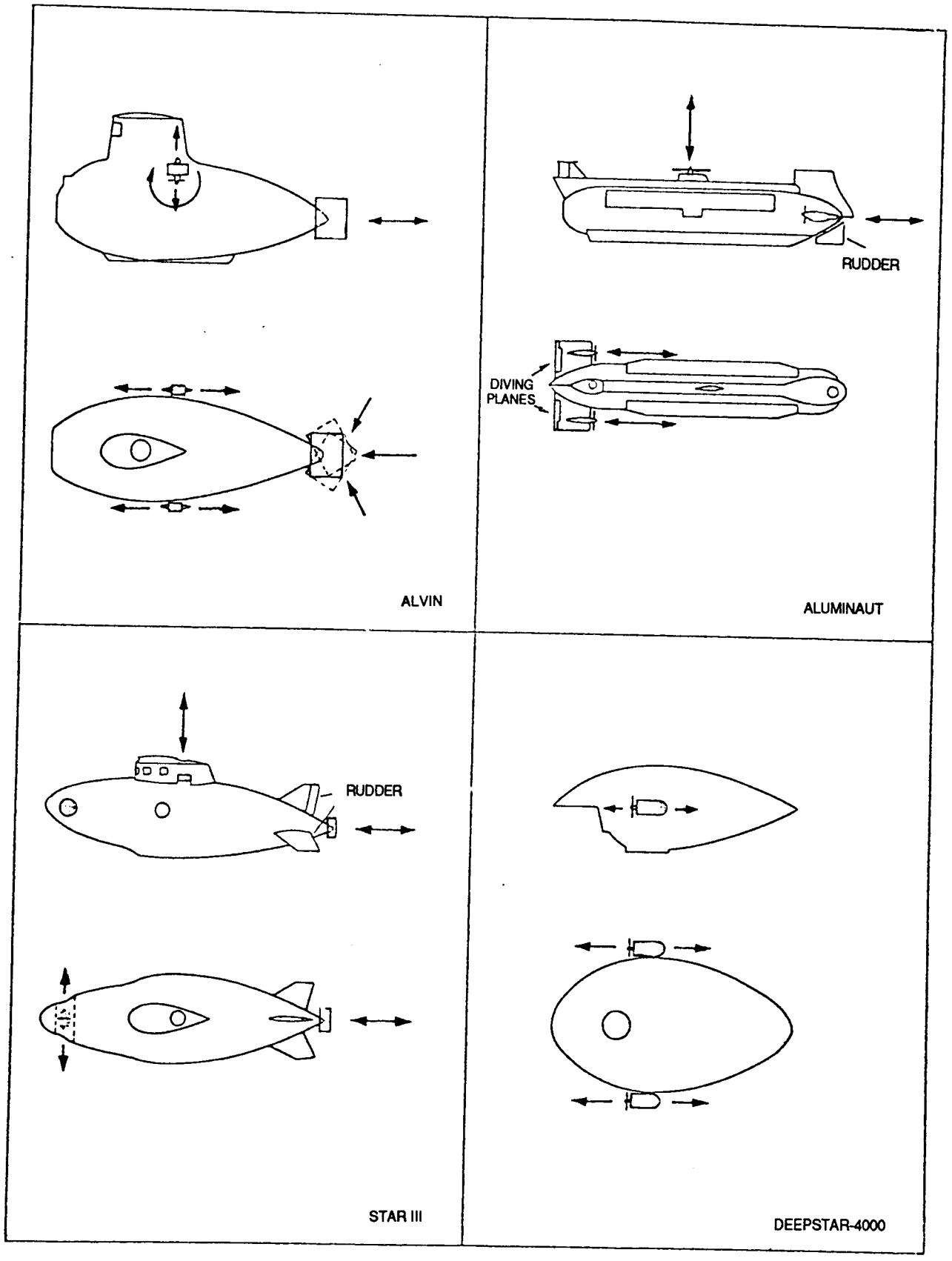


Fig. 6 Examples of types and arrangements of thrusters

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

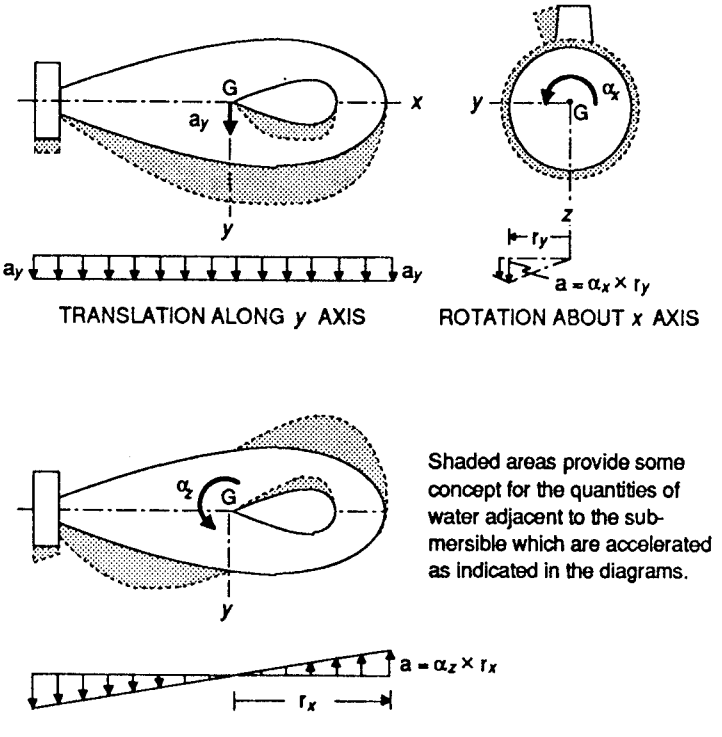


Fig. 7 Concept of added mass and mass moment of inertia

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

$$X = 0 \text{ or } \neq 0 \qquad K = 0 \text{ or } \neq 0$$
$$Y = 0 \text{ or } \neq 0 \qquad M = 0 \text{ or } \neq 0$$
$$Z = 0 \text{ or } \neq 0 \qquad N = 0 \text{ or } \neq 0 \qquad (2)$$

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 nonsteady-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

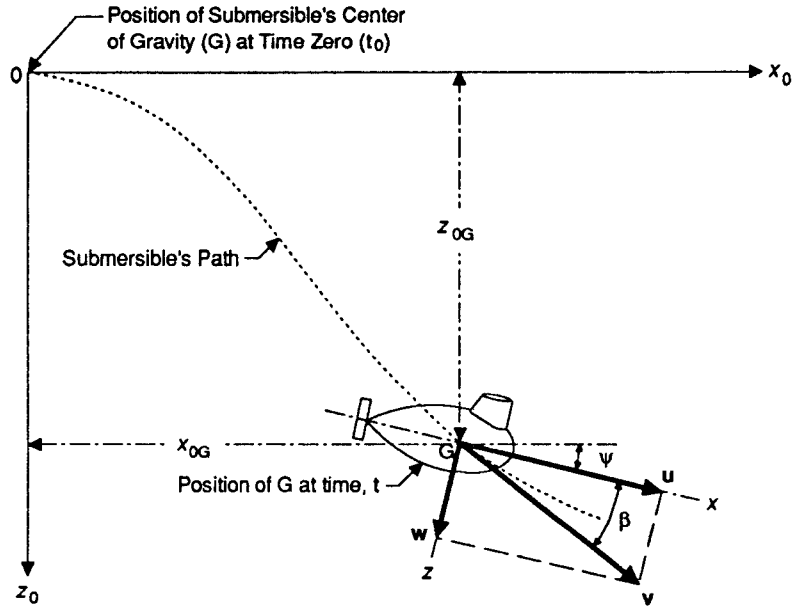


Fig. 8 Vertical plane motion of a submersible with reference to earth axes

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]).

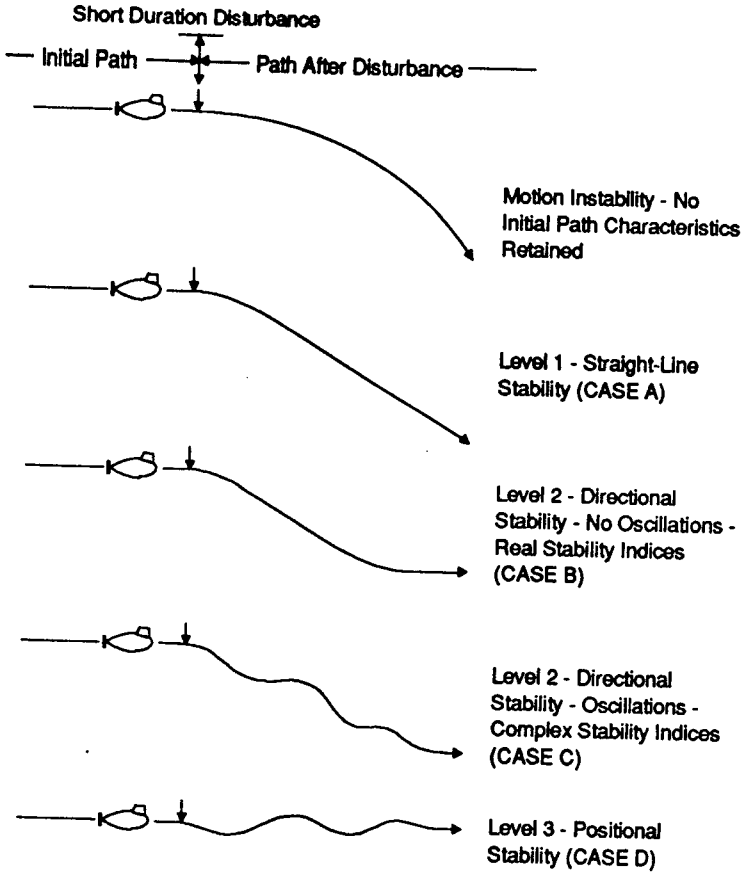


Fig. 9 Motion stability in the vertical plane

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

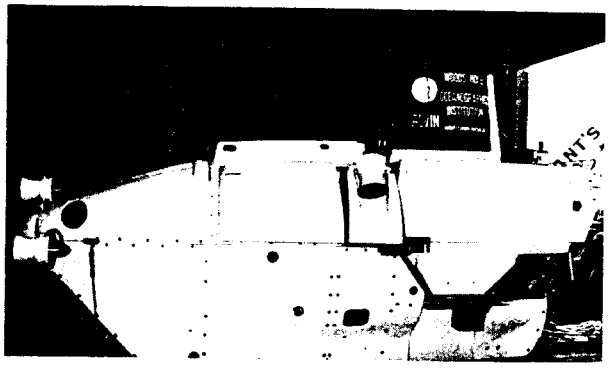


Fig. 10 Example of a neutral-buoyancy submersible. The manned, deep-submergence vehicle *Alvin* is the primary platform for conducting deep-sea science and is operated by the Woods Hole Oceanographic Institution. (Courtesy of WHOI)

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 nonzero 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.

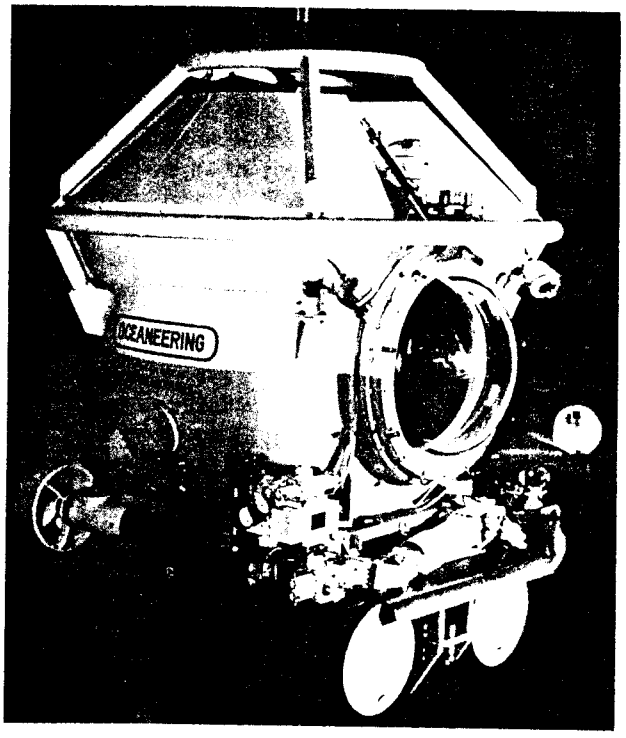


Fig. 11 Example of a statically-supported submersible. The manned Observation Manipulator Bell (OMB) was built by Perry Oceanographics and is used in support of undersea exploration and work missions. The bell can be made neutrally as well as negatively buoyant and is self-propelled. (Courtesy of Perry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

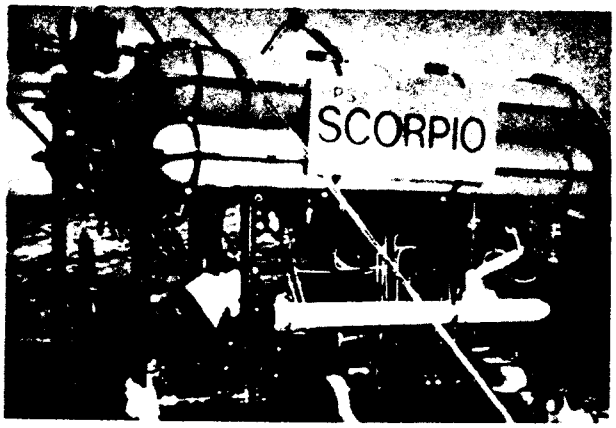


Fig. 12 Example of a thrust-supported submersible. The unmanned, remote-operated vehicle (ROV) *Scorpio* was built by Ametek-Straza for underwater work missions. It is controlled from the surface support ship through an umbilical cable to the vehicle. (Courtesy of Ametek-Straza)

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 (2*a*) merely states that the resultants of x and y components of hydrostatic pressure acting on the submersible are zero. The second force equation (2*a*) 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 (2*a*) 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.

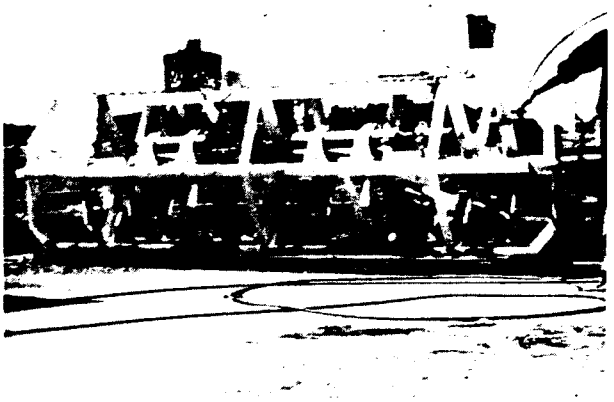


Fig. 13 Example of a lift-supported submersible. The unmanned, towed vehicle *Argo* was developed by the Woods Hole Oceanographic Institute for use in the visual and acoustical surveying of large areas of the seafloor. (Courtesy of WHOI)

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.

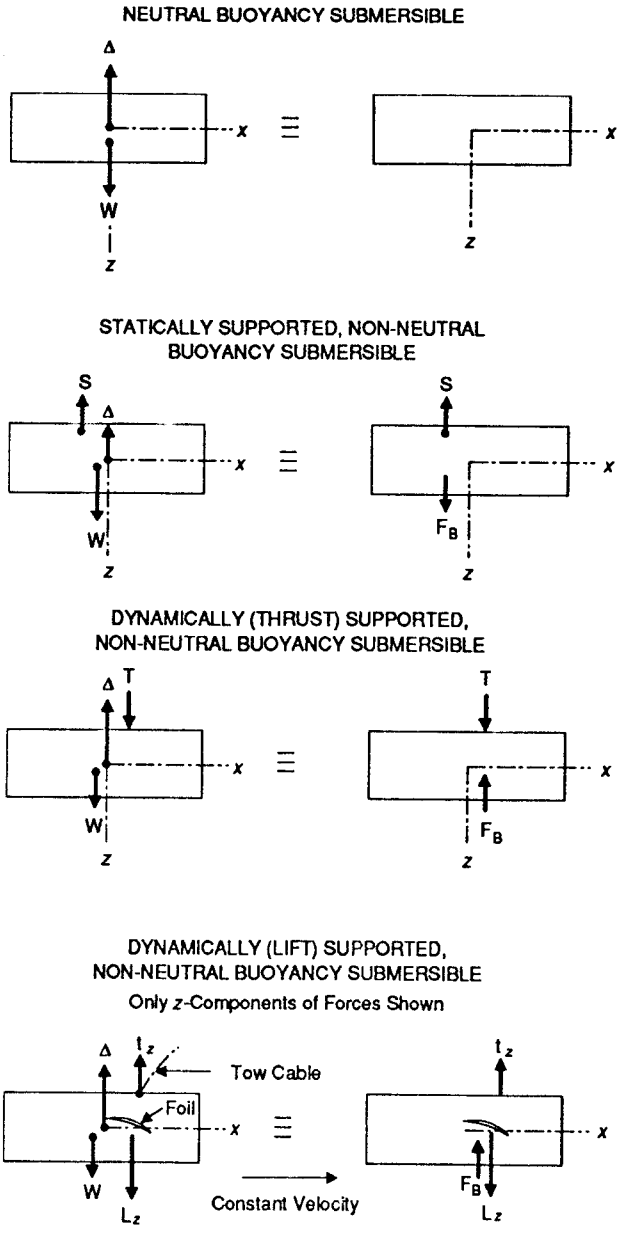


Fig. 14 Hydromechanical categories of submersibles based on design conditions

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 collinear 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

$$X = Y = 0$$

$$Z = W - \Delta \pm L_z \mp D_z - t_z = \mp F_B \pm L_z \mp D_z - t_z = 0$$

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

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 (2d) indicates that the sum of the horizontal components of all forces acting on the vehicle must be zero for dynamic equilibrium. The second equation (2d) 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.

(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

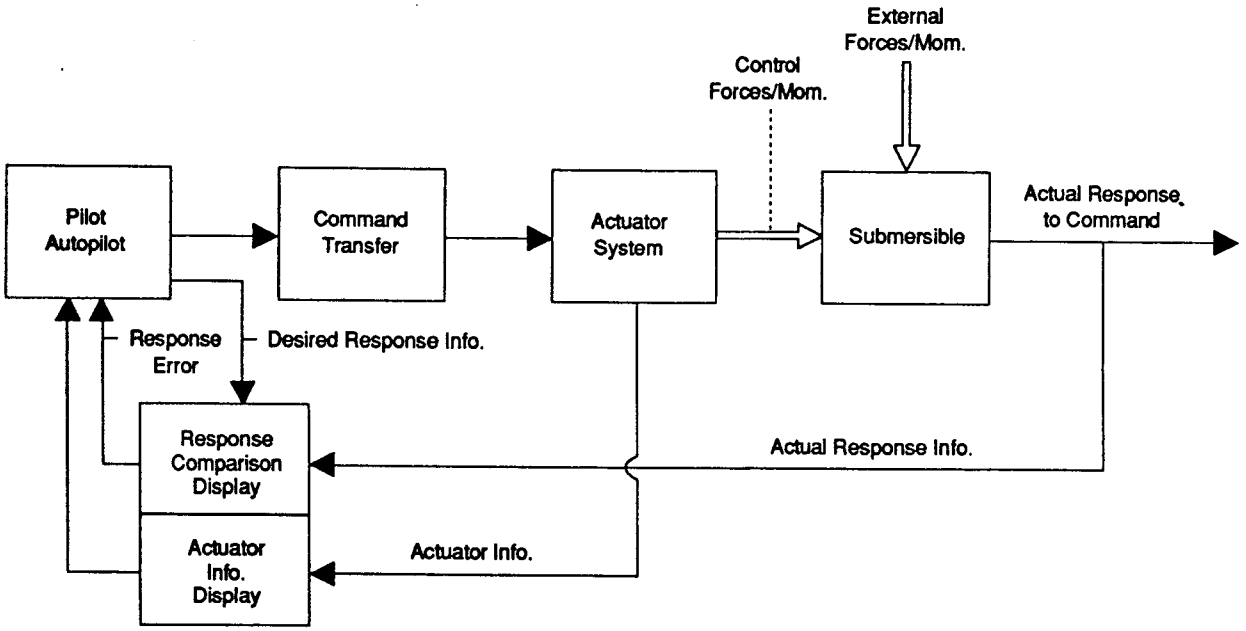


Fig. 15 Overall control system

b. Variable displacement system
c. Trim system
d. List system
e. Special compensating systems
f. Main ballast system
g. Descent weight system
h. Ascent weight system
(1) Normal
(2) Emergency

2. Dynamic control systems
a. Main propulsion system
b. Maneuvering system
(1) Thrust-producing systems
(2) Lift-producing systems

The functions of these control systems and underlying hydromechanical principles are discussed in Sections 3 and 4.

3. Hydromechanical Principles—Hydrostatics

3.1 Overview

The hydrostatic principles of the first two subjects of this section, *statical equilibrium* and *stability*, can be addressed by considering the submersible to be acted on by a simple, two-force system consisting of its total weight W and displacement Δ for the particular operating condition under study. The third subject, *dynamic stability*, requires the introduction of additional forces of waves, wind, and water resistance. Finally, the discussion of *weight-displacement-volume relationships*, *hydrostatic characteristics*, and *static control systems* necessitates breaking down the vehicle's total weight and displacement into their components and considering these components as separate forces interacting with one another so as to ensure the submersible's safety and effectiveness in all operating conditions. Thus, based on force systems acting, the text is arranged to proceed from simple to the more involved subjects.

3.2 Statical Equilibrium and Stability Fundamentals

Equations (2) express the first of two statical[12] equilibrium fundamentals which must be clearly understood: the resultants of all forces and moments acting on a submersible must be zero for equilibrium to exist. The second fundamental concerns the type of equilibrium—*stable*, *unstable*, or *neutral*.

Figure 16 shows ball bearings positioned on a curved track in the vertical plane at Points A, B, and C. At all positions the bearings are at rest and, hence, in states of equilibrium. But of what type are they? Suppose small, external forces are applied to the bearings just long enough to nudge them slightly away from their equilibrium positions and then removed, leaving only the forces of weight and track reaction acting. Non-zero moments now exist for the bearings initially at A and B, while a zero moment is maintained for the bearing at C. The moment acting on the bearing initially at A causes it to return to A, and, consequently, its initial state is described as one of *stable equilibrium*. The moment acting on the bearing initially at B causes it to depart further from B and to seek a new position of stable equilibrium. In this case, its initial state at B is defined as one of *unstable equilibrium*. No moment will act on the bearing initially at C, and, neglecting inertia forces, it will simply come to rest at some new position on the horizontal portion of the track. Therefore, its initial state at C is termed *neutral equilibrium*.

Now, note the diagrams directly below the ball bearings' positions A, B, and C. Shown are the transverse and longitudinal views of neutral-buoyancy submersibles in the design condition. As is the case for the bearings, the upper diagrams of both views show the submersibles both at rest and acted on by the basic force system of this category—an equal and opposite, collinear, two-force system. Again, let momentary external forces be applied so as to incline slightly the submersibles transversely and longitudinally as illustrated in the lower diagrams of these views. Upon inclination, the collinearity of W and Δ is destroyed for submersibles A and B while it is maintained for C. It is now possible to relate types of equilibrium and stability for the three situations.

Note that G[13] is below B for submersible A. In both transverse and longitudinal directions, the weight-displacement couple acts to return the submersible to its initial position, one of *stable equilibrium*. The restoring couple is called a *positive righting moment* ($+RM$), which is a measure of the *positive stability* existing at that small angle of inclination. This stability may also be measured by the positive righting arm ($+RA$), where RA is the normal distance between the parallel lines-of-action of W and Δ.

Note that G is above B for submersible B. In both views, the couple now acts to further incline the submersible away from its initial position, one of *unstable equilibrium*. The inclining couple is called a *negative righting moment* ($-RM$), which is a measure of the *negative stability* existing at that small angle of inclination. Again, this stability may also be measured by the negative righting arm ($-RA$).

Finally, note that G and B are coincident points in submersible C. No couples are created, $RM = 0$, with inclinations, and the submersible neither returns to

[12]The adjective "statical" will be omitted from the terms "statical equilibrium" and "statical stability" unless its usage is required for clarity.

[13]In this discussion, it is assumed that no effects of "free surface" or "suspended weights" are present. Their presence introduces the concept of "virtual center of gravity" (G_V), which is discussed in the next article.

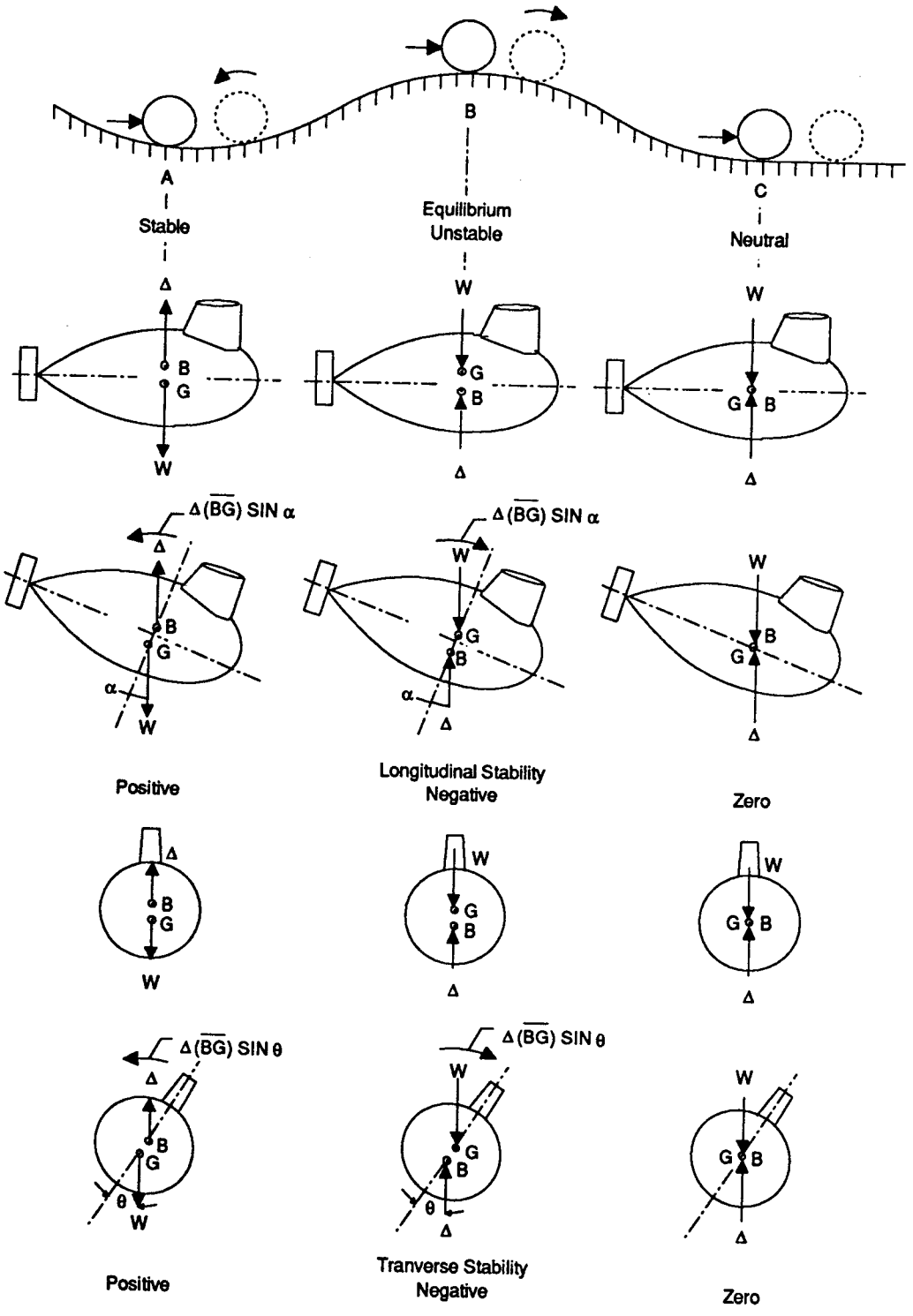


Fig. 16 Equilibrium and stability

nor departs further from its initial position, one of *neutral equilibrium.*

The impact of the equilibrium-stability relations on design can be fully appreciated only when stability curves of righting moment, or righting arm, versus angle of inclination are drawn. Figure 17 shows these curves for submersibles A, B, and C, in which inclinations extend over a theoretical range of 360 deg. It is assumed that *G*, as well as *B*, remains at a fixed location throughout this range of inclinations—an

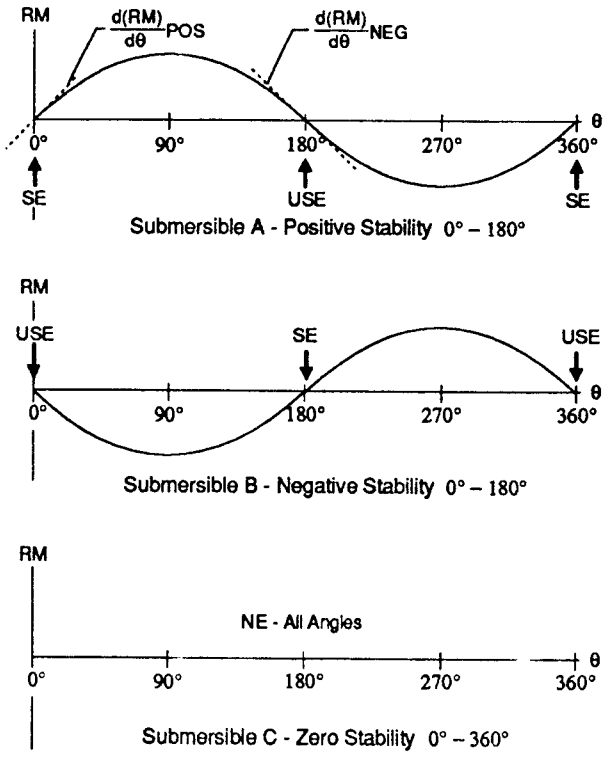


Fig. 17 Stability curves for submersibles A, B, and C. SE, USE, and NE note angles of stable, unstable, and neutral equilibrium, respectively

assumption, of course, which is valid only for this theoretical study. The equation for all three curves is

$$\pm RM = \Delta(\pm \overline{BG}) \sin \beta \qquad (3)$$

where

$\overline{BG}$ = metacentric height for submerged conditions,

β = either the heel angle θ or the trim angle α, and

$\overline{BG} \sin \beta$ = righting arm, RA.

From these curves it is seen that submersibles A and B have equilibrium positions at 0 or 360 and 180 deg while submersible C has an equilibrium position at any angle. Submersible A has stable equilibrium at 0 or 360 deg and unstable equilibrium at 180 deg. Note that as it is inclined beyond 180 deg, negative righting moments are developed, causing it to depart further from the 180-deg position of unstable equilibrium to seek the next position of stable equilibrium at 360 or 0 deg. Therefore, submersible A will "insist" on returning to its upright position when inclined to any angle once the inclining influence is removed. Conversely, submersible B has unstable equilibrium at 0 or 360 deg and stable equilibrium at 180 deg. Again, note that as it is inclined beyond 180 deg, positive righting moments develop, causing it to return to the 180-deg position of stable equilibrium. Consequently, submersible B will "insist" on floating in a "keel-up" position once it is inclined slightly away from the upright position. Submersible C has neutral equilibrium at any angle of inclination from 0 to 360 deg and, hence, zero righting moment through this range. When inclined away from the upright position, it will simply "lay over" at the position attained when the inclining influence is removed.

The type of equilibrium existing at any angular position of equilibrium is defined by the slope of the stability curve at that angular position where RM equals zero, either in the upright position of θ and α equal to zero or at other values of these angles. Mathematical expressions associated with these types of equilibrium are

$$\left.\frac{d(RM)}{d\beta}\right|_{\beta \text{ at } RM=0} \quad \text{positive—stable equilibrium}$$

$$\left.\frac{d(RM)}{d\beta}\right|_{\beta \text{ at } RM=0} \quad \text{negative—unstable equilibrium}$$

$$\left.\frac{d(RM)}{d\beta}\right|_{\beta \text{ at } RM=0} \quad \text{zero—neutral equilibrium} \qquad (4)$$

β is either θ or α in these expressions.

It is evident from the foregoing discussion that design condition criteria for a neutral-buoyancy submersible must not only provide for equilibrium but for stable equilibrium. A summary of design condition criteria, then, includes:

1. Neutral-buoyancy: $W = \Delta$
2. Zero trim and list (upright position): W and Δ are collinear forces with their common line of action normal to submersible's x-y plane
3. Stable equilibrium: G below B

Criteria 1 and 2, of course, may change for other operating conditions. Criterion 3, however, applies to all operating conditions although stable equilibrium may be attained with G above B for surface conditions, which is a subject discussed later. Moreover, Criterion 3 applies to all conditions of all design condition categories of submersibles insofar as the requirement for stable equilibrium is concerned.

3.3 Statical Stability Details

The subject of stability for most marine vehicles is considered from two perspectives: *intact stability* and *damaged stability*, with the hull maintaining its complete watertight integrity in the first instance and not in the second. Only intact stability considerations are of importance in submersible design since small vehicles cannot withstand any damage due to flooding. The details of intact stability differ considerably between surface, transition, and submerged operating conditions and, consequently, must be studied with reference to each of them. Further, within these three studies, transverse and longitudinal stability must be discussed with reference to transverse inclinations, about the waterplane's x-axis or the body

x-axis, and longitudinal inclinations about the waterplane's y-axis or the body y-axis.

Several factors, if present, influence stability, including swinging weights, free-surface, Source 5 external forces, and weight changes. In the first three instances, it will be necessary to introduce the concept of virtual center of gravity Gv in a discussion of these influences.

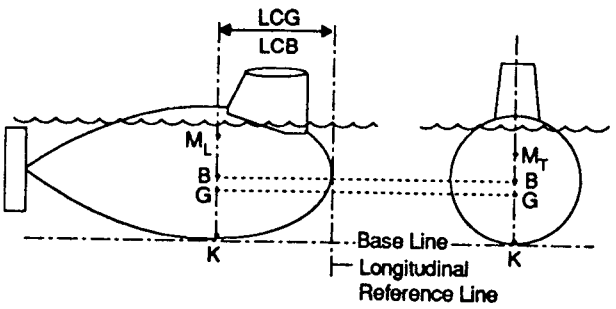
The presentation of principles associated with intact stability will be facilitated if the reader has stability dimension definitions clearly in mind. This subject introduces stability details.

3.3.1 *Stability Dimensions*

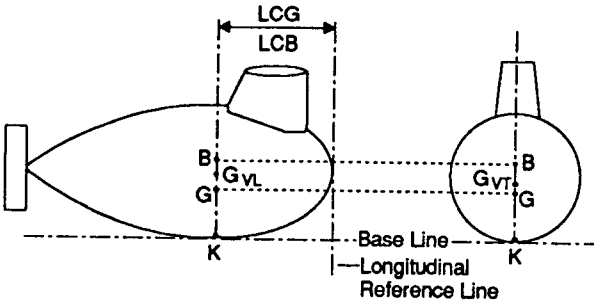
These dimensions locate six points of interest not only to stability but to trim and list calculations as well: the aforementioned center of gravity G and center of displacement B and also the transverse and longitudinal metacenters M_T and M_L. In the presence of certain influences, such as free surface, two additional points must be considered, the transverse and longitudinal *virtual centers of gravity*, G_{VT} and G_{VL}. As previously noted, the resultants of all gravitational and displacement forces act on a submersible at G and B. M_T and M_L are the instantaneous centers of curvature of the paths along which B moves as the surfaced vehicle is inclined in the transverse and longitudinal sense. When the vehicle is submerged, B does not move with inclinations and B, M_T, and M_L become coincident points. Finally, G_{VT} and G_{VL} are the points at which the resultant of all gravitational forces *appear to act* as far as transverse and longitudinal stability studies are concerned.

Stability dimensions, including those pertinent to trim, are measured vertically from the base line or bottom of the keel; K denotes a reference point on these lines. Longitudinally, they are measured from a transverse plane located at the bow, amidships, or at some other convenient point along the length of the vehicle. There are no stability dimensions in the transverse direction because all pertinent points lie in the same vertical plane, usually the vehicle's centerline vertical plane due to symmetry. Stability dimensions are shown in Fig. 18 and may be listed and defined as:

1. $\overline{KG}$ and $\overline{LCG}$: locations of vertical and longitudinal centers of gravity with respect to baseline and transverse plane references
2. $\overline{KB}$ and $\overline{LCB}$: locations of vertical and longitudinal centers of displacement with respect to baseline and transverse plane references
3. $\overline{BM_T}$ and $\overline{BM_L}$: transverse and longitudinal metacentric radii for surface conditions
4. $\overline{KM_T}$ and $\overline{KM_L}$: transverse and longitudinal heights of metacenter above baseline for surface conditions
5. $\overline{GM_T}$ and $\overline{GM_L}$: uncorrected transverse and longitudinal metacentric heights for surface conditions



SURFACE - DIVING TRIM CONDITION
No Free Surface Effects Assumed



SUBMERGED - DESIGN CONDITION
Free Surface Effects Assumed Present
Longitudinal Effects Assumed Greater Than Transverse Effects

Fig. 18 Stability dimensions

6. $\overline{BG}$: uncorrected metacentric height for submerged conditions
7. $\overline{GG_{VT}}$ and $\overline{GG_{VL}}$: transverse and longitudinal virtual rise of the center of gravity, corrections to be made to $\overline{KG}$, $\overline{GM_T}$, $\overline{GM_L}$, and $\overline{BG}$ if influences causing this rise are present

Criteria 2 and 3 for the design condition can now be stated in terms of these dimensions as

2. Zero trim and list: $\overline{LCG} = \overline{LCB}$, with G and B being in the same vertical plane
3. Stable equilibrium: $\overline{KG} < \overline{KB}$ or $\overline{KG_V} < \overline{KB}$, resulting in positive $\overline{BG}$ or $\overline{BG_V}$

3.3.2 *Surface Stability*

Hydrostatic principles underlying surface stability computations are the same for all free-floating marine vehicles having water-excluding volumes projecting above the waterline. However, as will be seen, the application of these principles to submersible and surface-vehicle stability yield different results due to shape, freeboard, and vertical center of gravity considerations. Generally speaking, as contrasted with surface vehicles of comparable displacements, submersibles have very poor shapes and excessively small waterplane areas and freeboards from the stability point of view. The panacea for these poor stability characteristics is relatively low vertical

centers of gravity as compared with these surface vehicles.

Shapes of submersibles vary widely from bodies of revolution to wall-sided prisms whose cross sections approach squares or rectangles. Consequently, a cylinder and a rectangular prism are chosen as shapes in the following discussion, the rectangular prism also being used to illustrate transverse stability characteristics of surface vehicles. Figure 19*a* shows three views of the cylinder: the transverse section containing *G*, *B*, and M_T; the longitudinal centerline plane containing *G*, *B*, and M_L; and the waterplane containing the *x* and *y* waterplane axes,[14] which intersect at the area's centroid called the *center of flotation CF*. Shown also is a transverse section through the prism containing *G*, *B*, and M_T. As illustrated in the figure, transverse and longitudinal inclinations cause "wedges" of displacement to be transferred from the high to the low side or ends. These emerging and immersed wedges must be of equal volume, of course, for the volume of displacement to remain constant. Wedge transference redistributes the volume of displacement, causing its center, *B*, to move away from its initial position along paths in the transverse and longitudinal planes as

[14]These waterplane axes should not be confused with the *x* and *y* body axes.

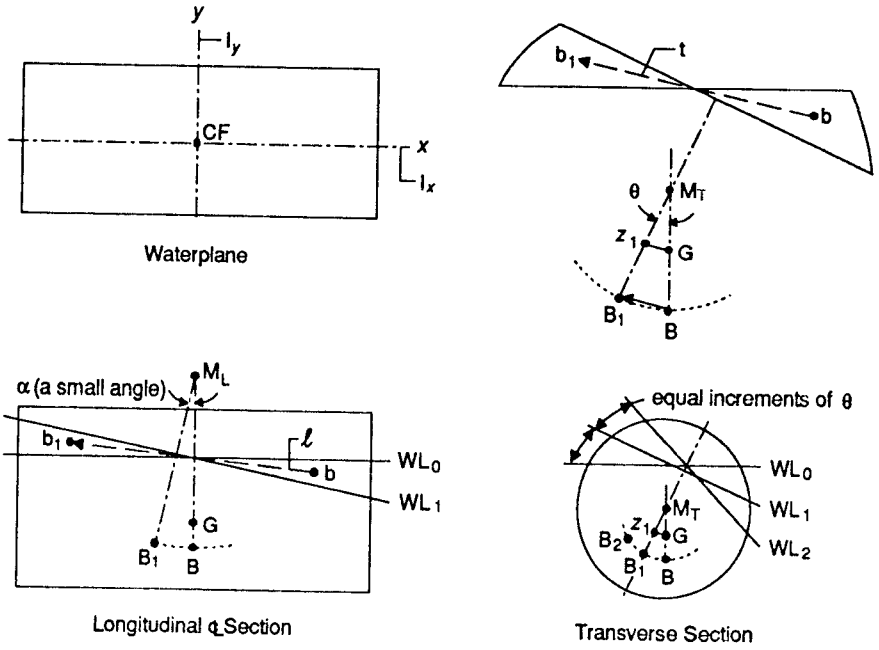


Fig. 19(*a*) Surface Stability

indicated. The transverse shift of B is given by the equation

$$\overline{BB}_1 = \frac{V_T t}{\overline{V}} \tag{5}$$

and the longitudinal shift by

$$\overline{BB}_1 = \frac{V_L l}{\overline{V}} \tag{5a}$$

In these equations, V_T and V_L are the volumes of the transverse and longitudinal wedges being shifted distances of t and l, while $\overline{V}$ is the volume of displacement. The products $V_T t$ and $V_L l$ may be called transverse and longitudinal *moments of transference.*

Transverse stability involves transverse sections of the cylinder and prism in Fig. 19*a* as well as the cylinder's waterplane. For the cylinder, it is obvious that equal increments of transverse inclination generate equal transverse moments of transference. These moments, in turn, cause equal $\overline{BB}_1$ shifts, which result in B moving along a circular path, a fact also evident from the observation that the volume of displacement and its shape remain constant for all inclinations. Thus, the radius of curvature $\overline{BM}_T$ remains constant and the center of curvature M_T is a fixed point at the section's center. In general, from reference [1], $\overline{BM}_T$ for very small inclinations of any floating body is given by the expression

$$\overline{BM}_T = \frac{I_x}{\overline{V}} \tag{5b}$$

where I_x is the second moment of area of the waterplane about its x-axis. Again, in this instance, it is clear that $\overline{BM}_T$ must remain constant since the waterplane's area and shape, I_x and $\overline{V}$, also remain constant for all inclinations. Transverse stability of the cylinder, then, is a function of the transverse metacentric height $\overline{GM}_T$ for all inclinations being expressed by any one of the equations

$$RM = \Delta \overline{GM}_T \sin\theta$$

$$RA = \overline{GZ} = \overline{GM}_T \sin\theta$$

$$RA = (\overline{KB} + \overline{BM}_T - \overline{KG}) \sin\theta \tag{5c}$$

The situation is different for the rectangular prism. Equal increments of inclination result in approximately equal moments of transference for small inclinations only—from 0 to about 8 deg for the usual mono-hull shape. Consequently, as indicated in Fig. 19*a*, the path of B can be considered circular and M_T a fixed point in this range only. Equal increments of inclination at larger angles result in non-equal moments of transference; these moments increase until the corner of the rectangle, or deck edge, is immersed, after which they begin to decrease.

These increasing/decreasing moments result in B moving along a noncircular path. The instantaneous center of curvature for a differential element of path length, now called the pro-metacenter, moves away from the point M_T along a path which is the evolute curve of the path of B. In summary, then, transverse stability for shapes other than volumes of revolution is divided into two parts:

1. Initial stability: from zero to small values of θ for which the path of B is approximately circular, with the center of curvature M_T considered a fixed point on the vehicle's centerline.

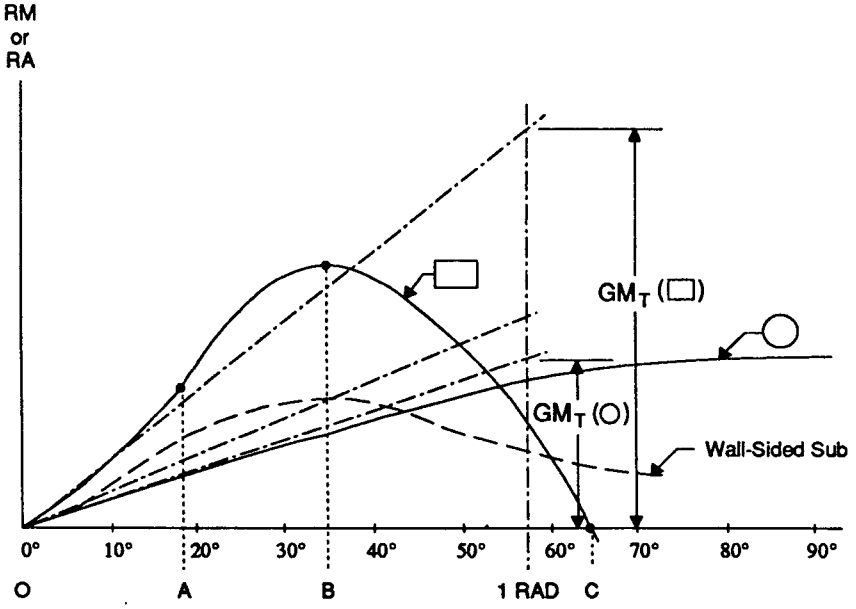


Fig 19(*b*) Transverse stability curves for cylinder and rectangular prism

2. Overall stability: for larger values of θ for which the path of B is noncircular, with the instantaneous center of curvature, the prometacenter, being a moving point.

Initial stability is a function of $\overline{GM}_T$, with equations (5c) being applicable. Overall stability is determined by other means, which usually involve cross-curves of stability, discussed in reference [1].

Figure 19*b* shows transverse stability curves for the cylinder and rectangular prism. Slopes at the origin for both curves is given by the expression

$$\left.\frac{d(RA)}{d\theta}\right|_{\theta=0} = \frac{\overline{GM}_T}{1 \text{ radian}} \qquad (6)$$

If the proportions of these shapes approach those of similar displacement submersible and surface vehicle, the $\overline{GM}_T$ of the cylinder will be considerably smaller than that of the rectangular prism primarily because of smaller water-plane areas and I_x. Consequently, the slope of the cylinder's curve at the origin is the smaller of the two. Its curve departs from the slope-line, constructed as indicated in the figure, and continues as a sine curve where the *maximum righting moment or arm* (*MRM* or *MRA*) develops at 90 deg and the *range of positive stability* is from 0 to 180 deg. As is characteristic of most mono-hull surface-vehicle shapes, the prism's curve departs from the slope-line as indicated with a point of inflection occurring at the Angle A of corner, or deck edge, immersion. It then continues to rise until Angle B is reached, which is the angle of *MRM* or *MRA*, where the values of B and *MRM* or *MRA* are critical to determining the amount of righting energy or dynamic stability developed. The curve then declines crossing the θ-axis at Angle C, which marks the beginning of negative stability development. Angle C is known as the *angle of vanishing stability* and *OC* is the range of positive stability. The actual shape of the curve for a particular surface vehicle depends on factors beyond the scope of this text (see reference [1] for further details).

Water-excluding items of a submersible involved in generating surface stability primarily include the pressure hull, tanks and containers external to this hull, and fixed buoyancy to the extent used in a design. If the assemblage of these items approaches a volume of revolution, as it does for most modern submarines in the interest of obtaining high submerged speeds, the transverse stability curve approaches the sine curve of Fig. 19*b*. If, however, this assemblage approaches a rectangular prism due to the straight or wall-sided shapes of items external to the pressure hull, the curve (indicated by the dashed line in the figure) will exhibit some of the characteristics of the curve for a similar-sized surface vehicle. Large righting moments or arms will not be developed due to its relatively narrow beam and low freeboard, but its range may be expected to be greater because of its comparatively low vertical center of gravity. Note that its $\overline{GM}_T$ may be expected to be greater than for a similar-sized, body-of-revolution submersible because its wall-sides generate a larger value of I_x. The transverse stability of all submersibles whose shapes depart from bodies of revolution should be investigated on both initial and overall bases.

It is evident from Fig. 19*a* that stable equilibrium for surface conditions can be obtained with G above B but below M_T or that

$$+\overline{GM}_T = \overline{KB} + \overline{BM}_T - \overline{KG}$$

Submarines and large submersibles have water-plane areas and beams which produce second moments of area I_x yielding sufficient $\overline{BM}_T$, and $\overline{KG}$ may be greater than $\overline{KB}$ with safety. However, smaller submersibles have relatively small waterplane areas and beams with commensurately low values of $\overline{BM}_T$ making it prudent, as indicated in Fig. 18, to have $\overline{KG}$ less than $\overline{KB}$. $\overline{GM}_T$ for most submersibles is in the order of a few centimeters or inches.

Longitudinal stability involves inclinations about the waterplane's y-axis. Again, as indicated in Fig. 19*a*, the transference of wedges of displacement from the high end of the cylinder to the low end of the cylinder causes B to move along a path in the longitudinal centerline plane. The longitudinal moment of transference $V_L l$ is usually much greater than the transverse moment of transference $V_T t$ for the same angle of inclination, resulting in a "flatter" curve with a greater radius of curvature $\overline{BM}_L$. For small inclinations, this radius is given by the expression

$$\overline{BM}_L = \frac{I_y}{\overline{V}} \qquad (7)$$

which parallels equation (5*b*). From Fig. 19*a*, it is evident that I_y is much greater than I_x because the waterplane's length is greater than its breadth, which results in a $\overline{BM}_L$ greater than $\overline{BM}_T$. The same relationship holds for submersibles and surface vehicles. Usually, only initial longitudinal stability is determined for intact vehicles, whereas both initial and overall transverse stability must be investigated. However, for small submersibles with low length/beam ratios, the differences between I_y and I_x and between $\overline{BM}_L$ and $\overline{BM}_T$ may be quite small. In these instances, it is best to investigate overall, as well as initial, longitudinal stability. Stability in this direction may be the more critical of the two if longitudinal free-surface corrections are greater than those in the transverse direction.

$\overline{BM}_L$ is used in formulating an expression called *moment to change trim one inch* (*MTI*), in which

$$MTI = \frac{W\overline{GM}_L}{12L} \doteq \frac{W\overline{BM}_L}{12L} \qquad (8)$$

The small difference in $\overline{KG}$ and $\overline{KB}$ make this approximation feasible. *MTI* is useful in obtaining change in trim δT for a given trimming moment *TM*, or the reverse.

3.3.3 *Transition Stability*

Consider an upright submersible passing from the surface to submerged condition or vice versa. As it does so, Points B, M_T, M_L, G, and G_V will move from their surface to submerged locations or vice versa. A study of their movements may be made by plotting their respective heights above the baseline for surface, submerged, and several intermediate displacements. The purpose of this study is to determine the minimum actual or virtual metacentric height developed during the transitional period for either the transverse or longitudinal direction, whichever one is critical. The transition period is usually longer for surfacing than for diving, thereby allowing more time for surface wave-wind heeling forces to act and thus making this operating condition the more critical of the two.

Actual paths followed by the various points in moving from their surface to submerged locations, or vice versa, depend on a number of variables, including shape and volume of above and below waterline structure, weight and its vertical distribution, and volume, shape, and arrangement of the main ballast tanks. Figure 20 illustrates a study of transverse transitional stability, showing paths of B, M_T, G, and G_V for an arbitrary set of variables. For diving, note that the B and M_T curves draw closer together as the waterplane area I_x and $I_x/\overline{V}$ decrease until, finally, this area becomes zero, B and M_T becoming coincident points in the submerged condition. Also, note the presence of $\overline{GG}_{VT}$, which is assumed to be due entirely to the unavoidable action of free-surface water in the main ballast tanks. In this case, $\overline{GG}_{VT}$ is called the *transverse free-surface correction* (*TFSC*). The small amount of *TFSC* in the surface condition assumes that the main ballast tanks contain some residual water—water that cannot be discharged from these tanks. Minimum $\overline{G_{VT}M_T}$, of course, should be positive and of sufficient value to limit dynamic inclinations on surfacing to reasonable amounts.

As indicated, studies of longitudinal transition stability should be undertaken if there is any question as to its criticality in the case of small submersibles. In this regard, it is important to note that the long dimension of main ballast tanks is usually oriented in the fore-and-aft direction, a situation making the *longitudinal free-surface correction* (*LFSC*) much greater than the *TFSC*. Thus, it is entirely possible for minimum transition stability to be based on $\overline{G_{VL}M_T}$ rather than on $\overline{G_{VT}M_T}$.

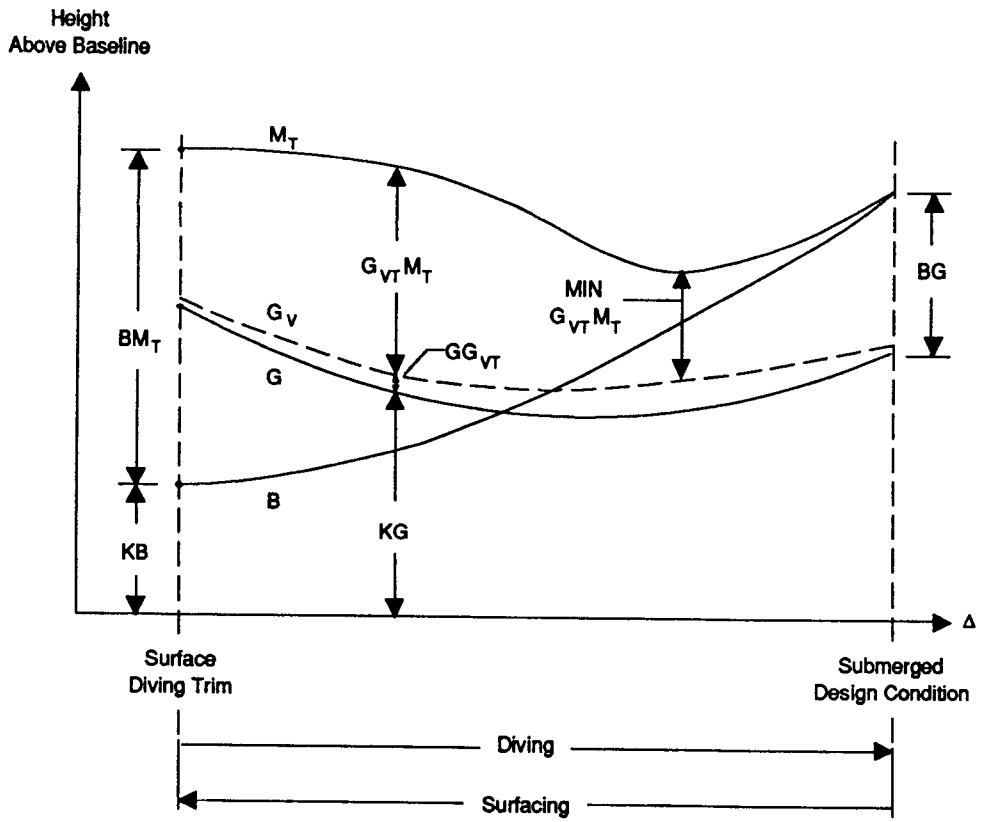


Fig. 20 Transition stability

3.3.4 *Submerged Stability*

It has been seen that B, M_T, and M_L are coincident points for all submerged operating conditions and that B remains "fixed"[15] in the submersible for these conditions. Consequently, unlike surface stability, submerged stability is a function of metacentric height $\overline{BG}$ throughout the entire range of transverse or longitudinal inclinations. Equation (3) is restated here in terms of righting arm $\overline{GZ}$ as

$$\overline{GZ} = \overline{BG} \sin \beta$$

where β is either θ or α. Thus, the transverse and longitudinal stability curves are identical sine curves in the absence of effects causing virtual rises of G. Positive stability is theoretically generated over a range of 0 to 180 deg, this range being practically limited to that throughout which G can realistically be assumed to remain fixed. The operating range of inclinations is further limited by considerations of crew comfort and maximum angles for safe and effective operation of system equipment. $\overline{BG}$ has upper and lower limits. The upper limit is related to maneuvering requirements, as previously discussed, whereas the lower limit must be such as to insure adequate positive stability in the presence of effects causing a virtual rise of G.

The presence of effects causing a virtual rise of G will, most likely, also cause the transverse and longitudinal stability curves to depart from pure sine curves in different degrees. Consider, for example, a submersible whose design includes a variable ballast system using water as ballast. The tanks of this system are only partially full during all or most of the submerged operations, as they also are for the surface and transition operations. Hence, free-surface effects create the aforementioned *TFSC* and *LFSC*. Most likely, these free-surface corrections will not be the same due to tank geometry and orientation. Paralleling the discussion of transition stability, if the long dimensions of the variable ballast tanks are in the fore-and-aft direction, *LFSC* will be greater than *TFSC*, causing the longitudinal stability curve to be lower than the transverse stability curve, less effort being required to trim the submersible than to heel it.

3.3.5 *Influences on Stability*

Influences on stability include the effects of swinging objects, free-surface, certain Source 5 forces and weight-displacement changes. The first three effects act to cause shifts in G, in an *actual* or *virtual* sense, although swinging objects may also cause these shifts in B if the objects contribute to the displacement as well as to the weight of the submersible. The fourth effect causes *actual* shifts of G and B, if the objects involved contribute to displacement, in one, two, or three dimensions. These influences, as they affect the stability of the various operating conditions, must be investigated carefully.

Swinging objects—The distinction between actual and virtual shifts of G is most easily demonstrated by considering swinging, or pendulating, objects. Figure 21*a* diagrams a situation representing an object of weight w suspended from the overhead of the pressure hull at point P on a submerged submersible whose total weight is W. For simplicity, only the line containing the pertinent stability points is shown. If w is not allowed to swing, the vehicle's stability, as measured by the righting arm $\overline{GZ}$ at any angle β, either θ or α, is

$$\overline{GZ} = \overline{BG} \sin \beta$$

If w is allowed to swing, it causes G to actually shift to G_1 with the line of action of W intersecting its upright line of action at G_V, which remains fixed for

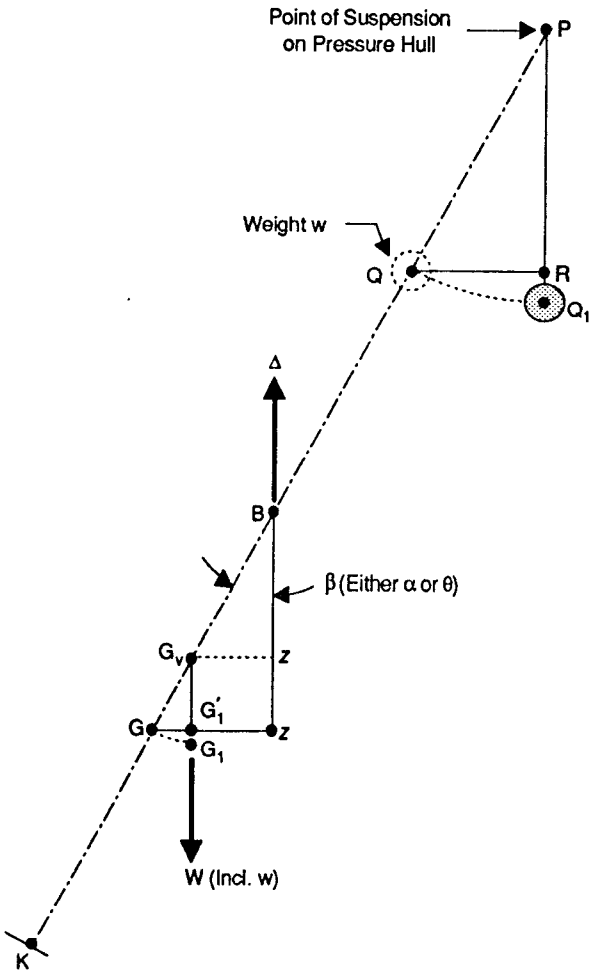


Fig. 21(*a*) Swinging weights

[15]The location of B will vary slightly in submersibles whose designs include variable-buoyancy systems.

all angles of inclination. Thus, the righting arm is reduced to

$$\overline{G_1Z} = \overline{GZ} - \overline{GG_1'} \tag{9}$$

where

$$\overline{GG_1'} = \frac{w \times \overline{QR}}{W}$$

and $\overline{QR}$ is the horizontal distance from the upright position of w to its line of action in the inclined position. The righting arm can also be expressed as

$$\overline{G_VZ} = \overline{BG_V} \sin \beta \tag{9a}$$

In equation (9), the actual horizontal shift of G, $\overline{GG_1'}$, varies with β while, in equation (9*a*), $\overline{BG_V}$ remains constant. Consequently, it is more convenient to use equation (9*a*) in calculating the effect of swinging objects on stability in the equation

$$\overline{BG_V} = \overline{BG} - \overline{GG_V} \tag{9b}$$

where

$$\overline{GG_V} = \frac{w \times \overline{PQ}}{W}$$

Thus, $\overline{GG_V}$ is the virtual shift caused by assuming w shifts from Q to P rather than to Q_1. Hence, for convenience, swinging objects may be considered as being located at their points of suspension for stability purposes, in which case W is considered as acting at G_V rather than at G_1.

Consider a situation in which an object is suspended from the grip of a manipulator of a submersible in the water column. In this instance, it contributes to the submersible's displacement as well as weight. If it is allowed to swing, it will cause a shift in B as well as G. This shift can be found by the method employed above and combined with the shift in G to determine its effect on stability. It can be shown that the virtual rise in B partially offsets the virtual rise in G.

Free surface—Free-surface effects are present any time the surfaces of liquids in tanks, in the bilge, or on topside areas of a submersible are free to seek horizontal positions. Such would be the case with partially filled tanks, with filled tanks containing two or more liquids of different specific volumes, with liquids which have leaked into the bilges or low parts of the pressure hull, and where the structure prevents the complete and rapid draining of water from topside areas on surfacing. These effects are also present, in modified form, any time the surfaces of semi-liquids are free to seek horizontal positions, for example large samples of seafloor ooze or mud being brought to the surface for analysis. The surfaces of liquids immediately seek the horizontal position upon the slightest inclination from the upright, while the surfaces of semi-liquids delay seeking horizontal positions until certain angles are reached. Called *angles of repose*, these angles vary with the nature of the material. Only the free-surfaces effects of liquids will be considered here.

The free-surface effect on stability is considered in two parts: that of the initial free-surface on initial stability or stability at small angles of inclination and that of the overall free-surface on overall stability or stability at large angles of inclination. Studies of initial and overall free-surface effects employ virtual and actual shifts of G, respectively.

Initial free-surface effects are illustrated in Fig. 21*b*, which shows a partially filled tank inclined to some small angle β, which is either θ or α. The surface of the liquid in the tank is free to seek a horizontal position, and, in so doing, a wedge of liquid is transferred from the high to low sides in a manner similar to the transfer of wedges of displacement in the study of surface stability. Thus, the liquid's center of volume moves from b to b_1 along an approximately circular path in the manner paralleling the movement of the center of displacement B, the center of curvature of the path being at a fixed point m corresponding to M_T or M_L. If the liquid's weight w is considered to be concentrated at b, then the free-surface effect is analogous to the swinging weight effect, where m is the point of suspension of w. Following the rationale established for the swinging weight effect, it is again convenient to use equation (9*b*)

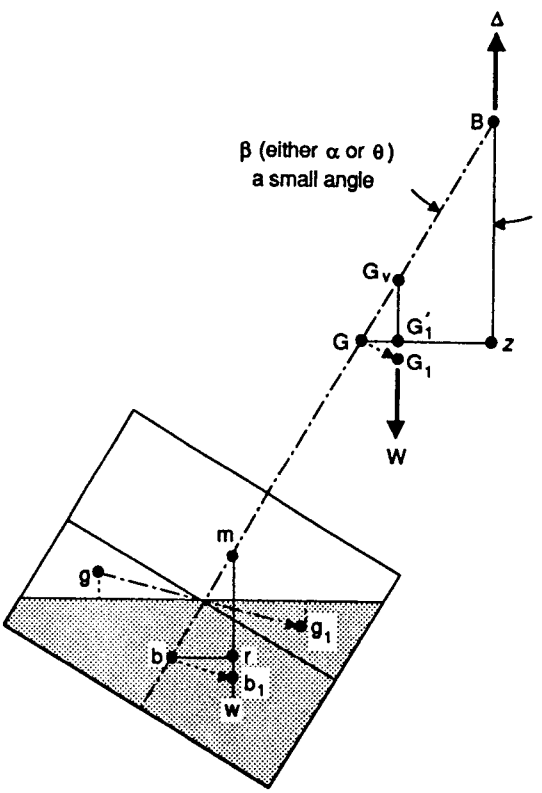


Fig. 21(*b*) Free surface

rather than (9), making use of the virtual rather than the actual shift of *G*. Then,

$$\overline{GG_V} = \frac{w \times \overline{bm}}{W}$$

In this equation, the following substitutions may be made:

$$w = \frac{v}{\delta_T}$$

$$\overline{bm} = \frac{i_0}{v}$$

$$W = \frac{\overline{V}}{\delta_S}$$

where v is the volume of liquid in the tank and δ_T and δ_S are specific volumes of the liquid in the tank and water in which the submersible is floating. Since the transfer of wedges of the tank liquid and displacement of a surfaced submersible for small angles of inclination lead to similar results, $\overline{bm} = i_0/v$ parallels $\overline{BM} = I_0/\overline{V}$. In this case, i_0 is the second moment of free-surface area of the liquid in the tank about the transverse or longitudinal axes through the area's centroid when the surface's angle of inclination is zero. The expression for $\overline{GG_V}$, or *TFSC* or *LFSC*, then becomes

$$\overline{GG_V} = \frac{\delta_S}{\delta_T}\frac{i_0}{\overline{V}} \qquad (10)$$

or

$$\sum \overline{GG_V} = \frac{\delta_S}{\overline{V}} \sum \frac{i_0}{\delta_T} \qquad (10a)$$

which accounts for the total free-surface effect from several different sources on the submersible. For two liquids of different specific volumes in a full tank, equation (10) becomes

$$\overline{GG_V} = \frac{\delta_S i_0}{\overline{V}}\left(\frac{1}{\delta_{T1}} - \frac{1}{\delta_{T2}}\right) \qquad (10b)$$

where δ_{T1} is less than δ_{T2}.

Great care must be taken in calculating i_0 in choosing the correct axis about which the free-surface area *inclines* and the proper *surface permeability* μ_S, which is the fraction of the total area within a tank or space which is occupied by the free-surface of the liquid.

An example of this situation is given in Fig. 22, which shows a transverse section through two tanks of equal dimensions which are partially filled with a liquid. The tanks are vented at the top and are connected by a large transfer line with a valve in it. With the valve closed, the two tanks are isolated and

$$i_{\text{Total}} = 2i_0$$

With the valve open, the two tanks are connected and a condition known as free-communication between them exists. Then

$$i_{\text{Total}} = 2(i_0 + a_0 d^2)$$

where a_0 is the free-surface area of one tank in the $\theta = 0$ position and d is the distance between axes of the tanks and the free-communication axes. This situation is analogous to a pair of port and starboard main ballast tanks with flooding holes in the bottom and vents at the top. If other items in the tank prevent the liquid's free-surface from occupying the total area a_{T0}, then

$$a_0 = \mu_S a_{T0}$$

and

$$i_0 = \mu_S i_{T0}$$

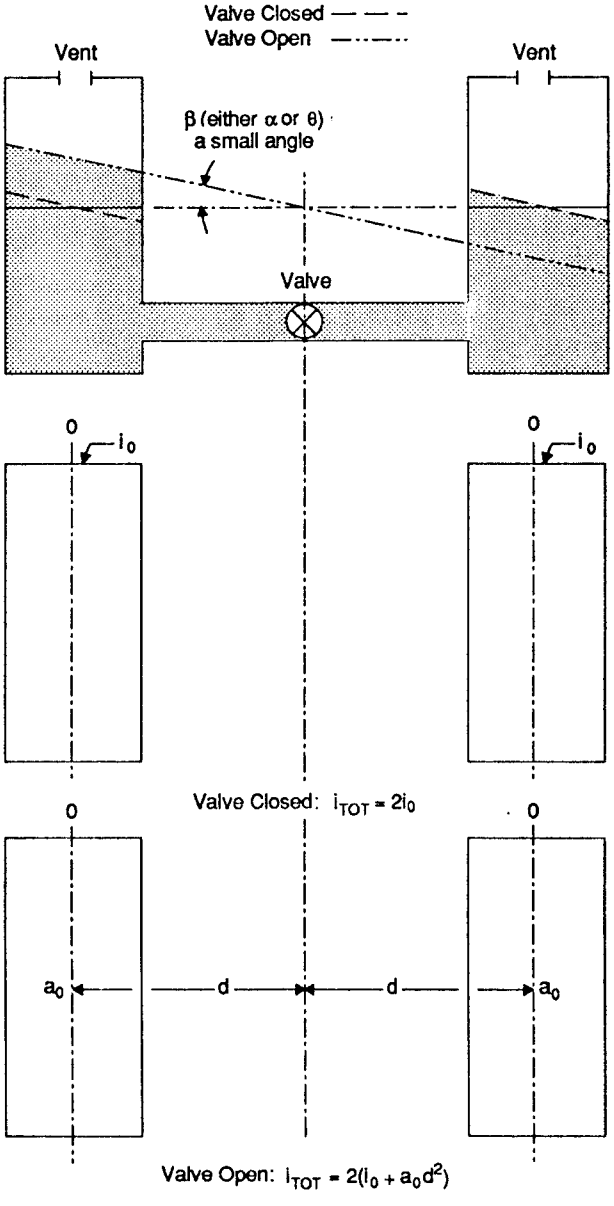


Fig. 22 Isolated free surface and free communication effects

where i_{T0} is the second moment of the total area about the axis through its centroid in the $\theta = 0$ position. In the case of tanks, structure and piping are usually the only items occupying a small portion of a_{T0}, and μ_S is about 0.97. In some designs, the main ballast tanks also house high-pressure air flasks, which would significantly reduce μ_S in the range of tank levels occupied by the flasks.

Theoretically, equation (10) is valid for very small angles of inclination only since this limitation is the basis for the derivation of $\overline{bm} = i_0/v$ as it also is for $\overline{BM} = I/\overline{V}$. Practically, for tanks, they may be used for larger angles of inclination as long as the difference between i_0 and i of the free-surface area in the inclined position is relatively small and as long as this area does not reach the tops or bottoms of the tanks.

Overall free-surface effects on stability must be calculated on the basis of the actual shift of G. The swinging weight analogy is no longer applicable since the liquid weight's "point of suspension" m can no longer be considered even reasonably fixed. Equation (9) is appropriate where

$$\overline{GG_1'} = \frac{w \times \overline{br}}{\Delta}$$

or

$$\overline{GG_1'} = \frac{M}{\Delta}$$

M is called the *moment of transference* and is given by the expression

$$M = C \times \frac{i_0}{\delta}$$

For tanks, C is a coefficient depending on the degree of "fullness" of the tank, its depth/breadth ratio, and angle of inclination. Data for calculating C and other information on this subject are given in reference [1].

Free-surface cannot be completely eliminated in submersibles whose systems include tanks containing liquids. In order for the total i_0 to be reduced to a minimum, in design the free-surface effect on stability can be minimized by careful attention to details of tank dimensions and shape, subdivision, orientation, and piping and cross-connections. Also, the topside structure should be designed so that it sheds water quickly and completely when the submersible surfaces, thereby avoiding a deterioration of stability during the transition and on-surface periods.

Source 5 external forces—A prime example of Source 5 external forces is provided by a submersible's contact with the seafloor in a condition of negative buoyancy. Such contact may be further described as *area, point,* or *line contact* in which the seafloor's reactive force acts on the vehicle over an area or, theoretically, at a point or along a line. Area contact is usually the case, and stability concerns do not arise. However, if the submersible inadvertently grounds, or "sits down," on a pointed rock or on a sharp ridge of coral reef, its stability is impaired.

A grounding situation involving point or line contact is illustrated in Fig. 23. Shown is the actual force system in the grounded condition and the equivalent, free-floating force system. The grounded force system consists of the design condition displacement Δ, the negative-buoyancy condition weight W_N, and the seafloor reaction R, which is equal to negative buoyancy and is assumed to be acting at the keel. The equivalent, free-floating force system consists of the design condition's weight W and displacement Δ. The relationship between the actual and equivalent conditions may be studied by inclining the submersible's centerline θ degrees as indicated. It is seen that the line of action of the resultant of W_N and R, W, passes through G_1 and G_V, $\overline{GG_1}$ varying with θ while $\overline{GG_V}$ remains constant because the line of action passes through G_V for all angles. Consequently, it is again convenient to consider grounded stability in terms of the virtual rise of G. The following expressions obtain:

$$\overline{GG_1} = \frac{R\overline{KG}\sin\theta}{W} = \overline{GG_V}\sin\theta$$

$$\overline{GG_V} = \frac{R\overline{KG}}{W} \qquad (11)$$

and

$$\overline{KG_V} = \overline{KG} + \overline{GG_V} = \frac{W_N\overline{KG}}{W} \qquad (11a)$$

Thus,

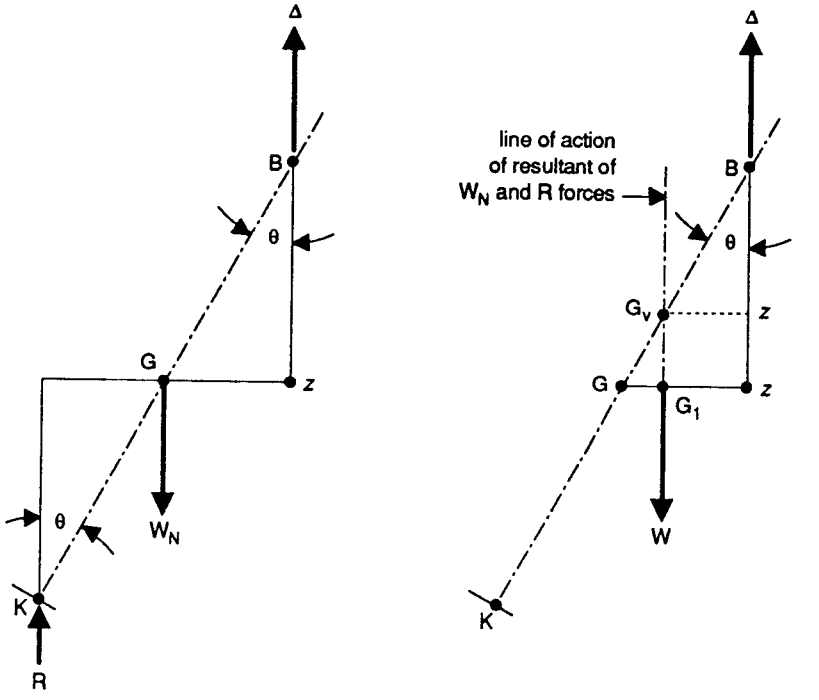
$$\overline{BG_V} = \overline{KB} - \frac{W_N\overline{KG}}{W} \qquad (11b)$$

and stability at any angle θ, while firmly grounded, is given by the expression

$$G_VZ = \overline{BG_V}\sin\theta \qquad (11c)$$

In other words, grounded stability is the same as free-floating stability, the equivalent free-floating condition assuming that a weight equal to the R has been removed from a position at the keel, which causes the submersible's center of gravity to rise from G to G_V.

Weight-displacement changes—These changes involve a shift of objects already on board a submerged submersible or the addition or removal of objects. If the submersible is initially in the design condition of neutral bouyancy, zero trim/list, and stable equilibrium, these changes may cause alterations in this condition's weight, displacement, trim/list, and stability. They may be compensated for to return the submersible to the design condition or uncompensated to allow the vehicle to attain another submerged condition. In both instances, the submersible's control systems are used. Under all conditions, ade-



GROUNDED FORCE SYSTEM

EQUIVALENT FREE-FLOATING FORCE SYSTEM

Fig. 23 Grounded stability

quate positive stability must be maintained and, consequently, the effects of weight-displacement changes on stability are of primary interest here.

Objects of interest in this study either contribute only to the submersible's weight, as would be the case if they were housed in the pressure hull or any container, or they contribute to both its weight and displacement if they are located outside any container. In the first instance, they effect stability by causing shifts in G only, while in the second they cause shifts in both G and B.

Consider the shift of an object already on board a submersible presumed to be in the design condition prior to the shift. If the object contributes to both weight and displacement, the following equations obtain for shifts in G and B:

$$\overline{GG_1} = \frac{w_0 \times \overline{gg_1}}{W}$$

$$\overline{BB_1} = \frac{\Delta_0 \times \overline{bb_1}}{\Delta} \qquad (12)$$

where

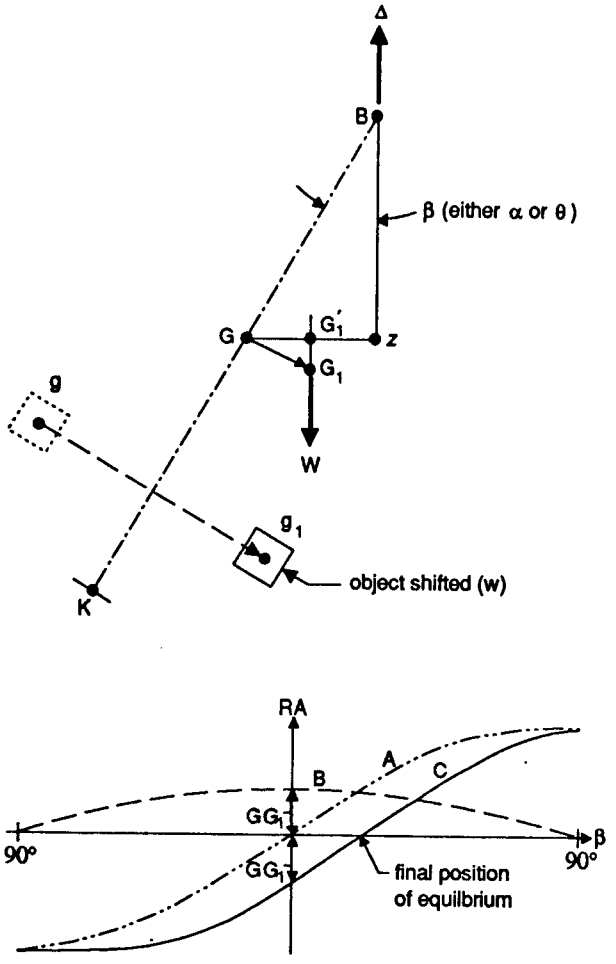
w_0 and Δ_0 = weight and displacement of object
$W = \Delta$ = weight and displacement of submersible in the design condition
$\overline{gg_1}$ and $\overline{bb_1}$ = shifts of object's centers of gravity and displacement

Shifts in the object's and submersible's centers of gravity and displacement may be one, two, or three dimensional with reference to the longitudinal, transverse, and vertical directions. Shifts $\overline{GG_1}$ and $\overline{BB_1}$, of course, parallel shifts $\overline{gg_1}$ and $\overline{bb_1}$, respectively. The effects of two- or three-dimensional shifts can be studied as a series of two or three one-dimensional shifts.

First, observe the effect of a one-dimensional longitudinal or transverse shift of an object contributing only to the submersible's design condition weight as illustrated in Fig. 24. For simplicity, only the line containing G and B in the upright position is shown. Let this line be inclined to some angle β which, as before, is either α or θ angles of trim and heel respectively. It is seen that the change in stability at a particular angle β is given by the expressions

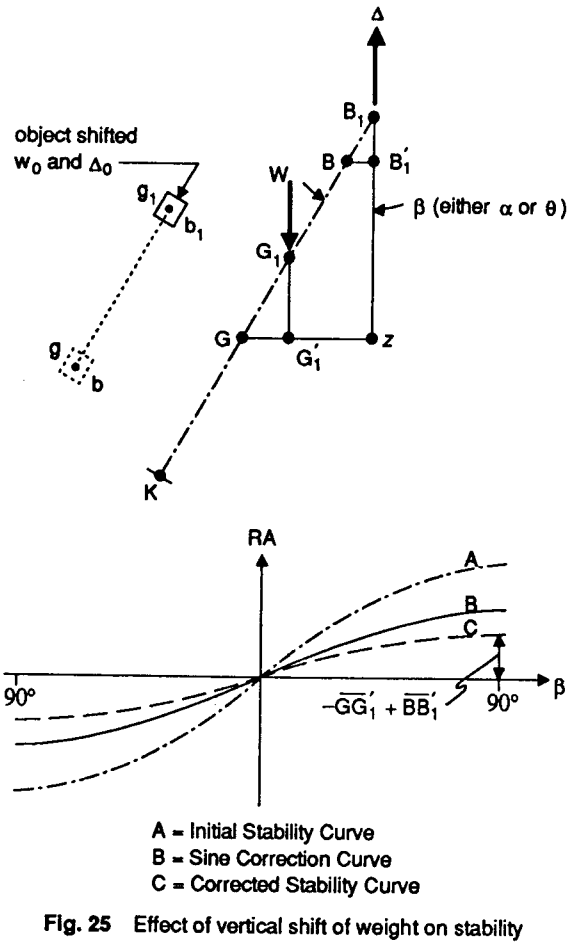
$$\text{Change} = -\overline{GG_1'} = \frac{\cos\beta}{W}\left(-w_0 \times \overline{gg_1}\right) \qquad (12a)$$

The second expression for all angles of β investigated generates the *cosine correction curve* shown in Fig. 24. Using the convention that the first and third quadrants show positive righting arms and the second and fourth negative arms, the figure shows the corrected stability curve as the algebraic sum of the design condition and cosine correction curves. Note the asymmetry of the corrected curve in the first and



A = Initial Stability Curve
B = Cosine Correction Curve
C = Corrected Stability Curve

Fig. 24 Effect of longitudinal or transverse shift of weight on stability



A = Initial Stability Curve
B = Sine Correction Curve
C = Corrected Stability Curve

Fig. 25 Effect of vertical shift of weight on stability

third quadrants and that a new position of trim/list equilibrium is established where the cosine correction and design condition stability curves intersect.

Now, consider the effect of a one-dimensional vertical shift of an object contributing to both the submersible's design condition weight and displacement. This situation is illustrated in Fig. 25 where, as before, the line containing G and *B* is inclined to some angle β which is either α or θ. It is seen that the change in stability at a particular angle β is given by the expressions

$$\text{Change} = -\overline{GG'_1} + \overline{BB'_1}$$
$$= \frac{\sin\beta}{W}\left(-w_0 \times \overline{gg_1} + \Delta_0 \times \overline{bb_1}\right) \quad (12b)$$

where upward and downward directions have a positive and negative sense, respectively. In this instance, note that a *sine correction curve* is involved. The algebraic sum of the design condition stability and sine correction curves yields the corrected stability curve which, unlike the previous example, is symmetrical in the first and third quadrants. In other words, the bow/stern or port/starboard corrected stability curves are the same and the angular position of equilibrium remains at β equal to zero degrees. In this instance, it is assumed that $w_0 \times \overline{gg_1}$ is greater than $\Delta_0 \times \overline{bb_1}$, which results in a decrease of stability for an upward shift of the object and an increase in stability for a downward shift. The opposite results would be obtained if the magnitudes of these products are reversed.

An example of the effect of an added object on stability is provided by the installation of a new item of equipment in a certain location on a submersible. If the design condition is to be maintained, it is necessary to compensate for the added object by adjusting the fixed ballast, fixed buoyancy, or a combination of both so that neutral buoyancy, zero trim/list, and adequate positive stability are maintained. If fixed ballast is assumed to be adjusted in this example, then ballast must be removed from locations such that the resultant force of this ballast

be in a vertical line containing the center of gravity of the object so as to eliminate trim and list. The weight of ballast removed is given by the equation

$$w_{FB} = w_0 - \Delta_0 + \Delta_{FB} \tag{13}$$

which reduces to

$$w_{FB} = w_0 \tag{13a}$$

if both fixed ballast and the added object are in containers such as the pressure hull and contribute only to the submersible's weight. Since fixed ballast is located at the lowest practical location on the submersible, the net vertical shift in G and B almost always results in a reduction of stability. If the situation of equation (13*a*) exists, then the added object's effect on stability is equivalent to shifting w_{FB} vertically upward a distance of $\overline{gg_1}$ where g and g_1 are the centers of gravity of the removed fixed ballast and the added object respectively. Maintaining adequate positive stability in added object situations may dictate adjustments in fixed buoyancy instead of or in combination with adjustments in fixed ballast. Object removal has the reverse effect on stability, nearly always resulting in an increase in stability.

An example of an uncompensated object removal is provided by emergency ascent procedures in which a battery pack is dropped to gain a large amount of positive buoyancy quickly. In this instance, the object contributes to both the submersible's weight and displacement, and change in stability is given by the expressions

$$\begin{aligned} \text{Change} &= -\overline{GG_1} + \overline{BB_1} \\ &= -\frac{w_B \times \overline{Gg}}{W_E} + \frac{\Delta_B \times \overline{Bb}}{\Delta_E} \end{aligned} \tag{14}$$

where

$\overline{GG_1}$ and $\overline{BB_1}$ = vertical shifts of G and B of the submersible, the negative and positive signs indicating reduction and increase in stability respectively

w_B and Δ_B = weight and displacement of battery pack

$\overline{Gg}$ and $\overline{Bb}$ = vertical distance between submersible's and battery pack's centers of gravity and displacement

W_E and Δ_E = emergency ascent weight and displacement of submersible created by dropping battery pack

If the battery pack is going to be used for emergency ascents, its location in the design needs to be carefully considered from the stability and trim points of view. Battery packs are located low in a submersible as it, together with the pressure hull, exert major influences in creating positive stability for normal operating conditions. Its center of gravity is usually somewhat below G, as it is assumed to be in this example, leading to a negative $\overline{GG_1}$, indicating an upward vertical shift and a reduction in stability. From the emergency ascent stability point of view, it would be well to make $\overline{Gg}$ as small as possible. From the trim point of view, the battery pack's g should be as nearly in the same vertical line with G as possible to reduce emergency ascent trim and list to a minimum.

3.4 Dynamic Stability

Dynamic stability has been defined as the righting energy possessed by a submersible or its ability to resist inclinations due to heeling moments from various sources. As seen, under the action of these moments, the vehicle will heel to an angle of θ_M such that righting and heeling energies are equal. In the further examination of dynamic stability, the primary source of heeling moments will be considered—those created by a steady wind force acting on the exposed lateral area of the submersible as it surfaces. Two cases of submersible surfacing are considered: in an upright position and inclined in the direction of the wind.

3.4.1 Case 1

An appreciation for Case 1 may be gained by considering the spring-weight system shown in Fig. 26. The spring, of spring-constant k, has a weight w attached to it in its unloaded position. The weight, when suddenly released, will descend until the spring's deflection is δ_{M1} and the areas of the rectangular weight and triangular spring diagrams are equal; these areas, of course, are the work or energy expended by the weight and stored in the spring, respectively. The spring-weight system will then oscillate until eventually, depending on damping involved, it attains a static equilibrium position at a spring deflection of δ_{S1}. Similarly, a submersible, surfacing in an upright position and meeting a steady beam wind, will heel to a maximum angle of θ_{M1}, comparable to δ_{M1}, and will oscillate until it reaches a position of static equilibrium at a list angle of θ_{L1}, comparable to δ_{S1}—a position at which the righting and heeling moments are equal. At θ_{M1}, as indicated in the figure, the areas under the righting and heeling moment curves are equal, neglecting rolling resistance, as are the areas under the spring and weight diagrams. In this instance, the heeling moment curve is assumed to be a cosine curve. The lever arm between the steady wind force and water resistance are assumed to vary as the cosine of θ. It should again be noted that these curves must be plotted on the basis of moment versus angle of inclination in radians for areas under them to have the dimensions of energy.

3.4.2 Case 2

The spring-weight system comparable to Case 2 is also shown in Fig. 26. In this instance, note that the

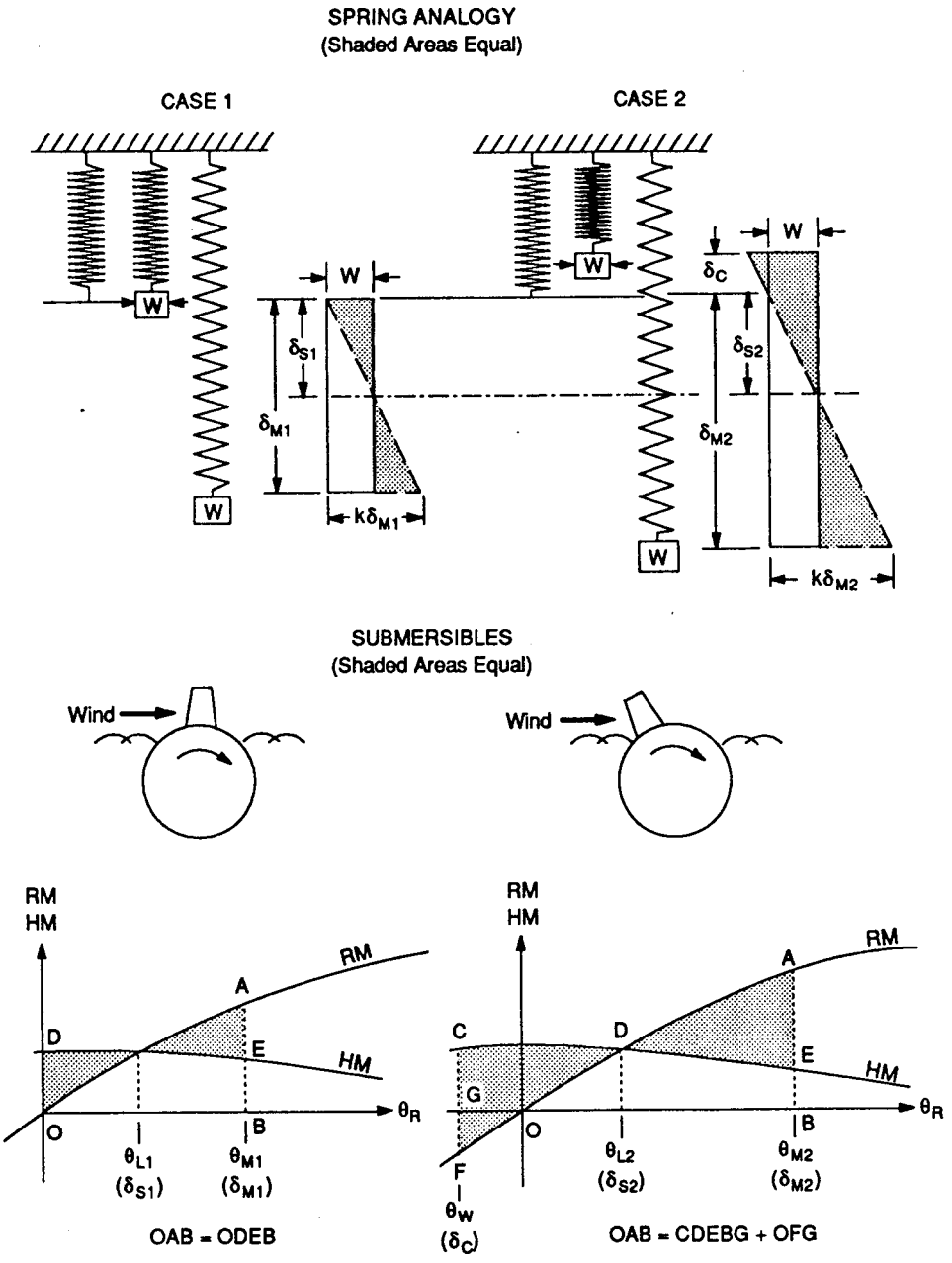


Fig. 26 Dynamic stability

spring is compressed prior to the weight's release. Upon the sudden release of the weight, it will descend to δ_{M2}, which is greater than δ_{M1}. The magnitude of δ_{M2} will be such that the area under the triangular spring diagram below the unloaded spring line equals the areas under the rectangular weight diagram plus the small triangular spring diagram above the unloaded spring line. The area of the small triangle represents energy stored in the spring due to compression and hence is released in the same sense as the potential energy of the weight is released in its vertical position, changing over the distance of $\delta_C + \delta_{M2}$ where δ_C represents the spring's compression. As before, the spring-weight system oscillates and eventually attains a position of static equilibrium at δ_{S2}, which is equal to δ_{S1}. Similarly, a submersible surfacing inclined into the wind will heel to a maximum angle of θ_{M2}, which is greater than θ_{M1}, and will oscillate until a position of static equilibrium is attained at a list angle of θ_{L2}, which is equal to θ_{L1}. At θ_{M2}, as shown in the figure, the area under the righting moment

curve on the side inclined by the wind is given by the expression

$$OAB = CDEBG + OFG$$

where *OAB*, *CDEBG*, and *OFG* are areas comparable to areas under the triangular spring diagram below the unloaded spring line, the rectangular weight diagram, and the small triangular diagram above the unloaded spring line.

It is obvious that Case 2 represents a potentially more dangerous situation than Case 1. Consequently, most weather-related stability criteria for surface vessels are based on situations similar to Case 2. Formulations for wind heeling moments and values for wind velocities and initial inclinations into the wind θ_W are set forth in the U.S. Navy and U.S. Coast Guard stability criteria. Since specified criteria do not exist for submersibles, the designer is left to exercise good judgment. However, since a submersible's transition stability on surfacing is apt to be low, due to free-surface in the main ballast tanks as they are being blown, the designer would do well to investigate stability from this perspective as well as from others.

3.5 Weight-Displacement-Volume Relationships

Hydrostatic principles discussed to this point have been concerned primarily with a simple, two-force system acting on a submersible, that system being composed of its total weight and displacement in the surface, transition, and submerged operating conditions. It is now necessary to consider the composition of total weight and displacement and the relationships between them and volumes which include the topics of weight/displacement ratios, volume-limited designs, and weight-limited designs. This overall subject necessitates the use of terminology and symbols which are by no means uniform throughout the submersible community. The terms and symbols used herein are by and large those used by the U.S. Navy, with departures where they seem appropriate for use with small submersibles.

3.5.1 *Composition of Weight and Displacement*

Figure 27 shows the composition of the total submerged condition weight and displacement, W and Δ, for a neutral-buoyancy, reserve-displacement submersible in which the submerged condition is the design or other underwater operating conditions. A figure for a neutral-buoyancy, zero-reserve displacement submersible would be similar except that main ballast L_{MB}, residual water L_{RW}, and volume of reserve displacement V_{RD} would be omitted. The following discussion concerns the former type of vehicle.

Composition of weight—The composition of the submerged condition weight W may be expressed by any one of three equations:

$$W = W_A + L_{TS} \quad (15)$$

$$W = W_{A1} + W_{FBB} + L_{DT} + L_{MB} \quad (15a)$$

$$W = W_N + L_{MB} \quad (15b)$$

The above major components of W are circled in Fig. 27. For these equations, U.S. Navy nomenclature refers to all items of W_A and L_{TS} as *weights* and *loads* or *ballast*, respectively. W_{A1} and W_A are the submersible's weight in non-operating conditions, while W_N is its weight in the surface, normal diving trim operating condition.

W_A is the *light ship* or *Condition A weight*, which is the submersible's weight in air when it is ready for operation in every respect except that no items of load to submerge, L_{TS}, are on board. It is composed of six or seven so-called SWBS weight groups, indicated in the figure, including "on-board" fluids in systems associated with these groups. In this case, two categories of system fluids exist: those which remain on board during the dive and those which vary during the dive. Examples of the former include mercury in the trim/list systems and oil in the variable buoyancy system; the weights of these fluids are included in W_A as system weights. Examples of the latter include water in the main ballast system and air in the high-pressure air system; the loads of these fluids are included in L_{TS} under the main ballast L_{MB} and the variable load L_V, respectively. Practically all individual weights of W_A remain constant during a dive; an exception is, for instance, liquid weights in the compensating systems of some designs.

