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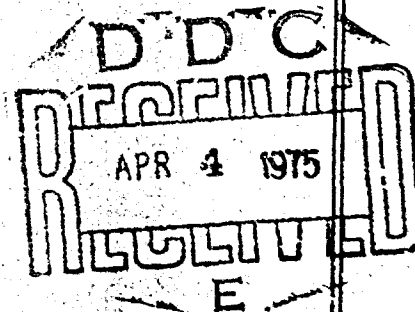
SUBMARINE SYSTEM FOR LOGISTICS AND REPLENISHMENT

by

Stephen T. W. Liang

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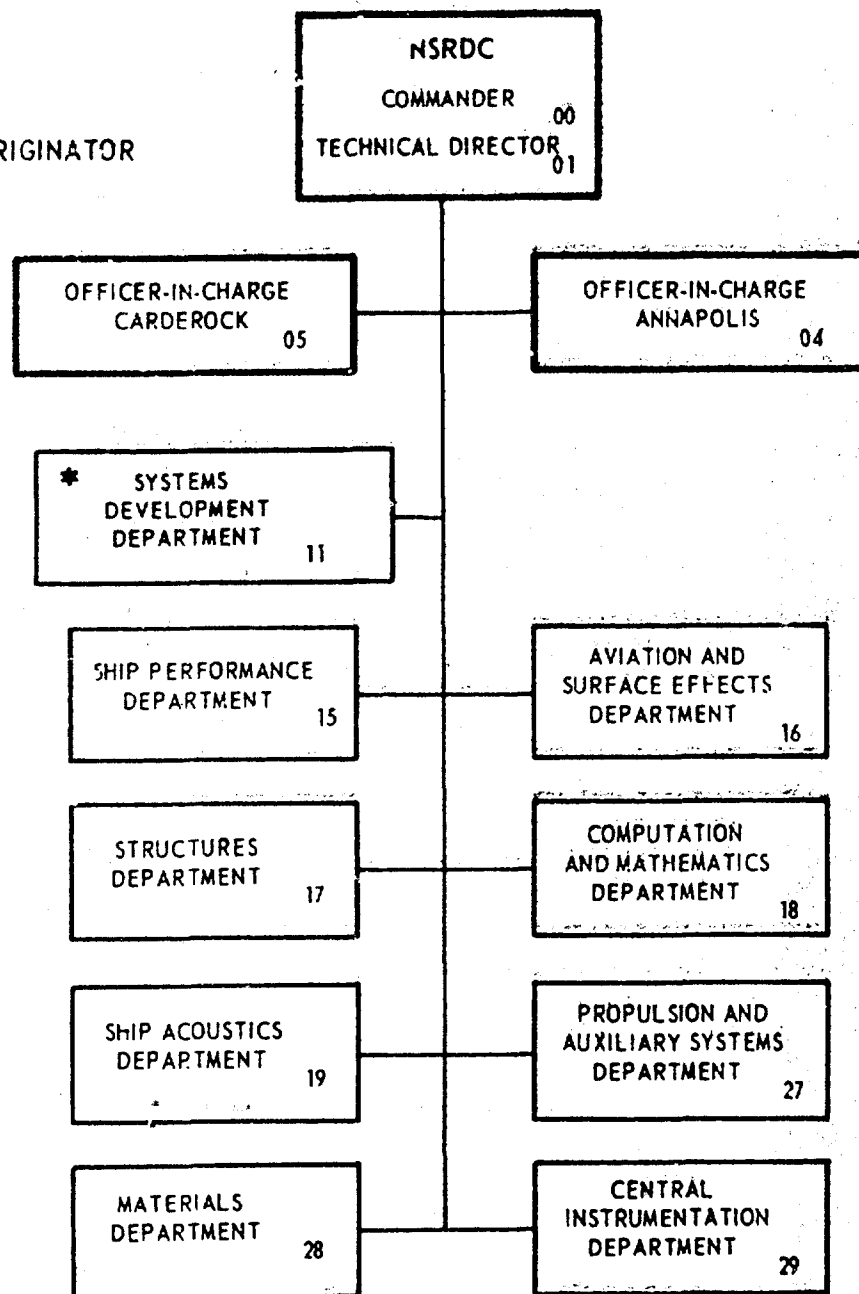
Report 4504

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER 4504	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) SUBMARINE SYSTEM FOR LOGISTICS AND REPLENISHMENT		5. TYPE OF REPORT & PERIOD COVERED
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Stephen T. W. Liang		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Ship Research and Development Center Bethesda, Maryland 20084		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS Task Area ZF61412001 Work Unit 1-1100-001
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Ship Research and Development Center Bethesda, Maryland 20084		12. REPORT DATE February 1975
		13. NUMBER OF PAGES 53
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report) Distribution limited to U. S. Government agencies only, Test and Evaluation Information. February 1975. Other requests for this document must be referred to Naval Ship Research and Development Center, Code 11.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number)		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) This report examines the technical feasibility of submarine alternatives to surface- borne cargo transport. Beginning with gross statements of potential missions and mission requirements, the study derives submarine concepts of transporting military loads which are assumed packaged in standard commercial containers. Submarines of conventional and tug-barge configurations are examined and estimates made of speed-power (Continued on reverse side)		

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performance near the surface, dynamic stability and control, surfaced motions and seaway loads. Several cross-sectional hull and structural geometries are examined for structural adequacy and weight. Possible weight allocations for a submarine cargo ship are presented.

Results of this preliminary feasibility investigation indicate that the concept of a cargo-carrying submarine may be technically feasible. It is found, however, that the transport efficiency of the submarine container cargo ship is significantly lower than that of a surface ship, and that unless other factors, such as attrition are considered, there is little incentive to consider submarine cargo transport.

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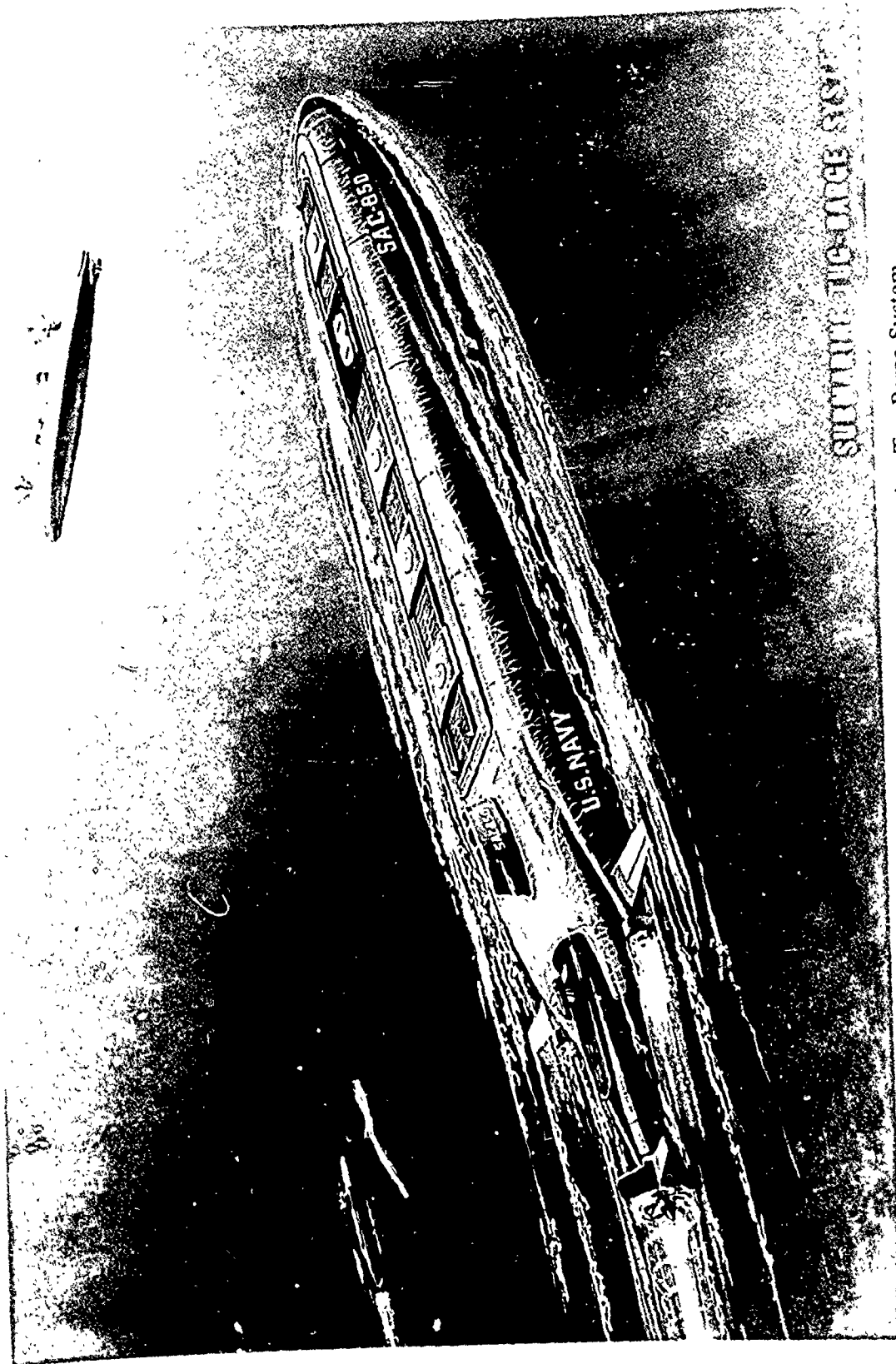
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Frontispiece - Artist's Concept of Submarine Tug-Barge System

ABSTRACT

This report examines the technical feasibility of submarine alternatives to surface-borne cargo transport. Beginning with gross statements of potential missions and mission requirements, the study derives submarine concepts of transporting military loads which are assumed packaged in standard commercial containers. Submarines of conventional and tug-barge configurations are examined and estimates made of speed-power performance near the surface, dynamic stability and control, surfaced motions and seaway loads. Several cross-sectional hull and structural geometries are examined for structural adequacy and weight. Possible weight allocations for a submarine cargo ship are presented.

Results of this preliminary feasibility investigation indicate that the concept of a cargo-carrying submarine may be technically feasible. It is found, however, that the transport efficiency of the submarine container cargo ship is significantly lower than that of a surface ship, and that unless other factors, such as attrition are considered, there is little incentive to consider submarine cargo transport.

ADMINISTRATIVE INFORMATION

This report describes the work accomplished during FY73 and FY74. Funding was provided by the in-house Independent Exploratory Development Program of the Naval Ship Research and Development Center (NSRDC) under Task Area ZF61412001, Work Unit 1-1100-001.

INTRODUCTION

The delivery of almost all logistic material from home bases to the Fleet, or to overseas sites, is presently accomplished by surface shipping. These ships have the advantage of greater economy of construction and operation than any other means of ocean cargo transport. They have the disadvantage of overt operation, and thus require control of the sea lanes.

In a naval conflict, large surface ships become easy targets for enemy submarines and aircraft. Losses in merchant ships will adversely affect movement of cargo by sea. It is conceivable that situations may arise in which there is a need for the transport of very high priority cargo over sea lanes which are contested. In such cases covert cargo transport may be desired, and a cargo-carrying submarine is one means of performing this mission.

The objective of the effort, reported here, was to explore the technical feasibility of such an option. Starting with a statement of potential requirements, exploratory concepts have been developed and analyzed to gain insight into various hydrodynamic, structural and design issues. The analysis includes data and findings related to propulsion near the surface.

stability and control, seaway loads when surfaced, structural concepts, structural strength and weight. Conclusions relative to major findings are included.

HISTORY AND BACKGROUND

GENERAL CARGO SUBMARINE

The concept of a logistic, cargo-carrying submarine is not new. In World War I, the German submarine freighter DEUTSCHLAND broke through the British sea blockade with a cargo of dyes, precious stone and mail, and proceeded to Baltimore, Maryland, U. S. A. She returned safely to Bremen, Germany, exactly three weeks after her departure from the United States bringing a return cargo of zinc, silver, copper and nickel. Both cargoes were composed of obviously high-priority items.

SUBMARINE TANKER

With the successful development of nuclear propulsion after World War II it became possible to consider submarines as noncombatant transport vehicles. Operating independently of sea states and weather conditions, and not limited by ice conditions, submarine cargo carriers could maintain year-round operations in areas presenting hazards to surface ships. In the past decade, many submarine tankers have been proposed by the marine industry;^{1,2} the characteristics of some of these are listed in Table 1.³

At present, a vigorous program is being pursued by the Maritime Administration to study the use of nuclear-powered submarine tankers to transport crude oil from the Alaskan oil fields to either the East Coast of the United States or European ports. It has been proposed that submarine tankers traverse under the polar ice cap and thus utilize a direct path and with no need for icebreakers.