L_{TS} is the *load to submerge*, which is the load required to be added to W_A to cause a surfaced submersible to submerge in the descent condition if descent ballast is on board or in the design condition if it is not on board. L_{TS} is composed of load and ballast groups indicated in Fig. 27 to the extent that they are included in a given design. Most of these loads and ballast vary during a dive; an exception is, for instance, the crew and effects of the variable load groups, L_V.

W_{A1} and W_{FBB} are the *Condition A1* and *fixed ballast/buoyancy weights*, where W_{A1} is the Condition A weight minus W_{FBB}. It is useful to separate W_{FBB} from W_A because of the special functions it performs in design margin, weight alterations, and stability considerations (to be discussed further in a review of static control systems). Fixed ballast can take a number of forms, such as pig lead, steel scrap impregnated concrete, and blocks of iron or steel. Fixed buoyancy can be used in the solid, structural, liquid, or gaseous form. Syntactic foam, titanium buoyancy spheres, aviation gasoline, and air are examples of these forms, respectively, the foam and spheres being used the most in designs.

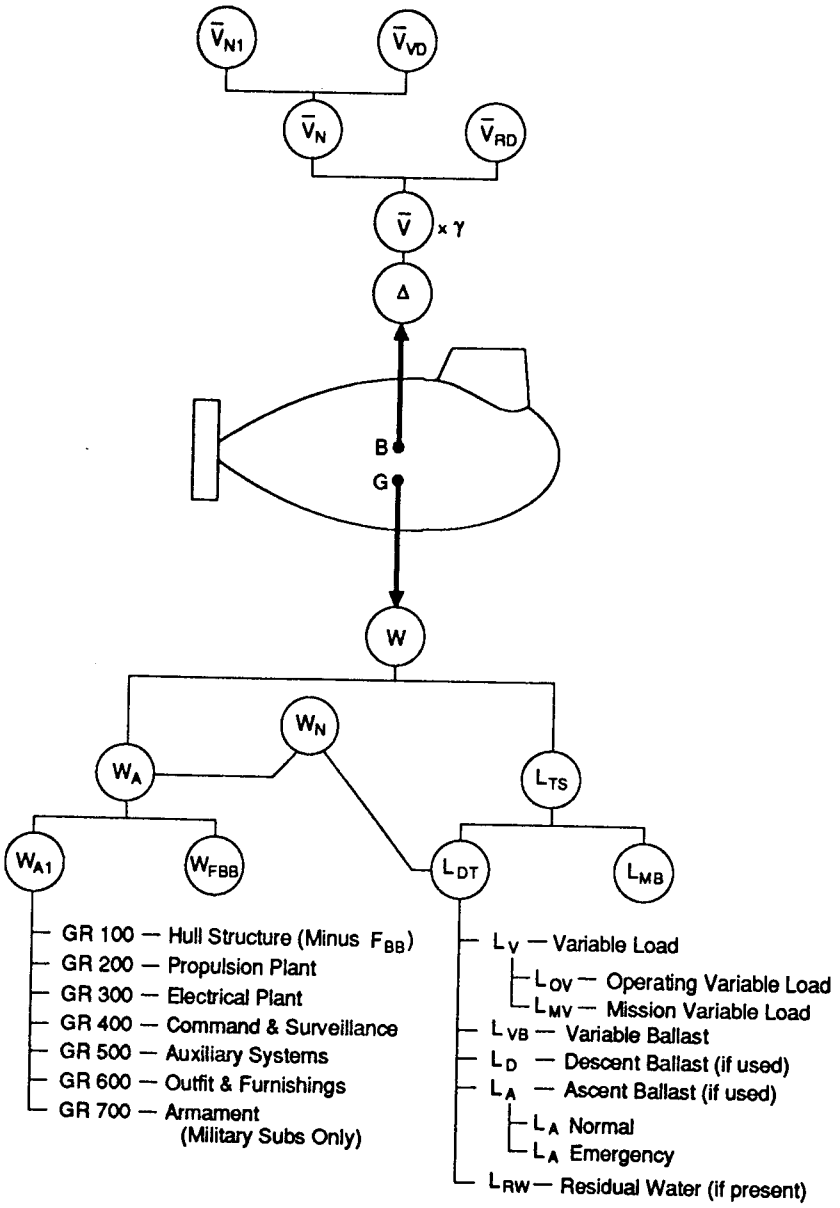


Fig. 27 Composition of weight and displacement

L_{DT} is the *normal diving trim load* which is that portion of the L_{TS} required to be added to W_A to place the submersible in the surface-normal diving trim condition. It is equal to L_{TS} minus L_{MB}.

W_N is the *Condition N* weight or the weight of the submersible in the surface-normal diving trim condition, an operating condition in which the submersible is ready to dive in all respects. In this condition, the vehicle is floating at one of two waterlines—the *normal diving trim waterline* (*NDTWL*), in which no descent ballast L_D is on board or the *normal diving trim waterline-heavy* (*NDTWL-H*), in which this ballast is on board in the form of solid weights, iron/steel pellets, or water ballast. "Normal" is used to define these waterlines when the main ballast tanks are dry except for *residual water* L_{RW}, if any, which cannot be discharged from them. The submersible can be in diving trim condition at other waterlines, generally called *DTWL*, when the main ballast tanks are partially filled with water.

L_{MB} is the *main ballast* or the load which must be added to W_N to cause a surfaced submersible to just submerge in the design condition if descent ballast is not on board or to submerge in the descent condition if this ballast is on board. In the first instance, dynamic forces of thrust and lift are required to achieve the descent condition. The major function of L_{MB} is to reduce the reserve displacement to zero on diving and to restore it on surfacing by discharging this water ballast from the tanks. Consequently, the

total volume of main ballast water, excluding any residual water which cannot be discharged, must be equal to the volume of reserve displacement above the normal diving trim waterline (*NDTWL*). At relatively shallow depths, main ballast is used to initiate the normal ascent condition by discharging a small portion of it, that is, by "putting a bubble" in the main ballast tanks, this bubble expanding as the submersible rises. These tanks are emptied only near or at the surface to conserve pneumatic energy. Emergency conditions require discharging a much greater portion of this ballast at depth.

Yet further subdivision of W_A and L_{TS} involves extensive detail requiring introductory comment. First, consider W_A, which is, of course, composed of a myriad of individual weight items which are parts of all the systems comprising the submersible. These weights, and their centers of gravity and displacement, if involved, must be accounted for with varying degrees of precision—from rough estimates to exact determinations as the design proceeds from its conceptual through detail phases. It is obvious that some kind of orderly system of weight accounting must be adopted in order to avoid confusion and errors and to facilitate design procedures. Although no uniform weight accounting system exists, this publication utilizes the U.S. Navy's system entitled *Ship Work Breakdown Structure* (SWBS), Navships 0900-039-9010, because a large amount of useful weight data has been accumulated and analyzed on the basis of this system.

The SWBS system divides W_A into seven weight groups, as indicated in Fig. 27. In turn, each weight group is divided into weight subgroups, entitled so as to identify the general function or make-up of the items comprising them. Finally, each weight subgroup is divided into weight elements, which are entitled in a similar manner. The following illustration provides an overview of the SWBS system, its three-digit numbering system and hierarchical weight subdivision being presented in sufficient detail for the purpose at hand. In this system, weight

Groups are numbered X_G00, $X_G = 1$ through 7
Subgroups are numbered X_GX_S0, $X_S = 1$ through 9
Elements are numbered $X_GX_SX_E$, $X_E = 1$ through 9

As an example, consider Weight Group 100, Hull Structure:

Group 100—Hull Structure
  Subgroup 110—Shell and Supporting Structure
    Element 111—Shell Plating, Submarine Pressure Hull
    Element 112—Shell Plating, Submarine Non-Pressure Hull
    Etc.

Some confusion arises concerning the classification of weight items serving dual or triple functions. In such cases, the item is always classified by its primary function. For example, a battery pack has the primary function of providing power, but it also may be a part of the trim system, in which its longitudinal movement trims the submersible, or the emergency ascent system in which its release creates positive buoyancy quickly, or both. It is classified according to its primary function as Weight Element 223 (Main Propulsion Batteries) of Weight Subgroup 220 (Energy Generating System—Non-Nuclear) within Weight Group 200 (Propulsion Plant). Composition of the seven weight groups is given in Chapter I. (See the SWBS document, reference [2], for further details.)

The load to submerge, L_{TS}, must now be considered in more detail concerning the load or ballast groups and subgroups comprising the normal diving train load L_{DT}. In this regard, it should be emphasized that the SWBS system is not used for identifying load or ballast groups as it is for the weight groups of W_A. Rather, with the exception of the variable load groups L_V, they are identified by association with the static control systems of SWBS Subgroup 560 (Ship Control System) of Group 500 (Auxiliary Systems), which they serve. It should be remembered that, with few exceptions, items of the various load ballast groups and subgroups vary during a particular dive.

L_V is the *variable load* which, for convenience in defining payload at a later point, may be divided into two subgroups—the *operating variable load* L_{OV} and the *mission variable load* L_{MV}. As its name suggests, L_{OV} is composed of load items necessary for the operation of the submersible including, for example, crew and effects, fuel, lube oil, fresh water, air, oxygen, and so forth to the extent that these items are carried on board in a particular design. L_{MV}, on the other hand, includes load items which are required only to accomplish mission tasks and which are not concerned with the vehicle's operation. Some examples include non-crew, non-crew effects, cargo being transported such as spare parts for seafloor oil/gas installations, and objects and specimens being brought to the surface. As will be seen, L_{MV} may be considered as part of the payload.

L_{VB} is the *variable ballast group* which consists of solid or water ballast or both, which vary during the dive. It is associated with static control systems which provide buoyancy, trim, and list control for two purposes: to maintain the design condition by compensating for changes in the loads and moments of L_V and in Δ due to variations in its components, volume of displacement, and specific weight of water; and to change from the design condition to other submerged operating conditions as necessary to accomplish mission tasks. L_{VB} or variable displacement (to be discussed later) or both, must be used for buoyancy control. L_{VB} may also be used for trim control, or this control may be exercised by static control system weights of W_A which remain on board during a dive. L_{VB} is usually not used for list control.

Static control system weights are used for this purpose. An example of L_{VB} use for both buoyancy and trim control is provided by the traditional trim system of naval submarines employing forward trim, auxiliary, and aft trim tanks. Ballast water is pumped to/from the sea and between these tanks to achieve required buoyancy and trim control. An example of L_{VB} used exclusively for buoyancy control is provided by a design including one variable ballast tank located near the submerged center of displacement, while trim control is exercised by moving the battery pack fore and aft. There are, of course, several other types of static control systems.

L_D is the *descent ballast* consisting of water, carried in a "negative tank," solid ballast, or iron/steel pellets, use of the latter enabling more control over rate of descent. L_D is associated with static control systems, and its function is to create negative buoyancy required to descend and, in some designs, also to create neutral buoyancy at the desired depth by compensating for added displacement due to pressure-resisting shell structures being less compressible than seawater. The design condition is attained at the desired depth by the release of all L_D, in the first case, or by retaining a part or all of it in the second case. In the latter instance, L_D becomes all or a part of L_A for ascent purposes. L_D may be located so as to create a "down-angle" trim and, in some instances, may cause list to facilitate descent. If L_D is not used, descent must be achieved by thrust, an energy-consuming alternative which becomes less and less feasible as the operating depth increases.

L_A is the *ascent ballast* which may be divided into two subgroups—the *normal* and *emergency ascent ballast*. It, too, is associated with the static control systems and has the sole function, upon release, of initiating normal or emergency ascent, thus creating positive buoyancy and, in some instances, being located so as to create an "up-angle" trim to facilitate ascent. L_A-*normal* is composed of solid or pellet ballast, the latter providing for finer control over rate of ascent. The importance of including L_A-*normal* in a design increases with operating depth. For shallow depths, because ascent with intent to surface may be initiated by a slight blowing of the main ballast tanks, use of L_A-*normal* is questionable. As depth increases, however, it becomes increasingly difficult to discharge main ballast water against the higher hydrostatic heads, so the use of L_A-*normal* becomes increasingly important. At very deep depths, its use becomes mandatory. L_A-*emergency* is composed of a heavy item of solid ballast such as a droppable keel, the release of which generates a large amount of positive buoyancy quickly. The use of L_A-*emergency* in a design may be avoided if an item of heavy weight in W_A, such as the battery pack, is used to fulfill this function. Emergency-ascent systems always provide for a large load or weight release and, in some designs, also for the release of the pressure hull from the rest of the submersible.

L_{RW} is the *residual water ballast* or the main ballast water which cannot be discharged from the main ballast tanks because of their shape and location of the flood holes. If it is part of a given design, it is a ballast which remains constant for all operating conditions.

Payload—The definition of payload P_L is debatable within the submersible community. As defined herein, payload consists of the weights and loads of all items on board a submersible which are required to accomplish specific mission tasks but which are not pertinent to its operation. In simple terms,

$$PL = W_{MS} + L_{MV} \tag{16}$$

where W_{MS} is the *weight of the mission systems*—the weights of the scientific and ocean engineering systems and diver support and protection systems which, in the SWBS system, are Elements 591 and 592, respectively, of Subgroup 590 of Group 500. L_{MV} is the mission variable load, as previously identified. Of course, components of certain systems may be required for both the submersible's operation and for accomplishing mission tasks, in which case their weights are excluded from the payload.

Composition of displacement—The composition of the submerged condition displacement Δ may be expressed by any one of the following equations:

$$\Delta = \overline{V}\gamma \tag{17}$$

$$\Delta = (\overline{V}_N + \overline{V}_{RD})\gamma \tag{17a}$$

$$\Delta = (\overline{V}_{N1} + \overline{V}_{VD} + \overline{V}_{RD})\gamma \tag{17b}$$

The above major components of Δ are circled in Fig. 27.

$\overline{V}$ is the total *volume of submerged displacement*. It is the sum of the water-excluding volumes of all items comprising the submersible, all items which it may be carrying as variable loads, and the variable displacement if a variable displacement system is provided in the design. Volumes of the first group include those of the pressure hull and other pressure and non-pressure-resisting containers, fixed-buoyancy to the extent it is included in the design and the water-excluding volumes of structure and system components which are not containerized. Volumes of the second group include those of mission variable load items which may be attached to the submersible such as spare parts being transported to a seafloor installation. The volume of the last item is that created by the variable displacement system, if present in a design, usually by pumping oil to/from bladders to inflate/deflate them.

$\overline{V}_N$ and $\overline{V}_{RD}$ are the *sums of the water-excluding volumes below and above the normal diving trim waterline* NDTWL, respectively. $\overline{V}_{N1}$ is simply $\overline{V}_N$

minus the variable displacement volume, $\overline{V}_{VD}$. $\overline{V}_{RD}$ is the *volume of reserve displacement* above this waterline, which, as noted, must be equal to the total volume of main ballast water. This volume provides freeboard, which elevates the access hatch above the waterline and allows for a semblance of a topside deck area for facilitating launch/retrieval procedures. Note that the use of descent ballast L_D in a design converts some of the $\overline{V}_{RD}$ to displacement, causing the submersible to float at the deeper normal diving trim waterline (*NDTWL-H*).

$\overline{V}_{VD}$, as noted, is the *variable displacement volume* created by the variable displacement system. If used in a design, this system, alone or together with the variable ballast system, has the functions of maintaining neutral buoyancy of the design condition or of creating positive or negative buoyancy as may be required to perform mission tasks. It is not used for list control.

γ is the specific weight of water surrounding the submersible at any time during its operation. The product of $\overline{V}$ and γ, of course, is the weight of the water displaced Δ or the algebraic sum of all vertically directed components of hydrostatic pressure forces acting on the vehicle.

3.5.2 *Weight-Displacement-Volume Relationships*

These relationships are given below for the two surface-normal diving trim conditions and for the submerged-design condition. The "primed" notations indicate the inclusion of descent ballast L_D in the design.

1. Surface-normal diving trim: submersible floating at the *NDTWL* without L_D on board; $\overline{V}_N$ is the volume of displacement below this waterline.

$$W_A + L_{DT} = W_N = \Delta_N = \gamma \overline{V}_N \tag{18}$$

2. Surface-normal diving trim heavy: submersible floating at the *NDTWL-H* with L_D on board; $\overline{V}'_N$ is the volume of displacement below this waterline.

$$W_A + L'_{DT} = W'_N = \Delta'_N = \gamma \overline{V}'_N \tag{18a}$$

3. Submerged-design condition: submersible floating submerged in a condition of neutral buoyancy with zero trim and list L_D if used, having been dropped or retained in the amount necessary to compensate for pressure-resisting shell structures being less compressible than seawater.

$$\left.\begin{matrix} W_A + L_{TS} \\ W_A + L_{DT} + L_{MB} \\ W_N + L_{MB} \end{matrix}\right\} = W = \Delta = \begin{cases} (\overline{V}_N + \overline{V}_{RD})\gamma \\ \overline{V}\gamma \end{cases} \tag{18b}$$

Other submerged operating conditions are attained by altering L_{DT}, $\overline{V}$, or both. The statement previously made that the volume of main ballast water must equal the volume of reserve displacement is apparent from equations (18) and (18*b*) since the specific weight of water admitted to the main ballast tanks is also γ.

Equations (18*b*) for the submerged-design condition are deceptively simple in appearance, but complexities arise from the fact that variations in the environmental parameters of *water temperature, salinity,* and *pressure* usually cause changes in components of W and Δ which must be compensated for if this condition is to be maintained. Consequently, a study of the composition of W and Δ must include a brief review of these variations and their effects which are critical to depth-control considerations. A more detailed discussion of these and other environmental parameters is given in Chapter III.

3.5.3 *Environmental Parameter Variations and Effects*

Sea water temperature, salinity, and pressure variations with unit change in depth, Δh, are called gradients and are designated Δt, Δs, and Δp, respectively, in the following text. Gradient curves are shown in Fig. 28.

Temperature gradient curves are simplified in Fig. 28*a*. As indicated, the water column can be divided into three layers: the surface or mixing layer; the thermocline layer; and the deep-water layer. The thermocline layer, as seen, provides the transition between the surface and deep-water layers in the low and mid latitudes and, by and large, is absent in the high latitudes. Δt may have negligible to large negative values, called seasonal thermoclines, in the surface layer depending on the period of the year. In the low- and mid-latitude thermoclines, it has large, negative values; very large values are associated with "sharp thermoclines" and "density stratification." Finally, below about 1067 m (3500 ft), Δt values become negligible.

Salinity gradient curves are generalized in Fig. 28*b*. As indicated in these example curves, Δs has positive/negative values in the upper part of the water column while it is negligible at deep depths. It should be noted that any positive Δt values must be accompanied by positive Δs values at the same depth for "heavier" water to be below "lighter" water, as must be the case in a stable water column. While open-ocean values of salinity vary from about 33‰ to 37‰, these values may become very small to negligible in local areas such as those near the mouths of fresh-water rivers. These areas give rise to Δs anomalies which can significantly affect submersible operations.

Pressure gradients, as shown in Fig. 28*c*, are constant in the upper portion of the water column and slightly variable in the lower portion due to the increasing significance of sea water compressibility. In the upper water column,

$$\Delta p = 10.04 \times 10^3 \text{ pascals/m depth salt water}$$
$$= 0.444 \text{ psi/ft depth salt water} \tag{19}$$

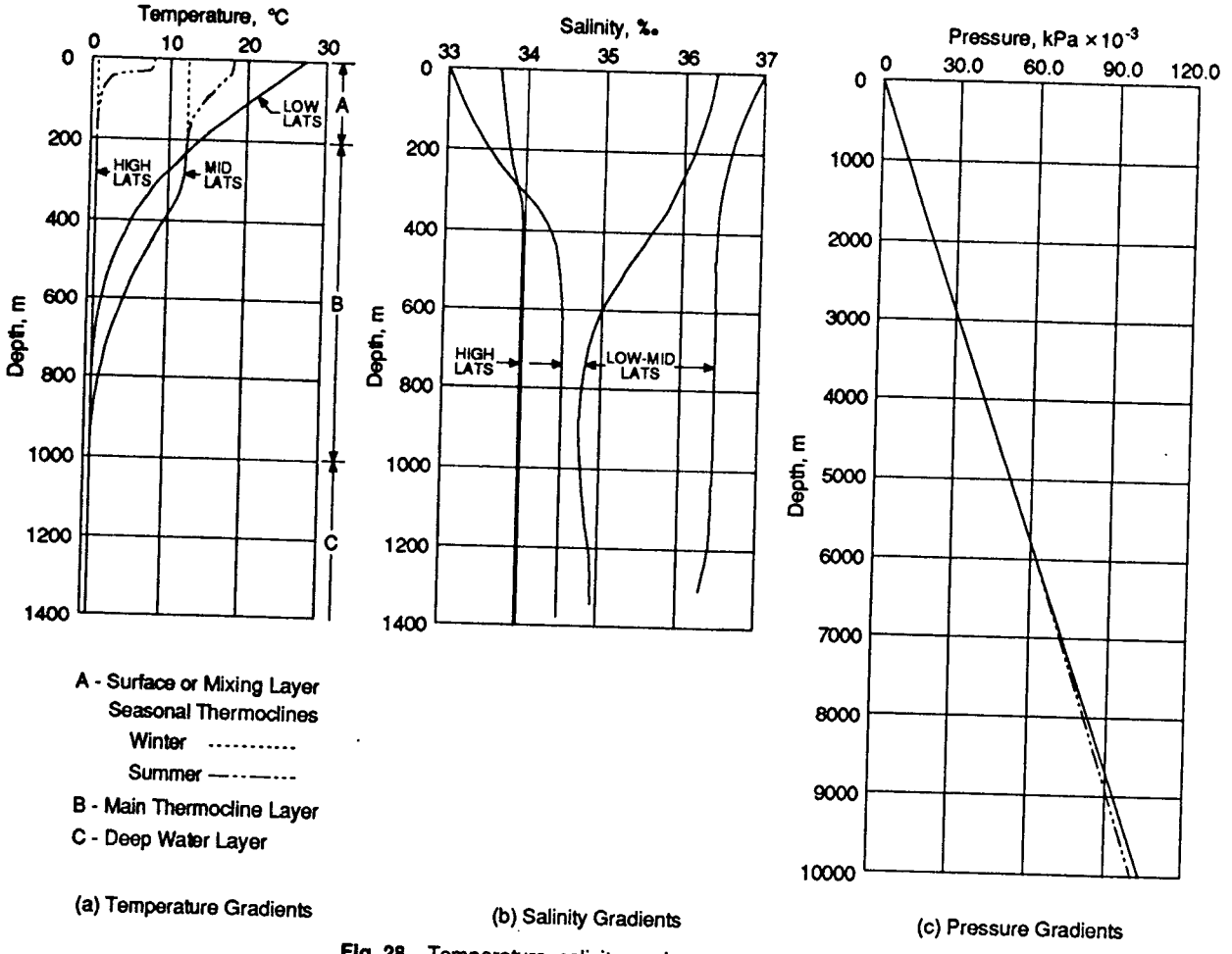


Fig. 28 Temperature, salinity, and pressure gradients

while in the lower water column,

$$\Delta p = 10.04 \times 10^3 + 13.57h \times 10^{-3} \text{ pascals per meter depth of salt water}$$

$$= 0.444 + 0.6h \times 10^{-6} \text{ psi per foot depth of salt water} \qquad (19a)$$

Sea water temperature, salinity, and pressure gradients cause changes in a submersible's buoyancy by altering both its displacement and weight. All three gradients cause these alterations as the submersible changes depth. The first two gradients may also be in effect as it moves from one location to another at constant depth. Temperature gradients cause *thermal expansion/contraction* of sea water, liquids in the submersible's systems and shell structures thereby changing the specific weights of the liquids and the volumes of the structures. Salinity gradients also cause changes in the specific weight of sea water. Finally, pressure gradients cause changes in volumes of all items comprising the submersible which are susceptible to compression,[16] including pressure-resisting shell structures, liquids in certain systems, and sea water itself. The total change in buoyancy, then, is the algebraic sum of the displacement and weight alterations caused by each parameter gradient. These alterations are now considered in more detail.

Displacement alterations—Both the volume of displacement and specific weight of sea water are affected by parameter gradients. The change in the volume of displacement per unit change in depth is

$$\Delta \overline{V} = \sum \Delta \overline{V}_i \qquad (20)$$

where $\Delta \overline{V}_i$ is the change in the volume of displacement per unit change in depth of an individual item, principally, the pressure hull and other pressure-resisting containers. In turn,

$$\Delta \overline{V}_i = -\left.\frac{\partial \overline{V}_i}{\partial p}\right|_{t=\text{const.}} \Delta p + \left.\frac{\partial \overline{V}_i}{\partial t}\right|_{p=\text{const.}} \Delta t \qquad (20a)$$

$$= -A + B$$

[16] Fixed buoyancy, in the form of syntactic foam, is subject to compression; its compressibility at design depth must be very small or negligible for the foam material to be satisfactory.

The signs in equation (20*a*) arise from the fact that the volume of the individual item decreases with increasing pressure and decreasing temperature. Terms *A* and *B* represent the instantaneous and gradual effects, respectively, of the pressure and temperature gradients on $\Delta\bar{V}_t$. The gradual, thermal expansion/contraction effects are awkward to calculate if a submersible changes depth relatively fast. As noted, the temperature gradient Δt tends to become negligible below about 1067 m (3500 ft), and the *B* term disappears. Actually, even in the upper portion of the water column, simplified formulations for $\Delta\bar{V}_t$ often neglect this term as being of secondary importance to compressibility. Term *A* will be considered further presently.

The change in the specific weight of sea water per unit change in depth is given by the expression

$$\Delta\gamma = -\left.\frac{\partial\gamma}{\partial t}\right|_{s,p=\text{const.}} \Delta t + \left.\frac{\partial\gamma}{\partial s}\right|_{t,p=\text{const.}} \Delta s + \left.\frac{\partial\gamma}{\partial p}\right|_{t,s=\text{const.}} \Delta p \qquad (20b)$$

$$= -A + B + C$$

Again, the signs arise from the fact that the specific weight of sea water increases with decreasing temperature, increasing salinity, and increasing pressure. Term *A* affects $\Delta\gamma$ in the upper layer of the water column, if seasonal thermoclines exist, as well as in the main thermocline layer. If the thermoclines are sharp, very large, negative Δt values create large, positive $\Delta\gamma$ values, leading to density stratification layers. A submersible which is negatively buoyant above such a layer may come to rest and "sit" on it in a condition of neutral buoyancy advantageous from the operational point of view.[17] Sharp and unanticipated thermoclines may pose ballasting and control problems, leading to difficulties in penetrating the layer. Term *A* disappears at depths below about 1067 m (3500 ft) or where more or less isothermal conditions exist. Term *B* plays a minor role in affecting $\Delta\gamma$ in the open ocean, where salinity varies within the relatively narrow range of 33‰ to 37‰. However, it has a very marked effect in areas where Δs gradients are pronounced, such as would be the case of a submersible entering/exiting an area in which salinity is substantially outside this range. As noted, the environs of mouths of fresh and brackish water rivers are examples of such areas. Finally, in Term *C*, both elements of the term, and hence the term itself, may be assumed, for purposes at hand, to be constant throughout the water column, as indicated in Fig. 28.

Equations (20) are useful primarily in illustrating the mathematical relationships between the variables involved. With this background, a more design-related approach is now pursued.

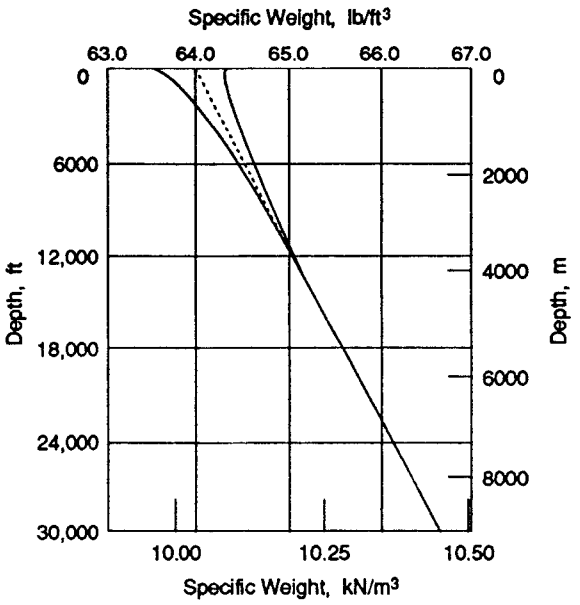


Fig. 29 Specific weight of sea water versus depth

The specific weight of seawater, γ, as a function of depth is shown in Fig. 29 for world-average, open-ocean conditions of temperature, salinity, and pressure gradients. First, consider only the effect of pressure and compressibility. A given volume of sea water at the surface is reduced by 4 percent at 9144 m (30 000 ft). Consequently, since Term *C* of equation (20*b*) is assumed constant, γ at the surface for standard condition of temperature and salinity will be 10.05 kN/m^3 (64.0 lb/ft^3) and 10.46 kN/m^3 (66.6 lb/ft^3) at 9144 m (30 000 ft). γ, varying linearly with depth, is shown in Fig. 29. Now, consider temperature and salinity effects. The variability of these parameters in the upper portion of the water column causes departures from standard sea water; see Fig. 30, which shows σ as a function of temperature and salinity.[18] From this figure, the limits of variations from standard sea water at the surface can be calculated as 9.99 and 10.10 kN/m^3 (63.6 and 64.3 lb/ft^3). Variations in these parameters with depth lead to the "envelope" curves shown in Fig. 29. Submersibles diving in the open ocean must be capable of operating satisfactorily when encountering sea water of specific weights anywhere within this envelope. Note that in the deep portion of the water column, standard seawater conditions of temperature and salinity may be assumed to exist for the purpose at hand.

A comparison of Terms *A* of equation (20*a*) and *C* of equation (20*b*) indicates the relative compressibil-

[17] Military submarines may take advantage of stratification layers, such as by "sitting" on them while the vehicle is engaged in evasive procedures. Few advantages, if any, of this water-column condition are seen for submersibles, most of whose missions are sea-floor oriented.

[18] The oceanographer's seawater density measurement convention is given by the equation $\sigma = (\rho - 1)10^3$, where ρ is the mass density in grams/cm^3. Hence, if ρ is 1.02523 grams/cm^3, σ is 25.23. σ for standard sea water is 25.00.

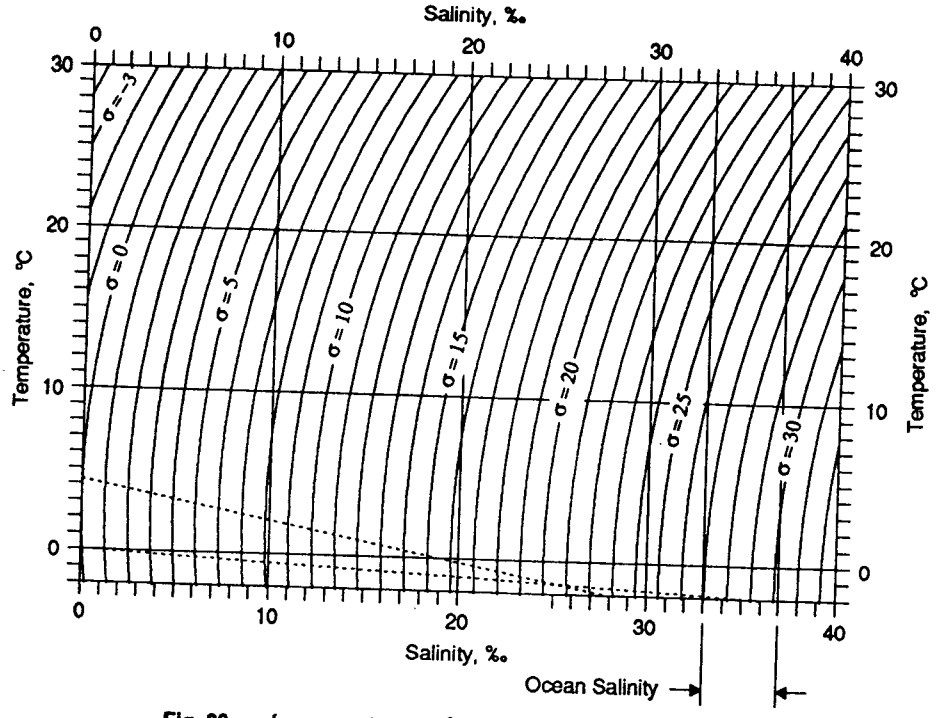


Fig. 30 σ for sea water as a function of temperature and salinity

ity of a submersible's pressure-resisting shell structures and sea water, thereby providing guidance for buoyancy-control measures to counter these effects of compressibility. If shell structures are less compressible than sea water, these effects will cause the submersible to gain positive buoyancy and a decreasing rate of descent as the vehicle dives. They may be countered by retaining all or part of the descent ballast, increasing variable ballast or decreasing variable displacement in order to attain neutral buoyancy at the operating depth. Conversely, if these structures are more compressible than sea water, these effects will cause the vehicle to gain negative buoyancy and an increasing rate of descent as it dives. In this instance, no descent ballast is retained and the effects are countered by decreasing variable ballast or increasing variable displacement to attain neutral buoyancy at the desired depth.

The Appendix to this chapter details this subject by providing compressibility formulations for sea water and for spherical and ring-stiffened cylindrical pressure-resisting shell structures. In these formulations, the Terms *A* and *C* are called compressibility factors; C_{sw}, C_s, and C_c are these factors for sea water and for spherical and cylindrical shell structures, respectively. In addition, unit compressibility *U* is used where

$$U = \frac{\text{change in volume}}{\text{unit change in depth}} \frac{1}{\overline{V}_0} = C \frac{1}{\overline{V}_0} \qquad (21)$$

$\overline{V}_0$ is the fully submerged but uncompressed volume of the item under consideration. Subscripts of *U* match those of *C*. Figure 49 in the Appendix provides an illustration of the compression of a spherical pressure hull and sea water. Figures 50 and 51 are diagrams of unit compressibility for steel spherical and ring-stiffened cylindrical hulls of varying radii and shell thicknesses compared with the unit compressibility of sea water. As noted from the formulations, the compressibility of shell structures varies directly with the material's yield strength and inversely with its elastic modulus as well as with depth. In general, metallic spherical shell structures are less compressible than sea water, whereas nonmetallic shell structures are more compressible. For ring-stiffened cylindrical shell structures, thick-shell structures of deep-diving submersibles are less compressible than sea water, whereas thin-shell structures of shallow-diving submersibles are more compressible.

Weight alterations—Submersible systems containing liquids whose volumes and densities are altered by sea water temperature gradients, pressure gradients, or both experience changes in system weight or load. The primary systems involved are compensating systems used in conjunction with (1) systems protecting equipment located external to the pressure hull, such as battery packs and motors, (2) buoyancy systems using liquids, usually aviation gasoline, as the buoyancy material, (3) liquid-fuel storage systems located external to the pressure hull, and (4) liquid-cargo storage systems. Usually, only Systems

(1) are of concern in the design of small submersibles. Systems (2) were used in the design of a unique type of vehicle called a *bathyscaph*. Systems (3) might be applicable to larger submersibles using diesel engines, and Systems (4) to submarine tankers. In addition to compensating systems, main and variable ballast systems may also experience some change in system loads.

The composition of compensating system liquid weights W_{CL} occupying fixed volumes such as those of tanks or battery-pack containers is given by the expression

$$W_{CL} = W_L + W_{SW} \tag{22}$$

where W_L is the weight of liquids associated with the four systems noted above—oils used to protect equipment, aviation gasoline, fuel oils, and crude oils or other liquids which may be transported. W_{SW} is the weight of the sea water in the system. Sea water transmits ambient hydrostatic pressure to all parts of the system, thereby equalizing internal/external pressure, which permits shell structures associated with compensating systems to be thin and relatively light. As the submersible moves vertically through the water column, W_L remains constant, but the volume it occupies changes. This change per unit change in depth ΔV_L is given by the expression

$$\Delta V_L = + \left.\frac{\partial V_L}{\partial t}\right|_{p = \text{const.}} \Delta t \; - \left.\frac{\partial V_L}{\partial p}\right|_{t = \text{const.}} \Delta p \tag{22a}$$

$$= + \; A \qquad - \; B$$

Signs of Terms A and B indicate that ΔV_L decreases with decreasing temperature and increasing pressure or vice versa. As before, temperature effects of Term A are gradual while the pressure effects of Term B are instantaneous. If time is sufficient for internal/external temperature equalization, Term A has a greater effect on ΔV_L than does Term B. The change in the compensating system's liquid weight per unit change in depth is

$$\Delta W_{CL} = \Delta V_{SW} \times \gamma \tag{22b}$$

where ΔV_{SW} is the change in volume of sea water per unit change in depth and γ is the sea water's specific weight at the depth in question. ΔV_{SW}, of course, must be equal to a minus ΔV_L, since a decrease in ΔV_L must be equal to an increase in ΔV_{SW}, or vice versa, for the total volume of liquids in the compensating system to remain constant.

Equations useful in design which are equivalent to Terms A and B are, respectively,

$$\Delta V_L = \alpha_L \times V_L \times \Delta t \tag{22c}$$

and

$$\Delta V_L = \frac{\Delta p \times V_L}{K} \tag{22d}$$

where

α_L = coefficient of cubical expansion for the liquid in question,

V_L = volume of the liquid prior to change in temperature, and

K = liquid's bulk modulus of elasticity.

A dramatic illustration of change in weight is afforded by the example of a submersible tanker carrying 20 320 mt (20 000 lt) of 61° API gasoline. ΔW_{CL} is about a plus 122 mt (120 lt) for each 6°C (10°F) decrease in temperature and about a plus 6.1 mt (6 lt) for each 306 kPa (44.4 psi) increase in pressure.

3.5.4 *Weight-Displacement-Center Records*

Discussion of this subject necessitates recalling criteria for neutral-buoyancy submersibles in various operating conditions. Criteria for the two primary conditions are outlined as follows:

1. Surface-diving trim condition (Condition N)

   $W_N = \Delta_N$ for neutral buoyancy

   $\overline{KG} \lessgtr \overline{KB}$ for stable equilibrium, where the relationship between $\overline{KG}$ and $\overline{KB}$ is such that $\overline{GM_T}$ and $\overline{GM_L}$ are always positive

   $\overline{LCG} \geq \overline{LCB}$ for zero trim, if equal, or trim by the stern (drag) when floating at the normal diving trim waterline if $\overline{LCG}$ is greater than $\overline{LCB}$.

   $\overline{TCG} = \overline{TCB}$ for zero list

2. Submerged-design condition

   $W = \Delta$ for neutral buoyancy

   $\overline{KG} < \overline{KB}$ for stable equilibrium or positive $\overline{BG}$

   $\overline{LCG} = \overline{LCB}$ for zero trim

   $\overline{TCG} = \overline{TCB}$ for zero list

$\overline{TCG}$ and $\overline{TCB}$, the only dimensions in the above outline not previously defined, are the transverse centers of weight and displacement as measured from the submersible's centerline, or x–z, plane. Data must be acquired and calculations made to assure that the magnitudes and distribution of weight and displacement are such that the above criteria, and criteria for other conditions, are met. Of the calculations associated with these criteria, those for $\overline{TCG}$ and $\overline{TCB}$ are usually not explicitly made because almost all submersibles have centerline planes of symmetry which makes the requirement for the equality of these dimensions relatively easy to meet.

Records of weight, displacement, and vertical and longitudinal center data and calculations are compiled during the design process, finalized during construction, and revised throughout the submersible's lifetime to reflect weight and displacement alternations. They are kept with varying degrees of

detail and precision as warranted by the nature of the input data—from little detail for data estimated during the early stages of design to great detail and precision for data from the working plans of the detail design stage and as returned from the building yard during construction. Data and calculations comprising the final record are assembled into a booklet called, for example, *Weight Control Report.*

Record booklets can be assembled in various ways but, in general, are divided into "summary sheets" and "detail sheets." Summary sheets contain weight, displacement, and center information for the major weight and load or ballast groups and for the nonoperating (A1 and A) and operating conditions. Detail sheets contain this information for the myriad of individual items comprising the groups shown on the summary sheets. Figure 31 is an example of a form[19] used for both summary and detail sheets. In this case, the figure shows summary information for the SWBS weight groups, load/ballast groups and the A1, A, surface-normal diving trim, and submerged-design condition conditions. Using SWBS terminology, detail sheets would contain weight, displacement, and center information for all items comprising the subgroups of each SWBS weight group and, in turn, for all items comprising the elements of each weight subgroup. Other detail sheets would contain this information for individual items comprising the load/ballast groups.

[19]Forms used often contain additional information which is omitted in Fig. 31, such as, in this example, shipyard plan numbers and weight accounting codes associated with individual items.

3.5.5 *Weight/Displacement Ratios*

Two weight/displacement ratios are important to the hydrostatic aspects of submersible design: those of pressure-resisting shell structures, primarily the pressure hull, and those of fixed buoyancy. The pressure hull weight/displacement ratio, also called *hull fraction*, is the weight of the empty pressure hull

Title	Weight					Displacement				
	WT	KG	VMW	LCG	LMW	DISP	KB	VMΔ	LCB	LMΔ
100—Hull Structure (excluding FBB)										
200—Propulsion Plant										
300—Electrical Plant										
400—Command and Surveillance										
500—Auxiliary Systems										
600—Outfit and Furnishings										
700—Armament (military subs. only)										
CONDITION A1										
Fixed Ballast/Buoyancy to Balance										
CONDITION A										
Normal Diving Trim Load										
CONDITION N										
Main Ballast										
SUBMERGED—DESIGN CONDITION	W	KG	ΣVMW	LCG	ΣLMW	Δ	KB	ΣVMΔ	LCB	ΣLMΔ

equal

$BG = KB - KG = \Sigma VM\Delta/\Delta - \Sigma VMW/\Delta$

equal

CONDITION A1										
Normal Diving Trim Load										
Main Ballast										
SUB. CONDITION (excluding FBB)	W′		ΣVMW′		ΣLMW′	Δ′		ΣVMΔ′		ΣLMΔ′
Fixed Ballast/Buoyancy to Balance	W_{FBB}		VMW_{FBB}		LMW_{FBB}	Δ_{FBB}		$VM\Delta_{FBB}$		$LM\Delta_{FBB}$
SUBMERGED—DESIGN CONDITION	W		ΣVMW		ΣLMW	Δ		ΣVMΔ		ΣLMΔ

Fig. 31 Weight-displacement-centers summary

W_{PH} divided by its displacement when fully submerged, Δ_{PH}, and is designated as $(W/\Delta)_{PH}$. It is a parameter expressing the efficiency of this structure in terms of its ability to provide for an excess of displacement over that required to support its own weight. This excess of displacement, to the extent it exists, is used to support the weights of other items comprising the submersible. $(W/\Delta)_{PH}$ is a function of the design depth h_D, the type of structure used, its material and assumptions concerning geometric imperfections (out of roundness), and residual stress, considerations which are detailed in Chapter VI.

For the purpose at hand, Fig. 32 will suffice in demonstrating $(W/\Delta)_{PH}$, h_D, and material relationships for an unstiffened sphere assumed to have perfect geometry and no residual stress. It is seen that 1.0 is the critical value of $(W/\Delta)_{PH}$, where the empty pressure hull is in a condition of neutral buoyancy with zero efficiency with respect to its ability to support other weights. Below and above 1.0, the empty pressure hull floats or sinks, possessing positive or negative efficiency, respectively. Note that for a specified h_D, $(W/\Delta)_{PH}$ decreases as the strength/density ratio of the material increases, a consideration which becomes increasingly critical as h_D increases from the view of efficient pressure hull design. Note also that for a specified $(W/\Delta)_{PH}$ and hull type, increases in the design depth h_D can be achieved only by using materials of increasing strength/density ratios. These considerations remain the same irrespective of types of pressure hulls used, the most common of which is ring-stiffened cylinders, closed by elliptical or hemispherical heads, for the shallower design depths and spheres or combinations of spheres for the deeper design depths.

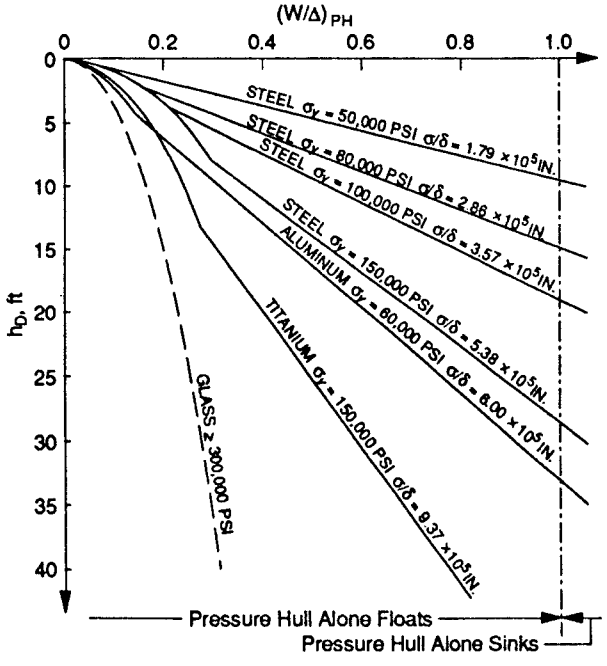


Fig. 32 $(W/\Delta)_{PH}$ versus depth for spherical pressure hulls

Fixed buoyancy weight/displacement ratio $(W/\Delta)_{FBY}$ is the weight of the fixed buoyancy W_{FBY} divided by its displacement Δ_{FBY}. $(W/\Delta)_{PH}$ is a parameter expressing the efficiency of the fixed buoyancy in providing positive buoyancy. It is also a function of both the design depth and the type of fixed buoyancy and materials used. Types of fixed buoyancy primarily used are hollow spheres and solids; liquids had been used to a limited extent in the past. Liquids nominally are called "fixed buoyancy" even in view of the facts that their volumes change significantly with changes in temperature and pressure and that some of their volumes may be released to control the ascent rate. Titanium, syntactic foams, and aviation gasoline are examples of materials used with hollow-sphere, solid, and liquid types of fixed buoyancy. Representative $(W/\Delta)_{FBY}$ values are 0.40 (for 44 in. titanium spheres), 0.60 to 0.65 (syntactic foams), and 0.80 to 0.85 (for aviation gasoline including weight of containment structure). Information on fixed buoyancy, in general, is contained in Chapter IV.

3.5.6 *Volume and Weight Critical Designs*

This subject is introduced by considering the composition of a submersible from a different perspective than heretofore discussed. In this perspective, items comprising the submersible, excepting the pressure hull and fixed ballast/buoyancy, are placed into one of three categories: those that

1. must be placed inside the pressure hull (personnel, life-support items, equipment and instruments which must be readily accessible, etc.)—whose combined weight is W_I and volumetric requirements, including adequate space for movements and the operation and servicing of equipment, is V_I,
2. must be located outside the pressure hull (propellers, manipulators, control surfaces, high-pressure air flasks, etc.)—whose combined weight and displacement are W_o and Δ_o, and
3. can be located either inside or outside the pressure hull (motors, tanks, and liquids associated with static control systems, batteries, etc.).

A weight/displacement ratio for the pressure hull $(W/\Delta)_{PH}$ can be found for a specified design depth h_D and a selected hull type and material and assuming certain hull geometric imperfections and residual stress. Thus, if V_I, the minimum volume that must be enclosed by the pressure hull, is known, its displacement and weight, Δ_{PH} and W_{PH}, can be calculated. Then

$$\Delta = \Delta_{PH} + \Delta_o \quad \text{and} \quad W = W_{PH} + W_I + W_o$$

The relationship between Δ and W tentatively identifies the design as being *volume critical* or *weight critical* depending on whether Δ is *greater* than W or *less* than W. Decisions concerning the locations of Category 3 items can now be made based on safety, cost, efficiency, reliability, accessibility, attainability, second-function (for example, using the battery pack as an emergency ascent weight), and buoyancy considerations. From the buoyancy consideration alone, Category 3 item location depends on whether a design is tentatively volume or weight critical. For a volume-critical design, the item should be located so that the positive buoyancy is either reduced or increased the least. For a weight-critical design, the reverse situation obtains, the item being located so as to be the more effective in reducing negative buoyancy.

The location of Category 3 items either inside or outside the pressure hull, as based on all considerations noted, permits equations (23) to be rewritten as

$$\Delta = \Delta_{PH} + \delta\Delta_{PH} + \Delta_o + \Delta_{IO}$$

and (23*a*)

$$W = W_{PH} + \delta W_{PH} + W_I + W_o + W_{IO}$$

where $\delta\Delta_{PH}$ and δW_{PH} are the changes in pressure hull displacement and weight as the result of locating Category 3 items inside/outside the pressure hull. δW_{PH} includes added weight in this hull by any inserts and penetrators made necessary by locating the item on the outside. Δ_{IO} is the combined displacement of Category 3 items located outside the pressure hull, and W_{IO} is the combined weight of these items located both inside and outside this hull. Great care must be taken in calculating Δ_o, Δ_{IO}, W_o, and W_{IO}. In this regard, items located outside the pressure hull may be called *free-flooding* or *isolated* items to indicate that they are either in direct contact with sea water or isolated from it in pressure-resisting or pressure-compensated containers. The weight of a "free-flooding" item of Category 3 may be heavier if located outside the hull than it would be if located inside due to added requirements for protection in the more hostile environment. If the outside item is "isolated," the weight recorded must include its weight and the weight of the container and liquids, while if it is inside a pressure compensated container the displacement recorded is essentially that of the container. The Δ and W relations of equations (23*a*), then, actually identify the design as being volume critical or weight critical. In practice, however, since the pressure hull provides the great bulk of displacement, a volume-critical, or weight-critical, design may nominally be said to be one in which the pressure hull provides more, or less, displacement than is required to support a weight of the submersible.

A volume-critical design requires fixed ballast W_{FB}, usually lead, to be added to equations (23*a*) to achieve neutral buoyancy. If this ballast appears excessive, one recourse the designer has is to use another material of a lower strength/density ratio, thereby raising the $(W/\Delta)_{PH}$ ratio, a procedure which is likely to result in cost savings and easier fabrication since these attributes are associated with lower strength/density materials.

A weight-critical design, for a specified design depth, must attain neutral buoyancy by (1) lowering the $(W/\Delta)_{PH}$ ratio to the extent feasible, (2) enlarging the pressure hulls' internal volume beyond that needed to contain the items in it, (3) adding fixed buoyancy, or (4) a combination of these options. Option (1) involves using pressure hull materials of higher strength/density ratios or more structurally efficient pressure hull types or both, which, as noted, forces the adoption of spheres or combinations of spheres for the more weight critical designs. Options (2) and (3) are best considered together since the choice between them is related to the relative magnitudes of their weight/displacement ratios, $(W/\Delta)_{PH}$ and $(W/\Delta)_{FBY}$. The choice should favor the lower of these two ratios; that is, if $(W/\Delta)_{FBY}$ is lower than $(W/\Delta)_{PH}$, neutral buoyancy should be attained by adding fixed buoyancy and vice versa. This practice will result in the least increase in size and weight of the submersible and, consequently, the least impact on other design considerations involving size/weight dependent items such as propulsion, energy storage, static/dynamic control systems, handling systems, and, of course, costs. For example, as the size of a submersible increases, more power is required to propel it at the design speed which, in turn, requires an increase in energy storage capacity to meet the design endurance—both design speed and endurance are specified in the mission/performance requirements. It is evident, then, that any increase in size/weight in an effort to attain neutral buoyancy will cause second-order displacement/weight increments in equations (23*a*), requiring further increments in positive buoyancy by Options (2) or (3) perhaps in combination with Option (1). Thus, attainment of neutral buoyancy is an iterative procedure.

A summary may now be made with respect to shallow- and deep-diving submersibles. Shallow-diving submersibles may have pressure-hulls of various geometries which are best suited for design considerations relating to interior arrangements and hydrodynamics. Materials used may have comparatively low strength/density ratios and are relatively low in cost and easy to fabricate. These vehicles are in the lower range of $(W/\Delta)_{PH}$ values and are usually volume critical, requiring fixed ballast to attain neutral buoyancy. Deep-diving submersibles must give priority to structural considerations, normally narrowing the choice of pressure-hull geometries to spheres and combinations of spheres. Hull materials have relatively high strength/density ratios usually obtainable in materials which are more costly and more difficult to fabricate. These vehicles are in the

higher ranges of $(W/\Delta)_{PH}$ values and are weight critical, usually requiring fixed buoyancy to attain neutral buoyancy.

3.6 Hydrostatic Characteristics

The characteristics are dependent on the form of the submersible and include

1. Displacement Δ
2. Stability dimensions *KB*, *LCB*, KM_T, KM_L
3. Waterplane area A_w
4. Longitudinal center of flotation *LCF*
5. Tons per inch of immersion *TPI*
6. Moment to change trim one inch *MTI*
7. Change in displacement per foot trim by stern $\delta\Delta F$
8. Form coefficients
   —Block coefficient C_B
   —Longitudinal prismatic coefficient C_p
   —Midship coefficient C_0

Excepting Characteristics 1 and 2, which have been discussed previously, brief comments on the remaining hydrostatic characteristics follow (see reference [1] or other texts on naval architecture for more detailed discussions and methods of computation).

The *waterplane area*, at various waterlines, is that area occupied by the submersible's structure or equipment which excludes sea water. The transverse and longitudinal distribution of this area about the x and y axes through its centroid determine the area's moments of inertia which, as has been seen, are required for the calculation of the transverse and longitudinal metacentric radii $\overline{BM}_T$ and $\overline{BM}_L$. The *longitudinal center of flotation* is the waterplane area's centroid. In the static sense, inclinations of a surfaced submersible occur about the x and y axes, heel about the x axis and trim about the y axis.

Tons per inch of immersion is given in English units by the expression

$$TPI = \frac{A_w}{12\delta} \tag{24}$$

where δ is the specific volume of water in which the submersible is floating. Equation (24) is useful in determining the parallel change in draft of a surfaced submersible caused by a given weight change δw or vice versa, where

$$\text{Parallel change in draft} = \frac{\delta w}{TPI} \tag{24a}$$

Moment to change trim one inch is given in English units by the expression

$$MTI = \frac{\overline{GM}_L}{12L} \tag{25}$$

where $\overline{GM}_L$, as previously defined, is the longitudinal metacentric height and L is the distance between fore-and-aft draft marks. It is used in determining the change in trim δT_R of a submersible caused by a weight change creating a trimming moment T_RM where

$$\delta T_R = \delta T_F + \delta T_A = \frac{T_RM}{MTI} \tag{25a}$$

δT_F and δT_A are the changes in fore-and-aft drafts, respectively.

Considered together, the four expressions above are useful in calculating weight adjustments and distribution required in order for the submersible to float at the normal diving trim waterline. In this regard, its fore-and-aft drafts can be changed by any of the following: a longitudinal shift of weight on board; weight addition or removal at or from a position in a vertical line with centroid[20] of the added or removed "layer" of water; and weight addition or removal at or from a position not in a vertical line with this centroid. The first change causes a change in trim only; T_RM is the weight times the longitudinal distance it is moved. The magnitudes of δT_F and δT_A are proportional to the distances of the fore and aft draft marks from *LCF*, since the initial and final waterlines intersect at this point, and are of opposite signs. The second change causes a parallel change in waterlines only—δT_F and δT_A are equal and of the same sign. The third change causes both a parallel change in waterlines and a change in trim. These combined effects in changing drafts are best considered independently, using, as an example, a weight w being added to the submersible at a distance a aft of the centroid of the "layer" of displacement $\delta\Delta$, whose magnitude, of course, is equal to w. The procedure is outlined as follows:

1. Assume w is added directly over/under the centroid of the added layer of displacement and calculate the parallel sinkage caused.
2. Shift w to its actual location, creating a T_RM of w times a, and calculate the change in trim caused.
3. Calculate δT_F and δT_A, proportioning their magnitudes to the distances of the fore-and-aft draft marks from the *LCF* of the final waterplane area.
4. The final fore-and-aft drafts are the original fore-and-aft drafts plus the algebraic sum of changes in these drafts due to parallel sinkage and change in trim by the stern.

Calculating a change in drafts due to the removal of a weight would involve the reverse of this procedure.

The *change in displacement per foot trim by the stern* is used in calculating the correction to the displacement which must be made when the submersible is floating at a waterline inclined to the

[20] This centroid is very nearly the *LCF* of the waterplane prior to the weight change if the weight involved is small. If the weight change is large, the centroid of the layer is the average of the *LCF*s of initial and final waterplanes.

waterline used for making displacement calculations. In English units, it is given by the expression

$$\delta\Delta F = 12TPI \times \frac{d}{L} \tag{26}$$

where d is the distance of *LCF* from amidship, or the center of length point, and L is essentially the submersible's length. $\delta\Delta F$ is positive or negative depending on whether *LCF* is aft or forward of amidship. Note that it is zero when *LCF* is at amidship even though an inclination, or trim, exists. The correction to displacement $\delta\Delta$ is

$$\delta\Delta = \delta\Delta F \times T_R \tag{26a}$$

where T_R is the trim in feet.

Coefficients of form are used to aid in defining the overall forms of submersibles and are meaningful parameters in speed-power, envelope volume, and other calculations. Three of the more common are the *block*, *longitudinal prismatic*, and *amidship* coefficients. Expressions for them are

$$C_B = \frac{\overline{V}}{L \times B \times T}$$

$$C_p = \frac{\overline{V}}{L \times A_o}$$

$$C_o = \frac{A_o}{B \times T} \tag{27}$$

where

$\overline{V}$ = volume of displacement at draft T,
L = submersible's length,
B = its maximum breadth or diameter, and
A_o = area of its amidship section at draft T.

Hydrostatic characteristics dependent on form are calculated for the submersible floating at a number of parallel waterlines, these waterlines also being either parallel or nonparallel[21] with the heel or base line. Data thus generated are plotted as curves in a diagram variously called Curves of Form, Displacement and Other Curves, or Hydrostatic Curves. These diagrams may contain all or part of the hydrostatic characteristics listed with the range of waterlines varying from "zero" to "fully submerged" to "Condition A" to "fully submerged" waterlines. In any event, it should be noted that data derived from these curves are correct only when the submersible is floating at waterlines parallel to those used for constructing the curves. "Displacement" is the only data corrected for other waterlines. However, all data can be used with sufficient accuracy when floating at waterlines inclined a maximum of about two degrees with the "curve" waterlines. Figure 33 is an example of Curves of Form.

3.7 Static Control Systems

This subject is introduced with a brief review of submersible control systems in general and the func-

[21] If calculations are based on waterlines which are not parallel to the keel or baseline, T_F is always less than T_A, with their difference being called drag and not trim, trim being reserved for the difference between T_F and T_A when the submersible is floating at waterlines inclined to those on which the calculations are based. The purpose of drag is to improve surface operating characteristics which primarily involves submerging the propeller as deeply as possible.

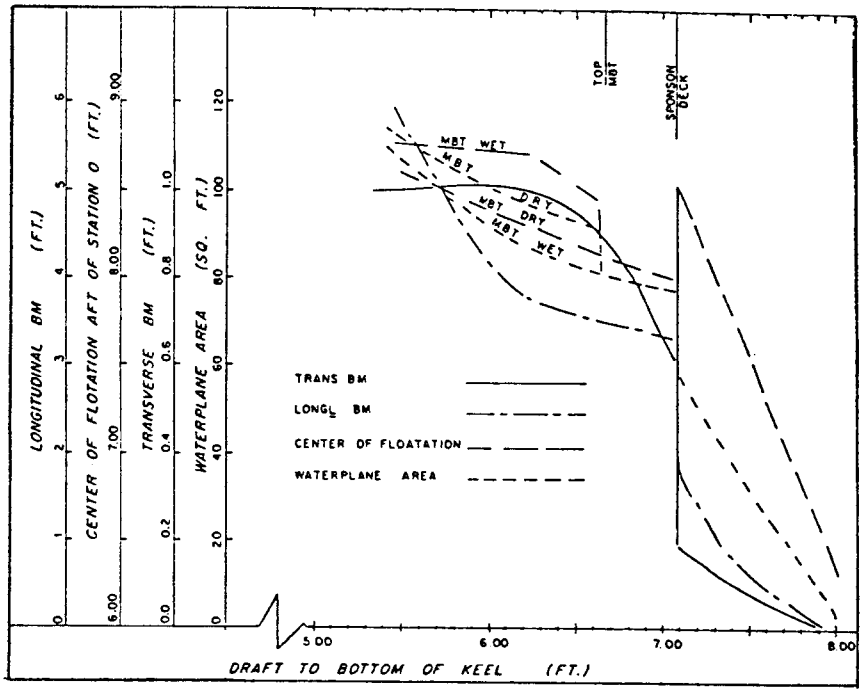


Fig. 33 Example of curves of form for *Alvin*

tions they serve. Submersible control involves the *retention of* or the *purposeful departure from* existing states of motion or design condition criteria or both, excepting the criterion for positive statical stability, which must always be retained. As has been seen, this control is exercised by static control systems, using static forces of weight and displacement, and by dynamic control systems using dynamic forces of thrust and lift. It was noted that the effectiveness of static control systems is limited to control in three degrees of freedom (translational in the $\pm z_0$-axis direction and rotational about the x and y axes), whereas dynamic control systems are capable of exercising control in all six degrees of freedom if required. Both systems, listed on pages 209 and 210, have been previously discussed in various contexts. Principles underlying static control systems are further considered at this point, while those pertinent to dynamic control systems are presented in the next section of this chapter.

3.7.1 *Fixed Ballast-Buoyancy Systems*

These systems have the functions of achieving and retaining the design condition criteria of neutral buoyancy, zero trim/list, and positive stability, acting alone or together. As has been seen, this ballast or buoyancy or both is fixed during a dive or series of dives, but may be altered while the submersible is on the surface, usually out of the water, to compensate for Condition A (Light Ship) weight or displacement or both changes. Consequently, these systems may be considered static control systems even though they cannot be used to control the submersible during actual operations.

The nature and functions of fixed ballast-buoyancy systems are revealed by further consideration of the weight-displacement-moment summary shown in Fig. 31. From this summary, it is apparent that

$$W_{FBB} - \Delta_{FBB} = \Delta' - W' \tag{28}$$

for volume-critical designs and

$$\Delta_{FBB} - W_{FBB} = W' - \Delta' \tag{28a}$$

for weight-critical designs where W_{FBB} and Δ_{FBB} are the weight and displacement of fixed ballast-buoyancy and W' and Δ' are the total submerged weight and displacement of the submersible, excluding the fixed ballast-buoyancy system. For fixed ballast, associated with equation (28), Δ_{FBB}, of course, is zero if the ballast is located in the pressure hull or other containers. Otherwise, the material's weight/displacement ratio should be as high as possible—this ratio is 11.09 for lead, with the left side of equation (28) becoming $0.91 W_{FBB}$, which is why this material is commonly used. On the other hand, for fixed buoyancy associated with equation (28*a*), the weight/displacement ratio should be as low as feasible. Hollow spheres provide the lowest ratios; for example, this ratio for high-strength titanium alloy spheres is about 0.40 for a design depth of 6096 m (20 000 ft), with the left side of equation (28*a*) becoming $0.60\Delta_{FBB}$. Despite this attractive feature and their use in deep-diving submersible designs, buoyancy spheres have drawbacks in the fact that they provide "all or nothing" buoyancy. Consequently, they must be considered as critical structures along with the pressure hull insofar as reliability of material and fabrication is concerned. Additionally, spheres are awkward shapes for packaging within a given envelope tending to increase its size. Solid fixed buoyancy, in the families of polyurethane and syntactic foams, provide the next lowest ratios. As an example, a syntactic foam for 6096 m (20 000 ft) would have a weight/displacement ratio of about 0.63, with the left side of equation (28*a*) becoming $0.37\Delta_{FBB}$. Despite the higher weight/displacement ratio, this form of fixed buoyancy, sometimes called "negative lead," is not concerned with the critical problem of "all-or-nothing" buoyancy although its characteristics of strength and resistance to compression and water absorption must be of the highest order. Other advantages include the fact that it can be shaped to fill voids and to provide a fair surface for a portion of the submersible's envelope. Liquid "fixed" buoyancy, usually aviation gasoline, has the highest weight/displacement ratios, about 0.80 to 0.85, which includes the weight of the containment structure. As has been noted, this material is no longer used because of this high ratio, its compressibility (which is about twice that of sea water), and dangers of handling and loss due to a rupture of its container.

Equation (28) states, simply, that negative buoyancy furnished by fixed ballast must equal the positive buoyancy of the total submerged weight and displacement, excluding fixed ballast, for neutral buoyancy to exist. Equation (28*a*) states the reverse of this situation for fixed buoyancy. This system must perform other functions as well, including providing for (1) adequate stability, (2) zero trim/list,[22] (3) a change in weight and/or displacement of Condition A1 and/or loads which remain fixed during a dive, and (4) a weight margin in anticipation of weight estimating errors during design. Adjustments in all terms of equations (28) and (28*a*) may be made during the design process to maintain equality. These adjustments become smaller as the design progresses until they disappear and the terms are fixed when the submersible is completed.

Functions (1) and (2) allow for a certain degree of flexibility in choosing vertical and longitudinal locations of items comprising the submersible, thus facilitating its general arrangement. Stability and trim

[22]Compensation for list usually presents few problems due, largely, to the more or less symmetrical arrangement of items about the submersible's x-y plane and will not be considered further.

fixed ballast-buoyancy assure that the centers of gravity and displacement are properly separated on the common line of action of weight and displacement forces and that this line is normal to the vehicle's x-axis. Wherever possible, however, item location should always promote achieving these design condition criteria so as to minimize the need for fixed ballast-buoyancy for this purpose. For example, a heavy item, such as a battery pack, should be located as low on the centerline as possible and near the longitudinal center of displacement. In addition, a conscious effort should be made to "balance" weights and displacements of items located fore and aft to the extent permitted by other design considerations. For stability,

$$\overline{BG} = \frac{\sum VM\Delta - \sum VMW}{\sum \Delta} \tag{29}$$

where $\sum VM\Delta$ and $\sum VMW$ are the sums of vertical moments of displacement and weight, with reference to the keel, as indicated in Fig. 31. It can be seen from equation (29) that fixed ballast and buoyancy should be located as low and high, respectively, as feasible, low fixed ballast making only a very small contribution to $\sum VMW$, whereas high fixed buoyancy makes a relatively large contribution to $\sum VM\Delta$. For zero trim,

$$\sum LM\Delta = \sum LMW \tag{29a}$$

where $\sum LM\Delta$ and $\sum LMW$ are the sums of the longitudinal moments of displacement and weight, with reference to the bow or amidship, as also indicated in Fig. 31. The longitudinal moment of fixed ballast-buoyancy must be such as to assure the equality of these moments.

Function (3), to the extent that fixed ballast-buoyancy is incorporated into a design for this purpose, permits a ready means of compensating for changes in a submersible's weight-displacement made between dives or on a more permanent basis. Suppose, for example, that the mission tasks of a shallow-depth submersible require different suits of equipment at the same location on the vehicle's bow on two consecutive dives, suit A being considerably less negatively buoyant than suit B, $W_A - \Delta_A < W_B - \Delta_B$. Attachment of suit B could quite easily be compensated for by removing and shifting fixed ballast, say, lead bars at two locations within the pressure hull, the weight of the lead removed equalling the difference in the negative buoyancies of suits A and B, $-F_B$, and the moment of lead shifted being equal and opposite to the moment, or couple, formed by $-F_B$ and the weight of lead removed. Note that if $-F_B$ had been in a position between the locations of the lead instead of at the bow, compensation could also be made by removing lead from each location. The total lead removed equals $-F_B$, with the proportion removed from each location being such that the center of gravity of the total lead removed is in the same vertical plane as $-F_B$. As an example of a more permanent change, consider a new item of equipment installed inside the pressure hull of a deep-diving submersible. In this case, weight and moment compensation could be made by adding fixed buoyancy, in the form of syntactic foam, having positive buoyancy, $+F_B$, where $+F_B$ equals the new equipment's weight, with the added syntactic foam being distributed longitudinally so the $+F_B$ and the added weight are in the same vertical plane. In both examples, as cautioned in Section 3.3, the impact of these changes on stability must be investigated. This involves determining the net change in $\overline{BG}$ due to vertical shifts in B and G caused by changes in buoyancy and weight. Obviously, there are limits to which these changes can be accommodated, these limits being established by minimum values of $\overline{BG}$. Extending these limits by increasing the vertical separation of B and G is a reason for using fixed ballast and buoyancy together in a design.

Function (4) involves a weight margin which is included in the design weight-displacement summary usually as a percentage of the surface displacement. This percentage increases as confidence in the accuracy of weight estimates decreases. For example, this percentage might be relatively high, perhaps about 10 percent, for an entirely new design because of limited specific, or "hard," weight data available and uncertainties in the vertical and longitudinal distribution of weight which, in turn, lead to uncertainties in fixed ballast-buoyancy requirements for stability and trim. Conversely, the weight margin percentage would be considerably lower for a design similar to that of an existing submersible for which ample weight, displacement, and center data are available. The weight margin which remains when the submersible is completed must be placed on board as margin ballast to achieve neutral buoyancy. The goal, of course, is to have little or no margin ballast on board while avoiding negative margin ballast, a situation which requires a reduction in other weights to retain neutral buoyancy. This procedure usually leads to a lowering of some aspect or aspects of vehicle performance, including a reduction in payload, which often results in *failure to meet contract penalties*. Thus, the importance of good judgment in selecting weight-margin percentages is underscored.

It is sometimes necessary to adjust the fixed ballast-buoyancy system after the submersible is built to allow the variable ballast-buoyancy system to compensate for all conditions of variable loading. This subject will be pursued shortly in the context of its relationship to the equilibrium polygon.

3.7.2 *Variable Ballast-Displacement and Trim Systems*

These systems may be used separately or combined in a wide variety of buoyancy-trim control systems. The functions of any one of them are (1) to

retain the design condition criteria of neutral buoyancy and zero trim or (2) to purposefully cause departures from these criteria to facilitate underwater operations and the accomplishing of mission tasks. In contrast with fixed ballast-buoyancy systems, the magnitude or longitudinal moment or both of weight, ballast, or displacement of these systems vary during the dive. In performing the first function, the buoyancy-trim control system compensates for changes in magnitude or longitudinal moments or both of weights, variable loads, and displacement which occur during the dive. Performing the second function requires that this system also has the capacity to create negative/positive buoyancy and nonzero trim to the extent specified either directly or indirectly by the performance requirements.

Variable ballast and displacement systems may be designed to perform fully both functions or perform them partially in retaining or causing departures in neutral buoyancy only. In the latter instance, a trim system must be used in conjunction with them to retain or cause departures in zero trim. If these functions are fully performed, the system's ballast or displacement magnitudes and their longitudinal moments must both be capable of being varied during a dive—otherwise, only their magnitudes need be varied. The major reason for creating negative/positive buoyancy is to facilitate the submersible's descent/ascent. This subfunction is often relegated in whole or in part to special-purpose variable ballast called descent/ascent ballast. Descent/ascent systems will be discussed separately.

Variable ballast-displacement systems may be classed as *reversible* or *irreversible* depending on whether or not ballast or displacement can be repeatedly increased and decreased. Examples of reversible systems include those using sea water which can be repeatedly pumped to/from/between variable ballast tanks and oil which can also be repeatedly pumped to/from/between bladders or bellows, inflating/deflating them to increase/decrease displacement. Examples of irreversible systems include those using iron/steel pellets (shot) as variable ballast and aviation gasoline as the buoyancy material. Neither pellets nor gasoline is retrievable once released from the submersible. For a reversible system, the number of reverses or cycles possible is dependent on power required by pumps and energy storage considerations. These considerations, as well as pumping rates, become more critical as operating depths increase, since the pumps now work against higher hydrostatic heads. For irreversible systems, their capacities depend on the pre-dive quantities of ballast or displacement materials or both on board. Since pumps are not involved in these systems, they are not depth-sensitive.

The trim system's only function, where used, is to retain or cause a departure in the criterion for zero trim. Consequently, only the longitudinal moment of this system's weight varies during a dive, its magnitude remaining constant. These systems used "dedicated" weights or the weights of other systems. Dedicated-weight materials should be as dense as possible since lever arms are relatively short on submersibles and since space is at a premium. Mercury, lead and steel are used. Mercury is pumped between fore and aft spherical trim tanks, whereas lead and steel weights are moved longitudinally on tracks. Weight of other systems almost invariably involves that of the battery pack, which is also moved on tracks. Use of the battery pack for this purpose saves weight, but often results in increased space requirements in allowing for the pack's fore and aft movement.

The capability of producing compensating or trim-creating moments, as seen, may be incorporated into the design of variable ballast-displacement systems or delegated to a trim system. Compensating moment requirements will be reviewed presently in connection with a discussion of equilibrium polygons. Trim-creating requirements of these systems are given by the expression

$$\omega l = W\overline{BG} \tan \alpha \qquad (30)$$

where

ωl = trim-creating moment, equal to the trimming weight or displacement times the longitudinal distance it is moved,
W = total submerged weight,
$\overline{BG}$ = submerged stability, and
α = trim angle specified in the submersible's performance requirements.

Maximum angles in past designs have varied from about 10 to 30 deg by the bow and stern, although this range has been exceeded in a few instances. Large angles of about 30 deg are useful if "down" and "up" angles are used in descent and ascent, particularly for deep operating depths, these angles orienting the submersible so as to reduce drag and nonproductive time in transit to/from the seafloor. Trim for this purpose may be provided by the regular buoyancy-trim control system or by the descent/ascent systems. Reference [3] addresses this subject in connection with the use of trim, descent/ascent ballast, and thrust forces.

It is seen from equation (30) that the magnitude of $\overline{BG}$ is an important factor in determining trim moment requirements—the higher the $\overline{BG}$, the larger the ωl for a given α. In turn, these larger trim moments require larger and heavier system components, consuming more power and energy in operation. Consequently, $\overline{BG}$ should be no higher than is necessary for adequate stability, unless performance requirements dictate the necessity for a high degree of resistance to static and dynamic inclinations of trim/pitch and list/roll.

Design considerations concerning compensation to retain the criteria of neutral buoyancy and zero

trim depend on the nature of the compensation requirements. For example, a submersible may have only one item of variable load at a fixed position requiring compensation—say, a payload of rock specimens contained in a basket at the bow. In this case, buoyancy and moment compensation might be made using an object recovery tank, sized and located to provide compensation for a range of specimen loads, and a relatively simple ballast or buoyancy system to compensate for changes in displacement. On the other hand, if the submersible design must accommodate several items of weight and load that can vary both in magnitude or position or both during the dive, it is usually worthwhile to investigate a number of *equilibrium conditions* and to base the design of the buoyancy-trim control systems on an *equilibrium polygon*. A discussion of the underlying hydrostatic principles for these conditions and the polygon follows.

Equilibrium conditions represent specific sets of weight-displacement-longitudinal moment summaries. These summaries are composed of different combinations of variable weights/loads, displacements, and their longitudinal moments, which are balanced by the variable ballast-displacement system alone or in conjunction with a trim system. A basic equilibrium condition can be established initially. This is a design condition in which the submersible has just submerged from its surface-diving trim condition, in "standard" seawater of 10.05 kN/m^3 (64.0 lb/ft^3), by the complete flooding of its main ballast tanks with this standard water. All variable weights/loads, variable ballast-displacement, trim weights, if used, and their moments are those associated with the surface-diving trim condition in standard seawater. It can be imagined that descent ballast, if used, is added to or superimposed on the basic equilibrium condition.

It is important that the status of the buoyancy-trim control system in the basic equilibrium conditions be such that the system can perform all duties assigned to it. For example, suppose this system is composed entirely of a variable ballast system, consisting of fore and aft tanks, which has both compensating and trim-creating duties. For compensation, there should be sufficient ballast water on board in this condition available for discharge to compensate for so-called "heavy" equilibrium conditions, a term to be defined shortly. If the system is also required to trim the vehicle to some non-zero fore and aft angles in these and other conditions, additional ballast for this purpose must be on board since the variable tanks are essentially emptied of "compensating" water in "heavy" conditions. The magnitude and distribution of this "trim-creating" ballast between the two tanks, separated by a distance l, must be such that the required fore and aft trim angles are obtained using equation (30).

Other equilibrium conditions are formulated based on realistic assumptions concerning variable weight/load, displacement and longitudinal variations from the basic condition which the submersible may encounter. As a minimum, the designer would do well to investigate the following:

1. Light condition: the lightest variable weight/load feasible combined with maximum displacement anticipated results in a maximum of positive buoyancy requiring a maximum of variable ballast or a minimum of variable displacement to retain neutral buoyancy.
2. Heavy condition: the heaviest variable weight/load feasible combined with minimum displacement anticipated results in a maximum of negative buoyancy requiring a minimum of variable ballast or a maximum of variable displacement to retain neutral buoyancy.
3. Heavy forward condition: a group of variable weight/load items so located longitudinally that their longitudinal moments result in a maximum forward, down-by-the-bow, trimming moment requiring maximum aft or forward compensating moments by the variable ballast/trim or variable displacement systems, respectively, to retain zero trim.
4. Heavy aft condition: a group of variable weight/load items so located longitudinally that their longitudinal moments result in a maximum aft, down-by-the stern, trimming moment requiring maximum forward or aft compensating moments by the variable ballast/trim or variable displacement systems, respectively, to retain zero trim.

Thus, it is evident that equilibrium Conditions 1 and 2 establish the extremes in positive/negative buoyancy compensating requirements for these systems, whereas Conditions 3 and 4 establish extremes in their fore and aft moment compensating requirements.

A generalized discussion of equilibrium condition summaries can become involved and confusing due to the wide variety of buoyancy-trim control systems in use. Consequently, the discussion will be based on two examples: Example A, based on a variable ballast system consisting of fore trim tanks (FTT) and aft trim tanks (ATT), which compensate for both positive/negative buoyancy and non-zero trim, and Example B, based on a variable displacement system, compensating for positive/negative buoyancy and a mercury trim system compensating for non-zero trim. The summaries for buoyancy-trim control systems A and B are shown in Fig. 34. A brief description of summary items follows.

—A,B1. Fixed weights/loads are the weight of all items of the weight and load groups comprising the submersible which remain fixed in magnitude and longitudinal location for all equilibrium conditions. Examples include Condition A weights, except compensating system liquid weights, and fixed load items such as crew, non-crew, and their effects. If dedicated ascent ballast is used in a design, it may also be included in this item since it is constant for all equilibrium conditions.

—A,B2. Variable weights/loads are all weights and loads which are presumed to vary in magnitude

CONDITION ____ @ γ = ____

No.	Item	Wt. or Disp.	Fwd. Mom.	Aft Mom.
1	Fixed Wts./Loads			
2	Variable Wts./Loads			
3	Main Ballast @ γ = ____			
4	Variable Ballast			
5	Total Weight			
6	Total Disp. @ γ = ____			

System A—Variable ballast system consisting of fwd. and aft trim tanks which compensate for both positive/negative buoyancy and non-zero trim. γ = 63.6 lb/ft^3 (Heavy), 64.3 lb/ft^3 (Light), 64.0 lb/ft^3 (other conditions).

CONDITION ____ @ γ = ____

No.	Item	Wt. or Disp.	Fwd. Mom.	Aft Mom.
1	Fixed Wts./Loads			
2	Variable Wts./Loads			
3	Main Ballast @ γ = ____			
4	Trim Moment	—		
5	Total Weight			
6	Primary Disp. @ γ = ____			
7	Variable Disp. @ γ = ____		—	—
8	Total Disp. @ γ = ____			

System B—Variable displacement system, compensating for positive/negative buoyancy, and trim system compensating for non-zero trim. γ = 63.6 lb/ft^3 (Heavy), 64.3 lb/ft^3 (Light), 64.0 lb/ft^3 (other conditions).

Fig. 34 Equilibrium condition summaries

or longitudinal location or both during a dive in such a manner as to produce the equilibrium conditions discussed above as well as others which may be investigated. Examples are compensating system liquid weights, payloads, supplies, fuels, and potable water.

—A,B3. Main ballast is the volume of main ballast water times its specific weight which is assumed to have the same value as that used in calculating displacement for a specific equilibrium condition.

—A4. Variable ballast is ballast of such magnitude and longitudinal center of gravity location that it compensates for both positive/negative buoyancy and non-zero trim, thereby establishing equality between the total weight and displacement and between their longitudinal moments. If descent ballast is used, any remaining on board at end of descent is considered to be part of this ballast.

—B4. Trim moment is the moment of trim system weight with its center of gravity located so as to compensate for the non-zero trim, thus establishing equality between the longitudinal moments of total weight and displacement. Only the compensating moment is recorded for this item, the trim weight being included in Item 1 since it remains constant for all equilibrium conditions.

—A,B5. Total weight and its moment are the algebraic sums of weights and moments of A1,2,3,4 or B1,2,3,4, which must equal the total displacement and its moment for neutral buoyancy and zero trim to exist.

—A6. Total displacement is the displacement, and its longitudinal moment, associated with a particular equilibrium condition. As noted, the maximum anticipated displacement is associated with the "light" condition and the minimum anticipated displacement with the "heavy" condition, thus generating extreme requirements for the buoyancy-trim control system.

—B6. Primary displacement is the total displacement excluding the variable displacement. Otherwise, Statement A6 also pertains to this item.

—B7. Variable displacement is displacement of such a magnitude and center location at the longitudinal moment reference line that it compensates for positive/negative buoyancy but creates no fore or aft moment, thereby establishing equality between the total displacement and weight while relying on the trim system to provide for moment compensation.

—B8. Total displacement and its moment are the algebraic sums of the displacements and moments of Items B6 and B7 which must equal the total weight and its moment for neutral buoyancy and zero trim.

The tabulation of equilibrium condition summaries permits the calculation of coordinates of *equilibrium points* on the equilibrium polygon. The ordinate of these points is the magnitude of the variable ballast or displacement required to achieve neutral buoyancy, whereas the abscissa is the longitudinal moment of variable ballast, variable displacement, or trim weight required to achieve zero trim. In Fig. 34, these coordinates are given in the data of Item A4, for System A, and in B4 and B7 for System B.

The equilibrium polygon is a graphical representation of the ability of the buoyancy-trim control system to meet compensating requirements for all equilibrium conditions investigated. This ability is demonstrated by the polygon's enclosing all equilibrium points. Construction of the polygon is illustrated using example buoyancy-trim control Systems A and B. The following steps outline the procedure for System A.

1. Coordinates are drawn as shown in Fig. 35*a*. The abscissa is fore-and-aft moments with reference to some convenient point, say, at mid-length of the submersible designated as Point 0, and the ordinate is ballast.
2. Equilibrium points are plotted on these coordinates—the basic equilibrium condition point (BP), light (L), heavy (H), heavy forward (HF) and heavy aft (HA)—as well as for other conditions which may be investigated.

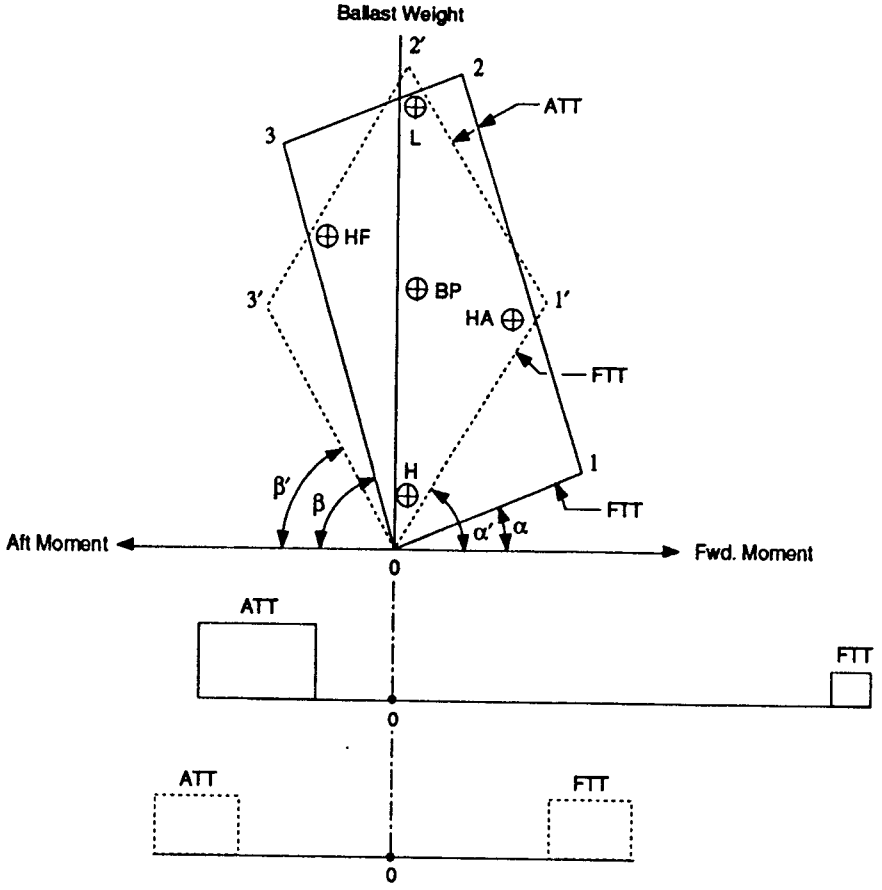


Fig 35(*a*) Equilibrium polygons for variable ballast systems using forward and aft variable ballast tanks (FTT and ATT)

3. A polygon is drawn so as to enclose the equilibrium points in the most expeditious manner feasible. In this illustration, the four-sided figure is formed by two sets of parallel lines of equal lengths with one set being labelled FTT and the other ATT, as indicated in the figure. All lines are straight, which assumes that the longitudinal center of gravity of the ballast water remains constant as the tanks are filled or emptied. Starting with both tanks empty at 0, the status of the tanks at other corner points is as follows—FTT filled at 1, FTT and ATT filled at 2, FTT empty and ATT filled at 3, and both tanks again empty at 0, closing the polygon. Variable ballast is assumed to be "standard sea water."

The design variables of *tank size* and *longitudinal location* are involved in constructing the polygon, or, conversely, the shape of the polygon finally selected determines these variables. Longitudinal locations determine the slopes of the polygon's sides while tank sizes and line slopes determine their lengths. Numerous polygons can enclose a given set of equilibrium points; two possibilities are shown in Fig. 35*a* by solid and dashed lines. The solid-line polygon is generated by a small FTT located well forward of 0, as indicated by the small slope or angle α, and a large ATT located a short distance aft of 0 as shown by the large slope or angle β. The dashed-line polygon represents a more balanced design in terms of approximately equal sizes of FTT and ATT at nearly equal distances fore and aft of 0; here, α' being nearly equal to β'.

In summary, it can be said that the compensating function alone can most efficiently be performed if the equilibrium polygon just encloses the extreme equilibrium points, thereby avoiding unnecessarily large tanks and longitudinal moments. Most likely, the final polygon will reflect the necessity of compromising with other design considerations influencing general arrangements of the submersible. If the system is charged with trim moment-creating as well as compensating requirements, the two tanks must accommodate trim ballast as well as compensating ballast and be longitudinally separated so as to generate adequate trimming moments in shifting of ballast between them. Consequently, the polygon

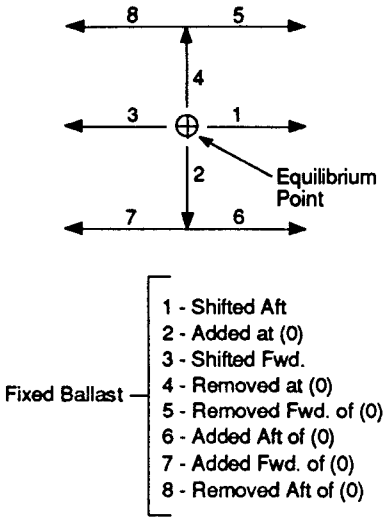


Fig. 35(*b*) Altering fixed ballast to shift equilibrium points

associated with compensation only will most likely be expanded to meet these dual requirements.

There may be instances when it is necessary to shift the equilibrium points in order to enclose all of them within a given equilibrium polygon; these instances sometimes occur in the advanced design stages or after the submersible is constructed. Shifts are made by altering the fixed ballast or buoyancy. Figure 35*b* illustrates this procedure of altering fixed ballast. It should be emphasized that a specific alteration causes all equilibrium points to shift in the same manner. Thus, care must be taken that shifting one point into the polygon does not result in shifting another point out.

The procedure for constructing the equilibrium polygon for System B is outlined as follows (see Fig. 36 for the polygon):

1. Coordinates are drawn in a manner similar to those of System A except that the ordinate is displacement.
2. Equilibrium points are plotted on these coordinates, two positions being the reverse of those on the System A coordinates—that is, L and H are reversed, while the HF and HA positions remain the same.
3. A polygon is drawn to enclose the equilibrium points in the most expeditious manner feasible; in this figure, the polygon is a rectangle. At 0, the fore and aft trim tanks are partially filled with mercury so as to create no moment with reference to 0 and the variable displacement bladder, or bellows, is fully deflated. The status of the trim tanks and bladder at other points on the polygon is as follows: fore trim full/aft trim empty and bladder deflated at 1, fore trim full/aft trim empty and bladder fully inflated with oil at 2, fore trim empty/aft trim full and bladder fully inflated at 3, fore trim empty/aft trim full and bladder fully deflated at 4, and fore/aft trim partially filled and bladder fully deflated to return to 0.

As has been noted, the mercury trim system could be replaced with a movable battery pack or weight with equivalent results.

Trim system design factors involved in constructing the polygon include weight of mercury ω and the longitudinal distance between the fore and aft trim tanks l. This system must be capable of creating fore and aft compensating moments, $(\omega l)_F$ and $(\omega l)_A$, equal to polygon dimensions 0–1 and 0–4, respectively. If the system has the added responsibility for creating trim, its moment capacity must be increased accordingly. Constraints are placed on ω, or sizes of the tanks, and l by other general arrangement considerations. It suffices to say that l should be as large as feasible in view of these considerations. Variable displacement design factors involved in constructing the polygon include the amount of oil required to fully inflate the bladder, and its longitudinal location. The volume of oil, or volume of the fully inflated bladder, times the lowest specific weight of sea water anticipated must equal the displacement dimension 1–2 on the polygon. Knowing the volume of oil required, or fully inflated volume of the bladder, also determines the size of the pressure-resisting, oil-storage sphere. Again, if the variable displacement system is also expected to create positive/negative buoyancy, its capacity would have to be increased accordingly. This requirement, together with added requirement or the trim system, expands the polygon beyond boundaries which just enclose the equilibrium points. The longitudinal centroid of the bladder should be at, or very near, the reference point for longitudinal moments, 0, so that no, or very small, moments are created.

3.7.3 Descent/Ascent Systems

These systems have the special function of facilitating descent/ascent by causing departures from the design-condition criterion of neutral buoyancy and, in some designs, also from the criteria of zero trim and list depending on the desired attitude of the submersible during descent/ascent and path—vertical, inclined, or spiral. As seen, the forces involved in these systems may come from a variety of sources or combination of sources, including dedicated, special-purpose variable ballast systems consisting of descent/ascent ballast; a variable ballast or displacement system serving this as well as its other functions; a main ballast system serving this function for ascent; and heavy, releasable weights of other systems serving this function for emergency ascents. Descent/ascent trim and list, to the extent used, are provided by trimming/listing moments caused by the longitudinal/transverse locations of these forces or

by trim/list systems. As noted, these forces may be created by varying displacement or ballast. Variable displacement and sea water ballast are restricted to shallow and mid-depth submersibles[23] because of the difficulty of pumping or blowing against high hydrostatic heads, whereas the use of ballast weights or pellets is not depth-limited.

A discussion of descent/ascent system design considerations begins with the submersible just below the surface in the basic equilibrium condition with the main ballast tanks full, variable ballast or displacement and trim system weights adjusted for this condition, and ascent ballast, if used, on board. Imagine for this purpose that a downward motivating force is now added to the submersible to initiate the dive, this force being negative buoyancy, created by the descent system, thrust, or both. If the descent path is assumed to be vertical, the equation of motion is

$$\sum F_{z_0} = m_v \ddot{z}_0 \qquad (31)$$

where

$\sum F_{z_0}$ = the algebraic sum of the forces acting on the submersible in the z_0, or vertical direction,

m_v = virtual mass composed of the masses of the submersible and adjacent water accelerated by its motion, and

$\ddot{z}_0$ = downward acceleration/deceleration.

The sum of the z_0-direction forces in general is

$$\sum F_{z_0} = F_B \pm T - F_D \qquad (31a)$$

where

F_B = negative buoyancy, in this case,

T = thrust, and

F_D = drag force.

The thrust force is usually absent in the case of untethered submersibles to conserve energy, although it may be used sparingly to help control rate of descent or for deceleration when the submersible approaches the bottom or desired depth. On the other hand, this force is often used for tether submersibles, for which the energy supply is virtually unlimited, in conjunction with negative, neutral, or even positive buoyancy if the "fail-safe" design philosophy obtains. Assuming here that the thrust force is absent, equation (31*a*) becomes

$$\sum F_{z_0} = F_B - F_D \qquad (31b)$$

which may be expressed in terms of the algebraic sum of the forces acting on the submersible at any depth h as

$$\sum F_{z_0 h} = F_d \pm \delta F_B \pm (+\delta w \pm \delta \Delta) - F_{Dh} \qquad (31c)$$

where signs are positive if force is in the direction of motion. In this equation:

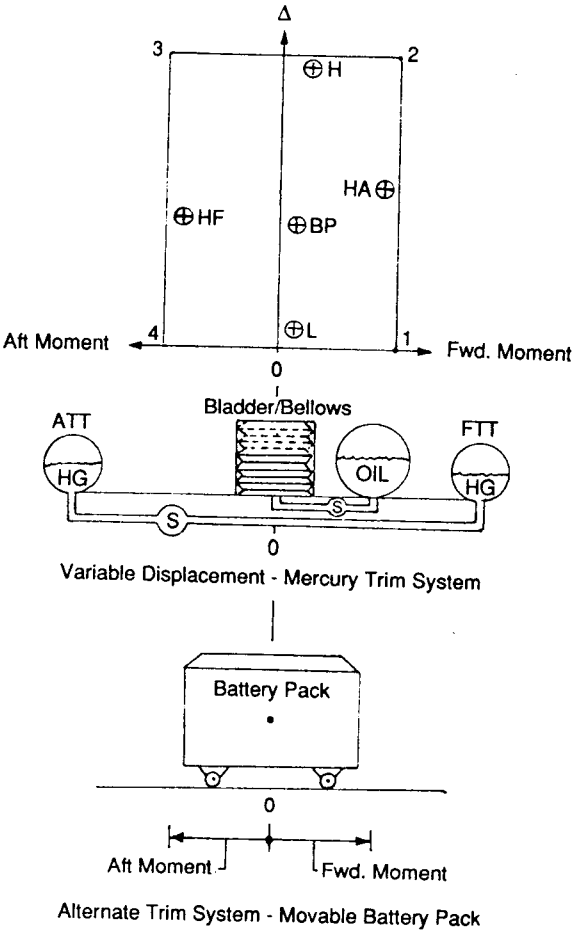


Fig. 36 Equilibrium polygon for variable displacement-trim systems

F_d = descent motivating force added to the basic equilibrium condition at beginning of dive, which equals initial negative buoyancy,

δF_B = change in negative buoyancy between surface and depth h caused by adding/discharging sea water, releasing descent weights or pellets or increasing/decreasing variable displacement by inflating/deflating oil-filled bladders. This change occurs continuously over the depth range (as in the controlled release of pellets or the admission of sea water), at intermediate depths (as in the release of individual descent weights one at a time), or at or near the desired operating depth (as in the release of all descent weight at once),

δw = change in the basic equilibrium condition weight between the surface and depth h assumed here to be entirely due to change in the weights of compensating system liquids because of compression

[23]Reference [3] describes a pressurized ballast system which extends the use of seawater ballast to deep-depth designs.

and temperature reduction. These weights increase with an increase in depth (and decrease with a decrease in depth during ascent),

$\delta\Delta$ = change in the basic equilibrium condition displacement between surface and depth h due to changes in volume of displacement $\overline{V}$ and specific weight of sea water γ. The displacement either increases or decreases with an increase in depth (with the reverse situation occurring with a decrease in depth during ascent) depending on whether the product $\overline{V} \times \gamma$ increases/decreases in response to changes in temperature, salinity, and pressure with change in depth. $\delta\Delta$ can also be zero if compression and γ changes offset each other.

F_{Dh} = drag force at depth h.

If the rate of descent (or ascent) is reasonably constant, the last term may be written as

$$F_D = C_{Dd} \times \frac{\rho}{2} \times Ap \times V^2 \qquad (31d)$$

where

C_{Dd} = drag coefficient[24] for descent, which is usually somewhat different from the drag coefficient for ascent, C_{Da}, because of the lack of shape symmetry about the submersible's x–y plane and, often, different descent/ascent trim angles,

ρ = mass density of sea water,

Ap = maximum projected area of the submersible normal to the direction of flow, and

V = descent velocity.

At or near the desired operating depths, neutral buoyancy must be reestablished by reducing negative buoyancy to zero, which requires that

$$Fd - \delta F_B + (+\delta w \pm \delta\Delta) = 0 \qquad (31e)$$

if the parenthesized quantity increases in the direction of motion as depth increases and

$$Fd - \delta F_B - (+\delta w \pm \delta\Delta) = 0 \qquad (31f)$$

if it increases in the opposite direction to motion as depth increases. Satisfying equation (31*e*) requires, of course, that δF_B equal Fd plus $+\delta w \pm \delta\Delta$. If, for example, descent and variable ballast are used in a design, all the descent ballast must be released as well as an amount of variable ballast equal to $+\delta w \pm \delta\Delta$, which is on board in the equilibrium condition. Quite another situation exists in satisfying equation (31*f*). Using the above example descent system, δF_B might be zero with the descent ballast F_d equalling $+\delta w \pm \delta\Delta$, a condition in which the submersible would be decelerating throughout descent until coming to rest at the desired operating depth. To decrease time of descent, the vehicle could be "ballasted heavy"; that is, the amount of F_d added to the basic equilibrium condition could be made greater than $+\delta w \pm \delta\Delta$ at operating depth. This excess ballast, equal to δF_B at operating depth, would be released at or near this depth to regain neutral buoyancy.

Ascent is initiated by adding an upward motivating force to the submersible in the design condition at operating depth, thereby creating positive buoyancy. Trim and list for ascent, if desired, may be obtained in a manner similar to that previously described. Ascent equations similar to equations (31) through (31*d*) may be written with the algebraic sum of the vertical forces acting on the submersible at any altitude a above the operating depth being

$$\sum F_{z_0 a} = F_u \pm \delta F_B \pm (+\delta w \pm \delta\Delta) - F_{Da} \qquad (31g)$$

where F_u is the initial upward motivating force at the beginning of ascent, F_{Da} is the drag force at a, and the terms δF_B, δw, and $\delta\Delta$ are similar to those in equation (31*c*) except that they note changes in buoyancy, weight, and displacement between the operating depth and altitude a. Ascent terminates when the submersible breaks the surface. Upon complete dewatering of the main ballast tanks, it may or may not float at the normal diving trim waterline (*NDTWL*), depending on the type of ascent system used. For example, if ascent ballast is used, the vehicle would float at a waterline below the *NDTWL* since ascent ballast, now released, was on board in the normal diving trim condition. The submersible would have to be reballasted so that it floated at the *NDTWL*, or *NDTWL-H*[25] before it could dive again.

Major descent/ascent system design considerations involve the terms F_d, F_u, and δF_B in equations (31*c*) and (31*g*), with F_d and F_u determining average descent/ascent velocity and δF_B whether motion is accelerated, decelerated, or at constant speed. As an example, suppose constant speed motion is desired—excluding initial acceleration or bottom/surface approach decelerations. Then

$$\delta F_B = (+\delta w \pm \delta\Delta) \qquad (31h)$$

for all values of h or a in the constant speed portion of descent/ascent. Assuming descent/ascent ballast is used, this stipulation would require continuous adjustment of this ballast such that

$$\frac{d}{dh}(\delta F_B) = \frac{d}{dh}(+\delta w \pm \delta\Delta) \qquad (31i)$$

[24] C_{Dd} (or C_{Da}) is associated with pressure drag based on the projected area. Drag coefficients associated with friction drag are based on wetted surface area, A_{ws}, instead of Ap. Otherwise, the equation is the same for both types of drag or resistance.

[25] Descent ballast, if used, is of course also on board the submersible in the diving trim condition, causing it to float at the *NDTWL-H* as previously discussed. As noted, it is only to facilitate discussion of the descent system that descent ballast is "imagined" to be added after the submersible has just submerged and is in the basic equilibrium condition.

During constant speed motion

$$F_d \text{ or } F_u = C_D \times \frac{\delta}{2} \times Ap \times V^2 \qquad (31j)$$

where C_D is either C_{Dd} or C_{Du}. For given velocities, set forth in the performance requirements, and drag coefficients, values of F_d and F_u can be calculated. In this regard, a typical range of velocities is 30.5 to 45.7 m/min (100 to 150 ft/min), with deeper operating depths associated with higher velocities to reduce the ratio of descent/ascent time to total submerged time to the extent possible.

Descent/ascent systems must be flexible in the sense that their capabilities must permit safe and efficient operation under the variety of water column conditions which occur in anticipated dive areas as set forth in the mission requirements. In this regard, the nature of the water column in a particular dive area should be known prior to the dive to avoid "surprises" such as unexpected sharp thermoclines which may be difficult or impossible to penetrate.

3.7.4 *List System*

This system, where used, has the primary function of purposefully causing departure from the design condition criterion of zero list. Performing this function requires that the system's list moment-creating capacity provide the maximum list specified by the performance requirements. Usually these maximum angles vary only from a few to about 10 deg port and starboard. List moment-creating requirements are given by the expression

$$wt = W\overline{BG} \tan \theta \qquad (32)$$

where wt is the moment, equal to the list weight times the transverse distance moved from its position in the basic equilibrium condition, and θ is the specified maximum list angle. Again, as in equation (30), the effect of $\overline{BG}$ on list-system design is apparent. A mercury list system is generally used.

3.7.5 *Main Ballast System*

This system has the function of creating negative/positive buoyancy for the purpose of destroying reserve displacement on diving, restoring it on surfacing, and, in some designs, creating an upward motivating force to initiate ascent. Consequently, as has been seen, its design is directly related to reserve displacement requirements regarding freeboard, the height of the access hatch above the waterline, and the provision for some small measure of protection against flooding. Design factors considered at this point include the volume of main ballast water and the relation between the longitudinal centers of main ballast water and reserve displacement.

The volume of main ballast water V_{MB} is given by the expression

$$V_{MB} = V_{MBT} - V_{RW} - V_{STR} - V_I \qquad (33)$$

which presumes that residual water, as previously defined, exists to a certain level in the main ballast tanks. In equation (33), V_{MBT} is the gross volume of the main ballast tanks, V_{RW} is the volume of residual water, and V_{STR} and V_I are the water-excluding volumes of structure in the tank and any items which may be housed therein. V_{STR} may vary from one to three percent of V_{MBT}. V_I is usually zero for small submersibles, whereas it may be an appreciable percentage of V_{MBT} for large submersibles, as well as submarines, in which these tanks often contain high-pressure air flasks and fixed ballast. Since the specific weights of main ballast water and water in which the submersible is floating are equal, knowing the volume of reserve displacement requirements permits calculation of V_{MB} and V_{MBT}. In this regard,

$$\frac{V_{RD}}{\overline{V}_N} = \frac{V_{MB}}{\overline{V}_N} = f \qquad (34)$$

where $\overline{V}_N$ is the volume of displacement in the normal diving trim condition and f is a factor varying from about 0.05 to 0.12. This factor depends on the nature of the submersible's surface operations associated with launch/retrieval and in-water handling. f tends to increase as these operations become more extensive.

The relation between the longitudinal centers of main ballast water and reserve displacement depend on whether or not the submersible is to have zero trim or trim by the stern in the normal diving trim condition. The latter option is sometimes exercised to increase the propeller's submergence and improve surface propulsion. If zero trim on the surface is desired, these longitudinal centers must be equal so that complete flooding of the main ballast tanks will cause the submersible to just submerge in the basic equilibrium condition of neutral buoyancy and zero trim. On the other hand, if trim by the stern is desired, the longitudinal center of main ballast water must be a meters (feet) forward of the longitudinal center of reserve displacement to achieve the same results. In English units,

$$a = \frac{t \times MTI}{L_{MB}} \qquad (35)$$

where

t = desired trim in inches,
MTI = the moment to change trim one inch in ft·lb/in., and
L_{MB} = load of main ballast water in lb.

Main ballast tanks are almost always located outside the submersible's pressure hull. Their dimensions and internal subdivision, if used, should be such as to minimize free surface effects during diving/surfacing to the extent feasible.

4. Hydromechanical Principles—Hydrodynamics

4.1 Overview

Hydrodynamic forces and moments act on the submersible as a result of its motion through the water. The forces and moments are a function of the hull size, shape, and velocity **U** through the water. The relationships between the forces, moments, linear velocities, and angular velocities can be very complex. This section will briefly overview the equations of motion that describe these relationships and then the typical naval architectural design approaches to these problems. Following these will be discussions of the trade-offs that are made between speed, endurance, hull shape, and vehicle drag, followed by an introduction to propeller design. The section will conclude with a discussion of submersible hydrodynamic stability and control.

The forces acting on a submersible can be equated to time rate of change of momentum of the submersible and the entrained water moving with the submersible:

$$\mathbf{F} = \frac{d}{dt}(\textbf{Momentum}) = \frac{d}{dt}(\text{mass } \mathbf{U}_g) \tag{36}$$

$\mathbf{U}_g$ is a velocity vector relative to the center of gravity defined as:

$$\mathbf{U}_g = \langle u, v, w \rangle \tag{37}$$

where u, v, and w are the surge, sway, and heave velocities, respectively, and are shown in Fig. 37. (The origin of this vector is typically at the submersible center of gravity and vertically on the centerline.) The moments are equal to the time rate of change of angular momentum:

$$\mathbf{M} = \frac{d}{dt}(\textbf{Angular Momentum}) = \frac{d}{dt}(\text{mass } \boldsymbol{\Omega}) \tag{38}$$

$$\boldsymbol{\Omega} = \langle p, q, r \rangle \tag{39}$$

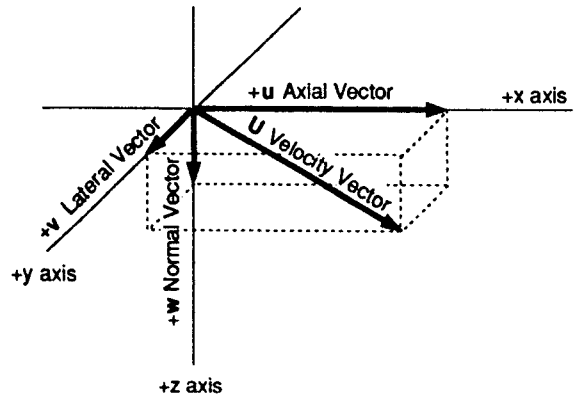


Fig. 37 Submersible coordinate reference system

where p, q, and r are the roll, pitch and yaw angular components, respectively. A vector diagram similar to Fig. 37 can also be drawn for the moments acting on the submersible.

Ignoring the ballast tanks for the general case and assuming no change in entrained water, the mass of the submersible can be considered constant and the equations reduced to

$$\mathbf{F} = m\frac{d\mathbf{U}_g}{dt} \tag{40}$$

$$\mathbf{M} = I\frac{d\boldsymbol{\Omega}}{dt} \tag{41}$$

These two equations can then be made relative to the earth axis system by the use of the vector $\mathbf{R}_g$. This vector extends from the center of the earth to the center of gravity. The force equation is shown below:

$$\mathbf{U}_g = \mathbf{U} + \boldsymbol{\Omega} \times \mathbf{R}_g \tag{42}$$

$$\mathbf{F} = m\frac{d}{dt}(\mathbf{U} + \boldsymbol{\Omega} \times \mathbf{R}_g) \tag{43}$$

Each of the above vectors is broken down into its components:

$$\mathbf{U} = \mathbf{i}u + \mathbf{j}v + \mathbf{k}w$$

$$\boldsymbol{\Omega} = \mathbf{i}p + \mathbf{j}q + \mathbf{k}r$$

$$\mathbf{R}_g = \mathbf{i}X_g + \mathbf{j}Y_g + \mathbf{k}Z_g$$

$$\mathbf{F} = \mathbf{i}X + \mathbf{j}Y + \mathbf{k}Z$$

Substitution and differentiation results in the three fundamental force equations:

$$X = m[\dot{u} + qw - rv - X_g(q^2 + r^2) + Y_g(pq - \dot{r}) + Z_g(pr + \dot{q})] \tag{44}$$

$$Y = m[\dot{v} + ru - pw - Y_g(r^2 + p^2) + Z_g(qr - \dot{p}) + X_g(qp + \dot{r})] \tag{45}$$

$$Z = m[\dot{w} + pv - qu - Z_g(p^2 + q^2) + X_g(rp - \dot{q}) \tag{46}$$

A similar development follows for the moment equations which provides the totals for roll, pitch, and yaw (K, M, and N, respectively) as shown below.

$$K = I_x\dot{p} + (I_z - I_y)qr + m[Y_g(\dot{w} + pv - qu) - Z_g(\dot{v} + ru - pw) + X_gY_g(pr - \dot{q}) - X_gZ_g(pq + \dot{r}) + Y_gZ_g(r^2 - q^2)] \tag{47}$$

$$M = I_y\dot{q} + (I_x - I_z)rp + m[Z_g(\dot{u} + qw - rv) - X_g(\dot{w} + pv - qu) + Y_gZ_g(qp - \dot{r}) - Y_gX_g(qr + \dot{p}) + X_gZ_g(p^2 - r^2)] \tag{48}$$

$$N = I_z\dot{r} + (I_y - I_x)pq + m[X_g(\dot{v} + ru - pw) - Y_g(\dot{u} + qw - rv) + Z_gX_g(rq - \dot{p}) - Z_gY_g(pr + \dot{q}) + X_gY_g(q^2 - p^2)] \quad (49)$$

The summations of the forces and the moments in the above equations are shown as simple expressions; they are, in fact, much more complicated and complex. The nature of these forces and moments is both hydrostatic and hydrodynamic. They can be listed as a function of several variables as

$$F = f(x_0, y_0, z_0, \phi, \theta, \psi, u, v, w, p, q, r, \dot{u}, \dot{v}, \dot{w}, \dot{p}, \dot{q}, \dot{r}, n, \dot{n}, \ddot{n}, \delta, \dot{\delta}, \ddot{\delta}, \ldots, \text{excitations, restraints})$$

where n represents the rotation of all the propellers and thrusters and δ represents the deflection of all of the control surfaces. The other variables will be discussed below. There are numerous conventions used to express the forces and moments, and the designer must be careful to be consistent in documentation. The approach taken here is taken from reference [5].

The terms x_0, y_0, z_0, ϕ, θ, and ψ deal with the orientation of the submersible in space and are determined by hydrostatics. The hydrostatic forces are described in detail in Section 3 of this chapter.

The hydrodynamic forces result from the terms u, v, w, p, q, r, $\dot{u}$, $\dot{v}$, $\dot{w}$, $\dot{p}$, $\dot{q}$, and $\dot{r}$. They are not easily handled, and in order to be able to cope with a mathematical solution, it is necessary to expand the above expression by Taylor's expansion theorem. Use of the Taylor expansion is limited to analytical functions, and it can be assumed that the hydrodynamic forces are of this type for the linear and third-order terms. For situations involving all crossflow drag (where u is zero) or when considering hovering, the second-order terms of the form u/u and v/v, etc., must be used. The resulting hydrodynamic force equation can be summarized as follows:

Surge:

$$\sum X_{hyd} = X_{\dot{u}}\dot{u} + X_{qw}qw + X_{rv}rv + X_{u/u}u/u + X_{rr}r^2 + X_{qq}q^2 + X_{rp}rp + X_{app} + X_{prop} + X_{thr} + X_{bal} \quad (50)$$

Sway:

$$\sum Y_{hyd} = Y_{\dot{v}}\dot{v} + Z_{\dot{q}}\dot{q} + Y_{\dot{p}}\dot{p} + Y_{r/u}r/u + Y_{pw}pw + Y_{qp}qp + Y_{v/w}v/w + Y_{p/u}p/u + Y_{v/u}v/u + Y_{app} + Y_{prop} + Y_{thr} + Y_{bal} \quad (51)$$

Heave:

$$\sum Z_{hyd} = Z_{\dot{w}}\dot{w} + Z_qq + Z_{pp}p^2 + Z_{p/v}p/v + Z_{pr}pr + Z_{rr}r^2 + Z_{u/u}u/u + Z_{w/u}w/u + Z_{q/u}q/u + Z_{app} + Z_{prop} + Z_{thr} + Z_{bal} \quad (52)$$

Roll:

$$\sum K_{hyd} = K_{\dot{p}}\dot{p} + K_{\dot{v}}\dot{v} + K_{v/w}v/w + K_{rw}rw + K_{qv}qv + K_{pw}pw + K_{v/u}v/u + K_{v/v}v/v + K_{p/u}p/u + K_{r/u}r/u + K_{p/p}p/p + K_{app} + K_{prop} + K_{thr} + K_{bal} \quad (53)$$

Pitch:

$$\sum M_{hyd} = M_{\dot{q}}\dot{q} + M_{\dot{w}}\dot{w} + M_{u/w}u/w + M_{pr}pr + M_{vp}vp + M_{u/u}u/u + M_{vv}v^2 + M_{rv}rv + M_{rr}r^2 + M_{q/u}q/u + M_{w/u}w/u + M_{rr}r^2 + M_{pp}p^2 + M_{app} + M_{prop} + M_{thr} + M_{bal} \quad (54)$$

Yaw:

$$\sum N_{hyd} = N_{\dot{r}}\dot{r} + N_{\dot{u}}\dot{u} + N_{pq}pq + N_{u/v}u/v + N_{r/u}r/u + N_{wp}wp + N_{rq}rq + N_{p/u}p/u + N_{w/r}w/r + N_{app} + N_{prop} + N_{thr} + N_{bal} \quad (55)$$

For a state of equilibrium to exist, the external forces and moments must equal those that are hydrodynamic in nature. The motion stability of a submersible deals with the response to small initial disturbances from an equilibrium condition to see if it will return to the original condition. It is necessary to consider only the linear terms of the Taylor expansion to obtain a satisfactory solution. The number of terms to be considered in this condition is such that they are rather easily handled. When an unstable submersible or one that is involved in violent maneuvers at either low or high speeds is considered, the nonlinear terms must usually be considered as well. These terms can become very numerous and can be reduced in number to some extent by considering the symmetry of the submersible and by discarding some of the less important terms. This leads to the concept of linear and nonlinear equations of motion.

The solution to the combined force, moment, linear momentum, and angular momentum equations can be used to predict the motion and path of a submersible under varying conditions. They can also be used to estimate the operating characteristics of a submersible. Although certainly a valuable design tool, the solution of these equations can be very time-consuming if done without the aid of computer programs and a database for estimating the numerous coefficients involved.

4.2 Resistance and Propulsion

One of the requirements of any submersible study is to predict the amount of power required to move the hull at a given speed, or conversely, to predict the speed obtained with a given amount of power. Traditionally, such predictions were made on the basis of steady-state conditions in level flight with no maneu-

vering, assumptions which greatly simplify the computations. In the equations of motion, all but the surge equation can be eliminated because the assumption of level flight and no maneuvering reduces all the other coefficients to zero.

The surge equation (50) is

$$\sum X_{hyd} = X_{\dot{u}}\dot{u} + X_{qw}qw + X_{rv}rv + X_{u/u}u/u + X_{rr}r^2 + X_{qq}q^2 + X_{rp}rp + X_{app} + X_{prop} + X_{thr} + X_{bal}$$

Since the only motion is assumed to be in the x direction, all other linear and rotary coefficients become zero. A simplified equation results:

$$\sum X_{hyd} = X_{u/u}u/u + X_{app} + X_{prop} + X_{thr} + X_{bal} \quad (56)$$

The above equation of motion is the basis behind the drag equation typically used in naval architecture. There is a difference in nomenclature due, in part, to the science of naval architecture predating the equations of motion. The drag, or resistance, of a submerged body moving along its longitudinal axis is given by

$$R_T = R_{BH} + R_{APP} \quad (57)$$

where

R_T = total resistance,
R_{BH} = bare hull resistance, and
R_{APP} = resistance of the appendages (such as the sail, control surfaces, etc.).

The power required to overcome this resistance is given by

$$EHP = \frac{R_T \times V}{550} \quad (58)$$

where EHP is the effective horsepower put into the water by the propeller and V is the speed of the submersible in feet per second.

4.2.1 *Bare Hull Resistance*

From basic fluid mechanics, the resistance of a body is predicted as

$$R_{BH} = \tfrac{1}{2}\rho A V^2 C_t \quad (59)$$

where

ρ = mass density of the operating fluid,
A = reference area of the submersible, usually the wetted area, the frontal area, or L^2,
V = velocity of the submersible, ft/s, and
C_t = nondimensional drag coefficient.

The above equation is applicable in surge, heave, and sway. The drag coefficient is nondimensional and is associated with a particular reference area. The designer must be careful to remain consistent with the use of area (that is, he or she should not use a drag coefficient based on frontal area with the submersible wetted area). The drag coefficient C_t can be broken down into four components:

$$C_t = C_f + \Delta C_f + C_r + C_w \quad (60)$$

where

C_f = frictional resistance coefficient,
ΔC_f = correlation allowance,
C_r = residual resistance coefficient, and
C_w = wave-making resistance coefficient.

The wave-making term C_w can be ignored when the submersible is operating at depths greater than five hull diameters. At slower speeds, typical of submersible operations, the frictional resistance dominates the bare hull resistance. This force is caused by the shearing of the fluid as it passes along the hull. The force is Reynolds-number dependent and can be estimated by using any of several equations based on model testing. The International Towing Tank Conference (ITTC) curve fit is given below.

$$C_f = \frac{0.075}{(\log_{10}\mathrm{Re} - 2)^2} \quad (61)$$

where Re is the Reynolds number based on hull length.

$$\mathrm{Re} = \frac{V \times L}{\nu} \quad (62)$$

where L is the characteristic length and ν is the kinematic viscosity of sea water. The kinematic viscosity is a function of the water temperature, pressure, and water salinity. Table 1 shows kinematic viscosity and density as a function of temperature for sea water.

Table 1 Kinematic viscosity versus temperature and density

Temperature, °F	Density, lb·s²/ft⁴	Kinematic Viscosity, ft²/s
32	1.9947	—
34	1.9946	—
36	1.9944	—
38	1.9942	—
40	1.9940	—
42	1.9937	1.6568
44	1.9934	1.6035
46	1.9931	1.5531
48	1.9928	1.5053
50	1.9924	1.4599
52	1.9921	1.4168
54	1.9917	1.3758
56	1.9912	1.2268
58	1.9908	1.2996
60	1.9903	1.2641
62	1.9898	1.2303
64	1.9893	1.1979
66	1.9888	1.1669
68	1.9882	1.1372
70	1.9876	1.1088
72	1.9870	1.0816
74	1.9864	1.0554
76	1.9858	1.0303
78	1.9851	1.0062
80	1.9844	0.9830

The ITTC curve fit was based on towing long very smooth planks in a model basin. However, because a submersible at sea will not maintain such a smooth finish, an additional resistance term is added—the correlation allowance ΔC_f. Typical values range from 0.0004 to 0.0009.

The residual resistance C_r represents the resistance due to the flow separation from the hull and the resulting adverse pressure gradient (also known as form drag). The residual resistance dominates the vehicle resistance at high speed. As will be discussed later, this portion of vehicle resistance is highly shape dependent.

The analytical prediction of C_r is very difficult due to the complexities of the physics of fluid separation. To overcome this, designers have conducted thousands of model tests and have measured the resistance of the models at various speeds. The frictional portion of the resistance was estimated using an ITTC-type equation and was subtracted from the total. The remaining resistance, called residual resistance, was assumed to represent the drag due to flow separation. (For surface ships, the residual resistance also includes the wave-making drag.) Over the years, tests have been conducted that varied a single hull parameter at a time to provide a "series" of hull forms. These series cover many of the possible combinations of length, beam, draft, speed, and hull coefficients. The designer interpolates between given cases to estimate the value of C_r for a new hull form. Unfortunately for the submersible designer, there have been relatively few tests of submersible hull forms compared to the number available for surface ships and submarines. This is not a severe limitation for slow-speed vehicles, but is a problem at higher speeds. Typical values of C_r based on wetted surface are given in Table 2.

4.2.2 *Appendage Resistance*

The appendage resistance can be a large percentage of the vehicle total, particularly if the vehicle is equipped with numerous lights, cameras, manipulators, etc. The drag of the propeller itself is not accounted for at this point, but the drag of the associated struts, exposed shafting, etc., should be estimated. As with the bare hull resistance, the appendage resistance is given by

$$R_{APP} = \tfrac{1}{2}\rho A V^2 C_t \tag{63}$$

Table 2 Typical values of C_r

Hull Form	$C_r \times 10^3$
Deep Quest	0.677
DSRV	0.435
Fleet Submarine	0.39
Albacore	0.1

where C_t is the drag coefficient for the appendages. Instead of breaking C_t into components, an appendage test usually only records the total drag coefficient. Much of the data for appendage resistance is based on frontal area, and the designer must be careful to use the correct combination of areas and coefficients. Table 3 lists drag coefficients for various shapes which are normal to the flow.

4.2.3 *Other Approaches*

In addition to relying on model data for estimating C_r, there are other approaches to estimating resistance. There are numerous potential-flow computer models that attempt to predict the fluid flow around a particular body. However, the description of the theory and computer programs used in these models is beyond the scope of this text.

S. F. Hoerner provides a simple algorithm for drag coefficient in his book *Fluid Dynamic Drag* [6]. This algorithm is based on a particular range of hull geometry, but can be useful for early estimates:

$$C_f = \frac{0.075}{(\log_{10}\mathrm{Re} - 2)^2} \tag{64}$$

Based on frontal area,

$$C_t = C_f\left[3\left(\frac{\text{length}}{\text{diam}}\right) + 4.5\left(\frac{\text{diam}}{\text{length}}\right)^{1/2} + 21\left(\frac{\text{diam}}{\text{length}}\right)^2\right] \tag{65}$$

Based on wetted surface,

$$C_t = C_f\left[1 + 1.5\left(\frac{\text{diam}}{\text{length}}\right)^{3/2} + 7\left(\frac{\text{diam}}{\text{length}}\right)^3\right] \tag{66}$$

4.2.4 *Sway and Heave Resistance*

Although the drag of the vehicle in the longitudinal direction is of primary concern, the resistance to sway and heave is important when considering station-keeping capability and maneuvering characteristics. Submersibles spend much of their time maneuvering at very slow speed. Under these conditions, the submersible is susceptible to ocean cur-

Table 3 Typical values of C_t for various appendages

Appendage	Area Basis	C_t
Small domes	Profile	0.015
Antennae	Projected	1.2
Cylinders	Projected	1.2
Arms	Wetted surface	0.005
Long faired protuberances	Wetted surface	0.005
Holes in skin	Projected frontal	0.5
Planes	Projected frontal	0.011

rents which can be as high as 4 knots. As the submersible turns perpendicular to the current, it is acted on by crossflow resistance forces. Depending on the shape of the hull, these forces can be much larger than the longitudinal drag forces. For this reason, submersibles usually cannot hold position perpendicular to a current and must head into it. However, the designer should be aware of the magnitude of these forces when designing control surfaces and maneuvering thrusters. Two rough approximations are to use either the profile area and the drag coefficient of a flat plate or the drag of a two-dimensional cylinder.

4.2.5 *Design Considerations*

The preceding discussion provides the designer with methods of predicting the vehicle speed/power profile prior to model or full-scale testing. The following paragraphs are intended to provide a broader understanding of the design considerations in terms of vehicle drag. In general, the designer is usually provided with a set of operational requirements in terms of endurance, range, speed, or operational time. These requirements are often loosely defined and subject to numerous changes. They are crucial to the design process and have a large impact on the resistance and energy analysis as shown below:

$$\text{Energy for propulsion} = \frac{\text{Energy}}{\text{Hours}} \times \frac{\text{Hours}}{\eta} \quad (67)$$

$$\text{Range} = \text{Speed} \times \text{Hours} \quad (68)$$

The Energy/Hour term represents the effective vehicle power required and varies with speed. η represents the total propulsion system efficiency, including the propulsive coefficient, losses in the motor and power lines, and losses in the battery or fuel cell. If the design has a specific range requirement, then the operational hours can vary. Thus, determining the maximum value of propulsion energy depends on a trade-off between speed, drag at each speed, range, efficiency, and operational hours. Although this seems obvious, the designer should analyze various operational profiles to determine the worst case. Having done so, the designer determines the appropriate propulsion equipment and energy storage system. As the design is developed and changes are made, the impact of the propulsion equipment (motors, controllers, cable, etc.) and the energy system (in the form of batteries, fuel cells, etc.) on the design is considerable. As the batteries or propulsion motors or both change in size, the entire vehicle size can change, which alters the drag profile and thus can require a further change in batteries or propulsion system or both. These iterations through the design spiral can go in either direction, the vehicle increasing or decreasing in size, complexity, and cost. As the design nears completion, these changes should become smaller and smaller until the design is considered accurate enough for construction to begin.

Slow-speed vehicles often do not have a rigid speed requirement. The designer assures that the vehicle falls within an acceptable range, typically 2 to 3 knots. In such a situation, speed is often determined by the availability of motor or battery sizes. The designer uses a combination of "off-the-shelf" motors or batteries that provide adequate performance while eliminating the need for a costly development program for a non-standard size. Low-speed requirements also result in the propulsion energy required being a smaller percentage of the total vehicle energy budget. This reduces the importance of a low-drag configuration arrangement. Thus many submersibles have a markedly non-aerodynamic configuration because the design places a higher priority on operational capabilities and does not pay a large drag penalty for doing so.

If the drag for a slow-speed vehicle must be reduced (to avoid being forced to the next larger motor size, for example), the primary contributors are the frictional portions of the base hull drag and the appendage drag. To minimize frictional drag, the designer should either make the vehicle longer to increase the Reynolds number and lower C_f or reduce the wetted surface. The reduction of appendage drag is a difficult task because submersibles typically require numerous appendages such as manipulators, tool baskets, video cameras, still cameras, thrusters, and control surfaces. These can occasionally be made retractable, but this leads to increased complexity, weight, and cost, with decreased operational flexibility.

Requirements for speeds greater than 4 knots provide additional difficulties for the designer. At such speeds, the residual portion of the bare hull drag begins to increase. The residual drag is strongly Reynolds number and shape dependent. For a streamlined body with few appendages, increasing the ratio of hull length to hull diameter reduces the residual drag, as shown in Fig. 38 [6]. Thus, a long tapered hull form is desirable, but it presents problems in terms of vehicle arrangements, crane handling, and on-deck stowage. The various appendages on the front of the submersible also diminish the benefits of a longer hull form.

For high-speed vehicles, the energy required for propulsion becomes a large portion of the vehicle energy budget. Thus, changes in the vehicle drag can have pronounced influence on the submersible size. This sensitivity emphasizes the importance of realistic drag estimates and makes the retraction of appendages a more attractive option.

4.3 Propulsors

4.3.1 *Propulsion System Overview*

The following discussion focuses on the use of propellers for propulsion, since they are the most

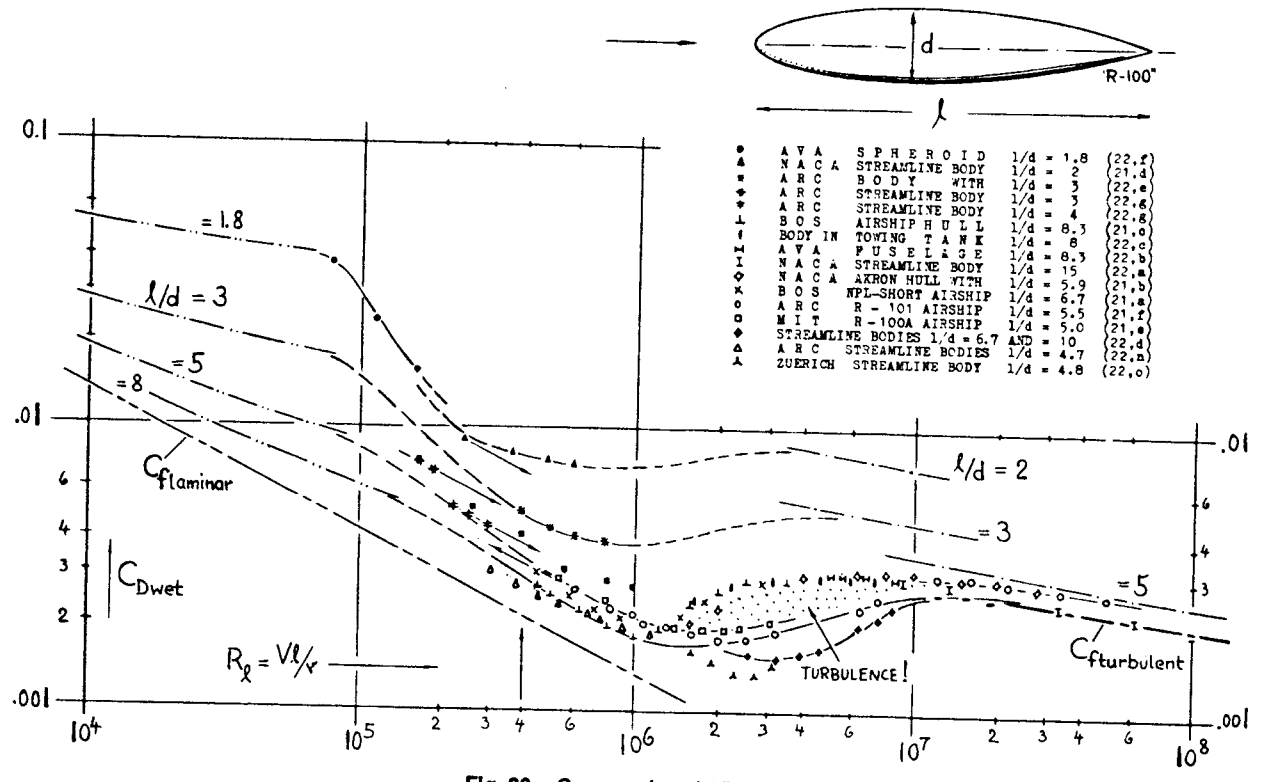


Fig. 38 C_f versus length/diameter

common propulsors. There are several considerations to be kept in mind during the selection of propeller type, size, rotational speed, efficiency, and other parameters. The basic principles will be presented here. The propeller should convert the power delivered by the propulsion motor (usually measured in horsepower) into thrust (usually measured in pounds) as efficiently as possible. This transfer of power involves several steps with each step producing losses. These are shown in Fig. 39; a more detailed description is also included.

EHP—Effective horsepower is the product of the total resistance and the submersible speed expressed in units of horsepower. The total resistance is estimated either by the drag calculations or by model tests.

$$EHP = \frac{R_t \times V}{550} \tag{69}$$

THP—Thrust horsepower is the product of the thrust delivered by power from the propeller and the submersible speed expressed in units of horsepower (the power just aft of the propulsor). The difference between *THP* and *EHP* is due to two factors. First, the drag caused by the propulsor itself is accounted for as a reduction in thrust produced; this is simply called the "thrust deduction" coefficient and is defined as a percentage of the total thrust:

$$\text{thrust deduction} = t = \frac{T - R_t}{T} \tag{70}$$

where T is the total thrust in pounds delivered by the propeller and R_t is the total resistance in pounds. Rearranging equation (70),

$$T = \frac{R_t}{(1 - t)} \tag{71}$$

The second portion of the losses between *THP* and *EHP* is due to the energy lost in the flow behind the vehicle before it enters the propeller. The speed of the water in the wake of the hull, V_A, is slower than the speed of the hull, V. V_A is known as the speed of advance. The wake fraction w is the ratio of the wake speed divided by the hull speed:

$$w = \frac{V - V_A}{V} \tag{72}$$

Rearranging equation (72),

$$V_A = (1 - w)V \tag{73}$$

Both the wake fraction and thrust deduction terms can be determined accurately only through the use of model tests but can be approximated from Fig. 40 as a function of diameter. Using the definition of horsepower and substituting the above relations yields:

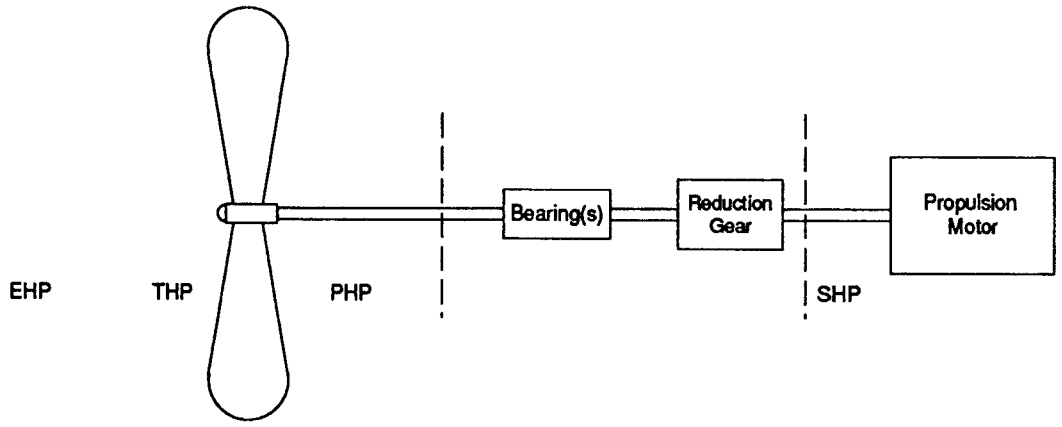


Fig. 39 Propulsion system efficiencies

$$THP = \frac{T \times V_A}{550} = \frac{R_t}{(1-t)} \times \frac{(1-w)V}{550} \tag{74}$$

$$= \frac{(1-w)\,R_t V}{(1-t)\,550}$$

$$= \frac{(1-w)}{(1-t)}\,EHP \tag{75}$$

$$EHP = \frac{(1-t)}{(1-w)}\,THP \tag{76}$$

The term $(1-t)/(1-w)$ is known as the hull efficiency, η_H. For a well-designed stern and propeller, the hull efficiency is approximately equal to 1.0. For a body of revolution with a single propeller on the axis, hull efficiency can be greater than 1.0. Therefore it should be obvious that this is not a true efficiency.

PHP—Propulsive horsepower is the power delivered to the propulsor (the power in the shaft just forward of the propeller). The difference between *PHP* and *THP* is due to the losses in the propulsor itself. (Different types of propellers have different efficiency ranges as will be discussed later.) The power delivered to the propeller can be determined as

$$PHP = \frac{2\pi n Q}{550} \tag{77}$$

where n is the shaft rotation in revolutions/s and Q is the torque delivered in ft·lb. The efficiency of the propeller η_p is defined as

$$\eta_P = \frac{THP}{PHP} = \frac{TV_A}{2\pi n Q} \tag{78}$$

The propeller efficiency also has two components: (1) the open water efficiency η_O and (2) the relative rotative efficiency η_R. The open water efficiency is measured for the propeller tested without interference from a wake. The open water efficiency depends on the propeller type, diameter, rpm, number of blades, and other parameters as discussed later. The relative rotative efficiency represents the difference between the open water efficiency and the efficiency actually measured in the hull wake. The value of η_R ranges from 0.95 to 1.0 for a submersible.

SHP—Shaft horsepower is the power delivered by the propulsion device (electric or hydraulic motor, etc.) to the propeller shaft. The difference between *SHP* and *PHP* is due to frictional losses in any bearings, linkages, or reduction gears. Usually the propulsion motor in a submersible is chosen which does not require a reduction gear; this saves weight and reduces propulsion system losses. The machinery efficiency η_M is defined as

$$\eta_M = \frac{PHP}{SHP} = \frac{Q}{Q_S} \tag{79}$$

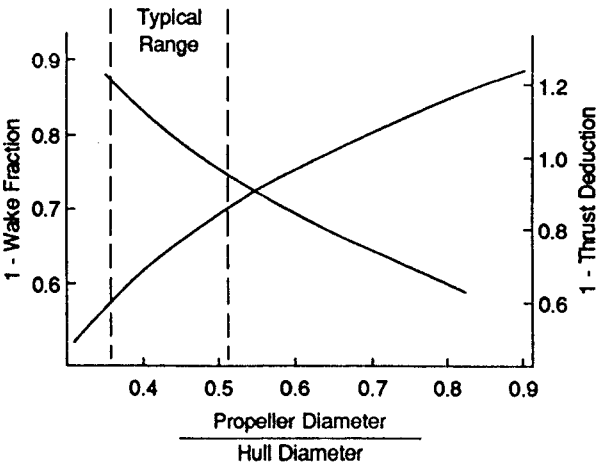


Fig. 40 Thrust deduction and wake fraction versus propeller diameter ratio

where Q_S is the torque at the propulsion device in ft-lb. Depending on the shaft length and the number of bearings, η_M ranges from 0.95 to 0.99.

All of the losses defined above can be combined into a term known as the propulsive coefficient, *PC*.

$$PC = \eta_H \eta_R \eta_O \eta_M \tag{80}$$

$$PC = \frac{EHP}{SHP} \tag{81}$$

The higher the propulsive coefficient, the higher the efficiency of the propulsive system and the smaller the size of the energy system required. Thus, *PC* is a good criterion to judge the efficiency of a particular design. Typical values range from 0.55 to 0.85. Another term frequently used is the quasi-propulsive coefficient defined as

$$QPC = \eta_R \eta_O \eta_H \tag{82}$$

It does not include the machinery losses and thus more clearly represents the efficiency of the propeller and the stern flow.



Fig. 41 Ducted propeller

4.3.2 *Propulsor Types*

There are several types of propulsors, including open propellers, ducted propellers, counter-rotating propellers, vertical axis propellers, and water jets. The purpose of all propulsors is to accelerate the water as it enters the propulsor. This acceleration produces a change in the water momentum per unit time—thrust. Momentum is defined as the mass flow times the change in velocity (acceleration). So to increase the change in momentum, the propeller can either increase the acceleration or increase the mass flow. In general, it is more efficient to increase the mass flow rather than increase the acceleration. Thus, the larger the propeller diameter, the greater the mass flow and the greater the efficiency. Although the hydrodynamics of propeller design are beyond the scope of this text, most propulsors use an airfoil-shaped blade to accelerate the flow. These blades provide lift and drag just as an airplane wing does.

Each type of propulsor has its advantages and disadvantages. The most prevalent type is the ducted propeller. The ducted propeller is basically an open propeller with a tunnel placed around it (see Fig. 41). This tunnel reduces the loss of lift (thrust) found at the blade tips and thus increases efficiency. The airfoil-shaped tunnel also produces additional thrust and can be used as a nozzle to direct the propeller flow and act as a rudder. From the designer's point of view, the selection of the open and ducted propeller follows very similar paths as discussed below.

Open and ducted propellers—Until the advent of the modern computer, propeller selection often was accomplished by building a model and adjusting the numerous design parameters until an acceptable design was reached. These parameters included the diameter, the rpm, the number of blades, the area of each blade, the distribution of area along each blade, the pitch (or angle of attack) of each blade, etc. A typical propeller drawing with these parameters labeled is shown in Fig. 42. This testing was a time-consuming and expensive task. Over the years, series were developed which varied a single parameter at a time to provide a database for propeller selection. These series provided the designer a method of obtaining a good estimate of performance for a propeller similar to a series type. These series are still widely used today to predict performance and will be discussed in detail below.

The computer provided a tool to model propeller flow and predict performance with reasonable accuracy. Computer design is often used to progress a new propeller design to an advanced stage, with model testing for final adjustments.

One of the more widely used propeller series is the Troost B series. Started over 30 years ago and still in use today, this series is appropriate for merchant and Navy auxiliary use and is suitable for submersible design. The nomenclature is

> B.4.55 indicates a B series propeller with four blades and an expanded area ratio of 0.55. The expanded area of a propeller is a measure of the blade area.

The summary of the Troost series is given in Table 4. Once the number of blades and area ratio was established, the pitch, rpm, propeller horsepower *PHP*, speed of advance, thrust, and diameter were

Table 4 Summary of Troost series propellers

Number of Blades	Blade Area Ratio												
2	0.30												
3		0.35			0.50			0.65					
4			0.40			0.55			0.70				1.00
5				0.45			0.60			0.75			1.05
6					0.50			0.65			0.80		
7						0.55			0.70			0.85	

varied and the open water propeller efficiency was recorded. This data is presented nondimensionally in one of two formats, each of which uses a different nondimensional term.

When installed horsepower *SHP* is known, then use B_p-γ curves.

$$B_p = \frac{NP^{1/2}}{V_A^{5/2}}$$

$$\gamma = \frac{ND}{V_A} \qquad (83)$$

where

N = rpm,
P = propeller horsepower *PHP*,
D = propeller diameter, and
V_A = speed of advance.

To obtain the open water efficiency, find the intersection of the γ curve and B_p and then obtain the open water efficiency and pitch to diameter ratio, H/D. An example curve is shown in Fig. 43.

When required horsepower, *EHP*, or R_t is known, use the B_u-γ curves.

$$B_u = \frac{NU^{1/2}}{V_A^{5/2}}$$

$$U = \frac{TV_A}{550} \qquad (84)$$

where the symbols have the same meaning as for equation (83). To obtain the open water efficiency, find the intersection of the γ curve and B_u and then obtain the open water efficiency and pitch to diameter ratio, H/D. An example curve is shown in Fig. 44.

To use an existing propeller shape based on a previous application, the designer needs to determine a new diameter, rpm, and efficiency to match the new requirements for torque and thrust. This requires the model test data from the existing propeller. This data is usually plotted nondimensionally. An example is shown in Fig. 45, and the coefficients are defined as

$$\text{Thrust coefficient, } K_T = \frac{T}{D^4n^2} \qquad (85)$$

$$\text{Torque coefficient, } K_Q = \frac{Q}{D^5n^2} \qquad (86)$$

As defined previously,

$$\eta_O = \frac{TV_A}{2\pi nQ} \qquad (87)$$

$$= \frac{K_TV_A}{2K_QDn}$$

$$= \frac{K_TJ}{2K_Q} \qquad (88)$$

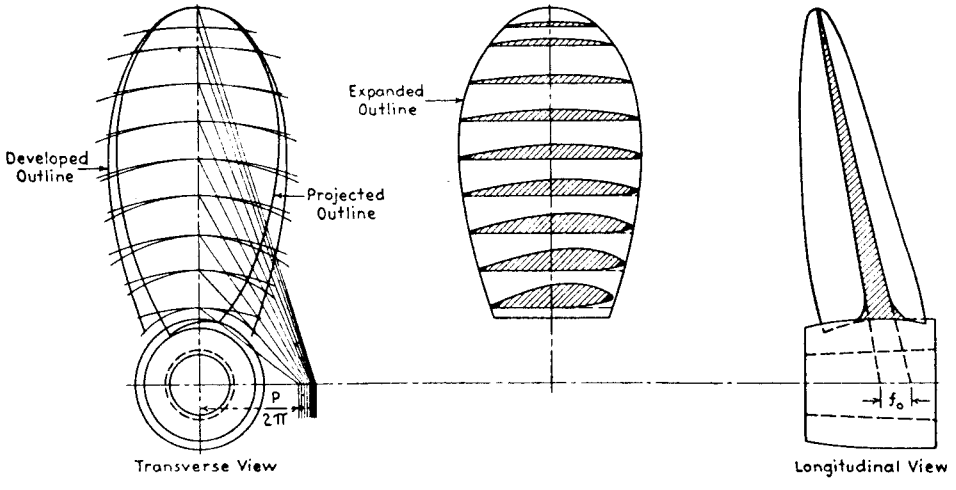


Fig. 42 Propeller geometry

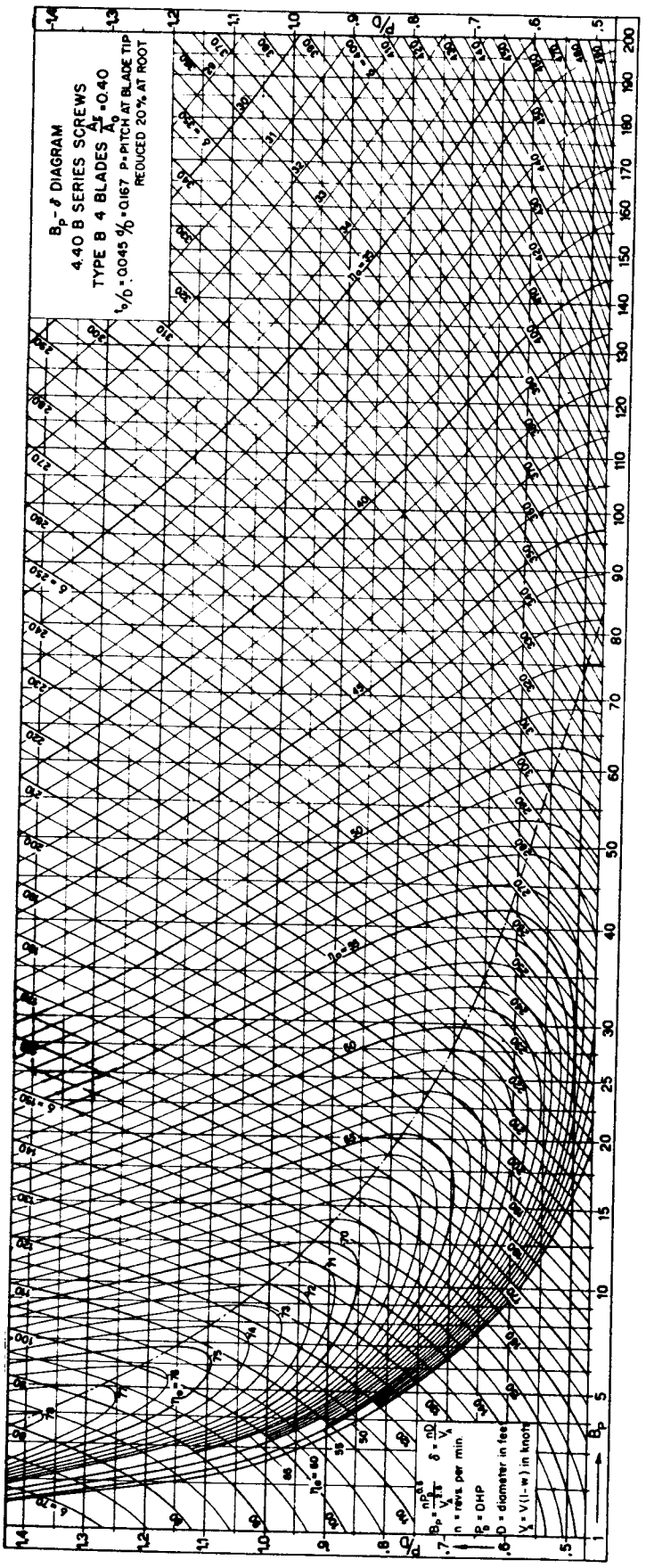


Fig. 43 B_P-γ diagram

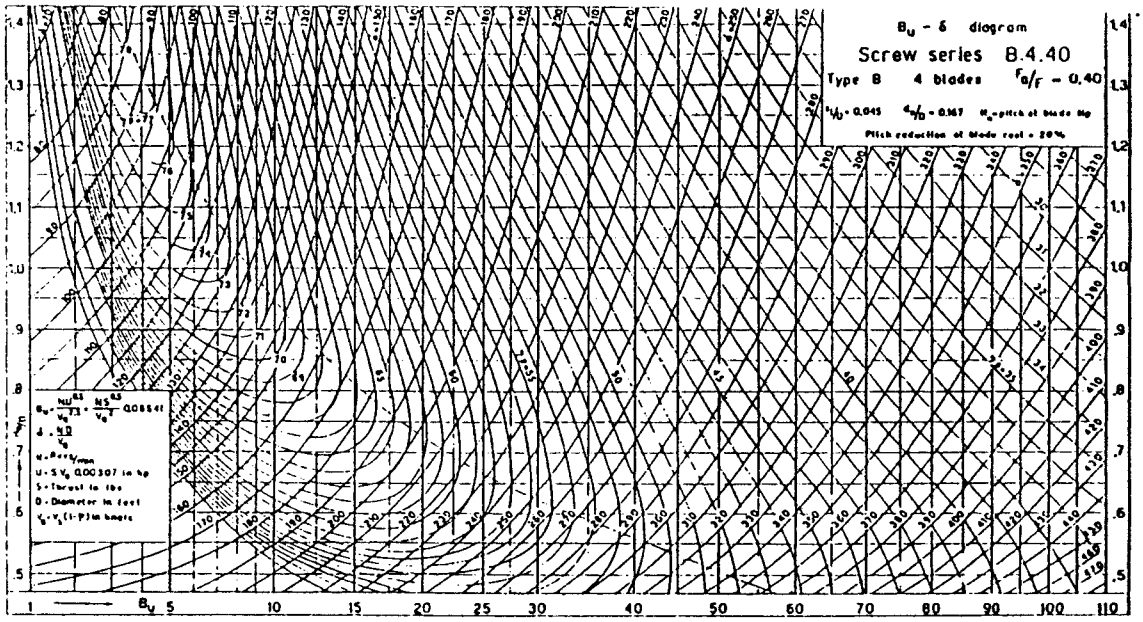


Fig. 44 B_u-γ diagram

where J is known as the advance coefficient,

$$J = \frac{V_A}{Dn}$$

The selection of the correct diameter and rpm using the model data is an iterative process. It is not sufficient to calculate the advance coefficient for a desired diameter and rpm and then use the corresponding open water efficiency. The designer must also check the corresponding torque and thrust from the K_T and K_Q curves to assure that there is sufficient torque and thrust. This usually involves attempting several combinations until an adequate solution is found. As evident from the coefficient definitions, the diameter has a large impact on propeller efficiency (for a constant pitch and blade area). A reasonable starting point is for a propeller diameter equal to one-half the hull diameter.

By using the above approaches, the designer can make a reasonable estimate of the open water propeller efficiency for use in calculating the propulsive coefficient and thus the size of the energy system. The selection of a propeller can be a time-consuming process. The propeller diameter is limited by its weight, the increasing risk of entanglement, and cavitation (cavitation will be discussed more thoroughly below). The rpm is limited by the selection of the propulsion motor or the reduction gears or both, the shaft size and strength, and cavitation. For high-speed submersibles, the designer must be careful not to choose a propeller that provides excellent high-speed efficiency yet has very poor low-speed efficiency and poor maneuvering. (If the propeller selection has a large impact on the design, it may be prudent to receive assistance from an expert in the field of propeller design.)

As the propeller blade passes through the water, it generates lift by inducing a static pressure difference between the two sides of the propeller blade. This difference in pressure across the blade area provides the thrust. If the static pressure is lowered to the water vapor pressure level, the water will begin to boil, which is known as cavitation. This boiling produces small vapor bubbles along the leading edge of the blade and at the tips. Small amounts of cavitation are acceptable, but as it increases, these small vapor bubbles collapse in the higher-pressure regions of the blade and cause erosion of the blade as well as decreased efficiency. To avoid cavitation, the designer may increase the blade area, increase the diameter, decrease the rpm, or increase the

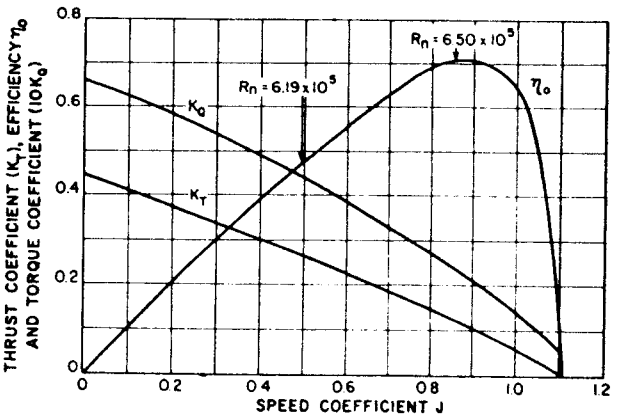


Fig. 45 K_T and K_Q versus J

depth of propeller submergence. This last option increases the ambient static pressure and usually provides submersibles with adequate cavitation protection.

The number of propeller blades can be an important consideration due to potential vibration problems. As the propeller rotates through the wake of the hull, it encounters varying levels of static pressure caused by the interference of the hull and appendages (this is particularly pronounced on surface ships). The varying pressures cause the blade to vibrate since it is acted upon by an oscillating load. This vibration can be severe enough to prevent the propeller from being turned at its maximum rpm. Fortunately, most submersibles are a body of revolution which provides a more uniform pressure distribution for the propeller. If there are an even number of appendages near the stern (stabilizer fins, for example) then a propeller with an odd number of blades should be used. The converse of this rule also applies.

The open propeller is not widely used in submersible design. The ducted propeller is chosen instead because of the maneuvering capability that a steerable ducted propeller provides. In addition, the duct helps protect the propeller from entanglement by fishing line, nets, rope, sea weed, and other hazards. From the design point of view, the ducted propeller follows the steps outlined above for the open propeller. The addition of the duct should add 5 percent to 10 percent increased efficiency if the clearance between the blade tips and duct is kept small.

Contra-rotating propeller—The contra-rotating propeller uses two open-water-type propellers on coaxial shafts, one turning in one direction, the other turning in the other direction (see Fig. 46). The purpose of this is to recover the rotational losses of the first propeller by the second propeller. Contra-rotating propellers also allow smaller diameters for the same rpm because of the increased blade area. They are rarely used on submersibles because they require a very complex drive shaft and gearing system. The additional weight and cost is not worth the small increase in efficiency.

Vertical axis propeller—The vertical axis propeller uses several blades mounted vertically as shown in Fig. 47. A complex cam system rotates these blades and changes their angle of attack as the blades rotate. The efficiency of such a propeller is lower than an open-water propeller but provides excellent maneuvering characteristics. Again, however, they are not a viable candidate for submersible use due to their complexity, weight, cost, and the chances for damage.

Controllable pitch propeller—The controllable pitch propeller has a mechanism on the propeller hub that changes the pitch of the blades. This allows the pitch to be set to maximize efficiency at various operating speeds. This is very useful for Navy ships which have a lower cruise speed and a high dash speed. The blade pitch can be changed so as to provide reverse thrust, which eliminates the reversing gear. However, submersibles do not have a large enough speed range to make this type of propeller attractive.

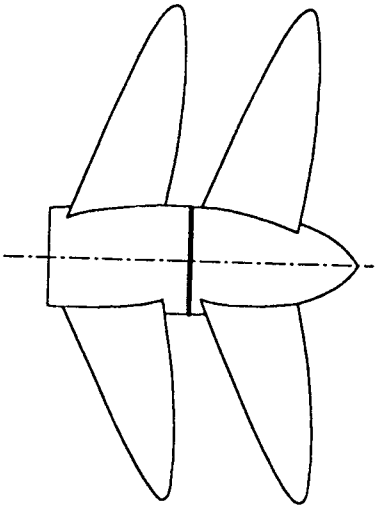


Fig. 46 Contra-rotating propeller

Waterjets—The waterjet falls into the pump category. It typically has a small diameter and thus operates at much higher rpm; it is often inside of the submersible, with an inlet and outlet duct leading to it. The waterjet operates at a lower efficiency than an open propeller at low vehicle speeds. Thus it is rarely used for submersible main propulsion, but its small size occasionally makes it useful as a maneuvering thruster (as in the *Deep Quest*).

4.4 Stability and Control

4.4.1 Stability

As mentioned earlier, submersibles spend a major portion of their time traveling short distances while conducting numerous stops, starts, and turns. To do so safely and efficiently, the submersible requires both stability and the control forces sufficient to maneuver the submersible. (It is assumed that the submersible has adequate hydrostatic stability, as discussed in Section 3 of this chapter.) There are several types of stability as discussed below. The greater the stability, the greater the resistance to deviation from the vehicle's original path, and the greater the control forces necessary. If the stability is insufficient relative to the control surfaces, the submersible can become dangerously uncontrollable

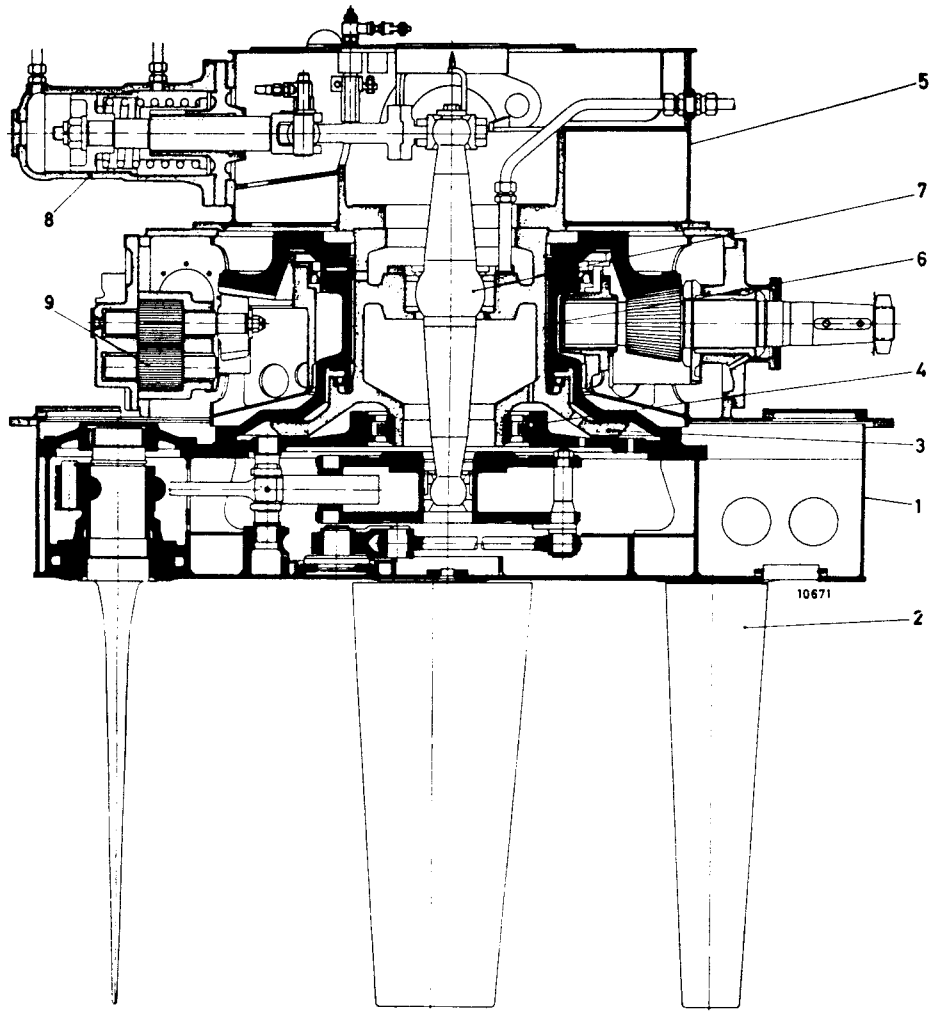


Fig. 47 Vertical axis propeller

as the control forces will generate forces too large for the vehicle stability to damp out. Thus the designer must trade-off between the two and reach an equilibrium where the vehicle maneuvers with reasonable ease yet will return to its original path upon completion of a maneuver.

The general approach to the problem of submersible hydrodynamics is to address the subject of motion stability first and then discuss the more vital subject of control. A body is considered dynamically stable if it returns to its initial state of equilibrium after an external force has been placed on the submersible and then removed. In discussing motion stability, the submersible is assumed to be initially traveling in a straight line and level path, as shown in Fig. 9. At time zero, it is disturbed by an external force which is then removed. In Case A, the final path after removal of the force is a straight-line but in a different direction. This type of motion stability might be called straight-line stability since this is the characteristic of the initial motion which is retained. In Case B, the final path after release of the external force retains both the straight-line and horizontal-plane direction characteristics and the submersible may be said to have directional stability. Case C is similar to Case B except that the submersible oscillates about the path prior to achieving its final state of equilibrium. In Case D, all characteristics of initial motion are retained and the submersible possesses positional stability. Unless the vehicle is at the surface, this type of stability in the vertical plane cannot be achieved with "controls fixed," but requires control forces and moments to bring the submersible back up to the initial depth.

The types of stability have been discussed in ascending order. For a submersible to be directionally stable, it must also have straight-line stability. A submersible that has positional motion stability must have both straight-line stability and directional stability.

Hydrostatic or metacentric stability and rest position stability are always associated with fixed con-

trols. With each kind of stability discussed, there is an index associated with it that indicates by the sign and the magnitude whether the body is stable or unstable in that particular sense. The conventional metacentric height is an index of the hydrostatic stability. For a submersible to be stable in this condition, the center of buoyancy must be higher in the submersible than the center of gravity. It should be noted that the hydrostatic stability, when submerged, is the same value and sign in the fore and aft direction as in the transverse direction. This is vastly different than when operating on the surface, where the longitudinal stability greatly exceeds the transverse stability. The moments for the pitch and roll directions are the derivatives of the moments multiplied by the appropriate angles and have been shown earlier. The yaw metacentric stability index N is always zero because there is no restoring force in the horizontal plane. Negative values of the indices indicate positive stability, and positive values indicate instability.

The indices for stability in the straight-line and the directional conditions are not so readily or simply stated. As mentioned in the overview, solving the equations of motion without computer models and a database for the selection of the various coefficients is a difficult process and is beyond the scope of this text.

4.4.2 *Control*

The control system for a submersible is comprised of static and dynamic components. The static systems include ballast tanks, trim tanks, and weight systems. These affect the static characteristics of the submersible, as discussed in Section 3 of this chapter. The control surfaces, main propulsion, and maneuvering thrusters comprise the dynamic systems. While most submersibles have all three types of dynamic maneuvering systems, they each have an application in which they are appropriate. In general, these applications fall into the low-speed and high-speed conditions. The pilot joy-stick, wheel, or other operating mechanism is usually used under both conditions. Associated with the pilot operating mechanism is a control system which converts the pilot's signals into the appropriate signals for the maneuvering device. This conversion can occur mechanically, hydraulically, or electronically. There is typically at least a rudimentary backup system.

As with the prediction of top speed, the sizing of the maneuvering systems is not always an exact science. From the longitudinal and transverse drag estimates, the designer has an idea of the thrust levels required to translate the vehicle along a particular axis. To provide rotation about an axis, a force must be applied at a distance from the center of rotation for that axis. The greater the force or greater the moment arm, the better the turning rate. Unfortunately, submersibles rarely have a steadfast requirement for a particular turn rate; as long as the submersible can turn, the design is usually acceptable. (The DSRV is an exception—its mission requires exceptional maneuvering capabilities to enable it to attach to a submarine.) Maneuvering performance is further complicated by currents. Certain maneuvers may not be possible if the current reaches a certain level. Thus, the designer should examine the forces involved and make a best estimate of control system type and size. Model tests, when available, and full-scale testing can be used to verify estimates and provide data for design modifications, if necessary.

Low-speed control—Submersibles spend much of their time at slow speed maneuvering around objects of interest. The designer must provide a control system that will produce the required motions. The dynamic maneuvering systems will be discussed individually below.

The main propulsion system is often a significant portion of the slow-speed control system. To be useful, it must be designed to provide thrust evenly and in small amounts to inch the vehicle back and forth slowly. This is best accomplished with low-speed propulsion motors and a fairly large propeller diameter. The ducted propeller is an ideal propulsor because it provides improved thrust at low vehicle speeds over an unducted propeller. If the main propulsion system is comprised of a single propulsor, then the propulsor is often allowed to pivot (as in the *Alvin*) to vector the thrust from side to side. In the case of the DSRVs, the propeller shroud can pivot in two degrees of freedom. On those vehicles on which two or more propulsors are used for main propulsion, such as the *Beaver IV*, then alternate propulsors can be used to turn as well as move the submersible. The disadvantages of the main propulsor for maneuvering include its typical energy inefficiencies at low speeds and the limited range of motions that it can generate. For example, the single main propulsor can swing the stern, but cannot swing the bow directly.

Control surfaces are not as effective as the main propulsion system and maneuvering thrusters for slow-speed operation. The forces generated by an airfoil-shaped rudder or elevator surface are given by the following equations and shown in Fig. 48. Values for the lift and drag coefficients depend on the airfoil section chosen. The NACA 0015 section is often used. (See reference [7] for a discussion of the NACA series and airfoil design.)

$$\text{Lift Force} = \tfrac{1}{2}\rho V^2 S C_l \qquad (89)$$

$$\text{Drag Force} = \tfrac{1}{2}\rho V^2 S C_d \qquad (90)$$

where

S = projected area of the control surface in ft^2,
C_l = lift coefficient as a function of the angle of attack,

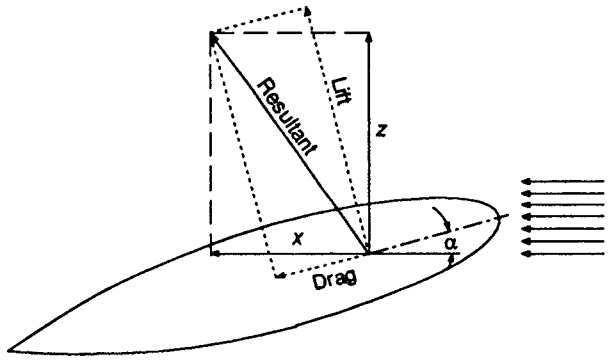


Fig. 48 Lift and drag of an airfoil section

C_d = drag coefficient as a function of the angle of attack, and
V = vehicle velocity in ft/s.

There are several parameters that impact the effectiveness of the control surfaces. These are summarized below:

1. As can be seen by the above equations, the forces are strongly dependent on the velocity. Thus, the control surfaces rely on the main propulsion system or other thrusters to move the submersible at sufficient speed for the surfaces to be effective. They will provide a small force when the vehicle is operating at low speeds. For this to be overcome, the surfaces are made larger which increases drag.
2. The further the surface extends into the flow away from the hull and out of the boundary layer, the greater the effectiveness at the risk of entanglement and damage.
3. The lift and drag coefficient values will increase with angle of attack up to a certain point, typically 12 deg. Thus, at low speed it is advantageous to rotate the entire control surface to provide an angle of attack as the submersible moves.

To determine surface size, an acceptable approach is to imitate the surfaces on an existing similar submersible. The surface sizes and placement can be scaled to alter performance as required. The estimates of forces required and forces produced should then be double-checked.

The duct around the propeller acting as a control surface overcomes some of the above deficiencies. The local fluid velocities are higher, and thus the duct generates larger forces. The duct can often pivot and thus also has an induced angle of attack.

Properly placed maneuvering thrusters can provide the submersible with excellent slow speed maneuvering. They can also be used in conjunction with the main propulsor to increase transit speed (as in, for example, the *Alvin*). Thrusters are usually either an electric or a hydraulic motor with a small-diameter propeller. The estimation of the thrust produced is fairly straightforward, and the designer can accurately select the motor and propeller required. They are used in many configurations but are often arranged in pairs and are either in a long tunnel through the vehicle (as in DSRVs) or attached to the side on a swivel mount (as in the *Alvin*). The placement in pairs provides thrust in opposite directions for precise positioning. The largest disadvantage is the fact that they will stir up the silt on the ocean bottom and can make vision very difficult. Other disadvantages include significant vehicle drag at transit speeds and risk of entanglement.

In summary, there is no clearly superior slow-speed control system. The designer has additional constraints if the submersible also has a high-speed requirement.

High-speed control—Most of the discussion of the three dynamic maneuvering systems for low-speed operation is applicable for high-speed use with a few notable exceptions, as follows. The intent was to make the submersible highly maneuverable at low speeds to allow positioning around objects; at high speeds the designer must assure that the vehicle is controllable. However, with submersibles, as with automobiles, the faster one travels, the faster accidents can occur.

The main propulsion system, by definition, provides the thrust necessary for axial motion. When fitted with a duct or shroud, the main propulsion system usually provides sufficient control forces, thus eliminating the need for other high-speed control systems (as with the DSRVs). If the duct or shroud has been designed for the low-speed case by increasing the area, then the large forces generated at higher speed can be dangerous. The control system must be designed with this in mind, and automatic limits should be put on the control surface movement as the vehicle increases speed.

Control surfaces such as rudders and elevators provide two functions at high speed: they influence the vehicle stability, and offer a way to alter the vehicle path. The higher fluid velocities make them markedly more effective in the high-speed range. Large, fixed surfaces designed for low-speed operation and placed at the stern will increase the high-speed dynamic stability (much as larger feathers on an arrow increase its stability). This arrangement will require larger control forces to overcome. If these surfaces are allowed to rotate to produce lift, then the forces generated may be so large as to make the vehicle uncontrollable. Automatic speed limits would be required to make the submersible safe. An alternative is to provide a smaller flap at the trailing edge of the surface that would be allowed to move, instead of rotating the entire surface. Both of these solutions increase the system complexity, weight, and cost. Appendages and control surfaces near the bow of the submersible, such as a sail, also complicate the design by providing additional forces on the submersible, particularly at high speed. For submarines at much

higher speeds, these forces can cause the submarine to roll significantly as it turns in the horizontal plane.

The maneuvering thrusters are of little benefit at higher speeds. The forces produced are usually minor compared to other forces acting upon the vehicle by the propulsion system and hydrodynamic forces. The drag the thruster causes is an added penalty.

Control-system design considerations—The designer should analyze the candidate control systems at both hovering- and top-speed conditions. As mentioned earlier, analysis of an existing design that is similar is an excellent starting point for a baseline approach. The designer should outline the existing characteristics and compare them to the new requirements, keeping in mind many considerations during the selection of the control system. These include maneuvering performance, dynamic stability, additional drag, additional energy required, complexity of the control system and pilot interface, cost, weight, buoyancy, reliability, and maintainability. The final selection of particular components must be made with consideration of the interaction with other systems.

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Appendix[26]

Compressibility of Submersible Pressure Hulls

Introduction

Hull compressibility equations are presented for spherical and stiffened cylindrical pressure hulls of undersea vehicles. Classical equations for hull compressibility are cumbersome because of term complexity and because they contain nonlinear functions of submergence depth. The author introduces an approximate, simplified equation for compressibility of spherical pressure hulls. An approximate equation, developed by the U.S. Navy, is also given for stiffened cylindrical pressure hulls. Unit compressibility, volume change per unit depth divided by uncompressed hull displacement, is used to compare hull compressibility with sea water compressibility. Curves are presented to illustrate the variation of hull compressibility with hull radius and thickness. Nomenclature associated with compressibility equations appears at the end of this appendix.

[26]The author of the Appendix is Edward A. Fernald.

Pressure Hull versus Sea Water Compressibility

If a pressure hull is more compressible than sea water, variable ballast must generally be discharged during increased submergence in order for the submersible to maintain neutral buoyancy. If a pressure hull is less compressible than sea water, variable ballast must generally be taken on in order to maintain neutral buoyancy with increased submergence.

The situation of a hull more compressible than sea water is illustrated in Fig. 49, where a sphere of sea water and a spherical pressure hull are both neutrally buoyant and have the same weight and outside diameter when fully submerged at the surface. As

both spheres submerge to equal depths, the pressure hull sphere is shown to compress to a smaller diameter than the sea water sphere. The sea water sphere always remains neutrally buoyant. The pressure hull remains equal in weight to the sea water sphere but, due to its smaller volume, is no longer neutrally buoyant. The volume difference between the compressed diameters of the two spheres is the amount of sea water which must be discharged from variable ballast of the pressure hull sphere, or the submerged weight of drop weights which must be released, to attain neutral buoyancy.

Spherical pressure hulls of acrylic material would respond as shown in Fig. 49. Spherical pressure hulls of steel, however, are invariably less compressible than sea water.

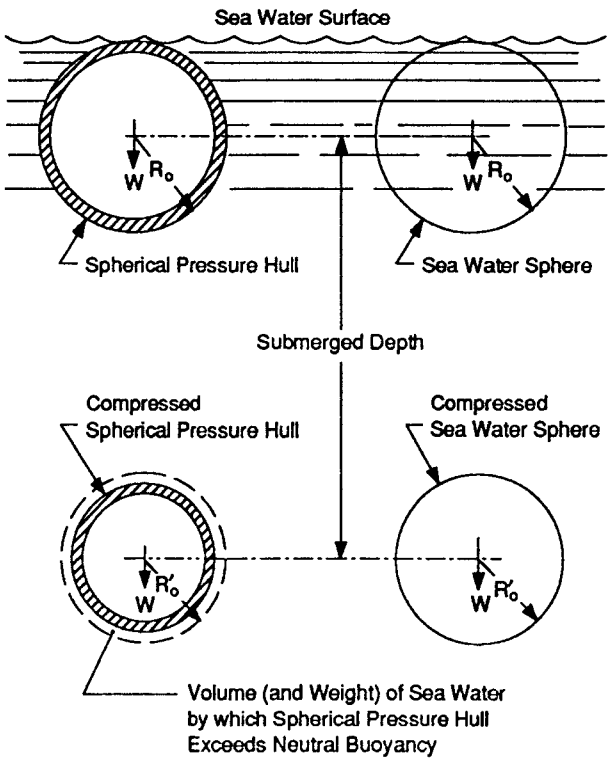


Fig. 49 Compression of pressure hull and sea water

Sea Water Compressibility

Compressibility factors can be calculated to represent the changes in pressure hull and in sea water volumes per foot of submergence depth. The compressibility factor for a sea water sphere, C_{sw}, is based upon unit weights of 64.0 lb/ft^3 at the surface and 66.6 lb/ft^3 at a depth of 30 000 ft [9].

$$C_{sw} = \frac{4/3\pi R_0^3\left(1 - \dfrac{64.0}{66.6}\right)}{30\,000 \text{ ft}} = (5.450878 \times 10^{-6})\frac{R_0^3}{\text{ft}} \tag{91}$$

Dividing C_{sw} by the sea water sphere volume gives the unit compressibility of sea water, U_{sw}:

$$U_{sw} = \frac{C_{sw}}{4/3\pi R_0^3} = \frac{1.3013013 \times 10^{-6}}{\text{ft}} \tag{92}$$

Hull Compressibility

Unstiffened Spherical Hull

The compressibility factor for an unstiffened spherical pressure hull can be determined from mechanics of materials [10] by the equation

$$C_s = \frac{4\pi}{3h}\left\{\left(R + \frac{t}{2}\right)^3 - \left[R + \frac{t}{2} - (1 - \nu)\frac{PR^2}{2Et}\right]^3\right\} \tag{93}$$

Inspection of equation (93) shows that compressibility of a spherical pressure hull is a function of submergence depth. Results, however, are sufficiently constant with depth that the following approximate equation may be used:

$$C_s = \frac{64\pi R^4}{144(1 - \nu)Et}\left[2(1 - \nu)^2 + \frac{t}{R}\right]\frac{\text{psi}}{\text{ft depth}} \tag{94}$$

If the t/R term of equation (94) is taken as zero, the equation for volume change of a sphere is the same as given by reference [11].

Unit compressibility of the hull can be obtained by dividing equation (94) by the pressure hull displacement of the fully submerged, but uncompressed hull:

$$U_s = \frac{C_s}{4/3\pi R_0^3} = \frac{R}{3(1 - \nu)Et}\left(\frac{R}{R + 1/2t}\right)^3\left[2(1 - \nu)^2 + \frac{t}{R}\right]\frac{\text{psi}}{\text{ft}} \tag{95}$$

Figure 50 illustrates unit compressibilities for ranges of thickness and mid-surface radius of a spherical pressure hull. The dashed line in Fig. 50 indicates unit compressibility of sea water, equation (92). Personnel spheres of selected submersibles are plotted on Fig. 50.

Stiffened Cylindrical Hull

The compressibility factor of a stiffened cylindrical pressure hull, the cubic inch change in volume of a typical cylindrical section between adjacent transverse stiffening rings per foot of sea water depth, is given as [12]:

$$C_c = \frac{64\frac{\text{lb}}{\text{ft}^3}}{144\frac{\text{in}^2}{\text{ft}^2}} \frac{\pi R^3 L_f (2-\nu)^2}{2Et} \left[1 + \frac{(1-\nu^2)}{(2-\nu)^2} - \frac{1}{\frac{L_f}{A_f} + \frac{\theta}{2N}} \right] \tag{96}$$

where

R = radius to mid-surface of hull plate,
t = thickness of hull plate,
L_f = center to center frame spacing,
A_f = area of frame,

$$\theta = \frac{[3(1-\nu^2)]^{1/4} L}{(Rt)^{1/2}}$$

L = unsupported length of shell between frames,

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

E = modulus of elasticity, and
ν = Poisson's ratio.

The unit compressibility of a stiffened cylindrical pressure hull is obtained by dividing equation (96) by the displacement of one frame space length of hull:

$$U_c = \frac{C_c}{L_f\left(R + \frac{t}{2}\right)} = \frac{2\pi R^3 (2-\nu)^2}{9Et\left(R + \frac{t}{2}\right)} \left[1 + \frac{(1-\nu^2)}{(2-\nu)^2} - \frac{1}{\frac{L_f}{A_f} + \frac{\theta}{2N}} \right] \frac{\text{psi}}{\text{ft}} \tag{97}$$

Unit compressibilities for various ranges of hull thickness and midsurface radius are given by Fig. 51, in which specific coefficients of hull thickness and radius have been assumed for determination of frame spacing, frame area, and frame attachment width. The compressibility factor of a multi-frame cylindrical hull can be calculated as the product of the unit compressibility determined for a typical frame and the uncompressed submerged displacement of the region which the typical frames reinforce.

Having determined compressibility for both the sea water and the submersible pressure hull, it is now possible to determine the corresponding variable ballast change necessary to maintain neutral buoyancy of the pressure hull. The volume of variable

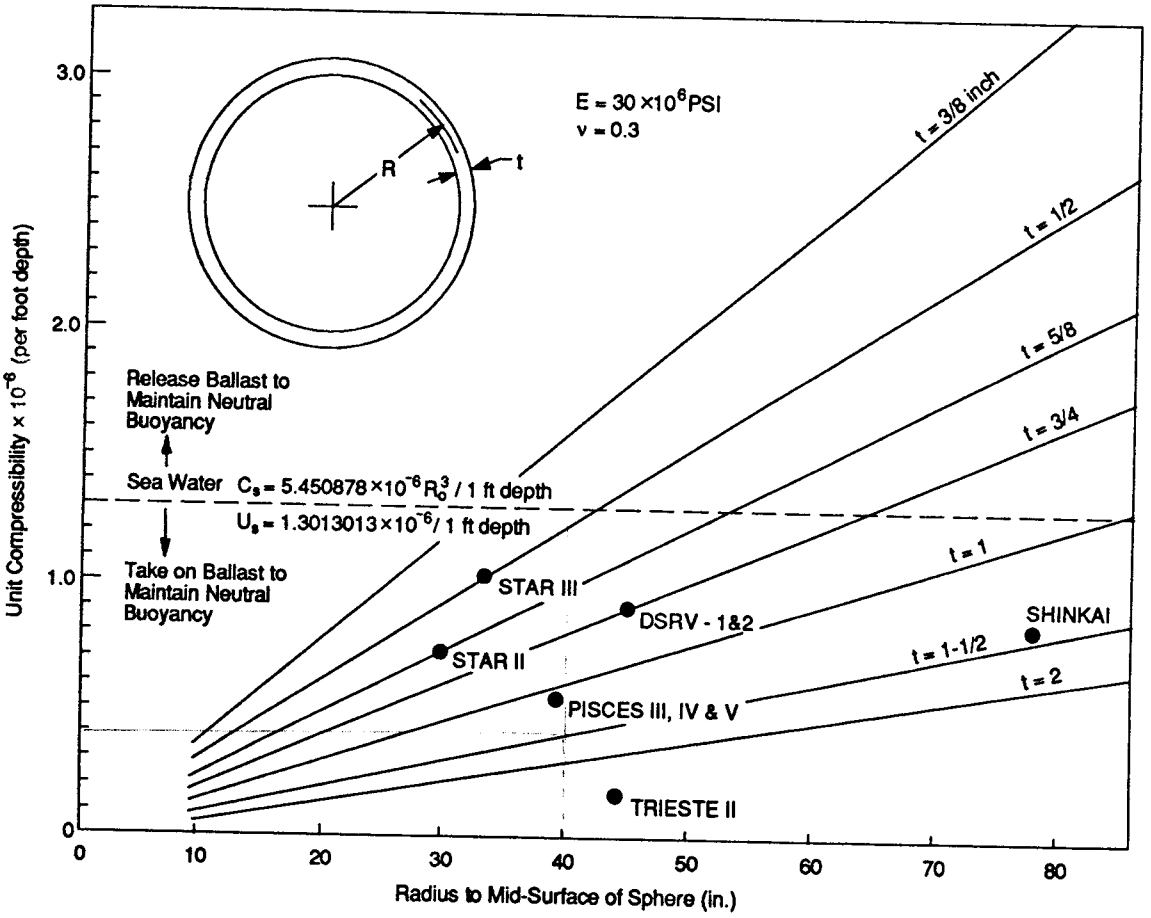


Fig. 50 Unit compressibility of spherical, steel pressure hulls

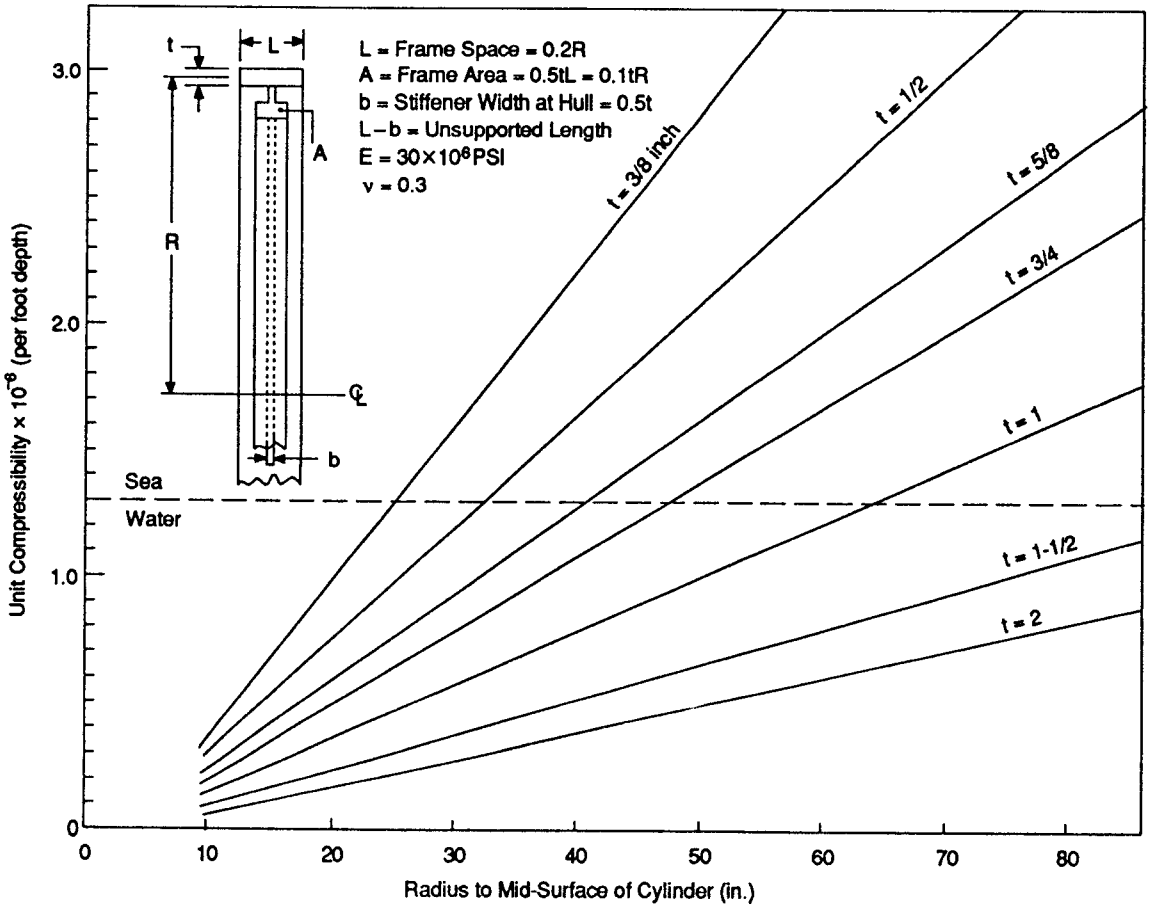


Fig. 51 Unit compressibility of cylindrical, stiffened, steel pressure hulls

ballast sea water which must be taken on (+) or discharged (–) in order to maintain neutral buoyancy with respect to increasing submergence depth can be obtained by the equation:

$$\text{Variable Ballast Change} = C_{sw} - C_{\text{Hull}}$$
$$= D_{\text{Hull}}(U_{sw} - U_{\text{Hull}}) \quad (98)$$

As an example, the steel, personnel sphere of Canada's *Pisces IV* has a 40-in. (1015 mm) external radius and a thickness of 1.038 in. (26.4 mm) [13]. The unit compressibility is 0.554×10^{-6}/ft, as shown in Fig. 50. The approximate amount of sea water which must be taken on as ballast per foot of submergence depth to maintain neutral buoyancy is $\frac{4}{3}\pi(40)^3(1.301 - 0.554) \times 10^{-6} = 0.200$ in.^{3}/ft.

The operating depth of *Pisces IV* is 6600 ft (2012 m) [13], resulting in a total of approximately 0.76 ft^3 (approximately 49 lb), which must be taken on for neutral buoyancy of the personnel sphere at maximum operating depth. *Pisces IV* also contains a balance sphere and two trim spheres.

As another example, the steel pressure hull of *Deepstar 4000* has a 39.5-in. outside radius and is 2.5 in. thick [13]. This corresponds to a unit compressibility of 0.202×10^{-6}/ft. The 4000-ft maximum operating depth requires taking on 0.66 ft^3 (approximately 42 lb) of sea water ballast to maintain neutral buoyancy of the pressure hull sphere, assuming neutral buoyancy existed when fully submerged at the surface.

Nomenclature

A_f = cross-sectional area of ring frame stiffener of cylindrical pressure hull

b = width at attachment of ring frame stiffener to pressure hull

C = compressibility factor, the change in volume per foot of submergence depth

C_c = compressibility factor of a stiffened cylindrical pressure hull

C_s = compressibility factor of a spherical pressure hull

C_{sw} = compressibility factor of a given volume of sea water

C_{Hull} = compressibility factor of a pressure hull

D_{Hull} = the fully submerged displacement volume of a pressure hull at the surface

E = elastic modulus of pressure hull material

h = submerged depth corresponding to submergence pressure P

L = unsupported length of pressure hull plate between ring frame stiffeners = $L_f - b$

L_f = center to center spacing of ring frame stiffeners

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

$$P = h\left(64.0 + \frac{2.6h}{2(30000)\ \text{ft}}\right)\frac{\text{lb}}{\text{ft}^3}\,\frac{1\text{ft}^2}{144\ \text{in}^2}$$

= hydrostatic pressure in psi at depth h

R = radius to mid-surface of pressure hull plating

R_o = radius to outside surface of pressure hull plating

t = thickness of pressure hull plating

θ = hull flexibility parameter

$= [3(1 - \nu^2)]^{1/4} L/(Rt)^{1/2}$

ν = Poisson's ratio

Chapter VI
Structural Principles

R. J. McGrattan and G. A. Peteros

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 "nonsystematic," 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

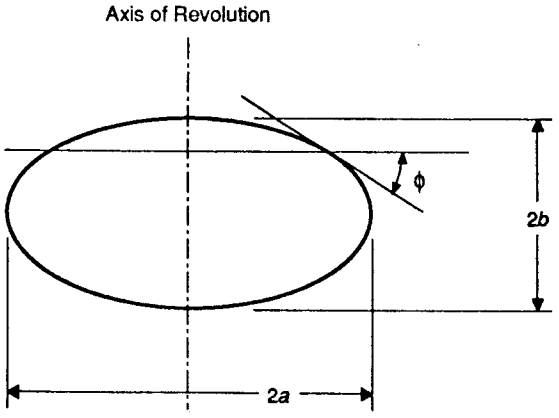


Fig. 1 Oblate spheroid

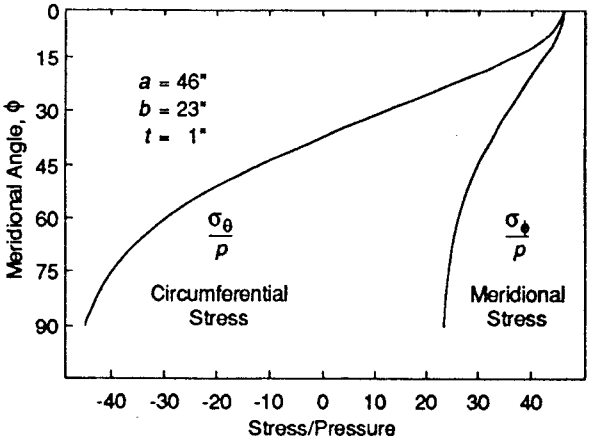
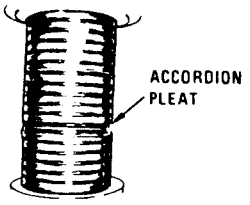


Fig. 2 Oblate spheroid-membrane stresses



YIELDING

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

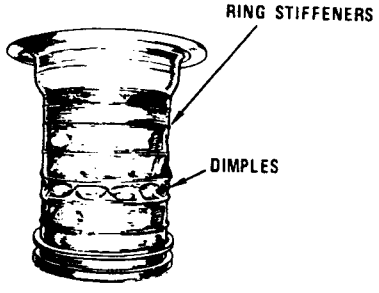
$$A_f \approx \beta_2 L_f t \tag{7}$$

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.