TOWED SUBMARINE BARGE

Another concept of shipping liquid cargo is that of using an icebreaker towing a submarine barge at a controlled depth below an icefield. According to Continental Oil Company, this system has been shown to be technically feasible and also economically competitive with other proposed systems for Arctic transportation. A design for a 250,000-deadweight (dwt)

¹Russo, V. L. et al., "Submarine Tankers," SNAME Transactions, Vol. 68 (1960). A complete listing of references is given on page 45.

²Todd, F. H., "Submarine Cargo Ships and Tankers," Ship Division, National Physical Laboratory, Great Britain (Jan 1961).

³"Is There a Future for the Giant Submarine Oil Tanker?" The Naval Architect, Journal of the Institute of Naval Architecture, England (Oct 1971).

barge was completed as shown in Figure 1.⁴ A detailed development program leading to designing, building and operating a prototype barge with controls has been planned, and a scale model construction is under way.

SUBMARINE ORE CARRIER

All previous submarine cargo carrier designs have been for tankers, except the DEUTSCHLAND, which carried war materials in the form of general cargo, and another cargo submarine, an ore carrier, named MOBY DICK.⁵ The latter was proposed by Mitchell Engineering Ltd. and Saunders-Roe Ltd., England, more than a decade ago. Having a length of 604 feet with a diameter of 72 feet, it was designed to operate at a depth of 300 feet at a speed of 25 knots. With a displacement of 50,000 tons submerged, it would carry 28,000 tons of ore. Propulsion was to include a boiling water reactor.

The shape of the hull included a parallel midbody enclosing the cargo spaces, a feature adopted to reduce cost of construction as well as to reduce draft in the surfaced condition. A double hull construction was proposed for this part of the ship, with a single hull construction elsewhere. The cargo hull was not completely pressurized internally.

As can be seen from the cutaway view in Figure 2, the vessel is a self-unloading ore carrier, with one conveyer belt extending along the bottom of the ship beneath the hoppers, and a second taking the ore upwards through a hatch.

Note that in most cases of previous cargo submarine designs, operating depths are considerably less than for military submarines.

SUBMARINE CONTAINER SHIP

In the Navy some work on structures for underwater replenishment vessels was done by the Naval Civil Engineering Laboratory. The Naval Undersea Center has also done analyses of cargo-carrying submarines, primarily with reinforced concrete as the structural material. The effort by this Center is directed toward concepts and technology for the dry-cargo-carrying submarine in which the cargo is prepackaged in standard commercial or military containers. Submarines designed to these cargo loads pose new problems because container stowage requires consideration of pressure hull geometries configured for the containers, and because the range of cargo densities is greater than for bulk-carrying cargo submarines. The submarine must provide ballasting capability ranging from no cargo load to full load of containers, each fully loaded to maximum rated capacity.

⁴Sudbury, J. D., "Submerged Barges for Arctic Transportation?" Ocean Industry (Mar 1973).

⁵"Submarine Ore Carrier," Shipping World, England (Sep 1959).

MISSION REQUIREMENTS

GENERAL

As this effort was intended to examine broad issues of technical feasibility, there was little need to firmly establish specific requirements for the cargo submarine. Instead postulated missions and general requirements previously identified were accepted and used as a basis for technical explorations herein. Missions for logistics and replenishment operations are listed in Table 2.⁶

CARGO LOAD

In recent years, the use of standardized containers has come into practice for most dry cargo shipment. In the past decade, the rate of growth of container ships has been much greater than that of tankers or other cargo ships. As the record shows,⁷ in January 1973, the United States leads the world in container ships, with 31 percent of the total dead-weight tonnage of the world fleet.

The containerized shipping of cargo has many advantages over the conventional storage system: increased efficiency and lower transportation costs, such as handling (including stevedoring); simplified inventory and delivery; and reduced costs in breakage, pilferage, and insurance. Because of the economic attractiveness, the popularity of such a mode of transportation will probably continue to grow for some time to come. In view of this situation, it is likely that the naval supplies will be carried wholly or in part by containers, and that new systems will be required for the transport and handling of containers. Cargo containers are designed to meet the requirements of the American Society of Mechanical Engineers, USA Standard Specifications for Cargo Containers (USASI MH5.1 - 1965). The length of containers may vary from 20 to 40 feet, with end dimensions of 8 x 8 feet. Permissible container loadings are shown in Table 3.⁸ The 20- by 8- by 8-foot container has been assumed as standard for this study.

In practice the average loaded container weight will be less than the 20 long tons for the 20-foot container shown in Table 3. A conventional surface container ship, operating on a trade route, will transport empty containers along with loaded ones. Of the loaded containers there will be a large distribution of weights caused by large variations in cargo density,

⁶"Summary of Logistic and Replenishment Submarines," Southwest Research Institute, prepared for the Office of Naval Research (Jan 1971).

⁷"Maritime Reporter and Engineering News," Maritime Activity Reports, Inc. (Nov 1973).

⁸USA Standard Specifications for Cargo Containers (USASI MH5.1), ASME (1965).

and by incompletely stuffed containers. In establishing a cargo load for commercial (non-military) design, an average 20-foot container is expected to weigh 11-15 long tons. Military loads are expected to be greater.

Knowledge of the cargo characteristics is, of course, important in the case of surface containership design. Cargo weight will affect the form and dimensions of the ship, as well as powering and range. When the design is established, however, the surface ship offers considerable latitude in accommodating to both the numbers and weights of containers carried. This is not the case for the submarine container-carrying ship.

In a submarine container transport all containers are carried within the pressure hull, and the pressure hull envelope prescribes the maximum number of containers to be carried. The watertight envelope of the submarine establishes the submerged displacement, and to achieve submergence, fully loaded weight must equal submerged displacement. The sum of all variable weights, i.e., the difference between submerged displacement and fixed weight must be constant to meet the condition of neutral submerged buoyancy. Variable weights include cargo load, consumables and seawater ballast. Weight variations in cargo and consumables must be compensated by equal changes in ballast. As the cargo load can vary from zero to maximum capacity, the demands upon variable ballast are potentially great.

For purposes of this study the author has elected not to provide variable ballast tankage for the no-load case. It has been assumed that entire cargo compartments can be flooded to achieve the required ship weight and static stability. Instead, average cargo weight is assumed to fluctuate over a much narrower region, and variable ballast capacity is provided to accommodate this smaller fluctuation. For purposes of examining exploratory designs, military cargo containers have been assumed to weigh between 15 and 20 long tons.

The depth of the ship determines the number of tiers of containers that can be accommodated in the hold. Current container design practices are based on the assumption that they are stacked to a maximum of six high.⁹ In a submarine, since the containers cannot be stacked on deck as on a surface ship, an increase in this capacity will require the stacking of containers exceeding the limitation of six high. The question arises whether all containers should be reinforced or just a selected few. It is noted that for the larger submarine carrying containers seven high, as later shown in the case of TB-1, only about 10 percent of the total number of containers on the bottom rows will have to support more than five containers. The choice of reinforced containers for certain locations would thus appear to be the obvious solution.

⁹Darman, W. J. and D. J. deKoff, "Characteristics of Recent Large Container Ship Designs," Marine Technology (Oct 1971).

SHIP SYSTEM CONCEPTS

SHIP CONFIGURATIONS

As stated previously, the objective of this study was to explore the technical feasibility of a cargo-carrying submarine. The principal interest was to identify the critical problems rather than to design a specific ship. For this reason, a number of concepts have been considered such as the conventional submarine, modular submarine, and submarine tug-barge system.

From the technological point of view, a conventional large cargo submarine (Figure 3) has the most advantages. It is not so costly to build and is perhaps easier to operate and maintain. For the flexibility of interchanging cargo modules to suit their missions, the modular configuration (Figure 4) may be applicable.

The advantages of the tug-barge configuration (Figure 5) over the others are:

1. Less Vulnerability—the loss of the barge may not mean loss of the tug or vice versa.
2. Lower Unit Cost—the most costly part of the ship system would be the nuclear power plant. The tug-barge system needs fewer power plants than a conventional ship system. Subsequently, this will reduce the initial cost, as well as operational cost.
3. More Operating Days—both tug and barge can be operated separately or as an integrated unit. This will minimize the idle time in port and, in turn, increase the overall operational efficiency.

The disadvantages include the need for advanced technology in construction and for more skillful personnel for operation. These disadvantages raise the cost slightly above that of a large single submarine.

For the purpose of general exploration, three sizes of cargo carriers, configured as conventional submarines or as the tug-barge combinations, were investigated. The tug-barge configuration was selected for further analysis in various options because of its genetic similarity to others, in addition to its own unique characteristics. Table 4 compares the principal characteristics of the three systems with the tug-barge configurations. TB-1 and TB-2 are essentially the same in design, except for their capacities. TB-3 is also similar in general arrangement, but designed for a relatively shallow draft. A conventional large submarine would have essentially the same overall characteristics as those shown in the table. In all cases, the tug-barge systems operate as an integrated system where a pusher submarine (power tug) is linked rigidly with a much larger cargo module (barge); and they are operated together as a single unit. The following discussion will apply to either a conventional or the tug-barge system.

Since virtually no work has been done on the containerized underwater logistic system, this effort has focused on submarines to carry dry cargo, and specifically configured to handle

the 8- by 8- by 20-foot standard containers. As noted earlier, a 15-ton minimum gross weight or an average gross volume density of 85 cubic feet per ton is assumed. Maximum container weight is 20 tons. These are referred to as the maximum and minimum container weights in Table 4.

For missions which have no cargo on their return trips, one out of three cargo holds will have to be designed for containers as well as for ballast. These cargo holds can be used as seawater ballast tanks on their return missions. This no-load condition will be examined and results will be presented later, in Table 13.

The cargo module is essentially a modern underwater container ship. The relatively light density of containerized cargo together with a moderate depth of submergence results in significantly different criteria for structural design of the submarine pressure hull than has been used for combatant submarines. Usually, the designer's concern is with minimization of weight; but here the problem is one of increasing weight to achieve neutral buoyancy and submergence.

For combatants, in almost all cases it is desirable that the pressure hull have as high a strength-to-weight ratio as possible so that hull structural weight is at a minimum. If the usual circular pressure hull is considered for cargo carriers, the combination of the lightweight pressure hull and low-density cargo requires great quantities of ballast for submergence. In some cases the required ballast water cannot be provided, and the cargo submarine cannot be submerged.

An array of arrangements and material combinations has been investigated in order to compare the efficiencies of the ship configurations. The results indicate the square-with-rounded-corners cross-sectional pressure hull with mild steel and partial concrete construction is promising. The general arrangements of submarine tug-barges TB-1 (Figure 6) and TB-3 (Figure 7) are basically identical except TB-3 is designed for a much shallower surfaced draft. Both employ rectangular-with-rounded-corners cross sectional pressure hulls. With this arrangement, cargo capacities (volume) for TB-1 and TB-3 can be improved by 22 and 24 percent, respectively, relative to a circular hull form.

A special structural problem of the pressure hull is the requirement for large access hatches through which to move cargo containers. These large accesses will place a severe requirement on the structural design. However, it is believed that these hatches can be developed with technology in-hand, and no serious problems are anticipated.

SPEED, POWER AND SUBMERGENCE

The resistance of a submarine is generally made up of a frictional component and form drag contributions of hull and appendages. Deeply submerged submarines create no surface disturbance and therefore the wavemaking drag is negligible. This is not the case for a

submarine operating near the surface, however. In this instance, the extent of surface disturbance increases with decreasing submergence and the wavemaking contribution and total drag increases accordingly.

The container-carrying submarine is envisioned as operating near the surface, as there is little need to operate at deep depths. To establish a minimum reasonable operating depth, computations were performed to assess the wavemaking drag penalty for near-surface operation. The effective horsepower (EHP) associated with a particular form, speed and depth of submergence were predicted by methods in use at the Center,¹⁰ augmented by Pien's Wave-Making Resistance Computer Program.¹¹

For estimating the total shaft horsepower (SHP) required for propulsion, a propulsive coefficient of 0.70, an appendage allowance of 15 percent (for the bridge fairwater, rudders, stern and bow diving planes, flooding holes, etc.) and service margin of 10 percent are assumed.

For determining the adequate operating depth, the speed and power curves (Figures 8 and 9) of both submarine tug-barges TB-1 and TB-3 were developed (TB-2 has the same hull form as TB-1; therefore no power curve was developed). In both cases, it is found that the maximum operating depth of three hull diameters (to the top of the hull) from the free surface are needed to avoid most wavemaking resistance in ship speed up to 30 knots.

The general relationships of the speed, power and submergence in different hull forms are presented in Table 5. It should be noted that the squared round-corner form has superior transport efficiencies in all speeds, and the efficiency gap gets wider when it reaches the higher speed regime. However, the rectangular form does have a higher cargo capacity factor, and can be operated at shallower draft.

OTHER HYDRODYNAMIC ANALYSES

This section addresses the submerged stability, control, and maneuverability as well as surfaced motions and wave-induced sea loads of the proposed submarine systems. Due to the large size and unusual configuration of the ship, its hydrodynamic characteristics may become key determinants of the feasibility of the concept. For this reason, the TB-1 configuration was selected to be more thoroughly studied.¹² The pertinent results are presented here.