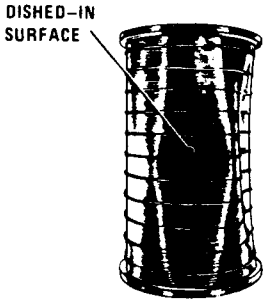
Three collapse pressures are calculated. For axisymmetric yielding (Fig. 3) the collapse pressure p_y is estimated by [1]

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



BUCKLING

Fig. 4 Model failure in shell buckling



GENERAL
INSTABILITY

Fig. 5 Model failure in general instability

where

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

b = width of the frame web

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

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

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

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

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

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

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

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

where

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

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

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

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

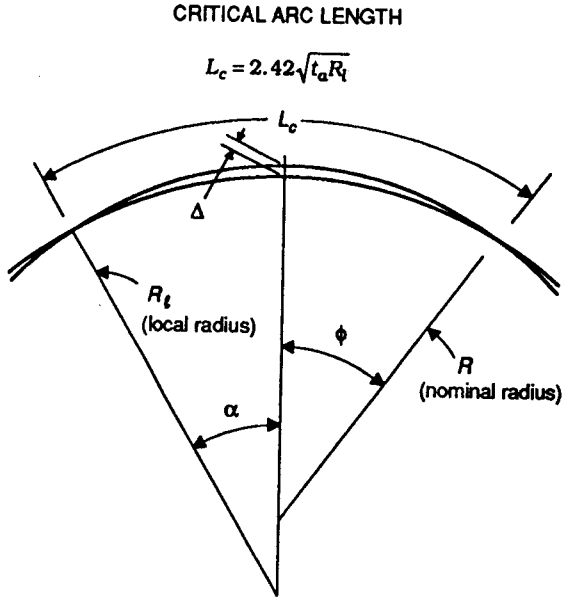
$$p_p = 0.84\sqrt{E_{scc}E_{tan}}\left(\frac{t}{R_o}\right)^2 \quad \text{for } \mu = 0.3 \qquad (12)$$

where E_{scc} 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,



$$\frac{\Delta}{t_a} = \left[(1-\cos\phi) - \frac{R_l}{R}(1-\cos\alpha)\right]\left(\frac{R}{t_a}\right)$$

$$\cos\alpha = \cos\left(1.21\sqrt{\frac{t_a}{R_l}}\right)$$

$$\cos\phi = \sqrt{1-\left(\frac{R_l}{R}\right)^2\left(1-\cos^2\alpha\right)}$$

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

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

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

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

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

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

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

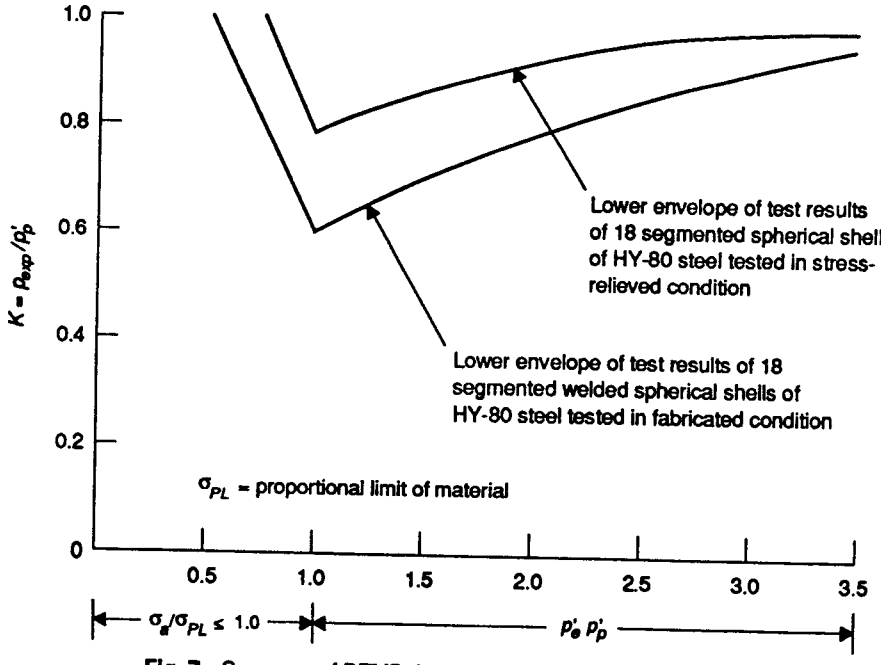


Fig. 7 Summary of DTMB data on unstiffened spherical shells

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

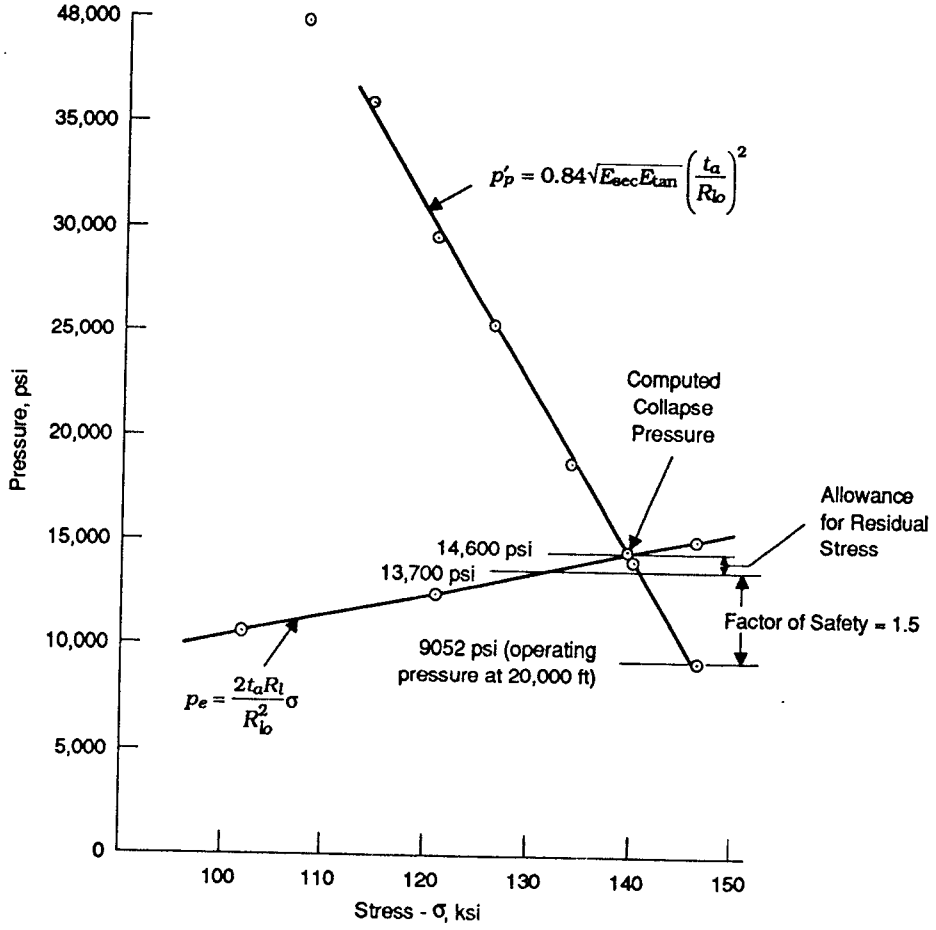


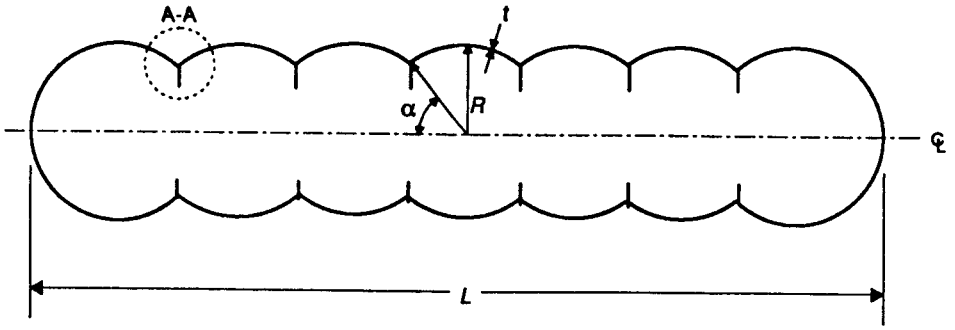
Fig. 8 Collapse pressure for a titanium sphere

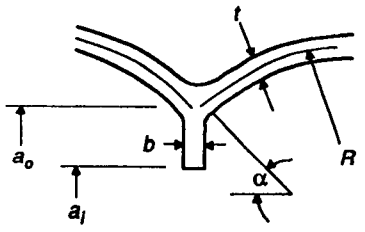
of p'_p versus the pressure corresponding to the average stress σ_a over the critical arc length for a titanium sphere. As a first step, a thickness is approximated from equation (16) by

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

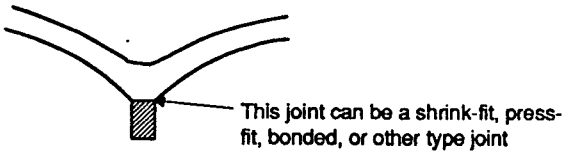
where σ_y is the yield stress and p_c is the design collapse pressure. With this thickness as a reference point, the true collapse pressures can be calculated, using the actual stress-strain plot of the material to compute E, E_{sec}, and E_{tan}. The collapse pressure can be found from the intersection 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 (Typical Intersection-Integral Ring)



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]

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

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

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

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

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

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,

$$E_R(1-\mu_s)\left(\frac{1+C^2}{1-C^2}-\mu_R\right)=K(1-C^2)$$

$$\frac{1+C^2}{1-C^2}-\mu_R=\frac{2E_R(1-\mu_s)}{K(1-C^2)} \qquad (20)$$

where K is the ratio of maximum stress to membrane stress in the shell. The stress in the ring at its inner radius is

$$\sigma_R=\frac{2pR\cos\alpha}{b(1-C^2)} \qquad (21)$$

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

$$b=\frac{4t\cos\alpha}{K(1-C^2)}$$

$$K(1+C^2)-K\mu_R(1-C^2)=2E_R(1-\mu_s)$$

$$C^2(K+K\mu_R)+K-K\mu_R=2E_R(1-\mu_s)$$

$$C^2(1+\mu_R)+(1-\mu_R)=\frac{2E_R}{K}(1-\mu_s)$$

$$C^2(1+\mu_R)=\frac{2E_R}{K}(1-\mu_s)-(1-\mu_R)$$

$$C=\left[\frac{2E_R}{K}\frac{(1-\mu_s)}{(1+\mu_R)}-\frac{(1-\mu_R)}{(1+\mu_R)}\right]^{1/2} \qquad (22)$$

From equations (20) and (22),

$$C=\left|\frac{2E_R}{KE_s}\frac{(1-\mu_s)}{(1+\mu_R)}-\frac{(1-\mu_R)}{(1+\mu_R)}\right|^{1/2} \qquad 0\le C\le 1 \qquad (23)$$

and

$$B=\left(\frac{6t}{KR}\right)\left\{\frac{n\rho_R\cos\alpha\sin^2\alpha+K\rho_s(1+n\cos\alpha)}{\rho_w\left[2+(n\cos\alpha)(2+\sin^2\alpha)\right]}\right\} \qquad (24)$$

For single material construction, the buoyancy factor is

$$B=\left(\frac{6t}{KR}\right)\left(\frac{\rho_s}{\rho_w}\right)\left[\frac{n\cos\alpha\sin^2\alpha+K(1+n\cos\alpha)}{2+(n\cos\alpha)(2+\sin^2\alpha)}\right] \qquad (25)$$

and

$$C=\left[\frac{(2-K)(1-\mu_s)}{K(1+\mu_s)}\right]^{1/2} \qquad 0\le C\le 1 \qquad (26)$$

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

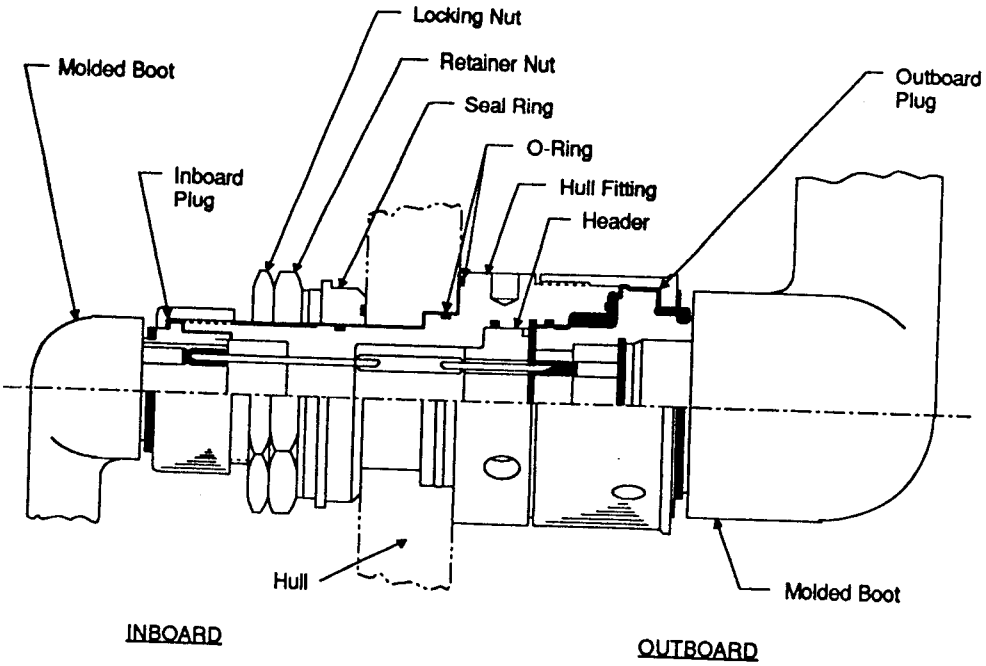


Fig. 10 Electrical hull penetrator

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].

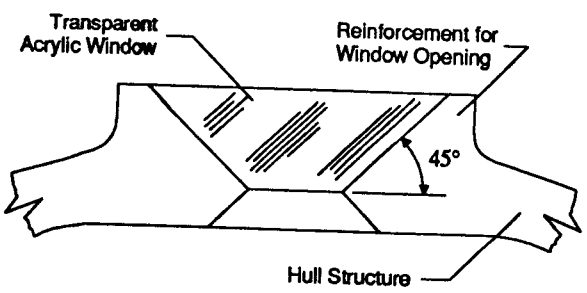


Fig. 11 Viewing port for oceanographic vehicle

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

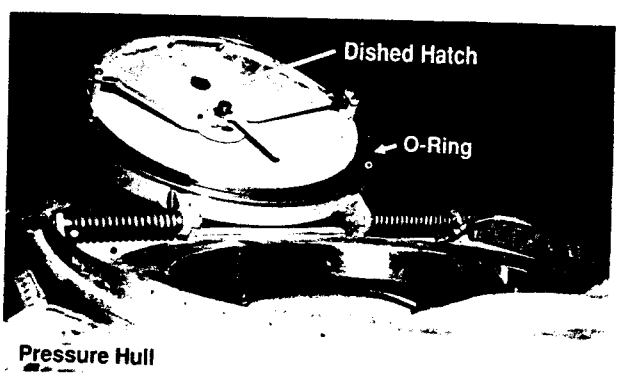


Fig. 12 Seat-type hatch

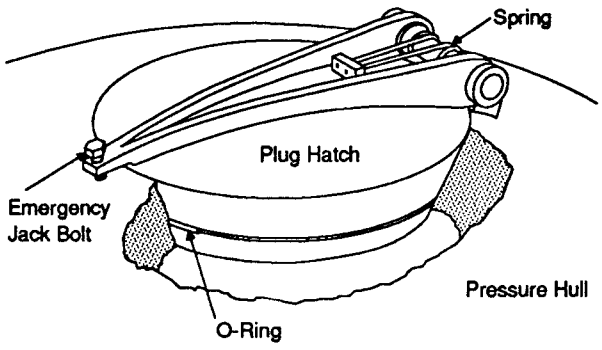


Fig. 13 Plug-type hatch

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

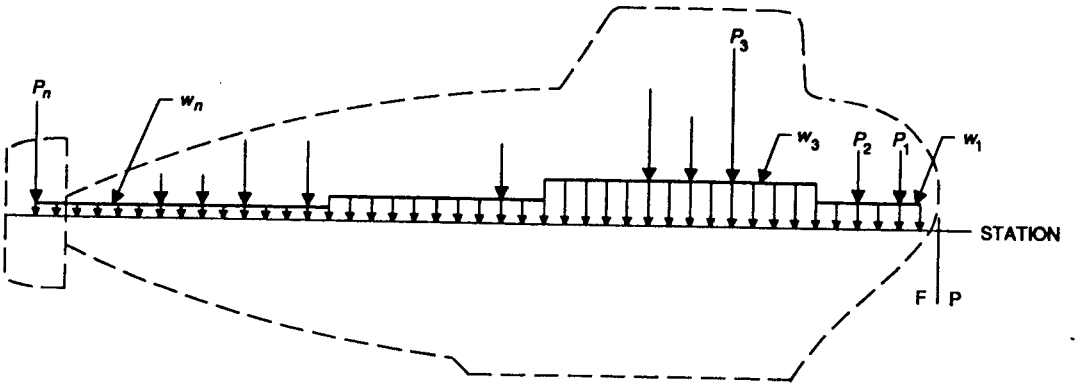


Fig. 14 Typical vehicle weight distribution diagram

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].

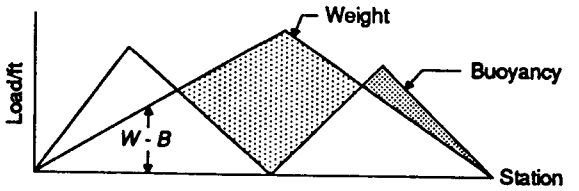


Fig. 15 Vehicle gravity load and buoyancy force distribution

$$R_x = m(\pm g \sin\theta \pm x\dot{\theta}^2 \pm y\ddot{\theta}) \quad (31)$$

$$R_y = m(-g \cos\phi \cos\theta \pm y\dot{\phi}^2 \pm z\ddot{\phi} \pm \dot{\theta}^2 \pm x\ddot{\theta}) \quad (32)$$

$$R_z = m(\pm g \sin\phi \pm z\dot{\phi}^2 \pm y\ddot{\phi}) \quad (33)$$

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.

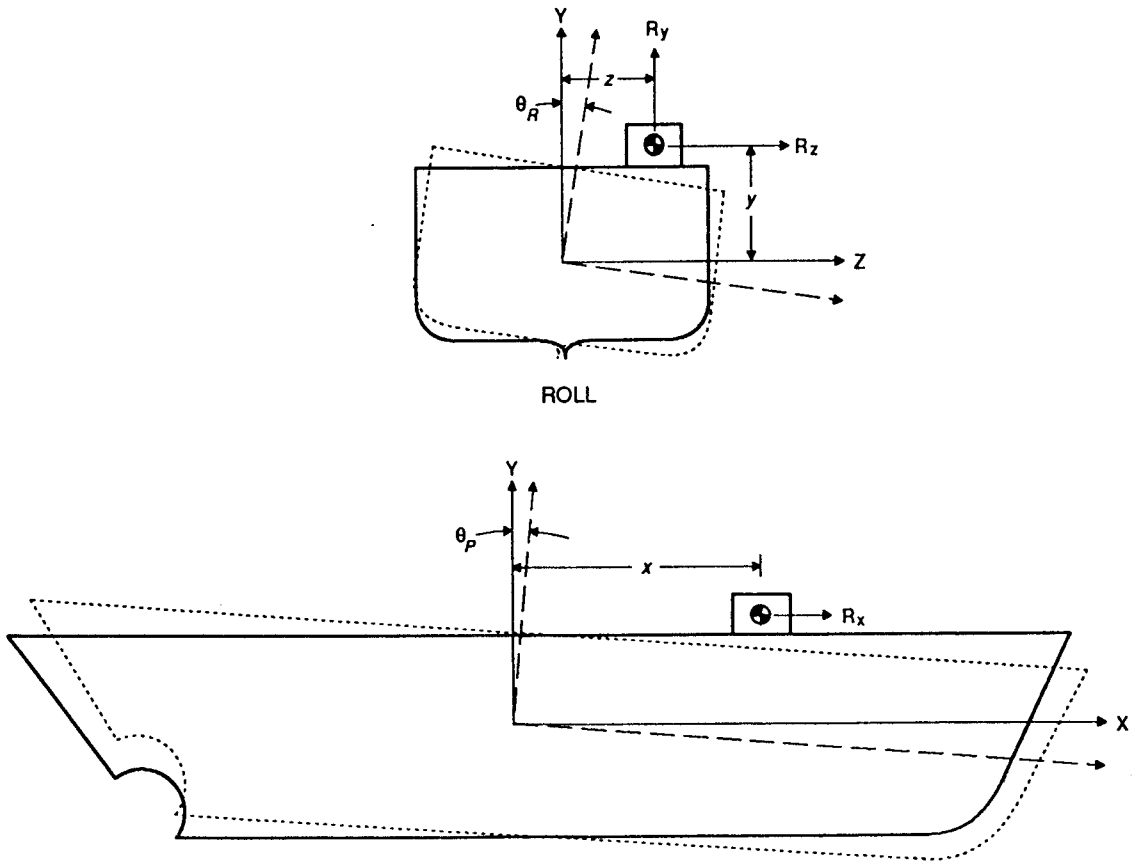


Fig. 16 Ship inertia force reference system

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.

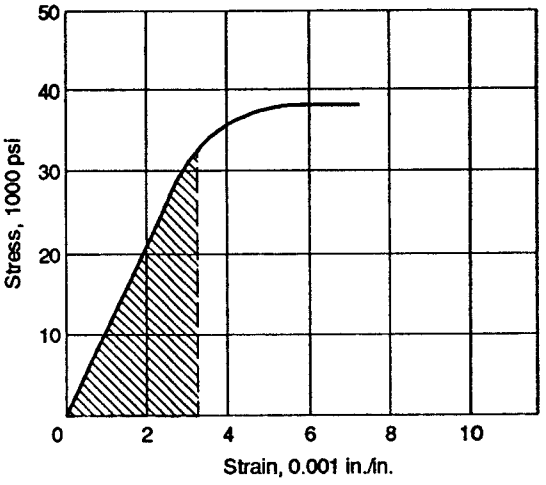


Fig. 17 Stress-strain curve—aluminum 5456-H321

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

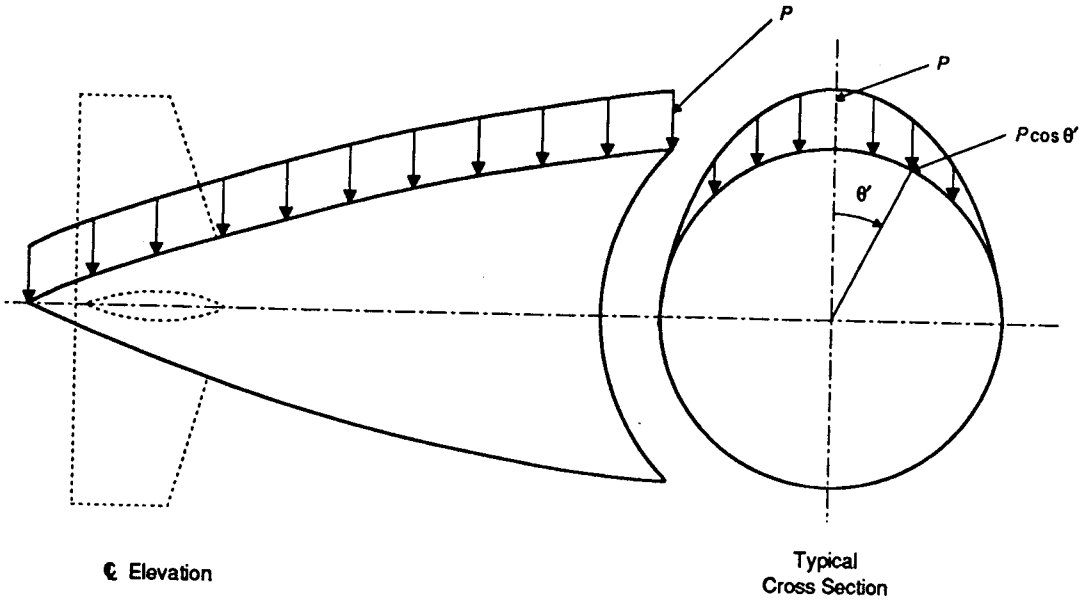


Fig. 18 Sea slap depicted as a cosine distribution

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

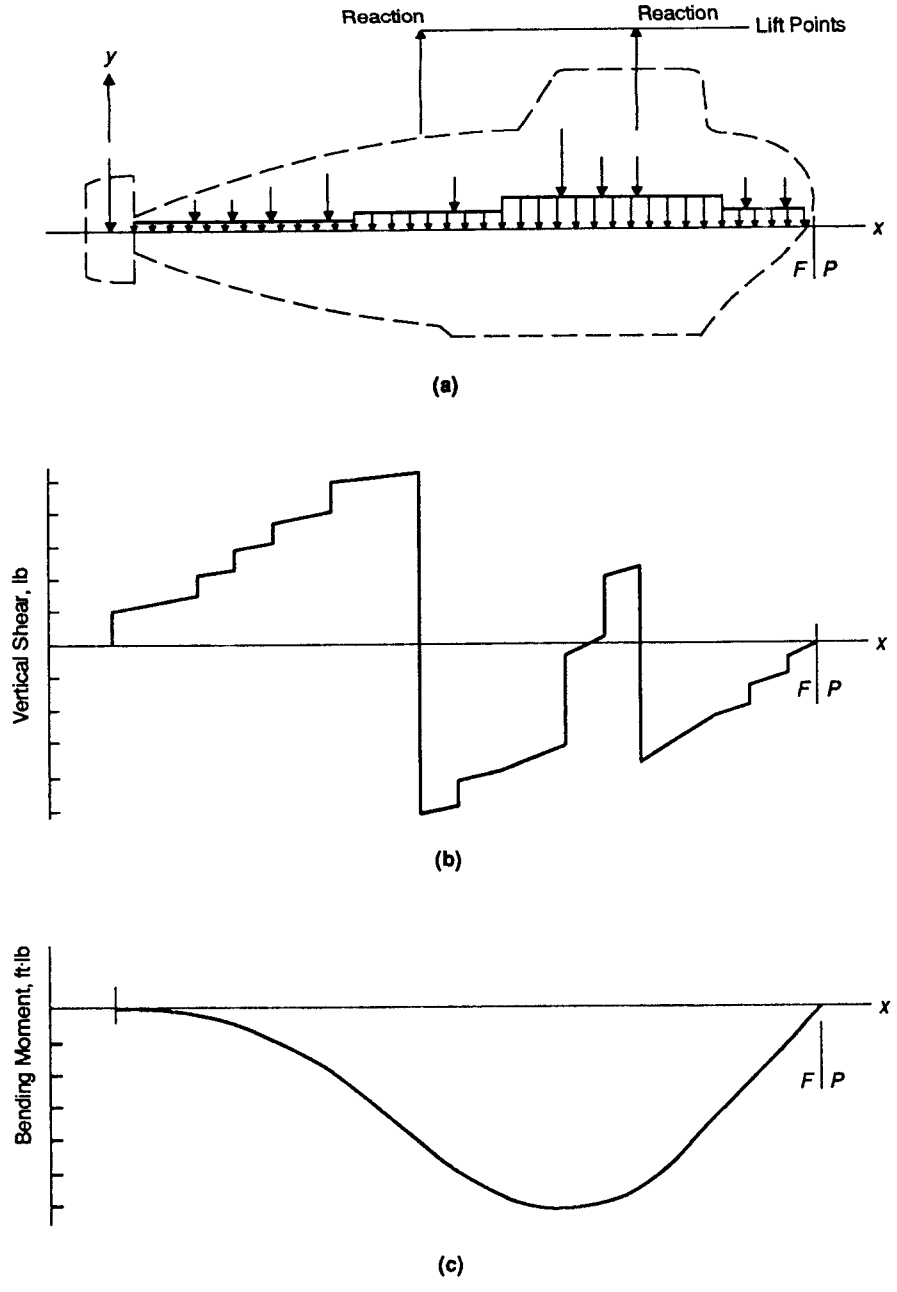


Fig. 19 Typical (*a*) load-lift, (*b*) shear, and (*c*) bending moment diagrams

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,

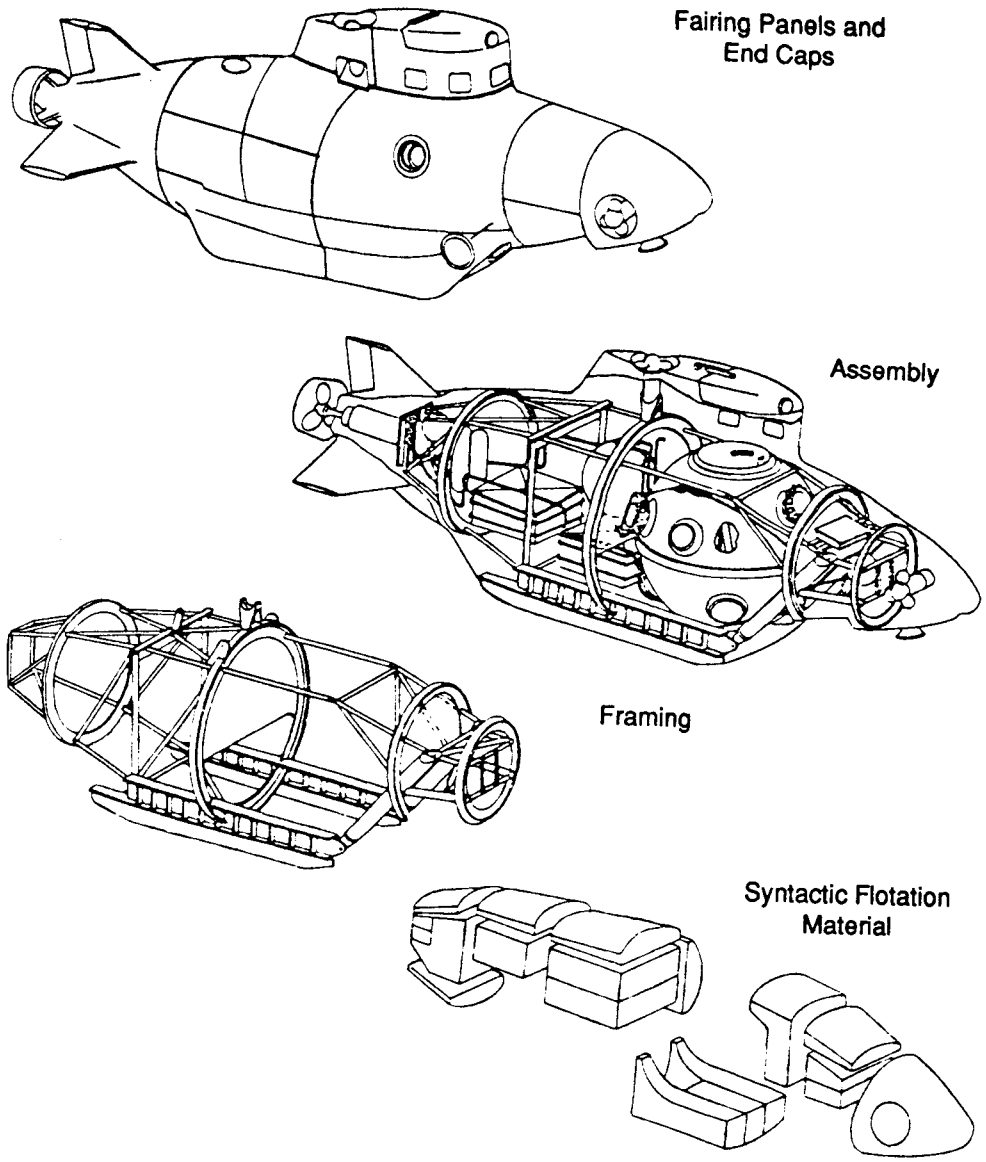


Fig. 20 Framed exostructure

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.

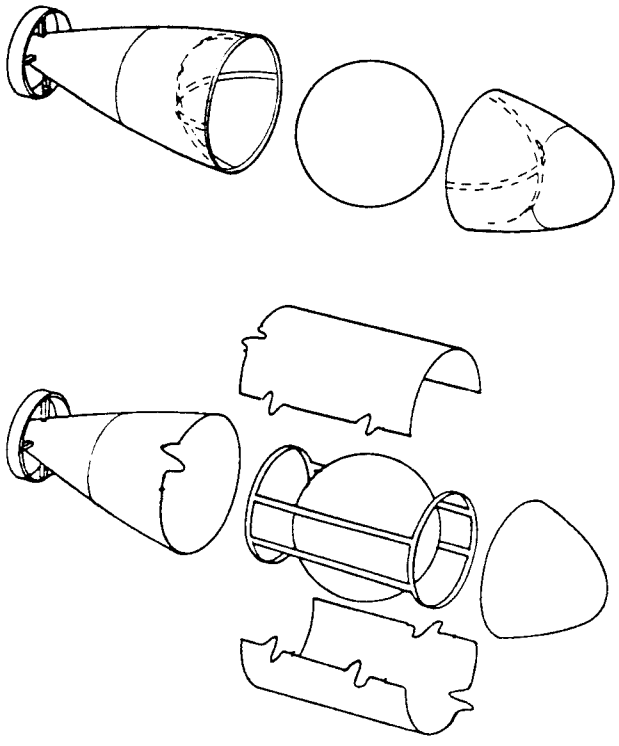


Fig. 21 Stiffened (*top*) and unstiffened (*bottom*) shell exostructures

While a framed structure affords the best access to enclosed equipment for the reasons cited earlier, certain interior arrangement and structural weight complexities arise from the orientation of the structural members in the framing system. In order to minimize the weight of framed structures (that is, to maximize structural efficiency) the members of the framework must be straight. As a consequence, such members normally will not conform with the curvature of the outer shell, and will thus intrude into interior space, which is needed for equipment, cables, and piping. Alternatively, the structure members may be curved to match the outer shell shape, but in so doing, eccentricities will be created along the longitudinal axes of the members, which give rise to secondary stresses that must be accommodated by larger, heavier members.

3.3.2 Stiffened Shells

This type of exostructure is characterized by a continuous (that is, unsegmented) shell in combination with internal stiffeners of the same or a different material that act integrally with the shell. Stiffeners may be of the ring (circumferential) type, commonly called transverse frames, spaced at intervals along the length of the shell, or a combination of transverse frames and longitudinal stiffeners (stringers). The frames and stringers serve to support the shell and increase shell strength by effectively subdividing the shell into smaller, more stable segments, and provide an efficient load transfer path for concentrated loads.

Because the shell acts integrally with the frames and stringers (if any) and consequently contributes to overall structural strength, stiffened-shell construction generally offers the least-weight outer hull. It also usually provides greater unobstructed interior space and more efficient support for attachment (foundationing) of equipment than a framed structure. More usable space results because the stiffeners conform to the shell curvature, whereas in framed structures with straight members, they do not. The superior foundationing capabilities of the stiffened shell are made possible by (1) the ability to employ stringers readily, spanning between transverse frames at any location so as to support equipment and (2) the absence of any of the load location limitations associated with the space truss framing system.

Unlike the framed structure, in which it is possible to remove the entire shell for service access, there are limitations on the number, size, and location of access openings in a stiffened shell. This is because the shell in a stiffened-shell structure is the main load-carrying member, and any removable panel provided for access in effect creates a hole in the shell, which has a weakening effect. While shell structure surrounding such openings can be reinforced to compensate for local loss of strength, there are practical limits to the extent that this can be done without compromising vehicle structural integrity or inordinately increasing the weight of the structure. This limitation on access openings, which makes servicing and replacement of equipment less than ideal, is a singular disadvantage of this type of exostructure.

3.3.3 Combined Structures

It follows from the foregoing discussion of the advantages and disadvantages of framing system alternatives that it might be useful in some cases to use different types of structure in different parts of the outer hull. For instance, a space frame structure could be used in the bow where access to sensors, underwater lights, tools, cameras, etc., is important; or the whole bow might be a removable monocoque shell (see Section 3.7). For the same vehicle, a stiffened shell fairing might be selected in way of the pressure hull, in order to reduce overall vehicle weight or size. (See, for example, Fig. 21.)

In the discussion that follows, information and guidance are presented for performing concept or preliminary designs for each of the framing systems noted above. In the technical data provided, a number of simplifying assumptions are made, and many of the equations presented are conservative approximations to the solution of complex problems. Further analytical refinements and the development of discrete structural details usually are left to the later stages of design.

3.4 Design Procedure for Framed Structures

Once the intended service and load conditions have been defined, the design of a framed structure begins by selecting a suitable framing material, and assuming sizes and locations of framing members. The location of framing members is influenced greatly by the arrangement of major components within the vehicle's envelope. Ideally, the exostructure should be framed in such a way as to provide an effective load path for the larger and heavier equipment. Launching and retrieval, load condition (A3), in particular, will be a determinant of where major transverse framing members are placed, since the vehicle-lifting lugs will be attached to those members.

Once an initial framing configuration has been established, a computer model (mathematical representation) of the structure is developed which considers the applied loads from each of the loading conditions described earlier, in preparation for a computer solution of stresses and deflections throughout the structure. Both space truss and space frame designs may be readily examined, using the same computer model and loading conditions, by the simple expedient of specifying in the computer input instructions "releases" at each joint for a space truss design, and "fixity" at every joint for a space frame design. The two joint conditions will require separate computer runs, which easily can be done in series. See Fig. 23 and Fig. 24 for a typical space frame structure and its model.

It is perhaps worth noting that space trusses are more amenable to hand analysis than space frames, which in the past might have had some influence on the choice of framing systems. In any case, hand solution of either is cumbersome at best and is discouraged, particularly when multiple loading conditions must be assessed. (See any standard text on the analysis of framed structures for a treatment of this subject, for example, reference [19].) Fortunately, there are many special and general-purpose structural analysis computer codes now available to facilitate the solution of such problems, including multiple loading conditions, for example, STRUDL. The STRUDL computer code [20], which employs a problem-oriented language for structural design, is perhaps more "user friendly" than most general-purpose computer codes, and is recommended for those who have had little or no experience with such codes.

The reader is cautioned that most structural analysis computer codes do not account routinely for the nonlinear, beam-column effects cited earlier. This is of particular importance in the analysis of space trusses with laterally loaded members. Typically, most computer codes treat laterally loaded members by replacing such loads with statically equivalent loads acting at the joints, which results in axial tension or compression forces in the members. The user then is obliged to extract the member forces

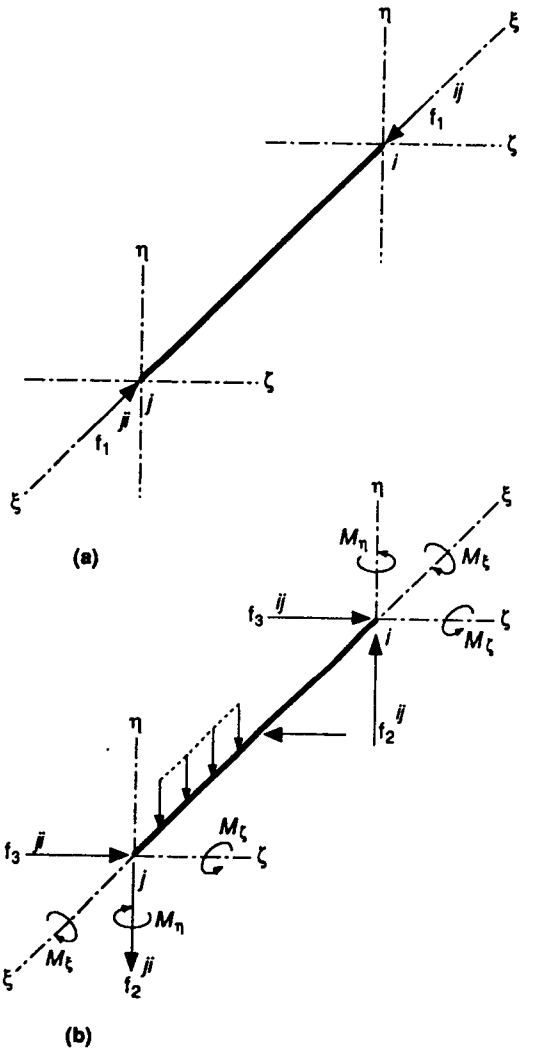


Fig. 22 Space (*a*) truss and (*b*) frame member forces

thus obtained from the computer output and, in conjunction with the lateral loads, analyze each such member as a free body in order to account for geometric nonlinearities.

In general, a framed-structure outer hull will be somewhat heavier and will occupy more interior space than a stiffened shell. For any weight comparisons between space trusses and space frames or, for that matter, any framing system, it is important also to consider equipment foundation weights for each configuration, since foundation weights frequently are affected by the choice of the framing system (see Section 3.6).

3.5 Design Procedures for Stiffened Shells

As noted earlier, a number of simplifying assumptions are made herein, consistent with the objectives of a conceptual or preliminary design. Some of the

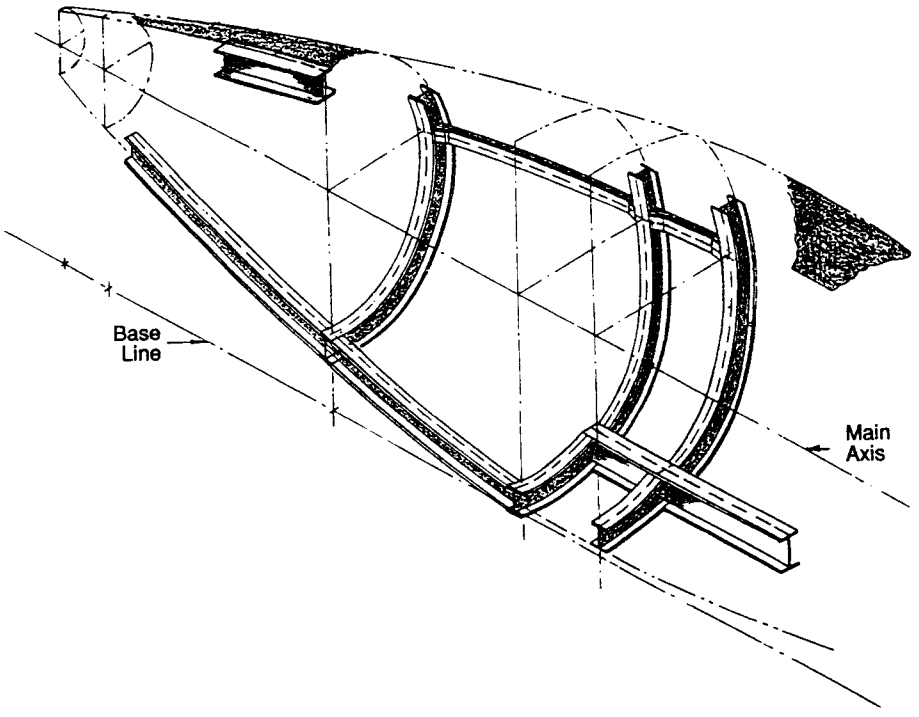


Fig. 23 Space frame exostructure

assumptions associated with stiffened shell exostructures are:

- In general, except where noted otherwise, linear-elastic behavior of all structural elements is assumed.
- Bending stress distribution in the shell and stiffeners is assumed to be in accord with elementary beam theory.
- Shell equations are based on thin-shell theory, with the range of applicability being $t/R < 1/20$. (Most, if not all, exostructure shells will be found to fall into this range.)
- Shell buckling equations presented are semi-empirical. The theoretical equations are complicated, and a computer program is required to calculate buckling loads accurately. Much of the complexity arises because the buckling strength of thin shells is sensitive to geometric imperfections and, in some cases, to the supporting boundary conditions.

It also was noted that, although there is a wide range of potential exostructure external shapes, many such structures either possess the form of, or can be approximated adequately by, cylinders or shallow-angle cone elements. These will be the forms addressed in the discussion that follows, for which bending of the shell is the dominant mode of behavior from the design loadings described earlier. For our purposes, the design information presented here for cylinders also can be applied to cones by use of an equivalent-cylinder method, that is, by converting the length and radius of the conical shell into that of an equivalent cylinder as shown by Fig. 25, and using the same stress equations as for cylinders.

The exostructure features that must be defined are: shell and stiffener material; shell thickness; frame size and spacing; and, if stringers are used, stringer size and circumferential spacing. Before proceeding, it might be well to review the role or function that each of these structural elements serve. The shell is designed to resist the applied loads described above by a system of normal and shearing stresses in the shell. Transverse frames serve a threefold purpose: to transfer concentrated loads to the shell; to increase the overall buckling strength of the shell; and, when stringers are used, to provide column and end support for the stringers. Stringers are sometimes used to reduce shell thickness by increasing the moment of inertia of the exostructure cross section, to increase local buckling strength of the shell, and to transfer concentrated loads to the frames and shell. Their use generally is decided by structural efficiency (weight) and cost considerations. Ideally, the structure should be designed in such a way that all modes of failure have an equal probability of simultaneous occurrence, but such attempts invariably prove futile and impractical.

The structure to be designed must be of sufficient strength, of course, to resist applied loads without yielding or buckling. Not only must stresses not

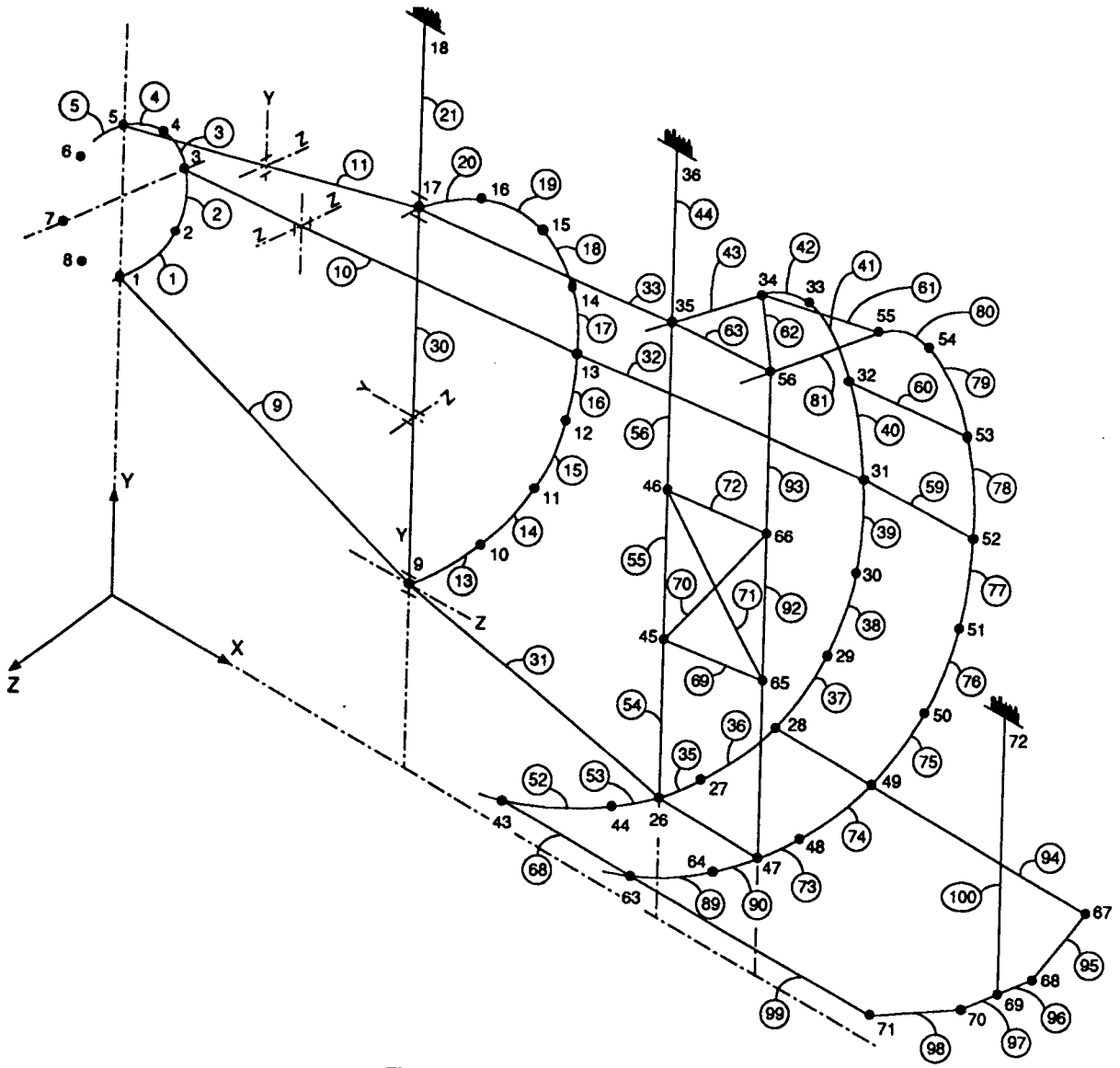


Fig. 24 Space frame computer model

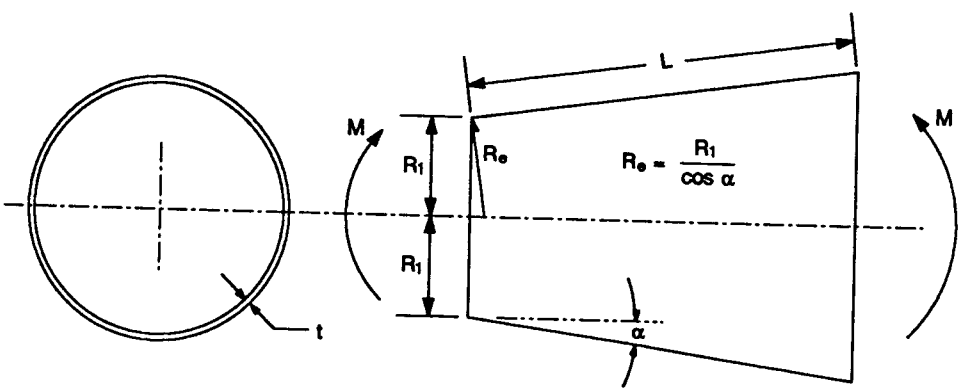


Fig. 25 Equivalent cylinder representation of a conical shell

exceed the yield strength of the material, but they must be kept within some prescribed limit, at a value less than the yield strength, in order to preclude failure by fatigue under repeated loading. In some instances, stiffness becomes a governing consideration in order to limit the elastic deflection of the structure. For the purposes of concept or preliminary design, for which only the gross strength of the structure is of interest, without regard for secondary or discrete discontinuity stresses at local details, some safety factor (often chosen is 1.5) is assigned to both applied stress and buckling values in order to ensure the feasibility of the structure.

After selection of the shell and stiffener materials, initial sizing assumptions are made for shell thickness and frame size and for spacing with and without stringers. The weight of the assumed structure is calculated and combined with the various vehicle system weights, in order to develop a longitudinal weight distribution diagram, as shown, for example, in Fig. 15. This weight distribution diagram then is combined with each of the loads discussed earlier, in order to formulate shear and bending moment diagrams for the various loading conditions. (For a treatment of shear and moment diagrams, see any standard text on strength of materials, for example, reference [21].) Shear and moment values thus obtained are then used to calculate shear and bending stresses and, where appropriate, deflections. These calculated values of stress are compared with allowable values in order to evaluate the adequacy of the assumed structural scantlings. If calculated stress values are unsatisfactory (either too low or too high), the scantlings are revised and stresses recalculated until a satisfactory design is attained.

3.5.1 *Shell Design*

Shear and bending stresses in the shell are calculated from the well-known beam equations (found in any standard text on strength of materials)

$$\text{Deflection:}\quad \Delta = \iint \frac{M}{EI}\,dx^2 + Ax + B \tag{36}$$

$$\text{Rotation:}\quad \theta = \frac{dy}{dx} = \int \frac{M}{EI}\,dx + A \tag{37}$$

$$\text{Moment:}\quad M = EI\frac{d^2y}{dx^2} = \iint w\,dx^2 + Ax + B \tag{38}$$

$$\text{Shear:}\quad V = EI\frac{d^3y}{dx^3} = \frac{dM}{dx} \tag{39}$$

$$\text{Load:}\quad F = EI\frac{d^4y}{dx^4} = \frac{dV}{dx} = \frac{d^2M}{dx^2} \tag{40}$$

Bending stress σ_b at any point in the shell is given by the well-known flexure equations

$$\sigma_b = \frac{M_y}{I_x} \tag{41}$$

$$\sigma_b = \frac{M_x}{I_y} \tag{42}$$

The bending stress about axis x–x is given by equation (41), while the bending stress about axis y–y is given by equation (42). (If σ_b exceeds the proportional limit of the material, these equations no longer apply.) I_x and I_y are principal moments of inertia; x and y are distances measured from the principal axes. The moment of inertia I of a thin-walled circular cross section can be approximated by

$$I_x = I_y = \pi R_s^3 t \tag{43}$$

where R_s is the radius to mid thickness of the shell and t is much less than R (see Fig. 26). Where stringers are used and are of the same material as the shell, normal stresses in the shell and stringers are calculated in the same manner as above, except for the moment of inertia I, which is calculated by the equation

$$I = \pi R_s^3 t + \sum A_s d^2 \tag{44}$$

where A_s is the cross-sectional area of each stringer and d is the distance from the neutral axis to the centroid of any stringer cross section. If the stringers and the shell are not of the same material (for example, glass-reinforced plastic shell and metal frames and stringers), the equation for calculating the moment of inertia of the stringer-shell cross section, equation (44), must be modified as shown in equation (45) below to account for the different moduli of elasticity of the two materials,

$$I^* = \pi R_s^3 \left(\frac{E_{sh}}{E_{str}}\right) t + \sum A_s d^2 \tag{45}$$

where E_{sh} is the modulus of elasticity of the shell material and E_{str} is the modulus of elasticity of the stringer material. Axial stresses in the stringers are then calculated using equation (41) or (42) with the revised moment of inertia I^* given by equation (45) as

$$\sigma_b = \frac{M_{dn}}{I^*} \tag{46}$$

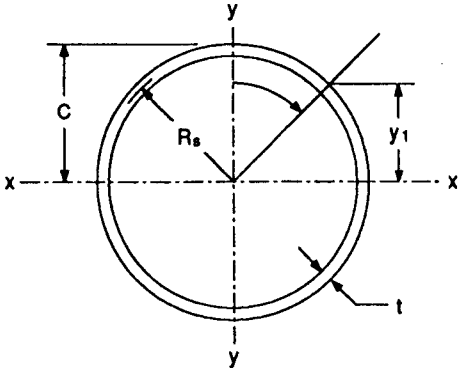


Fig. 26 Thin-walled circular shell cross section

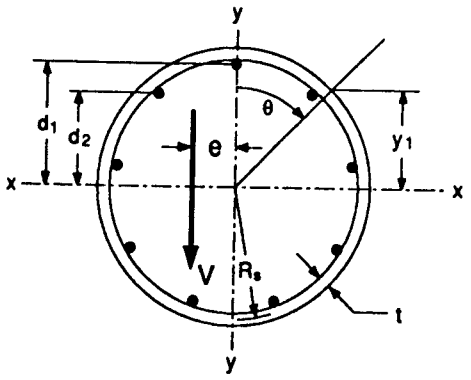


Fig. 27 Asymmetrical loading condition in the shell

while shell stresses are calculated from the equation

$$\sigma_b = \frac{M_y}{I^*}\left(\frac{E_{sh}}{E_{str}}\right) \tag{47}$$

For loading conditions in which the exostructure is loaded symmetrically with respect to the centerline of the shell, the shear stress τ at any point in the shell is calculated by the equation:

$$\tau = \frac{V}{It}\int_{y_1}^{c} y\, dA \tag{48}$$

For a thin shell of circular cross section,

$$y = R\cos\theta \quad \text{and} \quad dA = Rt\, d\theta \tag{49}$$

and equation (48) becomes

$$\tau = \frac{V}{\pi Rt}\sin\theta \tag{50}$$

If the load is asymmetric to the shell's axis of symmetry as shown in Fig. 27, the load may be resolved into a load acting at the centerline, and a couple equal to the load multiplied by the eccentricity *Ve* as

$$q = \tau t = \frac{V}{I}\int y\, dA + \frac{Ve}{2A} \tag{51}$$

where A is the area enclosed by the shell in the vertical plane; it is equal to πR_s^2. Instability or buckling strength also must be assessed in order to ensure the adequacy of the structure.

The critical buckling stress of a cylindrical shell between transverse frames under bending load can be calculated from

$$\sigma_{cr} = C_b\frac{Et}{R_s} \tag{52}$$

where

σ_{cr} = critical buckling stress of the shell,
E = modulus of elasticity of shell material,
t = shell thickness,
R_s = radius to mid thickness of shell, and
C_b = buckling stress coefficient given in Fig. 28.

The critical shell bending load can be expressed in terms of critical bending moment M_{cr}, where

$$M_{cr} = \pi R^2\sigma_{cr}t \tag{53}$$

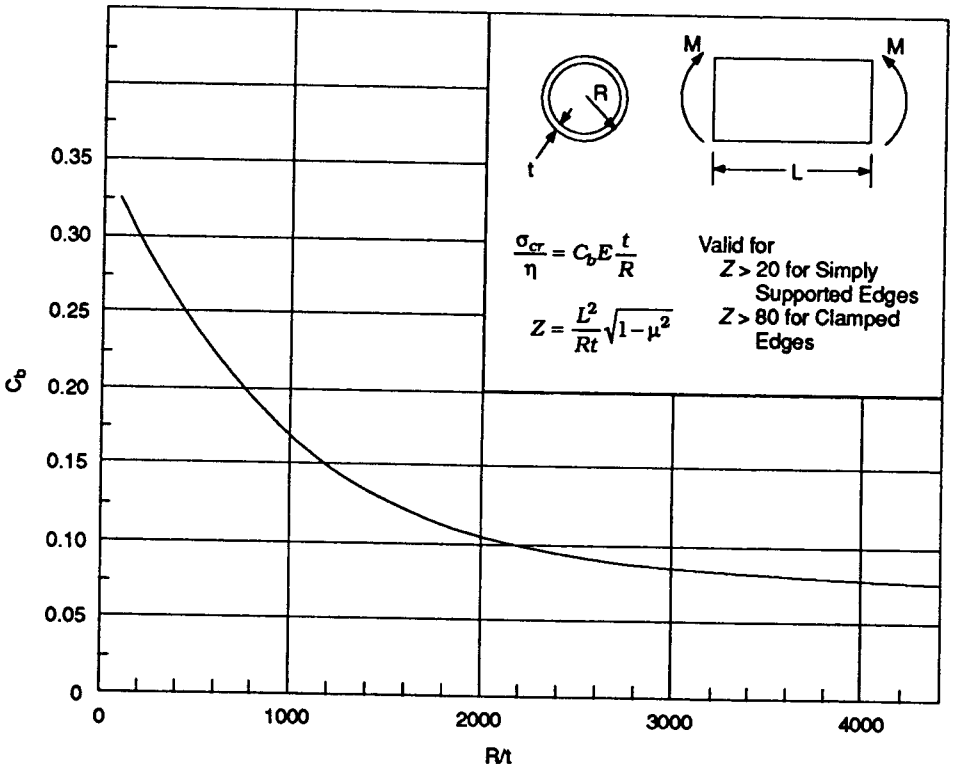


Fig. 28 Buckling coefficients C_b for cylinders

For conical shells, using the equivalent cylinder method described earlier, equation (53) becomes

$$\sigma_{cr} = C_b \frac{Et}{R_e} \tag{54}$$

where

σ_{cr} = maximum stress at the small end of the cone,
R_e = equivalent cylindrical radius (see Fig. 25), and
C_b = buckling stress coefficient given in Fig. 29.

and equation (53) becomes (see Fig. 29)

$$M_{cr} = \pi R_1^2 \sigma_{cr} t \cos \alpha \tag{55}$$

If stringers are used, the critical buckling stress of the shell panel, loaded in axial compression, and bounded along its sides by stringers and at its ends by transverse frames, is calculated from

$$\sigma_{cr} = K_c \frac{\pi^2 E}{12(1-\mu^2)} \left(\frac{t}{b}\right)^2 \tag{56}$$

where

b = panel breadth (circumferential direction),
t = shell thickness,
μ = Poisson's ratio, and
K_c = buckling stress coefficient given in Fig. 30.

3.5.2 *Frame Design*

In framed structures, the location of frames in a stiffened shell is influenced greatly by the arrangement of major components and the manner in which they are to be supported in the exostructure. Ideally, the exostructure should be framed in such a way as to provide an effective load path for the heaviest components while maximizing the structural efficiency of the outer hull in a gross sense. The location of the pressure hull, and lifting points for the vehicle, in particular, will determine the largest frame sizes and locations.

Each frame should be designed with sufficient strength not only to support the imposed loads safely without yielding or excessive deformation, but to preclude a general instability mode of failure characterized by simultaneous buckling of shell, stringers (if any), and frames, resulting in a gross inward bulge on the compression side of the stiffened shell.

In deciding on frame size, it is best first to calculate the stiffness required of the frames in order to preclude a general instability mode of failure, since this determines the minimum frame size required. Frame strength to sustain imposed concentrated loads safely then should be checked. The minimum frame stiffness EI_f required to preclude general instability can be calculated from

$$EI_f = (6.84 \times 10^{-5}) \frac{M(2R)^2}{L} \tag{57}$$

where

E = modulus of elasticity of the frame material,

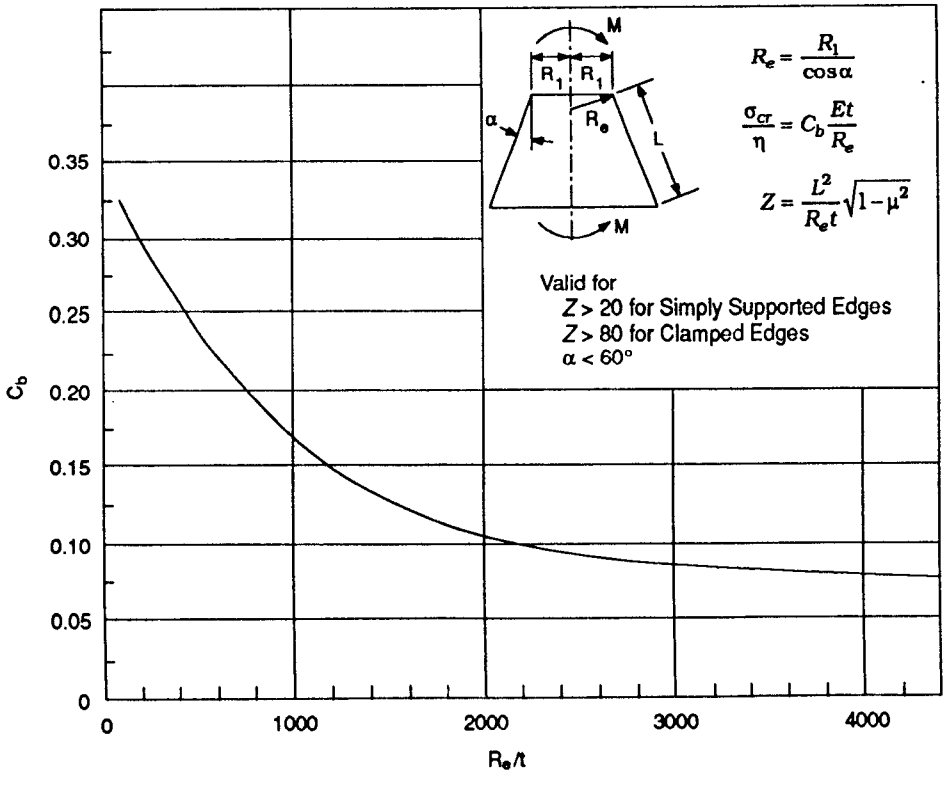


Fig. 29 Buckling coefficients C_b for cones

I_f = moment of inertia of frame cross section,
M = applied bending moment (from moment diagram), and
L = frame spacing.

Stresses and deflections in the frame produced by concentrated loads then can be calculated as follows: For a single concentrated load P acting through a vertical axis of symmetry, bending moments, shear and axial forces, deflections, and rotations can be calculated readily from the following equations in conjunction with the coefficients given in Fig. 31.

$$M = K_M PR \tag{58}$$

$$V = K_V P \tag{59}$$

$$F_a = K_{F_a} P \tag{60}$$

$$\Delta = K_\Delta \frac{PR}{EI_f} \tag{61}$$

$$\theta = K_\theta \frac{PR}{EI_f} \tag{62}$$

where, at any location ϕ on the frame

M = bending moment,
V = shear force,
F_a = axial force,
Δ = radial deflection,
θ = rotation,
K_M = coefficient of bending moment,
K_V = coefficient of shear force,
K_{F_a} = coefficient of axial force,
K_Δ = coefficient of radial deflection,
K_θ = coefficient of rotation,
R = radius to centroid of the frame cross section, and
I_f = frame moment of inertia.

Bending stress in the frame σ_b can be calculated using equations (58) and (41), while shear stresses, τ, in the frame are calculated using equations (59) and (48). Bending moment, shear, axial force, deflection, and rotation equations for other loading conditions can be found in reference [22].

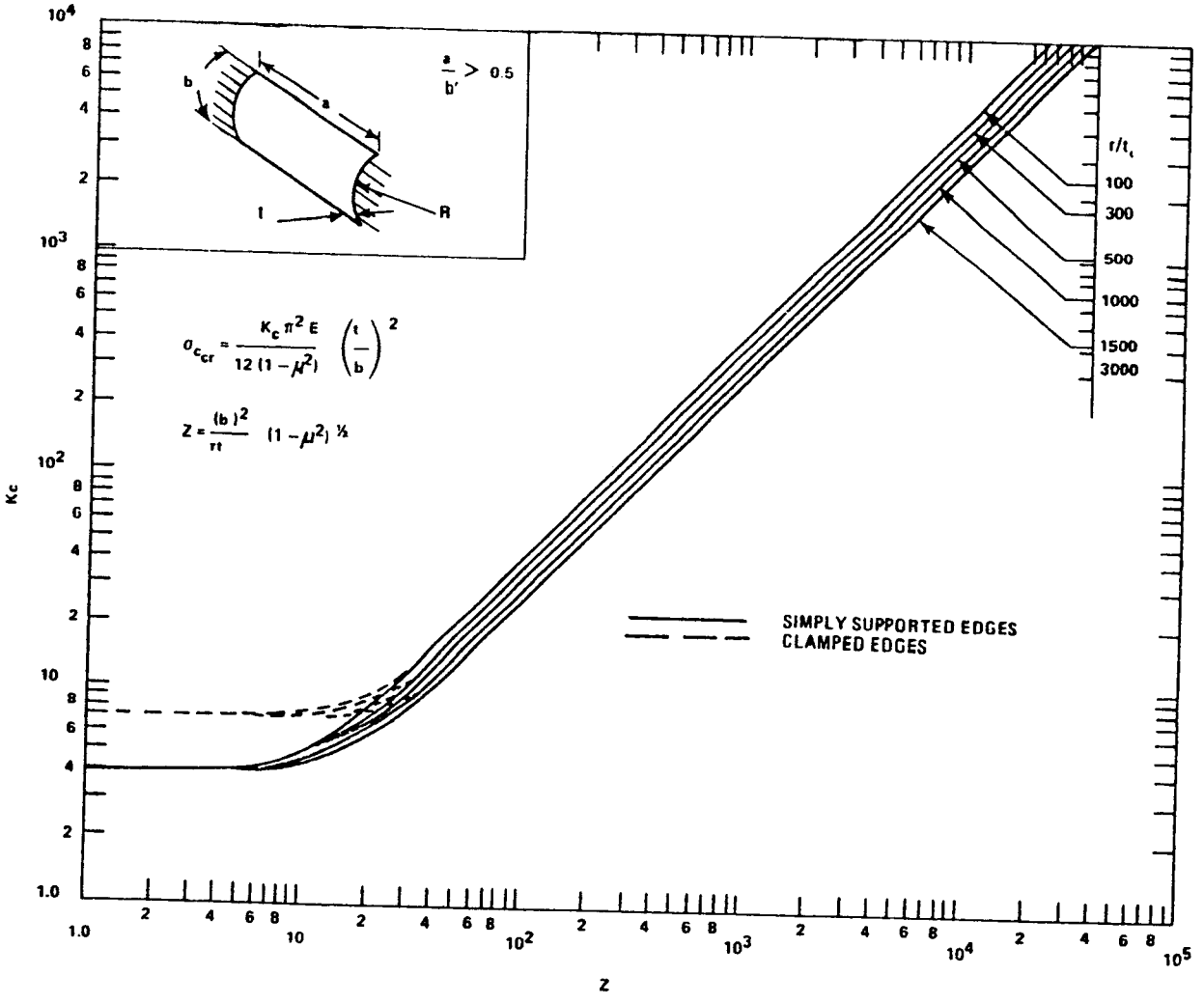


Fig. 30 Buckling stress coefficients K_c for unpressurized curved panels subjected to axial compression

3.5.3 *Stringer Design*

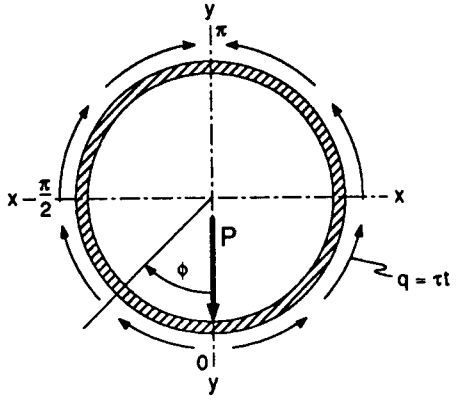
Stringers must be designed to carry the axial forces resulting from the overall bending of the stiffened shell, as well as local concentrated or distributed lateral forces or both, as illustrated in Fig. 32. For the load condition in which only axial forces act on the stringers, this is essentially a column buckling problem, in which an effective width of shell b_e acts with the stringer to behave as a column of length L equal to the transverse frame spacing. The critical buckling stress can be predicted conservatively from the equation

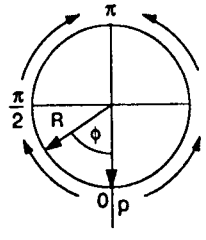
$$\sigma_{cr} = \frac{\pi^2 E I_s}{A_s L_f^2} \tag{63}$$

where

I_s = moment of inertia of the stringer including an effective width of shell b_e, as shown in Fig. 33,

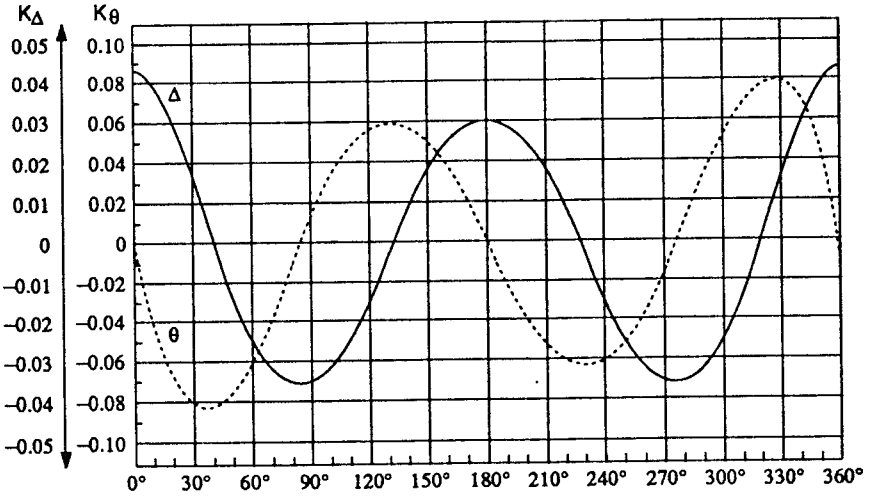
A_s = cross-sectional area of the stringer including an effective width of shell b_e, as shown in Fig. 33, and

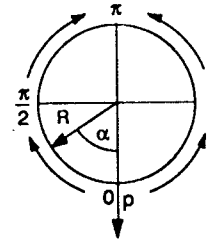




$$\Delta = K_\Delta \frac{PR^3}{EI}$$

$$\theta = K_\theta \frac{PR^2}{EI}$$





$$M = K_M PR$$

$$V = K_V P$$

$$N = K_N P$$

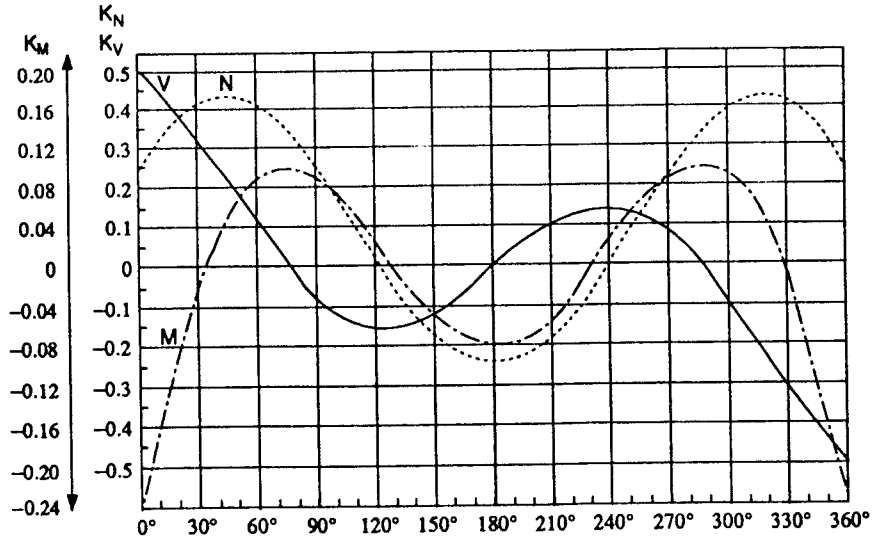


Fig. 31 Frame coefficients

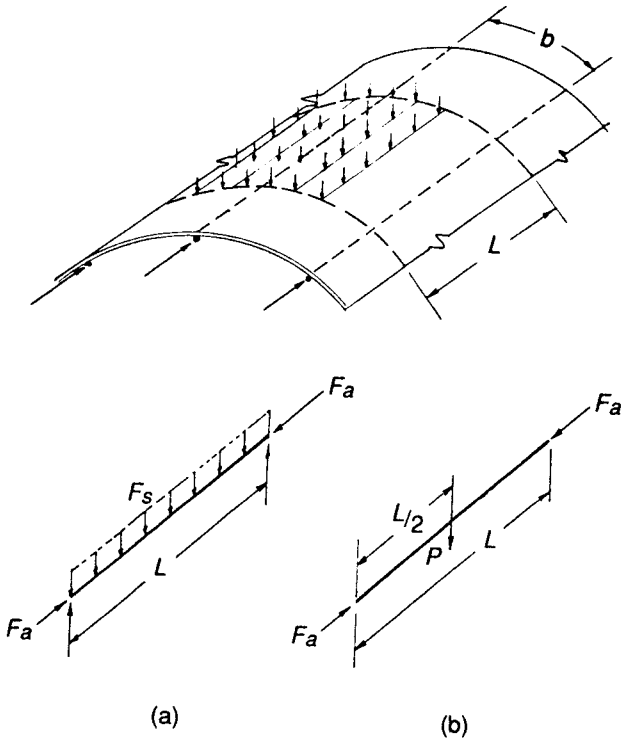


Fig. 32 Stringer loadings

L_f = length of stringer = frame spacing.

When both compressive axial and lateral forces act simultaneously on a stringer, the nonlinear beam-column effect previously mentioned for framed structures arises. For the surfaced condition (A1), in which the slamming of the vehicle in a seaway is combined with sea slap forces, the stringer force system can be idealized as shown in Fig. 32*a*. Here, the distributed sea slap forces are represented as a line load F_s (lb/in.) equal to the product of the sea slap pressure (500 psf) times the circumferential spacing of the stringers. The magnitude of the axial end load F_a is calculated from equations (41) or (46), in which F_a equals σ_b times the cross-sectional area, A_s, of the stringer. Once F_a and F_s have been defined, the maximum bending moment in the stringer can be calculated from

$$M_{max} = \frac{F_s L^2}{8} \frac{2(1 - \cos u)}{u^2 \cos u} \tag{64}$$

where

$$u = \frac{L}{2}\left(\frac{F_a}{EI}\right)^{1/2}$$

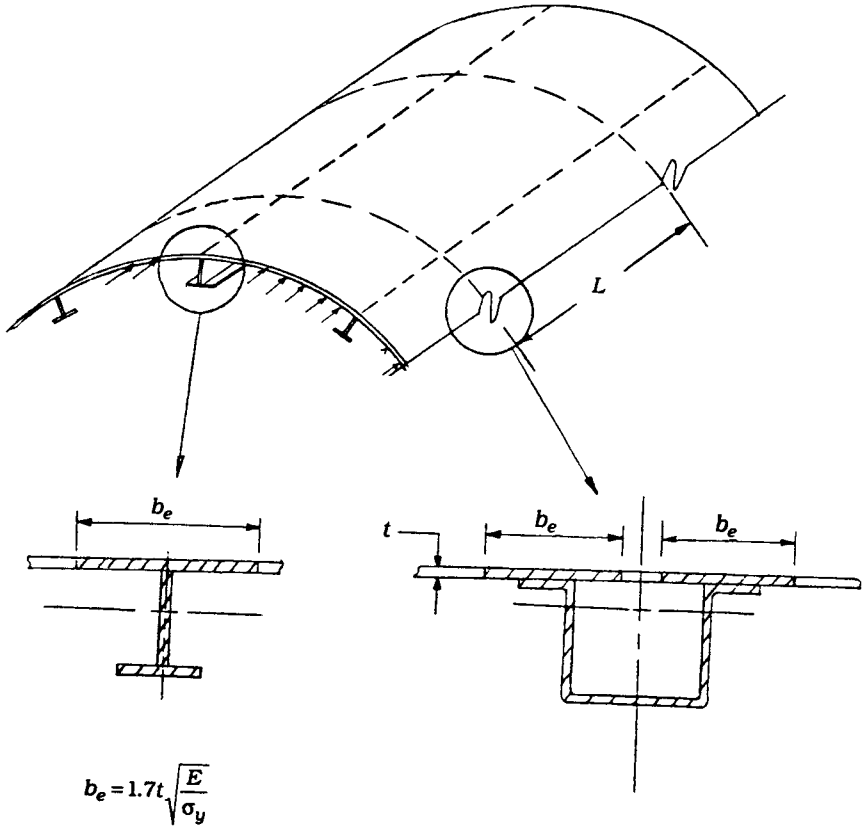


Fig. 33 Effective stringer-shell widths where b_e is the effective width of the shell to be used in calculating the moment of inertia I_s and area A_s in conjunction with the buckling strength of stringers. The shell material is the same as that of the stringer

When the maximum bending moment M_{max} for the stringer is calculated in this manner, the maximum stress in the stringer is obtained by combining the direct stress produced by force F_a with the maximum bending stress as follows:

$$|\sigma|_{max} = \frac{F_a}{A_{ss}} + \frac{M_{max}}{Z} \quad (65)$$

where

A_{ss} = cross-sectional area of the stringer-shell section,
Z = section modulus of the stringer-shell section, and
$|\sigma|_{max}$ = maximum stress in the stringer-shell section.

For load condition (A3), in which a concentrated lateral load P, representing a component gravity load acting at the midspan of the stringer, is combined with an axial stringer force F_a, induced by the bending moment in the stiffened shell as shown in Fig. 32*b*, the maximum bending moment in the stringer from load P can be obtained from

$$M_{max} = \frac{PL}{4} \frac{\tan u}{u} \quad (66)$$

The maximum stress in the stringer is found as before, by combining the direct stress from force F_a with the maximum bending stress

$$|\sigma|_{max} = \frac{F_a}{A_{ss}} + \frac{M_{max}}{Z} \quad (67)$$

Equations for other beam-column load cases can be found in references [22] and [23].

Although the preceding has addressed the modes of structural behavior that are generally of primary importance, other forms of instability such as local crippling of stringer elements and torsional instability have not been covered. The topic of instability of stiffened, unpressurized shells is complex and is not amenable to thorough treatment in a publication of this nature. The interested reader will find useful and concise coverage of the subject in references [24] and [25], in addition to the traditional texts such as references [26] to [28].

3.6 Foundations

A foundation is the structure interposed between components or equipment and the vehicle's main structure for connection and effective load transfer between the two. The equipment supported by the foundation usually is secured by mechanical attachments such as high-strength bolts, while the foundations are attached to the ship's structure frequently by welding and to a lesser extent by mechanical attachment. The type and size of a foundation will vary to suit the size and arrangement of the supported equipment.

For the purpose of concept or preliminary design, foundations for only the largest and heaviest components are sized roughly in order to establish feasibility and develop weight estimates. At this stage of the design process, the following information must be known or defined for the component or system or both to be supported:

1. The component's arrangement
2. The component's weight and center of gravity
3. Access requirements for system maintenance
4. Electrical and piping requirements
5. Component alignment criticality
6. Component operating loads
7. Method of attachment of component to foundation
8. Special requirements

The foundation must be of sufficient strength to withstand safely the various prescribed load conditions, specifically,

1. Gravity forces
2. Equipment operating loads
3. Sea slap, if outside the vehicle
4. Loads produced by vehicle impact
5. Dynamic loads induced by vehicle handling
6. Loads produced by vehicle motion

Absent any extraordinary operating or environmental conditions, experience has shown, in general, that a dynamic load factor of 2 is sufficient to account for the dynamic effects of conditions 5 and 6, above. The use of dynamic load factors facilitates the designer's task in that it essentially reduces a dynamic problem to a static equivalent. This is done by simply multiplying the weight of supported equipment, plus the estimated weight of the foundation, by the load factor to acquire the magnitude of force used to design the foundation. Recognizing that the dynamic load conditions specified in 5 and 6, above, have no directional preference of limitations (that is, the line of action of the inertial forces on the equipment can occur in any direction), it is customary to account for these multidirectional possibilities by performing strength calculations for each of three mutually perpendicular axes; normally, fore-and-aft, vertical, and athwartship. The foundation also should be of sufficient stiffness so that there will be no resonant frequency within the range of frequencies developed by any attached machinery and so that misalignment would be prevented that would interfere with operation of the supported equipment.

The vehicle structure to which the foundation is attached also should be reviewed to ascertain whether it has sufficient strength and stiffness to carry such loads safely or whether extensive modifications or strengthening of the main framework are required. This consideration might influence the designer to relocate the component.

Later in the design development, mockups of the vehicle primary structure, equipment, and associ-

ated electrical cables and piping may be built in order to avoid potential interferences and to meet other system construction and servicing access requirements. From such mockups, the designer can ascertain the most effective and practical method of foundationing equipment. It seems likely that economical computer graphic techniques generally will supersede the construction of mockups in the future.

3.7 Outer Hull Shells

There are two kinds of shells not specifically addressed in the previous discussion: (1) monocoque or unstiffened exostructure shells and (2) removable shell panels associated with access openings.

In this context, shell panels refer to removable shell segments that cover access openings in an exostructure. Such panels are normally, although not exclusively, rectangular in planform, and are sized to provide the requisite opening for servicing or removing enclosed equipment, piping, cables, etc. Typically, the shell openings for such panels are bounded with stiffeners, or the shell is thickened locally. This serves both to reinforce the shell at the opening and to provide edge support for the panels. Access covers are attached to the shell opening boundary reinforcement by mechanical fasteners, often of the quick-release type for ease of removal.

Access covers are not considered to contribute to overall shell strength; rather, the existence of access panels is disregarded for exostructure strength calculation purposes because of the way in which such panels are attached to the surrounding shell—only a few small fasteners normally are used for ease of removal and assembly. These are sufficient in strength to hold the panel in the shell, but not normally of sufficient strength to enable such panels to act integrally with the exostructure. Also, fastener holes generally are of larger diameter (usually by 1/32 in.) than the fastener in order to facilitate assembly, thus permitting relative movement between the panel and shell when the exostructure is stressed. For these reasons access panels are treated for design purposes as curved or flat plates, simply supported along the edges and of sufficient strength to sustain sea slap loading (represented as a uniformly distributed load of 500 psf).

For rectangular panels, ignoring shell curvature, the thickness required of the panel can be calculated conservatively from (see Fig. 34).

$$t = \sqrt{\frac{\beta p_s b^2}{\sigma_a}} \qquad (68)$$

where

t = plate thickness,
p_s = uniform pressure load = 500 psf,
b = width of panel (short side),

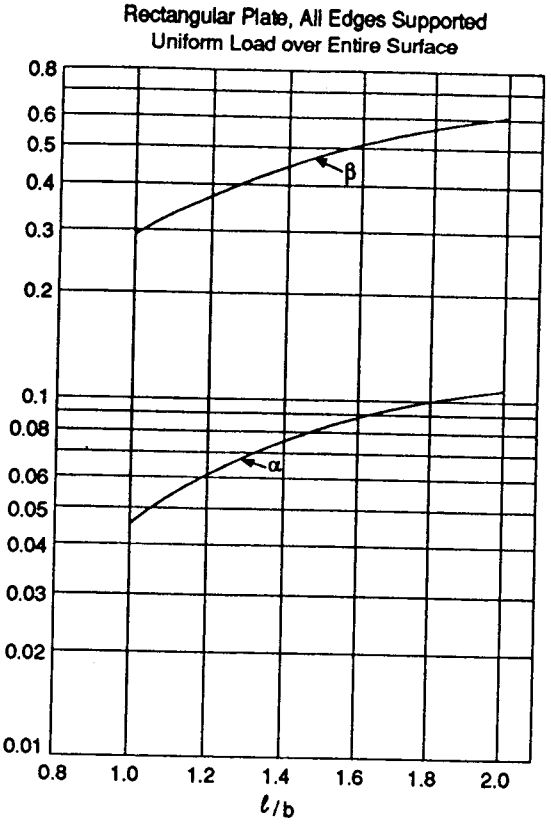


Fig. 34 Flat panel stress coefficients

σ_a = allowable stress, and
β = stress coefficient from Fig. 34.

If the access panel is circular, the thickness required can be obtained from

$$t = \sqrt{\frac{1.24 p_s R^2}{\sigma_a}} \qquad (69)$$

where R is the radius of the circular plate.

4. Summary

The intent of this chapter is to present an overview of the design process involved in developing a submersible. Simplicity of design is imperative for achieving minimum weight. The choice of the pres-

sure hull form is very restricted, and the need to minimize the number and types of penetrations is mandatory. Consequently, the structural design of a deep-diving submersible vehicle is restricted, essentially, to relatively few design considerations and is much less encumbered with structural details than the design of military submarines and other vehicles. The design of the exostructure involves many potential options depending on the mission of the vehicle, its size, and the pressure hull design selected for it.

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Nomenclature

A_f area of frames
A_h area of pressure hull shell's outer surface
A_s stringer cross-sectional area (in.^{2})
A_{ss} stringer-shell cross sectional area
a major semiaxis
B buoyancy factor (weight of pressure hull/weight of displaced water)
b minor semiaxis
b width (of frame web, panel, ring, shell)
C ratio of inner ring to outer ring radius
C_b shell buckling stress coefficient
c distance from neutral axis to extreme outer fiber (in.)
d distance from neutral axis to centroid of any stringer (in.)
E modulus of elasticity in compression (psi)
E_R Young's modulus of ring
E_{sec} secant modulus of compression
E_s Young's modulus of sphere
E_{sh} modulus of elasticity of shell (psi)
E_{str} modulus of elasticity of stringer (psi)
E_{tan} tangent modulus of compression
e eccentricity (in.)
F_a axial force or load
F_θ circumferential force
F_ϕ meridional force
F_{1ij} axial force in member ij (lb)
F_{2ij} vertical force in member ij acting at joint i (lb)
F_{3ij} lateral force in member ij acting at joint i (lb)
F_s sea slap line load on stringer
g gravity constant (32.2 ft/s^2)
h operating depth
I moment of inertia (in.^{4})
I^* moment of inertia of a composite section (in.^{4})
I_f moment of inertia of a transverse frame (in.^{4})
I_s moment of inertia of stringer (in.^{4})
K ratio of maximum stress to membrane stress in shell
KE kinetic energy (ft·lb)
K_M frame influence coefficient (bending moment)
K_V frame influence coefficient (shear force)
K_Δ frame influence coefficient (axial force)
K_θ frame influence coefficient (radial deflection)
K_{F_a} frame influence coefficient (rotation)
k_s manufacturing process coefficient
K_c panel buckling stress coefficient
L bulkhead or deep frame spacing
L_c critical arc length
L_f frame spacing
M applied bending moment (in.·lb)
M_{cr} critical bending moment (in.·lb)
M_η moment acting about member axis η
M_ξ moment acting about member axis ξ
M_ζ moment acting about member axis ζ
m mass (lb·s^2/ft)
n number of rings
P concentrated load (lb)
p hydrostatic pressure
p_b collapse pressure for lobar buckling
p_c design collapse pressure
p_{cr} collapse pressure, general instability failure
p_e elastic buckling pressure
p'_e elastic buckling collapse pressure
p''_e design elastic buckling pressure
p_p inelastic or elastic-plastic buckling pressure
p'_p inelastic buckling collapse pressure
p''_p design inelastic buckling pressure
p_s uniform pressure load (sea slap)
p_y collapse pressure for axisymmetric yielding
q shear flow (lb/in.)
R radius (in.)
R_e radius of an equivalent cylinder
R_o outer radius
R_s radius to mid thickness of shell
R_x resultant force acting in x direction (lb)
R_y resultant force acting in y direction (lb)
R_z resultant force acting in z direction (lb)
R_θ circumferential radius
R_ϕ meridional radius
R_l local midsurface radius over the critical arc length
R_{lo} local outer radius over the critical arc length
T torsional moment (in.·lb)
T_θ pitch period (s)
T_ϕ roll period
t thickness (in.)
t_a average thickness over a critical arc length
u amplification factor
U internal strain energy (ft·lb)
V shear force (lb)
V_h volume of pressure hull shell
v current velocity
v_i impact velocity (ft/s)
W weight (lb)
W_s weight of stiffeners and reinforcements
W_R ring weight
Z section modulus (in.^{3})
α angle of spherical pressure hull shell's intersection
β_1 reduction factor for frame size and spacing
β_2 reduction factor for frame area
δ_R deflection of ring at shell intersection
δ_s pressure hull shell membrane displacement normal to axis of revolution
Δ deflection (in.)
θ angle, rotation, generally
θ pitch angle (rad)
$\dot{\theta}$ pitch velocity (rad/s)
$\ddot{\theta}$ pitch acceleration (rad/s^2)

μ Poisson's ratio
μ_R Poisson's ratio of ring
μ_s Poisson's ratio of sphere
ρ density (lb/in.^{3})
ρ_R density of ring
ρ_s density of pressure hull shell
ρ_w density of sea water
σ stress
σ_a average stress in critical arc length
σ_b bending stress
σ_{cr} critical buckling stress (lb/in.^{3})
σ_{PL} proportional limit of material
σ_R stress in ring at its inner radius
σ_y yield strength
τ shear stress (psi)
ϕ roll angle
$\dot{\phi}$ roll velocity (rad/s)
$\ddot{\phi}$ roll acceleration (rad/s^2)

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

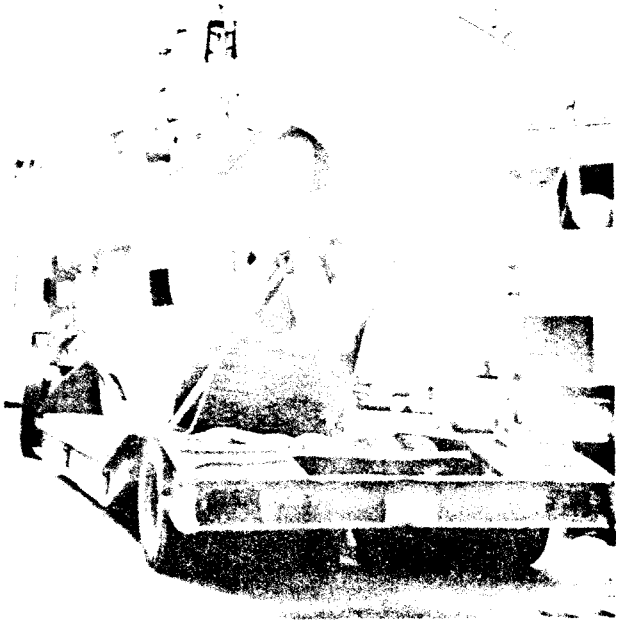


Fig. 1 Ground transportation of the submersible *Alvin*. (Courtesy of WHOI)

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.

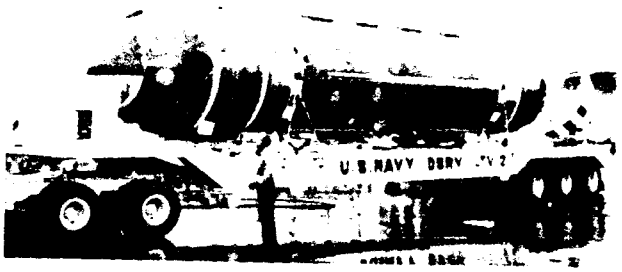


Fig. 2 DSRV land transportation vehicle. (Courtesy of Lockheed Missile and Space Co.)

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

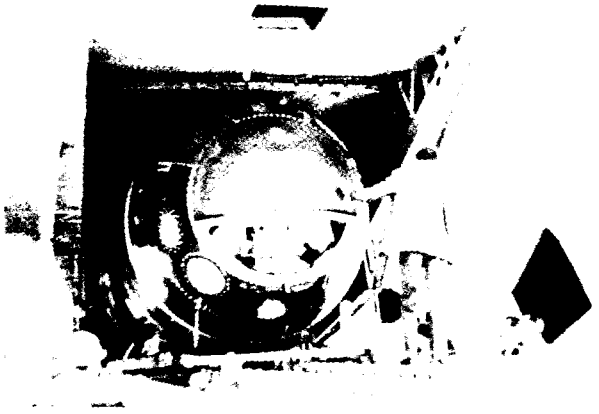


Fig. 3 Land transportation vehicle roller system. (Courtesy of Lockheed Missile and Space Co.)

- 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.

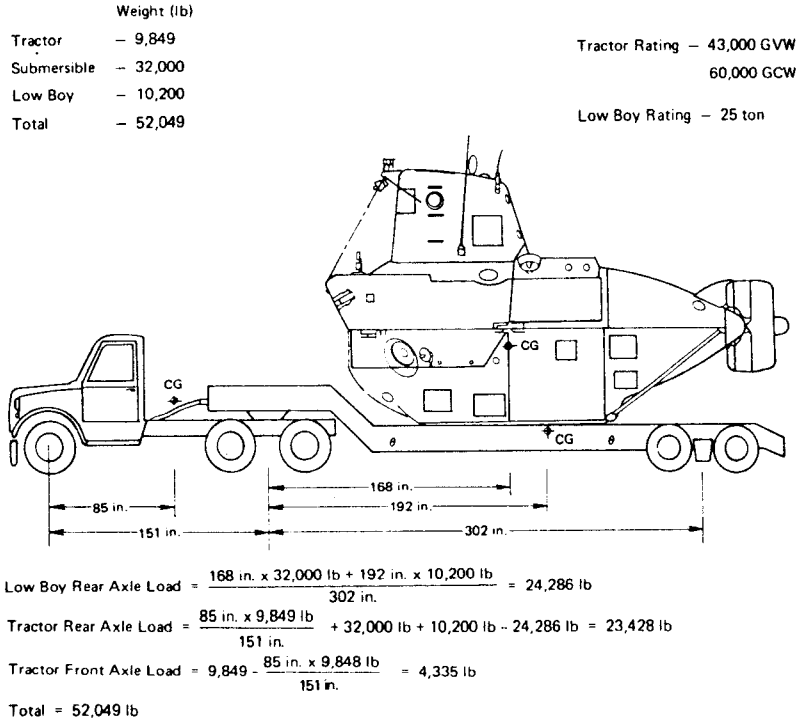


Fig. 4 Example of trailer loads

- 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].