¹⁰Gertler, M., "The Prediction of the Effective Horsepower of Ships by Methods in Use at the David Taylor Model Basin," David Taylor Model Basin Report 576 (Dec 1947).

¹¹Chen, R., "The Pien's Wave-Making Resistance Computation Program," NSRDC Report 4370 (Mar 1974).

¹²Sheridan, D. J., "Analytical Evaluation of the Submerged Dynamic Stability and Control and of the Surfaced Motion's and Dynamic Loads for a Proposed Tug-Barge Submarine, TB-1," NSRDC Report 4379 (in preparation).

PHYSICAL CHARACTERISTICS

To provide the basic information used for the initial hydrodynamic analysis, only the plan and profile of TB-1 were roughly faired as shown in Figure 10. The pertinent geometric properties and weight distribution for the ship floating at full load draft of 70 feet were generated by a computer program.¹³ The results are summarized in Table 6.

The weight distribution curve (Figure 11) was developed by assuming that 91 percent of the barge weight was evenly distributed between Stations 2 and 15, 3 percent was forward of Station 2, and 6 percent was aft of Station 15. For the power tug itself, it was assumed that 45 percent of its weight is distributed aft the midship of the tug and 55 percent forward of the midship.

Estimates of appendage sizes and locations required to achieve acceptable submerged dynamic stability characteristics were made either by extrapolating available empirical data or by analytic techniques. For simplicity rectangular planform shapes and sectional shapes typical of submarine appendages were assumed. Their pertinent characteristics are listed in Table 7, and their principal dimensions are shown in Figures 12 and 13. It should be noted that the barge bridge fairwater was assumed to be 30 percent flapped so that the fairwater could be used as a rudder to assist in turning the submerged TB-1. No bridge fairwater was designed for the tug since such an appendage of reasonable size would contribute little to the dynamic stability of the whole system.

SUBMERGED DYNAMIC STABILITY

The dynamic stability characteristics of the TB-1 when deeply submerged were assessed on the basis of Bottaccini's stability criterion¹⁴ which measures the relative dynamic stability in both horizontal and vertical planes by an "index of stability," G .

The value of G is speed independent in the horizontal plane. In the vertical plane the value of G indicates the degree of dynamic stability at an infinite speed. If the submarine is unstable at infinite speed, the extent of the unstable region must be determined including the effects of the specific speed considered. This approach is considered valid since, if a ship is stable at infinite speed, it will be stable throughout its entire speed range.

A value of G close to 1.0 implies a very high degree of stability, while a value of G only slightly greater than zero indicates a marginal stability. Therefore, a ship must possess a reasonable positive value of G , if it is dynamically stable. The indexes of deeply submerged

¹³Sheridan, D. J. et al., "Manual of NSRDC Ship Motion and Sea Load Computer Program," NSRDC Report 3376 (in preparation).

¹⁴Bottaccini, M. R., "The Stability Coefficients of Standard Torpedoes," NAVORD Report 3346 (Jul 1954).

stability for TB-1 in various configurations have been calculated and summarized in Table 8. The results indicate that the fully appended configuration has good stability in both the horizontal and vertical planes.

It should be noted that operation of TB-1 in the submerged mode, but near the free surface, may result in a degradation of the controllability of the vehicle. Positive vertical stability seems to be quite adequate for the system, and at 300 feet, free surface effects should be relatively small.

TURNING

The submerged turning diameter for a steady flat turn in the horizontal plane, and the time to reach executed pitch angle in the vertical plane have been estimated using the methods of Gertler and Hagen.¹⁵

The results indicate that the TB-1 submarine will have a submerged turning diameter of about 3 shiplengths when both the barge and the tug rudders are set at 35 degrees deflection at speed range of 10–20 knots. No assessment of the surfaced turning diameter was made, but it is expected to be considerably greater than that of the submerged condition, especially when only the tug rudders would be available for producing turning forces.

For assessment of the effectiveness of control of vertical plane, two conditions with 15-degree sternplane deflection were made. First, the sternplanes of both the barge and the tug were deflected simultaneously, and second, the tug sternplanes alone were deflected. The results are presented in Figure 14.

It should be noted that the curve for applying the tug sternplane alone is not extended below 10 knots. This means that the TB-1 could have difficulty in attaining a pitch angle of 5 degrees below this speed, because the tug sternplanes cannot generate enough moment to overcome the metacentric restoring moment in this condition. If the barge sternplanes are used together with that of the tug, a similar situation exists for a speed of 4 knots or less.

It appears that the TB-1 can be better controlled by using tug and barge sternplanes together. Although it will be sluggish in its response, at its maximum submerged speed of 20 knots the TB-1 will require about 18 seconds to reach a pitch angle of 5 degrees. Such sluggishness can be attributed primarily to its large size.

SURFACED MOTIONS

The surfaced motions of the submarine tug-barge TB-1 and its sea loads induced by waves were predicted.¹³ They included the motions of heave, pitch, sway, roll and yaw as well as the vertical shear forces, bending, and torsional moments.

¹⁵Gertler, M. and C. Hagen, "Standard Equations of Motion of Submarine Simulation," NSRDC Report 2510 (Jun 1967).

Calculations were made for 3 headings, 2 ship speeds, and 15 wavelengths. The headings were 180 (head seas), 135 (bow seas), and 90 (beam seas) at speeds of 0 and 5 knots. The wavelengths were expressed in terms of wavelength to shiplength (λ/ℓ) ratio. The significant range of the ratio was considered from 0.8 to 2.2. All calculations were performed for a ratio of wave height to wavelength of 1/20. Critical maximum transfer functions for heave, sway, and pitch are presented in Figures 15 to 17. Motions for those headings, which were small and which provided little insight into the dynamic characteristics of TB-1, were omitted.

Heave motions (Figure 15) are most severe in beam seas with the submarine moving forward at a higher λ/ℓ ratio; and this same trend is also true at zero speed. However, heave motions in head seas at zero speed are considerably less than at 5 knots, until λ/ℓ becomes smaller than 1.0; then the trend is reversed.

Sway motions (Figure 16) are maximum in beam seas, but they are not affected by the speed of the ship in all headings.

Pitch motions (Figure 17) are most severe in head seas while the vessel is moving forward. They are diminished gradually as the headings change toward beam seas. The speed of the ship seems less a factor in pitch motions except where the λ/ℓ is higher than 1.0 and the headings approach the head seas.

Judging from the results given above, wavelength plays a very significant role in all motions induced by waves. It should be noted that for λ/ℓ below 1.0, the motions become almost independent of both heading and speed, except for some headings in heave.

Wavelength and steepness depend on wind speed and duration as well as geography. For example, the average length of storm waves varies from less than 300 feet in the Mediterranean to about 500 feet in the North Atlantic, and a little over 500 feet in the Pacific Ocean.¹⁶ Wave periods corresponding to these wavelengths range from 8 to 10 seconds.

According to the statistical survey¹⁷ of most shipping routes around the world, 95 percent of wave periods are between 5 1/2 and 11 1/2 seconds. For a submarine of 886 feet in length, the ratio of wavelength to shiplength (λ/ℓ) corresponding to these periods will be in the range of 0.2 to 0.8. The motions at this range, as shown in Figures 15 to 17, are substantially lower.

Roll motions are very large in the vicinity of the resonance point (at about $\lambda/\ell = 2.8$ for beam seas) as shown in Figure 18. However, for a submarine length of 886 feet, the waves would have to be on the order of 1900 to 2500 feet long to excite the vessel in the

¹⁶Kinsman, B., "Wind Waves," Prentice-Hall Publishing Corp. (1965).

¹⁷Hagben, N. and I. E. Lumb, "Ocean Wave Statistics," Great Britain National Physical Laboratory and Meteorology Office (1967).

resonance regime. The likelihood of experiencing such steep waves of this wavelength is quite remote (for comparison a Pierson-Moskowitz State 9 sea has an average wavelength of about 1313 feet). Roll motions in head seas are zero.

SEA LOADS

The wave-induced sea loads are presented in the form of nondimensional transfer functions. They include functions for vertical and horizontal forces, vertical and horizontal bending moments, and torsional moments for head, bow, and beam seas at speeds of 0 and 5 knots. For each condition, i.e., each heading and speed, only the wavelength at which the individual functions are maximum is plotted.

A typical plot of the longitudinal distribution of vertical loading is presented in Figure 19, which illustrates the most severe condition found. The maximum values of vertical loading transfer function are summarized in Table 9. The maximum bending moments for all headings generally occurred in the midship region (Stations 8 to 10). Shear forces and torsional moments are maximum in the vicinity of the tug-barge couplings (Stations 15 to 17 as expected).

STRUCTURAL FEASIBILITY

Considerable effort has been expended in conceptual and actual design of transport submarines over the past two decades. Most of the effort, however, has been directed toward the development of submarine tankers. Virtually no work has been done on structures for large dry-cargo submarines, except an ore-carrying submarine.⁵

For liquid cargo, densities are not greatly different from that of seawater; loading and unloading is relatively simple; and few limitations are placed on the shape of the cargo holds. For containerized dry cargo, all these conditions will be adversely different. The most critical of all is that the cargo must be contained in a pressure hull and the loaded ship must have sufficient weight and ballast to make it submersible. Another requirement for the container-carrying submarine is that large cargo hatches within the pressure hull itself are necessary, calling for a hull arrangement which would differ in design from a conventional modern submarine.

With the above in mind, structural feasibility for three variations of tug-barge systems, designated TB-1, TB-2 and TB-3 (Figures 6 and 7) were investigated.

PRESSURE HULL ARRANGEMENT

Because a higher volumetric efficiency is essential to the performance of a dry-cargo submarine, the noncircular pressure hull was chosen for all configurations instead of a

conventional circular one. The configurations of TB-1 and TB-2 are essentially identical in structure as well as arrangement except for the total number of cargo holds. TB-3 is designed for a relatively shallow draft; therefore a rectangular cross-section pressure hull was introduced.

Five structural configurations depicted in Figure 20 were analyzed. Configuration I represents the hull cross section where no cargo loading hatch opening is located. Such a section can be treated as a continuous ring; and it can be strengthened with a vertical support at the center of the section, as shown in Configuration II.

Configuration III shows the hull section which contains a hatch opening at its top center. If reinforcement is required, a vertical support can be added at the fore and aft boundaries of the hatch as in Configuration IV, or a longitudinal beam may be used on each side of the hatch as in Configuration V.

All hatches are oriented in a single row longitudinally at the top center of the cargo holds and their openings are assumed to be 10 by 22 feet for shipping/unshipping 8 by 8 by 20 foot containers.

STRENGTH ANALYSIS

The collapse depth for the submarine pressure hull was chosen as 1000 feet,* which corresponds to an external hydrostatic pressure of 445 psi. The design approach taken for these hulls differs from a conventional approach by nature of the noncircular hull shape and the presence of large hatches along the top-center of each hull. Because of this geometry, the hull structure will experience considerably more bending than a circular hull, and the structural members will tend to be more massive. High yield strength steels can effectively reduce the structural mass but will cause increased costs for the material and its fabrication. The structural design problem, then, is to establish configurations which use moderate strength materials that are easily fabricated.

For a conventional submarine, the design of a cylindrical hull consists principally of ensuring against three types of failure: circumferential yielding of the shell and/or frame which leads to collapse, local buckling of the shell between frames, and overall buckling of the shell-frame combination. Any circumferential bending, if present at all, is only incidental, arising from the out-of-roundness of the hull.

For noncircular hulls, such as those for TB-1, TB-2 and TB-3, however, substantial bending will occur for all loads, because of the marked deviation from circularity. The problem is to provide sufficient rigidity in the shell structure to resist this bending, since overall buckling is not considered to be a serious problem.

*Calculations were not made at this stage of concept feasibility to determine the speed/vertical plane control/depth envelope.

A method was developed for analyzing these bending moments and stresses. The same analysis was used to estimate the effect on hull strength of large openings in the hull for cargo access. A variety of configurations were treated; some use vertical diametrical hull stiffening, particularly in the vicinity of the access hatches.

The circumferential bending moments for each of the configurations of Figure 20 were calculated. Profiles of the unit circumferential bending moments along the perimeter of the typical hull sections for the TB-1, TB-2, and TB-3 are presented in Figures 21 and 22. A unit load of one pound per square inch was used in all computations. The structure was considered as homogeneous, with stiffness of the ring members averaged over the frame spacing.

A series of five internal-frame-stiffened shell designs was devised (Figure 23) for structural analysis; their physical properties are listed in Table 10.

The direct compressive stresses at selected locations on the pressure hull were calculated for the design pressure of 445 psi. The allowable bending stress was found by subtracting the direct stress from the yield stress of the selected material. Based on this allowable bending stress, the corresponding allowable unit bending moment was determined. Designs C and E appear to be the most satisfactory. Results of computations are listed in Table 11 and compared with the maximum unit circumferential bending moments shown in Figures 21 and 22. All designs are within the material allowable limit.