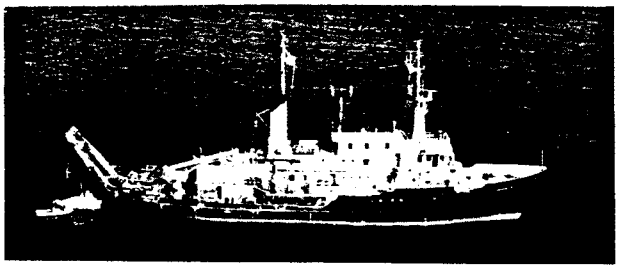
Overall costs may be considerably less with a ship-of-opportunity, though the efficiency of the total system may be reduced. Navigation and control systems for the submersible's underwater operations can be difficult to integrate into a ship-of-opportunity. Likewise, the time and costs of outfitting 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

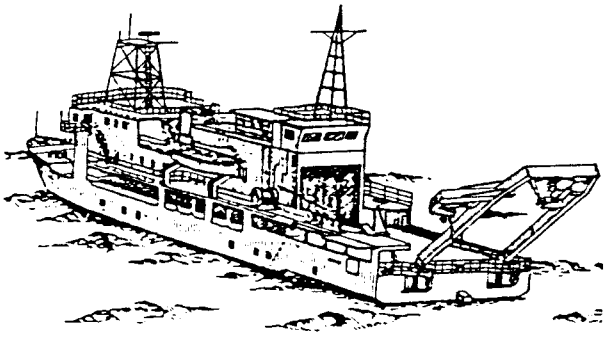
Table 4 Characteristics of some dedicated, manned-submersible support ships

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

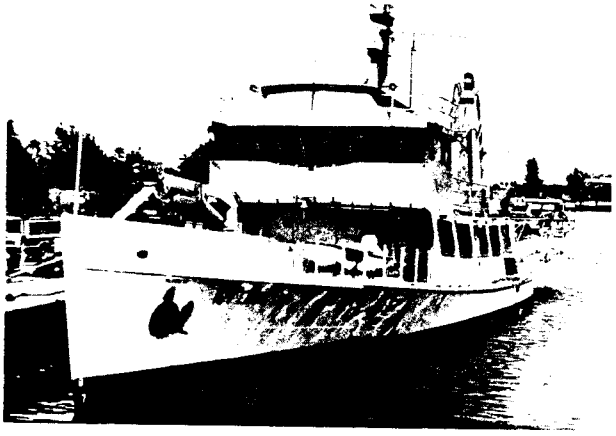
[a]Woods Hole Oceanographic Institute.
[b]Total accomodations.
[c]Harbor Branch Oceanographic Institution.
[d]French Institute of Research for Exploration of the Sea.
[e]Japan Marine Science and Technology Center.
[f]Information not available.
[g]Deep Submergence Vehicle Support Ship.



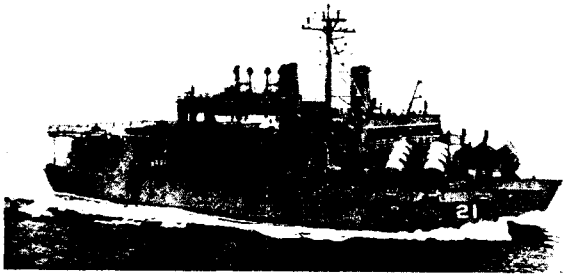
(a)



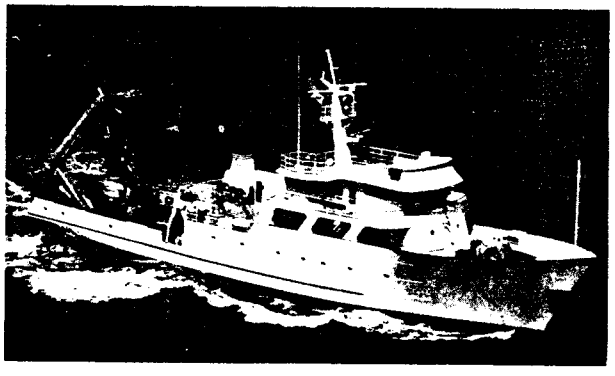
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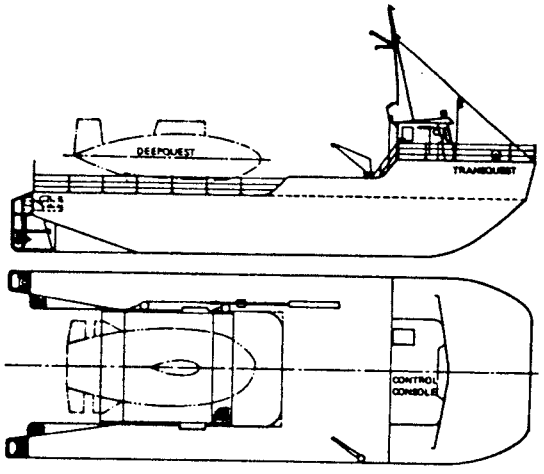
(b)



(e)



(c)



(f)

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).

Hullforms—The following is provided as an overview of the advantages and disadvantages of existing hullforms.

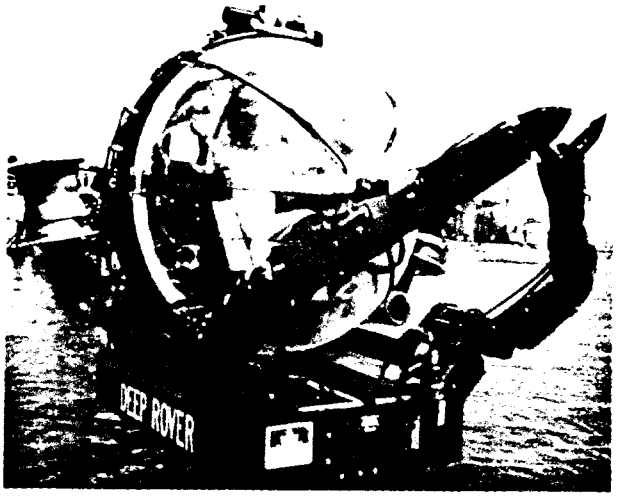


Fig. 6 *Deep Rover* submersible

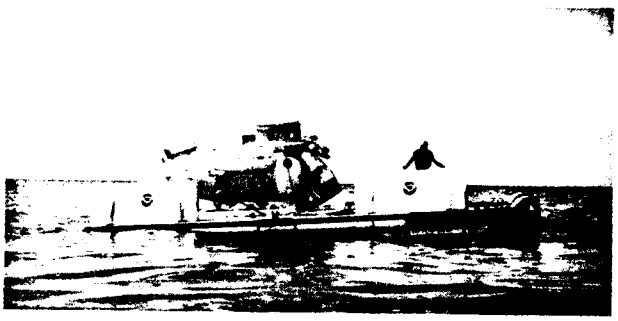


Fig. 7 Launch, recovery, and transport—12-ton aluminum (LRT-12A). (Courtesy of NOAA's Office of Undersea Research)

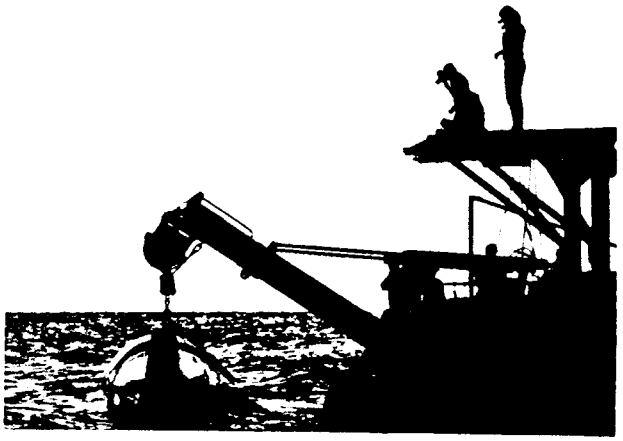


Fig. 8 Launching the one-man submersible *Deep Rover* from the support ship *Egabrag III*

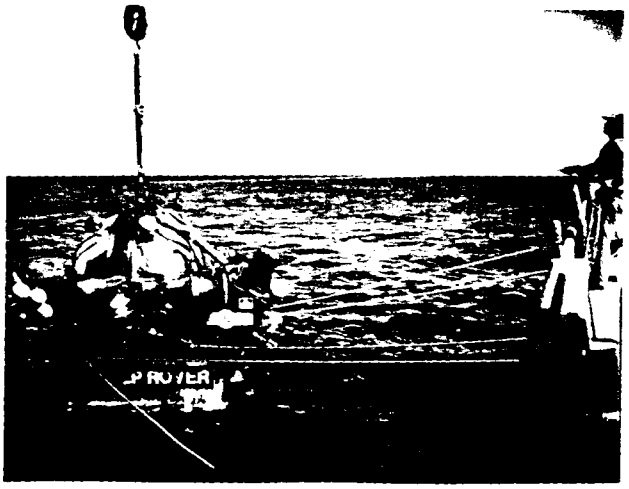


Fig. 9 Deployment of *Deep Rover* using crane from the vessel R/V *Wecoma*



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

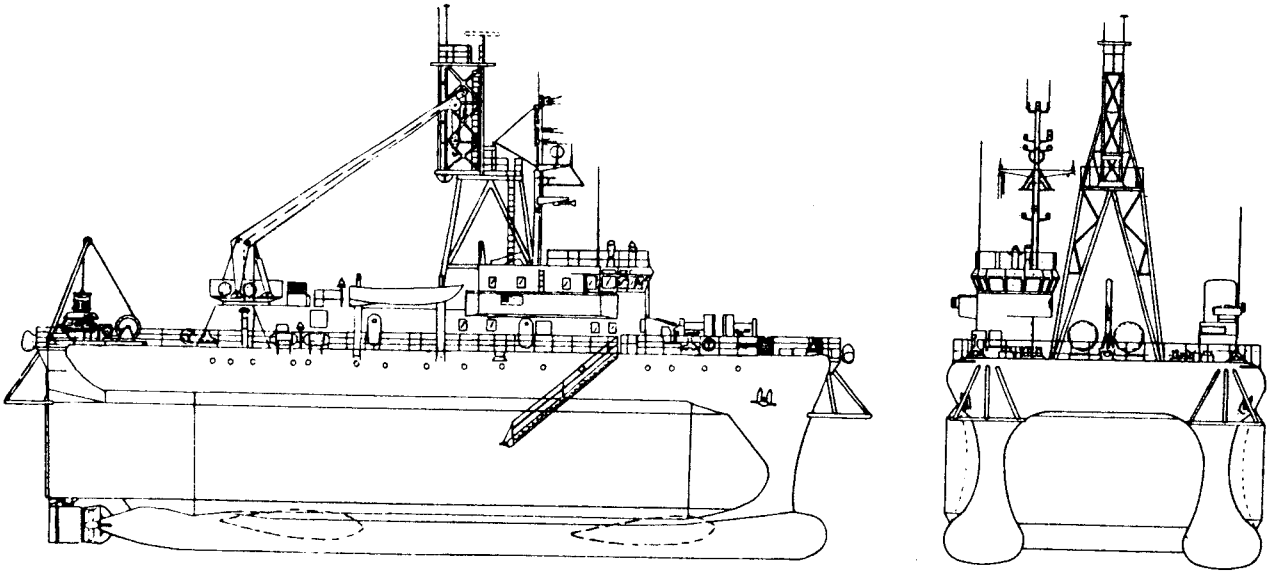


Fig. 11 Manned-submersible support ship *Twin Drill*

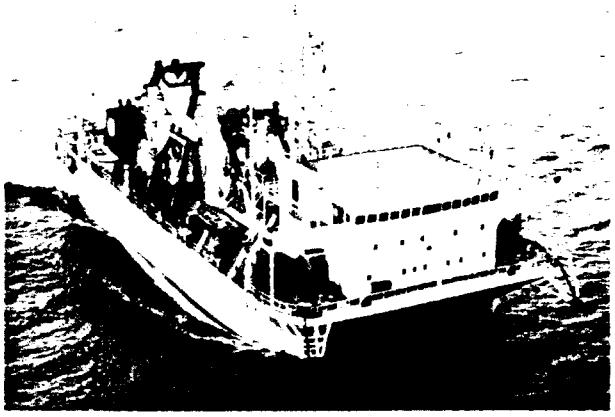


Fig. 12 Manned-submersible support ship *Kaiyo*

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-

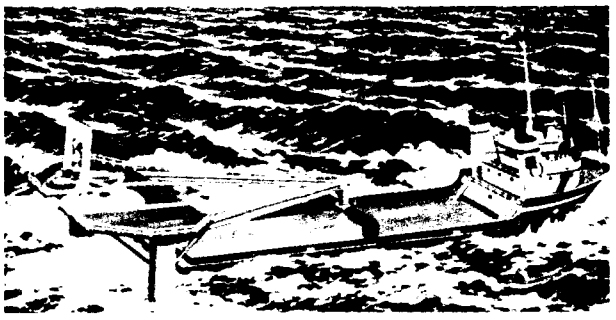


Fig. 13 SeaTrailer™

strumentation, and navigation system are normally carried on board the support platform. Spare thrusters, batteries, electrical connectors, cables, oxygen, and oils for hydraulic and pressure compensating systems are also required. If ascent and descent weights are used, spares of these must be carried as well. Electrical power supplies compatible with the submersible's systems are also required to operate test equipment and to charge the submersible's batteries. Secondary considerations for spare parts include shipping, customs, local supply, and the support ship's capability to assist in the repairs [2].

Submersible support ships should have adequate space for stowage of spare parts and supplies, a diving locker, a boatswain's locker, and maintenance, machining, and electrical/electronic shops to maintain and repair the various systems of the submersible. For vehicles that operate from ships-of-opportunity, support vans may be utilized to house the necessary shops and supply materials. Table 5 lists the physical characteristics of the support vans for the U.S. Navy's *Sea Cliff* and *Turtle* [6].

In addition to the spaces listed above, wet and dry laboratories should also be included in the arrangement of a support vessel. These are for use by the scientific and technical personnel on board the ship. A support platform should also provide a means for taking on stores, supplies, and equipment. Utility cranes are typically used in this capacity. Special handling facilities may be needed to assist in removing and replacing heavy items such as batteries and motors or thrusters from the submersible. An example of such facilities is the small elevator that is used on the *Atlantis II*. It is located just outside the hangar and is used to replace *Alvin*'s batteries.

Table 5 Physical characteristics of support vans for U.S. Navy submersible operations

Equipment	Space Requirements			Weight, lb
	Length, ft	Width, ft	Height, ft	
Supply Van	20.0	8.0	9.7	14 000
Electrical Van	16.0	8.0	9.7	16 500
Machine Shop Van	14.8	8.0	9.7	12 000
Electronics Van	14.8	8.0	9.7	17 000
Diving Locker	4.0	8.0	7.0	2 000
Bosun's Locker	5.0	6.0	6.0	3 000

Table 6 compares working and storage spaces as well as other features of four support ships. Data for this table were obtained from references [6] through [9].

2.2.4 *Manned Submersible Handling Systems*

All manned submersibles that operate at sea require a mechanism for deployment of the submersible before, and retrieval after, each dive. Three basic methods are used to accomplish this: a device to which the vehicle is attached and lifted out of the water, such as a crane or an A-frame; a platform onto which the submersible is maneuvered, either with the submersible fully submerged or with the submersible surfaced; or a mother submarine, by which the submersible is transported, deployed, and recovered entirely underwater.

This section describes the handling systems for a manned submersible. The following types of launch/retrieval systems will be discussed:

- Cranes
  - —fixed boom (non-articulated)
  - —articulated
  - —bridge/gantry
- A-frames
- Cradles/Elevators



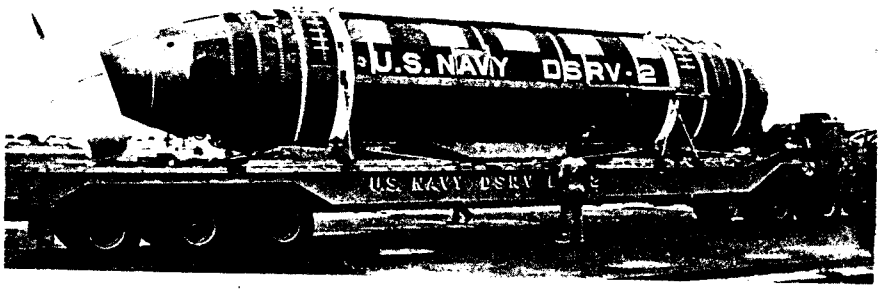


Fig. 14 Submarine rescue vehicles: (*top*) U.S. Navy's *Mystic* and (*bottom*) U.S. Navy's *Avalon*. (Courtesy of U.S. Navy)

Table 6 Comparison of some of the features on four dedicated support ships

Ship	Mechanical, Machining, or Maintenance	Electrical or Electronic	Supply	Diving Locker	Boatswain's Locker	Wet/Dry Labs, ft^2
Atlantis II	Mechanical and hydraulic shops—64 ft^2 each; plus 60 ft^2 in the hanger	Electrical—64 ft^2	21 000 ft^3	13 ft^2—holds enough gear for 3 divers (4 tanks)	None dedicated for *Alvin*	Total of 1700 ft^2 of lab area
Seward Johnson	Maintenance—272 ft^2 (used for storage, too)	Electronics—152 ft^2	Two tanks: 300 gal of hydraulic oil; 110 gal of mineral oil	24 ft^2—holds enough gear for 6 divers	None dedicated. Maintenance space is used	Wet—220 Dry—370
Deep Submergence Vehicle Support Ship *Laney Chouest*	Machine shop housed in a van (see Table 5). Also, a DSV hanger shall be incorporated into the ship	Electrical and electronic shops housed in vans (see Table 5)	250 gal for oil storage tanks. 960 ft^3 lockable storage space on main deck. Enclosed storage for 8- to 50-gal drums of oil	800 ft^3	Housed in a van (see Table 5)	Wet/Dry—400
Nadir	Mechanical van—20 ft ISO container	Electronic van—20 ft ISO container	850 ft^3 for ballast in holds. Spare parts van—20 ft ISO van	Information not available	Information not available	Information not available

Ship	Utility Cranes and Hoists	Expandable Weights, lb	Electrical Power Generation	Maneuverability	Propulsion	Small Boats
Atlantis II	One 40 000-lb crane. One small elevator outside the hangar for pulling *Alvin*'s batteries	60 000	Two 300-kw generators	One trainable bow thruster—800 hp	2400 shp	One 16-ft rigid inflatable boat
Seward Johnson	One 10 000-lb light-weight crane. One side A-frame, 3000-lb capacity	None: submersibles use an air/water ballast system	Three 295-kw generators	Two 360-degree rotatable thrusters—500 hp each	1700 shp	One 13-ft Boston Whaler. One 22-ft rigid inflatable boat
Deep Submergence Vehicle Support Ship *Laney Chouest*	One crane capable of lifting 20 000 lb; outreach of 15 ft for deep ocean lifting. One 5-ton utility crane (operable in sea state 4). One 5-ton overhead hoist	100 000	Four 300-kw generators	Three 360-degree azimuth thrusters—1200 hp each	9200 hp	Two 18-ft rigid inflatable boats, each powered with a 40 hp outboard motor
Nadir	One 30 000-lb crane. One 6000-lb overhead crane in the hangar. One 30-ton elevator.	30 tons for 30 dives	Three 300-kva, 300-V 50-Hz alternators	One directional thruster with 4465 tons of thrust	1200 hp	Information not available

- Docking well
- Submerged systems
  - —submarines
  - —platforms

Each system is defined in the following paragraphs, and examples are given to acquaint the reader with its operation. Advantages and disadvantages are also discussed for each of the various handling systems. Appendix 1 of this chapter summarizes the advantages and disadvantages of the systems.

Motion compensation techniques are defined and described in Appendix 2 in order to assist in defining the performance limitations of each system. Both passive and active systems are discussed and examples given to explain the application of motion compensating systems in the launch and recovery of manned submersibles.

Cranes—The types of cranes used to launch and recover manned submersibles are fixed boom (non-articulated), articulated, and gantry cranes. Of these three, non-articulated and articulated cranes have been the most widely used and are adapted from those used on land. The gantry crane has only been utilized by the U.S. Navy on the ASR-21 class submarine rescue ships to support the DSRVs.

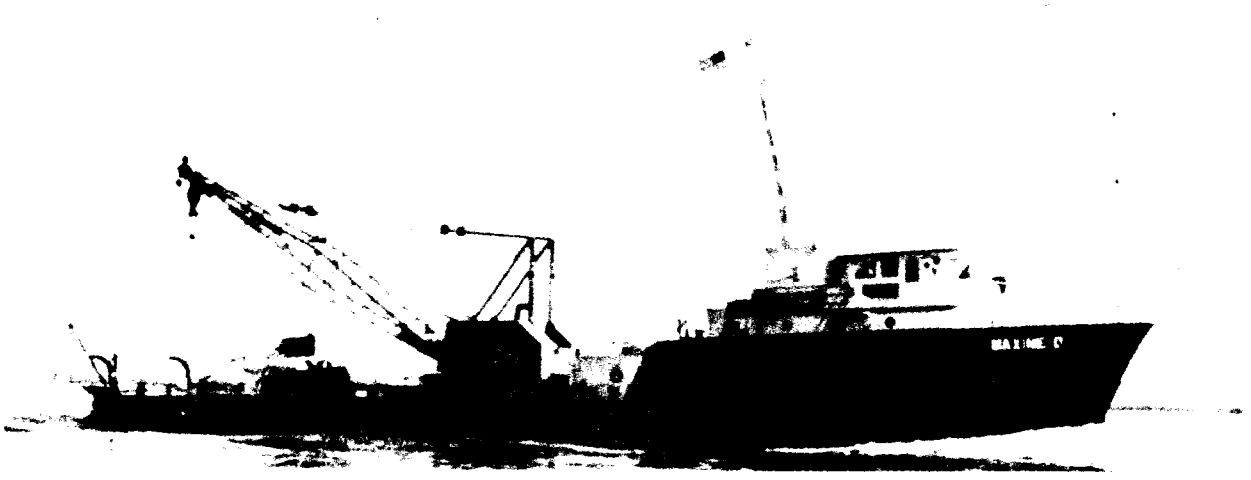


Fig. 15 Fixed-boom crane on the M/V *Maxine D*

Fixed-boom cranes: Fixed-boom cranes (Fig. 15) employ a rigid boom, which may or may not have the ability to be linearly extended and retracted. Typically, with this kind of crane, the submersible is suspended by a wire rope and is swung out over the side or quarter of the support ship. Tag lines are required to minimize the pendulum-like motion of the suspended submersible. This motion is due to the response of the support ship to the sea state. When the submersible is in the water, a diver is required to release the lift lines and tag lines during deployment and to attach them during recovery operations.

The load capacities of fixed-boom cranes have ranged from approximately 10 tons for the *Nekton Alpha, Beta,* and *Gamma* submersibles, to 100 tons for the *Turtle* (DSV-3) and the *Sea Cliff* (DSV-4), when they were operated off the *Maxine D.* In the early 1970s, the *Nekton* submersibles weighed 2.4 tons dry, and in the late 1960s to mid-1970s, when the *Maxine D* was used, the *Turtle* and the *Sea Cliff* each weighed 24 tons dry [1]. (Since then, both the *Turtle* and the *Sea Cliff* have undergone major modifications, so that the *Turtle* now weighs approximately 26 tons, and the *Sea Cliff* approximately 29 tons, each being dry weight as opposed to in-water weight or displacement.)

The submersible *Star III* was also deployed and recovered at sea with a fixed-boom crane. At that time, in the early 1970s, the *Star III* weighed 10.5 tons in air.

When a submersible is in the water, just after deployment or prior to recovery, the lifting mechanism and tag lines must be released or secured. Both *Star III* and *Sea Cliff/Turtle* used a four-point bridle system for lifting, and at least four tag lines for steadying purposes during lifting and lowering operations, which meant that eight attachment points had to be dealt with for every launch and recovery evolution. Today, as well as in the early 1970s, launching is easier than retrieval because of the quick-release mechanisms that can be included in the system and utilized during deployment operations. However, in order to secure the lifting mechanism and handling lines for recovery, divers have to be used. It is very dangerous to attach the lifting and tag lines on a rolling, pitching, and slippery submersible, especially when the heavy lift hook or bridle system is swinging. Figure 16 illustrates the number of lines that were required to hoist the *Sea Cliff* when launched/recovered by the fixed-boom crane on the *Maxine D.* This is no longer true today because *Sea Cliff* was changed to a single-point lift capability during her 20 000-ft depth modification. Therefore, the I-beam system is no longer required, but the four tag lines would still be necessary if the *Sea Cliff* were ever to be deployed or recovered with a crane. In addition to the lines shown in Fig. 16, another line was run to a small boat which acted as a variable resistance sea anchor. The small boat was fitted with two outboard motors and helped steady the *Sea Cliff* during deployment/retrieval operations.

Advantages and disadvantages: The main advantages of a fixed-boom crane system for deploying and recovering manned submersibles are its simplicity and cost. No special equipment other than the crane is needed on board the support ship. This allows a variety of vessels to be considered for conversion to submersible support ships. It also means that the whole submersible system, including the handling equipment, could be flown anywhere in the world and put on board a compatible ship for transport to the dive site. This was the case of the U.S. Navy submersible handling crane used on the *Energy Services I.* Its shipping weight was 100 000 lb, and it could be loaded on board a C-5A aircraft. It was designed to be mounted on an offshore supply boat no smaller than 200 ft long with a 40-ft beam, and could be made operational within 12 h [11]. The economic advantages (due to the potentially minimum modifi-

cations required of an existing ship) and its simplicity make a fixed-boom crane system attractive.

Other advantages of a fixed-boom crane system are its ability to handle other pieces of equipment, capacity for growth, adaptation of subsystems, and favorable launch and recovery time in light to moderate seas. If designed and placed on the support ship properly, the crane can also be used to lift the submersible on and off the ship in port. This means the support ship's crew would not have to rely on shore services and could on-load and off-load the submersible anywhere in the world.

Fixed-boom cranes have many disadvantages, some of which are very difficult to overcome. A major obstacle is the pendulum motion and the need to detach/attach the lifting hook or mechanism. The pendulum motion is affected by the motion of the wire rope suspension point, the forces due to the ship's motion, the frequency of the pendulating submersible relative to the ship's period, and the wind forces [12]. Line handlers using tag lines provide some control over the suspended submersible's motion; however, their effectiveness is limited by the need for coordination between the various groups involved with the deploying/recovering operation, and by the angles of the lines. Pitchersky and Southerland [12] state that a considerable training effort is necessary to obtain the degree of coordinated response that is necessary for reasonable control under adverse conditions. Consistent coordination is very difficult to achieve, and controlling tag lines from bitts or gypsy heads is not conducive to positive and accurate control.

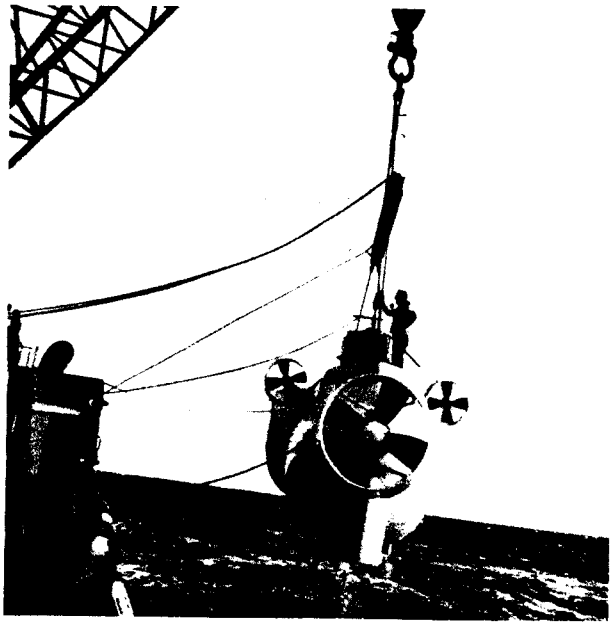


Fig. 16 Manned submersible *Sea Cliff* being launched/recovered by the fixed-boom crane on the *Maxine D*

When the submersible is suspended outboard of the support ship's side, the only motion of the submersible that can be restrained is the outward motion, away from the ship. Line handlers can only apply motion restraint from the side facing the ship. Nothing prevents the submersible from swinging inboard, towards the ship. Therefore, a long crane boom is required to get the vehicle far enough outboard of the ship so that no contact is made when the submersible swings toward the support vessel.

Besides pendulation while the submersible is suspended and detachment/attachment problems, there are other disadvantages associated with a fixed-boom crane system. One is the number of people required to launch and recover the submersible. The following people are necessary for the operations: at least three line handlers (*Sea Cliff/Turtle* and *Star III* used four) to man the tag lines; the crane operator; someone to direct the crane operator; at least one diver; and, in the *Sea Cliff/Turtle*'s case, two people in a small boat. Therefore, an absolute minimum of eight people is required for each launch and recovery procedure. As mentioned previously, it can be very difficult to coordinate the actions of all the people involved, especially in adverse conditions.

Another disadvantage is the snap loading on the system that can occur when the submersible is partially or totally in the water and still secured to the crane. Snap loads occur when waves passing under the submersible pick it up. If the submersible is partially or completely buoyant, its weight will be partially or totally removed from the hoisting line by the force of the water and then suddenly reapplied as the wave passes [12]. This can be very dangerous to the crew in the submersible, as well as to the support personnel on the deck of the support ship and in the water.

Discussion: Except for the *Deep Rover*, fixed-boom cranes are no longer routinely used to launch and recover manned submersibles at sea. They were commonly used in the 1960s and 1970s because of their minimal impact on a support ship and because they were easily adapted from land-based cranes. Very few submersible owners or operators built a dedicated support platform for their vehicle. Usually, the submersible was built, and then means of supporting and handling it were considered. A fixed-boom crane, therefore, was the most simple and least costly method for use in handling a submersible.

The flexibility, simplicity, and transportability of fixed-boom cranes are very attractive advantages. Nevertheless, the dangers to the support ship, the submersible, and the crews of both, due to the pendulating motion of the submersible when suspended, overcome those advantages. Furthermore, the number of lines that must be secured to the vehicle increases the recovery time. While a diver is securing the lift hook, the crane operator must be alert and ready to anticipate the movements of the

ship, sea, and submersible to prevent the lifting device from striking the diver or the submersible. The entire recovery operation is a potential hazard to the divers and submersible.

The intensity of the problems fixed-boom cranes experience is more pronounced with increasing sea states. Therefore, these systems are limited to state 3 seas or less [12,13]. Even in sea state 3 conditions, launch and recovery operations have proven to be very difficult.



Fig. 17 Two types of articulated cranes that have been successfully used to retrieve and deploy submersibles

Articulated cranes: An articulated crane differs from a fixed-boom crane in that its boom is made up of distinct sections that are separated by joints. This kind of crane is also called a backhoe-type crane. Figure 17 illustrates two articulated cranes that have been successfully used to deploy and retrieve manned submersibles.

The essence of an articulated crane was probably best stated by Mr. Edwin A. Link: "The world's best lifting device would be a man's arm, if it could lift eight tons" [5]. The articulated crane was originally designed to virtually snatch the submersible from the water when the vehicle is on the crest of a wave or swell. In addition, the pendulum motion of the vehicle when suspended is diminished because the boom tip is as close to the submersible's lift point as possible during the transition of ship from sea and vice versa.

To date, five support ships have used an articulated crane to deploy and recover manned submersibles. The vehicles ranged in weight from Jacques Cousteau's 4.5-ton SP-350 (also known as *Diving Saucer*), which is supported by the R/V *Calypso* [1], to the 12.5-ton French submersible *Griffon*, which was supported by the *Triton* [15].

Other submersibles which were launched and recovered by an articulated crane were the *Deepstar 4000* which weighed 9 tons in 1976 [1] and the two 12-ton Harbor Branch Oceanographic Institution (HBOI) submersibles *Johnson-Sea-Link I* and *II* (*J-S-L I* and *II*). The *Deepstar 4000* was supported by the M/V *Searchtide*, a leased 155-ft-long offshore supply boat. The *J-S-L I* and *II* were operated from both the R/V *Sea Diver*, a 100-ft ex-trawler, and the R/V *R. V. Johnson* (renamed the R/V *Edwin Link*), a 125-ft converted Coast Guard cutter. The R/V *Edwin Link* supported the *J-S-L* for 14 years until 1988 when she was replaced by a new R/V *Edwin Link* with an A-frame handling system. HBOI also operates the *J-S-Ls* from another support ship, the R/V *Seward Johnson*. This ship and its A-frame handling system will be described in the subsection that discusses A-frames.

For examples of launch and recovery operations using an articulated crane, the *Deepstar 4000/Searchtide* and *J-S-L/Edwin Link* systems will be described. An articulated crane with a lift capacity of 25 tons deployed and retrieved the *Deepstar 4000*. A crane operator and two people controlling tag lines were required when the vehicle was suspended in the air. When the vehicle was placed in the water, a diver was not necessary to trip the lift hook. However, a diver was needed to conduct a pre-dive checkout underwater and also to release the tag lines. Retrieving the *Deepstar 4000* involved having a diver swim a tow line out and secure it to the submersible, and then hauling the vehicle close enough to the ship to enable the diver to connect the lift hook. The bottom part of Fig. 17 shows the *Deepstar 4000* being hoisted from the water.

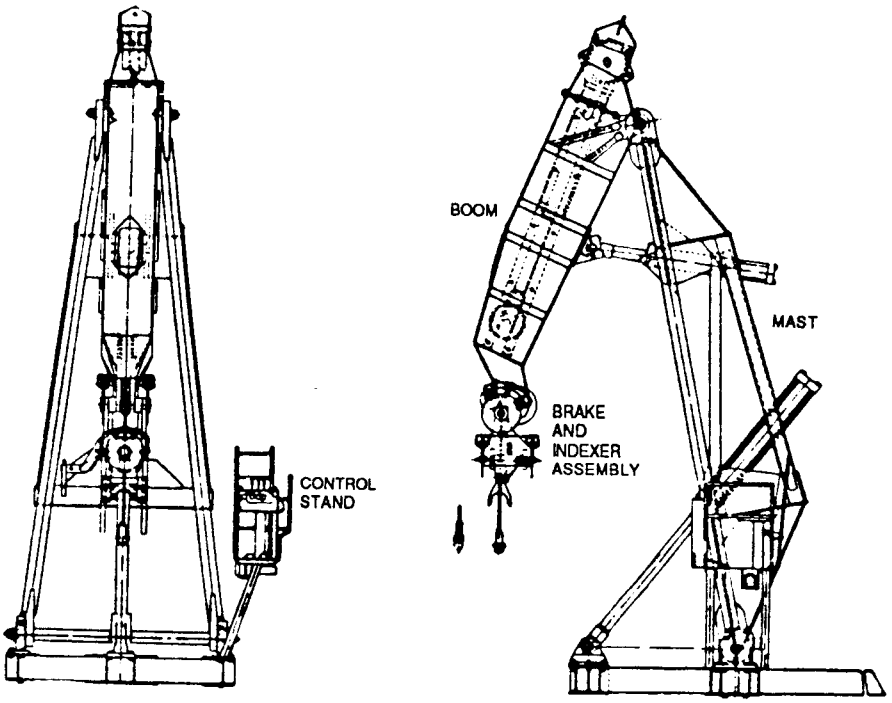


Fig. 18 The hydrocrane, an articulated crane used to launch and recover the *Johnson-Sea-Link* submersible [16]

The articulated crane that is used to support the *J-S-L* submersibles is called the Aluminum Hydrocrane, and is illustrated in Fig. 17(*top*) and 18. It was designed and developed specifically for the *J-S-Ls* in the early 1970s by Mr. Edwin A. Link and Alcoa, and began supporting the *J-S-Ls* by the end of 1974 [16]. The Hydrocrane is constructed entirely of 6061-T6 aluminum and is hydraulically operated. The design criteria for the Hydrocrane are given in Table 7. A hydraulic motion compensating system was incorporated in the crane head to dampen both the roll and pitch of the suspended submersible. See Appendix 2 of this chapter for more details on this motion compensating system.

The retrieval procedure of the *J-S-L* is as follows (summarized from reference [16]): When the *Edwin Link* was approximately 50 ft from the *J-S-L*, the swimmer secured a line to the port quarter of the submersible and then climbed aboard it. The *J-S-L* was winched into position parallel to the stern of the *Edwin Link*. Then the bow and stern lines were passed to the swimmer, who secured them to the vehicle. When these two lines were secured at their proper lengths, as marked, the *J-S-L* was in a position such that the lift attachment point and the lifting mechanism could be guided together by the swimmer. The swimmer then swam to the recovery line and the hydrocrane lifted the submersible out of the water. The support ship maintained minimum turns on the port screw until the *J-S-L* was winched into position and the lift connection made.

No swimmers were required for deployment of the *J-S-L* submersibles from the *Edwin Link*. The bow and stern mooring lines were not used, and the lifting mechanism could be released without the aid of a diver in the water. The attachment point on the submersible (still used by HBOI today) is referred to as the drop-lock quick release (DLQR) body. It is a rectangular housing bolted around the submersible strong-back and provides a lifting point near the centers of gravity and buoyancy. At the end of the lifting cable is a torpedo-shaped device called the drop lock which fits into the DLQR body. Four flukes on the side of the drop lock fall open when it is

Table 7 Design criteria[a] for the Harbor Branch Oceanographic Institution's Aluminum Hydrocrane [16]

Submersible Weight	20 000 lb
Breaking Strength of Cable	80 000 lb
Average Pick-Up Velocity of Submersible	1.4 ft/s
Time to Get Submersible on Deck After Locking	25 s
Time to Get into Water	20 s
Maximum Hydraulic Pressure	2500 psi
Maximum Sea State for Operation	state 5
Roll Amplitude of *R. V. Johnson*	10 deg
Roll Period of *R. V. Johnson*	8 s
Pitch Amplitude of *R. V. Johnson*	10 deg
Pitch Period of *R. V. Johnson*	6 s

[a]The data in this table is from a paper presented in 1976 and may since have changed. For example, the vehicle itself now weighs approximately 24 000 lb. This table is included to give the reader an idea of what was included in the design of a very successful submersible handling system.

inserted into the body. When tension is applied to the cable, the flukes make contact with a rim inside and at the top of the body. This provides the lift. To release the drop lock, the cable is slackened and high-pressure air is piped into the DLQR body from the pilot's sphere. Both hook-up and disconnect are accomplished in seconds.

Advantages and disadvantages: Articulated cranes used for manned submersible handling systems have the same advantages as fixed-boom cranes. However, articulated cranes have a clear advantage over the fixed-boom type because the boom tip is close-coupled to the submersible's pick-up point and is as close to its center of gravity as possible. This minimizes the hazardous pendulum motion of other handling systems, therefore making launch and retrieval safer and more efficient.

Another attractive feature of an articulated crane is its extreme maneuverability. It is capable of handling loads above and below deck, through restricted openings, and also close to its base. Therefore, it is suited for many applications, as well as for submersible support. When not in use, it can be folded into a relatively compact form, minimizing the required deck space for stowage.

An additional advantage of articulated cranes is their low center of gravity, which contributes to the seaworthiness of a support ship. This can be important since submersible support ships are usually small.

The main disadvantage of an articulated crane system is the potential danger to the submersible and swimmer because of the close proximity of the boom tip during attaching/detaching operations, especially in all but light sea states. HBOI skirted this problem by lowering the drop lock at the end of a cable a few feet to the lift point of the submersible. However, a swimmer is still necessary to guide the drop lock into place.

Other disadvantages of these cranes are the severe side loads that can occur and the additional operator training necessary. The severe side loads are a result of suppressing the pendulating motion imparted to the submersible. Additional training is required for the crane operator because he or she must be able to pluck the submersible from the water at the proper time and before the next wave or swell reaches it.

Discussion: Articulated cranes have been used by at least five manned submersible operators in the past. In fact, the *Edwin Link* supported one of the *J-S-L* submersibles during the dives on the Civil War ironclad ship, the USS *Monitor*, in 1985. This crane, as well as an identical one on the R/V *Sea Diver*, were used to launch and recover the 12-ton *Johnson-Sea-Link* submersibles for more than 2000 successful launch-and-recovery operations with few maintenance problems.

Why then did HBOI switch to an A-frame on their new support ships, the R/Vs *Seward Johnson* and the new *Edwin Link*? The new system will allow launch-and-recovery operations in heavier seas than the articulated crane and the R/Vs should be able to retrieve a submersible in up to sea state 5 or 6 seas [18]. A-frames will be discussed in more detail later in this section.

Bridge/Gantry Cranes: Figure 19 illustrates the components of a travelling gantry crane. The basic description of a gantry crane is as follows: Gantry cranes are modifications of travelling cranes. The bridge is carried at the ends by the legs and is supported by trucks with wheels so that the crane can travel on fore-and-aft- or athwartships-oriented tracks on the deck of the support ship. The bridge carries a hoisting unit over which a cover should be used to protect the machinery from the weather. The crane is driven by the motor through a gear reduction to the horizontal shaft which, in turn, drives the two vertical shafts through bevel gears. Bevel and spur gear reductions connect the axles of the wheels with the vertical shafts. Many gantry cranes are built without the cross shaft, employing separate motors, brakes, and gear reducers at each leg of the crane.

Bridge cranes, as used on support ships, are similar to gantry cranes. The only difference is that they do not ride on trucks with wheels. Therefore, the entire crane is stationary; only the hoisting mechanism can move, and it moves athwartships from one end of the bridge structure to the other. Shore-based bridge cranes found in industry normally run along the length of a ship on elevated tracks located at the ends of the bridge structure. A gantry crane is used on the USS *Pigeon* (ASR-21) and the USS

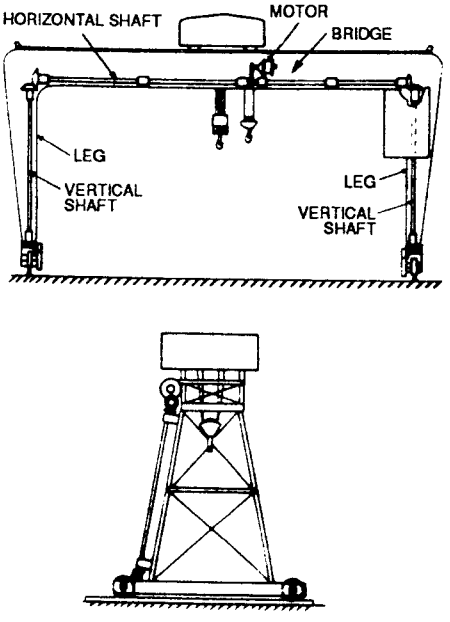


Fig. 19 Components of a gantry crane [18]

Ortolan (ASR-22) to support the DSRVs and other deep-submergence systems (DSS). Specifically, it is used for the following operations:

- Launch/recovery of the DSS components
- Handling of DSS components during deck transfer or pierside loading/unloading
- Workboat launch and recovery

The gantry crane on board the ASR-21 class ships (Fig. 20) is made up of a basic steel frame structure, four end truck assemblies, an operator's cab, forward and after trolley assemblies, hydraulic piping, deck tracks and track extensions, cable rigging, and electrical power and control systems. Its hydraulic system is self-contained, and the ship's electrical system supplies the necessary power (400 VAC, 60 Hz). The gantry crane is approximately 75 ft long and weighs 360 000 lb. It can travel 86 ft athwartships, plus an additional 17 ft over each side when the outriggers are extended. The outriggers allow over-the-side handling of the DSS components at sea, or the loading/off-loading of them in port. The forward and after trolley assemblies move fore and aft on rails atop the bridge frame. Each trolley has a double drum hoist with a lift capacity of 87 500 lb, a hydraulic power unit, an auxiliary hoist, ram cable tensioners, and associated electrical and hydraulic components. The two trolleys can be incorporated to provide a total lift capacity of 175 000 lb. The aft trolley auxiliary hoist has a lift capacity of 40 000 lb, while the forward trolley auxiliary hoist has a lift capacity of 10 000 lb.

Table 8 lists the general specifications of the ASR-21 class ship bridge crane system.

The launch and recovery procedures of the DSRVs are detailed in the subsection in which elevators are discussed.

Another type of overhead crane system used in the past for manned submersible support was an overhead rail. Two vehicles were launched and recovered by this type of system: the 9.4-ton (dry weight) *DOWB*, and the 2.4-ton (dry weight) *Nekton* submersibles. Figure 21 shows the handling system for the *DOWB*.

This method of submersible launch and recovery was not very successful because the pendulum effect was substantial. Also, it required more maneuvering by both the ship and submersible since the rail was untrainable.

Advantages and disadvantages: Bridge/gantry crane systems offer many advantages if an elevator or cradle mechanism and heave compensation systems are incorporated, as in the case of the ASR-21 class ships. The greatest advantages are avoidance of pendulum motion problems, and disconnecting/connecting operations at the air-sea interface. However, the elevator system is very costly and complex.

Discussion: A bridge crane or gantry crane that is not combined with an elevator is an inefficient

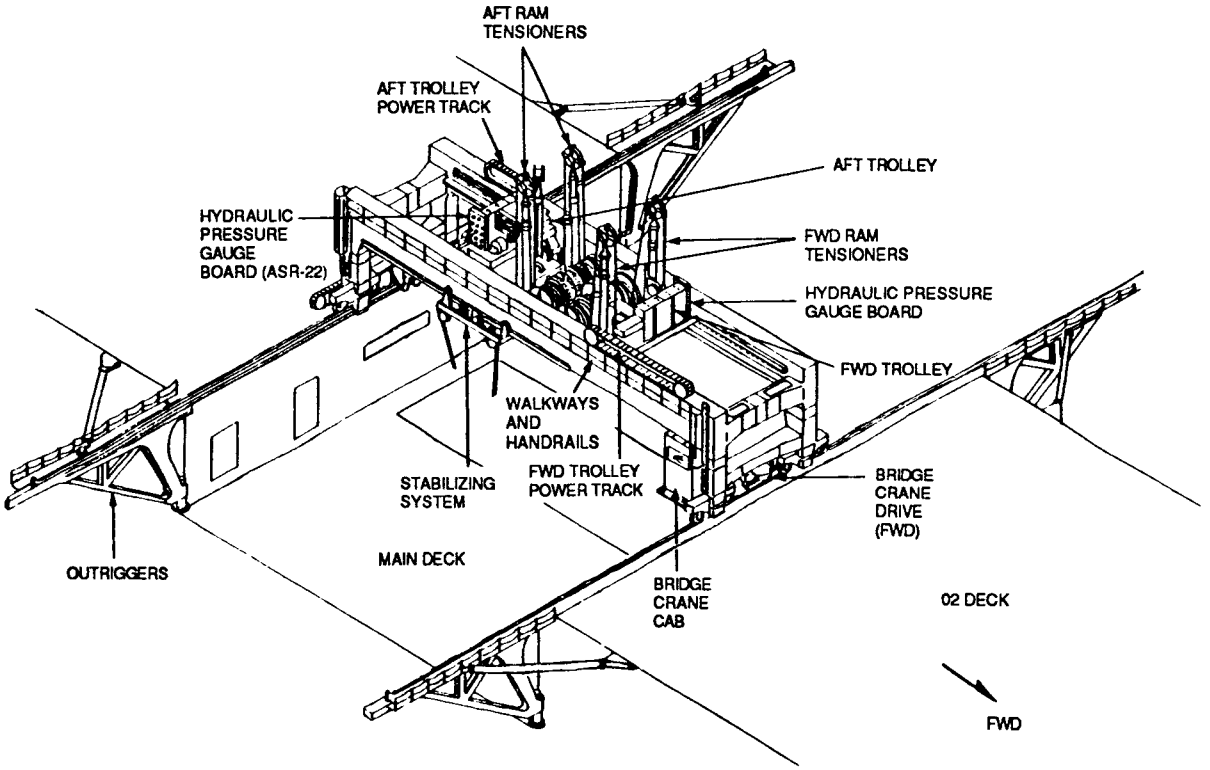


Fig. 20 Bridge crane assembly on the ASR-21 class submarine rescue ships [19]

Table 8 ASR-21 class ship bridge crane specification [19]

Bridge Crane Frame Assembly	
Overall Height (above 02 deck, ram tensioners extended)	46 ft 11 in.
Overall Height (above 02 deck, ram tensioners retracted)	34 ft 11 in.
Maximum Clearance Above Main Deck	27 ft
Overall Width	27 ft
Overall Length (fwd to aft)	75 ft 2 in.
Span, Clear, Between Cab and Aft Frame	59 ft
Total Weight	370 000 lb
Power Requirement	440 vac, 60 Hz

	Forward Trolley Assembly	Aft Trolley Assembly
Height (main ram tensioner extended)	35 ft 1 in.	37 ft 1 in.
Height (main ram tensioner retracted)	23 ft 1 in.	25 ft 1 in.
Width	17 ft 3 in.	17 ft 3 in.
Length	18 ft 3 in.	17 ft 9 in.
Weight	73 000 lb	91 000 lb
Travel Speed	30 fpm	30 fpm
Capacities		
Main Hoist		
Lift Capacity	87 500 lb	87 500 lb
Vertical Lift Distance	168 ft	168 ft
Speed, Rated Load	22 fpm	22 fpm
Main Hoist Ram Tensioner		
Maximum Operating Pressure	2500 psig	2500 psig
Rated Line Pull	43 750 lb	43 750 lb
Maximum Line Takeup	24 ft	24 ft
Auxiliary Hoist		
Lift Capacity	10 000 lb	40 000 lb
Vertical Lift Distance	63 ft 6 in.	46 ft 6 in.
Speed, Rated Load	28 fpm	28 fpm
Auxiliary Hoist Ram Tensioner		
Maximum Operating Pressure	—	2500 psig
Rated Line Pull	—	41 250 lb
Maximum Line Takeup	—	24 ft

method for handling a manned submersible due to its inflexibility. The surface ship would have to maneuver dangerously close to the submersible, or the submersible would have to reserve some of its valuable energy in order to approach the ship, or both. In heavy seas this could be both dangerous and time-consuming.

A-frames—A-frames are used extensively in the launch and retrieval of manned submersibles. However, they got a slow start in the 1960s and early 1970s when the use of manned vehicles was widespread. Only one company, Vickers Oceanics, used them with their fleet of *Pisces* submersibles. By the late 1970s, 15 of the 29 submersibles listed in reference [15] in use world-wide were deployed/recovered

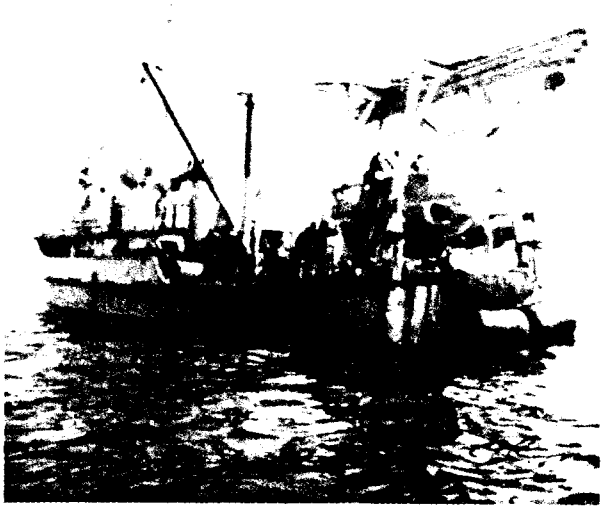


Fig. 21 Handling system for the *DOWB*

with A-frames. All the vehicles that operated in the North Sea employed this method. Most of the manned submersibles operating today are launched and recovered with an A-frame, including the *Sea Cliff, Turtle, Shinkai, Nautile, Johnson-Sea-Link I* and *II*, and *Alvin*. Figure 22 shows the various configurations of A-frames used for launching and retrieving submersibles. All are mounted on the stern of their prospective support ship.

The basic configuration of the A-frame in Fig. 22(*a*) consists of the main structural A-frame with a smaller, inverted A-frame hanging from its apex. The smaller A-frame swings longitudinally within the main frame.

The main frame is extended outboard over the stern or retracted inboard by hydraulic rams. The lift line is a 5-in. braided nylon line attached to a winch. It is passed over a sheave at the top of the A-frame and then down through the inverted "A" as illustrated in Fig. 22(*a*). British Oceanics, Ltd. utilized this A-frame during the 1970s and into the 1980s.

To retrieve the submersible, the operating procedure of the British Oceanics, Ltd. A-frame was as follows [15]: when the submersible surfaced, the support ship positioned itself approximately 100 ft from the vehicle. An inflatable rubber raft carried a diver and the end of a braided nylon tow line to the submersible. The diver attached the tow line to the vehicle and then climbed aboard and into the sail. The support ship got underway into the prevailing seas at a speed of 1 to 2 knots, whatever was adequate to maintain headway with the submersible under tow. The submersible was drawn close to the stern to a point where the lift hawser could be

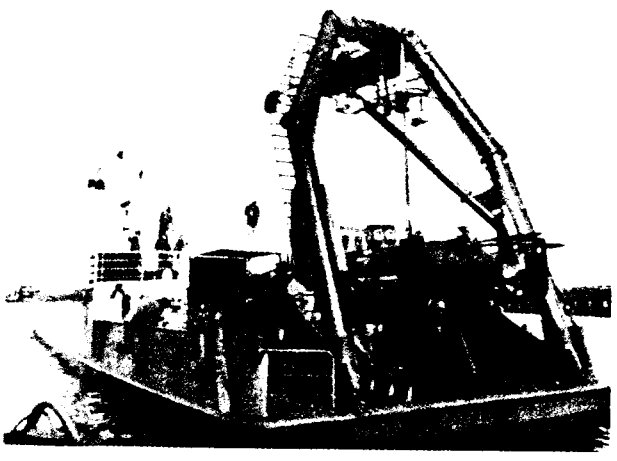
attached to the lift hook by means of a conventional shackle. The submersible was then lifted by the winch until it was almost two-blocked to the small inverted A-frame. The diver, who rode on the submersible throughout the retrieval procedure, engaged a double-looped safety strap (wire rope) to a "ram's horn" lift hook, and the braided nylon lift hawser was lowered until the safety strap supported the full load of the vehicle. The main A-frame was then retracted inboard, and the submersible was lowered onto a deck trolley, where it was trundled into its shop. The original tow line became a steadying line during the retraction phase of launch. Operations could be conducted in up to and including sea state 6 with this system, according to British Oceanics, Ltd.

The success of this A-frame system is due primarily to its inboard motion-compensating winch, with high-speed capabilities. A high hoist speed was used to overcome wave-induced upward motion. This feature was designed to reduce shock loading.

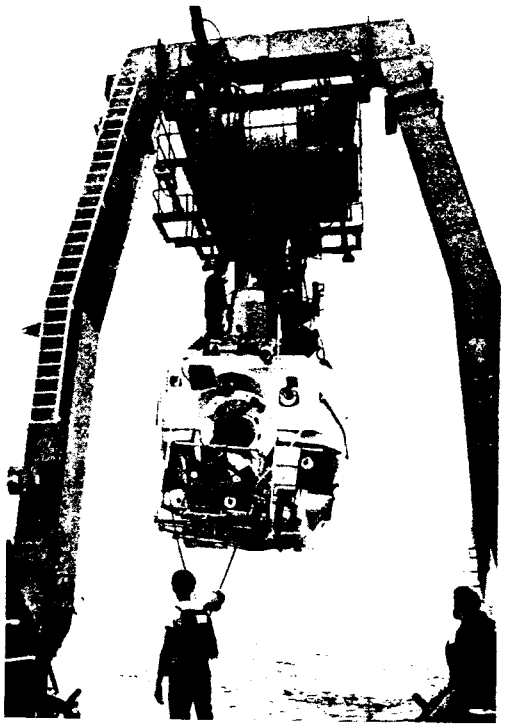
The operator of the A-frame shown in Fig. 22(*b*) was Northern Offshore, Ltd. (Intersub). The main difference between this A-frame and British Oceanics, Ltd.'s A-frame is that the winch was located at the apex of the A-frame structure. Therefore, valuable deck space was saved for other services. It also did not include the smaller inverted A-frame in its main A-frame structure. The Northern Offshore, Ltd. system followed the same general operating procedure as the British Oceanics system to bring the vehicle into a position near the support craft for attachment of the main lift line (that is, rubber raft, swimmer, tow line, etc.). During the towing phase (conducted while the support ship was underway), a small parachute was deployed from the submersible. The purpose of the parachute was to act as a drogue to provide dynamic stability. At a point directly under the extended A-frame, the lift hawser was attached by a swimmer. Because there was no inverted A-frame, there was approximately 9 ft of lift hawser between the top of the A-frame and the lift point; the pendulum motion induced by a sea state could be severe. To limit this motion, the swimmer attached four smaller lines to the submersible, two port and two starboard, which were controlled aboard the support craft and steadied the submersible as it was brought aboard. The lift points on the submersible consisted of a standard hook configuration. The hook opening was closed by a counterweighted bar that was opened to attach the main lift line. The system could operate routinely



(*a*)



(*b*)



(*c*)

Fig. 22 Various A-frame configurations used to deploy/retrieve manned submersibles: (*a*) British Oceanics, Ltd. system; (*b*) Northern Offshore, Ltd. system; (*c*) Caley Hydraulic system [1,10]

in sea state 4, and retrieval was possible in sea state 5 if necessary [15].

The third A-frame, shown in Fig. 22(c), is the Woods Hole Oceanographic Institution's (WHOI) system, which in conjunction with the R/V *Atlantis II* currently supports the manned submersible *Alvin*. Caley, Ltd., of East Kilbride, Scotland, designed and developed the A-frame, pendant/winch assembly, and the hydraulics and controls. The newest and largest A-frame handling system is on the Deep-Submergence Support Ship *Laney Chouest*. This ship is owned and operated by Edison Chouest Offshore and is contracted by the Military Sea Lift Command to support the U.S. Navy's *Sea Cliff* and *Turtle*.

Figure 23 illustrates the components of the Caley A-frame system on board the Harbor Branch Oceanographic Institution's (HBOI) ship R/V *Seward Johnson*. A low inertia lift winch is located below the top beam of the A-frame and also below the fore and aft pendulation control device. Besides saving useful deck space, placing the winch at this location also minimizes installation time and costs, and eliminates

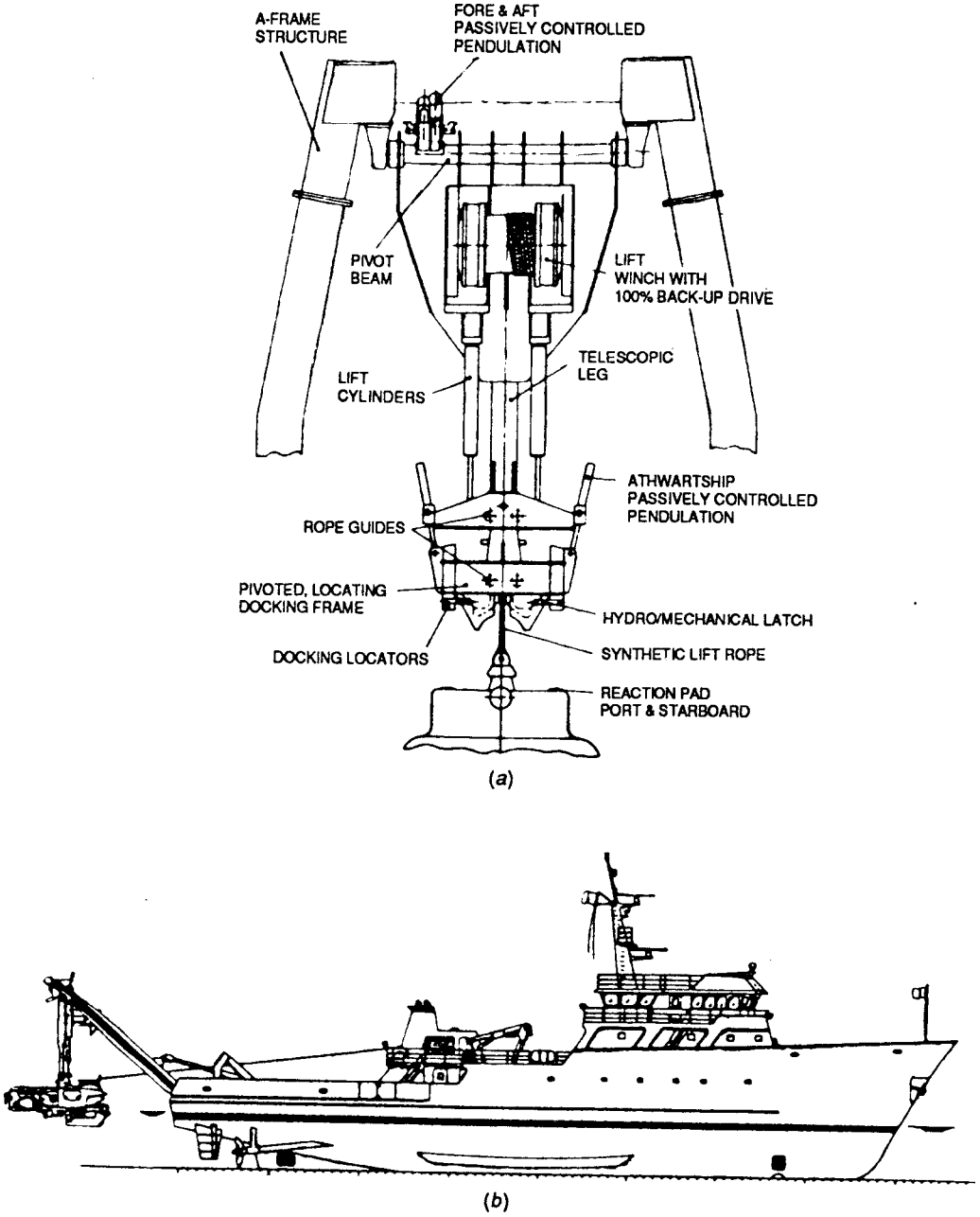


Fig. 23 Harbor Branch Oceanographic Institution/Caley Hydraulics submersible launch and recovery system: (*a*) lift winch and telescopic leg; (*b*) system on the R/V *Seward Johnson*

problems associated with the lift line on deck [20]. Below the winch is a vertical telescopic leg which supports the locator-docking frame. The docking frame, which is capable of pivoting, includes a hydromechanical submersible latch, rope guides, and a passive port/starboard-controlled pendulation system. These elaborate mechanisms allow the docking frame to follow and control compounded submersible pendulation. In this manner, overstressing of the submersible and docking frame components are prevented.

During launching operations the entire system is controlled by one person. The submersible is secured at the telescopic leg, which provides the lift from the deck and limits pendulation, thereby steadying the submersible during the transition to the water. When the A-frame is in the outboard position, 41 degrees from the vertical, the lift winch takes the strain of the submersible. This unlocks the mechanical latches and allows the vehicle to be lowered into the water. A drop lock similar in design and operation to the one used with the articulated crane is actuated, allowing the submersible to get underway.

The recovery of the submersible is similar to the two preceding examples. The submersible is towed to the transom area by a tow winch and the lift rope is attached by a diver. The lift winch is set to raise in its fast-response auto-tension mode, preventing slack rope problems. This also acts to reduce the vertical displacements between submersible and A-frame. The submersible is then hoisted to locate and automatically latch the docking frame at the telescopic leg. Thus, controlled pendulation is achieved. Two very important refinements make the operator's task simpler, particularly in heavy seas where considerable support ship roll can be created:

- On recovery, during the lift winch hoist, the docking frame is allowed to "float" and be aligned by the lift rope angulation, presenting the docking interface parallel to the submersible.
- The telescopic leg has an overhoist feature which dispenses with the need to stop the lift accurately at the latches.

During both launch and recovery operations, the R/V *Seward Johnson* is underway at approximately two knots. The system capabilities are listed in Table 9.

Table 9 Harbor Branch Oceanographic Institution/Caley Hydraulics A-frame system typical capabilities [20]

Rated Capacity	30 tons
Maximum Recommended Operational Support Vessel Roll	30 deg total
Maximum Recommended Operational Heave Relative Position Vessel and Submersible (vertical position)	10 ft
Maximum Hoist Speed	60 ft/min
Tow Winch Capacity	3 tons
Approval	ABS, Lloyds, and DNV

Table 10 *Alvin/Atlantis II* A-frame test loads, lb [21]

1.	Weight of *Alvin*	35 000		
2.	Dynamic Overload	17 500		
3.	Entrained Water	3 500		
4.	Design Load (1 + 2 + 3)	56 000		
5.	Rated Load Test (dynamic)	35 000		USN
6.	Functional Test (dynamic)	43 600	1.25 × rated	ABS
7.	Overload Test (dynamic)	84 000	1.50 × rated	USN
8.	Static Test	112 000	2.00 × rated	USN

The operating procedures that WHOI uses to deploy and recover the *Alvin* from the *Atlantis II* are similar to those for HBOI: the main difference is the attachment mechanism between the vehicle and telescoping boom. Instead of a pneumatically-operated drop lock, WHOI uses a T-shaped hook on the *Alvin* (Fig. 24), over which a soft eye in a 4-in., double-braided nylon line is placed. This system requires divers for both launch and recovery operations.

Table 10 summarizes the test loads conducted on the *Alvin/Atlantis II* A-frame. Both static and dynamic loads were applied. The system was tested to U.S. Navy, U.S. Coast Guard, and American Bureau of Shipping (ABS) requirements.

Advantages and disadvantages: A-frames are good, stable structures for launching and recovering manned submersibles. They are more rigid for a given deck area than either a fixed boom or an articulated crane and offer side loading advantages over the



Fig. 24 The *Alvin* T-hook [21]

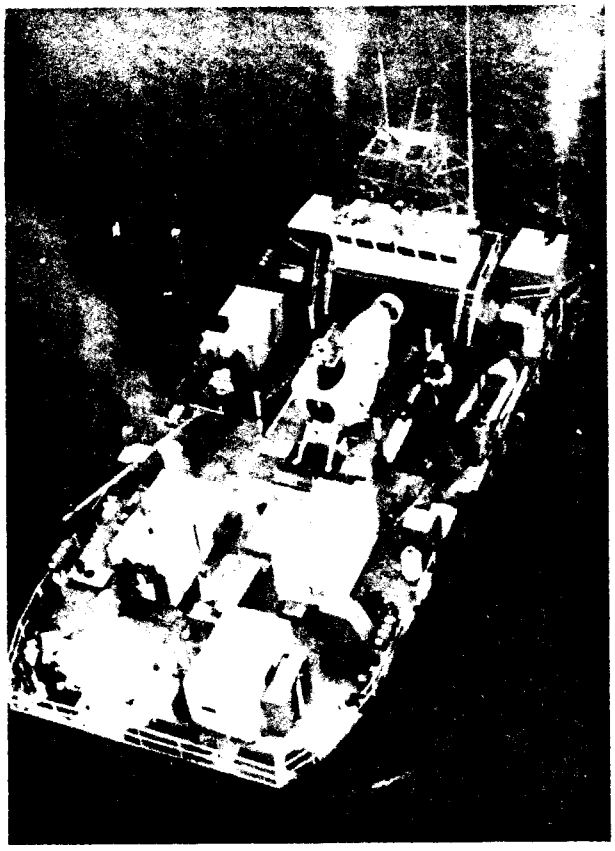


Fig. 25 R/V *Lulu* with the *Sea Cliff* onboard [22]

articulated crane. When anti-pendulation devices are installed, A-frames can be used in higher sea states. For example, British Oceanics, Ltd. have reportedly operated in the North Sea and northeastern coastal Atlantic areas in sea state 4 [13]. Caley Hydraulics systems have successfully recovered submersibles in winter conditions, during which the support vessel often exceeded 50 degrees total roll [20].

On the other hand, A-frames can have severe load pendulation motions if not equipped with some kind of motion compensating system. As with fixed-boom cranes, some means of controlling this motion is necessary to prevent damage to the submersible, ship, or A-frame itself, as well as to prevent injury to personnel in the vehicle and on the ship.

Other disadvantages are the amount of deck space required and the lift line from the deck winch to the A-frame when the winch is not located inside the A-frame structure. With conventionally configured A-frames, the deck winch is forward of the A-frame structure. In this configuration, the entire system occupies a much greater area of deck on the support ship than does a crane system. When suspended over the deck from the winch to the A-frame, lift lines under tension are extremely dangerous and are a potential hazard to personnel.

A-frames are also limited in their use. Whereas cranes can load/unload equipment and the submersible from/to the pier or move them around on deck, A-frames can only be used to raise and lower equipment over the side or stern. The HBOI and WHOI A-frames were designed and built to handle specific submersibles. The lifting points and attachment methods are distinct and therefore unique for handling their particular submersible system only. These A-frame systems would probably not be used for anything other than submersible handling.

Discussion: A-frames are the most common piece of equipment in use today to launch and recover manned submersibles. Most of the deep ocean manned submersibles use them, excepting the U.S. Navy's DSRVs.

To be supported by an A-frame, a submersible must not only be capable of a single-point lift, but also be suited for towing. This means that the appendages and external mission equipment items must be secured with enough strength to withstand the drag and slamming forces created by towing the submersible to the support vessel during recovery operations. The tow speed historically has been around one to two knots, but in any case must be of sufficient speed to maintain control of the submersible in heavy seas.

Cradles/elevators—With this kind of launch-and-recovery system, the submersible is maneuvered onto a platform. The platform can be just below the surface of the water, such as on the M/V *Transquest* and the R/V *Lulu*, or submerged deep enough so as not to be affected by the waves and wind on the surface. The ASR-21 class ship utilizes the latter method with its gantry crane and cradle dolly/elevator system.

The elevator system can be located at either the center of the ship or at the stern. Both positions have been used successfully to launch and recover manned submersibles. The R/V *Lulu* and the ASR-21 utilize an inboard-mounted elevator system, whereas the M/V *Transquest* uses a stern-mounted elevator. All three ships are still operational; however, the *Lulu* is no longer actively used by WHOI's research submersible, *Alvin*. She was reassigned to the U.S. Navy when *Atlantis II* was converted to the *Alvin*'s support ship.

Launch-and-recovery procedures for all three ships are given to clarify the different cradle/elevator methods used to support manned submersibles. R/V *Lulu*'s procedures are included because of her successful operations for almost 20 years.

The R/V *Lulu* is a unique catamaran made from two surplus U.S. Navy pontoons. The pontoons are 96 ft long, 14 ft wide, and 9 ft deep, and are connected by two bridge structures. They are separated by 20 ft. A deck, 70 ft long and 45 ft wide, is supported by the bridge structures. The cradle is located aft of mid-

ships and is controlled by four electrically-driven chain hoists, tested to 27 tons. R/V *Lulu* is shown in Fig. 25. Note that the *Sea Cliff* is on board *Lulu* in this photograph.

The launching procedure requires that the cradle be lowered into the water and stopped prior to the submersible becoming buoyant. At this point, the pilot and observers enter the personnel sphere and close the hatch. This procedure is used for submersible crew safety and to keep water from splashing into the sphere.

Allowing the crew to go aboard the submersible just before it becomes waterborne is a result of an accident involving *Alvin* in August 1968. As *Alvin* was being lowered on the cradle with the crew aboard and hatch open, one of the wire ropes parted and the submersible slipped into the sea. All three crew members managed to get out before the *Alvin* sank in 5500 ft of water. A year later, *Alvin* was salvaged by the U.S. Navy and was back in service again in the summer of 1970.

After the crew is on board and the hatch secured, the cradle is lowered until it is about 10 ft below the submersible. The submersible then backs out from between *Lulu*'s hulls as six tending lines, three on each side, are passed aft. Divers aboard the submersible cast off the lines and conduct pre-dive checks.

Retrieval is accomplished in the reverse sequence of launching. When the submersible surfaces, the *Lulu* is used to close the distance. The submersible maneuvers into position over the submerged cradle under her own power and, with the aid of the six line handlers, is properly positioned. As in launching, the pilot and observers disembark before the submersible is completely out of the water. Figure 26 shows the *Alvin* between the hulls of the *Lulu* during a launch or recovery. Note the number of tending lines required.

One reason why the *Alvin* and the *Lulu* were successful for so long is that their motion responses to the sea are very much alike [12], which has not been true for other submersible and support ship combinations. The limiting factor in the *Alvin/Lulu* handling process was the control of the submersible between the hulls of the ship, before the tending lines could be secured. There was only 4.5 ft of clearance on each side of *Alvin* when between the *Lulu*'s hulls, which proved to be a major hazard in high sea states. However, launch and recovery operations were conducted in up to sea state 4.

Another example of an inboard-mounted cradle system is the ASR-21 class submarine rescue ships handling systems. These ships utilize a gantry crane to raise and lower the deep ocean systems carried on board.

To launch and recover the DSRV, a cradle dolly is used in conjunction with the gantry crane. The cradle dolly (Fig. 27) provides deck stowage and transportation for the DSRV. The self-propelled dolly is used to move the DSRV athwartships to the deck decompression chamber hatchway, or to the centerwell hatchway and onto the lift platform. It has shock absorbers installed that dampen the landing shock of the DSRV when it settles into the cradle underwater. The dolly is attached to the lift cables from the bridge crane by cable sockets that are located inside the upper extremity of the cradle and assemblies. Six guide wheels, located on the vertical arm of each end assembly, are used to guide and steady the cradle dolly when it passes between the two hulls of the ASR. The upper wheels are made of solid polyurethane. The rest of the tires are pneumatic.

Also shown in Fig. 27 are the two wire rope guides which prevent the four main lifting cables from chafing against the ASR ship structure. They also assist in guiding the lift platform and cradle dolly into the guide arms. When the platform and dolly are lowered into the water, the wire rope guides remain two feet below the main deck level, and are picked up again when the platform and dolly are brought up.

Figure 28 illustrates the DSRV lift platform. It is located within the centerwell hatchway and is used as an elevator for the submerged launch and recovery of the DSRV, and also for the Personnel Transfer Capsule (PTC). To accomplish the vertical lift and lowering of the system, cables from the gantry crane main hoists are secured to a cable clamp at each corner of the platform. Television cameras and lights and their associated topside and underwater instrumentation and equipment are used to assist the DSRV during submerged launch and recovery operations. This underwater television surveillance system includes one forward and one after television camera, seven lights, and three television monitors. The monitors are located in the pilothouse, Rescue Control Center, and the Deck Operations Control Station. The cameras incorporate self-contained automatic controls with all adjustments internal, except for a remotely operated focus control. The wide angle lenses are optically corrected for the normal underwater distortion to provide maximum underwater visibility. The forward television camera is mounted in a fixed position at the center at the forward end of



Fig. 26 The *Alvin* between the hulls of the R/V *Lulu* [1]

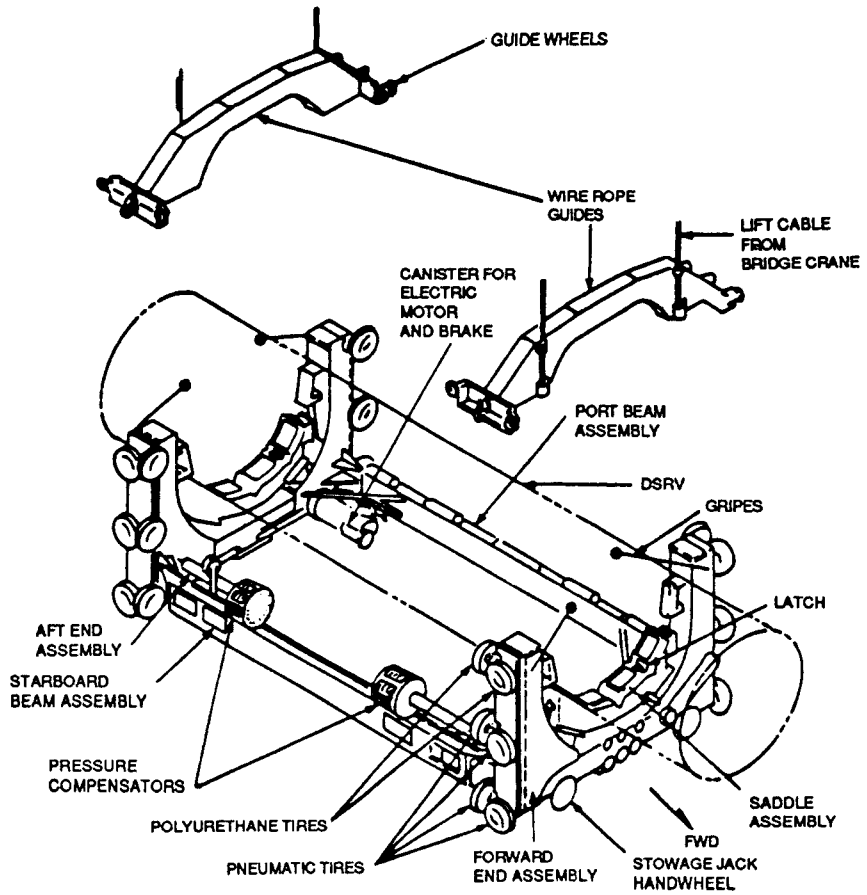


Fig. 27 DSRV cradle dolly components [19]

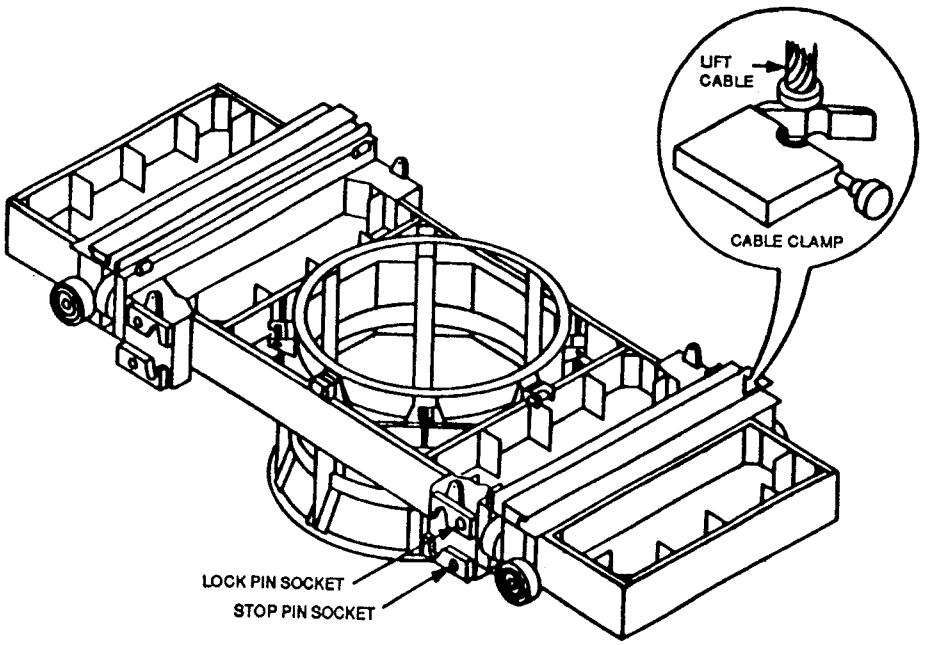


Fig. 28 DSRV lift platform [19]

the platform for viewing upward and aft. Its purpose is to view the DSRV as it engages and releases the cables and while latching to and unlatching from the cradle dolly. The after television camera is mounted to a remotely controlled underwater pan and tilt unit located at the center of the aft end of the platform. Controlled from the DSRV console in the Deck Operations Control Station, this camera can be aimed in any direction for any viewing purpose. An array of lights is used as illumination sources for the television cameras.

Table 11 lists the underwater surveillance system components and the quantity per ship [20].

Under normal conditions, the DSRV is launched and recovered with the submerged platform and cradle dolly. To launch the DSRV, it and the cradle dolly are positioned on the lift platform over the centerwell and lowered to approximately 100 ft below the surface of the sea. After the DSRV is released from the cradle dolly, by unlocking the latches, it moves away from the platform/cradle and goes to the stricken submarine.

Recovery of the DSRV is accomplished by the vehicle first ascending to approximately 100 ft. Then the DSRV is maneuvered between the after lifting cables, until the capture arms on the forward section of the DSRV encounter the forward lifting cables. It is then latched onto the cradle dolly with the hydraulic latches and hoisted to the main deck of the ASR. Command and control of the cradle dolly and DSRV latches are provided from the Deck Operations Control Station during the launch and recovery operations. Figure 29 illustrates the submerged launch and recovery of the DSRV [20].

During emergency situations the DSRV can be recovered on the surface. This operation requires that tow and tending lines be secured to the vehicle and that it be winched into position over the lift platform and cradle dolly assembly. The DSRV must be brought into position between the hulls from astern of the ship for it to be properly oriented in the cradle dolly. The cradle dolly and platform are raised

Table 11 DSRV platform underwater television surveillance system equipment list [19]

Description	Quantity per Ship
Forward TV Camera	1
After TV Camera	1
Port Cradle Incandescent Light (wire rope)	1
Starboard Cradle Incandescent Light (wire rope)	1
Port Forward TV Light	1
Starboard Forward TV Light	1
After TV Light	1
Pan and Tilt Unit	1
Port Beacon Strobe Light (battery operated)	1
Starboard Beacon Strobe Light (battery operated)	1
14-in. TV Monitor, RCC	1
14-in. TV Monitor, Pilothouse	1
17-in. TV Monitor, DSRV Console	2
Ballast Transformer, DSRV Console	3
TV Camera Control and Power Supply Unit	1
Pan and Tilt Control Unit	1
Lighting Control Panel	1
Slipring Assembly	2
Platform Underwater Junction Box	1
Cradle Underwater Junction Box	1
DSRV In-Position Indicator	2
Cradle Forward Illumination Light	1
Cradle After Illumination Light	1
Port Spot Light	1
Starboard Spot Light	1

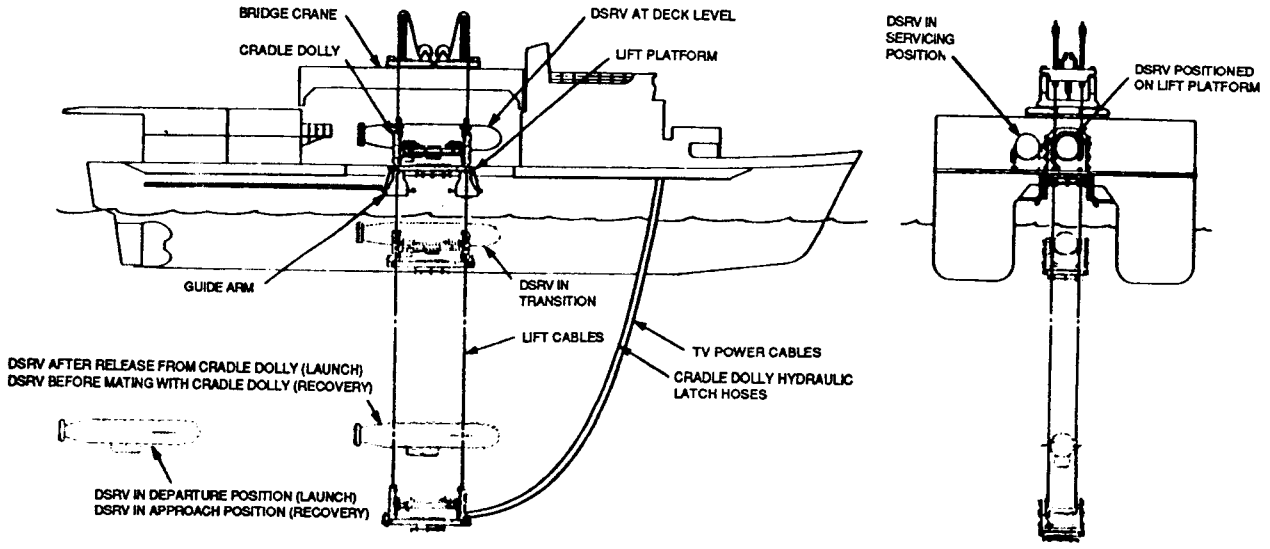


Fig. 29 DSRV submerged launch and recovery arrangement from the ASR-21 class ships [19]

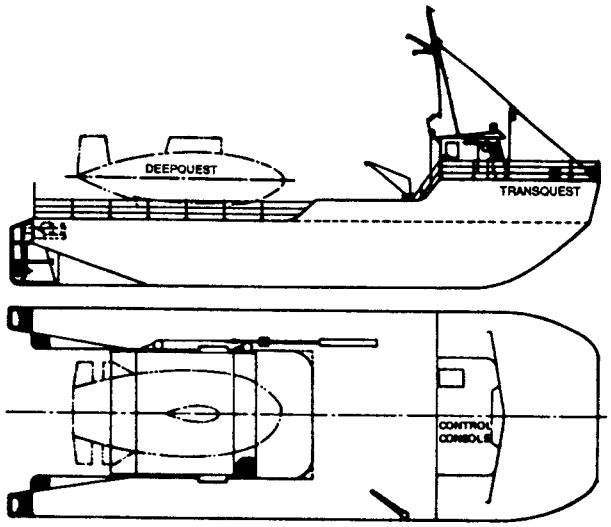


Fig. 30 M/V *Transquest* with *Deep Quest* aboard [1]

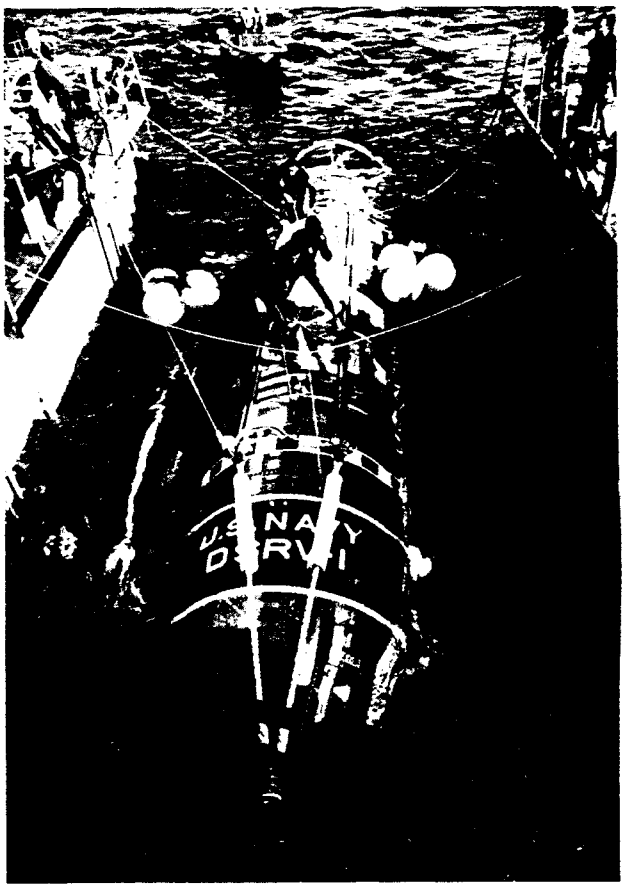


Fig. 31 DSRV *Mystic* recovered by the support ship *Transquest*. (Courtesy of the U.S. Navy)

to the DSRV, which is then locked into place and hoisted to deck level.

The third example of an elevator system is the one utilized on board the M/V *Transquest*. The *Transquest* (Fig. 30) uses a stern-mounted elevator to launch and recover manned submersibles. *Transquest* was built by Lockheed Missile and Space Company (LMSC) to support its 52-ton (dry weight) manned submersible *Deep Quest*. Today, the *Transquest* is used primarily to support the *Sea Cliff*, *Turtle*, *Mystic*, and *Avalon* during close-to-home operations.

The stern of the *Transquest* is fitted with an open well that is 62 ft long and 25 ft wide. Within the well is a hydraulically powered elevator platform, 28 ft long and 23 ft wide, capable of lifting 60 long tons.

To launch and recover a submersible, three tending lines are required per side to ensure that the submersible does not collide with either side of the well or the forward bulkhead. During launch, the *Transquest* maintains enough headway to effectively pay out the submersible. To assist the submersible out of the well, a bow line is secured to it from a capstan. As the elevator is lowered into the water, the submersible floats free of the *Transquest*. For additional protection, rubber fenders are mounted on the sides, a 2-in.-thick neoprene mat covers the deck, and a net spans the forward end of the well.

During retrieval, *Transquest* maintains a speed of 1 to 2 knots into the sea. The bow line is used to haul the vehicle into the well. With assistance from the line handlers and the submersible's pilot, it is eased into position where the cradle can lift it. Figure 31 shows the *Mystic* (DSRV 1) being retrieved by the *Transquest*, following the vehicle's first dive.

The *Transquest* is designed with four ballast tanks, port and starboard, that permit the ship to change its trim from 6.5 to 10 ft to facilitate launch-and-recovery operations. Also, the elevator platform can submerge to about 15 ft so as not to obstruct the vehicle.

The U.S. Navy, in the late 1960s, included two other stern elevator systems in concept designs of DSV support ships. Figure 32 shows the concept design of a support platform for the *Sea Cliff* and *Turtle* when they were still designated *Autec I* and *II* respectively [11]. The general configuration of the vessel itself is similar to that of the *Transquest*. Construction of these support ships did not proceed.

Figure 33 illustrates a design developed by Ocean Systems, Inc. to secure an elevator to the stern of an offshore supply ship. This handling system was designed specifically for the *Turtle* [23,24]. The operating characteristics and launch and retrieval methods are similar to those of the *Transquest*.

Table 12 lists the support ship and lift system characteristics.

The following is a general description of the lift system designed by Ocean Systems, Inc. [25]. Fundamentally, the system comprised hydraulic lifting cylinders operating a stern-located elevator platform by means of a reeved cable system. The proposed system used two hydraulic cylinders which raised the platform on the inhaul stroke of the rod. With a

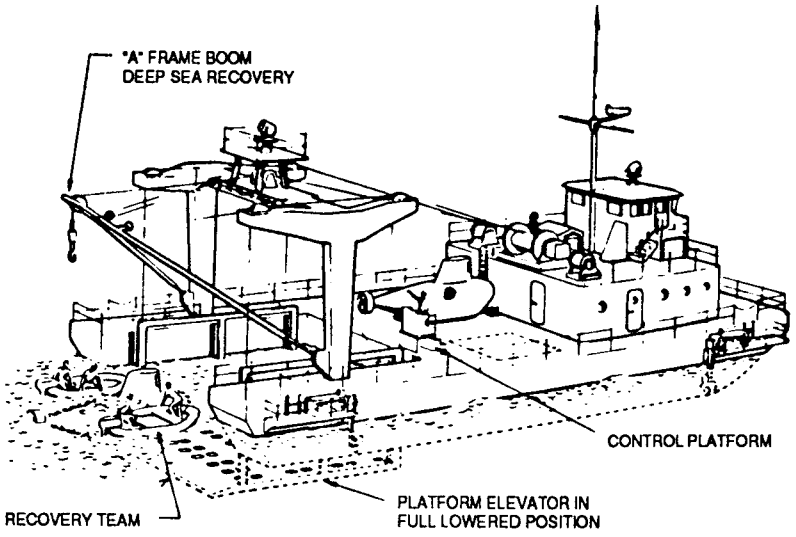


Fig. 32 Artist's concept of the *Autec* support barge [12]

maximum total load capacity of 200 000 lb, the lift system mechanism was designed with a minimum factor of safety of four to one based on ultimate strength. During normal launch and recovery operations, the *Sea Cliff* or *Turtle* would have been cradled on a specially designed transportation cart, enabling it to be moved on and off the elevator platform to a maintenance location further forward on the ship.

The lift system consisted of four wires, each attached at one end to a point on the elevator platform. The wire ropes extended vertically upward to a sheave system mounted to the ship's stern well sponsons, then forward to sheaves mounted on the ends of the guided piston rods of the lift cylinders. After passing through these piston rod-mounted sheaves, the wire ropes terminated at a deck structure. The reeving was such that the working stroke of the cylinder (rod inhaul) was one half of the distance traveled by the platform at one half its speed. The wire rope termination points incorporated spring-type shock mitigation devices, which reduced the effects of sudden load transients applied to the lift wires.

In the event of a malfunction of this primary lift system, a backup lift system was provided for emergency use. As a further safety feature, four emergency lifting padeyes were provided on the platform for use if the primary lift wires got fouled, or were otherwise unusable. More detailed information on this system can be found in reference [24].

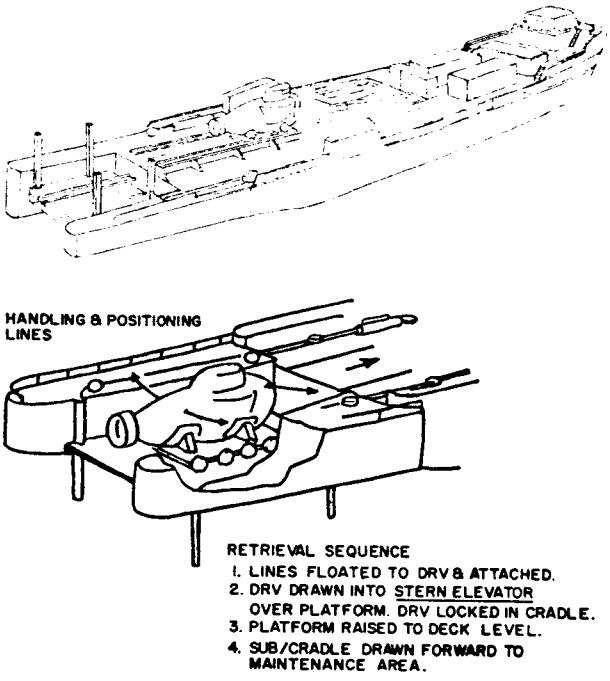


Fig. 33 Stern elevator system developed by Ocean Systems, Inc. in the late 1960s [24]

Advantages and disadvantages: Like A-frame systems, the surface cradle/elevator systems, such as those on the *Lulu* and *Transquest*, have been tried and proven. The elevator systems on both vessels have been observed to successfully recover their particular submersibles in sea state 4 conditions [12]. In addition, the *Alvin* and Navy crews who have used the cradle/elevator systems on the *Lulu* or *Transquest* have been pleased by their performances [25,26]. Other indications of past approval with these systems are the two concept designs described above.

Further advantages of a surface cradle/elevator system are elimination of the pendulation motion problem, a potential for moderate ship modifications if a stern elevator similar to the Ocean Systems, Inc. design was used, a capacity for growth and adaptation of subsystems, and favorable launch and recovery times. This system can also act to free valuable deck space that is required for a submersible support ship.

Table 12 Support ship (as modified) and lift system characteristics summary of the Ocean Systems, Inc. design [24]

Length and Beam	180 by 35 ft overall (including sponsons)
Propulsion	Two 1000-hp diesel, twin screws, and twin rudders
Generators	Two 60-kva, 3-phase, 60-cycle 115/208 vac
Speed (normal)	12 knots
Cruising Radius	6000 mi
Mean Operating Draft and Displacement with Fuel, Potable Water, and Washdown Water Tanks 100% Full	7.50 ft—731 tons
Mean Operating Draft and Displacement with Fuel, Potable Water, and Washdown Water Tanks 50% Full	6.35 ft—592 tons
Potable Water Tank Capacity	11 400 gal
Washdown Water Tank Capacity	31 170 gal
Fuel Oil Tank Capacity	36 200 gal
Habitability	16 berths, air conditioned and heated
Crew	5
Auxiliary Electronic	Fathometer
	Gyrocompass
	Navigation system radiotelephone
	Radar
	RDF
	All-wave receiver
	Underwater communications
	Submarine tracking gear
Endurance	30 days of ship operations and 15 days of DSV operations
Transverse KM @ 750 Tons	20 ft
Distance from Rear Deckhouse to Stern	109-ft deck space
Maximum Lift Capability	200 000 lb
Lift Speed	0–4 ft/s
Additions to Ship	Two 30 ft long, 3 by 5 ft box girders
Platform Vertical Travel	25 ft
Platform Size	28.25 ft long, 18 ft wide
Lift System Hydraulics	75-hp electric motor
	40-gpm pump (3000 psi)
	Two 14-in. bore by 150-in. stroke cylinders
	Ten hydropneumatic accumulators backed with nitrogen (43 bottles)
	Recycle time of 5 min
Rigging	Four 1¼-in. wires, 16-in. sheaves
Auxiliary Equipment	200-kw generator
	12-ton crane
	12 000-lb line pull hydraulic winch
	100-hp thruster
Total Weight Added to Ship (including DSV, but excluding tonnage)	128 long tons

An important disadvantage of a surface cradle/elevator system is the need for shock mitigation during recovery. Some means of protecting the vehicle must be incorporated. The Ocean Systems, Inc. design used airplane wheels on the cart and springs in the hydraulic lift system.

Other potential problems involve positioning and restraining the submersible in order to secure it in the cradle during recovery operations, and possible adverse stern well dynamics. In addition, a large number of personnel are required for launch-and-recovery operations. The *Sea Cliff* and *Turtle* require three line handlers on each side, a winch operator, two people in a small boat, one diver, and the person in charge or deck coordinator. That is a total of at least eight people who must act together to deploy and retrieve the vehicles. As mentioned before, maintaining control under adverse weather conditions requires training and experience for the launch-and-recovery team.

The main advantage of the submerged elevator system on the ASR-21 class ships is that the DSRVs

can land and take off without being affected by the relative motions caused by the air-sea interface. No lift, tow, or handling lines have to be secured or released by divers in the water. Pendulating motions are also eliminated due to the wheels and guide system included in the cradle dolly/elevator design.

The disadvantages associated with the ASR-21 system are mainly its complexity and cost. In addition, the system has poor sea mitigation qualities and is therefore limited to sea state 3 seas and less.

Discussion: Surface cradles/elevators have good technical and operational track records and have been liked by the various submersible crews that have used them. Mr. Arnold Sharp of WHOI stated that they would have seriously considered a system similar to the one on *Lulu* if a support platform was found for *Alvin* that could have incorporated an elevator system [25]. However, he also stated that they are very happy with the A-frame system now used by the *Alvin*, and that the *Alvin* has regularly operated in sea state 3 to 4 conditions. The attitude expressed by Mr. Sharp regarding the elevator system on the *Lulu* is understandable. *Lulu*'s elevator system served WHOI well for about 20 years. It is difficult to go from a tried-and-proven technique to a new one.

Likewise, the elevator on the stern of the *Transquest* has served LMSC and the Navy favorably for many years. All four of the U.S. Navy's deep submergence vehicles—*Sea Cliff*, *Turtle*, *Avalon*, and *Mystic*—have been supported successfully by the *Transquest*. Any problems have usually been associated with the limited capabilities of the ship itself and not the handling system.

Surface elevator systems have proven to be a successful and effective means of launching and recovering manned submersibles. However, at the present time there is evidence of a bias favoring A-frame systems. The main reason for this is that A-frames can operate in higher sea states. Other reasons are the costs and complexity of modifying a ship to provide an elevator system, as compared to that of installing an A-frame system. A cradle/elevator handling arrangement requires a special purpose vessel that is not always commercially viable.

Deploying and recovering a manned submersible with a cradle/elevator system underwater has only been accomplished by the U.S. Navy's ASR-21 class submarine rescue ships. The idea of this arrangement is to mate the DSRV with the support platform at about 100 ft to avoid the effects of the air-sea interface. The system is very complex and costly and requires a specially built vessel.

Docking well—This kind of manned submersible handling system incorporates a well deck in the stern of the support ship. For it to deploy and recover a submersible, the ship is trimmed by the stern to a sufficient depth to allow the vehicle to become buoyant and then be maneuvered out and away from the ship. The U.S. Navy is the only submersible operator with a support vessel, the USS *Point Loma* (AGDS-2), that has this capability (Fig. 34).

The *Point Loma* is a Landing Ship Dock (LSD) type ship that was originally built as the wet-well dock cargo ship *Point Barrow* (T-AKD-1). She is 465.5 ft long, has a 22.5-ft draft, and has a full load displacement of 10 025 tons. The ship was originally converted to support the *Trieste II* (DSV-1) and was then modified again to support the *Sea Cliff* and *Turtle*. The *Point Loma* is also capable of conducting deep-sea recovery operations, as well as assisting in Trident missile test launches. The Point Loma supported the Sea Cliff on its 20 000-ft qualification dive in 1985.

To launch a submersible, the ship is normally underway with a heading into the sea at a slow speed to limit rolling motions as much as possible. After the ship is ballasted down sufficiently to enable the submersible to float and the stern gate to be opened, a small boat, using a line to the submersible's stern, guides the submersible out of the well area. This small boat acts as a variable drogue to assist in the controlled deployment of the submersible. In addition to the stern line, a bow line is secured to the submersible and fair-leaded through a ring to a capstan forward of the well deck on the *Point Loma*. The ring is underwater and located on the centerline of the well deck. The bow line controls the speed at which the submersible is launched. There are also line handlers positioned on the wing walls of the well deck. In all, about 20 line handlers are used. A diver is required to release the tow lines and handling lines when the submersible is clear of the ship. The recovering operation is the reverse of the deployment procedures.

Advantages and disadvantages: The advantages of using a support ship with a well deck are elimination of the pendulation problem and the ability to accommodate any size of submersible that is within the lift capacity and well deck draft limitations of the sup-

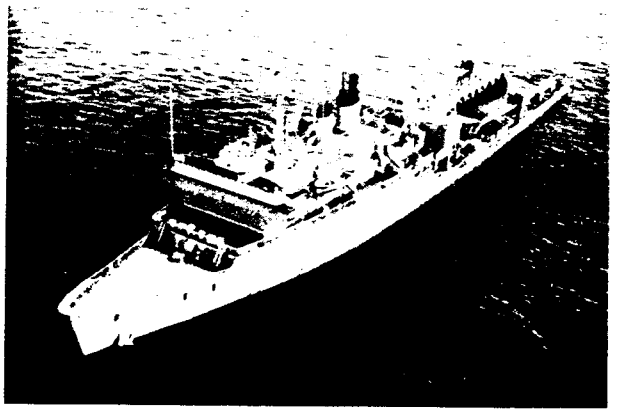


Fig. 34 Support ship USS *Point Loma*. (Courtesy of the U.S. Navy)

port ship. The *Point Loma* is actually the only LSD type ship that is set up to handle manned submersibles. Because of her size, she is capable of supporting submersibles as large as the *Trieste II*, 84 tons dry weight, excluding the almost 66 000 gal of aviation gasoline used for buoyancy.

The disadvantages of this system outweigh the advantages, however. Two main problem areas are the size of the ship and the crew required to man it, and the free surface effect of the water inside the well when the ship is ballasted down. Even though the size is advantageous for submersibles as large as the *Trieste II*, it is impractical for modern submersibles. The *Point Loma* is manned by 264 personnel: 14 officers, 230 enlisted, and 20 technicians. It is inefficient from cost, operational, and practical points of view to support a manned submersible in the size range of modern submersibles.

A serious and hazardous problem is the free surface effect, which results in the build-up of waves inside the well area. A ship rolling four to five degrees can cause a build-up of transverse waves in the ship's well of sufficient magnitude to make launch/recovery operations dangerous. The rolling motion of the ship causes the water in the well area to slosh back and forth between the wing walls bounding the deck area. Pitching motion, while capable of producing waves as high as 4 to 6 ft, is more acceptable when operating submersibles in the well.

When the vehicle is pitching, as the stern goes down, the water level in the well moves below the sea level and a surge front starts into the open well from the sea. When the stern moves up, this surge of incoming water is moved above the sea level and travels into the well as a wave front. This surge may be deflected by the ship's roll or a wave entering the well at an angle, causing it to travel diagonally across the well. Such a diagonal wave will frequently strike the walls with sufficient force to throw green water as high as 30 ft [12]. Diagonal waves may cross the well three or four times as they progress to the forward end of the well.

The *Sea Cliff* experienced other difficulties during launching and recovery operations on the 20 000-ft qualification dive [26]. There was no provision made to tie the vehicle down to the deck when the well deck was being flooded and therefore it was difficult to keep *Sea Cliff* stationary. Not only was this uncomfortable for the crew inside the pressure hull, but it was potentially hazardous to the vehicle and all personnel involved in the operation.

Another disadvantage is the large number of personnel required to conduct launch and recovery operations. Twenty people alone are required just for lines in addition to a capstan operator, divers, the small boat crew, and a deck coordinator or officer in charge. This large number of people, plus the over-50-ft distance between the wing walls, makes coordination of this particular handling system difficult.

Discussion: The problems associated with the free surface effects in the well deck area of these types of ships have been recognized, and action has been taken to overcome them. However, what may work for landing craft operations will not necessarily work for manned submersible launching-and-recovery operations. The *Point Loma* uses a wire mesh platform at the forward end of the well deck in which a number of layers of wire mesh are used in an attempt to dampen waves inside the well area, with little success, judging from the experience of the *Sea Cliff*.

2.2.5 *Submerged Launch/Recovery Methods*

Fundamentally, there are two ways to overcome the problems associated with deploying and retrieving manned submersibles under adverse weather conditions. The difficulties can be faced head-on or avoided entirely.

Confronting the problems head-on (as discussed in the last subsection) requires that the ships and handling systems be designed properly for operation in rough weather, including the possible need for stabilizing, tensioning, damping, and shock absorbing devices to avoid excessive dynamic loads. At present, these types of handling systems can conduct operations in sea states no greater than 4 or 5, which can restrict their use in some areas of the world, such as the North Sea and Northern Pacific Ocean during the winter months.

Avoiding the problems generated by unpredictable motions at the air-sea interface requires that the submersible be launched and recovered at a depth where weather conditions have no effect. The support system is then completely independent of topside weather conditions.

This section discusses this latter approach, featuring two types of submerged launch and recovery systems: mother submarines and submersible platforms or barges.

Submarine support ships—As previously mentioned, the U.S. Navy first deployed and recovered a manned vehicle from a submarine in the early 1970s, when one of the DSRVs was successfully supported by a modified SSN 637 class nuclear submarine. Since that time, both DSRVs have been launched, recovered, and mated successfully many times with submarines. Figure 35 shows the *Mystic* mated to the back of the USS *Pintado* (SSN 672).

In the event that a submarine becomes disabled and an ASR-21 class ship is not immediately available, a DSRV can be flown to a port near the emergency site. It is then placed on the after hatch of a submarine at the pier and transported underwater to the site of the disabled submarine. At the launch site, the DSRV is unlocked from within the mother submarine to conduct its rescue mission. Horizontal and vertical sonars, as well as a transponder on the

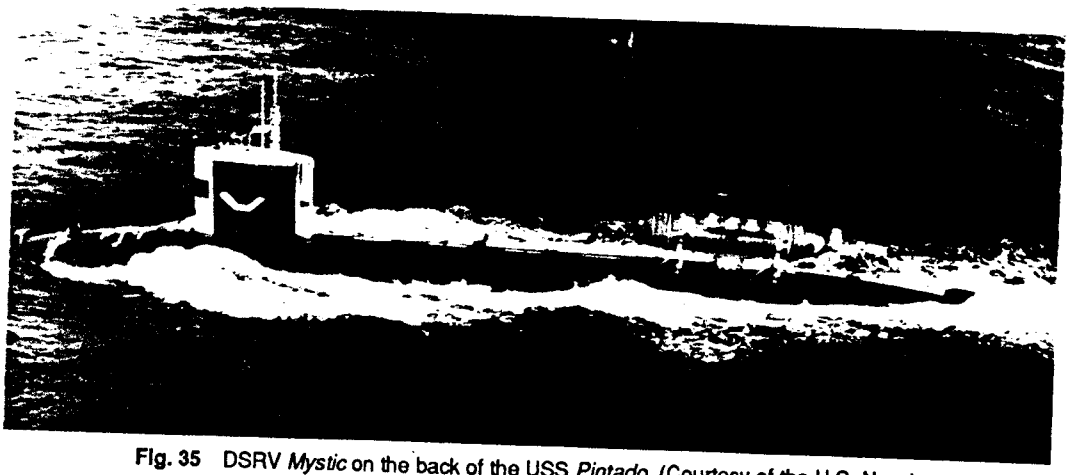


Fig. 35 DSRV *Mystic* on the back of the USS *Pintado*. (Courtesy of the U.S. Navy)

submarine, enable the DSRV to find its way to the disabled submarine and then back to the mother submarine. External television on the DSRV is used to monitor its final approach and tie-down. Reflective paint marks obstructions on the submarines and highlights the mating areas and guide lights on individual pylons. When the DSRV is secured to the mother submarine, ballast and life-support replenishment, battery charging, and other servicing and minor repairs can be conducted.

A Submarine Support Vessel (SSV) was designed and developed by Lockheed Petroleum Services, Ltd. and Kockums Shipyard in Sweden in the early 1970s [28]. Kockums modified this design to allow it to support a Submarine Rescue Vehicle, developed by them and the French company COMEX. Figure 36 illustrates the SSV and its general arrangements. Table 13 lists the specifications of the Kockums and COMEX SSV.

Both designs of the SSV require it to launch and recover a manned submersible for the inspection, maintenance, and servicing of sea-bed installations. The Submarine Rescue Vehicle designed by Kockums and COMEX would be converted to an offshore support submersible for civilian use. The entire undersea work system consists of a 1600-ton SSV equipped with a diver's system and a 50-ton submersible with a diver lockout system and a payload of 3 tons. The SSV is designed to operate down to 400 ft, where the water is calm, and launching and recovering the submersible can be performed regardless of surface weather conditions and without the use of heavy lifting gear. The SSV was designed to carry a sizable amount of equipment and tools to the site. It may also be possible to modify the design to carry one or more submersibles. The diver lockout system of the 50-ton submersible was designed for operations down to 1000 ft. The submersible itself was designed to operate down to 1500 ft.

The price of one SSV was estimated in November of 1974 to be $30 million U.S.

Advantages and disadvantages: The greatest advantage of a mother submarine launch-and-recovery system for a manned submersible is the elimination of the difficulties associated with the air-sea interface. Diving operations can be conducted year-round in any kind of weather conditions. In addition, pendulation of the submersible is nonexistent, and there

Table 13 Characteristics of the Kockums and COMEX SSV [27]

Displacement	1600 tons
Length Overall	65 m (213 ft)
Width	8.5 m (27.9 ft)
Draught (surfaced)	6 m (19.7 ft)
Height from Keel to Top of Mast	23 m (75 ft)
Diving Depth	125 m (400 ft)
Electric Propulsion Motor	1700 kw
Diesel Generators	1 × 230 kw + 2 × 1000 kw
Storage Battery	85 tons
Fuel Oil	150 tons
Speed	
Surfaced	12 knots
Snorting	10 knots
Submerged on Battery	10 knots
Range	
Surfaced at 12 Knots	3300 nmi
Surfaced at 10 Knots	4500 nmi
Endurance on Battery	
Hovering	50 h
3 Knots	32 h
5 Knots	14 h
10 Knots	1.5 h
Time for Charging Battery	3 h
Complement	
SSV, Captain	1
Crew—3 × 3	9
Cook	1
Submersible Crew— 2 × 3	6
Divers and Installation Workers	7
Total Personnel	24

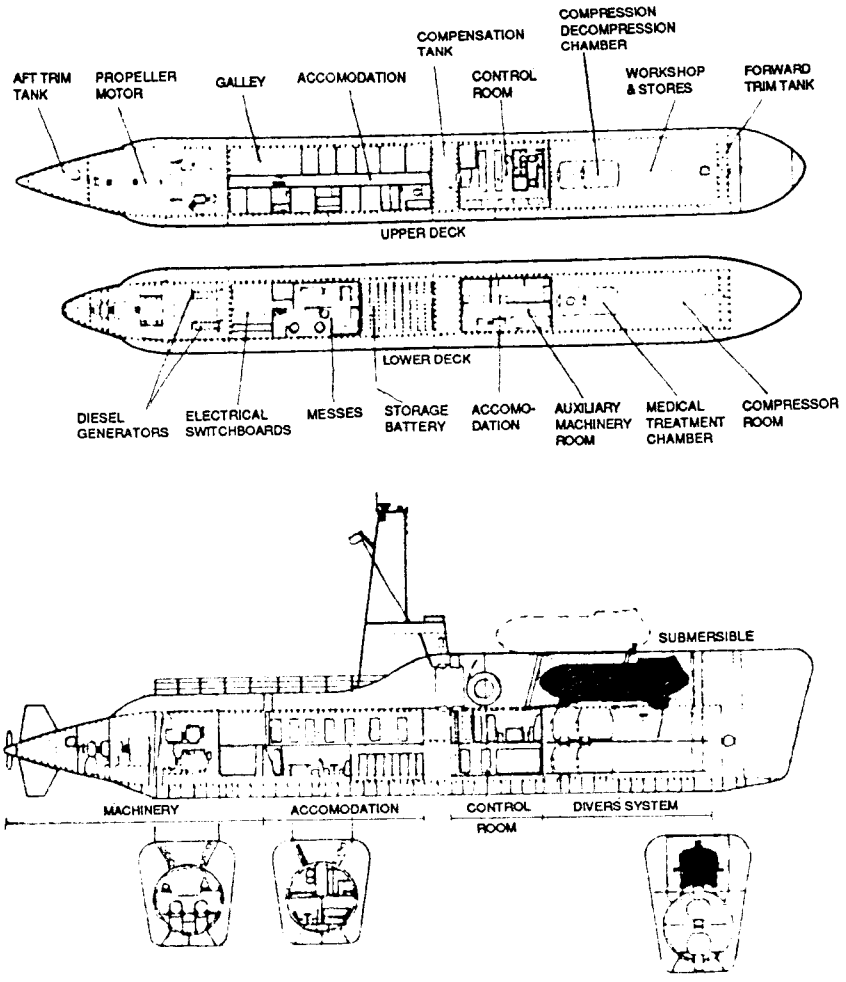


Fig. 36 The SSV and its general arrangements [27]

is no requirement for divers or line handlers during launch and recovery operations.

The main disadvantage of the submerged submarine launch-and-recovery system is that it is impossible to recover a disabled submersible that has been forced to surface. This topic is not discussed in either reference [26] or [27], and from Fig. 36 it does look as if the SSV design were developed for underwater mating only.

Other drawbacks to this system are the very specialized crew that is required, the reduced work and storage space, and the operational costs. Submariners are highly skilled seamen who require many months of training to become proficient at their jobs. Additional training would be required for launching and recovering a manned submersible underwater.

To work on the equipment outside the pressure hull requires a lot of deck space, which is not available on a submarine. The SSV design calls for a garage for the submersible when it is mated to the mother submarine. When the SSV surfaces, the water inside the garage empties, leaving a dry area for servicing the vehicle. However, it is not economically or practically feasible to incorporate a deck space in the submarine as large as that found on most surface support ships. In addition, the motion characteristics of submarines on the surface are much worse than those of surface ships. Therefore, handling heavy loads such as batteries, motors, and ascent/descent weights is dangerous on an SSV.

Operational costs may be higher for the SSV system than for the surface support vessel. This is primarily due to the fact that the submarine's batteries must be replaced every five to seven years [29].

Submerged platforms: An alternative method for launching and recovering manned submersibles underwater is to use a submersible platform or barge, as illustrated in Fig. 37. This type of arrangement is much simpler and more commercially acceptable than an SSV. Basically, it consists of an unpowered stable platform supported between two pontoons. It is also used for transporting submers-

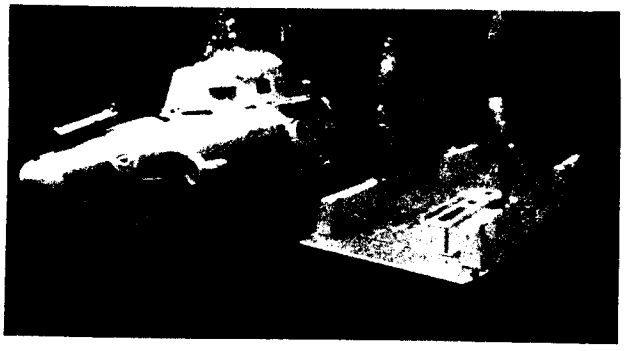


Fig. 37 The submersible *Makalii* approaching the launch recovery and transport (LRT-12A) platform. (Courtesy of NOAA's Office of Undersea Research)

ibles and therefore must be towed to and from the dive site. When launching and recovering submersibles, the platform hovers below the surface at such a depth that it is not affected by surface waves and wind forces.

This type of system has been used successfully since 1968, when the Launch Recovery and Transport (LRT) was designed by Makai Range, Inc. (now Makai Ocean Engineering, Inc.). The Launch and Recovery Platform (LARP) developed by the Naval Undersea Research and Development Center (now the Naval Ocean Systems Center, NOSC) came next in 1970. In the early 1970s, the LRT was modified and, in the mid-1970s, was lost in an accident while launching the *Star II*. In 1976 another LRT, the LRT-12A (12 000-lb payload, aluminum) was launched and is currently being used by the Hawaiian Undersea Research Lab (HURL) to support the *Makalii* (ex-*Star II*). The LRT-30A was designed and built to launch and retrieve the *Pisces V* submersible, also for HURL. As a side note, HURL is also planning to support the *Pisces V* with a modified tuna seiner and A-frame.

The basic configuration of an LRT was formulated by Makai Range, Inc. and is shown in Fig. 38. The hull configuration is that of a catamaran, chosen for stability while on the surface, with two cylindrical hulls joined by crossbeams. Each hull has a bow shape for towing and is divided into several large compartments and one small one, centrally located. The large compartments serve as main ballast tanks for submersion of the platform and the small ones serve as control or variable ballast for hovering and depth control. Each compartment is fitted with a large bottom flood valve and a smaller vent valve on top. Compartmentation is to permit attitude control and to minimize the free surface effects of a partially ballasted hull. A keel is placed on the bottom of the hulls to protect them and the flood valves from the sea bottom or deck when out of water. Strong points for a tow bridle are located at the forward inner faces of the hulls.

Over the top of the hulls and crossbeams is a lightweight grating which serves as a deck for personnel and offers little resistance to the ascent or descent of the platform. Located at deck level, outboard, are compressed air flasks for deballasting the hulls when flooded. Control of submergence, hovering, and deballasting is normally relegated to divers in the interest of simplicity. For this purpose, a pilot control station is located forward and fitted with the appropriate valves for flooding and blowing ballast tanks. Adjustment of weight and fixed buoyancy is

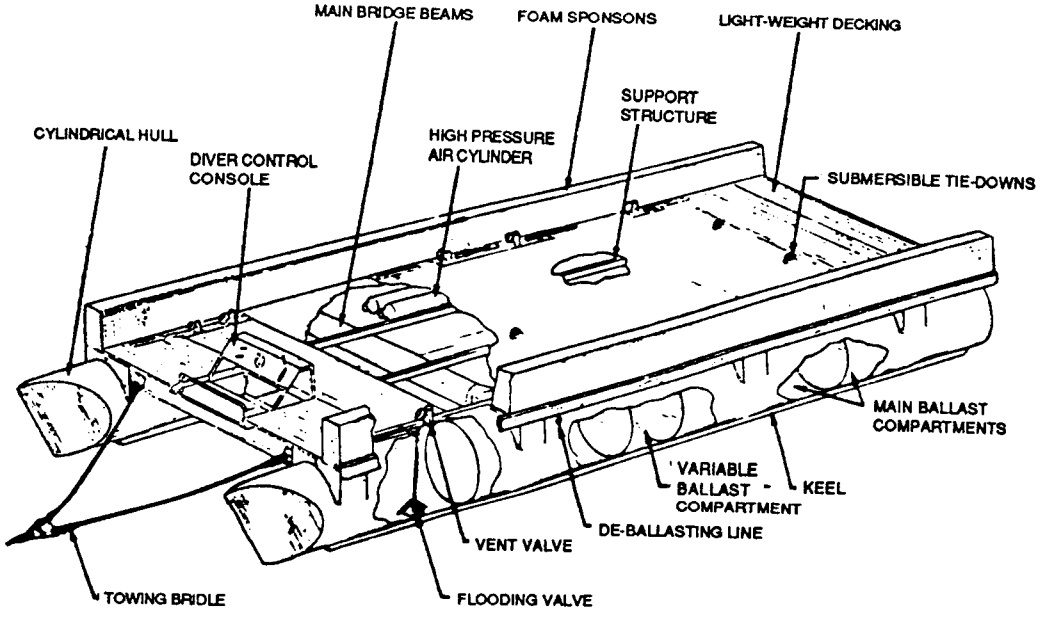


Fig. 38 Basic configuration of the LRT concept [28]

such that sinking and rising are controlled by manipulation of the small variable ballast tank. The small ballast tank minimizes air consumption during hovering.

Foam sponsons are located above decks on each side. There are several reasons for this fixed buoyancy material. First, it provides needed buoyancy to offset the underwater weight of the platform and bring it within range of a neutral state such that the variable ballast tanks can control it. Second, its outboard position and air weight increase the moment of inertia of the platform and hence increase resistance of the platform to rapid attitude changes due to passing swells. Third, it aids in stability during the critical passage through the air/water interface. The foam sponsons act here as "buoyancy stabilizers" by offering a restoring moment should any imbalance take place. Finally, the sponsons serve as an effective means for raising the center of buoyancy and increasing the platform righting arm after total submergence.

The deballasting system on the LRT included twelve 300-ft^3 2400-psi compressed-air cylinders. Two of the cylinders were used for emergency blow reserve and had a separate manifold system. Deballasting the main tanks as well as controlling the variable ballast system was accomplished at the diver control pit.

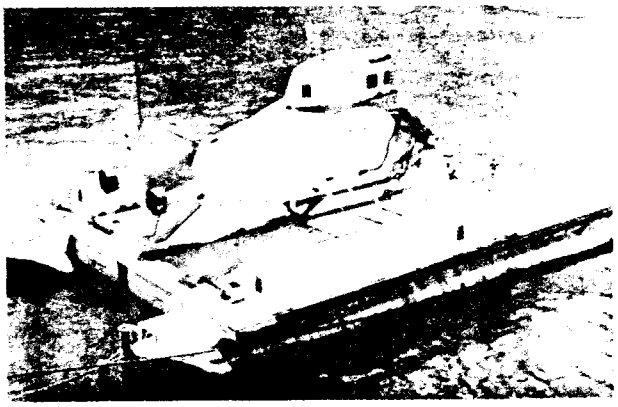
The LRT measured 33 ft long by 18 ft wide and weighed 24 000 lb. The hulls were straight cylinders, each with an inside diameter of 52 in. divided into five compartments. Each bow end was shaped to form a 60-degree half cone below the mid-diameter and was cut off to a 60-degree parallel-sided wedge above this. This design provided bow lift and forced the bow waves to intersect ahead of the forward bridge.

Another platform, the LARP, was 35 ft long, 18 ft wide, 7 ft deep, and had an air weight of 17 000 lb. Its cylindrical hulls had an inside diameter of 48 in. and were made of fiberglass. The 60- by 60-degree chisel-shaped bow was also included in this platform.

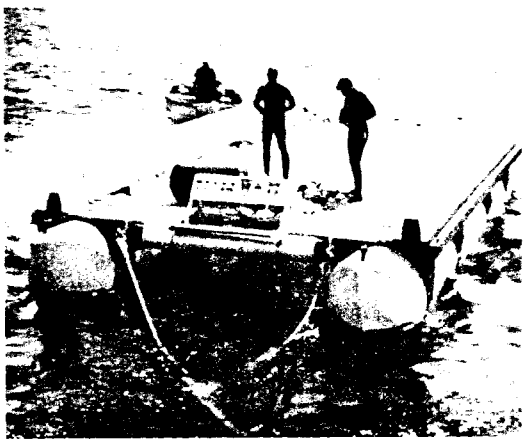
The LARP was designed to be lightweight; therefore, low-corrosion materials were used throughout the platform. It could be disassembled and transported by air to any location in the world and had an optional remote control capability. It was designed to handle the submersibles *Makalii* (10 500 lb) and *Deep View* (11 000 lb).

Figure 39 shows the original LRT, the modified LRT, and the LARP platforms. The modified LRT had the same dimensions as the original LRT, but weighed 20 000 lb and could support up to 12 000 lb. By 1976 the modified LRT was supporting the *Star II* submersible, averaging three dives per week in seas up to state 4 [21].

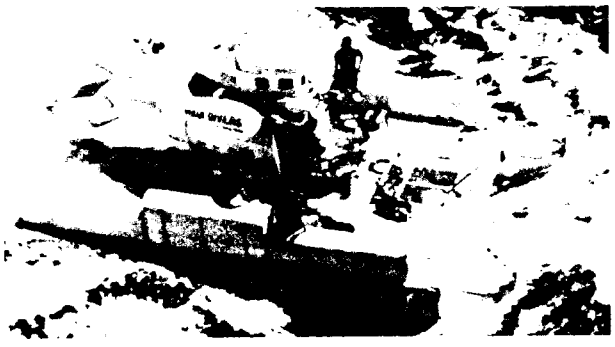
The LRT-12A (A is for aluminum) is larger than the LRT, and is shown in Fig. 40. It was designed this way to enable it to endure long sea passages and to be safer than the previous designs [9]. The LRT-12A was also designed by Makai Ocean Engineering, Inc. and was launched on May 24, 1976. It is 37 ft long, has a beam of 17 ft, and weighs 17 500 lb dry. Its operating depth is 50 to 100 ft and it has a maximum tow speed of 10 knots. The LRT-12A has recovered *Makalii* in 18-ft waves and 36 knot winds [29].



(*a*)



(*b*)



(*c*)

Fig. 39 Submerged platform launch and recovery systems: (*a*) original LRT-4 with the *Star II* aboard [1]; (*b*) LARP [22]; (*c*) modified LRT-8 being towed with the *Star II* aboard [22]

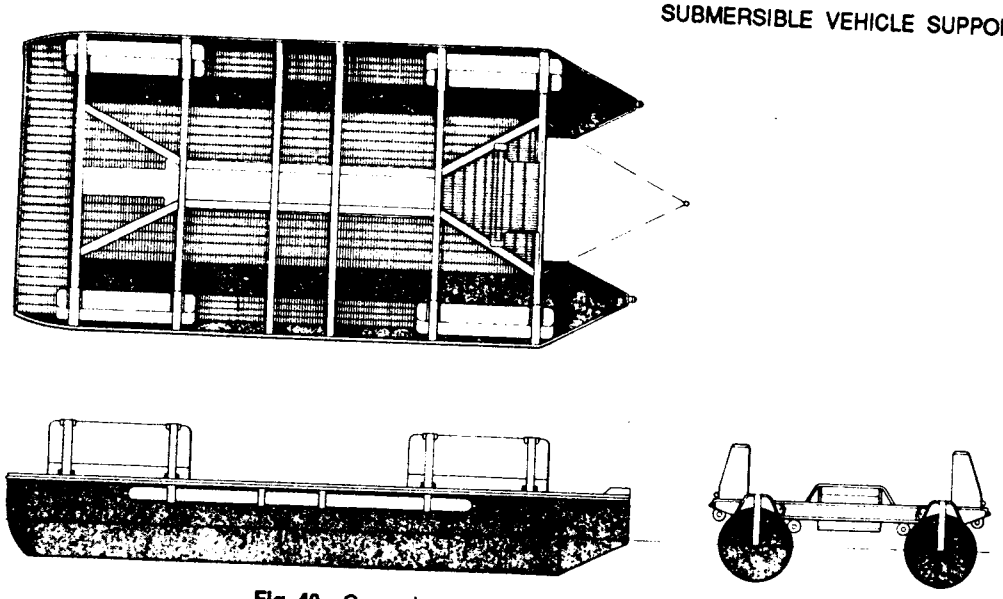


Fig. 40 General arrangement of the LRT-12A [10]

The manned submersible launch-and-recovery operations are as follows [27]: The submersible is checked out in transit to the dive site. Once on station, the submersible operators are locked in and the platform submerged, usually to a depth of 60 to 65 ft. While the platform operator remains on station, divers release the tie-downs, freeing the submersible to commence its mission. The platform is blown to the awash surface condition (to conserve air) or, in the case of the LARP, it can be "hung" on a surface buoy to await the return of the submersible.

Upon receipt of a signal from the submersible of its return, the platform is again positioned and receives the submersible from the stern to avoid interference from the barge towline. Since there is no relative motion between sub and platform, the divers can attach the hold-down lines and pull the sub down securely. Again, the platform is brought to the surface, this time to full freeboard, and towed back to port. A typical time span for a launch is 5 to 6 min, with 8 to 10 min required for a recovery.

Advantages and disadvantages: Estabrook and Horn [28] discuss the advantages of using a submerged platform to launch and recover submersibles:

By increasing the limiting sea state in which a submersible can operate, its useful time is increased and it becomes more cost effective. Furthermore, the gross cost of an LRT should be less than a marine-type crane with comparable manned submersible handling capacity. Finally, the following points can be made to further illustrate the economic attractiveness of the concept:

- Fabrication is inexpensive and can be accomplished in any small "boilerworks" rather than a high-overhead shipyard.
- The basic platform needs no pressure hulls, propulsive systems, complicated or sensitive plumbing, electrical systems, or winches, and no exotic materials.
- Whereas a deep submersible should be hauled out and rinsed with fresh water after each dive, an LRT platform need only be hauled out once a year for painting and hull inspection.
- For most operations, only a tow boat of opportunity is required, not a large seagoing surface ship.

The main disadvantage of these platforms is that they must be towed. Towing is extremely slow, especially as the sea state increases. Both the LRTs and the LARP were designed for towing at speeds of 6 to 10 knots. Extended missions, hundreds of miles from land, may prove to be expensive and logistically impractical. Another ship or barge, in addition to the tow boat, may be required to carry all the support vans, people, and spare parts. In addition, there are questions of how maintenance would be performed on equipment within the fairing of the submersible, and how batteries, motors, or other components could be replaced at sea.

Operating in murky waters, where the visibility is poor, is hazardous to the divers releasing or securing the submersible to the platform. The submersible is supposed to make its approach over the stern of the platform; under reduced visibility conditions this may be a very difficult operation. It would require the divers to relay the pertinent information to topside personnel, who in turn must relay the information to the submersible crew. In addition, the topside crew would have to watch the operation on sonar to assist the submersible's pilot if he went off course, or if other problems arose. Also, the divers' equipment would have to include an underwater communications microphone and receiver. At the present time, they use SCUBA and hand signals.

2.3 Airborne Transportation Systems

With the availability of large cargo aircraft, the shipping of small and medium size submersible vehicles by air has become a feasible mode of transportation. The U.S. Navy's DSRV, weighing approximately 82 000 lb, can be flown aboard a U.S. Air Force C-141 Starlifter (Fig. 41).

The large cargo air transports can provide a rapid, efficient, reliable and long-range method of transporting submersibles to most locations throughout the world. However, it must be pointed out that most submersibles will probably be transported primarily by ship. For these vessels, transportation will be provided from their home base to the site of operation by their support ship.

Having assumed that air transportation is a system requirement, this section will discuss the pertinent characteristics of transport aircraft which form mission internal constraints on the overall submersible system design. These characteristics will be identified by going through the steps that must be taken during a conceptual design process. To do this, a specific aircraft, a C-5 military transport, has been chosen in order to list specific details. At the end of this section, the results of a study done by the Lockheed Missile and Space Company (LMSC) [30] are given, as well as a loading configuration for transporting the DSRV aboard a C-5. Although the equipment chosen for our examples is military, the principles highlighted are applicable to any submersible system.

The objectives of this section are to give a background to the system designer who will be responsible for choosing aircraft as a mode of submersible vehicle transportation, and to provide the submersible designer with a knowledge of the constraints that the aircraft characteristics impose on his submersible design. In choosing an aircraft, the system designer must evaluate the overall system requirements to determine what support equipment, in addition to the submersible, may have to be carried aboard the aircraft. The support equipment may include items such as spare parts, handling fixtures, wheeled vehicles, and auxiliary power equipment.

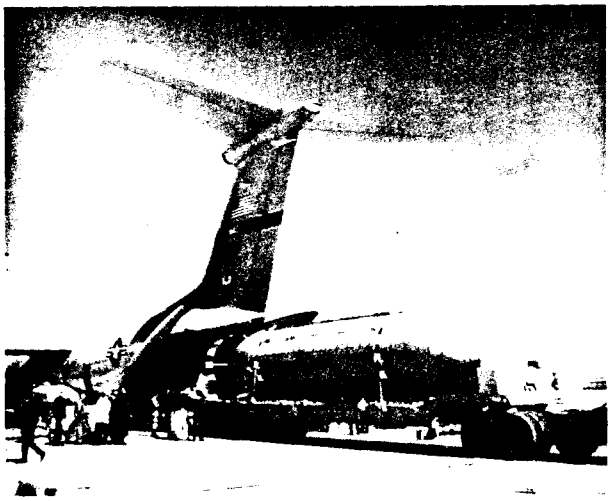


Fig. 41 DSRV loading aboard a C-141. (Courtesy of Lockheed Missile and Space Co.)

To determine what type of aircraft will be used, the system designer proceeds from overall system requirements. These mission requirements may call for a particular aircraft or class of aircraft to be used, or they may contain a less specific requirement such as "the submersible will be air transportable."

In the first instance, the designer investigates the characteristic of the aircraft specified. Many of these characteristics are mission internal constraints, and are used as inputs to the submersible design iteration process. Depending on the operational requirements of the submersible, the aircraft requirements may impose severe constraints on the submersible design in the areas of weight, size, configuration, and resistance to shock and vibration. If these constraints affect the operational capabilities of the submersible, it may be that certain aircraft will have to be rejected.

In the second instance in which the aircraft is not specified, the system designer will wait for a number of preliminary submersible design iterations, based on operational requirements, to be performed. As the preliminary submersible design takes shape, the designer is now in a position to make estimates on what the submersible's size and weight will be and to determine what aircraft can carry it. At this point also, he or she must determine if the submersible will be airlifted infrequently or will be transported regularly in this manner. If air transportation is used quite often, the designer must look at the possibility of using the smallest aircraft to do the job, without compromising the operational capabilities of the submersible. The economic savings, over the life of the system realized by the use of a smaller aircraft, must be determined. These savings have to be compared to the additional cost of designing and building a submersible compatible with this aircraft.

If the submersible is flown rarely, then the designer should look at aircraft that will "comfortably" accommodate the submersible; he or she should also point out to the user the cost of having this capability.

The above discussions point out that the mission internal constraints of an air transport system generate influence vectors that can vary in magnitude from mild, where the least impact aircraft is chosen, to severe, where the aircraft may have a dominant influence on the physical characteristic of the submersible. The system designer must be aware of the economic and operational implications that may be imposed by air transportation requirements and must weigh them in his or her analysis.

The maximum weight and volume of submersible equipment carried by an aircraft and their arrangements in the aircraft are limited by

- the payload capability of the aircraft,
- the cargo compartment size,
- the cargo loading door dimensions,
- cargo loading zone limitations,
- the cargo center-of-gravity limits, and
- the cargo restraint and tiedown fittings in the aircraft.

The following paragraphs will review some of these important details. This will help the system engineer realize what capabilities and characteristics exist and will cause the submersible designer to consider the constraints they impose.

The first step in any preliminary analysis to determine potential aircraft for submersible transportation should start with a list of aircraft that have a payload capability which may meet the requirements of the submersible system. The payload capabilities of a number of large cargo aircraft are given in Table 14.

The next step in the design study is to determine the size, weight, and configuration of the submersible system equipment that can be loaded into an aircraft without damage to the aircraft structure. The engineer can make this determination from each aircraft's loading and operations manuals.

Table 14 Payload capability

Aircraft	Payload, lb (kg)	Gross Volume, ft^3 (m^3)
C-141B	70 847 (32 136)	8722 (247)
C-5	291 000 (131 998)	34 795 (985)
747 (Container)	245 000 (111 132)	37 485 (1062)

C-5 cargo area dimensional information is shown in Fig. 42 [31]. This degree of detail is adequate for the conceptual design phase. Estimates of the length, width and height of a submersible vehicle that can be located within the aircraft can be made from this figure. However, the cargo doors on an aifcraft may be the limiting factor on the submersible's configuration.

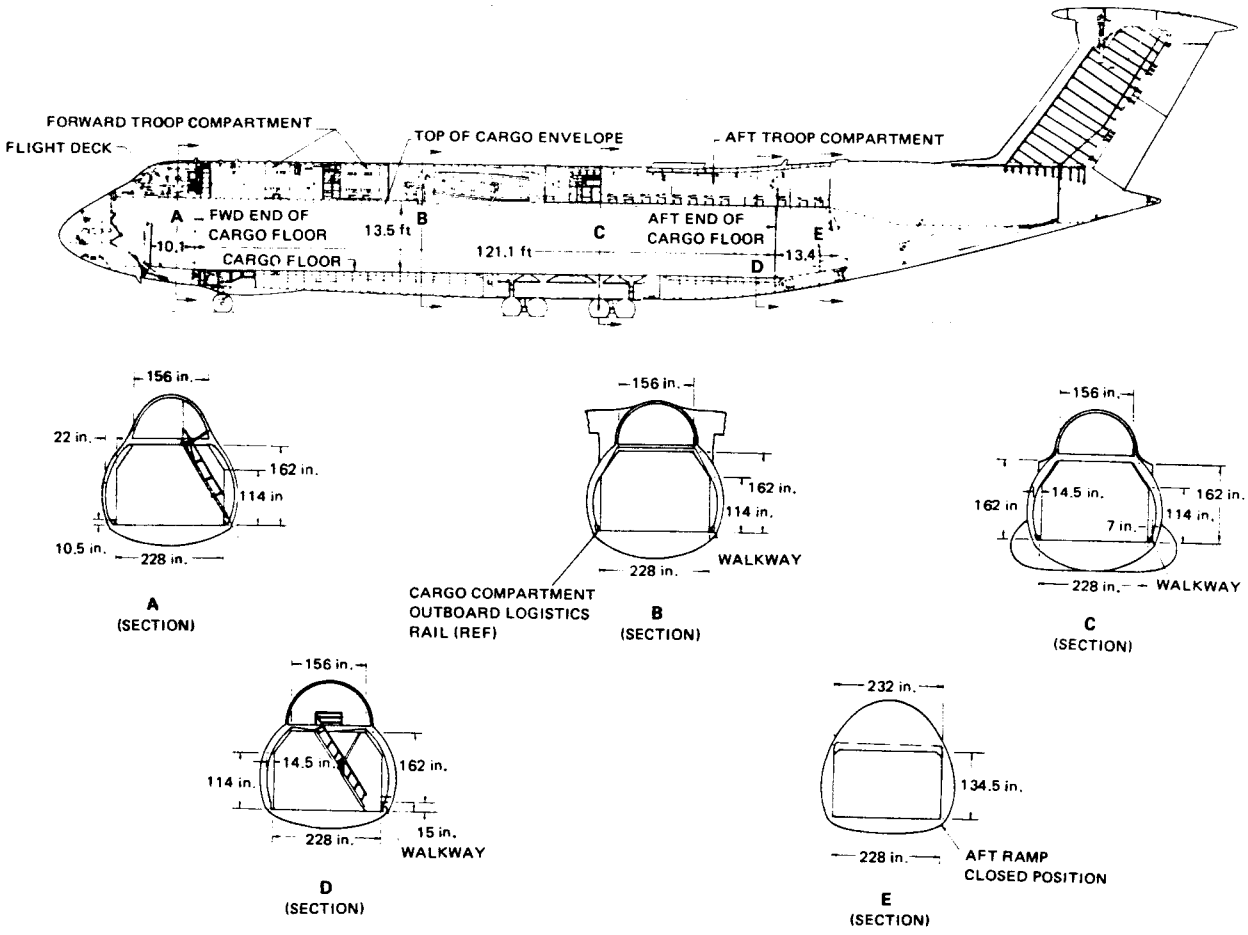


Fig. 42 C-5 aircraft

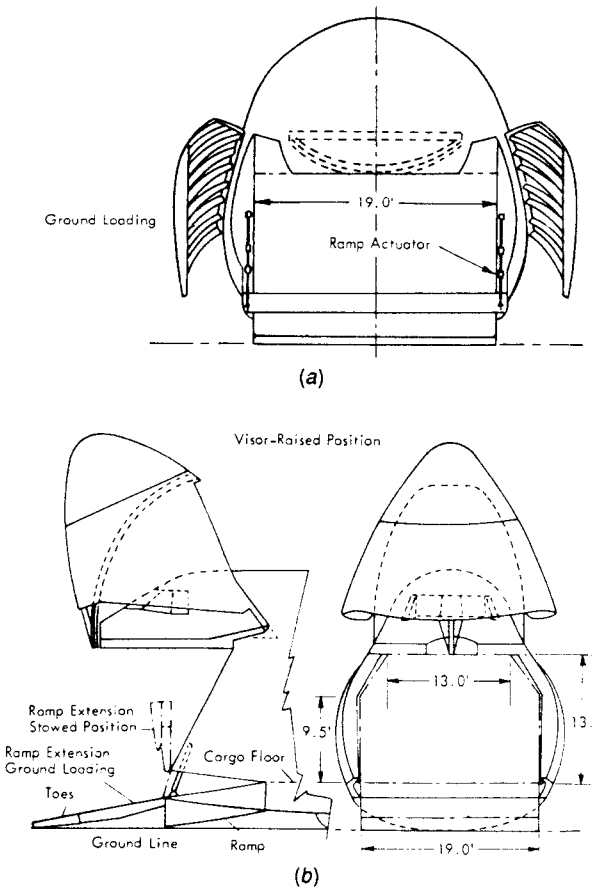


Fig. 43 (*a*) C-5 aft cargo opening; (*b*) C-5 forward cargo opening

Figure 43 shows the two cargo openings on the C-5 aircraft. For truck bed loading where the aft ramp is level with the cargo floor, the height of the rear entrance is 9.5 ft. With the aft ramp lowered to the ground, this height can vary from 12 ft 10 in. to 13.5 ft, depending on the kneeling position of the aircraft. Cargo floor load diagrams and tables provide information on the load-carrying capability of the cargo compartment floor of an aircraft. The system engineer uses this information to determine how the weight of the submersible system must be distributed. The floor load limits generate mission-internal design constraints on the submersible system and the handling fixture or vehicle it may be mounted on during air transportation. For wheeled vehicles, the floor limits govern the maximum wheel and axle loads.

During the conceptual design phase, the floor loading diagram for the C-5 shown in Fig. 44 provides adequate information. As the design proceeds to the preliminary phase, more detailed information can be obtained from the aircraft loading manuals. Personnel qualified in aircraft load analysis should be consulted for verification of the preliminary design.

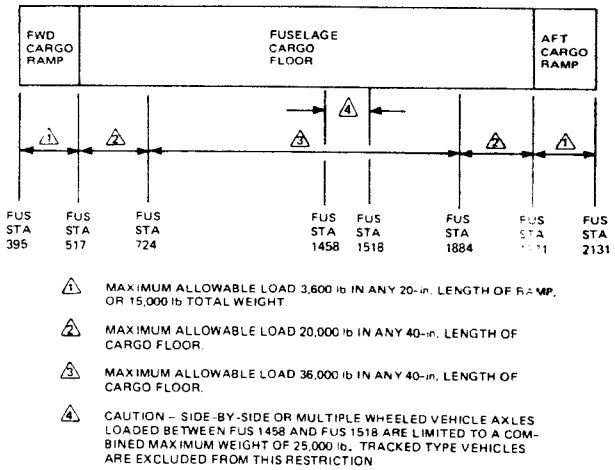


Fig. 44 C-5 floor loading

The location of the center of gravity of the total cargo load in an aircraft must be controlled. Figure 45 shows the maximum forward and aft limits for various loads in the C-5. These limits are related to fuselage stations and reflect the aircraft maximum fuel weight condition with regard to the aft boundary and flight with empty tanks in the case of the forward end of the envelope. For cargo that is made up of a number of items, the total cargo center-of-gravity CG is found by using the relation

$$CG = \frac{\sum \text{Moments}}{\sum \text{Weights}}$$

where the moments are the products of the item weight and the item fuselage station in inches.

Cargo aircraft are equipped with tiedown fittings and receptacles for cargo restraint. (The C-5 has 304 flush, permanently installed rings, each designed for a load of 25 000 lb.) Design of the submersible and its carrying devices requires the consideration of the location of these devices and their load limits. Points must be provided on the submersible system that will accommodate the tie-down devices. The submersible equipment should be restrained with a symmetrical tie-down pattern, using the same size and type of tie-down chain at each point. The chain lengths should be of the same final length if possible. Cargo floor drawings showing details of tie-down fittings and other restraint hardware are contained in the aircraft loading manuals. The tie-down floor plan for the DSRV mounted on the Land Transport Vehicle (LTV) is shown in Fig. 46. This configuration, secured for flight, is shown in Fig. 47.

Cargo aircraft also may be equipped with roller conveyors installed in the cargo floor. These systems can be utilized for on-loading and off-loading submersibles that are not mounted on wheeled vehicles. The details of interfacing with these systems, and

also with such systems as the cargo winches and ramps, are left to the contract design phase.

Aircraft-induced environmental factors are of prime concern to the submersible subsystem designers, particularly to the structural engineers. The submersible equipment must be designed to withstand the acceleration and vibrational loads generated during taxiing and during flight without any degradation in performance. In addition, they must be designed to be restrained so that during a controlled crash landing no parts will break loose and there will be no catastrophic structural failure. Military specification MIL-A-8421 details general design and performance requirements for air transportability of military equipment. For flight and taxiing loads, it specifies that equipment shall be designed to withstand acceleration loads of 3 g for a minimum of 3 s applied independently along each of the longitudinal and vertical axes in each direction, and 1½ g for a minimum of 3 s applied independently along the lateral axis in each direction. The controlled crash landing loads imposed are a minimum of 9 g in either direction applied independently along each horizontal axis for a minimum of 3 s, and a minimum of 4½ g vertically downward again for the same duration.

When the actual load factors for a specific aircraft are known, they should be used as the mission internal constraints. For example, the C-5 load factors are given in Table 15.

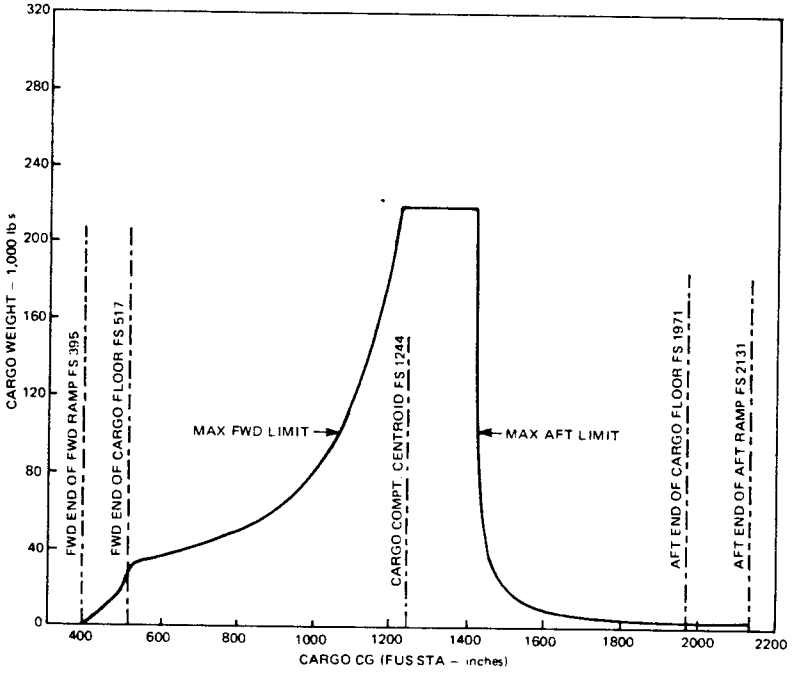
Vibration-induced loads require that the system be designed to withstand +2.5 g peak amplitudes that vary in frequency from approximately 5 to 200 Hz.

As an example of an aircraft loading arrangement study, one result is given of a preliminary analysis performed by LMSC to determine the feasibility of using a C-5 to transport the DSRV system (including support equipment) [30].[3] Figure 48 shows the proposed loading configuration. Table 16 lists all the cargo items with their weight, fuselage station location, and moments. For the total cargo weight of 192 085 lb, the cargo CG station location was found to be at fuselage station 1299. Checking with Fig. 45, it is seen that this CG location is within the acceptable limits.

Aircraft floor loading limits were verified by plotting the item weights on a floor loading limits (maximum shear) envelope. Figure 49 shows that this arrangement stayed within the envelope.

The loads plotted in this figure were all assumed to be concentrated at their respective CGs. During the next iteration of the design process, the distribution of the loads should be considered. For example, the axle or wheel loads for the wheeled vehicles and dolly-mounted equipment should be determined. In addition, the distribution of the weight of the submersible mounted on its cradle and other equipment

[3]Note that this does not reflect the actual arrangement used, which is shown in Fig. 46.



NOTE THESE DATA ARE FOR PLANNING PURPOSES ONLY. THE RESULTS ARE APPROXIMATE. FINAL LOADING FOR OPERATIONS MUST BE CHECKED FOR PARTICULAR AIRPLANES USING THE HANDBOOK OF WEIGHT AND BALANCE DATA.

Fig. 45 C-5 cargo loading envelope. (Courtesy of Lockheed Missile and Space Co.)

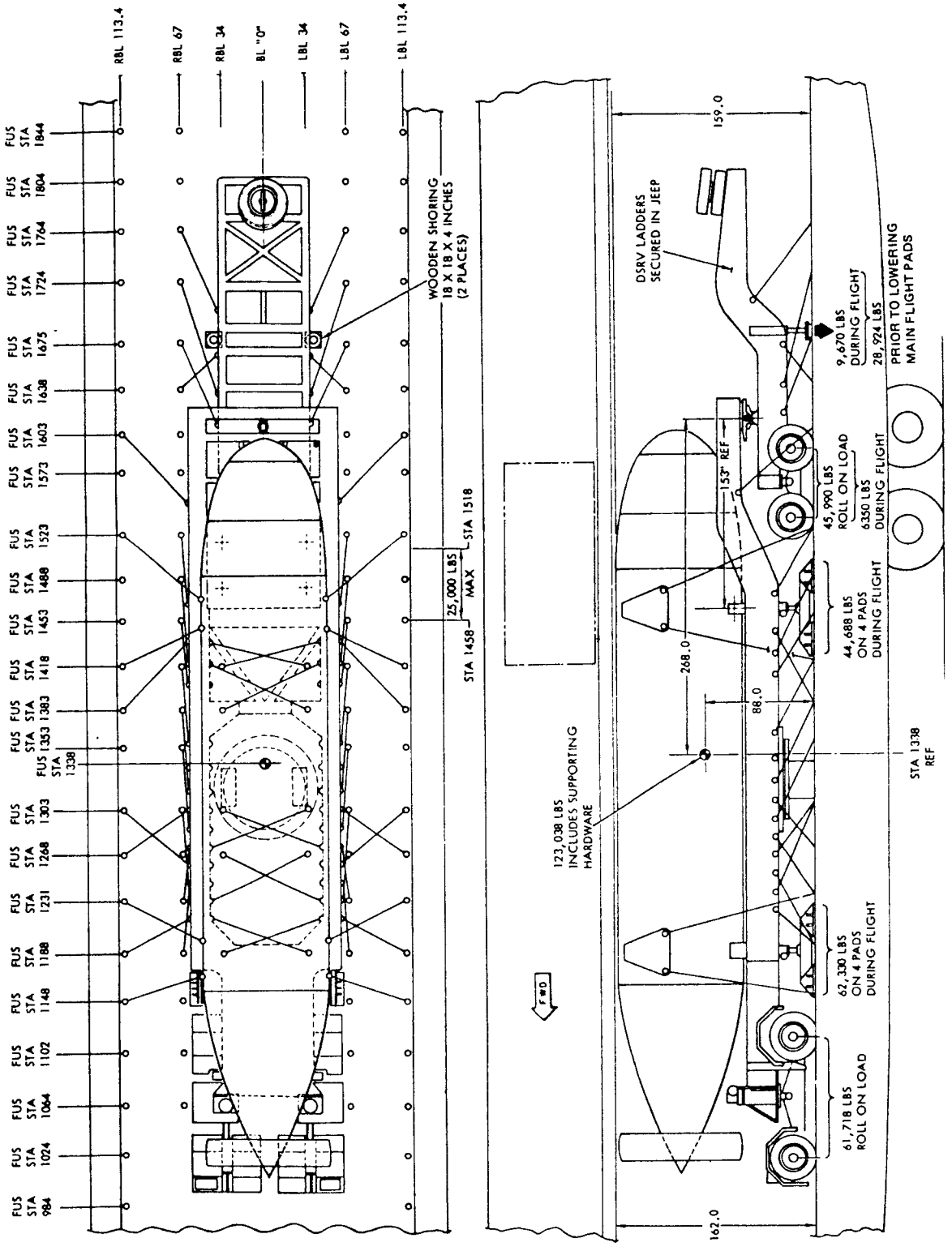


Fig. 46 C-5 tie-down floor plan for the DSRV system. (Courtesy of U.S. Department of Defense)

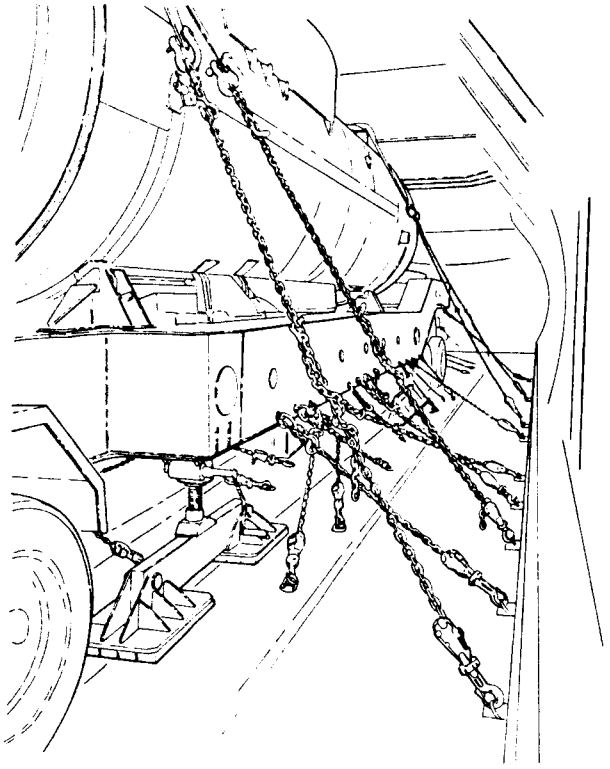


Fig. 47 DSRV on LTV secured for flight. (Courtesy of U.S. Department of Defense)

mounted on pallets or shoring must also be calculated. With these new weight locations for the cargo, a more detailed plot can be generated.

Another area of concern that must be considered during air transportation of submersible systems is that of handling flammable and toxic materials and high-pressure gases. Typically, such materials as hydraulic fluid, silicon oil, alcohol, and high-pressure air are carried in the aircraft. These materials must be carried in marked drums which can be strapped down on aircraft pallets. High-pressure air must be carried in tanks that have documented certification and must also be packed in foam-lined boxes that are tied down on pallets. Specific information on the methods for handling the above types of materials aboard aircraft should be obtained from personnel responsible for cargo handling planning. They can supply details such as which materials are allowed on board, what the tie-down and locating requirements are, and which materials have venting requirements.

Table 15 C-5 load factors

Direction of Load	Amplitude, g	
	Transportation Load Limits	Controlled Crash Ultimate Loads
Longitudinal	±0.2	+3, −1.5
Lateral	±0.6	±1.5
Vertical[a]	+2, −3.5	+4.5, −2

[a]Includes acceleration due to gravity.

3. Maintenance and Repair Facilities

As part of an overall submersible system, a maintenance and repair facility is an important element. It is the responsibility of this facility to maintain the submersible vehicle in a full state of operational capability, to repair submersible subsystems after failures or damage, and to perform overhauls and preventive maintenance to achieve the expected life of the vehicle.

The size and complexity of maintenance and repair facilities reflect the type of submersible that must be maintained. At one end of the spectrum, one might have to maintain a small towed vehicle. This requires a small work area with sufficient tools and equipment to service a few specialized subsystems. At the other end, one can have a larger manned submersible with complex subsystems. Supporting this type of vehicle requires a sizeable facility with

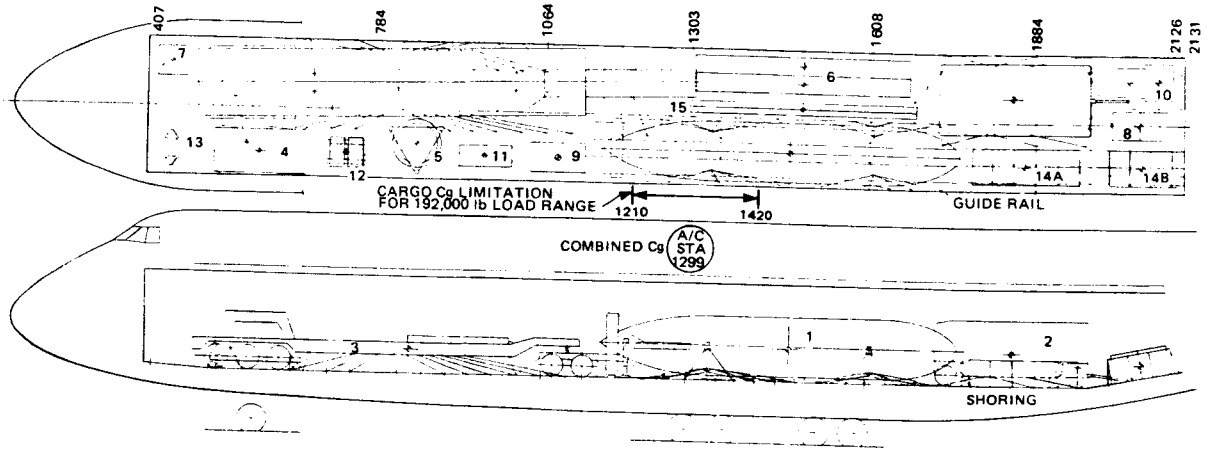


Fig. 48 DSRV C-5 loading configuration

Table 16 Loading arrangement study for a DSRV system aboard a C-5

Number	Equipment	Weight	Σ Weight	Station	Moment	Σ Moment
1	DSRV and Cradles	73 000	73 000	1470	107 310 000	107 310 000
2	Support Van with DSRV Sling	21 000	94 000	1845	38 745 000	146 055 000
3	Land Transport Vehicle	45 385	139 385	845	38 350 325	184 405 325
4	DSRV Pylons	9 150	148 535	585	5 352 750	189 758 075
5	Transfer Skirt	3 600	152 135	850	3 060 000	192 818 075
6	(IRS) DSRV Flotation Devices	6 800	158 935	1490	10 132 000	202 950 075
7	(IRS) Umbilicals	800	159 735	435	348 000	203 298 075
8	3 Rubber Boats	1 200	160 935	2050	2 460 000	205 758 075
9	Charging Station	2 000	162 935	1085	2 170 000	207 928 075
10	DSRV Towing Bridle (IRS)	650	163 585	2085	1 355 250	209 283 325
11	Diesel Generator (IRS)	5 400	168 985	965	5 211 000	214 494 325
12	IRS Restraint Devices	600	169 585	730	438 000	214 932 325
13	Two 30-ton Jacks	1 500	171 085	435	652 500	215 584 825
14A	Five Mission Containers	4 200	175 285	1865	7 833 000	223 417 825
14B	Three Mission Containers	1 800	177 085	2060	3 708 000	227 125 825
15	Pylon Adapter Beam and End Dollies	15 000	192 085	1490	22 350 000	249 475 825

$$\text{Cargo CG} = \frac{\text{Total } \Sigma \text{ Moments}}{\text{Total Weight}} = \frac{249\,475\,825}{192\,085} = 1298.8 \text{ in.}$$

a number of people trained as electronics and electromechanical technicians, machinists, and electricians.

Maintenance facilities are normally located at the home base of the submersible vehicle. However, some portable portions of the facility are taken to sea to support a submersible operation on site.

Figures 50 and 51 depict facilities that are used to support manned submersibles. Figure 50(*a*) shows the maintenance facility at the U.S. Navy's Submarine Rescue Unit, which is used to support the DSRV. Shown is an overhead movable crane used during assembly and disassembly and when moving the vehicle. Support dollies can be seen in Fig. 50(*b*). Figure 51 shows the high bay area at the maintenance facility for the manned submersible *Alvin*, which is operated by the WHOI. Again, an overhead crane is shown in addition to welding facilities and a metalwork area.

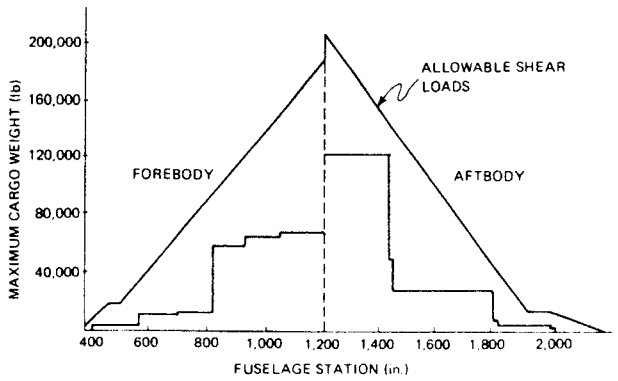


Fig. 49 C-5 shear diagram

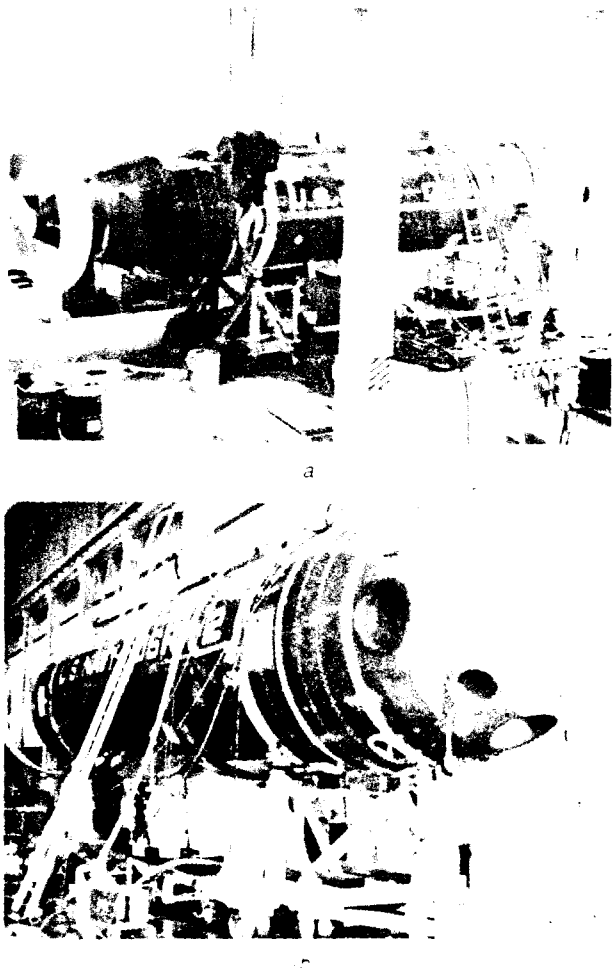


Fig. 50 (*a*) DSRV at the maintenance facility; (*b*) DSRV on support dollies. (Photographs by P. Lanner)

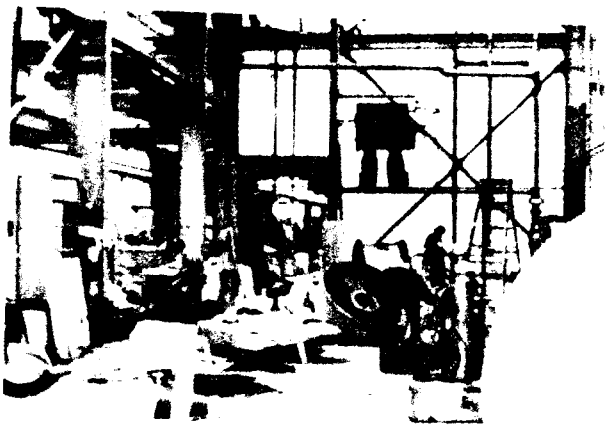


Fig. 51 *Alvin* maintenance facility. (Courtesy of WHOI)

The following list, based on the typical requirements to support a manned vehicle, indicates the types of items required to establish a maintenance facility for a submersible.

a. building to contain the facility
b. vehicle work area sufficiently large to support assembly/disassembly
c. overhead movable crane with adequate weight capacity
d. offices for technical personnel
e. electronics test area
f. hydraulic area
g. battery charging and maintenance area with possibly a cold storage area
h. machine shop
i. welding facility
j. holding fixtures, blocks, and dollies
k. paint area
l. scales
m. fork lift
n. storeroom for mechanical, electrical, and electronics spare parts and vehicle consumables
o. water tanks
p. source of running water for washing the vehicle and for cooling it during operational test
q. files and storage areas for technical manuals, test and repair procedures and maintenance/repair records
r. air conditioning and dehumidifying equipment
s. storage area for hydraulic fluids
t. source of compressed air

This list is not all-inclusive and depends on the degree of maintenance and repair that must be supported at the facility. There is a choice between setting up a given in-house capability and using outside support to cover a given area. The choice is made based on the trade-off between convenience, turn-around time, and economics. For less complex submersible systems, some of the above facility items are not necessary. A cable-towed vehicle, for example, which receives power from a support ship has no need of a battery charging and maintenance area.

The following is a list of equipment required in an electronics shop:

a. digital voltmeter
b. oscilloscope
c. signal generator
d. power supplies
e. frequency counter
f. pulse generator
g. resistor and capacitor decade boxes
h. time delay reflectometer (cable testing)
i. attenuators
j. electronic technician tools

In Fig. 52, a hydraulics laboratory equipped to test and maintain submersible hydraulic components is shown. The unit undergoing tests is a 3000 lb/in.^{2} hydraulics power unit. The facility has the ability to analyze oil samples to Class 1 cleanliness.

The DSRV is maintained at the Submarine Rescue Unit with facilities similar to those just listed. In contrast, the manned submersible *Turtle* also maintained at the same facility as shown in Fig. 53, has a slightly different maintenance operation. The *Turtle* has four portable vans:

- an electrical van containing a battery-charging system and various electrical tools
- a mechanical van containing a drill press, vise, welding equipment, and mechanical tools



Fig. 52 Submersible hydraulics laboratory. (Courtesy of Lockheed Missile and Space Co.)

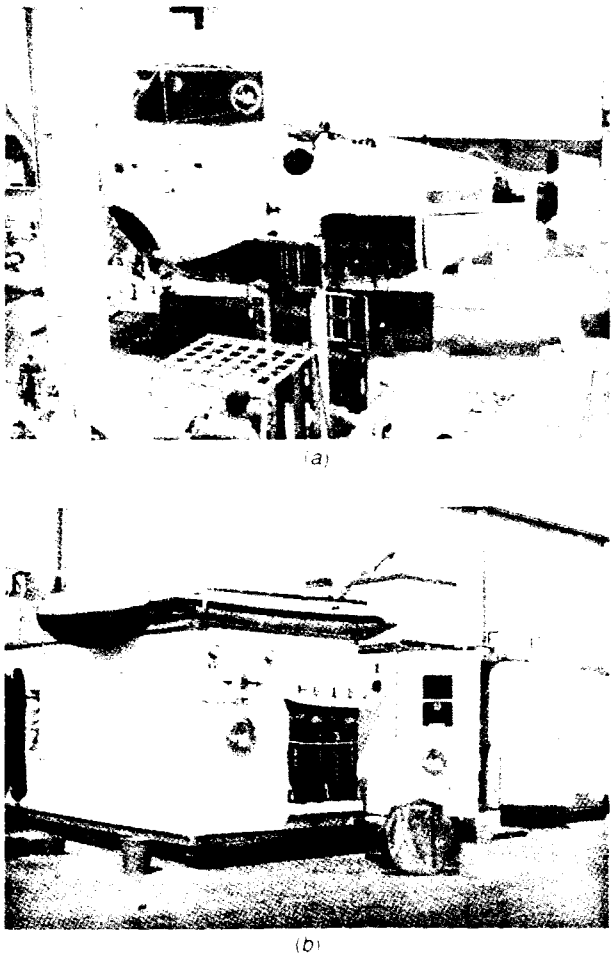


Fig. 53 (*a*) *Turtle* at the submarine rescue unit; (*b*) *Turtle* maintenance van. (Photographs by P. Lanner)

- an electronics van containing meters, digital multimeters, power supplies, frequency counters, and other electronics test equipment
- a supply van containing spare parts and equipment

These vans are shown in Fig. 53(*b*). The *Turtle* technicians maintain the vehicle from these vans. When the *Turtle* goes to sea, these portable vans are brought along on the support ship.

When the submersible *Alvin* was supported by the support ship *Lulu* a similar arrangement was used. Figure 54 shows the different support vans mounted on the deck of the *Lulu*.

Implicit in the requirements for onshore and at-sea maintenance facilities is the requirement for trained and experienced personnel. A typical maintenance facility will have mechanical and electrical technicians, together with engineering personnel. During sea operations, a small core of maintenance personnel can accompany the submersible. They bring along the necessary tools and equipment to perform maintenance down to the specified level defined for the operation. Normally a complement of spare electronics at the unit level and mechanical parts are brought. Replacement motors also may be brought. For manned submersibles, life-support consumables must be included. For battery-powered vehicles, a battery-charger system must be part of the at-sea complement of equipment.

The importance of a maintenance and repair facility to the overall success of a submersible operation should be pointed out. Submersible equipment malfunctions and failures at sea can cause expensive delays and can sometimes jeopardize the safety of the crew. A crew and maintenance team that knows thoroughly all aspects of safety, preventive maintenance, reliability, and the difficulties that result from the harsh environment in which equipment must operate can optimize the possibility of a successful mission.

4. Navigational and Positional Aids

In this section, a review will be presented of the major aids available for navigation and positioning.

In the design of an overall submersible system, consideration must be given to the navigational function. Attention must be placed on such items as overall mission efficiency, reliability, safety, and cost as they relate to navigation. Submersible systems have many applications. They are used in oceanographic research, offshore exploration and surveys, deep ocean mining, cable search, surveys and repair, personnel rescue, and in various military duties. These applications require that the submersible be transported to and from a work area and that its position while in the work area be known. In addition, recorded navigational data may be part of the end product of the submersible activity to be used for future analysis and correlation with other data. When the total system mission is considered, different navigational requirements may be encountered for different phases. For example, a submersible support ship travelling through the U.S. coastal confluence zone may have adequate navigational capability with a Loran C receiver, but on site, at some remote area outside the range of Loran C, a different system will be required in support of a submersible operation.

As with any support subsystem, the navigational requirements must be derived from the mission requirements. The navigational requirements of both the submersible and the support vehicle must be identified and the methods by which the requirements can be met by the total system analyzed. The navigational components that provide measurements of position, speed, and direction should be organized in a compatible manner to provide a reliable system that is capable of meeting the required performance. Where safety is a factor, multiple navigational aids may be necessary. High accuracy may

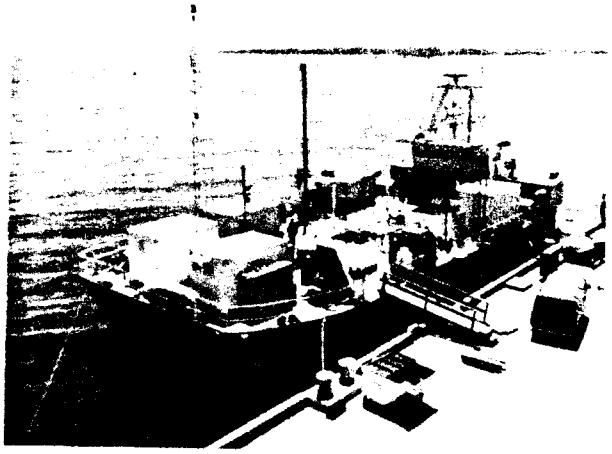


Fig. 54 *Alvin* support ship *Lulu* with support vans. (Courtesy of WHOI)

be achieved by processing multiple navigational inputs in some optimal manner.

Assuming a ship is beyond familiar landmarks and navigational markers, position may be determined at sea by two commonly used methods: by dead reckoning, a method by which the distance travelled and the direction of travel are accounted for; and by independent fixes whereby it is determined that the ship lies on two or more intersecting lines of position (position fixing). These two methods can be used to complement each other, with an intermittent position fix used to update a dead reckoning system. When position fixes are obtained continuously, then speed and heading can be estimated.

Dead reckoning (DR) in its simplest form can be implemented by using a ship's compass to determine a desired heading and by estimating speed by the setting of the propulsion system. Speed logs or Doppler sonar devices for measuring speed in conjunction with a gyro compass for sensing heading result in higher accuracy DR systems. Inertial navigation systems provide for a self-contained position, velocity, and heading measuring device which responds to inertial and gravitational forces and requires no optical or radio link with the outside environment.

Lines of position required in the position fix method are generated in a number of ways. If the direction to a known geographic object or point on the earth is measured relative to some reference frame, then the line formed between the object and the observation point is a line of position. Radio direction finders and radar and optical instruments can determine direction lines. When an observer measures the distance between his or her location and some known point, a circle of position is generated having a radius equal to the distance measured. Over the small region of concern, the arc of this circle can be considered a line of position. Distance can be measured by radar and by other radio devices. Another method by which a line of position is generated is by the hyperbolic method. Here the difference in range to two known locations is made. The family of functions generated by constant differences is hyperbolic. Again, over a small region, the curves can be considered lines of position. Hyperbolic lines of position are generated by a number of radio navigation and positioning systems.

The intersection of any two lines of position determines the observer's location; a third independent line provides confirmation of the measurement.

If the lines of position are made from a moving base at different times, one or more of them may have to be adjusted to account for the assumed motion during the time interval between measurements.

Navigational and positional aids in this section will be defined as radio, sonar, and inertial devices used for navigation at sea. However, the instruments of classical celestial navigation, the sextant, chronometer, and the magnetic compass will not be discussed since they are well-treated in the many works on that subject. The discussion of these devices will be directed toward the submersible designer and people involved with submersible system design and operations.

Therefore, the treatment has been limited to providing the underlying theory and the current practice on a number of principal aids in use today. The discussion will not include all units in use today, nor will it cover all configurations and combinations of their applications. To do so would be beyond the scope and limits of this chapter.

The intent of this section is to provide sufficient information so that potential navigation systems can be chosen during the planning and preliminary design phases of the submersible system design. As a minimum, the following will be presented for each navigation and position aid:

- a general description and theory of operation
- the area of coverage
- the accuracy over the coverage area
- the relative cost

Until the early 1990s when the Global Positioning System (GPS) becomes available, there is no global, high-accuracy, continuous navigation system available. The navigation system planner must economically and prudently choose between a number of navigational aids in order to meet the system requirements. The aids he can choose from can be broadly placed into three categories—radio aids, inertial devices, and sonar aids. Radio aids can be subdivided into hyperbolic radio navigation systems, radio ranging systems, and satellite navigation systems. Inertial devices are identified as gyroscopic devices and as inertial platforms which are made up of gyroscopic and specific force measuring instruments. The sonar navigational aids that will be discussed are the velocity sonar systems and the sonar beacon systems.

It may be found that the mission navigational requirements for a submersible support system can-

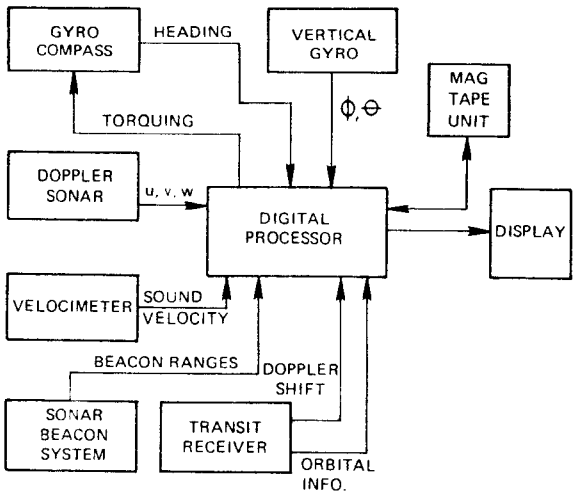


Fig. 55 Integrated navigation system

not be met by a single navigational subsystem. For a mid-ocean operation, if the ship's position and velocity must be determined to a high degree of accuracy, investigation of all available navigation subsystems may indicate that none will meet all the requirements. For example, the position accuracy may be met using a Transit satellite subsystem during the fix time, but between fixes no information will be available to determine position. On the other hand, using a Doppler sonar with a heading reference for dead reckoning may provide the required velocity accuracy, but the position accuracy will degrade beyond the required accuracy if the operation extends over any appreciable length of time.

Meeting the above requirements will necessitate the assembly of a number of navigation aids into an integrated system. Using a navigation filter that can combine the data from each aid in an optimal manner, the overall performance of the integrated system can be improved. Figure 55 shows one possible configuration for an integrated navigation system.

The feasibility of optimally integrated systems, in the statistical sense, has been made possible by two developments. One was the formulation of optimal estimation by R. E. Kalman [32]. Through utilization of his unification and simplification of optimal estimation, the formulation of algorithms capable of processing navigational sensor data in the time domain has become routine.

The other development that has led to the feasibility of optimally integrated systems has been the advances in digital processing equipment. Small onboard digital computers are now available which have the computational capacity for implementing Kalman filter navigation algorithms.

The design of an integrated navigation system using a number of navigation aids is not a straightforward process. The utilization of digital simulation using statistical and deterministic analyses can aid in arriving at a "best" configuration [33]. Each navigation aid in the simulation must be modeled. All the sources of navigation sensor measurement errors must be defined; these will include errors which are uncorrelated from measurement to measurement, and those which are correlated.

When different configurations have been simulated and analyzed, they then can be compared. The final choice of the best configuration for an application can then be made based on such parameters as performance, cost, complexity, size, and weight.

4.1 Hyperbolic Radio Navigation

The hyperbolic radio navigation systems make position determinations by the line-of-position method. The systems that will be discussed are Decca, Loran C, and Omega [34]. These systems are identical in principle, but they are all dissimilar in execution. They are called hyperbolic navigation systems because the lines-of-position they generate are of hyperbolic form. In Fig. 56 is illustrated two radio stations located at M (Master) and S (Slave), both emitting a radio pulse simultaneously. The two families of concentric circles, with M and S as centers, represent the location of these pulses at equal increments in time, assuming constant propagation velocity. To an observer located anywhere on the line AB, the master and slave pulses will arrive simultaneously. If the stations' positions are known, AB can then be

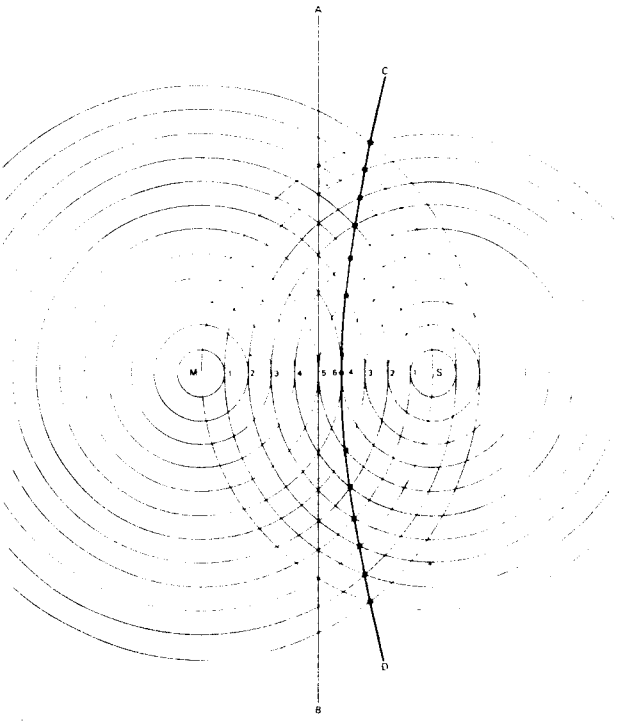


Fig. 56 Hyperbolic lines of position

used as a navigation line-of-position. Hyperbola CD, in similar manner, is formed from the locus of the intersection of the circles such that the circles generated by the slave station arrive two increments of time earlier than those from the master station. In other words, the line-of-position CD is such that the difference in the distance to M and S is a constant. An infinite family of lines can be generated in this manner, each having a specific but constant distance difference. Along the baseline MS, equal displacements along this line result in equal distance differences. Away from the baseline, the hyperbolas diverge, and the same displacement results in a smaller change in the distance difference. This expansion factor is found to be numerically equal to the cosecant of half the angle subtended at the receiver by M and S. During a typical operation, a user's distance from the stations seldom exceeds six times the length of the baseline.

The above discussion described the transmission of pulsed signals which, in fact, is the basic signal format for the Loran C system. However, both Decca and Omega utilize synchronized continuous wave signal transmission. For these two systems, time difference is measured by finding the phase difference between signals when compared at a common frequency.

To make a position fix, at least two lines-of-position are required. This means another pair of stations is required. In many cases, the master may be common to two slaves, forming the required two pairs.

The basic equipment for the hyperbolic navigation user is the receiver system. However, depending on the user requirements, the receiver and peripheral equipment used will vary in complexity. In its simplest form, an operator will manually tune a receiver to acquire and make the distance difference measurement. With a map, over-printed with a hyperbolic lattice, he or she can then make a position fix. On the other hand, many receiving systems provide fully automatic capabilities that can acquire and make all required adjustments. Processing equipment takes the hyperbolic inputs and derives position in X and Y or latitude and longitude coordinates, and drives pictorial or plotting displays. In compound, fully integrated navigation systems, the hyperbolic radio navigator may be only one source of inputs for a digital computer, which is also processing data from such elements as an inertial navigator, gyro compass, Doppler sonar, and sonar beacons.

4.1.1 Decca

The basic principles of Decca navigation were invented in the United States and developed in Great Britain during World War II. This system is heavily used for shipping in the northwestern part of Europe.

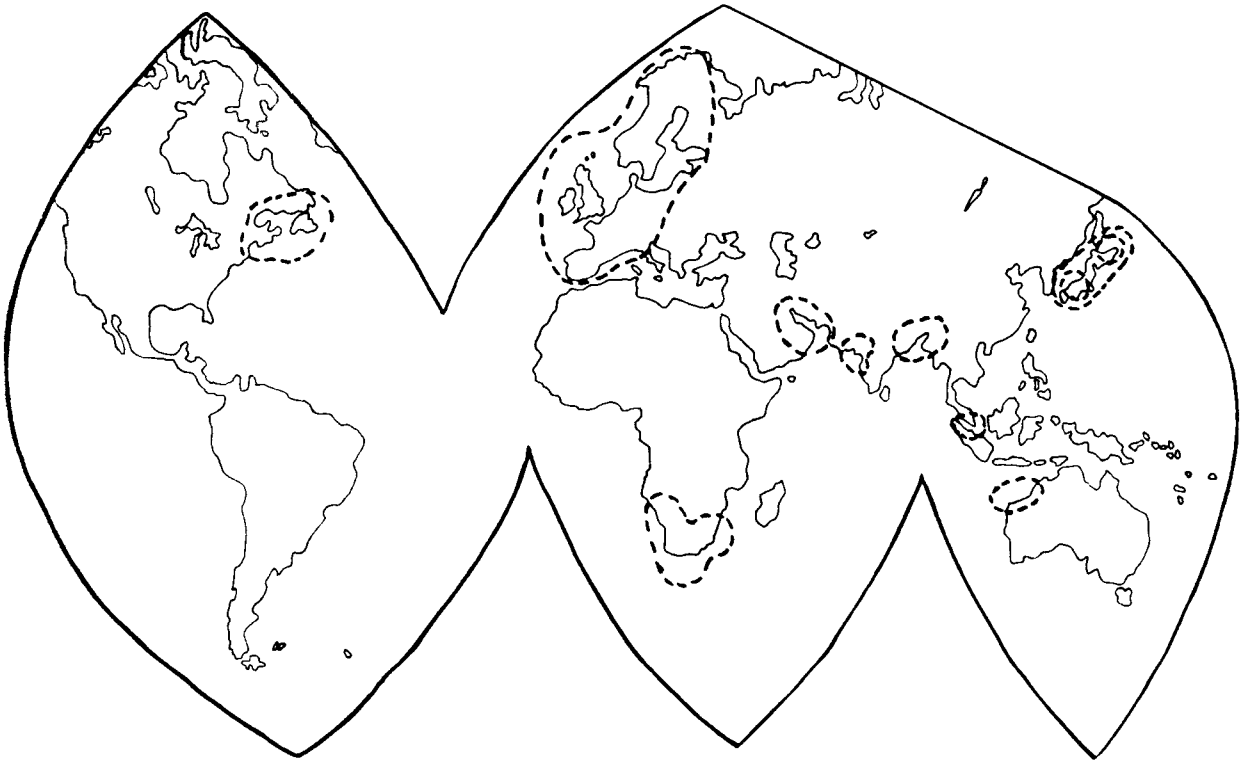


Fig. 57 Decca world-wide coverage. (Courtesy of ITT Decca Marine)

The present location of Decca chains throughout the world is shown in Fig. 57. The first commercial chain of Decca stations was established in Southeast England in 1946. By 1973 there were 43 chains in operation or under construction, providing accurate navigation in coastal confluence zones throughout many regions of the world. The locations of the Decca chains are

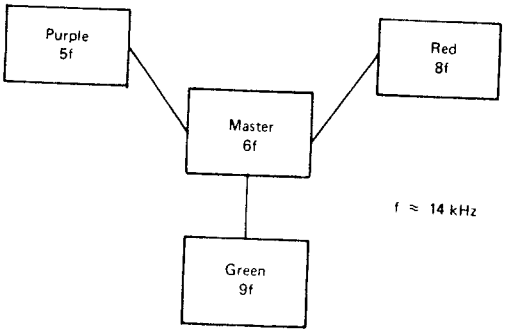
- 23 in Europe
- 2 in the Persian Gulf
- 4 in North America
- 2 in India
- 1 in Bangladesh
- 3 in Japan
- 5 in South Africa
- 2 in Australia
- 1 in Indonesia

As can be seen from this listing and from Fig. 57, the greatest concentration of stations is in Northwestern Europe. The coverage in Africa is marginal, and in South America it is nonexistent. The North America coverage is limited to the northeast Canadian Maritimes; there are no Decca navigation chains in the United States.

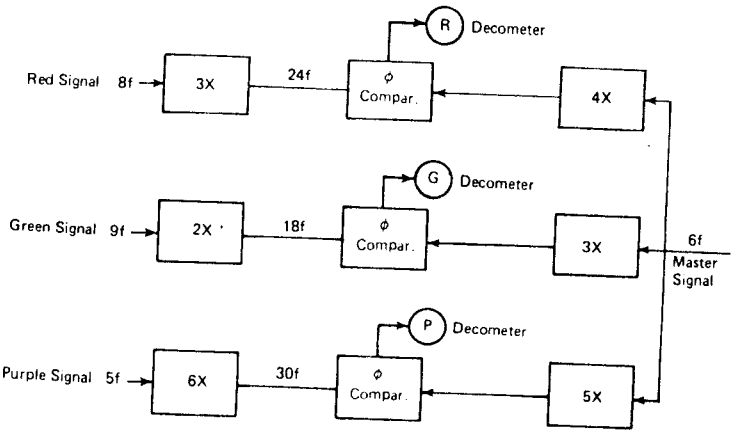
Decca uses unmodulated, continuous wave signals. It operates in the low-frequency band between 70 and 130 kHz, resulting in beyond-the-horizon coverage down to sea level [34]. The maximum range of operation from a master station is limited to regions where the reflected sky wave does not cause excessive interference with the normal ground wave signal. This range for both day and night operation is about 445 km (240 miles), which makes Decca essentially a medium-range coastal navigation aid.

A Decca chain is normally formed by three slave stations, typically spaced evenly about 113 km (70 miles) from the central master station. The omnidirectional signal from the master is received by the slaves and used by them to lock their own transmissions in phase with the master station. Each master/slave combination transmits phase-synchronized, unmodulated continuous radio waves, creating a pattern of hyperbolic position lines along which the phase difference between master and slave is a constant. Position fixes are made by identifying two of these position lines.

Since the user receiver cannot discriminate between different signals at the same frequency, the



(*a*)



(*b*)

Fig. 58 (*a*) Relation of master and slave transmitter frequencies; (*b*) multiplying process in Decca

master and slaves transmit at different frequencies. The master transmits on a frequency of 6f. The slave stations designated red, green, and purple (from the colors with which the hyperbolic grids—lines of position—are printed on Decca navigation charts) transmit at frequencies of 8f, 9f, and 5f, respectively. These relations are illustrated in Fig. 58.

The nonradiated fundamental, f, is given a specific frequency between 14 and 14.33 kHz for each chain. In the receivers, the master and each slave signal are processed to a common frequency for phase comparison. In marine receivers, the common frequencies are generated by frequency multiplication. One cycle of phase difference at this multiplied frequency represents one lane width. The comparison frequencies for the red, green, and purple are 24f, 18f, and 30f, respectively. The master signal is multiplied to these frequencies also, so that a phase comparison can be made for each master/slave pair. A simplified diagram of a receiver multiplying process is shown in Fig. 58(*b*). The processed phase difference signal drives a phase meter or decometer, which has a scale marked in fractions of a lane. At the multiplied common frequencies, the wavelengths are of the order of 700 to 1200 m. On the baseline, a change in position of one half of this wavelength will produce a change of 360 degrees in phase difference or one revolution in the decometer, corresponding to about 500 m. Position fixes are obtained by plotting two decometer readings on a Decca grid chart.

Since lane or phase difference measurements reoccur each cycle, an ambiguity arises in lane identification. This ambiguity is resolved by periodically, every 20 s, shifting the transmission frequencies of the chain to frequencies from which the receiver can generate comparison frequencies equal to f. One cycle of this much lower frequency represents a zone, and by measuring fractional zones, the correct lane within a known zone can be identified. Zones are far enough apart in marine navigation that zone ambiguities are not a problem.

Decca charts are printed by hydrographers and by the Hydrographic Department of the Admiralty. They are standard marine charts on which Decca grid lines of position are printed. The lines are identified by a zone letter and number representing a lane.

With a Decca receiver, one can determine one's position by plotting the decometer readings on a chart. The decometer readings provide zone, lane, and fractional (0.01) lane outputs. Two lines of position determined by two decometer readings give a position fix.

Decca receivers can be coupled to automatic navigation plotters. These continuous recorders provide a record of a ship's track which may be required in some missions. They also provide a convenient method for holding to predetermined tracks or returning to pre-plotted locations or surveyed areas.

Decca receivers can also be interfaced to navigation computers. The computer can take the Decca coordinates and convert them into rectangular, latitude/longitude, or North/South coordinates. These coordinates can then be displayed on a map or used as part of an integrated navigation system.

The Decca system has been proven to be a fine, simple, relatively inexpensive short-range navigation system with an average fix accuracy of about 0.4 km (0.2 nmi). With favorable geometry during daytime operation, the accuracy approaches a few tens of meters. During nighttime operation, sky wave interference can degrade the accuracy to, at worst, a few kilometers.

Decca receivers cannot be purchased, but can be leased from the Decca Company.

4.1.2 Loran C

Loran C was developed in the United States and was the outgrowth of development efforts to provide a high accuracy system with a long range capability especially over land. Loran C is a pulsed system, with the transmitted pulses having a carrier frequency of 100 kHz [34,35]. The pulses are approximately 270 μs in duration, and their shape is rigidly controlled. These Loran C stations transmit groups of pulses with each station transmitting eight pulses; master stations transmit an extra ninth pulse for identification. In addition to its use for identification, the ninth pulse is also used for blink. Blink is accomplished by turning the ninth pulse on and off according to a specific code; the code tells the user that a specific station has an error and the master/slave pair corresponding to this station must be used with caution. Figure 59 shows the pulse sequence for a Loran C chain. The pulses in a group are 1000 μs apart with a ninth master identification pulse space of 2000 μs.

A Loran C chain is formed with a master station and two to four slave stations with baselines of approximately 1000 to 1600 km (540 to 860 nmi). The master station forms the time and frequency reference for the chain. The slave stations are each assigned a coding delay. This delay is such that each slave transmission maintains the same position in the master/slaves sequence of transmissions for any receiver position in range of the net. In other words, if station X was the first to transmit after the master transmission as shown in Fig. 59, its pulse group

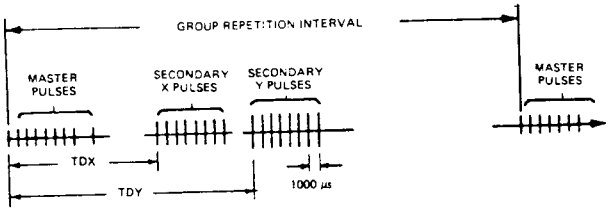


Fig. 59 Loran C pulse chain

would maintain this relative position for all receiver locations in the net coverage.

The Loran C receiver compares the groups of pulses from each master/slave pair to measure the time differences and subsequently to determine a hyperbolic line of position. The receiver processes the received signals by either envelope matching or phase matching. The former technique is used to measure the delay between master and slave signals, and provides course accuracies. To realize the full accuracy capabilities of Loran C, the receiver is designed to measure the time differences by phase matching using synchronous detection. Phase matching is used to lock onto a specific cycle in both the master pulse and the slave pulse, thereby allowing for a very accurate measurement of time difference, typically 0.1 μs. Furthermore, the multiple pulses are combined in such a way as to achieve a much larger signal-to-noise ratio than can be obtained from a single pulse.