The required sizes of the vertical center-support columns were also investigated. The maximum total forces to be supported by these columns occur for Configuration IV, although the loads for Configuration II are nearly as high, and the unit forces are 76.0 psi per unit length of hull for TB-1 and TB-2 and 208.0 psi for TB-3. The supports were treated as pinned, end-loaded columns. The standard column-buckling formula was applied, assuming a safety factor of four, based on the design collapse pressure.

The required moments of inertia of the center column, per hull frame space, were determined for all designs. These results appear in Table 10. These values are much smaller than those of the shell-and-frame combinations for the hull. They should not present any design or weight problem.

The end-closure sections would probably have short transition hull sections, having approximately the same scantlings as the main hull, to reduce the diameter of the hull and if necessary, produce a circular opening. This smaller opening could then be covered either with a hemispherical or a flat closure, fabricated from reasonably thin material.

The strength required for the pressure hull to withstand pressure loads down to specified collapse depth would normally be expected to ensure sufficient surface operating capability. In view of the unusual size of cargo submarines, however, longitudinal stresses due to bending of the entire hull by wave action during surface operation have been investigated,¹² by using the method of wave-induced loading, discussed earlier. The maximum was found to be a

vertical bending moment near the midship, occurring in a head sea. The dynamic shear loads and torsional moments were generally in the vicinity of the tug-barge coupling. Stresses resulting from application of these loads and moments to the pressure hull design were calculated. TB-1 is 28,000 psi, well within the working stress value.

DESIGN EVALUATION

The moment plots of the designs of TB-1, TB-2, and TB-3 (Figures 21 and 22) show that the critical moments occur either near the center of the flat sections of the hull perimeters with negative sign (maximum compressive circumferential stress on the external shell surface) or near the centers of the rounded corner sections with positive sign (maximum compressive circumferential stress on the internal frame flange surface).

Design A (Figure 23) was developed to utilize steel with a yield strength of about 50,000 psi. The 6-inch thick plate material is, perhaps, unrealistic considering problems of fabrication and welding.

Design B is essentially the same as Design A, using thinner but higher-strength material. Its strength is satisfactory for all three ship systems TB-1, -2, and -3, if material with a yield strength of 100,000 psi is used. However, interframe shell buckling may be a problem, as was indicated by its shell-buckling parameter, shown in Table 10.

Design C calls for 2 1/2-inch material, but with smaller, more closely-spaced frames than Design B. It is adequate for TB-1 and TB-2 with material having a yield strength of 100,000 psi. It would also be adequate for TB-3 with 50,000 psi yield strength material if vertical diametral reinforcement is used throughout the hull (Configurations II and IV).

For Design D, the thickness of the shell is reduced to 2 inches and the frame dimensions are reduced slightly from those of Design C. This design would be adequate for TB-1 and TB-2 with 130,000 psi yield strength material and nearly adequate with 100,000 psi material. It is more than adequate for TB-3 with Configurations II and IV and 80,000 psi yield strength steel.

Finally, Design E is considered, having a smaller frame and using 2-inch thick steel throughout. This design would require material with a higher yield strength than 130,000 psi for TB-1 and TB-2, but would be satisfactory for TB-3 with 80,000 psi yield material.

Diametral reinforcing (Configurations II and IV) is necessary in all cases for the portions of the hulls containing loading hatches. The possibility of reinforcing the hatch areas of TB-1 and TB-2 with longitudinal beams (Configuration V) instead of vertical support may be possible, although a more refined analysis to validate the results is needed.

Structural failure resulting from stresses other than those considered so far may be the possibility of buckling in the flat sections and interframe shell. Buckling pressure cannot be predicted for these structures with any precision, but rough estimates can be made according

to Reynolds.¹⁸ The values of the shell buckling parameter, β , given in Table 10, indicate that circular shell portions for all except Design B have nearly sufficiently close frame spacing.

It is believed that construction of pressure hulls of TB-1 and TB-2 of 50,000 psi yield strength material would be economically impractical since material thicknesses of nearly 6 inches are required (in Design A). A pressure hull could be fabricated from 2 1/2-inch 100,000-psi-yield-strength steel. The calculations indicate that this would be feasible, utilizing Design C. Even if 2 1/2 inches is taken as the upper limit of material thickness, it appears, moreover, that the use of 80,000 psi material might be possible, as maximum circumferential stresses for Design C would only reach about two-thirds of yield strength at an operational depth of 500 feet.

For TB-3, the beneficial effects of diametral vertical support columns at the center of the hull, and running the full length of the hull (Configurations II and IV), are so great that the unsupported cases will not be considered. For the supported hull, Design E should be most satisfactory with 80,000-psi-yield-strength material, and Design C would be feasible with 50,000-psi material.

It is apparent that some additional ballast will be needed. It could be carried in a useful form as in concrete cargo floor-support or concrete frame-support as shown in Figure 24. Frames could be deeper, and made of thinner material, and some or all the space between them could be filled in with concrete to prevent local buckling of the frame components. The maximum available weight of these two types of ballast that could be accommodated with Designs C and E are given in Table 12. This is evidently more than is needed for neutral submerged buoyancy in every case, so that only a portion of available weight capacity would actually be used.

Rough estimates of the pressure-hull weights for representative designs and some weight-displacement ratios have also been calculated and appear in Table 12. For these calculations it has been assumed that weights and displacements of the end closure structures will be comparable to hemispherical heads. The pressure hull weight (Design C) of TB-1 has been incorporated in the estimated total weight of the barge (Table 13). It appears that the weight of such a design augmented by permanent ballast could make this ship system submersible.

CONCLUSIONS

The concept of a submarine system for logistics and replenishment, either in a conventional or tug-barge configuration, appears technically feasible insofar as it has been examined.

¹⁸Reynolds, T. E., "Elastic Buckling of Ring-Supported Cylindrical Shells Under Hydrostatic Pressure," David Taylor Model Basin Report 1721 (Jul 1966).

This conclusion is tempered by the belief that more thorough investigations would be required to fully establish the feasibility and practicality of some of the more detailed aspects of the concept. Matters which have not been investigated include container handling and transfer, joining of power unit and barge, watertight hatch closures and stability under varying loading conditions. Negative findings in these areas may invalidate the concept.

The geometry of a container-carrying submarine requires substantial departure from the conventional, circular pressure hull shape. The analysis indicates that, for shallow operation a square section with rounded corners is the preferred configuration. This cross section yields significantly increased volume capacity relative to the circular shape.

The high volumetric capacity produces a weight problem which is the reverse of that normally encountered. Even in fully-loaded condition the cargo submarine cannot submerge without the addition of large amounts of fixed ballast. For the configurations examined, the space required to locate the required ballast appears to be available. This insensitivity to weight allows much more latitude in selection of structural concepts. With the requirement for minimum weight structure removed, it may be possible to develop hull and structural configurations more suitable for cargo carriage and less costly to fabricate. The assumption of 15-20 tons per container has an important effect on the conclusions of this study. Lighter, average cargo loads increase the magnitude of the weight problem in a manner detrimental to the concept.

Except at very low speeds, large submarines are dynamically stable and controllable. Low speed control requires further study. Primarily because of size, however, submarine response to control surface settings will be sluggish.

The transport efficiency (defined as ton-miles of cargo per shaft horsepower) of a cargo submarine is less than half that of a comparable surface containership at a similar speed. This lower efficiency, coupled with higher ship acquisition and operations costs, makes the submarine option unattractive in normal circumstances. Considerations related to covert transport, or high attrition rates for surface operation might, however, change the balance in favor of the submarine cargo ship.

ACKNOWLEDGMENTS

The author is particularly indebted to Mr. R. V. Raetz of the Structures Department for his contribution on structures, especially for the development of a method for analyzing bending stress in a noncircular hull, and to Mr. D. J. Sheridan of the Ship Performance Department for his contribution on hydrodynamics. Special thanks are also expressed to Mr. R. M. Stevens and others who provided technical reviews and valuable comments to improve the contents of the report.

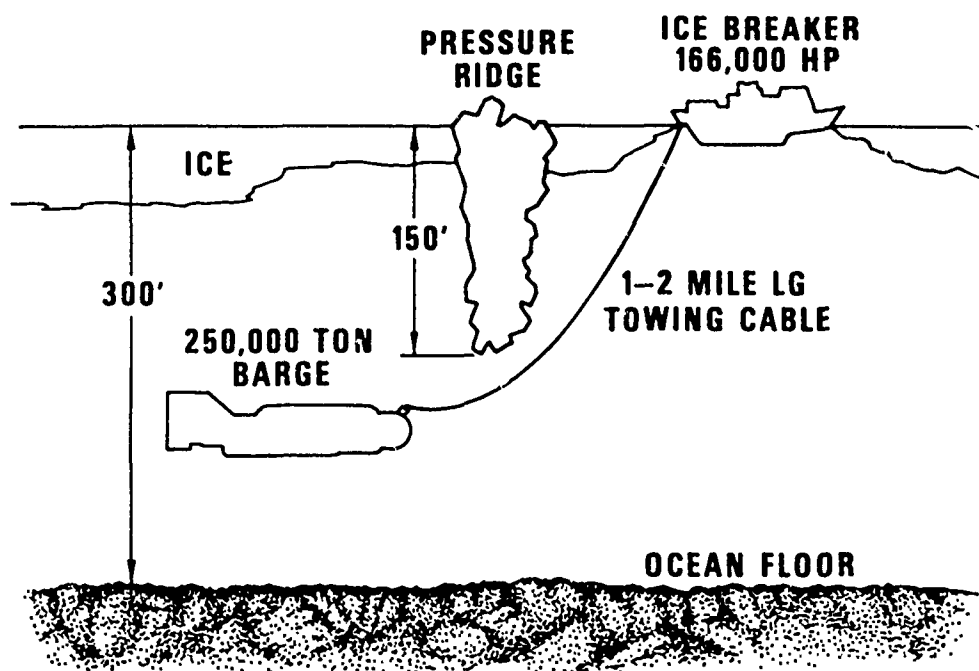


Figure 1 - Icebreaker Towing Barge at Controlled Depth under Ice Field

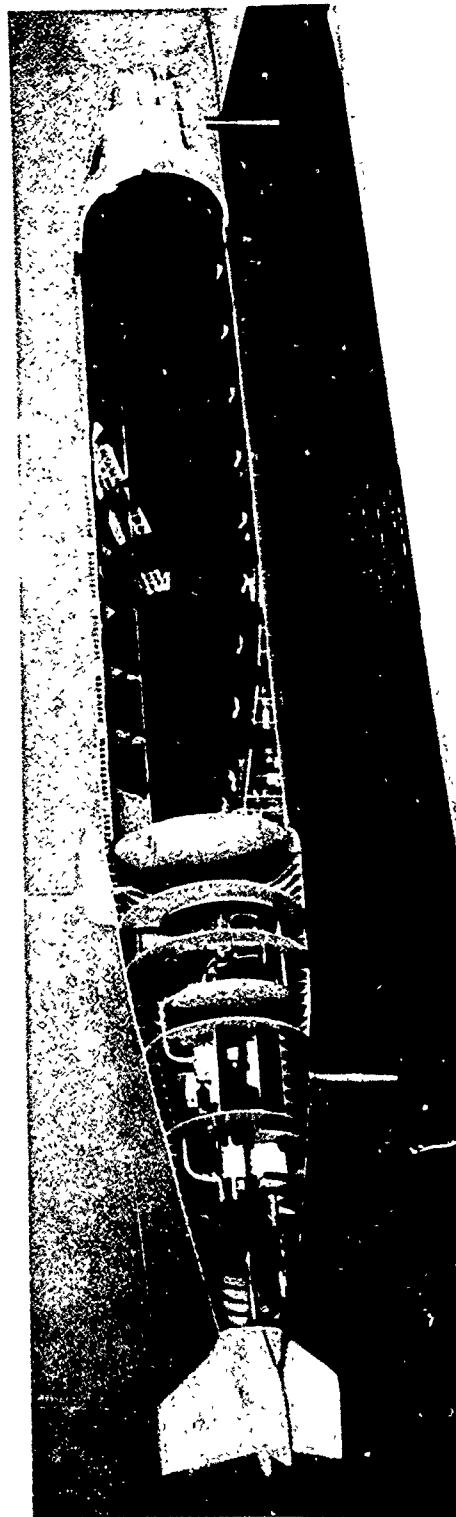
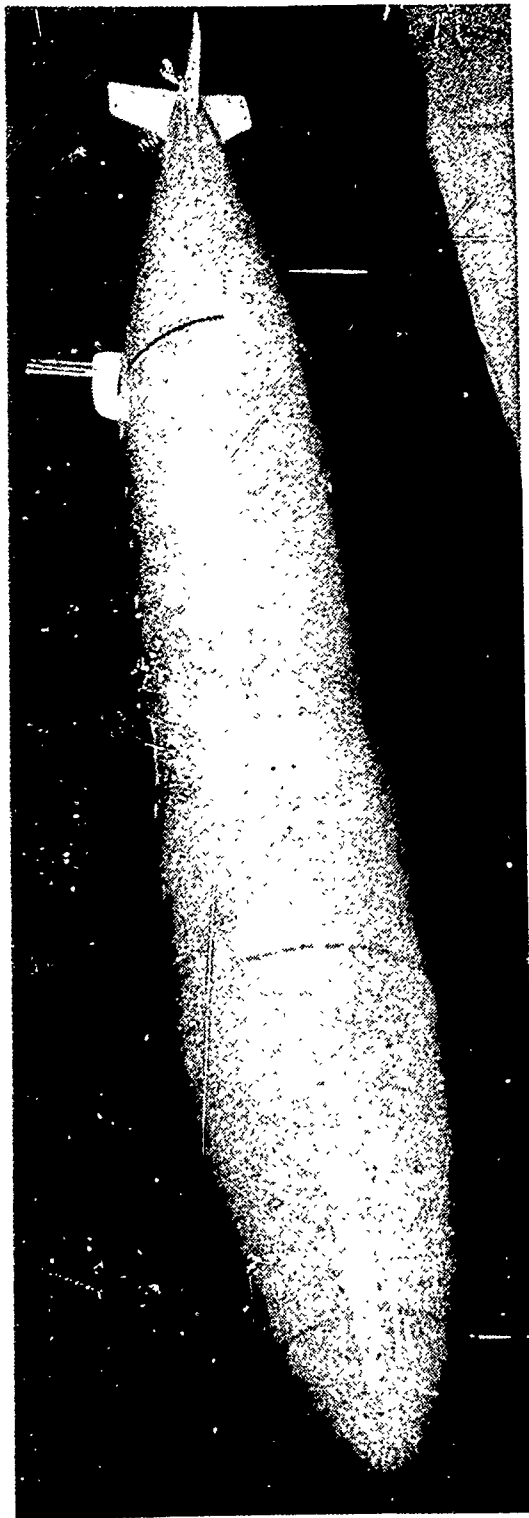