The receiver is normally locked to the third cycle of the pulse. The reason for this is that the contaminating sky-reflected signal (sky wave) may arrive after the main ground-wave signal with a delay as small as 35 μs. By "indexing" on the third cycle, which is 30 μs after the start of the pulse, the sky wave is ignored.

Sky waves can also arrive delayed by as much as 1000 μs. When this happens, the sky wave will overlap the groundwave of the succeeding pulse in a group, again causing pulse shape changes. Discrimination against this type of interference is provided by phase coding the pulses. The pulses will then have 0- or 180-degree carrier phases. By comparison of the phase of the received pulses with the phase of reference signals in the receiver, the sky wave signals can be discriminated against.

The modern Loran C receiver provides high accuracy position fixes at long range. It will automatically acquire and track the signals from a master and two slave stations simultaneously. The processed signals are displayed as two time difference readings which can be directly plotted on standard Loran C charts. The receiver can also be interfaced with a plotter to provide a continuous record of ship's position. In integrated navigation systems, the receiver time difference measurements are transferred to a digital computer for processing.

The groundwave operating range of Loran C is of the order of 2000 km (1100 nmi). At the 100-kHz transmission frequency, a sky wave propagation mode exists for both daytime and nighttime operation. During the daytime, the sky wave range can extend out to 3000 km, while during the night it can go out to as far as 6000 km.

Typical accuracies for the system are shown in Table 17.

Loran C systems provide excellent repeatability, allowing the user to return to a known location with less than 90-m (300 ft) error, in many cases to less than 15 m (50 ft).

Table 17 Typical Loran C accuracies [34]

Range from Master, km (nmi)	Accuracy
650 (350)	122 m (400 ft)
927 (500)	213 m (700 ft)
1390 (750)	335 m (1100 ft)
1853 (1000)	518 m (1700 ft)
1853 (1000)	7 km (4 mi)[a]
2780 (1500)	19 km (10 mi)[a,b]
3700 (2000)	33 km (18 mi)[a,b]

[a]1-cycle index error.
[b]Skywave operation.

The world-wide coverage of Loran C is shown in Fig. 60. The chain locations and their codes are listed in the Loran C user handbook published by the United States Coast Guard [35].

Loran C has become the principal navigation system for the U.S. Coastal Confluence Zone (CCZ); it has replaced the Loran A nets. Fourteen stations will be added to cover the entire U.S. CCZ and a large portion of the northeastern hemisphere. In the mid 1990s the United States will reduce its operation of overseas stations. The host nations will then maintain the stations and increase the number of chains.

4.1.3 Omega

The Omega system is a continuous wave (CW) radio navigation system operating in the 10- to 14-kHz (VLF) range that provides world-wide coverage. Historically, this system, developed by the U.S. Navy, evolved from work done on the Radux radio navigation system. Omega was added to the Radux system to act as a fine-phase indicator to the much coarser phase indication derived from Radux. Eventually, the Radux element was dropped and the Omega system was further developed.

The parameters of the Omega system were specially chosen to economically provide global coverage. In particular, the geometry of the system calls for the unusually long baseline of 9000 to 11 000 km (5000 to 6000 nmi) with only eight transmitters distributed globally. The sites of these transmitters are (A) Norway, (B) Liberia, (C) Hawaii, (D) North Dakota, (E) La Reunion Island, Indian Ocean, (F) Argentina, (G) Australia, and (H) Japan. All the operating sites have achieved full designed radiated power levels of 10 kW.

The extremely low radio frequency operating band was chosen to realize the long operating ranges and also to have a stable and predictable propagational environment. This band is lower than that employed by any other hyperbolic radio navigation service. The very long ranges at which stable signals can be

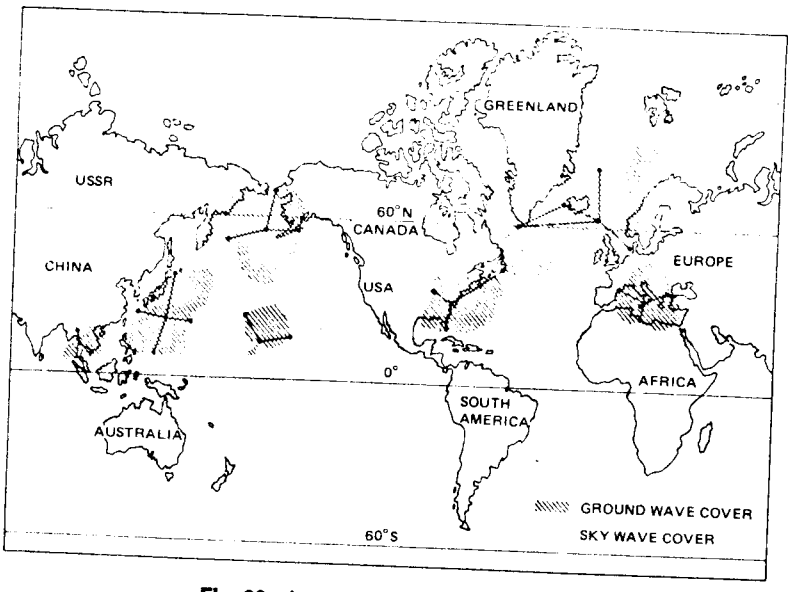


Fig. 60 Loran C world-wide coverage

received generally give the navigator the choice of considerably more than the minimum number of two lines of position required for a fix. Omega is the only global radio navigation system that is usable by completely submerged submarines; it is estimated that three frequency Omega navigation can be accomplished at antenna depths to 15 m (50 ft).

Each Omega transmitter broadcasts four frequencies, each for approximately 1 s. After 10 s the sequence is repeated. The frequencies transmitted are 10.2, 11.05, 11.33, and 13.6 kHz. Each station also transmits a unique frequency for identification. The transmission formats for all the stations is shown in Fig. 61 where the exact burst lengths are given.

The basic frequency for the Omega system is 10.2 kHz, corresponding to a wavelength of 29 km (16 nmi). A user determines a line of position (LOP) by measuring the phase difference of 10.2 kHz signals transmitted from two stations. If the observer proceeded from one transmitter to the other, he or she would find the phase pattern repeating at intervals of half a wavelength, or 15 km (8 nmi). These 15 km segments are identified as lanes. Measurement of phase difference in the receiver equipment is made in intervals of one-hundredth of a cycle, called centi-lanes. An LOP is defined in lanes and centi-lanes, and charts carrying this 10.2 kHz Omega lattice have been prepared by the U.S. Naval Oceanographic office. These charts normally cover an area of approximately 3700 by 1850 km (2000 by 1000 nmi).

In addition to the charts, correction tables must be used which account for the diurnal and seasonal variations in the propagation velocity of the signal. These sky wave corrections are predictable and result in position fixes that are accurate to within 1.9 km (1 nmi) by day and 3.7 km (2 nmi) by night. The correction tables are quite simple to use.

The only variables required are the user position to the nearest 10 degrees of latitude and longitude, pair of stations being observed, time to the nearest hour, and the date to within a half-month.

To provide for a coarser lane structure, Omega has three additional frequencies, 13.6, 11.33, and 11.05 kHz. This coarser structure is provided to resolve lane ambiguities. The 13.6, 11.3, and 11.05 kHz signals are processed in the receivers to effectively produce frequencies equal to 1/3, 1/9, and 1/36 the basic 10.2-kHz signal, respectively. This method of increasing spacing between ambiguities results in lane widths of 44, 133, and 534 km (24, 72, and 288 nmi).

Since the transmission from any one station lasts about 1 s on each frequency, a synchronously operating commutator in the user's receiver allows the carriers from the different transmitters to be separated into distinct channels. The commutator can be phased by reference to UT2, to which Omega is time related.

The Omega receivers are implemented with phase-locked loops and memory so that the phase difference of two signals, which are never present simultaneously, can be measured. Long time constant, averaging circuits greatly reduce the effect of noise so that Omega is operative even on signal strengths far below noise level.

For simple marine navigation, a single-frequency (10.2 kHz) receiver may fulfill most requirements. The navigator can establish the lane numbers at his or her point of departure and set automatic lane-crossing counters accordingly. During the voyage the navigator can plot his or her position from any two measurements of phase difference in lanes and centi-lanes.

Differential Omega is a method for obtaining increased accuracy in a small area such as an approach to a harbor. The method is implemented by

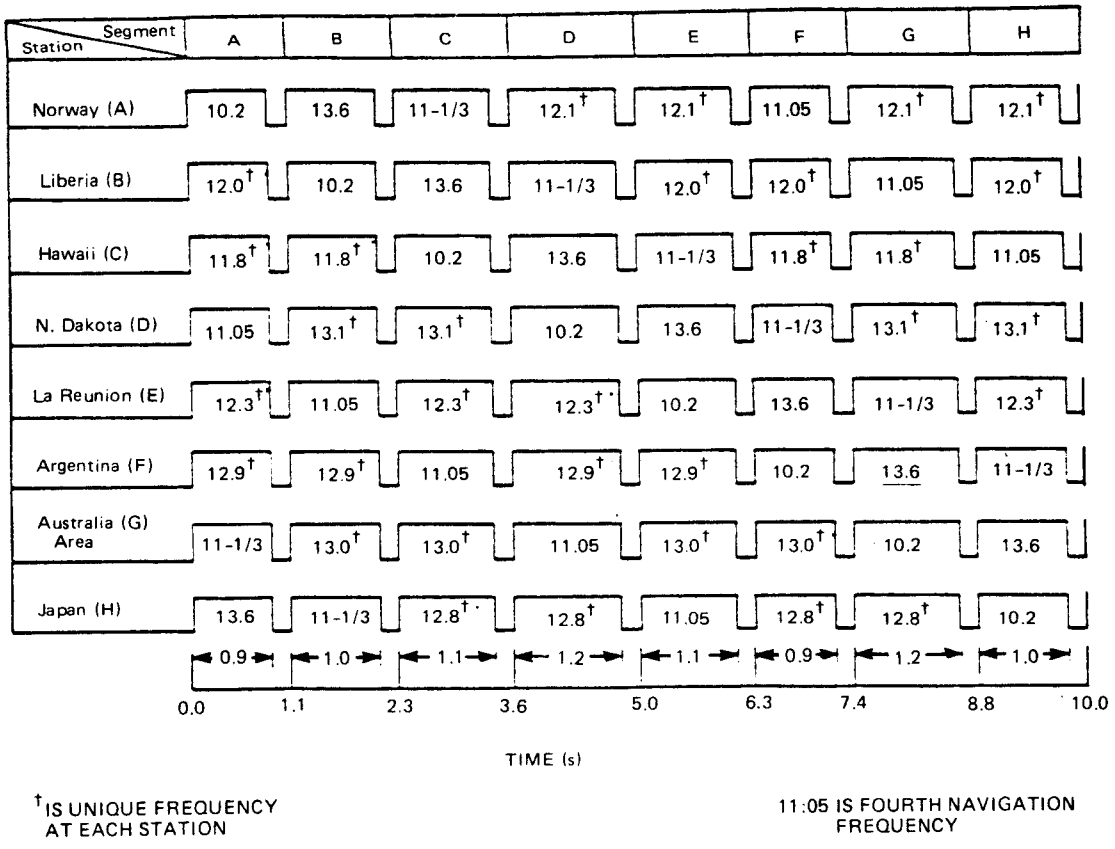


Fig. 61 Omega transmission format

placing a monitoring system at an accurately known location. The monitor determines the difference between its actual position and measured position. This difference is transmitted to Omega users in the area as up-to-date corrections. At distances of 40 km (22 nmi) from the monitor, the position error can be reduced to 0.4 km (0.2 nmi) or better.

The Omega system provides what the other landborne radio navigation systems cannot, namely world-wide coverage. It is an established and reliable system supported by the U.S. Coast Guard. It has a predicted accuracy of 3.7 to 7.4 km (2 to 4 nmi), but typically realizes accuracies in the 1.9 to 3.7 km (1 to 2 nmi) range. Omega with continuous position fixing capability overcomes the disadvantage of the other (early 1980s) world-wide coverage navigation system, Transit. This system, which is discussed in a following section, cannot provide continuous position fixes because of the nonavailability of satellites overhead at all times.

Omega receiver systems can be implemented with digital computers, which results in automatic operation and propagation velocity corrections. Through digital interfacing, Transit and Omega systems can be tied together to obtain the best features of each—the high accuracy of Transit and continuous operation of Omega.

Unpredictable changes in the ionosphere can cause errors in Omega position fixes [36]. Solar flares can cause these changes, one of which—sudden ionospheric disturbances (SID)—results from sudden increased x-rays generated by these flares and occurs during daylight hours. They last typically from 30 to 60 min and can cause line of position shifts on the order of 7 km along the baseline. Another major error producer is polar-cap absorption (PCA), which is caused by flare-emitted protons which are focused at the earth's magnetic poles. Signals which cross these regions can cause position errors of 30 km (16 nmi) or more and last for periods of usually several days. Event information on PCA is provided to users by the U.S. Coast Guard. U.S. Omega plotting charts and tables can be obtained from the Defense Mapping Agency Hydrographic Center, Washington, DC.

4.2 Electromagnetic Distance Measuring Systems

When submersible operation is to take place within a few hundred km of shore, electromagnetic distance measuring (EDM) systems can be considered for position determination where accuracies in the range of 5 to 20 m are required. These systems

are particularly attractive in offshore regions where no other navigation systems are available and where a system such as Omega does not provide the required accuracy.

A user survey of these positioning systems was performed by the National Oceanic and Atmospheric Administration [50]. In this survey, EDM systems were divided into two different types; one, short-range systems with operating ranges which vary from 15 to 250 km (8 to 135 nmi) and the other, medium-range systems, with ranges which vary from 370 to 740 km (200 to 400 nmi). These ranges are for daylight operation. The range for nighttime operation of the medium-range systems can go down by a factor of about two.

The short-range EDM systems generally operate in the microwave region, measuring range from a mobile transmitter to a pair of transponders which are accurately positioned on shore or on some fixed reference station. Some of the commercially available short-range systems are

- Artemis
- Autotape
- Miniran
- Mini-Ranger III
- Syledis
- Tellurometer
- Trident III
- Trisponder

Most of these short-range systems operate in a range-range mode for determining position, while the Artemis system makes range/bearing measurements. Most of these systems can provide positioning capability to multiple users except for Artemis and Autotape, which are single-user systems.

The short-range systems can be further divided into three classes—pulsed systems, continuous wave (CW) systems using phase measurements on modulation frequencies, and long coded pulsed CW systems. The phase-measuring systems in general have better performance than the pulsed systems, but are more expensive.

Performance of the short-range EDM systems which operate in a pulse mode can be affected by variations in the transit time caused by variations in the internal equipment delays in the transmitter, transponder, and receiver. Performance can also be affected by multipath effects where reflections from the ocean surface and from nearby reflective objects can combine with the direct signal, causing fading and dropouts of the ranging signal.

The short-range EDM systems can be expected to have performance accuracies in the 5-to-10-m (16 to 33 ft) range.

Some of the medium-range EDM systems are

- Argo
- Hi-Fix
- Hydrotrac
- Lorac
- Maxiran
- Omi System
- Raydist
- Toran

Most of these medium-range systems are configured with three or more shore-based transmitters operating at around 2 MHz and any number of mobile passive receivers when operating in the hyperbolic mode. A number of these systems can operate in the active range-range mode, but the number of users then becomes limited.

When used in the passive hyperbolic mode, all the medium-range EDM systems suffer from lane ambiguity. They can measure within lanes, but cannot tell which lane they are in. Therefore, the correct lane must be set in at initiation time and correct lane counts must be monitored. Loss of signal or equipment turn-off can cause a loss of the correct lane count.

Variations in the signal propagation velocity can affect the performance of the medium-range systems. At the 2-MHz transmission frequency, the propagation velocity is affected by the electrical conductivity of the surface over which the signal passes; over-land and over-water velocities can vary enough to cause system errors. The medium-range system is also susceptible to errors caused by sky wave interference, especially at long ranges, in addition to being sensitive to electrical storms, which can cause signal loss.

The medium-range EDM systems can be expected to have performance accuracies in the 10-to-20-m (33 to 66 ft) range.

In summary, EDM systems are commercially available positioning systems that can be set up onshore for high-accuracy positioning. The short-range systems are more portable and flexible than the medium-range systems, having units that can be mounted on tripods at onshore reference points. Medium range systems can require 31-meter (100 ft) towers for antenna locations to achieve the desired performance.

At a location far from land, it is also possible to use an EDM system. This can be done by mounting the transponders on buoys or some other object such as a ship or platform. With a buoy, the variations in buoy position due to changing cable scope must be accounted for. This could be done with an acoustic beacon system which could measure the buoy positions. The buoy position information would then have to be sent to the control station where the EDM system solves the range measurements for the ship's position.

4.3 Satellite Navigational and Positional Aids

The basic concept of the use of artificial satellites for communication and navigation purposes originated in the late 1950s. Since that time, both communications and navigational satellites have been

placed in orbit and have performed successfully. The U.S. Navy Navigation Satellite System (NNSS) known as Transit became operational in 1964 as a precise position, location, and ship navigation system. In 1967, the system was declassified and the shipboard receivers were released for commercial manufacture and civilian use. During the past several years considerable effort has been devoted to the development of high-accuracy, space-based radio navigation systems. Defense, civil, and international agencies have been considering the application of these systems to such areas as aviation navigation, missile guidance, position surveillance for air traffic control, precise time and time transfer, international maritime and aeronautical position location and reporting services, and collision avoidance.

A review of all these activities is beyond the scope of this chapter. What will be presented is a discussion of the present Transit system and the NAVSTAR Global Positioning System (GPS) now under development. The latter system is an outgrowth of both the USAF System 621B and the Navy TIMATION programs. Competition between branches for developing the Defense Navigation Satellite System (DNSS) resulted because of budgetary constraints in a joint service program, now called NAVSTAR GPS. This system has borrowed elements from both of the previous systems, namely, the signal structure of 621B and the satellite configuration of TIMATION. The reader can find information on additional systems in reference [37] which contains a survey of principal systems and concepts which have been under analysis or development.

4.3.1 Transit System

During the period immediately after the launching of Sputnik I, scientists at the Applied Physics Laboratory (APL) of Johns Hopkins University studied the feasibility of determining orbital parameters by measuring the Doppler shift of radio signals transmitted from a satellite. Surprisingly, it was found that the Doppler curve generated during a single pass of the satellite contains sufficient information, which when properly utilized, gave orbital elements of reasonable accuracy [38]. The next thing that was considered was the inverse problem, namely the determination of the location of the receiver from the Doppler shift measurement, assuming that the orbit of the satellite is known. This navigation case can result in a very high-accuracy measurement since only two parameters, latitude and longitude, must be determined, as opposed to the former satellite tracking case, where six orbital parameters had to be estimated. Furthermore, the navigator can make a position fix each time a satellite passes within range. Initial studies into this navigation method indicated that accuracies of better than 0.8 km (0.5 miles) were feasible. The personnel at APL were aware that the Polaris fleet would need an accurate position fix capability, and this concept was presented to the Navy. The proposal was approved, and a team of APL scientists was assembled to develop an NNSS, which became known as Transit. The chronology and history of subsystem evolution can be found in reference [39] for the interested reader.

The Transit system is an all-weather, world-wide, passive-user navigation system that provides accurate navigation capabilities, normally latitude and longitude, to better than 0.19 km (0.1 nmi). The system is composed of three subsystems:

- a constellation of satellites
- a ground support network
- the user's receivers and computers

The present active satellite constellation normally consists of six nonsynchronous satellites which are in near-circular polar orbits at an altitude of approximately 1100 km (600 nmi) as shown in Fig. 62. The satellites each have an orbital period of approximately 107 minutes. Depending on latitude, most world-wide areas can realize 15 to 20 usable passes daily from the net.

The satellites continuously broadcast their respective orbital information with timing signals at the ultrastable nominal frequencies of 400 MHz and 150 MHz, as harmonically related to 5 MHz. The timing signals from the satellites are synchronized with Universal Time Coordinated (UTC) to within 200 μs, and are transmitted every 2 min on every even minute of UTC. Both signals carry identical phase modulated satellite orbital messages, which, updated at 2-min intervals, continuously describe the orbital parameters of the satellite, including

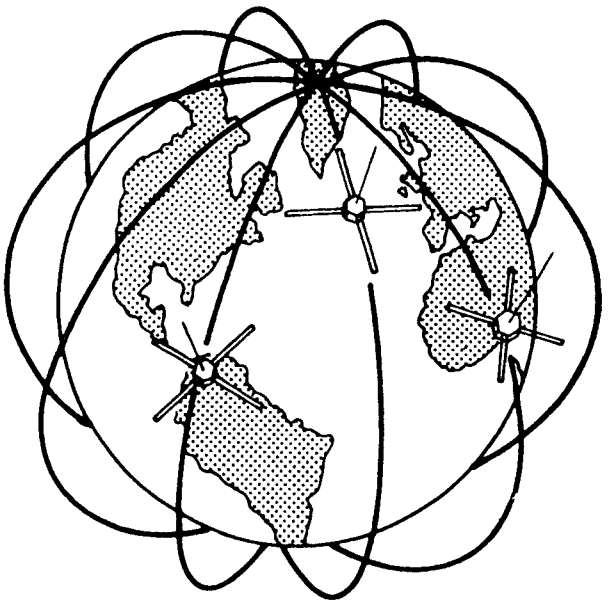


Fig. 62 Navy navigation satellites in circular polar orbits at altitudes of approximately 600 miles

ephemeral corrections for estimated orbit changes due to friction, gravity anomalies, and other predictable sources of orbit variations.

The ground-support network has the responsibility for tracking the satellites, computing their orbital information, and inserting this information into each satellite memory on the average of every 12 h. To accomplish this, there are four Doppler tracking stations which monitor the orbit of each satellite whenever in sight. The data from the tracking stations are sent to Point Mugu, California, where the Navy Astronautics Group, responsible for the ground network, is located. Here, the data are processed by computer and the orbits determined to an accuracy of 10 to 15 m rms and extrapolated into the future to provide Transit users with an orbital accuracy of about 40 m. Injection stations at Point Mugu and in Minnesota take the predicted orbital information and update each satellite twice a day.

To complete a position fix the user will obtain the following data during a satellite pass:

- measured Doppler frequency shift
- a two-frequency ionospheric refraction correction
- satellite orbit parameters
- accurate time marks

The earlier radio receiver systems used to measure the frequency shift were implemented to measure this frequency at discrete points in time. This method of measurement resulted in system complexity and has given way to the present method of integral Doppler measurement. Here, the Doppler difference frequency is counted over an interval of time, typically 2 min, with the count starting and ending with time marks from the satellite [40]. This method eliminates the need for a very accurate digital clock in the receiver, reduces the amount of data collected (Doppler measurements), and lends itself to a simpler way of calculating the navigation fix.

By using the integral Doppler measurement method, the system can be treated in theory as a hyperbolic navigation system [41]. In Fig. 63, the relative positions of a satellite at three different times and the user are illustrated. If the positions P1, P2, and P3 correspond to the location of the satellite at the instant the time marks were transmitted, then the separation between these positions forms the baseline of the hyperbolic system. For a 2-min interval, the separations are on the order of 960 km (500 nmi). If the user can measure the time difference corresponding to the range difference for positions P1 and P2 and also for positions P2 and P3, then two hyperboloids of position can be formed. Since the user can determine the locations of the satellites (P's) from the received orbital information, he or she can now determine his own location from the solution of the hyperbolic equations.

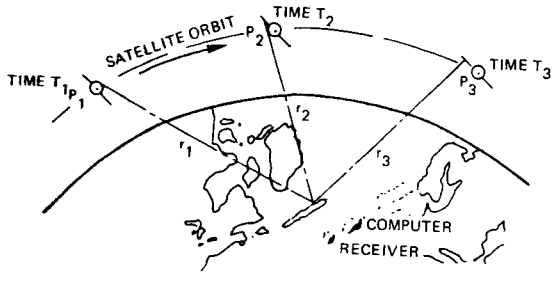


Fig. 63 Geometry of transit navigation

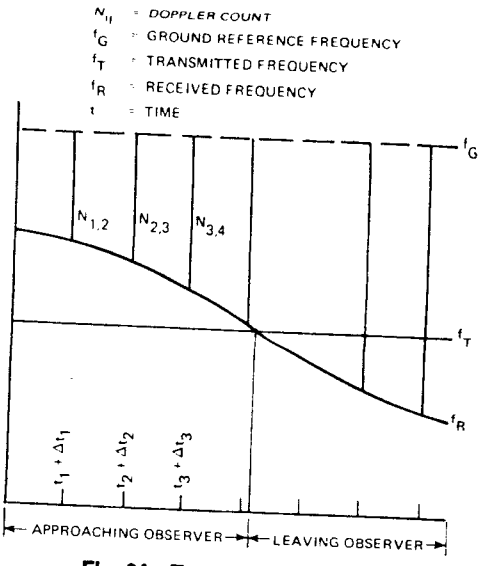


Fig. 64 Transmit Doppler signal

In the implementation of this theory, linearized equations are employed. Using an estimated observer position as a starting point, iterative computations are performed until the solution converges in a "least-squares" sense.

During the Doppler integration by the receiver, the receiver frequency f_R is compared to a stable oscillator output frequency f_G to produce a frequency difference

$$f = f_G - f_R \tag{1}$$

The relationship between the frequencies for one satellite pass is illustrated in Fig. 64. During a time interval, the beat cycles are counted or, essentially, integrated. This can be related to a slant range difference since each frequency count corresponds to one wavelength of distance travelled by the satellite. From Fig. 64 and using the first two intervals, the Doppler count is given by [40]

$$N_{1,2} = \int_{t_1 + \Delta t_1}^{t_2 + \Delta t_2} (f_G - f_R)\, dt \tag{2}$$

However,

$$\int_{t_1 + \Delta t_1}^{t_2 + \Delta t_2} f_R\, dt = \int_{t_1}^{t_2} f_T\, dt \tag{3}$$

Then equation (2) can be written as

$$N_{1,2} = (f_G - f_T)(t_2 - t_1) + f_G(\Delta t_2 - \Delta t_1) \quad (4)$$

Now the propagation delays Δt_i are equal to the slant range divided by the speed of light

$$\Delta t_i = \frac{r_i}{C} \quad (5)$$

The constants $(f_G - f_T)$ and $(t_2 - t_1)$ can have their constant product represented by $\Delta F \Delta T$. Note that the wavelength of the frequency f_G is given by

$$\lambda_G = \frac{c}{f_G} \quad (6)$$

Equation (4) can then be written as

$$N_{1,2} = \Delta F \Delta T + \frac{(r_2 - r_1)}{\lambda_G} \quad (7)$$

With the observer's position at $r(X,Y,Z)$, the slant range r_i is given by

$$r_i = \sqrt{(X_i - X)^2 + (Y_i - Y)^2 + (Z_i - Z)^2} \quad (8)$$

where the i's correspond to position P_i. Therefore, equation (7) can be written as

$$N_{jk} = \Delta F \Delta T + \frac{1}{\lambda_G}\sqrt{(X_k - X)^2 + (Y_k - Y)^2 + (Z_k - Z)^2} - \frac{1}{\lambda_G}\sqrt{(X_j - X)^2 + (Y_j - Y)^2 + (Z_j - Z)^2} \quad (9)$$

The solution of equations of this type with data (Doppler counts) taken during a satellite pass results in the position determination.

The major sources of errors in the Transit system are

- orbital determination accuracy
- signal propagation anomalies
- antenna height error
- observer velocity errors
- interference and sunspot activity

The orbital determination accuracy was seen to be greatly affected by the earth's atmosphere and gravity. Refinements to the gravity model have resulted in a substantial reduction in position determination inaccuracy. As previously mentioned, this error is now on the order of a few tens of meters.

Signal propagation anomalies are the result of the effects of the troposphere and ionosphere on the transmitted signal. Since tropospheric refraction, like Doppler frequency shift, is directly proportional to frequency, the tropospheric refraction error cannot be separated from the Doppler frequency shift. Studies that have been performed on the effects of refraction caused by the ionosphere have shown that this source of error can not be ignored for the Transit system to achieve its required navigational accuracy [38]. It was shown that for a transmit frequency equal to or greater than 100 MHz, the Doppler shift in the presence of the ionosphere is of the form

$$\Delta f = \frac{-f}{C}\dot{r}(t) + \frac{\alpha(t)}{f} + \frac{\beta(t)}{f^2} + \ldots \quad (10)$$

The first term is the Doppler shift if the atmosphere were absent and the following terms are the correction to the vacuum Doppler shift due to the presence of the ionosphere which corresponds to the time rate of change of the total number of electrons along the original path. By transmitting on two phase coherent frequencies, namely 400 MHz and 150 MHz, the predominant first-order term can be determined and eliminated from the Doppler measurement.

Antenna height error is another source of position estimation error. The vertical height of an antenna is measured from the reference geoid, not from sea level. A first-order approximation to the longitudinal error $\delta\lambda$ that results from this error δh is given by

$$\delta\lambda = \delta h \tan \Sigma_A \quad (11)$$

where Σ_A is the satellite elevation angle at closest approach. Therefore, the effect of height error is greater for high-elevation satellite passes, and results in useless data when the satellite passes directly overhead.

Figure 65 shows a typical fix error due to antenna height error for different longitudinal separations between the observer and the satellite [42]. The effect on latitude due to antenna height error is a second-order effect because of the predominantly north-south geometry of the orbital plane.

If the system is used for navigation on a moving vehicle and the velocity of the vehicle is not compensated for in the calibration of position, or the velocity measurement used for compensation is in error, then a position error can result. Typically, a 1-knot error in the ship's velocity measurement can add 0.37 km (0.2 nmi) of uncertainty to the position of a ship on a north-south course. Figure 65 illustrates how this type of error can vary depending on the longitudinal separation of the satellite from the observer. Velocity measurements for correction can be derived from an inertial navigation system, Loran C, Omega, Doppler sonar, or EM logs (if any of this equipment is available), or from dead-reckoning estimates.

Based on the overall mission requirements, there are many user configurations that might be utilized. Transit can be used to provide navigation for a support ship, with position fixes available about every hour and one half. It can be used for geodetic surveying [43] or where a very accurate, three-dimensional fix is required. In this application, the satellite data obtained over many passes can be stored on tape for later evaluation at a computer center. For fixed-position applications, the fix accuracy based on a few days of satellite data collection can be better than 10 m (30 ft). With two or more stations and observations made over several days, relative position and interstation spatial distances can be measured with an accuracy of better than 3 m (9 ft). Where

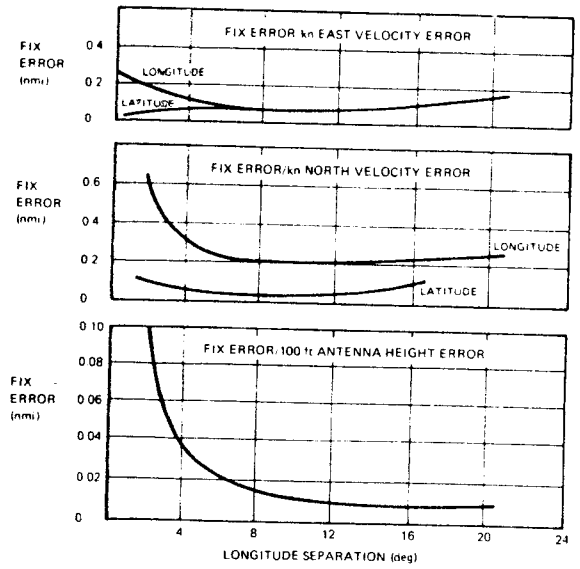


Fig. 65 Fix error versus velocity error and versus antenna height error

the application requires instantaneous computation of the position fixes, the receiver is interfaced with processing equipment.

Where high-accuracy and continuous navigation information is required, the transit receiver equipment is commonly integrated with ship's dead-reckoning sensors and other navigation equipment by interfacing with a digital processor. The configuration of any integrated navigation system results from the overall system requirements. One form can use a ship's heading reference and speed log or Doppler sonar. Another form can use the Transit system as an updating device to an inertial navigator. For positioning and site surveying, the Transit system can be integrated with a beacon navigation net which has sonar beacons placed on the ocean bottom. The Transit system can then accurately locate the beacon net geographically.

In addition to various system configurations, the user must consider the various receiver and processor options. For example, where a degraded accuracy of 0.3 km (0.2 miles) is acceptable, a single-channel Transit receiver is an economical choice.

One point must be brought up about Transit. It is a system run by the military for U.S. Navy applications, which can create problems for civilian users [44]. Orbit gaps and availability of satellites which may be allowable to the Navy could be unacceptable to some users. Occasionally, the Navy conducts a test in which a satellite can give a position fix significantly worse than normal. There are now no communications channels that civilians can use to discuss problems of the system with the Navy. As the GPS system becomes operational, Transit will most likely not be supported into the late 1990s.

4.3.2 Navigation Satellite Timing and Ranging Global Positioning System (NAVSTAR GPS)

The GPS system is a satellite based navigation system that will provide users with extremely high accuracy, three-dimensional position fixes, velocity measurements and timing information anywhere on or near the earth. It will provide continuous, all-weather availability, with good coverage geometry. Being developed for the military, it will have anti-jamming and anti-spoof capability. However, it will also be available to the nonmilitary user at a reduced accuracy. Ultimately, because of its universal coverage and accuracy, it is planned that this system will provide savings in the number and costs of navigation and position-fixing systems currently employed or projected. Error estimates for the GPS system indicate that it will provide position fixes to the order of 10 m (30 ft) and velocity measurement of 0.1 m/s (0.3 ft/s) or less.

For the GPS system to make a position fix, transit times for the propagation of signals between the satellite and the user must be measured. In addition, satellite orbital parameters and internal clock timing information must be given to the user, through signal modulation. The user, with algorithms in his or her digital processing equipment, converts the transit times and system data into position fixes and clock corrections. Ground monitoring stations determine the satellite positions, clock states, and other necessary system information and relay these parameters to the satellite in order to update their transmitted signal information.

Figure 66 illustrates the GPS configuration and identifies the three major segments, the space system, user, and the control system.

The space system will be formed in a manner to provide continuous global navigation service to any point on the surface of the earth [45,46]. To achieve this, 21 primary satellites plus three operational spares will be placed in circular orbits at an altitude of 20 242 km (10 924 nmi). (This altitude corresponds approximately to a 12-h period.) Six of these satellites will be equally spaced in each of three orbital planes. Each orbital plane will be spaced 120 degrees from the neighboring planes. The satellites will weigh about 800 lb and will transmit their navigation signal from an antenna sized for earth coverage. The carrier frequency of the signal chosen for the system is in the L-band and is a compromise between errors due to ionosphere delays and cost and complexity of both the satellite and user equipment. Each satellite will in fact transmit two simultaneous signals, one at approximately 1.6 gHz, the primary L1 signal, and the secondary L2 signal at 1.2 gHz. The delay between these two signals will be used to determine the ionospheric delay. Superimposed on both signals will be satellite ephemerides, constellation data, atmospheric propagation correction data, and satellite

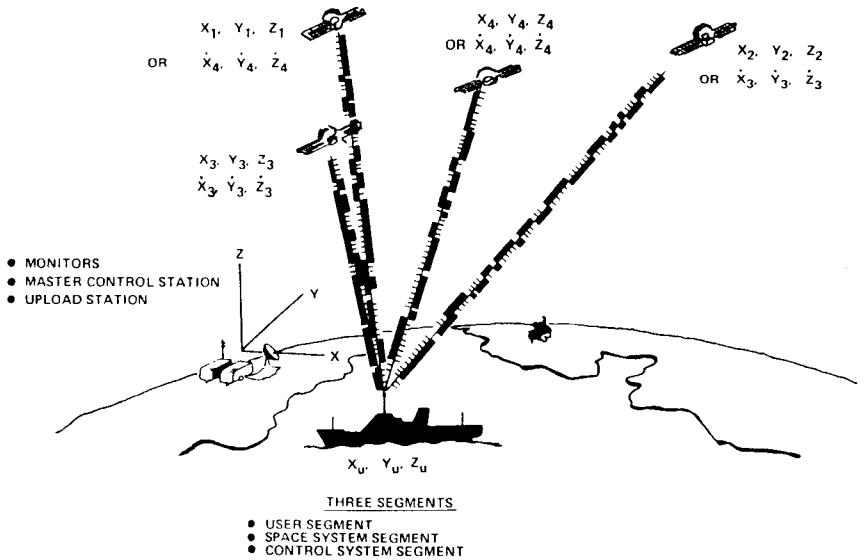


Fig. 66 Global positioning system configuration

clock drift information, all of which are periodically updated by a master ground station.

The operation of the GPS system requires that the satellites carry clocks of extreme accuracy. The ground station clock updates will maintain synchronization between satellite clocks on the order of a few nanoseconds. The clock update frequency contemplated for the system will require clocks with an accuracy of a few parts in 10^{13}.

The control system is composed of widely separated monitor stations which passively track all satellites in view, and a master control station which processes the ranging data accumulated by the monitors. An upload, also part of the control system and located in the Continental United States, at least once a day transmits the best (master control station) estimates of ephemeris, clock drift, and propagation delay data to each satellite.

The user equipment includes, in addition to equipment required to receive and demodulate the satellite signals, the computational equipment which determines the user's position and velocity. If the user equipment contains precision clocks synchronized to the satellite nets, then range measurements can be made to each satellite in view. To avoid this costly requirement, enough satellites are deployed so that at least four will be available at all times. Four pseudo-range measurements can then be made containing sufficient data to determine not only position but user clock bias. The measurement is called pseudo-range because it contains, besides range, a bias caused by the user's clock error. The pseudo-range equation can be expressed as

$$TC = \sqrt{(X - X_S)^2 + (Y - Y_S)^2 + (Z - Z_S)^2} + T_B C \quad (12)$$

where

X,Y,Z = the user's position coordinates
X_S,Y_S,Z_S = satellite position coordinates
T = difference between satellite time of signal transmission and the user time of signal reception
T_B = user clock bias
c = speed of light

T measurements from four satellites, which each provide position coordinates and time information, is sufficient to solve for the four unknowns, namely user position coordinates and user clock bias. In an analogous manner, pseudo-range rate measurements allow for the determination of user velocity.

The primary L1 carrier from a satellite will be biphase modulated by the navigation signals in phase quadrature. One signal, for military use, is called the protection P code, and the other, which is used by the nonmilitary and also as an aid in acquiring the P signal, is called the clear acquisition (C/A) code. The satellite position, clock, and atmospheric correction data are superimposed on both the P and C/A signals.

The P signal consists of a sequence of binary digits at approximately 10 MHz generated by a linear shift register which is reset every seven days. This seven-day segment is part of a linear code with a period of 267 days. Because of this long period, the signal effectively appears as a random sequence and is termed a pseudorandom noise (PRN) sequence. Each satellite will have its own sequence, making the system resistant to cross talk between satellites, multipath interference, jamming, and spoofing. The military user will have decoders to determine this sequence, which will be changed periodically, thus making the signal virtually secure.

The C/A signal consists of a sequence of binary digits at approximately 1 MHz. Each satellite has its own 1023-bit code word (Gold Code). This signal, with a lower inherent range accuracy, is accessible to nonmilitary users.

Both the P signal and the C/A signals are modulated with the system data at 50 bits per second. The secondary L2 carrier transmission will carry only the P code. The C/A code is normally acquired first by the receiver. The C/A code has, as part of its data stream, a handover word which has the information necessary for the receiver to transfer over to the P code.

Depending on the user requirements, the receiver equipment will have varying complexity. The most complex will operate on the L1 and L2 frequencies, utilizing both the P and C/A signals and will have the capability to track four or more satellites simultaneously. A nonmilitary, degraded-accuracy system will operate on the L1 frequency using the C/A signal.

When fully operational, the GPS system can provide positional accuracies better than 10 m (30 ft) in any direction 90 percent of the time.

The program is now being developed in three main phases. Phase 1, which was a concept validation phase, has been completed successfully. Satellites were launched into two orbits and user equipment was developed and tested. In 1982, the program was in Phase 2, in which prototype satellites and equipment were built and tested [47]. In the early 1990s, Phase 3 will culminate in a fully operational GPS system when the space segment is extended to the full set of satellites. During this phase, operational equipment testing will continue and will include all operational modes.

GPS will become very useful to both military and civilian users because of its accuracy and availability [44]. It is expected to have position accuracies from about 100 m (330 ft) for the least expensive commercial equipment to about 10 m (33 ft) in three dimensions for more complex military navigation sets; velocities along the three axes will be measured to an accuracy of 0.15 m/s (0.3 knots). The navigation coverage provided by GPS will be continuous, world-wide, and all-weather.

It must be kept in mind that the GPS system is being developed primarily for military use. However, a Federal Radio Navigation Plan (FRP) has been developed which describes how DOD and the Department of Transportation (DOT) will plan and implement radio navigation systems. Present plans only guarantee highest accuracy GPS functions to the military, with single frequency C/A signals for civilian use at reduced accuracy. The P-code dual frequency capability will probably not be made available to civilian users.

4.4 Sonar Navigational and Positional Aids

Sonar devices have been in ocean use since about the turn of the century. Their development has been accelerated by both world wars, especially World War II. During that period, systems had to be developed to detect and track submarines effectively. The forms of radiation that were utilized on the surface for military systems, namely light and radio waves, could not be used because of the high propagation losses in the ocean medium. Successful underwater systems had to resort to sound waves which can propagate over considerably longer ranges than electromagnetic waves underwater.

The sonar systems used today for anti-submarine warfare (ASW) are as complex in technology as the radar and electronic systems used for aerospace operations.

Nonmilitary developments have produced sonars used for depth sounding, fish finding, sub-bottom profiling, and side-scan mapping.

During the latter part of the 1960s, the development of a number of sonar devices for navigation and positioning applications took place. This development was encouraged by the many manned submersibles that were being built during this period. These vehicles had unique navigational requirements not present in other ocean vehicles. In addition, the oil industry needed better sensors for positioning during exploration and for navigation of the large tankers that were transporting crude oil. The development of these sensors was motivated by the needs of this industry for higher accuracy and for effectiveness from an economics standpoint.

The sonar systems used for positioning and navigation are active sonars. In active sonars, as opposed to passive sonars, one component of the system generates the underwater sound used by the system [34,48,49,51]. This transducer, which normally converts an electrical signal into a sound signal, is called a projector. With an active sonar there is another transducer called a hydrophone, which converts an underwater sound into an electrical signal. In some systems, the projector and hydrophone functions are performed by one transducer. Within an active sonar system are the electronic components that generate the wave shapes and timing signals necessary to drive the projector and amplify the hydrophone received sound wave for further electronic processing.

In the design and use of a sonar system, due recognition of the limitations of underwater sound propagation must be made. Electromagnetic waves propagate through space at 3×10^8 m/s, while sound waves propagate through water at the relatively low speed of 1500 m/s. The range of penetration of sound in water, another major parameter in sonar system design, is determined by two different effects. One is the spreading loss caused by the distribution of the sound energy over an ever increasing area as it propagates; the other is the attenuation loss caused by the absorption of energy by the medium and by the scattering of energy by discontinuities in the medium [49]. Spreading loss depends directly on the

distance of the sound wave from the source, while the attenuation loss is independent of this distance. The attenuation loss is a function of path length, however, and is strongly dependent on the operating frequency of the sound wave, roughly being a function of the square of frequency. This effect has to be considered in the choice of operating frequency in a sonar system. The resultant effect of the two types of transmission losses places a limit on the range over which a sonar system can operate.

The velocity of sound in the ocean is not constant, but is affected by depth, temperature, and salinity. This variation in the propagation velocity can have a marked effect on the performance of a sonar system. The refraction of sound waves caused by velocity variation results in the sound wave path being bent instead of straight-line.

All information transmission systems are ultimately noise limited. With sonar systems, this noise is the general sonic disturbance that exists in the water medium. This interference basically comes from a number of sources. First, there is the noise produced by external sources such as animals, ships and one's own ship, waves, and wind, in addition to the thermal background noise that is always present. Wave noise can be related to sea state [48,49]. There is noise from flow over a hydrophone due to ship motion. There is also the noise caused by the interaction of a sonar system with the ocean medium, called reverberation noise. This interference can exist in any active sonar and it is caused by the discontinuities in the ocean medium which scatter acoustic waves that are projected by the sonar system. Some of this scattered energy is received by the hydrophone.

In this section, two types of sonar systems will be discussed. The first is the beacon position measuring sonar system. The other is the velocity measuring sonars which can measure a ship's velocity.

4.4.1 *Sonar Transponder Navigation*

High-accuracy local navigation can be performed with an acoustical transponder net anchored on the ocean bottom. In the early 1960s, the ARIS sonar transponder system was developed to measure the position of Polaris submarines during missile firing tests. Since that time, this system has been improved and used for navigation and position keeping of both surface vessels and submersibles [52–54].

Oil and gas exploration have moved further from land, increasing the accuracy requirements for local navigation systems. Highly-repeatable accurate beacon navigation systems meet this requirement. In addition, they find applications on surveys, production platform siting, pipe laying, geological studies, submersible navigation, and surface ship and undersea devices positioning.

The basic system is made up of acoustical transponders fixed on the ocean bottom and an interrogator/receiver located in the vehicle (Fig. 67). The interrogator/receiver interrogates the transponder net, which in turn transmits reply signals. The reply signals are processed in the receiver to determine the total round-trip times from the start of the interrogation pulse to the received reply from each beacon. These round-trip times are equivalent to the slant ranges from the vehicle to each beacon. Slant ranges can be determined by multiplying one half the round-trip times by the average sound velocity.

In order for different beacons to be distinguished, each beacon normally replies on its own fixed frequency. These frequencies are usually in a band between 5 and 20 kHz, which provides a good compromise between range and transducer power and size. Adjacent transponder frequency spacing can be as close as 500 Hz. Interrogation frequency coding systems are also used where the beacons are instrumented to be interrogated by different signal frequencies.

The normal format of both the interrogation and reply signals is a pulsed sinusoid with the pulse duration of the order of 4 to 15 ms.

Given the coordinates of the beacons and slant range to these beacons, the navigation task is to determine vehicle position in a coordinate system related to the beacon net.

In order for a beacon net to be implemented, the transponders must first be planted on the ocean bottom. The transponders can be planted by releasing them from the surface and letting them free-fall to the bottom. Some transponder beacons are designed to be released from a submarine. Transponder systems are composed of three major components: a flotation device, an anchor/cable system, and the transponder electronics battery unit. The flotation device of a transponder may be a separate float, the housing for the electronics, or flotation material around the housing. The transponder floats on a cable above the anchor. Some floats are designed to have low drag in a current. This keeps the cable scope

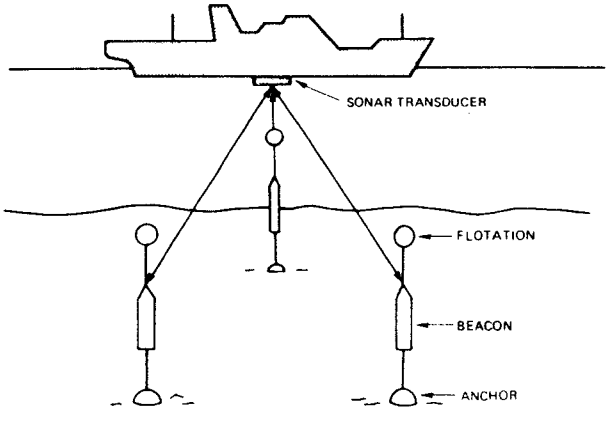


Fig. 67 Acoustic beacon navigation configuration

low; cable scope contributes to the uncertainty in the coordinates of a beacon.

The anchor/cable system is a weighted device that holds the system to the ocean floor. Deep Ocean Transponders (DOTs) that can be used to depths of 20 000 ft normally have the transponder floated on a line 300 ft above the anchor in order to reduce the effect of bottom reflections. On some transponders, the anchor/cable and float separate some time after the unit has settled on the ocean bottom. On other types, the anchor/cable unreels as the unit descends to the ocean floor. Descent rates are about 5 ft/s.

The transponder's electronics are designed to receive interrogation pulses, recognize these pulses, and transmit back the transponder's own acoustic pulse after recognition. The turn-around time for this operation ideally should be constant for a set of transducers in a net. Variations in the turn-around time contribute to slant range measurement errors.

Some beacons are designed to have commanded release mechanisms. When the electronics receives a pre-assigned coded signal, this release mechanism is activated and the transponder housing is freed from the anchor system and floats to the surface.

The batteries in a transponder beacon provide power during the life of the beacon. Typical battery lives can be from months to years depending on the particular design of the transponder and its frequency of interrogation.

In the implementation of a beacon net, the beacons must be placed in a given geometry. Beacon navigation can be done with two beacons, but normally three beacons are used. These three beacons are placed on the bottom to form an approximate equilateral triangle. For increased area coverage, additional triangular arrays can be placed on the bottom. The spacing between beacons is typically about 7 km (4 miles), depending on the transponder range capability and the application. Optimum transponder spacing which will provide the maximum usable net area based on fixed range uncertainty and a minimum positional accuracy has been determined in reference [55].

A beacon survey is the next thing that must be done to establish a beacon net. When the beacons were originally placed, their positions were noted. However, due to uncertainties in position at the time of release and because of the effects of underwater currents, these original positions will be in error. Errors in the positions of the beacons in fact contribute the most significant errors to beacon navigation. A beacon survey, which considerably reduces these errors, is done by using an interrogation/receiver system on the surface ship: the beacons are interrogated and the slant ranges are recorded.

In the past, the most common type of survey ran directly over each transponder to determine its depth and ran across the baseline centers (midpoint between two transducers) to determine the baseline separations [53,56]. Reference [57] shows the importance of the geometry used for a survey. The accuracy of the survey can be severely affected by the geometry of the survey points. Survey points should not lie on a straight line, nor should they lie on a conic section, that is, a circle, where the transponder positions are not uniquely determined.

One method for determining beacon coordinates, which eliminates the large amount of ship time and precise maneuvering necessary, uses a digital computer which is part of the beacon navigation equipment [52,58]. A sound velocity profile of the area is input into the computer in addition to the approximate coordinates of the beacons. The ship steams through the area while the beacon navigation electronics interrogate the beacon net. About 15 to 25 random slant ranges are recorded. The coordinates of the beacons are then estimated by the computer using least square methods. After the survey, the net can be used for navigation.

By the addition of a fourth transponder to a underwater object, such as a submersible or towed body, this object's position can be tracked.

Many factors affect the performance of a beacon navigation system [56]. Such environmental factors as bottom type (reflective or absorbent), roughness of terrain (which can cause shadows), and the magnitude and directional fluctuations of current which affect transponder cable scope all influence performance. Transponder characteristics such as beam pattern, height of transducer above the bottom, frequency and pulse width of the signals for both interrogation and response, source level, receiver design for rejection of both noise and multipath signal, and float design each contribute to the overall performance.

The greatest contributions to errors in the navigation system are the errors in coordinates of the beacons.

Another source of error occurs in the measurement of slant range. The slant range errors come from three sources. One error source is bias and random errors in the timing device or in the delays incurred in transponder response or receiver recognition. The next error source results from an incorrect estimate of the average speed of sound. The third source of slant range error is caused by the bending of sound rays due to refraction.

Still another source of error is failing to consider the curvature of the earth. Yet another error arises if the ship that is using a beacon navigation system is underway: then the ship's motion must be compensated for in the position calculations or an error will result.

Multipath reflections of the sound waves can also cause errors. Reflections can take place off the surface, the ocean bottom, or even parts of the transponder system. Ambient noise can affect performance. It can cause false alarm triggering of the transponders and receiver. It can mask the signal and cause delays in signal recognition. The effects on navigational

accuracy by all these error sources are analyzed in reference [56].

Beacon navigation systems can provide essentially continuous navigation information at any location in the ocean where they are deployed. This means that navigation is available in areas far from land, and where other forms of navigation are not available or are not practical. In reference [59], with three transponders placed to form an equilateral triangle of 3 to 4 km on each side covering an area of about 20 km^2 and with depths between 2.5 and 3 km, the ship positional accuracy was 10 m. In reference [52], a beacon navigation system achieved an accuracy of better than 4.7 m for a ship underway in 107 m (350 ft) of water. Commercial equipment is now available which allows transponders to operate to depths of 6000 m (20 000 ft), have ranges up to 18.5 km (10 nmi), and have an accuracy of 1 to 2 m; up to 32 active transponders can be handled. The life of the transponders for this type of system is typically 30 months and 600 000 replies, and can be extended to 5 years.

By using other navigational aids such as Transit, a beacon navigation net can be referenced to geodetic position. Since the beacon navigation system is readily transportable, it can be set up anywhere a submersible operation is located.

An acoustic beacon navigation system implemented with a digital computer and a video display can be part of an integrated navigation system. It can accept inputs from other navigational aids such as Omega, Transit, Loran C, GPS, a Doppler sonar, and the ship's heading reference.

4.4.2 *Velocity Sonars*

Doppler velocity sonar—It has been observed in wave propagation that if there is relative motion in the direction of propagation by the source of the wave or if there is relative motion by the receiver of the wave in the same direction, then the frequency of the wave is shifted proportional to the velocity of those motions. This is called the Doppler effect and is observed in sonar systems when this relative motion exists [48]. This effect has been utilized in Doppler sonar systems to measure the velocity of a ship [60].

The Doppler effect may be analyzed as follows. Let V_s be the component of velocity of the source of the signal between the source and a fixed reflector. V_s is positive when it is in the same direction as the signal propagation direction. The signal is transmitted at frequency f_p. The wavelength of one cycle of this signal is given by

$$\lambda_p = \frac{(C - V_s)}{f_p} \tag{13}$$

where C is the propagation velocity. Assuming that the reflector sends back a portion of the signal to the source along the same path, the received signal will have a frequency given by

$$f_r = \frac{(C + V_s)}{\lambda_p} = f_p \frac{(C + V_s)}{(C - V_s)} \tag{14}$$

The Doppler frequency is the difference between the transmitted and received frequency and is given by

$$f_D = f_r - f_p = \frac{2f_pV_s}{C - V_s} \approx \frac{2f_pV_s}{C} \tag{15}$$

With C = 2900 knots, V_s in knots and f_p given in kilohertz, equation (15) becomes

$$f_D = 0.69 f_p V_s \tag{16}$$

Doppler sonar systems use the ocean bottom as the reflector or target. The ship using the sonar has its velocity, a vector, basically in the horizontal plane. To make a Doppler measurement, a sonar beam is directed downward at an angle γ, normally about 45 degrees (Fig. 68). The component of velocity must be resolved along this beam axis and equation (15) becomes

$$f_D = \frac{2f_pV\cos\gamma}{C} \tag{17}$$

To remove sensitivity due to vertical motion and attitude, a second beam is added, pointing down at the same angle γ but directed aft. By taking the difference between the forward and aft beams, the Doppler measurement is given as

$$f_D = \frac{4f_pV\cos\gamma}{C} \tag{18}$$

A sonar Doppler system was developed for the Massachusetts Institute of Technology Instrumentation Laboratory by the Marquardt Corporation. This work was performed for the U.S. Navy as part of the DSRV submersible program. The system has four beams and measures the vehicle velocity in body (X,Y,Z) coordinates by taking the sums and differences of the measured signals. It uses phase-tracking electronics to make the frequency measurements and can achieve high accuracy. It has an inherent accuracy of about 0.2 percent and has achieved

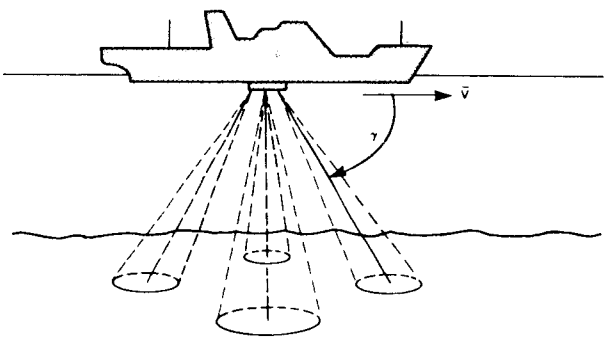


Fig. 68 Doppler velocity sonar four-beam construction

overall accuracies better than 1 percent. This system was the forerunner of high-accuracy systems that were developed for surface ships.

Doppler sonars normally operate at a frequency range between 100 to 500 kHz. They operate at these high frequencies to keep the transducer sizes small, while maintaining a narrow beamwidth—typically a few degrees wide. It is desirable to keep the beamwidth small from the power and spectrum standpoint because wider beamwidths cause increased signal spreading with a resultant reduction in accuracy.

One disadvantage of a high-frequency signal is that the absorption of the sound wave by sea water limits its range. Most Doppler sonars operate in a pulse mode, in contrast to CW signals. This allows for volume reverberation problems to be eliminated. In this mode, the sonars typically can operate to an altitude of about 150 m (500 ft) above the ocean bottom. Earlier sonars that operated in the CW mode could achieve only about 30 to 60 m (100 to 200 ft) of altitude.

When a Doppler sonar is beyond the range of the ocean bottom, it measures velocity relative to the water mass using the volume reverberation returns. For accurate navigation, a correction has to be made for currents if they are known or can be estimated.

As pointed out previously, multibeam (3 or 4) Dopplers measure velocities in vehicle fixed coordinates. Typically, one pair of beams measures longitudinal velocity while another pair measures athwartship velocity. The velocity outputs are in digital form, with pulses that correspond to a distance moved in a given direction. If a ship's gyro compass or heading reference is used, the pulses can be resolved and integrated to give North/South or along-track/cross-track distance traveled.

Where high-accuracy measurements of velocity are required, the velocity of propagation C in equation 18 must be known accurately. This velocity can be estimated fairly accurately by measuring the temperature of the sea water. Temperature is the predominant factor in the velocity variations of sound in water. If greater accuracy is required, a velocimeter, a device that measures propagation velocity, can be used.

Another factor that can cause an error in the measurement of velocity is the attitude of the ship. When a Doppler sonar is measuring the longitudinal and athwartship components of the ship's velocity vector, it is usually assumed that the plane formed by these two components is parallel to the local horizontal plane. However, if the ship is rolling and pitching, the measured velocity components are not in the horizontal plane. In a high-accuracy integrated system with a digital computer, the outputs from a two-axis (roll/pitch) vertical gyro or an inclinometer can be used to resolve the velocity components into horizontal components.

Doppler sonars have been used to aid in the docking and berthing of large ships [34]. In some cases, an additional pair of athwartship sensors has been added to one end of the ship in addition to the navigational sensors on the other end. In this manner, the pilot of a large ship is given independent measurements of the athwartship motion of the bow and the stern. Since the tolerance on impact velocity of large ships is so small, typically on the order of 6 m/min, the ability to measure these low velocities is invaluable.

Typically, Doppler sonars are used with gyro compasses for dead-reckoning navigation. They have been integrated with other navigational systems such as Transit through digital processing equipment. Used in this manner they can provide velocity inputs to reduce velocity errors during Transit position fixes besides providing dead reckoning between fixes. The velocity measurements can also be used for gyro compass torquing to reduce the velocity sensitive errors in that device. In applications in which the Doppler sonar system is integrated with an inertial navigator, the velocity measurement can be used to damp the inertial platform.

In summary, a Doppler sonar sensor can provide accurate velocity information to about 0.1 knots and provides for measurements of distance travelled with an accuracy of 0.5 percent or better. It works off bottom returns at distances from the bottom of about 150 m (500 ft). At larger distances, it measures velocity relative to the water mass.

Correlation velocity sonar—The General Electric Company has been involved in the development of a sonar velocity sensor [61]. This work started in the mid 1970s and has been sponsored by the U.S. Navy. A commercial version of this sonar has been marketed.

The geometry of the correlation sonar is shown in Fig. 69. A single projector transmits a wide beam pulse straight down that is reflected off the ocean bottom. The echoed sound waves are received by an array of hydrophones that are accurately spaced. The hydrophone signals are cross-correlated and analyzed to estimate the distance the ship traveled during the time between each transmit pulse which is precisely known. So that the reader can better

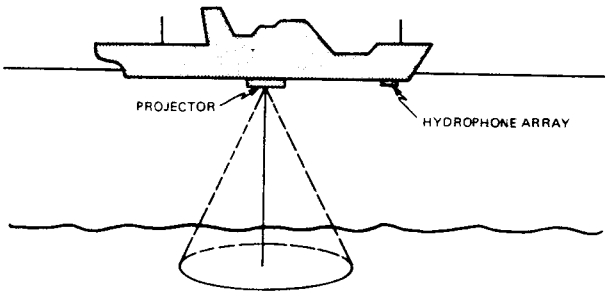


Fig. 69 Correlation velocity sonar

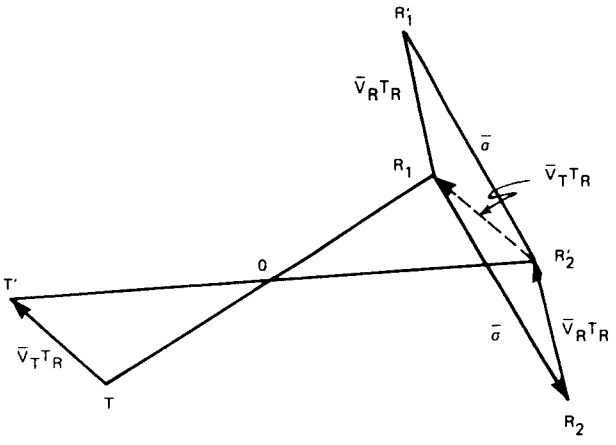


Fig. 70 Correlation sonar geometry [64]

understand the principle involved, Fig. 70 and the following argument are presented [61].

From the figure, we have a transmitter at T and a pair of receivers at R_1 and R_2 having a vector separation $\boldsymbol{\sigma}$. A signal is transmitted from T. A time T_R later, the transmitter moves a distance $\mathbf{V}_R T_R$ to T' and transmits the identical waveform again. Sometime later, the bottom scattered signals are received. R_1 and R_2 represent the receiver positions when the first transmission is received. R_1' and R_2' represent their positions when the second transmission is received. Their vector displacements can be designated by a receiver platform velocity $\mathbf{V}_R$ multiplied by the interpulse delay time T_R. If the midpoints between T and R_1 and between T' and R_2 coincide, the two waveforms will be identical. From Fig. 70, the relationship

$$\boldsymbol{\sigma} + (\mathbf{V}_T + \mathbf{V}_R)T_R = 0 \qquad (19)$$

is satisfied. This relationship depends only on the vector separation of the receivers. It is independent of propagation velocity, C, and the beam widths of the projector and hydrophones.

To illustrate equation 19, if a sonar is pulsing at a 100-ms period and if we have two hydrophones spaced 33.6 in. apart, one aft of the other, assuming V_T and V_R are equal, we find that identical waveforms are received by these hydrophones if the ship's speed is 8.3 knots.

In a practical sonar system, one does not have a pair of hydrophones whose vector separation can be adjusted to achieve maximum cross-correlation. Instead we have a plane of sensors, typically six in number arranged in a T or C configuration. These configurations provide fore/aft and port/starboard spacings and are sensitive to the component of the ship's velocity that is in the plane of the array. However, measured correlation is dependent on precise time relationships and because of this dependence, is sensitive to vertical velocity. Therefore, if the space and time dependence of the correlation functions are used, all three components of a ship's velocity can be measured. However, since the array samples at only spatial discrete points, interpolation techniques are used to determine the values of separation where maximum correlation takes place.

The velocity and distance measurement accuracies of the correlation velocity sonar are comparable to those measured by a Doppler sonar. However, the correlation sonar uses a wide beamwidth for the transmitted signal, typically 10 to 40 degrees. Because of this, the transmitter can work at a lower frequency than a Doppler sonar, typically 10 to 20 kHz. This allows for the design of correlation sonars that can operate on ocean bottom returns for any water depth. As previously mentioned, the velocity component that is measured in the array plane is independent of sound velocity. The high-accuracy Doppler sonar must correct for sound velocity changes. Effects of ocean bottom characteristics on the performance of a correlation sonar are analyzed in reference [62]. The correlation sonar can also be designed to measure velocity relative to the water mass, if this is required, by changing the pulse rate.

4.5 Inertial Navigation

Inertial navigation is a method of navigation that utilizes the inertial properties of sensors to determine the position and velocity of the vehicle on which the sensors are mounted. This navigation process has been described as a mechanization of Newton's laws of mechanics and is accomplished through the appropriate processing of data obtained from inertial force and angular velocity measurements.

Inertial navigation systems provide continuous indications of vehicle position and velocity in a wholly self-contained manner without reliance upon either natural or man-made external radiation for their measurements. An inertial navigation system is nonradiating and cannot be jammed. It can operate at any location on earth and in all types of weather. It can perform while the vehicle is maneuvering and provide measurements of vertical and vehicle attitude.

During the last quarter century, great improvements in the precision of gyroscopes and accelerometers have made possible the development of high accuracy inertial navigation systems [63]. Random drift in gyroscopic units forms the limiting factor in inertial navigation technology. A particular class of gyro unit, the single-degree-of-freedom floated integrating gyro series, was developed at the MIT Instrumentation Laboratory. Under continuous refinement since 1948 at the Instrumentation Laboratory and elsewhere, they have provided the accuracy improvements necessary for inertial navigation systems from the 1950s through the 1970s. Today, in addition to the single-degree-of-freedom devices, there are many other types of instruments—electrostatically suspended gyros, laser gyros, and two-axis flexure con-

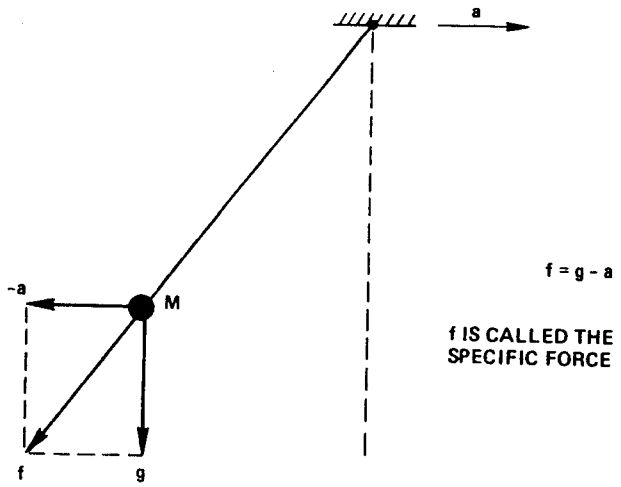


Fig. 71 Simple pendulum

figurations to name a few. The history of inertial guidance and developments can be found in references [63] through [66].

Inertial navigation systems can be divided into three categories—geometric, semianalytic, and analytic [67]. Almost all systems today are of the semianalytic type; the geometric system has become obsolete due to the advances in computer technology. However, this same technology has now made the analytic, or strapdown, system feasible for navigation applications, as have advances in gyro technology.

In an inertial navigation system, gyroscopes are used to measure vehicle attitude motions. Because these measurements are made with respect to inertial space, gyroscopes lend themselves to the establishment of a reference coordinate frame in which the motion of the vehicle can be measured. Acceleration information is obtained from accelerometer measurements. Accelerometers do not measure acceleration, but rather the difference between gravitation and acceleration, the nonfield specific force **f**.

$$\mathbf{f} = \mathbf{G} - \mathbf{a} \tag{20}$$

If accelerometer data is to be useful, then the gravity field must be considered in the system. The steps that must be taken by an inertial navigation system are [67]

- Establish a reference coordinate frame through gyroscopic measurements.
- Measure specific force in a frame related to the reference frame.
- Have a knowledge of the gravitational field **G**.
- Initialize system with original position and velocity.
- Compute position and velocity by integrating the inertial acceleration **a**.

Some basic concepts on which inertial navigation is based will now be reviewed [68,73]. In Fig. 71, we have a simple pendulum formed by a mass, M, suspended by a weightless string from a platform which is accelerating with a value, a, perpendicular to the local gravity field, g. The offset of the pendulum from the local vertical is seen to correspond to the specific force f. Some accelerometers are built on the principles of a pendulum; one of these units is shown in Fig. 72. For this instrument, any specific force directed along its input axis (IA) will cause rotation about its output axis (OA). A signal generator is located at one end of the OA and produces an electrical signal proportional to deflection. This signal is processed and sent back to a torque generator on the other end of the OA. The torque generator will return the pendulum to null, with the torque generator signal being a direct measurement of specific force f.

Let us see how the accelerometer measurement can be used. In Fig. 73 we have a very simple model, with an accelerometer mounted on a vehicle. The vehicle is constrained by tracks to move only in the x direction. The IA of the instrument is in the direction of motion; this direction is perpendicular to g and therefore the instrument does not measure it. Since the specific force measured is now equal to the

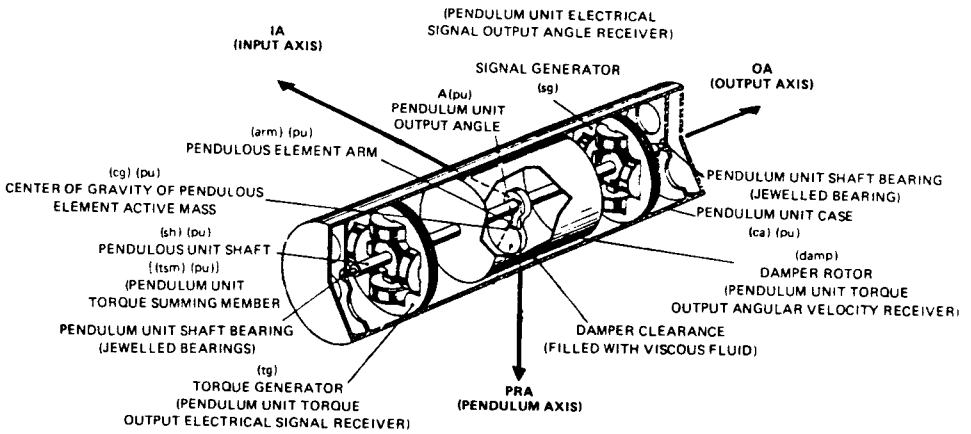


Fig. 72 Damped, single-degree-of-freedom pendulum unit

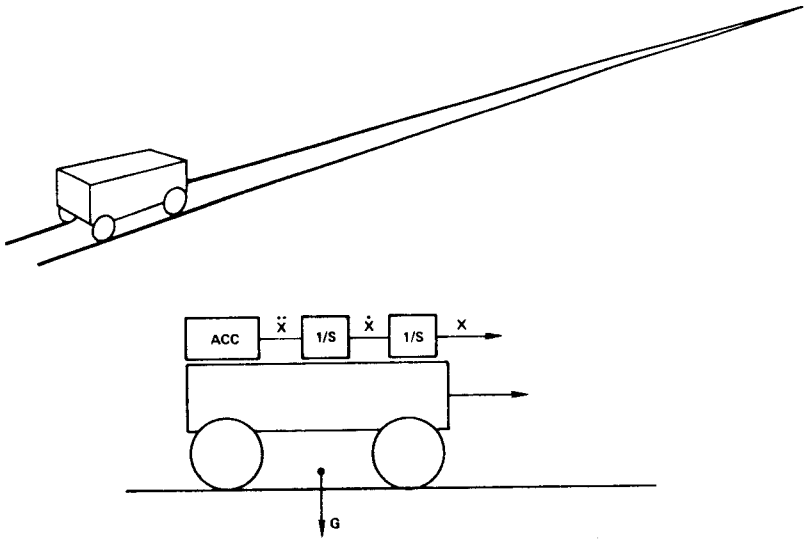


Fig. 73 Single-axis plane navigator

acceleration of the vehicle in the x direction, the accelerometer output is equal to $\ddot{x}$. By interpreting this signal, the velocity of the vehicle, $\dot{x}$ is obtained. In the figure, $1/S$ is the Laplace equivalent of integration. By performing another integration of the velocity signal, displacement in the x direction is obtained. Knowing the initial conditions of position and velocity, the position and velocity in the single dimension x coordinate system is measured continuously as a function of time.

A slightly more complicated system is shown in Fig. 74. Accelerometers are mounted in a horizontal plane with their IAs in the x and y directions, respectively. Again, the accelerometers sense no components of g along their IAs. Velocity and position measurements are made in the x and y coordinates of the vehicle. If the x axis of the vehicle maintains the same heading over the path or if the x and y measurement coordinate frame orientation relative to a reference navigation frame is known, then velocity and position in this frame can be correctly computed. However, if the measurements from the accelerometers are used directly as in the figure, and since the x,y coordinate system fixed in the vehicle in general can move freely, then the integrated outputs will be meaningless. Another problem exists. Suppose the IA of the x axis accelerometer in Fig. 73 or Fig. 74 is misaligned about the y axis so that it senses a component of g. If this misalignment is of the order of one arc minute, then about 0.01 ft/s^2 of apparent acceleration due to g is sensed. If this error is doubly integrated, a position error which grows

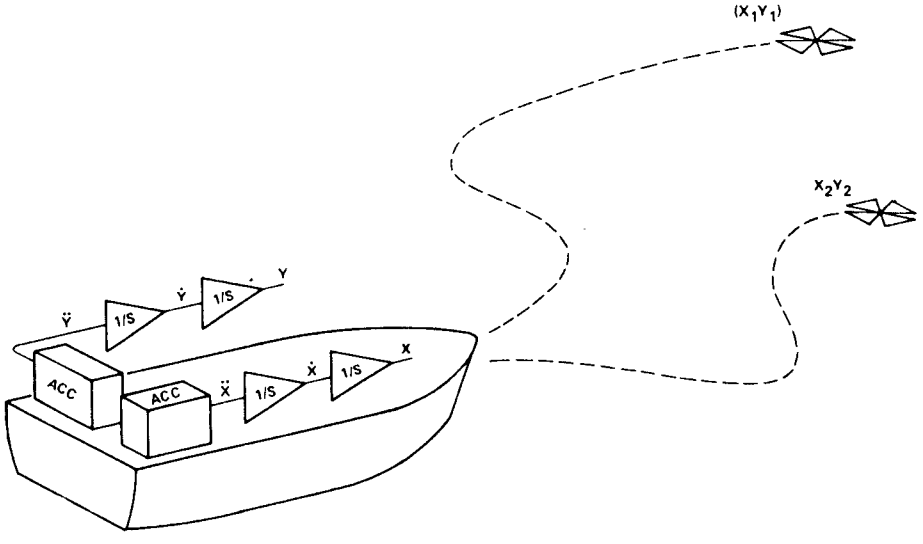


Fig. 74 Two-dimensional navigator

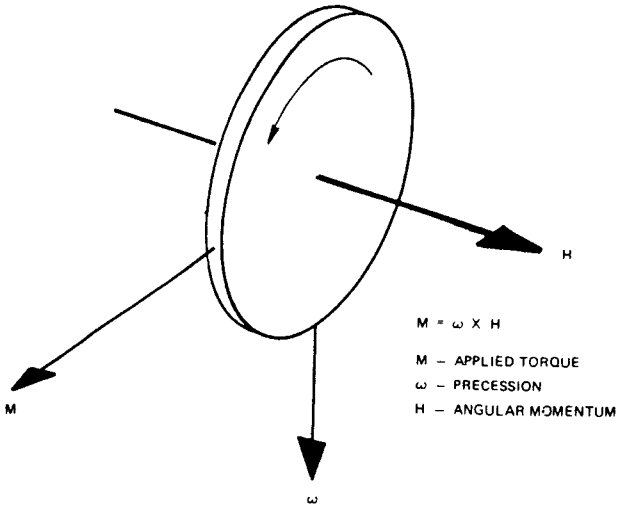


Fig. 75 Gyroscopic relationship

with the square of time will result, being about 40 nmi after 2 h, irrespective of the motion of the vehicle, even if it is stopped. From these effects it is apparent that accelerometer orientation must be known and controlled. This is normally accomplished by mounting the accelerometer on a stable platform which is controlled in orientation by gyroscopic instruments. The alternate approach is to mount the accelerometer and gyroscopic instruments directly on the vehicle frame and to implement analytically the reference frame in the navigation computer.

The stable platform approach will now be discussed. To implement a stable platform, gyroscopes are mounted on a base motion-isolation system composed of gimbals, gimbal torquers, gimbal angle resolvers, and gimbal servo amplifiers. Three gimbals are sufficient for most marine navigation requirements. Three single-degree-of-freedom (or two two-degrees-of-freedom) gyros are mounted on the stable platform with their input axes mutually orthogonal in order to stabilize the platform in three degrees of freedom. Angular rotations of the platform with respect to inertial space are detected and activate appropriate gimbal torquers so as to maintain the platform in the desired orientation regardless of vehicle motion.

A gyroscope can be considered a wheel rotating at high speed about its axis of symmetry, as shown in Fig. 75. If the wheel is suspended in a balanced frictionless gimbal system and a torque **M** is applied normal to the angular momentum vector **H**, then the gyro will rotate with angular velocity $\boldsymbol{\omega}$ transverse to both **M** and **H**. This is given by the relation

$$\mathbf{M} = \boldsymbol{\omega} \times \mathbf{H} \tag{21}$$

The magnitude of ω is M/H where M and H are the magnitudes of the respective vectors. Figure 76 shows a diagram of a single-degree-of-freedom integrating gyro. Inside the case is a float cylinder assembly which is pivoted to the case by a bearing at each end [63]. A viscous fluid fills the space outside the float and provides viscous damping; this fluid also provides buoyancy to the float thereby unloading the bearings. Inside the float is a high speed wheel mounted transverse to the axis of symmetry of the float—the gyroscopic element. At one end of the case is a signal generator which measures rotation of the float about the OA. At the other end is a torque generator which applies torque to the float about the OA.

The application of an integrating gyro for controlling a single-axis stable table is shown in Fig. 77. If a disturbance moves the stable table at a rate ω with respect to inertial space about the gyro IA, the case tends to rotate about its OA. This rotation is detected by the signal generator which sends an error signal

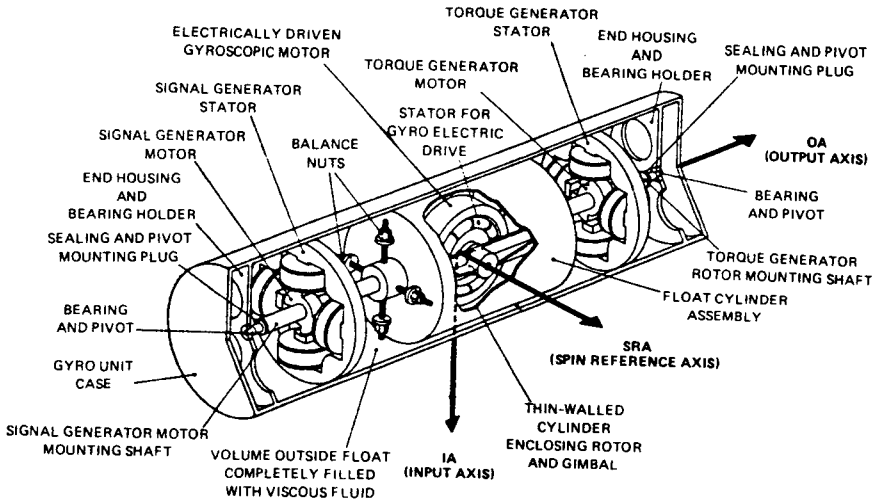


Fig. 76 Single-degree-of-freedom integrating gyro unit

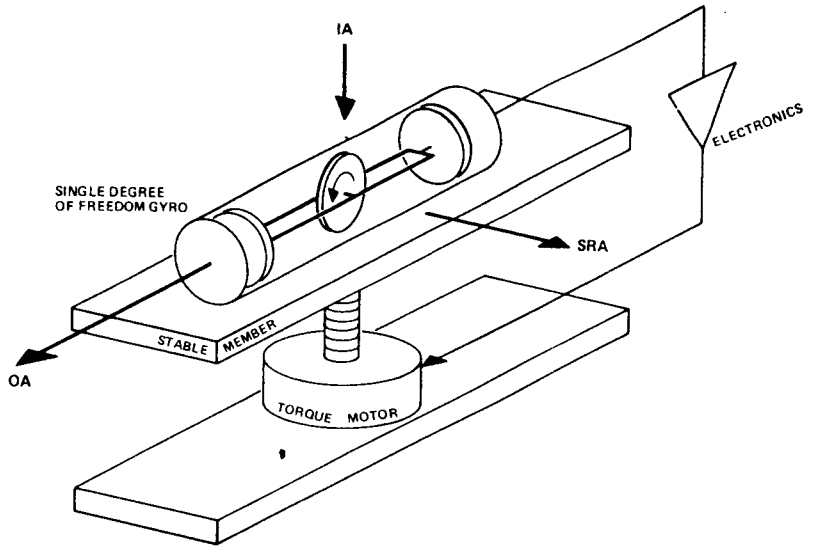


Fig. 77 Single-axis stable table or space integrator

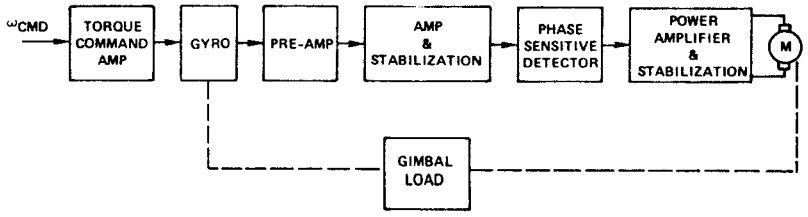


Fig. 78 Single-axis stable table functional diagram

to the electronics, which in turn drives the gimbal torque motor, moving the stable member back to its original position. If an electrical signal is applied to the torque generator, the torque about the OA will cause the table to revolve about the IA. This shows that a gyro may be used to control the motion of a stable table about an axis. A single-axis stable table functional diagram is shown in Fig. 78.

In an inertial navigation system, motion must be controlled about three axes, requiring three gyros and three drive motors. Figure 79 is a pictorial diagram of a three-degree-of-freedom stable table. By application of appropriate precessional torquing commands to the gyros, the angular velocity of the stable platform can be made to track a desired reference frame.

In Fig. 80 we have a simple inertial navigation system. The x and y axis accelerometers have their IAs in the horizontal plane (no g components), and are initially aligned to some horizontal X and Y coordinate system. As the vehicle moves, any rotations of the stable platform about the vertical will be sensed by the gyro and corrected by the torque motor, keeping the instrument x and y IAs aligned to the original X and Y navigation coordinate axes. Thus, all velocity and position measurements will be correct in the original coordinate system.

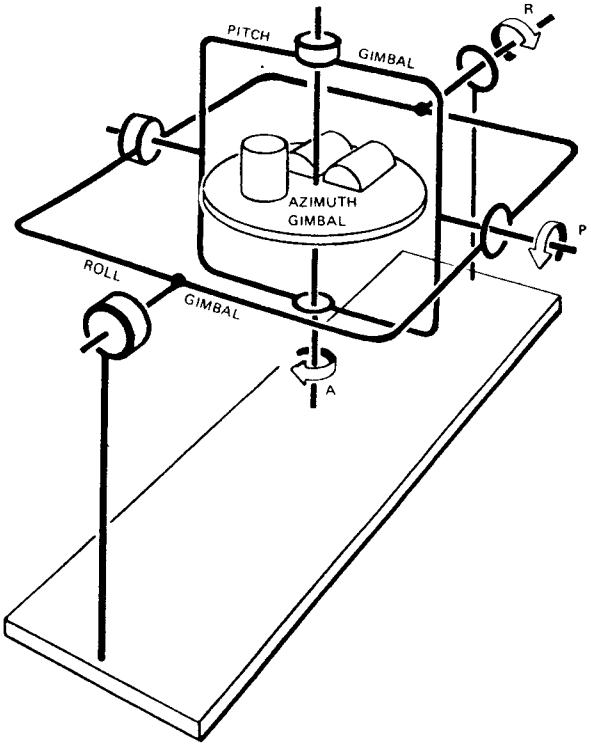
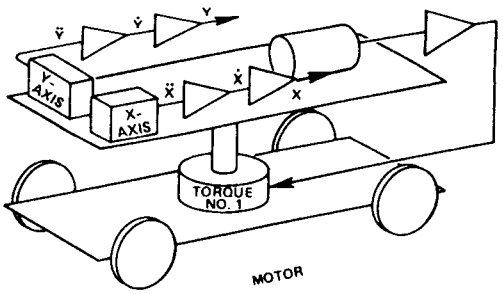


Fig. 79 Three-degree-of-freedom stable table



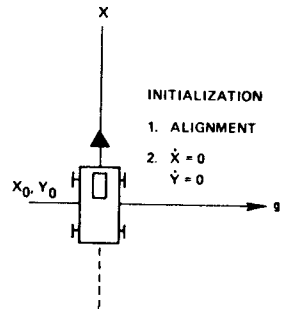


Fig. 80 Simple inertial navigator

In general, it was shown in Fig. 72 that a simple pendulum will be displaced from the gravity vector by an input acceleration. If the pendulum length, however, equals the radius of the earth, it is found that accelerations cause no deflections. The pendulum in this case can follow any movements over the surface of the earth without error in indicating the vertical. The natural period of such a pendulum, the so-called Schuler period, is given by

$$T = 2\pi\sqrt{\frac{R_e}{g}} \approx 84 \text{ min} \tag{22}$$

where R_e is the radius of the earth and g is the magnitude of acceleration due to gravity at the earth's surface. Of course, it is not possible to implement this configuration physically. However, any device which, for a small disturbance angle β from the vertical, has a restoring force equal to $g\beta$ and has

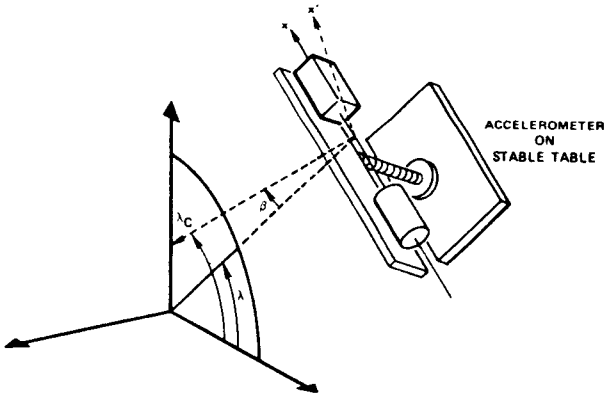


Fig. 81 Single-axis local-level inertial navigator geometry

a natural period equal to T, will simulate a Schuler pendulum [65].

A single-axis local-level inertial navigation system can be configured to have a dynamic behavior highly analogous to the Schuler pendulum [63]. The diagram for this system is shown in Fig. 81.

The vehicle containing this system is constrained to move on a meridian of longitude. For simplification, it is assumed the earth is nonrotating and spherical. The stable table has an accelerometer with an IA along x which is originally pointed North and a gyro with an IA pointed East. The vehicle may have any type of acceleration and may pitch but may not roll or yaw. In the figure, λ is vehicle latitude, λ_c is calculated latitude, and β is the error in vertical indication. The functional block diagram for this system is shown in Fig. 82.

The specific force/velocity relationships are given by

$$f = -g\beta - \dot{V} = -\dot{V}_{NC} \tag{23}$$

The miserection angle relationships are given by

$$\beta = \lambda - \lambda_c \tag{24}$$

$$\dot{\beta} = \dot{\lambda} - \dot{\lambda}_c = \omega_N - \omega_{NC} \tag{25}$$

The velocity/angular velocity relationships are given by

$$\omega_{NC} = KV_{NC} = \frac{K}{S}\dot{V}_{NC} = \frac{K}{S}(g\beta + \dot{V}) \tag{26}$$

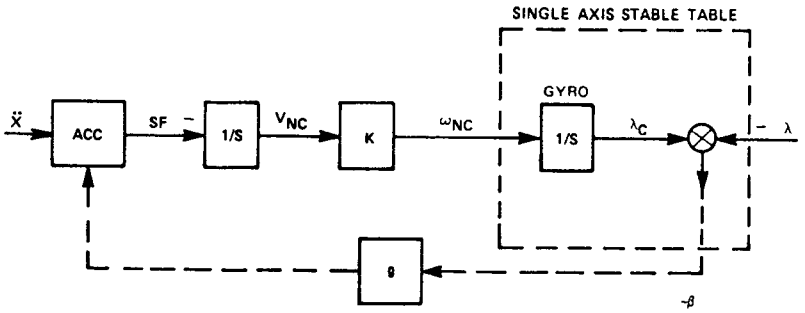


Fig. 82 Single-axis inertial navigation system functional diagram

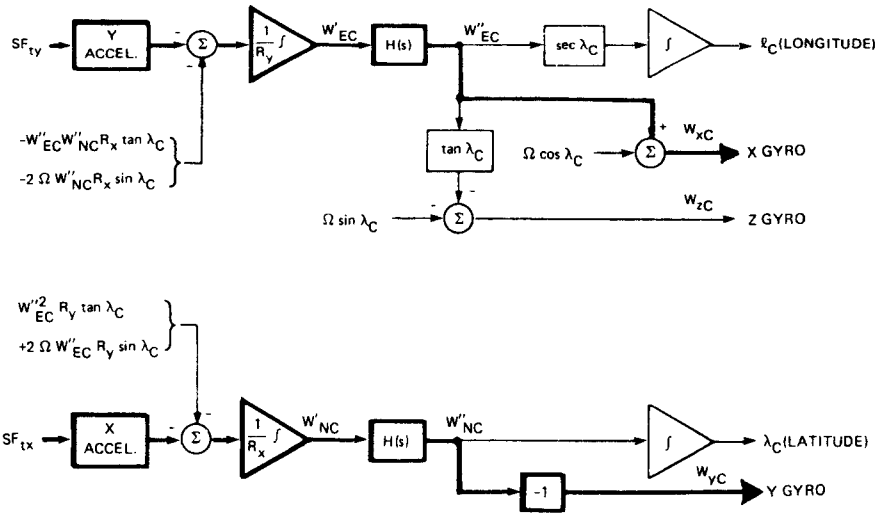


Fig. 83 Local vertical geographic reference inertial system functional diagram

Combining equations (25) and (26),

$$\dot{\beta} = \omega_N - \frac{K}{S}(g\beta + \dot{V}) \tag{27}$$

Differentiating equation (27) and rearranging,

$$\ddot{\beta} + Kg\beta = \dot{\omega}_N - K\dot{V} \tag{28}$$

For a spherical earth

$$V = R\omega_N \tag{29}$$

$$\dot{V} = R\dot{\omega}_N \tag{30}$$

If K is made equal to $1/R$, then

$$\dot{\omega}_N - K\dot{V} = 0 \tag{31}$$

Hence,

$$\ddot{\beta} + \frac{g}{R}\beta = 0 \tag{32}$$

This equation is identical to the equation for the Schuler pendulum and shows that the error of vertical indication β oscillates with an 84-min period. This error is not forced by tangential acceleration $\dot{V}$. This navigation system will indicate the vertical whether the vehicle is at rest or accelerating. This simple single-axis case contains many of the essential concepts of inertial navigation. The actual case of interest in real applications is the three-axis case, since vehicles move in three dimensions. However, for this discussion it is not practical to go into the details of these systems. Interested readers can consult the references given and investigate the broad technical literature that exists for this subject. They will find that a three-axis system has marked differences from a single-axis system. Certain of the system variables will be found to have 24-h components arising from earth's rate and beats will be found in the 84-min vertical oscillations.

Most marine applications can be performed with a local vertical geographic reference inertial system. This system can be implemented with two accelerometers and three single-degree-of-freedom gyros mounted on a three-axis platform. The functional block diagram for one such implementation is shown in Fig. 83. The coordinates for this system are shown in Fig. 84. The measurement system (x,y,z) moves with the vehicle and has the x axis pointed north, the y axis pointed east, and the z axis pointed down along the local vertical. In the figure, latitude is specified by λ, latitude rate by $\dot{\lambda}$, longitude by l, longitude rate by $\dot{l}$ and earth's rotational rate by Ω. Components of these rates appear on the x, y, and z axes. If the stable platform with gyros mounted with their IAs along these axes is to remain in coincidence with the geographic frame, then it must move at these angular rates. Looking back at Fig. 83 we see that torquing signals with these rate values are indeed sent to the gyros. As previously explained under the stable table discussion, the platform will move at a rate corresponding to a torque generator signal.

One more concept must be developed for this geographic reference system. As given in equation (20), an accelerometer measures a component of specific force. The acceleration portion of this specific force is the acceleration of the vehicle with respect to a nonrotating reference (initial space). What we want is the acceleration in the reference frame in which we are performing our navigation, in this case the geographic system. However, as just pointed out, this system is rotating. In general, the acceleration of a point relative to a fixed frame can be expressed in vector form as

$$\left.\frac{d^2\mathbf{R}}{dt^2}\right|_s = \left.\frac{d^2\mathbf{R}}{dt^2}\right|_R + 2\boldsymbol{\omega}\times\left.\frac{d\mathbf{R}}{dt}\right|_R + \left.\frac{d\boldsymbol{\omega}}{dt}\right|_R \times \mathbf{R} + \boldsymbol{\omega}\times\boldsymbol{\omega}\times\mathbf{R} \tag{33}$$

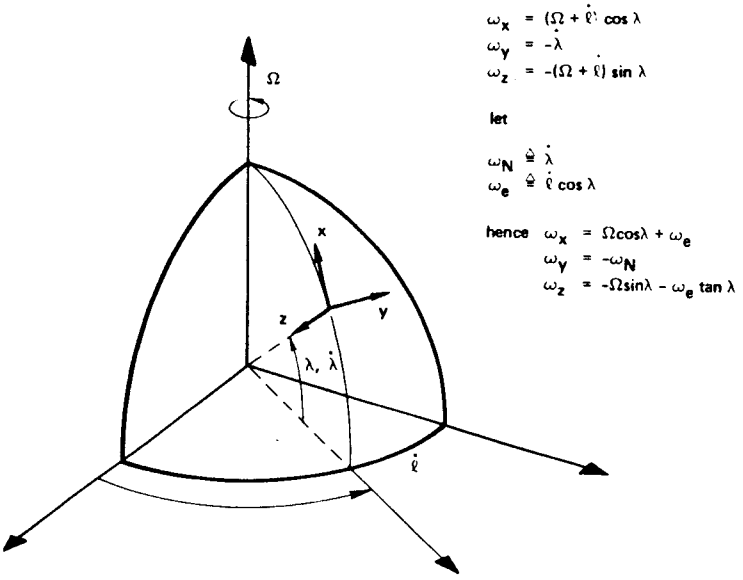


Fig. 84 Local geographic frame angular velocities, earth rate, and vehicle translation

The S notation on the derivatives correspond to the derivative with respect to a fixed, nonrotating frame, while the R notation refers to a derivative with respect to a rotating frame. $\boldsymbol{\omega}$ is the vector angular velocity of the rotating frame relative to the fixed frame. Accelerometers measure

$$\mathbf{f} = \mathbf{G} - \left.\frac{d^2\mathbf{R}}{dt^2}\right|_S \tag{34}$$

What we would like for navigation in the rotating frame is

$$\left.\frac{d^2\mathbf{R}}{dt^2}\right|_R \tag{35}$$

The second term in equation (33) is a Coriolis force per unit mass, and the last term is a centrifugal force per unit mass. In the local geographic system it is found that the centrifugal term, due to earth's rate, can be combined with the gravitational field vector $\mathbf{G}$ to give the local gravity vector $\mathbf{g}$, which is the direction a pendulum takes. Since the z axis is along this direction, the x and y axes sense no components of $\mathbf{g}$. In Fig. 83 we see that the accelerometer outputs are combined with signals that compensate for the Coriolis terms and thus give accelerations in the x and y directions.

What has just been presented are the important concepts of inertial navigation. Many other topics should be included to make the coverage complete, but this would go beyond the limited scope of this chapter. The effects of instrument errors in both gyros and accelerometers should be considered. Different configurations for implementing a navigator besides the geographic reference system briefly discussed should be treated. An inertial navigator can be configured to have the stable platform referenced to inertial space (nonrotating).

In the application of inertial navigation, one has to consider the problems of the initial erection and alignment of the reference axes. This can be done quite easily if the vehicle on which the system is mounted is not moving. On a moving vehicle, special methods have to be considered for erection and alignment. In addition to the initial alignment of the system, the initial conditions of velocity and position are important for system performance. As we know, an initial velocity error will be integrated with time, giving a position error.

Another subject that should be noted is that of damping. A Schuler loop as previously discussed is a free oscillator. In different system configurations, an undamped loop can be driven by a forcing function such as random gyro drift, resulting in an unbounded error. To compensate for this, damping can be added to the loop, thereby bounding this source of error. However, damping now makes the system sensitive to vehicle accelerations; obviously, compromises must be made depending on the application. For marine applications, in relatively slow moving vehicles, a damped system can be utilized. If an independent velocity measurement is available such as from a Doppler sonar, the system can be velocity compensated and made insensitive to vehicle motion.

Before ending this section, it should be noted that with the development of high-performance instruments, such as laser gyros and advances in microcomputers, strapdown inertial navigators are now practical [69]. In this type of system, the stable platform has been eliminated. Three accelerometers

and the gyros are mounted directly to the vehicle; they are no longer isolated from the vehicle motions. In effect, the instruments are analytically isolated from vehicle rotations by means of the digital computer in the system, which constantly takes the gyro output signals which result from vehicle motion and updates a coordinate transformation matrix which contains the direction cosines between the vehicle axes (roll, pitch, and yaw) and the navigation reference coordinate axes. In this manner, the accelerometer output signals in body (vehicle) axes can be resolved into navigation axes components. When laser gyros are used in this type of system, instead of the conventional spinning wheel gyros, an attitude reference is provided with no moving parts except for the accelerometers. By replacing electromechanical components with electronic ones, the overall reliability of the system is improved. In addition, the "warm-up" time of a laser gyro is greatly reduced over that of the conventional instruments.

An inertial navigator is a very expensive system. For high-accuracy performance, it requires very sensitive and precise inertial instruments. These sensors require great care in design and fabrication. In addition, they require elaborate testing to verify component performance. They then have to be integrated into a gimbal system with the same degree of care. All of these steps add to the overall cost of generating an inertial navigator. At the system maintenance level, skilled technicians are required to maintain and repair these systems.

In ending this section, we again point out that an inertial navigator is a self-contained system that provides position and velocity. In addition, it can provide attitude and act as a very accurate heading reference. Because of this capability it can be used to provide the ship's crew with steering information; for an automatic steering system, it provides the heading signal. The inertial navigator has found many applications in military systems, especially submarines. The system can provide position with an accuracy that normally ranges from 1.9 to 0.19 km/h (1 to 0.1 nmi/h). To realize a lower value takes accurate gyros with random drift rates of 0.001 degrees per hour.

4.6 Navigation and Positional Aids Summary

The cost of a submersible operation is generally high because of the amount of support equipment and number of personnel involved. A chosen navigation system must be capable of guiding a submersible support system to the operating location in the most expeditious manner. In addition, once on site, the navigation system must provide the necessary navigation and position fixing required for the surface operation while the submersible is underwater.

Many submersible vehicles do not contain navigation and position fixing equipment. When accurate knowledge of the position of these vehicles is required, the surface support ship tracks their positions. The position accuracy of the support ship's navigation system then contributes to the accuracy of the submersible's position estimate.

In the preceding sections, the theory and characteristics of a number of current navigational aids have been reviewed. The author feels that they are the important ones, though not the only ones, available today. He also feels that submersible system designers should have the basic knowledge that has been presented on these systems as part of their technical tools. This will allow them to make an initial choice on the type of navigation required during a submersible system conceptual design. They will also be able to define navigation requirements and interface with navigation specialists when complex navigation systems are required to meet stringent performance. It will also allow them to understand the limits and capabilities of the different systems available for solving the navigation problem.

To summarize this section, the salient features of navigation systems reviewed are presented in Table 18. One parameter in this table that has not been previously addressed is absolute cost. To give an absolute cost on the different systems reviewed would be most difficult. Because of advances in microelectronics and in digital processing equipment, there has been a generally downward trend in the cost of some navigation systems. Other factors that influence the cost of navigation systems are performance, reliability (mean time between failures), environmental resistance, and the market for which the system is developed, for example, a system developed for military use versus one for a commercial application.

Another thing that influences the cost of a system is the additional features that may be present in a particular navigation system; for example, a hyperbolic radio navigation system could be configured to provide the solution to the navigation problem with an output in latitude and longitude coordinates instead of range difference outputs. In addition, it might provide range and bearing to the next waypoint, and even steering commands to guide the course of the ship.

The costs indicated in Table 18 are given in relative terms. This will allow the reader to make comparisons between the different systems listed.

However, it must be kept in mind that with advances in solid-state electronic technology, application of such devices as VLSI (very large-scale integration) chips and microprocessors will bring about dramatic reductions in navigation system costs.

In the description on navigation systems, no effort has been devoted to the physical characteristics of weight, size, shape, etc. Such information is readily obtainable from specific manufacturers. All the sys-

Table 18 Comparison of navigation systems

System	Position Accuracy	Range	Cost	Notes
Decca	0.4 km (0.2 nmi)	445 km (240 nmi) from Master Station	Can be leased at a reasonable price	Medium range Continuous signal Limited coverage—mainly Northern Europe
Loran C	0.2 to 0.5 km (0.1 to 0.25 nmi) for range to 1800 km 7 to 33 km (4 to 17 nmi) for range out to 3700 km	1500 to 2000 km (800 to 1080 nmi) Ground Wave Operation from Chain 3000 km (1620 nmi) Skywave Operation from Chain	Moderate to medium expensive	Repeatability 15 to 91 m (50 to 300 ft) available in U.S., coastal waters, North Atlantic, Norwegian Sea, Mediterranean, Canadian East Coast, North-Central, Northwest Pacific, Alaska
Omega	7.4 km (4.0 nmi) 95% of the time	9300 km (5000 nmi) from Master Station	Moderate to medium expensive	World-wide coverage Requires propagation phase corrections
EDM Systems—Short Range	5 to 10 m (16 to 33 ft)	15 to 250 km (8 to 135 nmi)	Moderate	Requires setting up transponders and/or equipment at surveyed points
EDM Systems—Medium Range	10 to 20 m (33 to 60 ft)	370 to 740 km (200 to 400 nmi)	Moderate to medium expensive	Requires setting up transponders and/or equipment at surveyed points
Transit	0.19 km (0.1 nmi) Static measurement less than 50 m	Satellite must be in "view"	Medium expensive	Discrete coverage approximately every 90 min Velocity correction must be used for dynamic fixes
GPS	10 to 100 m (33 to 330 ft)	4 satellites must be in "view" for normal operation; with an accurate clock and altitude input, can operate with 2 satellites in "view"	Medium expensive to expensive	Measures velocity to an accuracy of 0.03 to 0.3 m/s (0.1 to 1.0 ft/s) Provides accurate time World-wide coverage when full constellation of satellites are available in the 1990s Civilian accuracy less than full military accuracy
Sonar Transponder Navigation	1 to 10 m (3.3 to 33 ft)	2 to 20 km (1 to 10 nmi)	Expensive	Requires beacon net survey; beacons have limited life Continuous coverage in area where net is set up
Doppler Velocity Sonar	0.2% to 1% of distance travelled	60 to 150 km (200 to 500 ft) ocean depth	Medium expensive to expensive	Requires accurate heading reference Must be corrected for the speed of sound variation to achieve high accuracy
Correlation Sonar	0.5% of distance travelled	Full ocean depths	Expensive	Requires accurate heading reference
Inertial Navigator	0.2 to 2.0 km/h (0.1 to 1.0 nmi/h)	Not applicable	Expensive to very expensive	Requires initial erection alignment and state information Self-contained operation No radiated energy required for operation

tems described are mostly electronic in nature and will require electrical power. The size of and the power required by these systems vary depending on the nature of the specific system. However, one will find that variations are more likely to be a factor of two rather than an order of magnitude. In addition, many of the electronic boxes will be found to be in the size range of one cubic foot or less. The number of boxes required depends on such things as the requirement for a display or plotter, the need for power supplies and for digital processing equipment.

Radio systems require antennas to be mounted externally on the support ship. Sonar navigation aids require sonar transducers mounted on the underside of the ship. In general, electronic boxes are mounted inside the ship. A sonar beacon navigation system requires the support ship to have the capability of handling and launching beacon transponders.

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Appendix 1

Advantages and Disadvantages of Some Handling Systems for Manned Submersibles

Handling System	Advantages	Disadvantages
Cranes		
Fixed-Boom	Easy to install and maintain Simple to operate Cost Flexibility—can be used to lift many loads Capacity for growth and adaptation of subsystems Favorable launch and recovery times in calm seas Requires little deck space	Provides no control over pendulating load Sea state 3 limited Requires line handlers to steady suspended submersible Requires a large number of people for launch and recovery operations Snap loads may occur if motion is not compensated for
Articulated	Easy to install and maintain Simple to operate Cost Flexibility—can be used to lift many loads Capacity for growth and adaptation of subsystems Favorable launch and recovery times in calm seas Requires little deck space Minimum pendulum motion imparted to submersible Maneuverable Small stowage envelope Low CG Used successfully for over 10 years by HBOI	Severe side loads Requires divers in the water Close proximity of boom tip to submersible is dangerous to the diver Requires additional operator training
Bridge/Gantry	None, unless used with a cradle/elevator. (See cradle/elevator row for advantages)	Expensive Inflexible—use with other systems is limited Severe pendulum motions if not used with a cradle/elevator
A-Frames	Stable structure Easy to install and maintain Simple to operate Good control of motion compensated Used successfully for at least 15 years New systems require one man to operate	Expensive Inflexible—use with other systems is limited Severe pendulum motions if not used with a cradle/elevator
Cradle/Elevator	Tried and proven systems Liked by submersible crews that have used them Favorable launch and recovery times May not require divers in the water	Expensive Inflexible—use with other systems is limited
Docking Well	No pendulum motion to contend with No constraint on the size of submersible that can be supported	Expensive Inflexible—use with other systems is limited Severe pendulum motions if not used with a cradle/elevator
Submerged Systems		
Submarine	Do not have to contend with problems associated with the air-sea interface Submersible can operate year-round anywhere in the world No divers or line handlers required	Very involved mating procedures Difficulties associated with retrieving a disabled submersible on the surface Difficulties associated with routine maintenance and servicing of the vehicle for the next dive
Platform	Do not have to contend with problems associated with the air-sea interface Cost Few people are required for launch and recovery	Must be towed to the dive site Reduced visibility operations are hazardous Additional submersible pilot and diver training Operations limited to coastal areas

Appendix 2

Motion Compensating Techniques

1. Introduction

Motion compensation and the techniques used to accomplish it are defined and discussed in this appendix. Examples of motion compensation applications are given for a fixed-boom crane, articulated crane, an A-frame, and an elevator.

All of these systems, with the exception of one of the articulated cranes, have been used to launch and recover manned submersibles. The fixed-boom crane was used on the *Energy Services I* to handle the *Sea Cliff* and *Turtle*, the A-frame system supports the *Alvin*, and the elevator is used onboard the ASR-21 class submarine rescue ships to deploy and recover the DSRVs. Two articulated crane systems are presented. The first, used to support ROVs and oceanographic instrumentation, is an example of a state-of-the-art handling system. The second is the HBOI's Aluminum Hydrocrane, used to launch and recover the *Johnson-Sea-Link I* and *II* submersibles.

2. Motion Compensation Defined

It is desirable to conduct all handling operations at sea over a wide range of environmental conditions. In order to extend these marine operations to higher sea states, specially designed handling systems must be incorporated. This is particularly true for small ships which are typically used for many handling or support operations. In this case even small waves may cause large response motions of heave, pitch, roll, and yaw during launch and recovery operations. These motions make handling difficult, subject the entire handling system and submersible to severe loads, and make the entire operation potentially hazardous to all the personnel and equipment involved.

Some form of motion compensation is required to protect the support ship, submersible, and handling system from severe loads, as well as to maintain relative safety for the personnel involved with the operation. One approach to the problem is to use a larger vessel. However, the cost may make this alternative unrealistic. The practical approach is to design equipment to operate safely and effectively under expected conditions.

The best location—for motion minimization—for a launch/recovery handling system is at the center of the support vessel's pitch, roll, and yaw axes, leaving only the heave of the vessel to be dealt with. This approach is rarely practical, however, and generally, operations must take place either over the stern, or over the side, or both. For stern operations, pitch offers the severest restrictions to handling and results in the highest loads. For over-the-side operations, roll is the primary concern. The period of both pitch and roll depends primarily on vessel characteristics, while amplitude depends on the sea state.

Short periods cause the highest loading and require the fastest responses from the motion compensator. The longer heave periods impose less severe loading restrictions, but for an improperly designed passive compensator can cause resonance problems.

Motion-compensating devices can be incorporated into a hydraulic winch, or into the handling system's structure, or as a separate subsystem. Ram tensioners are an example of a separate subsystem. When the motion compensating device is included in the boom of a crane, the boom is driven up and down in such a way that its tip stays in approximately the same horizontal plane.

3. Types of Motion Compensation

There are three techniques used to mitigate the motion of the vessel while handling loads over the side or stern: passive, active, and a combination of passive and active. Passive systems work by maintaining a constant tension on the lift cable, or tether, in the case of an ROV. A hydro-pneumatic ram tensioner is a typical passive compensation system, and is illustrated in Fig. 85. With this technique, the slack in the cable is taken up by a hydraulic ram located between the lift winch and the load. The ram cylinder is pressurized with hydraulic fluid, and the

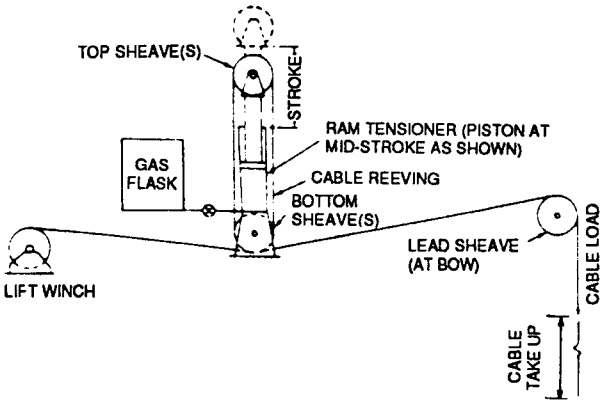


Fig. 85 A hydro-pneumatic ram tensioner; this is a passive motion-compensating system [30]

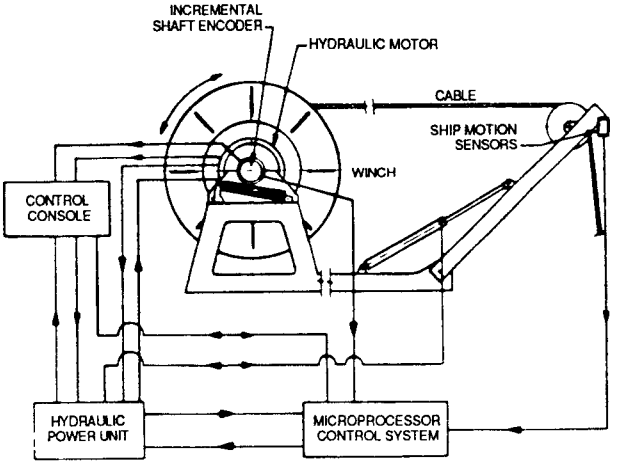


Fig. 86 Schematic of an active motion compensating system [32]

accumulator cylinder is charged with a gas, usually nitrogen. The position of the ram tensioner is maintained by the force equilibrium between the piston rod side and the gas side of the ram cylinder.

Another approach incorporates the passive system into the lift winch assembly. The operation, then, is as follows [70]:[4] gas accumulators are connected to either side of the hydraulic motor which drives the winch. When the mother ship moves downward and the cable tension drops, the gas pressure in the accumulators drives the motor to haul in the cable and maintain the tension. Conversely, as the ship moves upward, the increased cable tension drives the motor in reverse and pumps fluid into the accumulators. The hydro-pneumatic accumulators act in the manner of a vibration absorber, isolating the cable from the motion of the ship. By adjustment of the gas pre-charge pressure and volume, the absorber can be tuned to provide good compensation over a certain range, coinciding with the dominant periods of vessel motion. The range, however, is limited, and in some instances a passive compensator could amplify the vessel motion as felt by the submersible.

An active motion compensating system works by sensing the support ship's movement, predicting the future movement of the load, and taking appropriate action to minimize the load's movement [71]. A simplified schematic of this kind of system is shown in Fig. 86. The accelerometer, mounted on the apex of the A-frame, provides information on the vertical acceleration of the support ship. This acceleration information is measured at the point where the load is suspended from the A-frame. The information is supplied to a microprocessor unit inside the control box. This microprocessor controls the main active lease compensating pump. Feedback on cable motion is provided by a shift encoder mounted on the cable drum.

[4]References precede appendices.

Motion compensation systems that utilize both passive and active devices may be more practical because they combine the advantages of each. For instance, they join the low power requirement of a passive system with the precise motion control of an active, direct-drive hydraulic winch. The active part refines the response of the passive part by driving the compensator from an external power source [72]. This is in response to a signal from the accelerometer mounted close to the sheave on a crane boom or to the apex of an A-frame.

Active/passive motion compensators minimize the external power requirements because the static cable load is mostly supported by the passive system. The active system portion supplies only the power needed to accelerate the equivalent mass of the compensator. However, because the active part can cause system instability, the entire system must be designed carefully.

Motion-compensating systems are not without disadvantages. For example, passive systems, specifically ram tensioners, have limited travel. When the ram or the boom bottoms or tops out in high sea states, it is like turning off the compensator. In addition, changing the speed and scope of the cable can cause the nominal load on a compensator to change. If this is not accounted for, the system will become unbalanced and drift in a corresponding direction. However, a boom compensator can detect this change and will automatically make adjustments to compensate for the conditions [71].

Active winch systems consume a good deal of power and, since they are made up of many moving parts, are subject to a higher probability of failure than passive systems. Kidera [71] states that there are other problems with an active winch system. Duty cycle time tends to be limited due to overheating. One of the biggest problems with any active system is that of actually getting it to work. A successful system must be able to predict the future movement of the vessel with high accuracy. Although in theory it is possible, in practice it rarely is. An active system that is slightly inaccurate in its predictions of random motions is worse than no compensation at all. Active systems also tend to be considerably more expensive than passive systems, both in terms of initial cost and in terms of maintenance.

Both ram tensioners and active/passive systems subject the lift cable to large excursions over one or more sheaves. This can cause fatigue in the lift cable and can severely limit its useful life.

For a more detailed description of these systems, see references [71] and [72].

4. Examples of Motion Compensating Systems

4.1 Fixed-Boom Crane

One approach to compensating for a ship's motion when using a fixed-boom crane was used in conjunc-

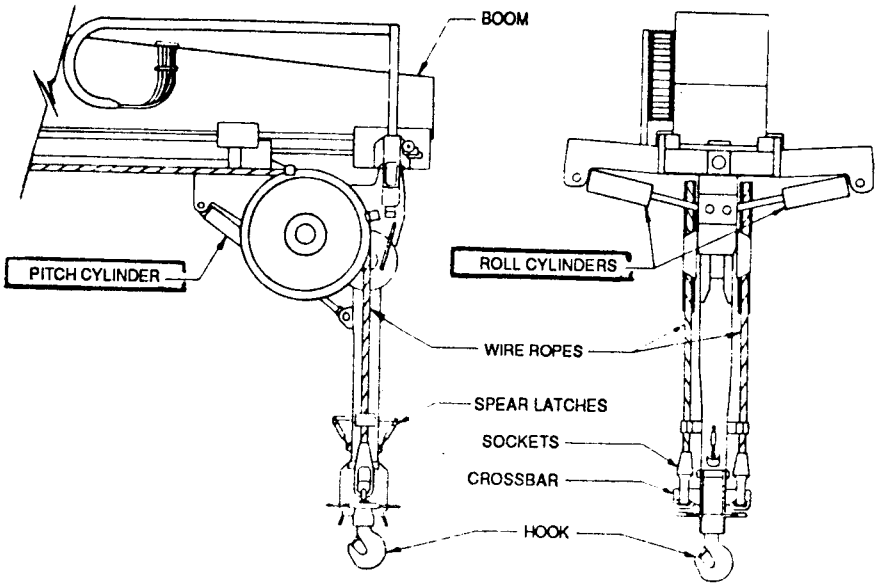


Fig. 87 Anti-pendulation devices on the DSV handling crane [11]

tion with the DSV Handling Crane (Fig. 87). This crane was used to deploy and retrieve the *Sea Cliff* and *Turtle* from their support ship, *Energy Services I*, during the late 1970s to early 1980s.

Information on this system was taken from reference [11]. It is a passive system in which the major components of the hydraulic motion-damping system are the two roll cylinders, the pitch cylinder, and the accumulator. Associated with the system are relief valves and a pressure-reducing valve.

The pitch-damping cylinder is attached between the hook snout and the inboard travelling cart. It dampens the swing motion of the snout in the vertical plane parallel to the boom axis, up to 60 degrees inward and 15 degrees outward. The pitch cylinder has a 5-in. bore, a 22$\frac{5}{16}$-in. stroke, and is rated at 2800 psi. In the pitch-damping system there is a low-pressure accumulator to accommodate the unequal fluid volumes between the rod end and the head end of the pitch cylinder. The roll-damping cylinders damp the swing motion of the cart/snout in a vertical plane perpendicular to the boom axis, up to 30 degrees each side of the center. There are two roll-damping cylinders, one on each side of the cart, attached between the cart and the other outboard truck. The roll-damping cylinders have a 6-in. bore, a 16$\frac{1}{4}$-in. stroke, and are rated at 3050 psi.

The crane control panel contains two damping control valves. These valves adjust the spring pressure of the relief valves for the roll- and pitch-damping systems, which in turn vary the degree of motion damping.

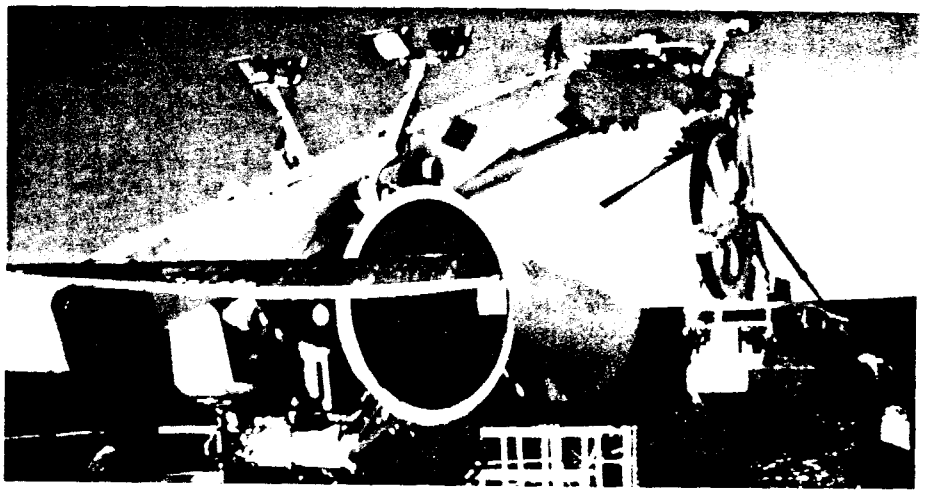


Fig. 88 Motion compensation launch/recovery crane

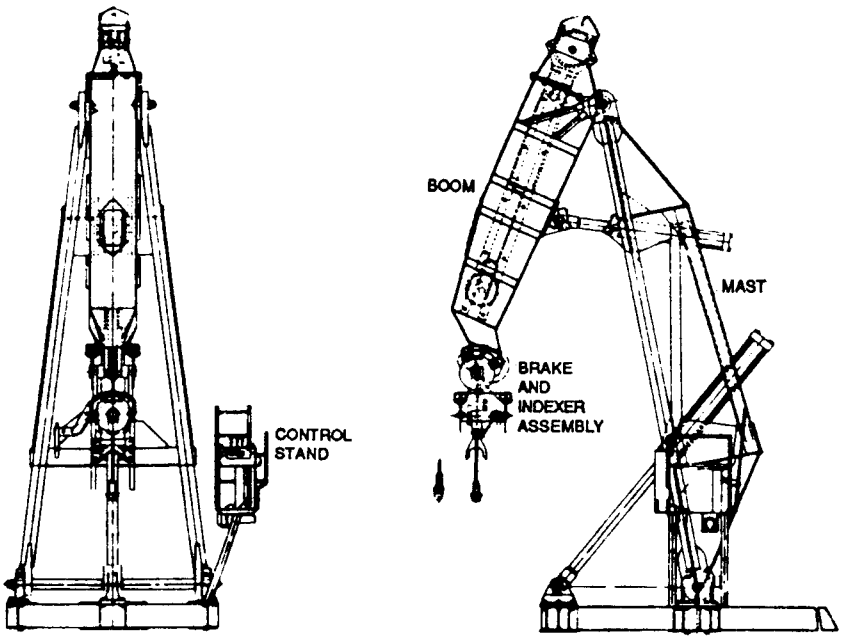


Fig. 89 Roll- and pitch-damping brake system on the Harbor Branch Oceanographic Institution aluminum hydrocrane [16]

4.2 Articulated Crane

Ocean Equipment Development, Inc., has designed a number of passively compensated cranes to support ROVs, oceanographic instrumentation, diving bells, and towed systems, or any other handling situation within the crane's weight range. These cranes are boom compensated. The crane used in this example is the Launch/Recovery Motion Compensator (LRMC-18) capable of launching a 6700-lb load and compensating an 8000-lb load with reduced stroke. The maximum compensation stroke is 26 ft. The crane system is shown in Fig. 88.

Key features of the LRMC-18 include isolation of the compensation hydraulics from the crane hydraulics, a special sheave/saddle arrangement, a "doughnut" fairleading system, and the capability of unattended operation.

A brief outline of a launching operation (for recovery, follow in reverse order) is presented below so that the reader will have a better idea of how the system is used:

1. Crane is positioned over vehicle, tether put through sheave, and connected to vehicle.
2. Vessel is brought to proper heading for launch.
3. Crane is in launch mode—main compensation system off, launch compensation system on.
4. Operator lifts vehicle, extends boom over the stern, and checks status.
5. Boom is lowered until vehicle is submerged.
6. Winch pays out cable, and the main compensator is turned on.

Another example of the use of a motion-damped articulated crane to launch and recover manned submersibles is HBOI's launch-and-recovery operation for the submersibles, *Johnson-Sea-Link I* and *II*. Their motion-damped system uses roll- and pitch-damping brakes which are incorporated into the

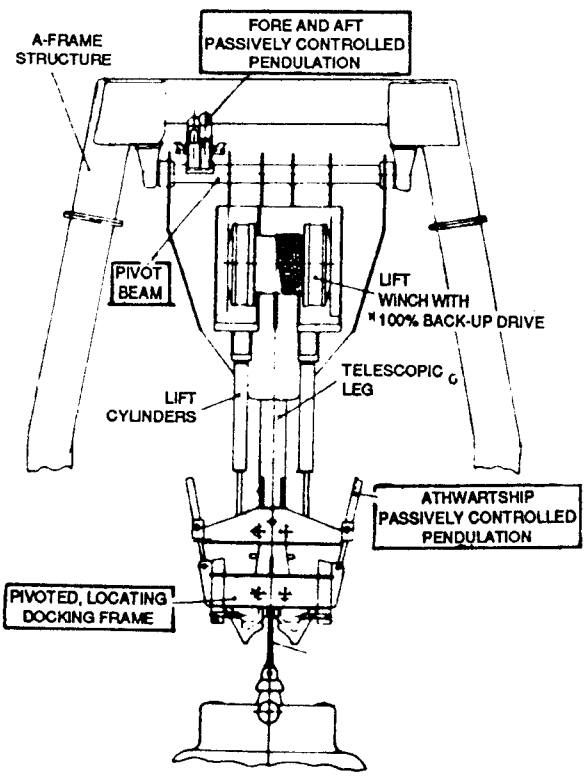
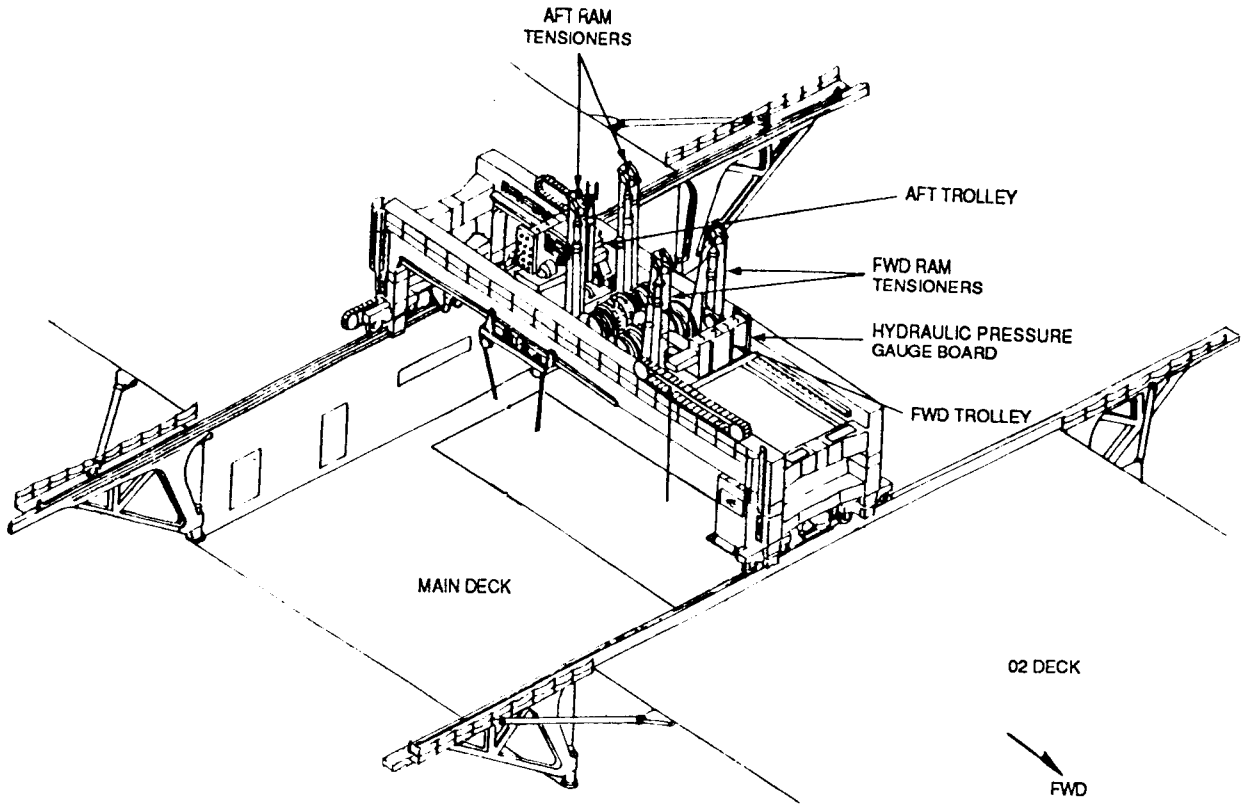
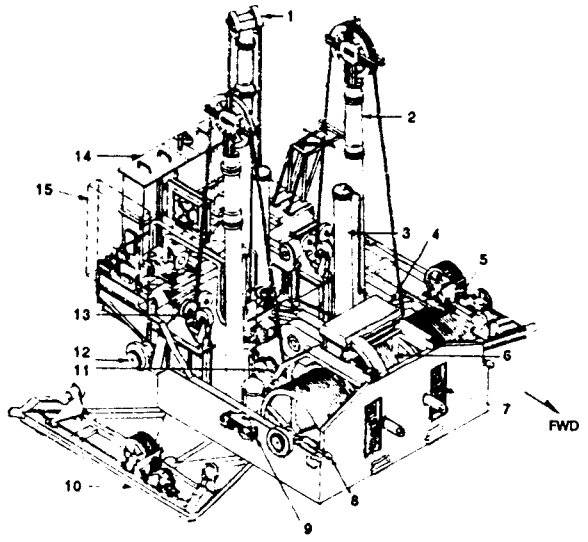


Fig. 90 Motion compensated A-frame [20]

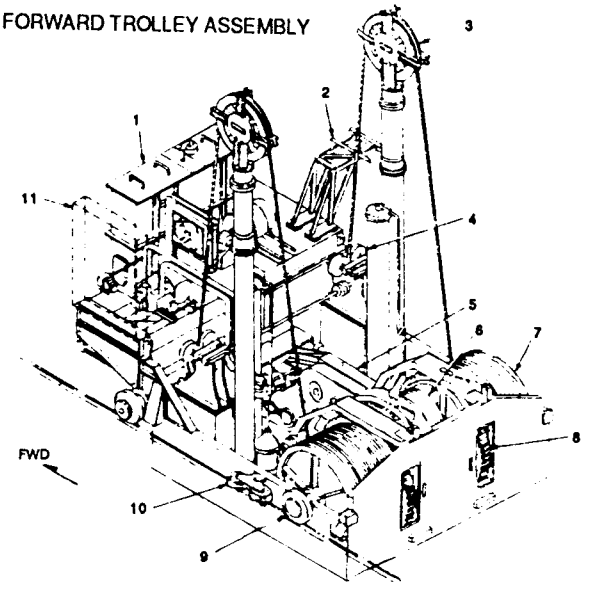


AFTER TROLLEY ASSEMBLY



1. AUX. HOIST RAM TENSIONER
2. MAIN HOIST RAM TENSIONER
3. RAM TENSIONER ACCUMULATORS
4. LEVEL WIND UNIT
5. MAIN HOIST DRUM
6. AUX. HOIST DRUM
7. BULL GEARS
8. SHOCK ABSORBER
9. LATERAL GUIDE ROLLERS
10. STABILIZING SYSTEM
11. MAIN AND AUX. HOIST DRIVE GEARBOX
12. TROLLEY DRIVE PINION
13. SLACK PULLER
14. HYDRAULIC POWER UNIT
15. GAUGE BOARD

FORWARD TROLLEY ASSEMBLY



1. HYDRAULIC POWER UNIT
2. POWER TRACK PICKUP ARM
3. RAM TENSIONER
4. SLACK PULLER
5. MAIN AND AUX. HOIST DRIVE
6. AUXILIARY HOIST
7. MAIN HOIST
8. BULL GEARS
9. CARRIAGE WHEELS
10. LATERAL DRIVE ROLLERS
11. GAUGE BOARD

Fig. 91 ASR-21 class ship bridge crane assembly showing ram tensioners [19]

head of the boom. The hook is held immobile by the socket, as illustrated in Fig. 89.

Each unit of the dual-axis brake system is a completely integrated system. The reservoir, pump, and motor are unitized and operate at a maximum of 1000 psi. Caliper type brakes are used in which a composite shoe bears on a machined stainless-steel disc. Brake pressure is applied by a foot-operated electric switch.

4.3 A-frame

Caley Hydraulics' A-frame design is chosen as a representative state-of-the-art example to illustrate motion compensation techniques as applied to an A-frame. Figure 90 illustrates this concept. It is a design utilized by HBOI.

The Caley Hydraulics design, like all A-frames to date that have utilized motion-compensating devices, uses a passive system. The lift winch, telescopic leg, and other associated submersible handling equipment are secured to a pivot beam at the apex of the A-frame structure. This pivot beam allows the lift winch and the entire mating system to rotate fore and aft, which helps to align the locator-docking frame with the attachment point on the submersible. Two hydraulic cylinders, fastened to the pivot beam and the top of the A-frame, retard the pitching motions. To dampen rolling motion, hydraulic cylinders are secured to the locator-docking frame. This arrangement allows the locator-docking frame to follow and control compounded submersible pendulation. In addition, a constant-tension lift winch is used to prevent slack in the lifting line. It also reduces vertical displacements between the submersible and support system.

4.4 Cradle/Elevator

ASR-21 class submarine rescue ships which support DSRVs with a bridge crane and cradle/elevator system use hydro-pneumatic ram tensioners for motion-compensation purposes. The main and auxiliary ram tensioners are used to maintain even tension on the bridge crane wire ropes and to prevent excess stress on the handling system due to the ship's motions in a seaway. They also absorb shock to the deep-submergence systems when launched or retrieved. A ram tensioner is included with each trolley hoist except the forward trolley auxiliary hoist. Each tensioner is pressurized with nitrogen provided by a nitrogen flask bank in the ship. Figure 91 illustrates the ASR-21 class motion compensation system. Both forward and aft ram tensioners have a maximum operating pressure of 2500 psig, a rated line pull of 43 to 750 lb, and a stroke of 24 ft [19].

5. Summary

Most, if not all, handling systems for manned submersibles which compensate for ship motion do so by using the passive technique. This is true because of problems associated with active systems, discussed in previous sections of this Appendix. However, active motion-compensating systems are now being designed and developed for ROVs. McCallum and Clark [20] state that an active system is the ultimate solution for manned submersible handling. They feel that as microprocessor-based systems become more reliable, active systems will prevail. However, they also note that if these sophisticated systems fail, there should be a passive mode on which the handling system can fall back.

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 alterna-

tives. 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 in-

clude 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].

use. The proof of acceptability of the material or component and justification of the acceptance criteria must be provided by the designer. For example, the designer should demonstrate the effect of defects, manufacturing tolerances, and production variations upon the reliability of the material or component by appropriate model or prototype testing or both in a simulated service environment. The designer must establish that the new material or component possesses at least the same factor of safety as is provided by proven materials in a similar application. As a minimum, the designer should provide justification for the Group 3 material/components.

5.3 Pressure Hull and Hard Structure

In addition to considering conditions such as effects of shock and vibration, creep, thermal transients, or material deterioration due to radiation effects, the designer must consider the failure modes caused by fatigue or fracture.

5.3.1 Fatigue

When pressure hull and hard structure materials are from Groups 1 or 2, a fatigue analysis should be performed; the analysis should generally be based on specimen or model tests or both. Suitable fatigue strength reduction factors should be applied to the specimen or model test results to account for variations in properties, scatter in test results, and the uncertainties involved in applying a fatigue analysis. The designer should consider the following design parameters:

a. magnitude and nature of peak stresses: Stress concentration factors used in the calculation of peak stresses should be based on experimental data on similar structures;
b. material properties and method of fabrication;
c. notches and flaws in the material;
d. geometry of structure and details and penetrations and attachments;
e. previous fabrication history, stress history, and operating history;
f. effect of residual stresses, thermal stresses, and strain rate; and
g. type and method of loading and environmental conditions such as corrosion/erosion: When the pressure hull/vessel hard structure is constructed of Group 3 material, sufficient destructive fatigue tests of full-scale prototypes, models, or penetration mock-ups must be performed to determine experimentally the fatigue life of the design.

5.3.2 Fracture

The designer should assure that the pressure hull and hard structure have adequate resistance to fracture. Specifically, the design analysis by the designer should demonstrate that brittle-fracture or low-energy test failures are not possible modes of failure by considering at least the following:

a. magnitude and nature of peak stresses, both applied and residual;
b. the minimum temperatures to which the structure may be subjected;
c. the size, location, and number of flaws initially present in the material and which may occur as a result of cyclic operation;
d. environmental conditions such as corrosion or erosion or both. Specifically, the effect of sea water on crack initiation and propagation should be evaluated;
e. the effect of engineering stress/strain levels and rates on notch toughness (dynamic versus static);
f. the effects of creep and strain cycling on notch toughness;
g. localized effects due to penetrations, attachments, and other hull or component restraints (stress risers); and
h. the effects of fabrication processes and heat treatments on the fracture characteristics of the material. In particular, for welded construction the properties of the weld metal, base metal, and heat-affected base material should be considered.

The material properties used in the fracture analysis should be based on appropriate tests such as tensile and compressive tests, Charpy V-notch transition curves, dynamic tear test, drop-weight tests, and explosion bulge tests. Where appropriate for low-ductility materials, fracture-mechanics type tests may also be conducted. Further, the design analysis should consider possible variations in material properties and, in particular, the effect of material thickness on fracture characteristics. The structural design basis used by the designer for the analysis of brittle fracture should be verified by destructive testing of pressure hull and component models and structures or, where possible, by reference to existing information and service experience.

For Group 3 materials which are not ductile in the usual sense, the designer should demonstrate that, for all the conditions listed above, a sufficient margin against catastrophic brittle fracture is provided.

5.3.3 Design Strength Parameters of Externally Loaded Pressure Hulls

In general, the designers should demonstrate that the collapse pressure of the hull and hard structure is at least 1.5 times the design operating pressure under loading conditions (environment, loading rate, and duration) representative of those expected in service, whether the collapse is by elastic or inelastic instability. The collapse pressure to which this and

subsequent requirements apply is the lowest pressure at which any one of a series of nominally identical hull structures would collapse. As such, the collapse pressure (analytical and experimental) must take into consideration the effect of basic material characteristics, including creep, ductility, and anisotropy, and must account for statistical fabrication and geometrical variations to assure adequate reproducibility. Examples of pertinent variables include in-service material reproducibility, fabrication flaws and defects, hull openings and intersections of different shells of revolution and attendant reinforcement or reinforcements, residual fabrication stresses, and deviations from the nominal geometry (flat spots, mismatch, tilt, out-of-roundness, out-of-planeness, and out-of-sphericity). The following requirements should be met in order to accomplish this objective.

Inelastic stability (yield)—For stable hull structures (that is, stiffened or unstiffened shells which permit the level of load-induced membrane stresses to approach the material yield point at collapse), the collapse pressure must be no less than 1.5 times the design operating pressure. In the determination of the pressure at which collapse occurs, all fabrication and design-induced restraint and geometrical variables must be considered and their effect or effects included, since the strength of moderately stable hulls can be detrimentally affected by such variables (that is, collapse can occur at actual levels of membrane stress below calculated levels of yield point intensity).

Elastic stability (buckle)—For stiffened or unstiffened hull structures fabricated from either Group 1 or Group 2 material and having a propensity for failure in an instability mode (that is, collapse occurring at actual stress levels appreciably below the material yield point), the collapse pressure at which failure due to instability occurs must be no less than 1.5 times the design operating pressure. When the hull requiring verification is not proof tested at a depth of 1.5 times the maximum operating depth, designers should justify applicability of proven fractional buckling coefficients used in strength calculations to account for the higher scatter normally associated with structures which collapse in this mode (see subparagraph 3 below).

For Group 3 materials, an appropriate ratio of collapse to operating pressure shall be justified by the designer. This ratio shall not be smaller than values given for Groups 1 and 2 materials.

Verification of calculated collapse or burst pressure—For Groups 1 and 2 materials, the calculated collapse or burst pressure or both should be verified by destructive testing or use of existing nondestructive tests. There are three alternative methods which can be used to satisfy this method of verification:

1. When comparable hull geometries and identical materials have been successfully tested to a pressure at least 1.5 times the maximum operating pressure of the structure (that is, achieved design collapse or burst pressure and failure mode or both), use of these test data can be substituted for destructive testing of the structure under review.
2. For new unproven designs that do not fit the above parameters, the calculated collapse pressure may be verified by performing representative destructive model tests, either full or reduced scale. When such testing is performed, the structural model should be sufficiently large to contain representative prototype geometries, material properties and fabrication process restraints, tolerances, and residual stress.
3. For as-built hull designs which by refined calculations have a collapse pressure in excess of 1.5 (for example, approximately 1.75 or greater) times design operating pressure, full-scale pressure testing of the hull structure requiring verification to 1.5 times the maximum operating pressure will be acceptable as verification of the calculations.

For Group 3 materials, the calculated collapse pressure and the reproducibility of this collapse pressure should be verified by destructive model tests or appropriate destructive proof tests or both of a duplicate prototype hull, as appropriate. An acceptable destructive model test is one performed on a model whose scale is such that the mechanical properties of the material either are identical to or differ by a known factor of magnitude from those in the full-scale hull built from the same material and incorporates all structural details of the full-scale hull.

For any material, the test method should duplicate the loading conditions expected in service and, where applicable, should be such that the mode of failure is identifiable. Where appropriate, use may be made of prior experimental results obtained on similar fabricated structures. The proof test should be supplemental data equivalent to that required by ASME, Section 8, Division 2.

Verification of design strength (proof test)—A proof test shall be performed on all pressure vessels, hard structure, and penetration fittings that are subject to external pressure. Hard spots (that is, areas of discontinuity) should be instrumented to establish stress levels. The test fluid should be representative of the expected environment and at a temperature equal to or colder than the coldest expected operating temperature. The test time should be sufficient to demonstrate that the combined stress/temperature/time loadings are not critical. For Groups 1 and 2 materials, the proof test pressure (for submersible hulls—between 1.10 and 1.25 times the design operating pressure) should not

produce permanent deformation in the structure or prevent the operation of a penetrating component. For Group 3 materials, a technically justifiable hydro proof test is in order. For example, a conventional proof test would not be appropriate for materials whose performance would not be reproducible in such tests. Guidelines for testing pressure vessels and hard structure which are of a size and shape not suitable for pressure chamber testing may be provided on a case basis.

Stress analysis—The designer should perform a complete stress analysis of the hull and hard structure and demonstrate that its fatigue life is adequate for the intended use period. It is also expected that static stress levels will be limited to values such as those given below by the collapse strength requirements discussed in Section 5.3.2. The following stress levels are given as guidelines for Groups 1 and 2 materials; with the exception of peak stresses, the designer is not required to demonstrate compliance with these specific stress values.

1. The average shell membrane stress at design operating pressure should be limited to approximately two thirds of the minimum specified yield strength of the material.

2. The highest valued combination of average shell membrane stress and bending stress (excluding effects of local stress concentrations, for example, small fillet radii) at design operating pressure should be limited to approximately three quarters of the minimum specified yield strength of the material. The effect of all external loads and transitions and stiffener-to-shell connections should be considered.

3. The maximum peak stress at any point in the hull, including effects of local stress concentrations, shall be limited by fatigue considerations as discussed in Fabrication Design Criteria (in reference [2]). In addition, to ensure an adequate fatigue resistance for some designs using Groups 1 and 2 materials, it may be necessary to reduce the level of combined membrane and bending stress to a level below three quarters of the minimum specified yield strength of the material.

4. The designer should calculate stresses in the pressure hull by means of any recognized stress formulas or proven computer programs. A computer program should be proven by demonstrating agreement with experimental test results or manually obtained predictions. The validity of the stress-analysis methods used should be demonstrated by experimental results and a prior experience with similar structures.

5. For hulls constructed of Group 3 materials, the foregoing design requirements and guidelines may not be appropriate or adequate. Therefore, the designer is not limited to use of the above design basis. The design basis used, however, must be comprehensive and provide at least the same degree of conservatism as the design basis for Groups 1 and 2 materials.

5.3.4 Design Strength Parameters of Internally Loaded Pressure Hulls

In general, the designer should demonstrate the structural integrity or the pressure vessel and hard structure under loading conditions (environment, loading rate, and duration) representative of those expected in service. As such, the design operating pressure (analytical and experimental) must take into consideration the effects of temperature, cycle loading, creep, ductility, and anisotropy. Examples of pertinent variables include pressure, temperature, cycle loading, in-service material reproducibility, fabrication flaws and defects, vessel openings and intersections of different shells of revolution and attendant reinforcement or reinforcements, residual fabrication stresses, and deviations from the nominal geometry (flat spots, mismatch, tilt, out-of roundness, out-of-planeness, and out-of-sphericity). The following requirements should be met in order for this objective to be accomplished.

Inelastic stability (yield)—For stable vessel structures (that is, stiffened or unstiffened shells which permit the level of load-induced membrane stresses to approach the material yield point), the ratio of the yield pressure to the design operating pressure should be justified. In the determination of the pressure at which yield occurs, all fabrication and design-induced restraint and geometrical variables should be considered and their effect or effects included since the strength of moderately stable vessels can be detrimentally affected by such variables (that is, yield can occur at actual levels of membrane stress below calculated levels of yield point intensity).

Verification of calculated design operating pressure (proof test)—For Groups 1 and 2 materials, the calculated design operating pressure should be verified by destructive model testing or use of existing nondestructive tests. There are three alternative methods which can be used to satisfy this method of verification.

1. Where comparable vessel geometries and identical materials have been successfully tested to a pressure greater than the maximum operating pressure of the structure, use of this test method can be substituted for destructive model testing of the structure under review.

2. For new designs that do not fall within the parameters described in the previous paragraph, the calculated maximum allowable pressure may be verified by performing representative destructive model tests, either full or reduced scale. Where such testing is performed, the structural model should be sufficiently large to contain representative prototype ge-

ometries and material properties as well as fabrication restraints, tolerances, and residual stress.

3. For as-built vessel designs, which by refined calculation have a yield pressure in excess of the design operating pressure, full-scale testing of the vessel structure to 1.5 times the maximum operating pressure may be acceptable as verification of the calculations.

For Group 3 materials, the calculated yield pressure should be verified by destructive model tests. A destructive model test is one performed on a model whose scale is such that the mechanical properties of the material either are identical to or differ by a known factor of magnitude from those in the full-scale vessel, built from the same material, and which incorporates all structural details of the full-scale vessel. A conventional nondestructive proof test may not be appropriate for materials whose performance would not be reproducible in such tests.

For any material, the test method should duplicate the loading conditions expected in service and, where applicable, should be such that the mode of failure is identifiable. Where appropriate, use may be made of prior experimental results obtained on similar fabricated structures. The proof test should be supplemental data equivalent to that required by ASME Section VIII, Division 2.

Stress analysis—The designer should perform a complete stress analysis of the vessel and hard structure and demonstrate that its fatigue life is adequate for the intended period. It is expected that static stress levels will be limited to values below the yield strength requirements discussed herein. The following stress levels are given as guidelines for Groups 1 and 2 materials, and, with the exception of peak stresses, the designer should demonstrate compliance with these specific stress values.

1. The average shell membrane stress at design operating pressure should be limited to approximately two thirds of the minimum specified yield strength of the material.

2. The highest valued combination of average shell membrane stress and bending stress (excluding effects of local stress concentrations, for example, small fillet radii) at design operating pressure should be limited to the minimum specified for the material. The effect of all external loads and transitions and stiffener-to-shell connections should be considered.

3. The maximum peak stress at any point in the hull, including effects of local stress concentrations, should be limited by fatigue considerations. In addition, to ensure an adequate fatigue resistance for some designs using Groups 1 and 2 materials, it may be necessary to reduce the level of combined membrane and bonding stress to a level below that given in the previous paragraph.

4. The designer should calculate stresses in the pressure vessel by means of any recognized stress formulas or proven computer programs. A computer program should be proven by demonstrating agreement with experimental test results or manually obtained predictions. The validity of the stress-analysis methods used should be demonstrated by experimental results or prior experience with similar structures.

5. For vessels constructed of Group 3 materials, because the foregoing design requirements and guidelines might not be appropriate or adequate, the designer is not limited to using the above design basis. The design basis that is used, however, should be as comprehensive and provide at least the same degree of conservatism as the design basis for Groups 1 and 2 materials.

5.4 Design of Piping Systems

Piping systems should be designed to withstand, without failure, all anticipated service loading such as weight of pipe, fittings, valves, and contained fluid; internal or external pressure, static and cyclic; deflections and rotations of structure and equipment at points of pipe attachment; restraint of hangers and support; thermal expansion and transients; and shock, vibration, and water hammer.

Piping designed to carry over 3448 kPa (500 psi) and piping which, if ruptured, would present a safety hazard, should be protected against damage by suitable routing or shielding. Systems adjacent to lines designed to carry over 3448 kPa (500 psi) should be protected by suitable shielding or routing at a safe distance. Lines should be routed so they cannot be used as handholds. Lines should be made of one piece, whenever possible, within assembly, disassembly, and maintenance restrictions. Connectors should be designed and arranged so that it is physically impossible to connect a system of one pressure level to one of another pressure level. If gas reservoirs such as cylinders are included in a system, a valve should be provided to stop gas flow from the reservoir. The valve must be able to withstand full reservoir pressure, plus a sufficient safety factor.

High-pressure lines and fittings made of special material should be identified to prevent replacement with a part made of unsuitable material.

Piping and tubing should be supported as close to bends as possible and should not, themselves, be used to support components such as valves, relief valves, check valves, and filters. One line should not be used to support another. Clamp blocks may be used to support two or more adjacent lines with the clamp blocks attached to suitable structural members.

The number of pipe joints should be kept to a minimum. The designer must justify the selection of the type of joint used. The justification includes such information as previous similar service experience, nondestructive tests, and developmental testing.

All pipe joints should be accessible for inspection. Antiseize material should be used on all male pipe thread fittings. Oxygen compatibility must be observed.

Piping should be designed to have sufficient flexibility to prevent failures such as

- failure of the system from overstressing materials or anchors,
- leakage of joints,
- distortion of connected equipment and structures which exceeds the limits specified for them or renders them inoperable, and
- stress corrosion failure.

The structural adequacy of the piping system should be demonstrated by the designer for all anticipated service loadings. This may be accomplished by meeting the design requirements for internally pressurized components, as outlined in Section 5.4.2, Design of Piping Components, or by use of established American Industrial standards where the designer can show that a substantial body of service experience exists in similar applications. In addition, the designer, where feasible, should show that the fatigue life of the piping system is adequate by performing a fatigue analysis as outlined herein.

5.4.1 *Piping Flexibility Analysis*

It is desirable that piping flexibility calculations be performed by the designer as a further measure of assurance. Detailed sketches of piping under examination may be submitted with calculations and form a part of the calculation report. Calculations should show maximum stresses and their location in each section of piping under examination. Calculations should be prepared in a detailed form that will permit their review without difficulty and should delineate

- theoretical basis of calculations,
- method of performing calculations,
- simplifying assumptions,
- sign conventions,
- assumed material and dimensional data, and
- other pertinent information such as hull deflections.

The piping flexibility analysis may consider the added flexibility of piping components. Flexibility factors should be in accordance with the USAS B 31.1 Code for pressure piping. Piping components for which there are no flexibility factors listed in USAS B 31.1 should be considered rigid unless the designer can justify the use of added flexibility.

The designer's piping flexibility analysis should include calculations of the bending moments, twisting moments, and reaction forces imposed on each component in the piping system.

All lines should be identified and labeled to indicate function, content, maximum pressure, and direction of flow preferably according to MIL-STD-1247, Markings, Functions, and Hazard Designations of Hose, Pipe, and Tube Lines for Aircraft, Missile and Space Systems, except for color coding. There are some situations where such standardization is considered important for continuity and personnel training. The color coding of MIL-STD-101 Color Code for Pipelines and for Compressed Gas Cylinders is recommended for submersibles design requirements except as noted in Table 1.

Table 1 Color coding in submersible design

Gas System	Designation	Color Paint
Helium	He	buff
Oxygen	O	green
Helium-oxygen mix	He-O	buff and green
Nitrogen	N	light gray
Exhaust	E	silver
Air (low or high pressure)	ALP, AHP	black
Chilled water	CW	blue and white
Hot water	HP	red and white
Potable water	PW	blue

The use of military standards is preferred to other standards that could be used for system identification. At times the standard invoked for a support ship and for a supported submersible may conflict. For example, a color code standard specifying orange for use in helium-oxygen applications is not compatible with a color code standard specifying orange for use in hydraulic oil applications. The designer must ensure the personal safety compatibility of existing standards at the intended use site.

5.4.2 *Design of Piping Components*

It is desirable that the designer perform a stress analysis of each piping system component design. This analysis should consider all anticipated loading conditions including the loads calculated in the piping flexibility analysis discussed above. Applicable requirements are given below.

1. The structural design requirements used for externally pressurized components should not be less than the requirements for the pressure hull and hard structure.
2. For internally pressurized components of Groups 1 and 2 materials, the following structural design basis should be used:
   a. The allowable operating stress, *Sm*, should be taken as the lesser of two thirds of the minimum specified yield strength or one third of the minimum specified tensile strength for ferrous materials and the lesser of two thirds of the minimum yield strength or one quarter of the minimum specified tensile strength for nonferrous materials.
   b. General membrane stress should not exceed *Sm*.

c. Local membrane stress must not exceed $1.5Sm$.
d. The highest valued combination of membrane stress and primary bending stress must not exceed $1.5Sm$.
e. The highest valued combination of primary and secondary stresses must not exceed $3Sm$.
f. Peak stresses, including effects of local stress concentrations, must be limited by fatigue considerations, as discussed in Section B.3, Fabrication Design Criteria, in reference [2].
g. The various categories of stress which should be considered in the designer's stress analyses are defined below:
   (1) Primary stress: A primary stress is one which is required to produce a state of equilibrium with the applied loads. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses are further classified as follows:

   General membrane stress: Average value of primary stress across a solid section, excluding discontinuities and concentrations. An example of a general membrane stress is the average stress across the thickness of a pressurized cylindrical shell.

   Local membrane stress: Average primary stress across any thickness at a particular location on the pressurized component. This includes the effect of discontinuities but not stress concentrations. Also, it is limited in extent to a relatively small area of the components. An example of a local membrane stress is the membrane stress in a shell produced by external load or moment at a permanent support or nozzle connection.

   Bending stress: Primary stress at a particular location on the pressure hull which is proportional to the applied or induced moments and is proportional to distance from the centroid of a solid section. The effects of discontinuities and concentrations are excluded. An example of a primary bending stress is the stress in the central portion of a flat head due to pressure.

   (2) Secondary stress: A secondary stress is one which is developed by the constraint of adjacent parts of structure and which is not required to produce equilibrium with the applied loads. Secondary stresses are self-relieving in nature and are necessary to satisfy continuity of the structure. Examples of secondary stress are the discontinuity bending stresses at a head-to-shell or shell-to-flange junction.
   (3) Peak stress: The maximum combined stress at any point in the structure. The peak stress at a given point will be the maximum combined primary plus secondary stress suitably increased to account for local stress concentrations. The basic characteristics of a peak stress are that it does not cause any noticeable distortion and is of concern primarily as a possible source of a fatigue crack or brittle fracture. An example of a peak stress is the maximum stress at a notch or other small discontinuity in a structure.

3. For components whose geometry is not amenable to analytical evaluation, the structural adequacy of piping system components may be verified experimentally when considered appropriate by the designer. In this case, a comprehensive experimental stress analysis or a burst test such as specified in the ASME Boiler and Pressure Vessel Code, Section VIII, or both may be performed in lieu of an analytical stress analysis.
4. For components constructed of Group 3 materials, the designer is not required to use the structural design basis outlined above. However, as in the case of pressure hulls constructed of Group 3 materials, the designer must show that the design basis used is at least as inclusive and conservative as the basis discussed above. Further, the structural adequacy of components constructed of Group 3 materials must be verified by tests of full-scale prototype components.
5. All piping components should be designed or selected for flow that is adequate for the mission conditions expected for the submersible. These conditions should be specified in justifying the design or selection. All manually operated piping components should be readily accessible and easily operated under mission conditions.
6. In order for interference with gas flow to be minimized, flow-control piping components must be capable of controlling gas flow within the required design limits and should be designed with as few moving parts as possible.

7. If the flow-control component is adjustable, it should provide a smooth, even flow charge as it is operated. If orifices are used to control flow, the orifice size should be easily identifiable. Provision should be made for clearing or bypassing clogged orifices during use.
8. Exhaust components may be either adjustable or nonadjustable.
9. A check or nonreturn piping component is to be provided in the gas system when one-way flow is required for the safety of the submersible personnel or for normal operation of the equipment.
10. Maintenance and inspection procedures for nonreturn components should be clearly stated to ensure proper operation at low-pressure differentials.
11. Mechanical friction or bite-type connections should not be used on piping components subject to internal or external operating pressure, unless the designer can furnish sufficient information to demonstrate that the integrity of the connection will be preserved under service conditions. This must include consideration of reuse of the fittings and possible work-hardening of the associated tubing.

5.4.3 Piping System Design Verification

Piping systems, either gaseous or liquid, located inside the submersible should successfully pass a hydrostatic or pneumatic strength test at a pressure equal to at least 1.5 times the design operating pressure to assure system strength. The pressure should be maintained for a period of time sufficient to permit a thorough inspection of all points of suspected stress raisers (for example, rapid changes in section or sharp discontinuities) and brazed, welded, or mechanical joints, or some combination of these joints. There should be no evidence of distortion or leakage except that leakage which may result from loose or faulty seals in mechanical joints. Following this test, the pressure should be reduced to 1.1 times the design pressure and the tightness of all mechanical joints checked. If the tests indicate a leak, it should be located and eliminated unless a special leakage acceptance criterion can be justified by the system designed which will not affect the safety of the occupants.

Subsequent tests for those same systems, assuming that they do not undergo a configuration change and that component replacements are made in kind, should be to 1.1 times the design operating pressure for tightness. Should a change in materials, components, or configuration be required, that portion which has been changed should be tested to 1.5 times the operating pressure for strength and then 1.1 times the operating pressure for tightness. The remaining portions should only require pneumatic or hydrostatic tests to 1.1 times the design operating pressure for tightness.

Piping systems or portions thereof, located external to the submersible, should be subjected to tests for strength and tightness at least twice:

1. With an external pressure equal to the design test depth and an internal pressure at one atmosphere, there should be no indication of leakage from outside the system to the internal portion of the system. Pressure should be maintained for a period of time sufficient to permit a thorough inspection of all accessible joints and test leak-off points. There should be no evidence of distortion or leakage.

2. With the external pressure at one atmosphere and the internal pressure equal to at least 1.5 times the system design operating pressure, strength and tightness should be tested in accordance with the criteria in Section B.2.4, Fabrication Design Criteria, of reference [2]. Burst discs or pressure-relief valves that are required by the various codes to prevent rupture if installed on the submersible must not be permitted to present a hazard to the operator or occupant by causing rapid decompression should they be actuated by a malfunction or an extreme in pressure.

5.5 Design of Penetration Fittings

Design calculation and analyses should be conducted to establish the adequacy of all piping, mechanical, and electrical penetration fittings of the pressure hull and hard structure. The fittings must be adequate for the design life and the intended service conditions. Possible modes of failure including leakage should be considered. When internal pressure is a loading, the requirements of Section 5.3.4 apply. The requirements of Section 5.3.3 should apply to those portions of the fittings subjected to external submergence pressure. Sufficient testing must be performed to confirm the structural adequacy and leak tightness of each of these fittings.

5.5.1 Penetrations—General Requirements

All readily accessible vents, drains, exits, or exhaust from the pressurized volume, which would not be in use during the period that the submersible is pressurized, should be provided with covers or stoppers so that the number of potential, accidental exhaust paths may be kept to a minimum. The designer should assure that all penetrations are adequately sealed.

The body of electrical pressure vessel penetrators and connectors that may be exposed to sea water or spray should be corrosion resistant. Electrical penetrations of the pressure-containing structure must

be gas-tight even in the event of damage to the connecting cables.

Electrical cable pressure vessel penetrators should provide a water barrier at the hull to prevent flooding of the submersible in the event of failure of the external cable. Stuffing-tube-type penetrations of the pressure capsule or other hard structure are not generally acceptable. Pin-type connections for cable entrances into compensated enclosures are preferred; however, terminal tube entrances are acceptable provided there is evidence of compatibility of the cable jacket and insulation with the compensating medium.

Provisions should be made to protect the pressure vessel from corrosion in the gasketed areas of the penetrators.

Placement of pressure vessel penetrations should be such that minimum flooding or loss of atmosphere or both will occur in the event of a broken line. Emergency shutoffs also should be provided for such a contingency.

5.5.2 Viewports

The designer must show that the viewport design is adequate for the purpose. Each port's field of view within the submersible should be shown on drawings. The materials used must have adequate fatigue strength for the stresses to be imposed upon them during their specified service life. Resistance to stresses applied continuously over a long period, as well as cyclic stresses, must be properly considered. Full specification must be made of the materials used; composition; thermal, chemical, or physical treatment required; dimensions and tolerances; and renewal or replacement criteria. Where necessary, ports must be protected against accidental blows or other mechanical abuse. Although the use of various materials for viewports is not restricted, acrylic plastic viewports have a high level of success. For additional design information, refer to Pressure Vessels for Human Occupancy ANSI/ASME PVH01-1987.

5.5.3 Hatches and Closures

The designer should show that all closures, including hatches for personnel or materials, port covers (deadlights), and caps or plugs for openings of other types, have adequate strength for their application. The method of sealing must be specified in sufficient detail to permit evaluation. Ease and rapidity with which a closure can be closed or opened from inside or outside and whether or not tools are required are design considerations which vary according to the application. In the case of any closure for which these factors are important, attention should be drawn to them and the appropriateness of the design explained.

Hatch hinging, closing, locking, and sealing elements must be made resistant to or must be protected either from abuse due to rough handling or from possible accidents such as blows from a personnel transfer capsule which is being attached to the submersible hurriedly in rough weather. Design of the hatches must permit safe operation under specified ship motions.

5.5.4 Seals

Sealing materials and techniques must be shown to be adequate for the range of pressures, gas mixtures, temperatures, vibrations, and atmospheric environments specified for the chamber. Seals should not be subject to failure due to the effects of a nonlethal extinguishable fire inside or outside the chamber, attack by the fire extinguishing agent used in the chamber, or thermal shock caused by its application. The effect of pressure cycling, stress concentrations, differential thermal expansion, differences in modulus of elasticity, and aging should be considered. Tolerances must be considered to prevent seal failure (for example, nibbling and extrusion).

5.6 Detail Design/Fabrication/Construction Techniques for Personnel Safety

In addition to understanding the basic design parameters described in Section 5, it is important to recognize that they must be judiciously applied to the detailed design, fabrication and construction criteria described in the following subsections.

5.6.1 Piping System Joints

The designer should justify the selection of the type of joint design and method of joining used. The number of piping joints should be kept at a minimum. The justification should include such information as previous similar service experience, nondestructive test requirements specified to ensure that constructed joints meet design requirements, and developmental testing to qualify the method of fabrication. General considerations are:

1. Lines which are required to be disconnected during normal operation of the submersible or during normal overhauls or inspections should be provided with suitable closure devices for each exposed connection to prevent ingress of foreign materials when the system integrity is broken. Caps which introduce moisture and tapes that leave adhesive deposits should not be used for this purpose. Closure devices should be stored, when not in use, in such a manner as not to become contaminated.
2. Flexible hose is a poor substitute for rigid tubing. It is to be considered only where flexibility or vibration dictates its use and where rigid tubing is neither acceptable nor practical. When flexible hose

is to be subjected to considerable vibration or flexing, sufficient slack should be provided to avoid mechanical loading. Sharp bends or twisting should be avoided. A pressure safety factor of four should be the minimum used with flexible hose.

3. All flexible hose connectors should be designed so they cannot be connected into the wrong system. This can be done by size selection, key fit, or other means. Coding or identification alone is not generally considered sufficient.

4. Where hand-operated water hoses are used, they should be electrically conductive and grounded to reduce the risk of electric shock to the user in case they should be used inadvertently on live electrical equipment.

Mechanical friction or bite-type connections—Mechanical friction or bite-type connections should not be used in applications subject to internal or external submergence pressure unless the designer can furnish sufficient information to justify the integrity of the connection under the service conditions. This should include consideration of reuse of the fittings and possible work-hardening of the associated tubing.

Threaded fasteners—The design of all threaded fasteners including bolts, studs, and nuts should meet the requirements of the National Bureau of Standards Handbook H-28 or a comparable standard for metric sizes. Studs and nuts should be of sufficient length so that nuts, when tightened, expose at least 1½ excess threads.

Locking devices for mechanical fasteners—The need for locking devices for mechanical fasteners should be evaluated by the designer. A locking device that meets the following functional requirements is generally considered satisfactory:

- provides a positive locking action,
- is relatively simple to install,
- has a design which lends itself to inspection that will ensure proper installation without disturbing the locking feature, and
- is not normally reused.

If the designer selects locking devices that do not meet the above criteria, such as blind self-locking hardware or reusable insert-type fasteners, he or she should furnish sufficient information to justify the safety and integrity of the connection under the service conditions. The justification should include recommended inspection procedures and applicable acceptance standards. Should locking devices not be practical, critical fasteners should be marked with a "torque stripe" which identifies relative locations of parts when properly torqued for ease of inspection.

For further information on fastening devices, see reference [3].

5.6.2 Finishes

Finishes applied to pressure-containing elements should not be of a type likely to permit the development of hidden pitting. Metal applied as a surface finish, coating, or cladding should be lower on the electrochemical scale than the metal to which it is applied. The use of dissimilar metals in contact, where dissimilar metals are defined by Military Standard 889, Dissimilar Metals, should in each case be justified by the designer as not presenting an unacceptable corrosion situation.

5.6.3 Fracture Toughness

All plates, parts, and components should demonstrate adequate strength and toughness at the design operating temperature. Toughness characteristics of ferrous materials should be referenced to the Nil Ductility Temperature (NDT).

5.7 General Parameters for Design/Fabrication of Life-Support Systems

The overall life-support system should include those items required to provide a safe environment for all personnel utilizing the submersible. The designer should provide a detailed design description of the life-support system. The design description should provide justification, as outlined below, as to the life-support system capability to provide required services consistent with design operational requirements of the submersible.

The justification should include consideration of the duration of operation, number of personnel embarked on the submersible, physical activity of personnel, volume of personnel enclosure, and ambient environment.

The amount of redundancy required in the life-support system should be based on the duration of the operation. As an example, a submersible that is to operate without a support ship should have sufficient duplication of life-support services so that, if a failure of a life-support service occurred, the submersible still could return the personnel to a safe haven.

The designer should furnish data on actual operating characteristics obtained by test and on reliability of the life-support system. These data may be based on either previous history of use or on the results of life test data.

The design of life-support systems should consider the following services necessary for atmospheric control and monitoring:

- breathing mixture supply system,
- carbon dioxide removal system,
- hydrogen, carbon monoxide, and toxin removal systems,
- air purification and filtering system,

- atmospheric monitoring systems, and
- emergency breathing supply system.

5.7.1 *Breathing Mixture Supply System*

The breathing mixture supply system capacity, as a minimum, should be sized on the basis of providing sufficient oxygen to meet the human oxygen and air consumption rates.

The volume of breathing gas (air) needed for a submersible at one atmosphere may be determined by using Fig. 1 and by taking into consideration the total of all aspects of submersible personnel activity.

The normal surface oxygen control range should be 18 percent to 21 percent (0.18 to 0.21 atm partial pressure). Alarm set points should be 17 percent to 23 percent (0.17 to 0.23 atm partial pressure). Under no circumstances should the oxygen concentration fall below 16 percent (0.16 atm partial pressure). If the alarm limits are exceeded, immediate corrective action should be taken. Because of fire-prevention considerations, if the concentration of oxygen ever exceeds 25 percent (0.25 atm partial pressure) or if the corrective action fails to return the oxygen concentration to within the limits, the submersible should return to the surface and ventilate.

5.7.2 *Carbon Dioxide Removal System*

The carbon dioxide removal system should be sized in accordance with the carbon dioxide generation rate. The design of the system involves consideration of the criteria below.

The production of carbon dioxide closely follows the consumption of oxygen, with respiration accounting for most of the carbon dioxide generated in a submersible. During the breathing cycle, oxygen is removed from the breathing gas and replaced in a certain proportion by carbon dioxide. The proportion of the volume of carbon dioxide released to the volume of oxygen consumed varies from 0.7 to 1.1, depending upon the chemical composition of the energy source, level of exertion, and other factors. The average resting oxygen consumption[2] is approximately 0.44 slpm (1.0 scfh) per man, and the corresponding average carbon dioxide generation rate has been observed to lie between 0.352 and 0.374 slpm (0.80 and 0.85 scfh) depending upon dietary and other considerations. This is equivalent to 46 g (0.1 lb)/man-hour. The ratio of the volume of carbon dioxide produced to volume of oxygen con-

[2]Consumption may be expressed in either standard liters per minute (slpm) or standard cubic feet per hour (scfh).

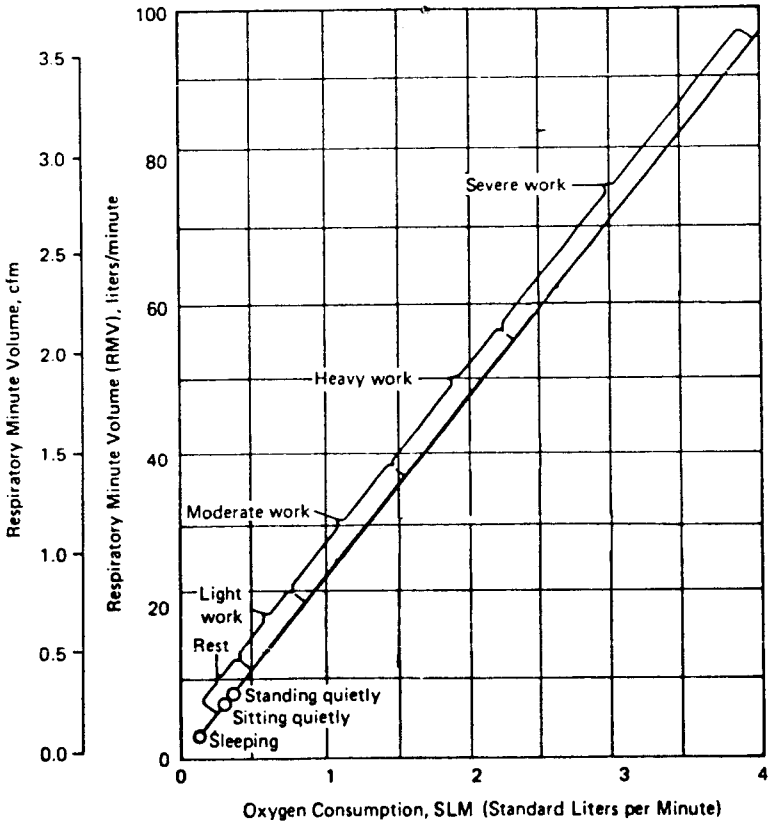


Fig. 1 Relation of respiratory minute volume and oxygen consumption to type and level of exertion [2]

sumed is termed the respiratory quotient (RQ). Therefore,

$$RQ = \frac{\text{Volume of } CO_2 \text{ produced}}{\text{Volume of } O_2 \text{ consumed}} = \frac{0.85}{1} = 0.85$$

Without removal equipment, carbon dioxide concentration will increase in accordance with the formula

$$\%CO_2 = 0.03 + \frac{RQ \times \dot{V}_{O_2} \times T}{V/N}$$

where

T = time in hours,
V/N = floodable volume per man,
$\dot{V}_{O_2}$ = oxygen consumption rate,
RQ = respiratory quotient, and
0.03 = percent of carbon dioxide in "clean" air.

If carbon dioxide is included in an inhaled gas, the partial pressure of carbon dioxide in the blood increases, and the respiratory center in the brain increases ventilation (tidal volume or breathing rate or both) in an attempt to restore blood carbon dioxide to normal levels. Excessive amounts of carbon dioxide in the breathing gas result in toxic effects, the severity of which depend upon exposure time and the partial pressure of carbon dioxide.

Figure 2 shows the relation of physiological effects of carbon dioxide for different concentrations and exposure periods. In Zone I, no perceptible physiological effects have been observed. In Zone II, small threshold hearing losses have been found and there is a perceptible doubling in depth of respiration. In Zone III, the zone of distracting discomfort, the symptoms are mental depression, headache, dizziness, nausea, air hunger, and decrease in visual discrimination. Zone IV represents marked physical distress leading to dizziness and stupor, with inability to take steps for self-preservation. The final state is unconsciousness.

The bar graph at the right of Fig. 2, for exposures of 40 days, shows that concentrations of carbon dioxide in air of less than 0.005 atm partial pressure (Zone A) cause no biochemical or other effects, concentrations between 0.005 and 0.03 atm partial pressure (Zone B) cause adaptive biochemical changes, which may be considered a mild physiological strain, and concentrations above 0.03 atm partial pressure (Zone C) cause pathological changes in basic physiological functions.

It is recommended that the short-time exposure limit of carbon dioxide be 0.015 atm partial pressure, with 0.020 atm partial pressure indicating an urgent need for correction. Long-term carbon dioxide exposure limits should be 0.005 atm (0.5 percent surface equivalent).

5.7.3 *Emergency Breathing Supply System*

The emergency breathing supply system should be used only in an emergency and not in normal operations. The system must be separate and distinct from other breathing supply systems. The system should be capable of providing proper breathing mixtures in the event of fire or failure of the normal system. The system should provide for voice communication and provide acceptable breathing (exhalation) resistance and comfort. All materials used in this system should be fire- and heat-resistant and their use justifiable. The capacity of the system should be sufficient to allow time either for normal ascent from operating pressure/depth plus at least a 50 percent margin, or

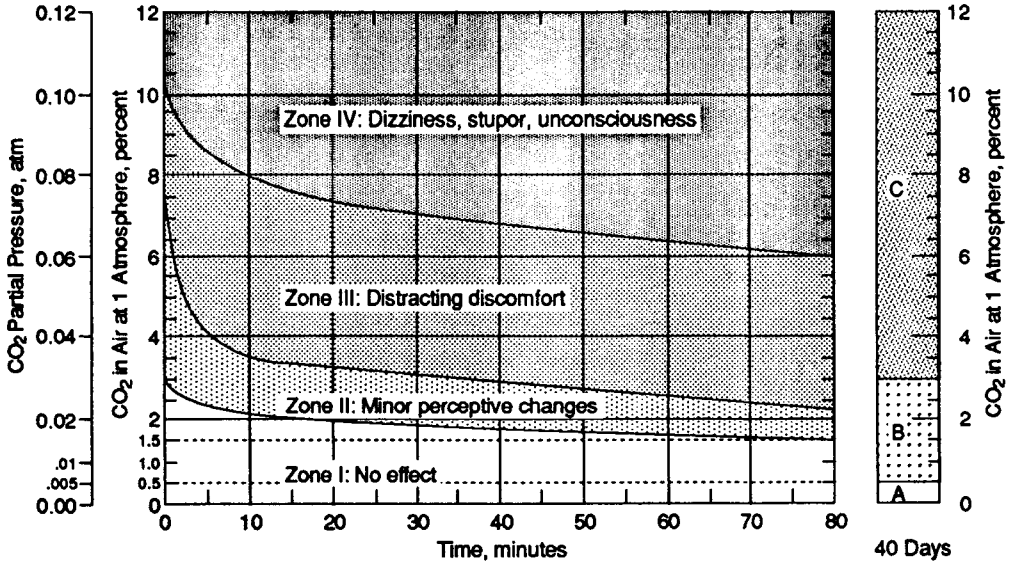


Fig. 2 Relation of physiological effects to carbon dioxide concentration and exposure period

for such other time that will permit salvage and rescue measures to be taken.

5.7.4 *Atmosphere Monitoring Instrumentation*

Atmospheric monitoring instrumentation should be provided to warn operators and passengers of unsafe atmospheric conditions. Suitable redundancy of instrumentation should be provided and should be particularly designed to operate in the event of normal power failure. Monitor features should include:

1. Oxygen sensors should be equipped with high- and low-limit alarms, and carbon dioxide sensors should be provided with high-level alarms. Alarms preferably should be both visual and audible.
2. Automatic transfer to reserve breathing gas supply, if provided, should actuate a warning signal, preferably both audible and visual. Provision may be made for turning off the warning device after it has been activated. Consideration should be given to providing automatic reactivation of the warning after a specified time.
3. Checking the status of the emergency supply should be required as part of the predive procedure.

5.7.5 *Breathing Gas System Components*

Breathing gas bottles are normally used to provide a breathing gas replenishment supply. They are used in the normal or emergency gas supply or both. When bottled breathing gas is installed, it is important to consider the following:

1. Assure that the normal supply container or containers will provide sufficient gas for the longest mission expected.
2. Assure that the emergency gas supply container or containers will provide sufficient gas for the possible emergency conditions.
3. Assure that each pressure-reducing device is designed to maintain the outlet pressure within required limits despite varying inlet pressure. When set to the required outlet pressure, regulation should not be markedly affected by position, motion, activity, or environmental temperature. Regulation should be shown to be satisfactory within the design operating depths. The regulating valve should be designed to fail shut.
4. There should be an isolation valve downstream of the regulating valve capable of withstanding full bottle storage pressure should the regulating valve fail. The isolation valve must be capable of manually regulating pressure. A bypass valve around the regulating valve should also be included in the event that the regulating valve fails shut.
5. There should be high-pressure oxygen piping of nonferrous material preferably of nickel-copper (monel) (MIL-T-1368) seamless, annealed tubing. High-pressure portions of the oxygen system should be welded wherever possible for maximum integrity of this vital and potentially dangerous system.
6. There should be suitable valve protection to prevent inadvertent depressurization of the system. For example, isolation valves should be provided for gages, regulating valves, or double-valve protection of fill lines for oxygen flasks that are permanently installed and recharged in place; double-valve protection should be provided on fill lines to prevent inadvertent depressurization of the flasks.
7. Oxygen quality meeting MIL-O-27210 or equivalent should be verified as to purity by the supplier.
8. Breathing gas systems should be designed so that they can be cleaned. The method specified for cleaning breathing gas systems must be capable of removing particulate matter as well as grease, oil, and other contaminants. It is important that no toxic, outgassing, or combustible residues remain after the cleaning operations and that the cleaning operations not cause degradation of the breathing gas system components.
9. Bottles should be secured to prevent their accidental detachment and supported or restrained so that they do not impose a load on other components that are not specifically designed as supports.
10. There should be assurance that the instantaneous release of the stored gas in the largest single container which cannot be isolated will not result in an increased pressure or concentration that will then result in an immediate and severe danger to the submersible personnel.

Carbon-dioxide absorbers: The carbon-dioxide level must be controlled in the breathing gas that is recirculated. This control generally is effected by passing the expired gas through a canister that contains a carbon-dioxide absorbent material. The carbon-dioxide absorbers must maintain the carbon-dioxide content of breathing gas below a specified maximum value for the duration of the longest mission expected, plus a specified safety factor (see Section 5.7.2). Carbon-dioxide canisters should be easily checked for absorbent content and condition, and should be easy to refill or replace.

5.7.6 *Toxic and Flammable Material Design Considerations*

The submersible occupants, having to create their own atmosphere, cannot tolerate toxic and flammable material which would upset the closely controlled environment or present a fire hazard. Every material that goes inside the pressure hull of a submersible must be carefully scrutinized to ensure that it does not offer a potential to generate a toxic environment or present a fire hazard.

Toxic materials—Materials such as paints, insulation, adhesives, fabrics, fittings, or other items and equipment containing material or components that

would give off noxious fumes at their operating temperature or at any temperature below 48.89°C ± 2.87°C (120°F ± 5°F) should not be installed or applied inside the pressure hull. For paint or adhesives, this requirement applies after drying or curing. Air sampling and analysis of the atmosphere for trace contaminants must be performed during simulated closed hatch operations to verify the above.

Mercury and beryllium and parts or components containing these metals should not be used in or introduced into the submersible's systems where that system could, during operation or due to a system fault, expose personnel to the toxic atmosphere that results from the presence of these metals.

Flammable materials—Every effort should be made to reduce or eliminate flammable materials. Flammable materials include materials in a form which will ignite or explode from an electric spark or from heating and which, if so ignited, will independently support combustion in the presence of air or in an oxygen-enriched atmosphere that might be encountered in the submersible under either normal or emergency conditions.

Magnesium and alloys containing magnesium should be avoided because of their combustibility. Also, materials or combinations of materials that may produce sparks when struck together should not be used. For instance, magnesium, aluminum, and their alloys can produce sparks when struck against rusty steel.

If aluminum paint is used inside a submersible, precautions should be taken to ensure that it is not applied over rusted steel. A primer containing neither red lead nor iron oxide should be used under the aluminum paint.

5.8 Design of Control Systems, Electrical Power Distribution Systems, and Lighting Systems

The following paragraphs cover the general design criteria applicable to control systems, electrical power distribution systems, and lighting systems with respect to their influence on submersible safety.

5.8.1 Control System

A control system is considered to be made up of three parts. The first part, the power supply, is the source of power from which the control derives its energy. The second part, the instrumentation, is composed of the devices used to measure the variable being controlled, the devices used to measure the response of the control system, the devices used to provide necessary information to determine when the control should be actuated, or some combination of these devices. The third part, control activators, is composed of the devices which accomplish the desired control function.

5.8.2 Power Supply

Control power supplies may be manual, mechanical, pneumatic, hydraulic, or electrical. The choice should be based on reliability in the environment in which the power supply must function. The designer should furnish information which substantiates the reliability of the power supply in the intended environment. This information may be based either on previous use of the power supply or on tests. Features that should be considered follow.

Electrical equipment should be protected against voltage variations in the power supply and against excessive flow of current when such occurrences present a hazard—direct or indirect.

All controls should have two independent sources of power. For purposes of discussion, these alternate sources of power are to be termed normal power and emergency power. Failure of one of the power sources should in no way hinder the use of the other power source. The emergency power source should be located so that it has as high a degree of survival as the personnel. In instances where redundant or backup electrical control circuits are provided to external equipment or safety devices, the circuits should be routed through different penetrators for maximum reliability. The emergency power should have, as a minimum, sufficient capacity to permit operation of systems and instrumentation to allow a safe ascent from maximum operation depth/pressure to the surface and to maintain the submersible at the surface. A load analysis should be furnished in sufficient detail to justify the power source capacity. This analysis should cover all possible combinations of power use.

5.8.3 Instrumentation Requirements

In a submersible, instrumentation requirements will vary widely depending upon depth and time. Generally, instrumentation includes sensors, warning devices and displays that are related to personnel safety. Instrumentation may include, but not be limited to:

- carbon-dioxide sensors,
- oxygen sensors,
- gas-flow sensors,
- electrical-defect detectors, and
- low-pressure warning devices.

General design considerations follow.

All instrumentation should provide response commensurate with the avoidance of hazardous conditions. The equipment should be electrically isolated from the personnel and should not be subject to erroneous readout.

All measuring devices used for control should have a backup means for providing measurements. Failure of one measuring device should in no way impair the use of the others.

5.8.4 *Control Actuators*

A control actuator is any device or group of devices used to accomplish a desired control function. From a safety standpoint, the basic design goal of any control system is reliability. In order for reliability to be obtained, the design should be such that the control actuator is not subject to false alarms or extraneous signals which produce undesired responses. Switches and controls that are used to energize manually a control actuator should be located so that they are not inadvertently energized. Features that should be considered in the selection of control actuators follow.

Control actuators should be designed to fail-safe. That is, failure of any portion of a control actuator should not in any way diminish the ability of the submersible to return to the surface. Indication of malfunction or failure in a control actuator should be relayed to the operator.

Individual control actuators should be capable of being isolated from other control actuators which share a common power supply. For electrical control actuators, this requires either fuses or circuit breakers on all lines connecting the control actuator to the power supply. For hydraulic or pneumatic control actuators, this requires an appropriate check valve or valve isolation on all lines connecting the power supply to the control actuator.

Where a control actuator energizes more than one device to accomplish the desired control function, each device should be capable of isolation from the remaining devices. For electrical devices, this requires circuit protection for both supply and return lines as well as a device that can manually disconnect the load. For hydraulic and pneumatic devices, this requires appropriate check valves or isolation valves on supply and return lines.

Where a control actuator normally operates automatically, provision should be made to allow the operator to manually control the actuator. The manual control should bypass as much of the control system as is practical.

5.8.5 *Electrical Power Distribution Systems*

Designed and installed with safety of the operators and passengers in mind, these systems should take into consideration the following attributes.

Ungrounded systems—All electrical power distribution systems should be ungrounded to minimize shock and fire hazards. Equipment connected to the power distribution systems should not ground the power distribution system. All equipment designed to employ chassis grounds should be isolated from the electric power system by a transformer or suitable isolation device. To minimize corrosion due to electrolysis, the hull must not be used as an electrical conductor or as a common reference. Ground detectors should be provided when possible.

Fault current protection—Fault current protective devices should be provided for each unit of electric-generating equipment or power supply and for each unit of power-consuming equipment connected to the distribution system. The design of the fault current protective devices should meet the basic requirements of section 21.57 of AIEE Standard No. 45, Recommended Practice for Electric Installation on Shipboard.

Cabling—Cable subject to deep submersible pressure should either be compensated or non-wicking (water blocking). In addition, cable runs for individual feeders should be selected so that the effect of any fire damage due to a fault will be minimized with respect to damaging other cabling in the distribution system. Cable and wire housed inboard should be arranged or protected to eliminate damage during normal personnel movement. During construction, installed cables should be protected against mechanical damage, burning by welders' torches, and contact with oils and solvents. All cables terminating in pressure-type connectors should have connectors applied in such a manner as to exclude all voids from the cable-connector assembly, to impose no undue mechanical stresses, and to minimize cable flexing during submersible transport and operations. A means of support should be provided at the cable/plug interface to minimize bending during operation. Sufficient slack should be left in the cable at penetrators to permit plug-in and disconnect operations without stressing the cable excessively. Additional considerations applicable to cabling follow.

Cables and wires should be protected against damage by accidental contact, crushing, or shearing, and being used as a handhold. Cables should be non-wicking and should be shown to be capable of withstanding the operating pressure plus a safety factor of approximately 125 percent.

Particular care should be taken in the case of portable electrical equipment to protect conductors from excessive flexure, kinking, tension, being caught between movable objects, or being stepped on. Conductors should be tolerant of such abuse.

Conductors within cables should be electrically insulated from each other and from the submersible. Electrical insulating materials should be selected on the basis of their ability to insulate the equipment and to provide the proper functional and mechanical characteristics. Also, electrical insulating materials should be nonflammable and nontoxic. Examples of electrical functional characteristics are dielectric strength, insulation resistance, and arc resistance. Mechanical properties include impact strength, elongation, flexibility, and adhesion and abrasion resis-

tance. Teflon-coated wire or kapton polyimide film over (FEP)-type insulation are generally preferred for these applications.

Connector design should permit the operator to disconnect the umbilical and any other electrical conductor readily without receiving an electrical shock. Electrical connectors should be sealed against water at operating pressure.

The materials and methods used to join connectors and fittings to umbilical cables and hoses should provide a strong bond capable of withstanding severe handling and operating conditions.

Connector pins and sockets should be corrosion-resistant or plated to prevent corrosion and electrical discontinuities.

Electrical connectors should be designed to prevent incorrect connection. This may be accomplished by size selection, key fit, or other means. Color coding or other visual identification alone is usually insufficient. Electrical connectors should also be designed to prevent accidental disconnect.

Electrical shock hazards—All electrically-powered equipment or enclosures should be adequately grounded to prevent shock hazards to personnel.

Environment—The environment for electrical components that are used on submersibles (for example, cabling, connectors, protective devices, motors, and power supplies) may differ markedly from normal shipboard conditions. The components may be oil immersed, be subject to full sea pressure, operate at low temperatures, or be subject to high vibration and high humidity.

Specific tests should be performed which justify the ability of the electrical component to function in the intended environment for its design life or over the design range of pressures, temperatures, humidities, voltages, etc. For flammability and toxic consideration, see Section 5.7.6.

Materials used in electrical equipment should be shown, by test or experience, to be resistant to deterioration by the environment, by migration of a plasticizer, or by the external application of heat.

Electrical penetrators and connectors—The body of electrical pressure-vessel penetrators and connectors exposed to sea water should be of a corrosion resistant material. Electrical cable pressure-vessel penetrators must provide a water-tight barrier at the hull to prevent flooding in the event of a failure of the external cable. Through-cable stuffing-tube-type penetrations of the pressure capsule or other hard structures are not generally acceptable. Pin-type connections for cable entrances into compensated enclosures are preferred; however, terminal tube entrances are acceptable provided there is evidence of compatibility of the cable jacket and insulation with the compensating medium.

Since the electrical hull penetrator is part of the primary pressure boundary, data should be provided to assure that its hydrocyclic life is defined in relation to the design life of the hull.

Provisions shall be made to protect the pressure vessel from corrosion in the gasketed areas of the penetrators.

5.8.6 Lighting Systems

The lighting system should have an alternate source of power, or emergency lamps should be provided. These lamps are backup in the event of total loss of the normal lighting system. The capacity and location of the alternate power supply and the wattage rating of the emergency lamps should be specified.

5.9 Operational Safety

There are several naval architectural requirements that should be considered for operational safety. These include static stability, both transverse and longitudinal; the conditions of list and trim; strength; and the dynamic considerations of stability and motion of the submersible in waves.

5.9.1 Stability

It should be demonstrated where applicable, both by calculation and by tests, that a submersible has adequate static and dynamic stability under the various loadings and conditions encompassed by the design (for example, surfaced, submerged, and all possible emergency surfacing conditions). The designer should also identify any limiting conditions for sea state, winds, temperatures, water-density variations, and so forth. Extreme loading conditions and the resulting stability should also be analyzed. For example, for those submersibles which jettison relatively large weights to achieve buoyancy in an emergency, there is an attendant risk that significant weights might be jettisoned inadvertently while in normal operations. It is important to show that there is adequate stability under these conditions. The criteria of adequate stability and reserve buoyancy with sufficient information pertaining to stability must be furnished to operating personnel to permit proper control of loading and to avoid danger of capsizing or foundering in heavy seas or swells.

5.9.2 Surfaced Conditions

There are periods when it may be necessary for the submersible to operate in a surfaced condition in the limiting sea-state and wind conditions. The designer should show that the submersible is safe for emergency egress under these conditions. It should also be demonstrated that there is adequate strength to withstand such factors as wave slap when the

submersible is in the surfaced condition. Protection should also be provided against foundering. The submersible should be designed to operate on the surface in the most severe sea conditions expected in its intended operating area.

5.9.3 Inclining Experiments and Trim Dives

If the submersible is of such complex geometry that reliable curves of form cannot be readily calculated, air, surface, and submerged inclining experiments must be performed to determine the stability in the surfaced and submerged modes. Precalculated form characteristics obviate the need for an air-inclining experiment; hence, the surfaced and submerged inclining experiments are sufficient to determine the stability surfaced and submerged when curves of form are available. If not too large, the longitudinal trimming moment could be determined by direct measurement or by a longitudinal inclining experiment. In addition, a trim dive is necessary to determine the proper weight and location of ballast, both permanent and variable, that will permit the vessel to operate under the design conditions of loading and in water of any density.

5.9.4 Damaged-Condition Information

The damaged-condition criterion should be identified early in the design phase. The ability to survive within the damaged condition criterion should be verified by calculation and testing, where possible. When at-sea testing of systems or components or both is impractical, simulated tests using appropriate test fixtures may be substituted. The means of detecting an abnormal situation or detecting a damaged condition should be evaluated. The corrective action intended by the designer should be implemented by written operating procedures. These actions should be demonstrated in simulated damaged conditions. The purpose of the demonstration is to verify the effectiveness, timeliness, and realism of the proposed corrective actions established by the designer.

5.9.5 Strength

It should be shown that the structural strength of the submersible in a seaway does not endanger the safety of personnel. Operations may be in close proximity to the ocean bottom. This means that there will be a hazard of striking objects, grounding, or even deliberate bottoming. There is also the hazard of bumping against or colliding with the surface support ship. The designer should either show that the submersible can withstand such incident or demonstrate that sufficient precautions can be taken to avoid such situations.

5.9.6 Rescue Features

The submersible should be provided with the necessary fittings and features that will enable lifting the vehicle for rescue or for recovery when flooded. As appropriate, a specific rescue plan should be included showing the intended means to accomplish rescue in the event of an accident while submerged. The limiting conditions for each of these operations should be included. It would be prudent to provide sufficient data to determine whether or not the submersible is compatible with existing U.S. Navy rescue and salvage equipment.

5.9.7 Stability and Buoyancy Systems

All systems and components affecting stability and buoyancy under operating conditions should be clearly identified. In addition, an analysis of the consequence of a failure or loss of displacement caused by loss of buoyancy components should be analyzed. Finally, the adequacy of specific materials and components should be justified as described under Section 5.2.

5.10 Waste Management

It is highly desirable that the designer provide for the storage, inactivation, and disposal of food and waste products, including metabolic waste.

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