Figure 2 — Model of Submarine Ore Carrier
(This photo is taken directly from Reference 5)

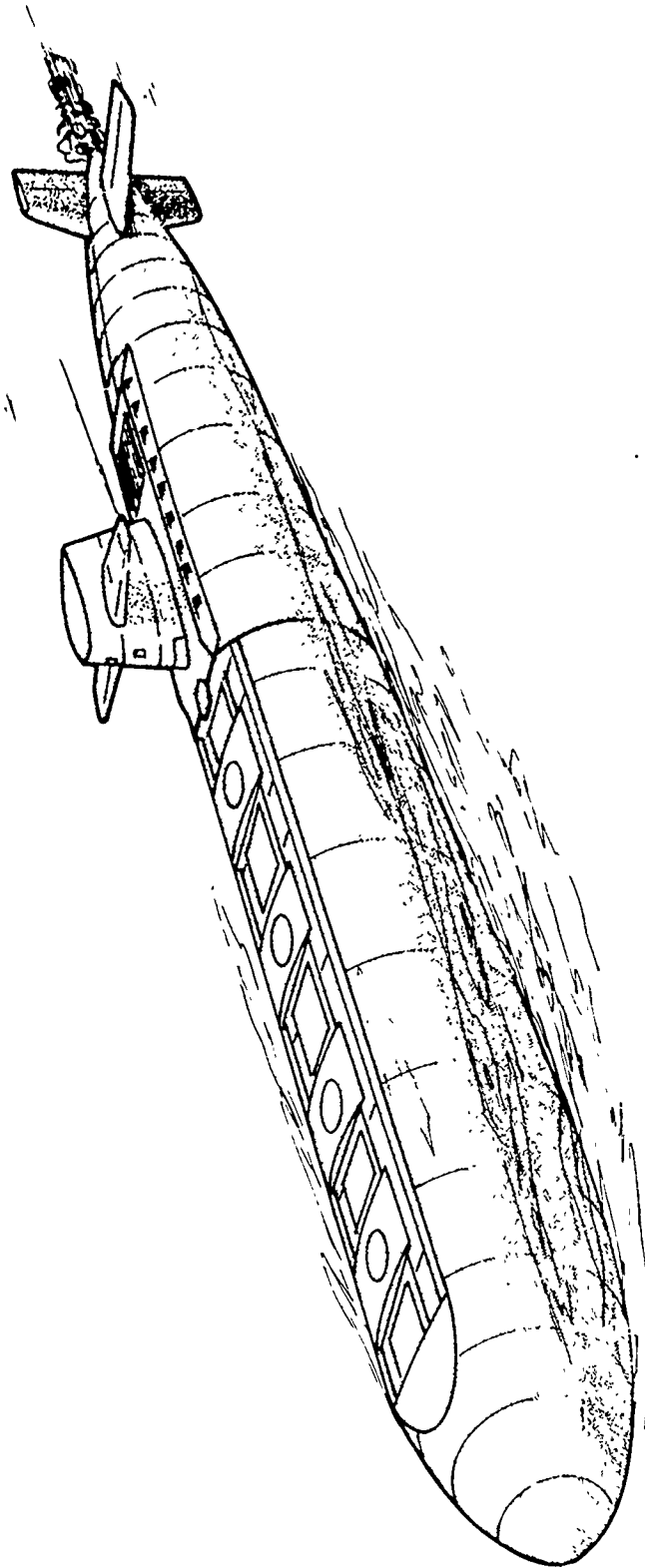


Figure 3 - Submarine Logistic System

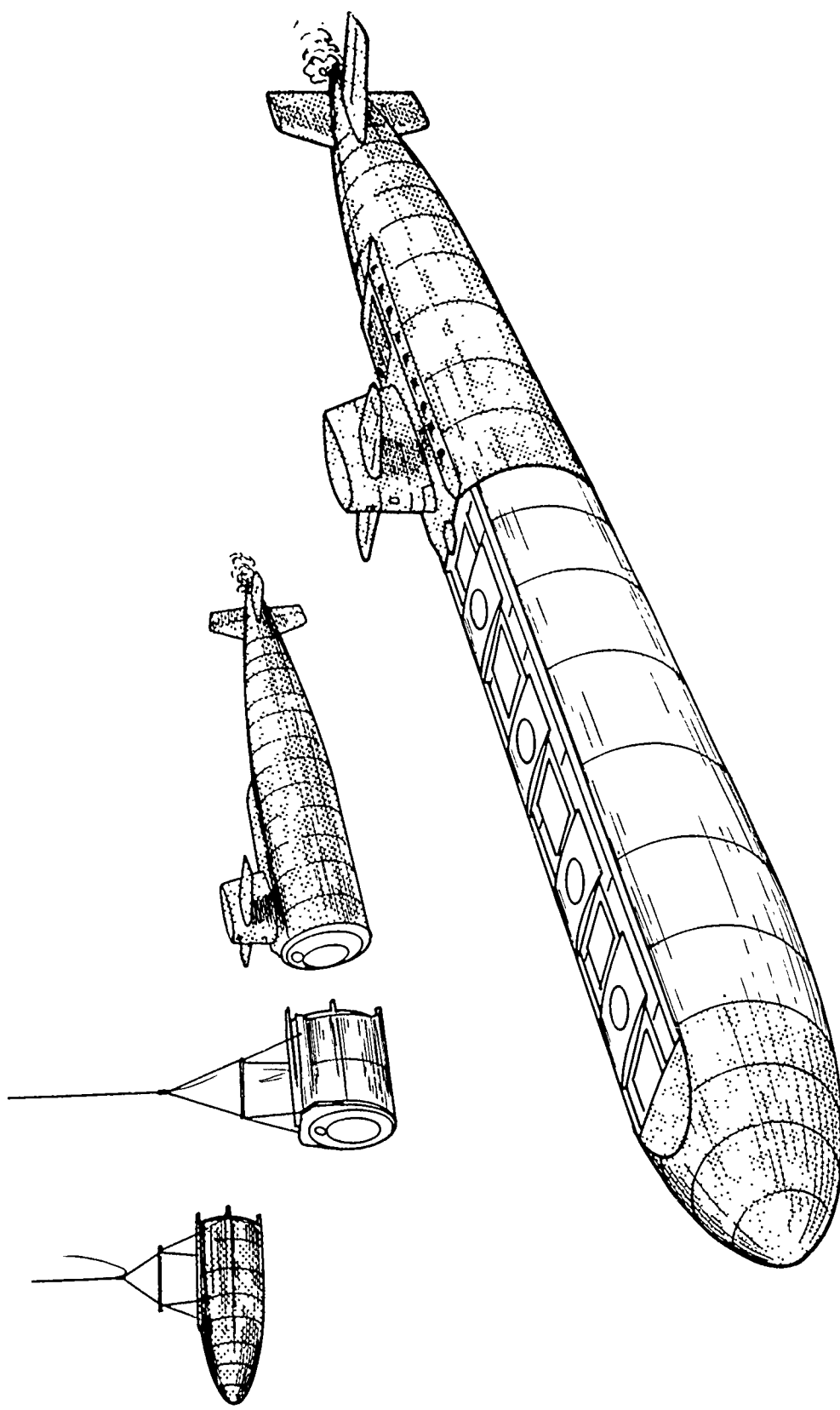


Figure 4 – Modular Submarine

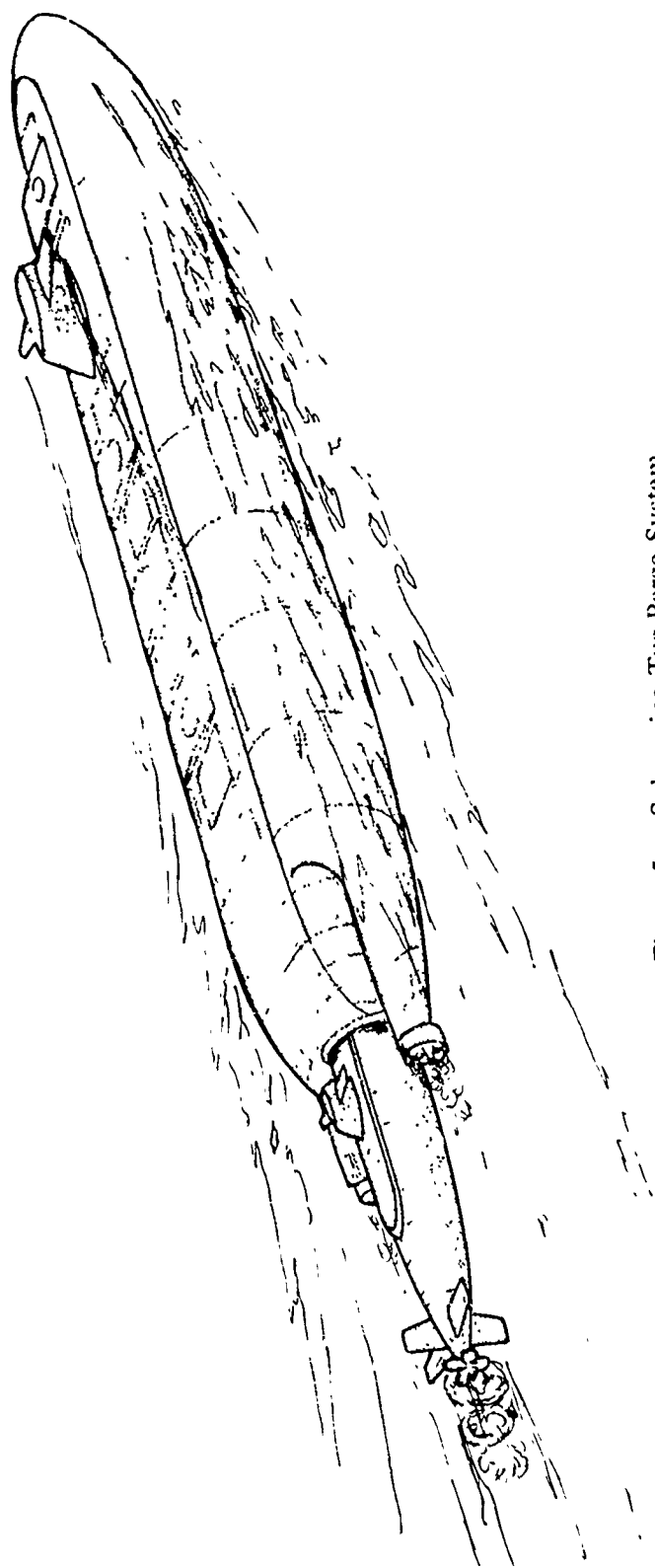


Figure 5 — Submarine Tug-Barge System

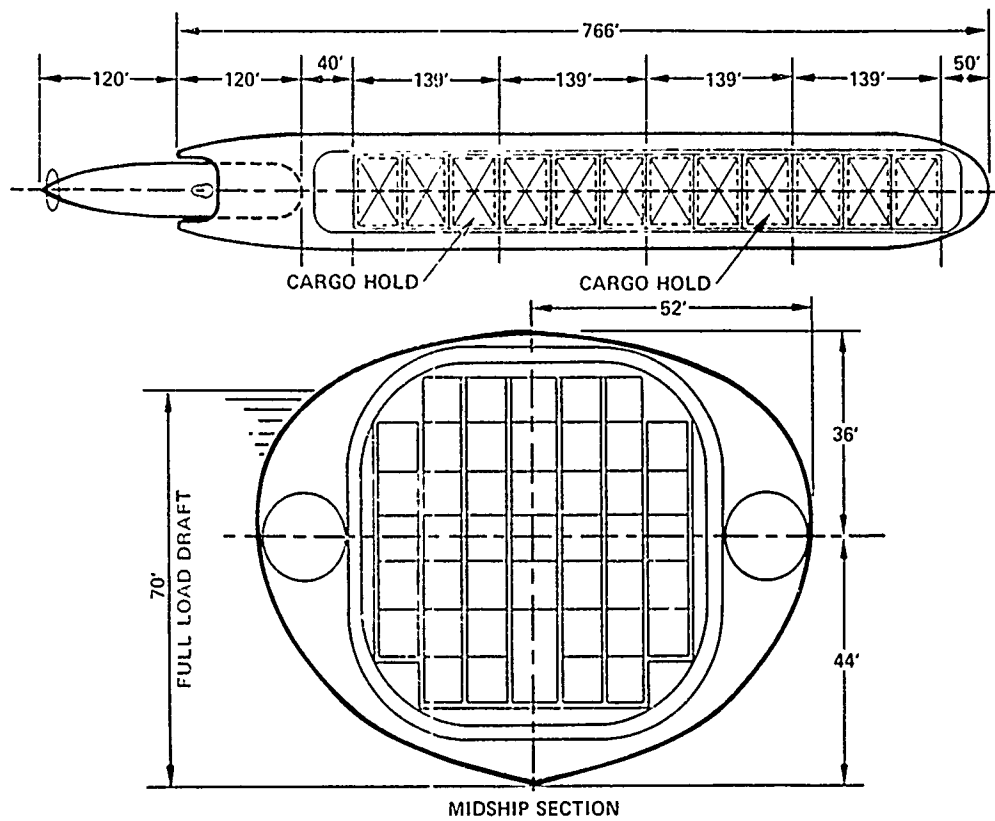


Figure 6 - General Arrangement of Submarine Tug-Barge TB-1

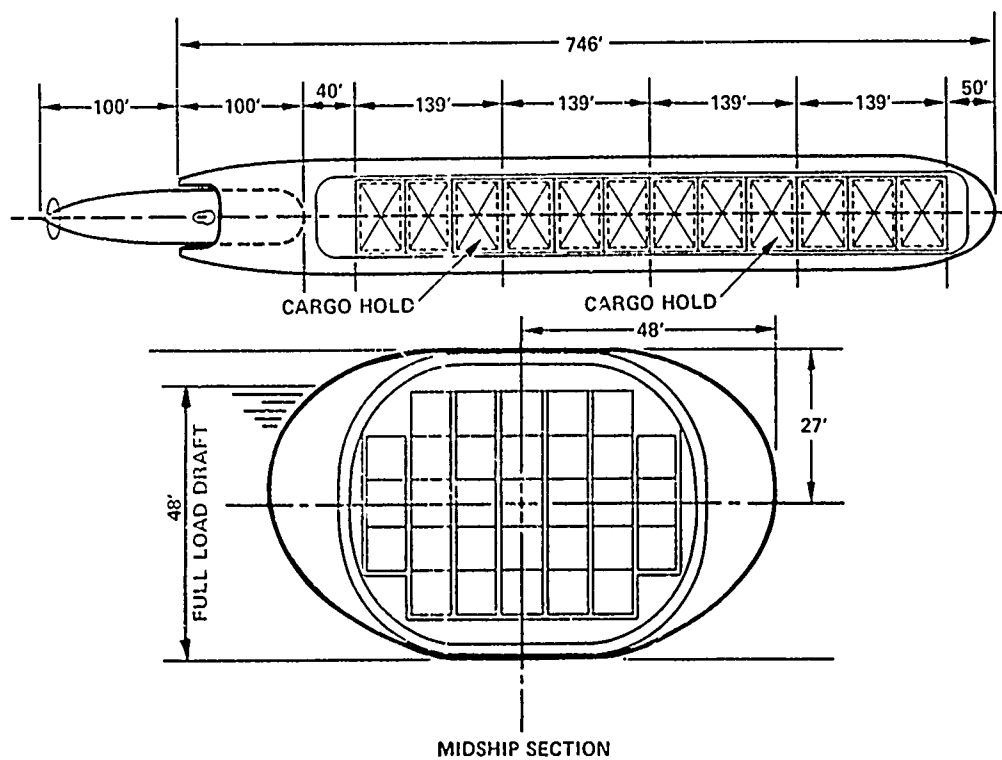


Figure 7 - General Arrangement of Submarine Tug-Barge TB-3

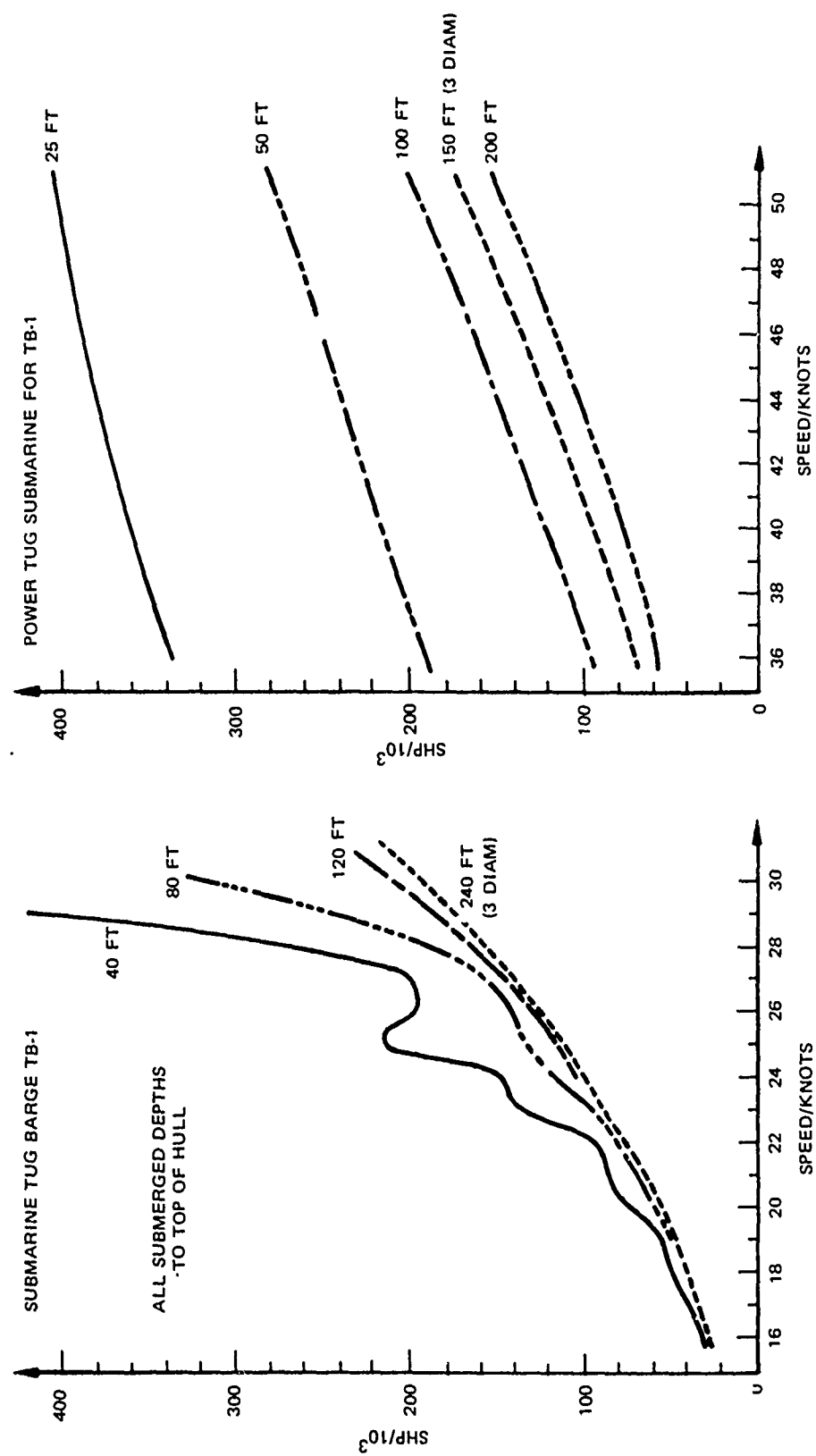


Figure 8 — Speed-Power Curve of the Submarine Tug-Barge TB-1

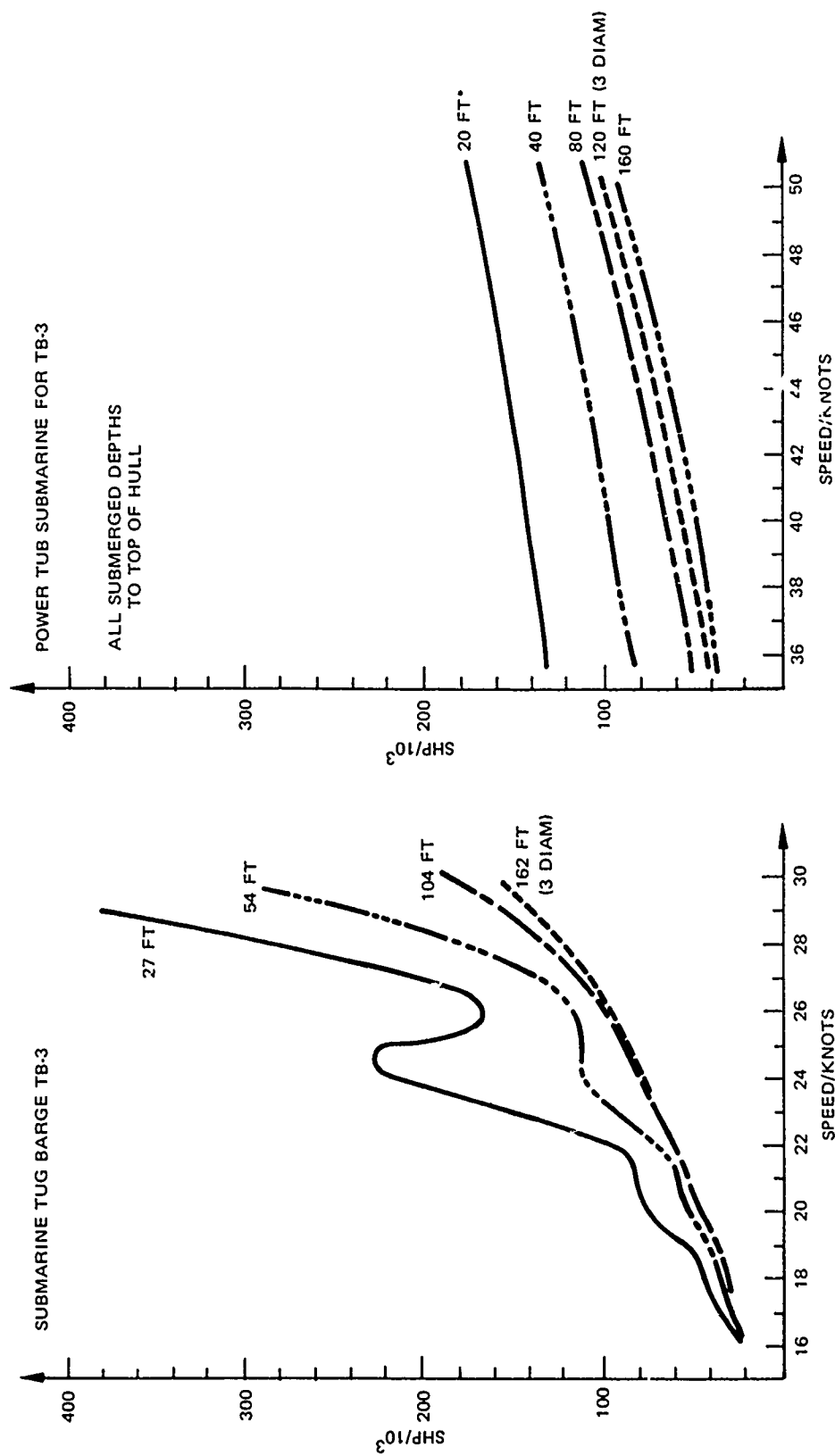


Figure 9 - Speed-Power Curve of the Submarine Tug-Barge TB-3

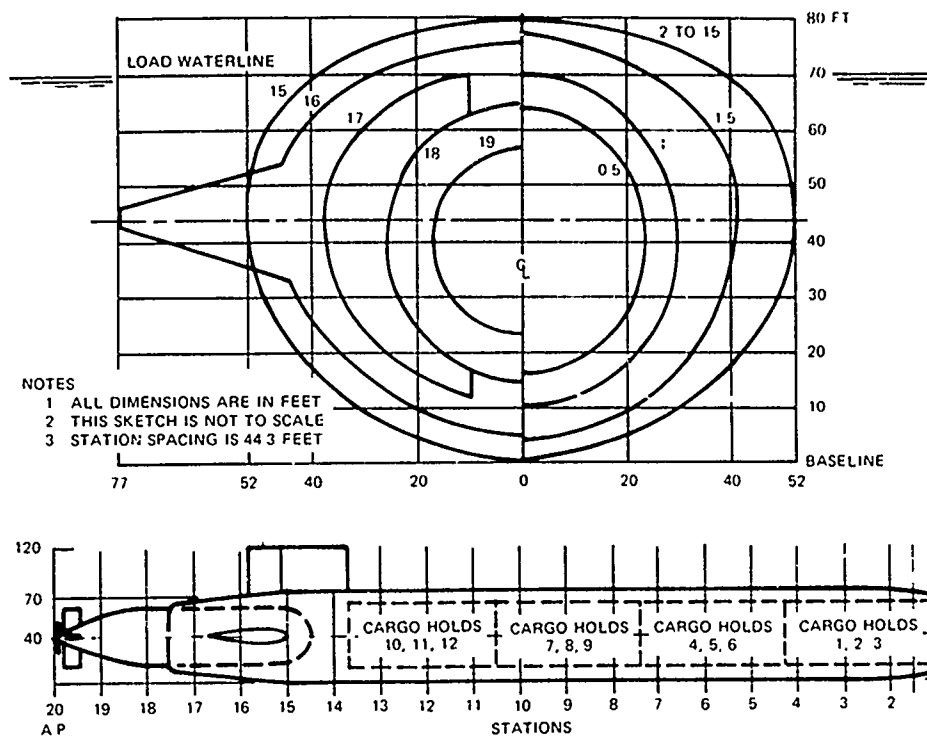


Figure 10 – Body Plan and Profile of Submarine Tug-Barge TB-1

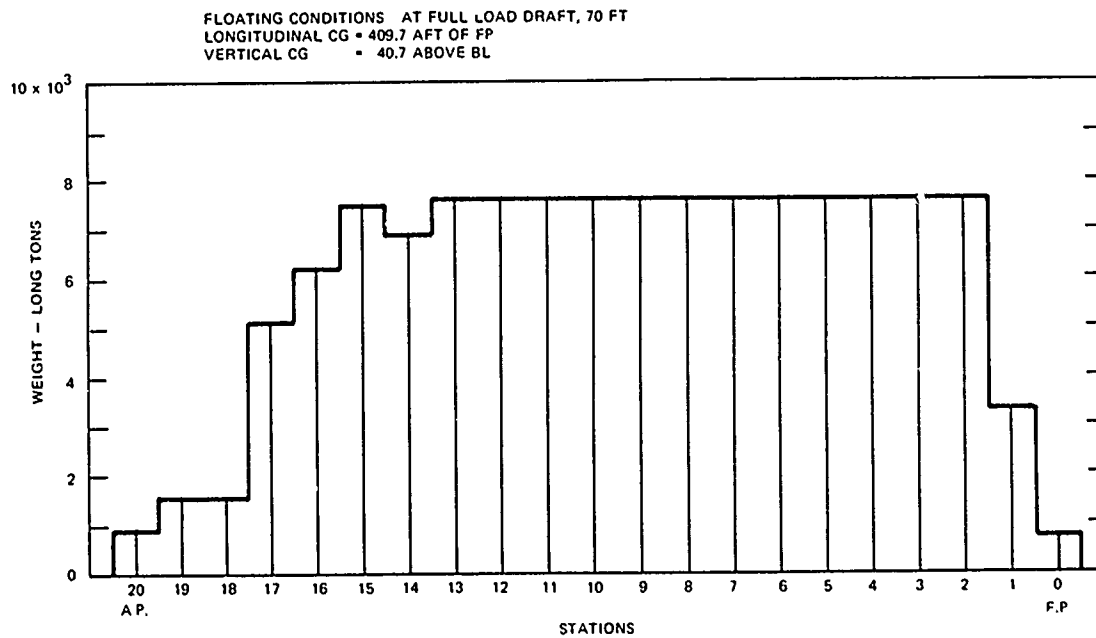


Figure 11 – Estimated Weight Curve for the Submarine Tug-Barge System, TB-1

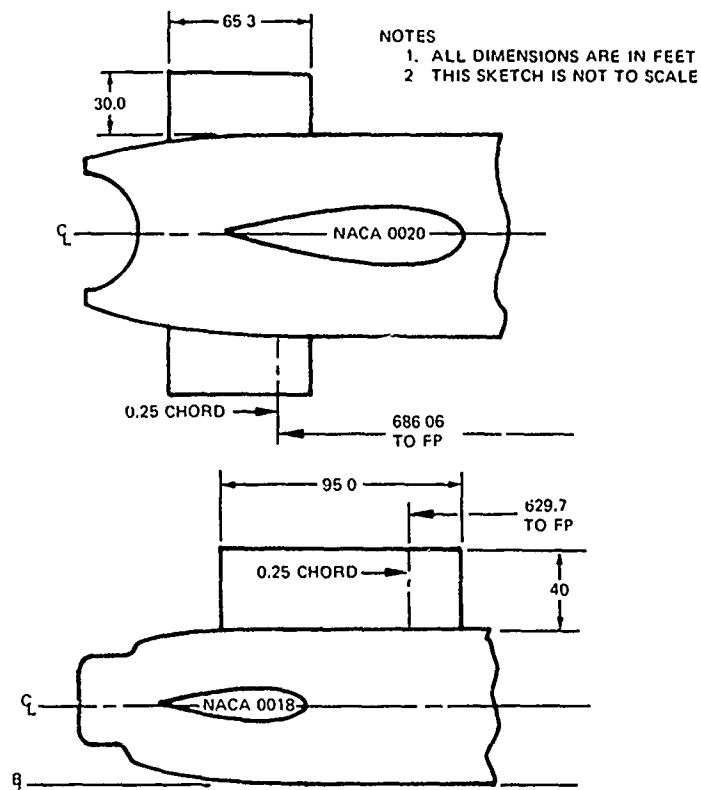


Figure 12 – Appendages for TB-1 Barge

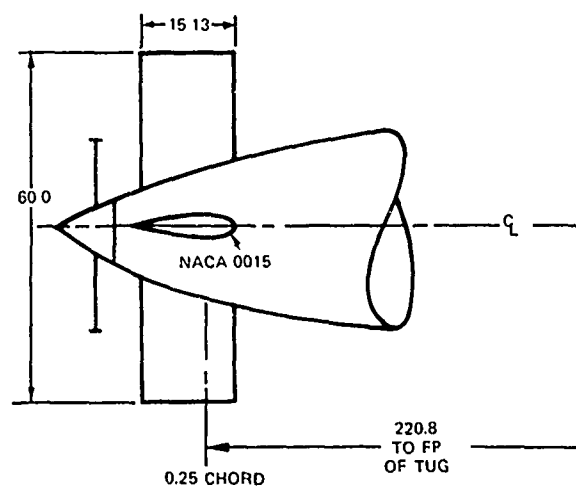


Figure 13 – Appendages for TB-1 Power Tug

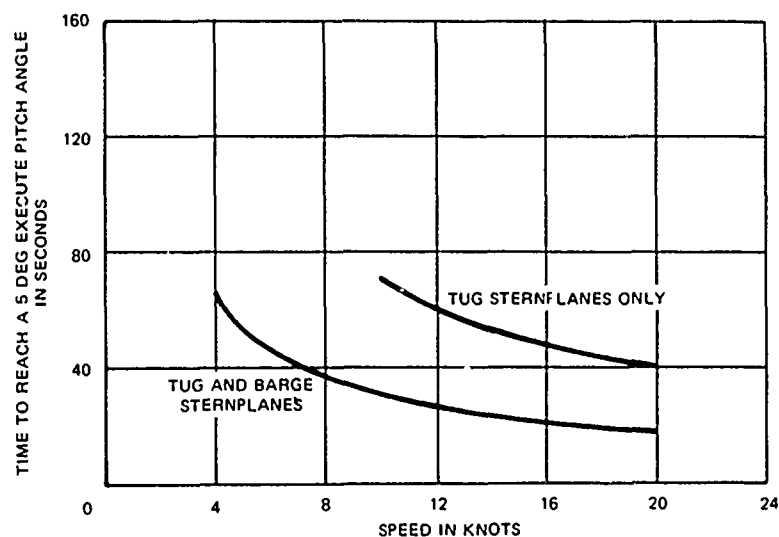


Figure 14 – Control Characteristics for Submarine Tug-Barge TB-1

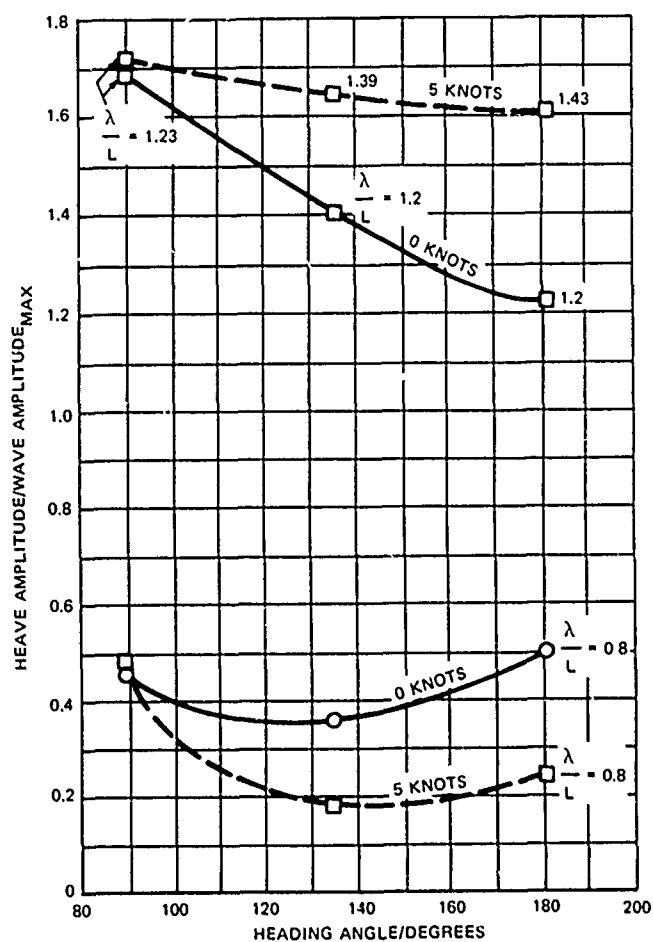


Figure 15 – Maximum Values of Transfer Functions for Heave Motions for 0 to 5 Knots

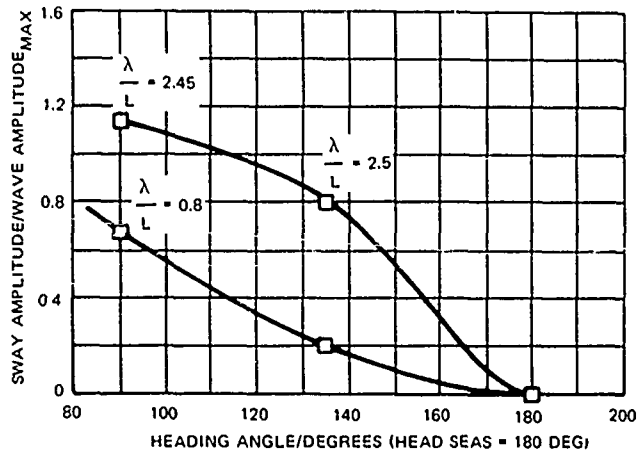


Figure 16 – Maximum Values of Transfer Functions for Sway Motions for 0 to 5 Knots

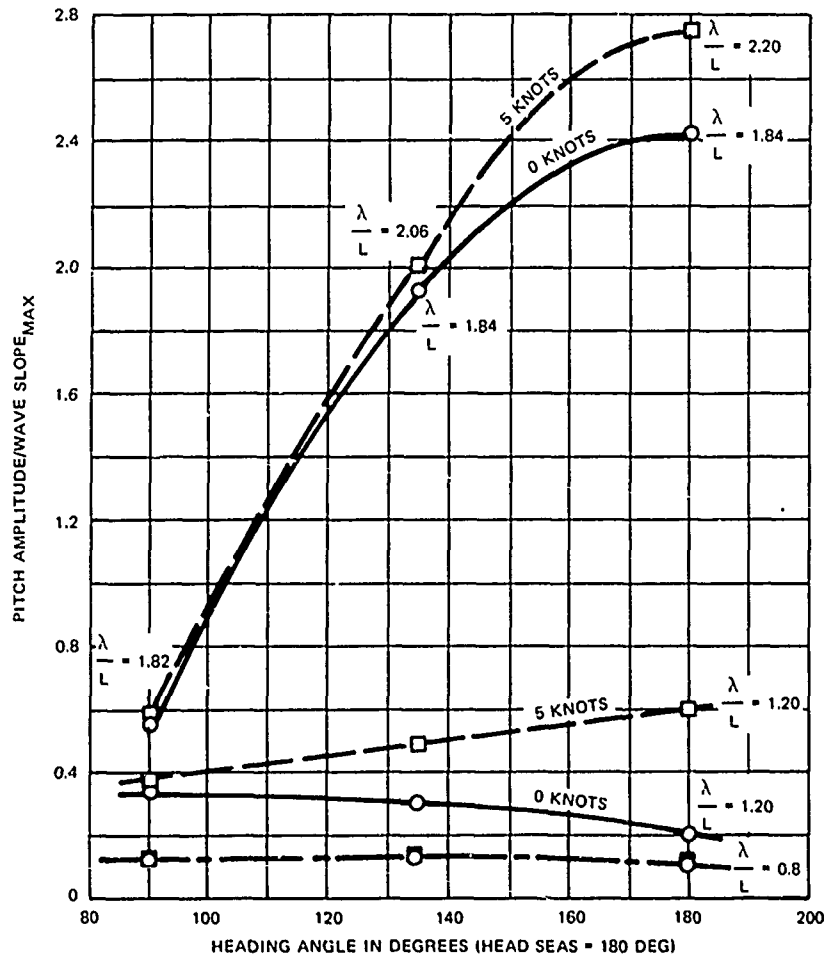


Figure 17 – Maximum Values of Transfer Functions of Pitch Motions for 3 to 5 Knots

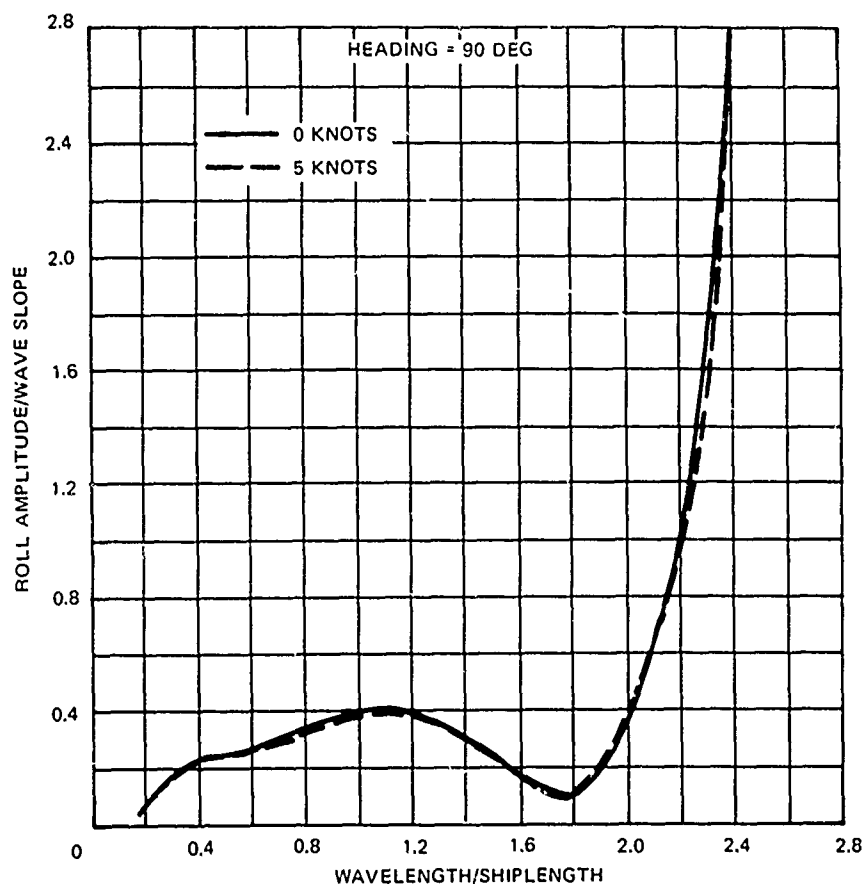


Figure 18 - Transfer Functions for Roll Motion versus Wavelength to Shiplength

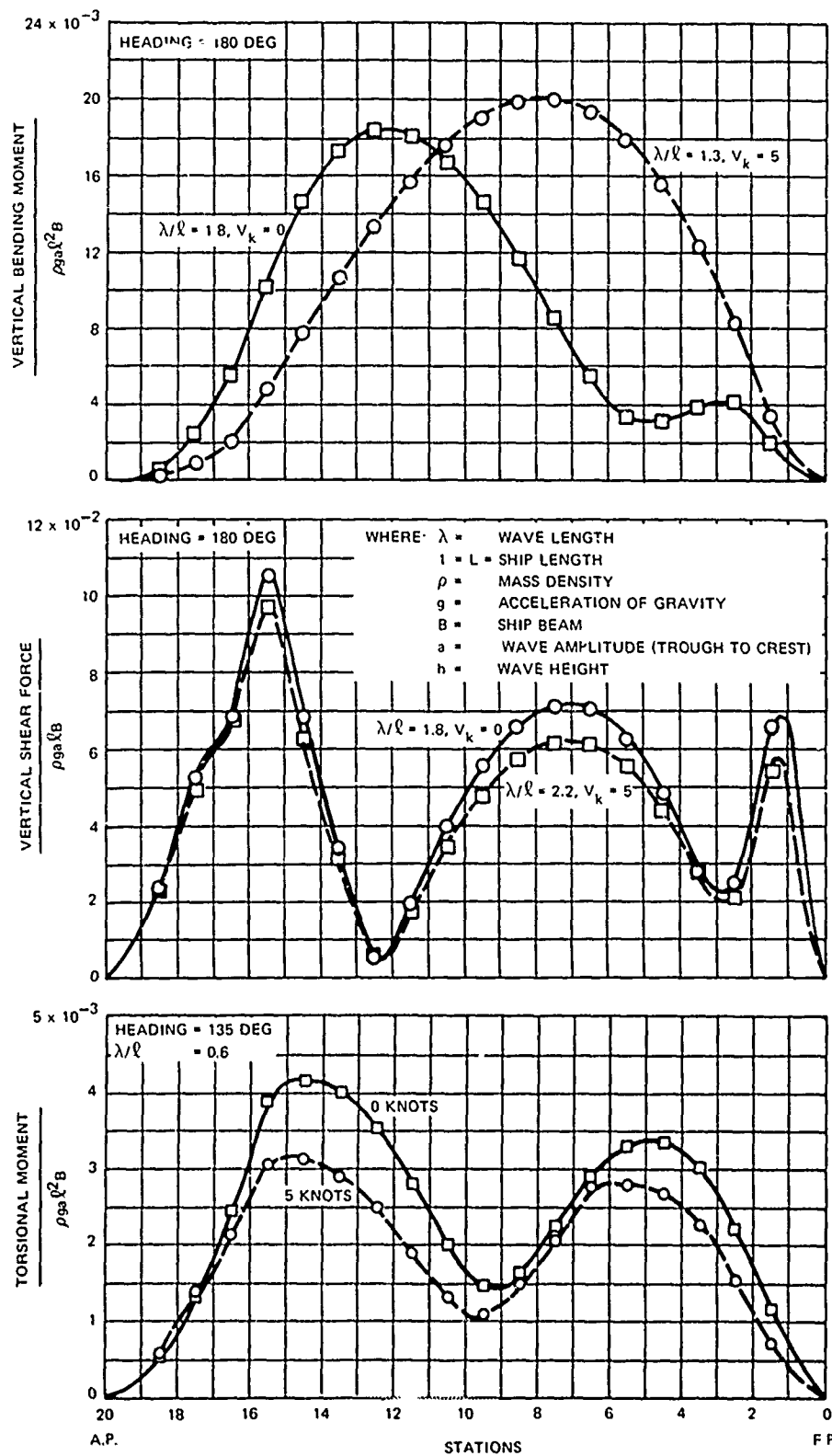


Figure 19 - Transfer Functions for Longitudinal Distribution of Wave Induced Sea Load

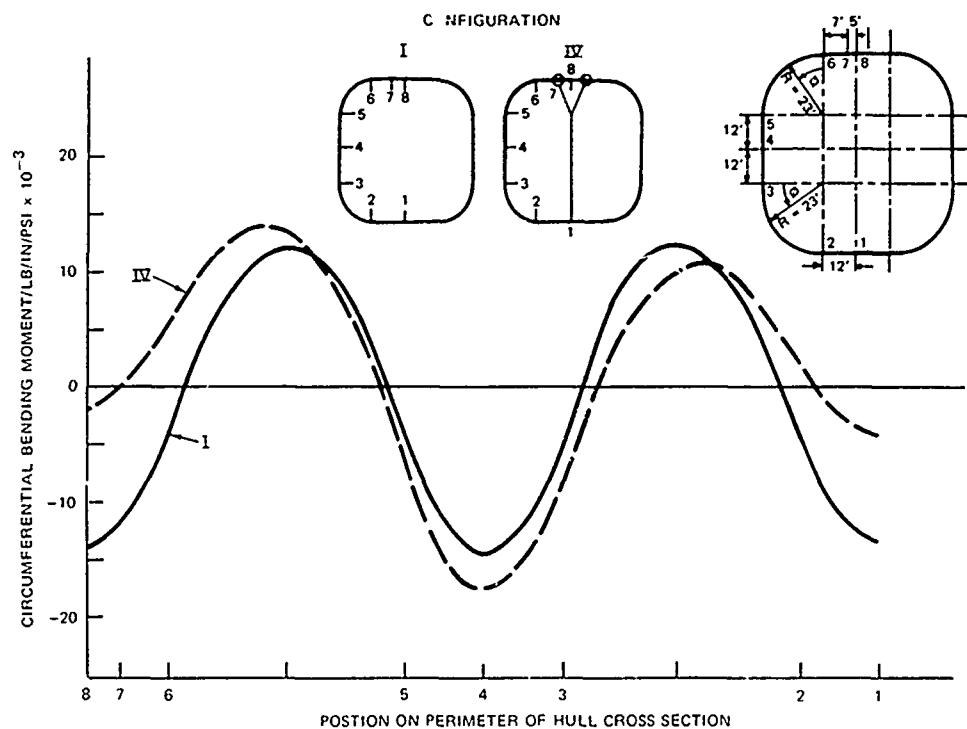


Figure 21 – Circumferential Bending Moments in Pressure Hull of TB-1 and TB-2

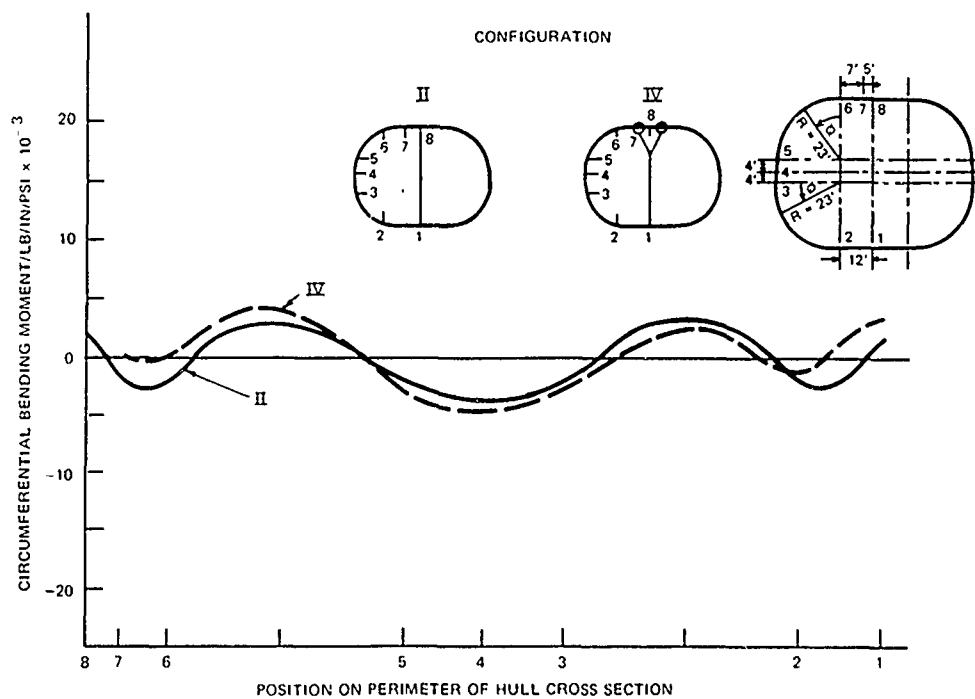


Figure 22 – Circumferential Bending Moments in Pressure Hull of TB-3

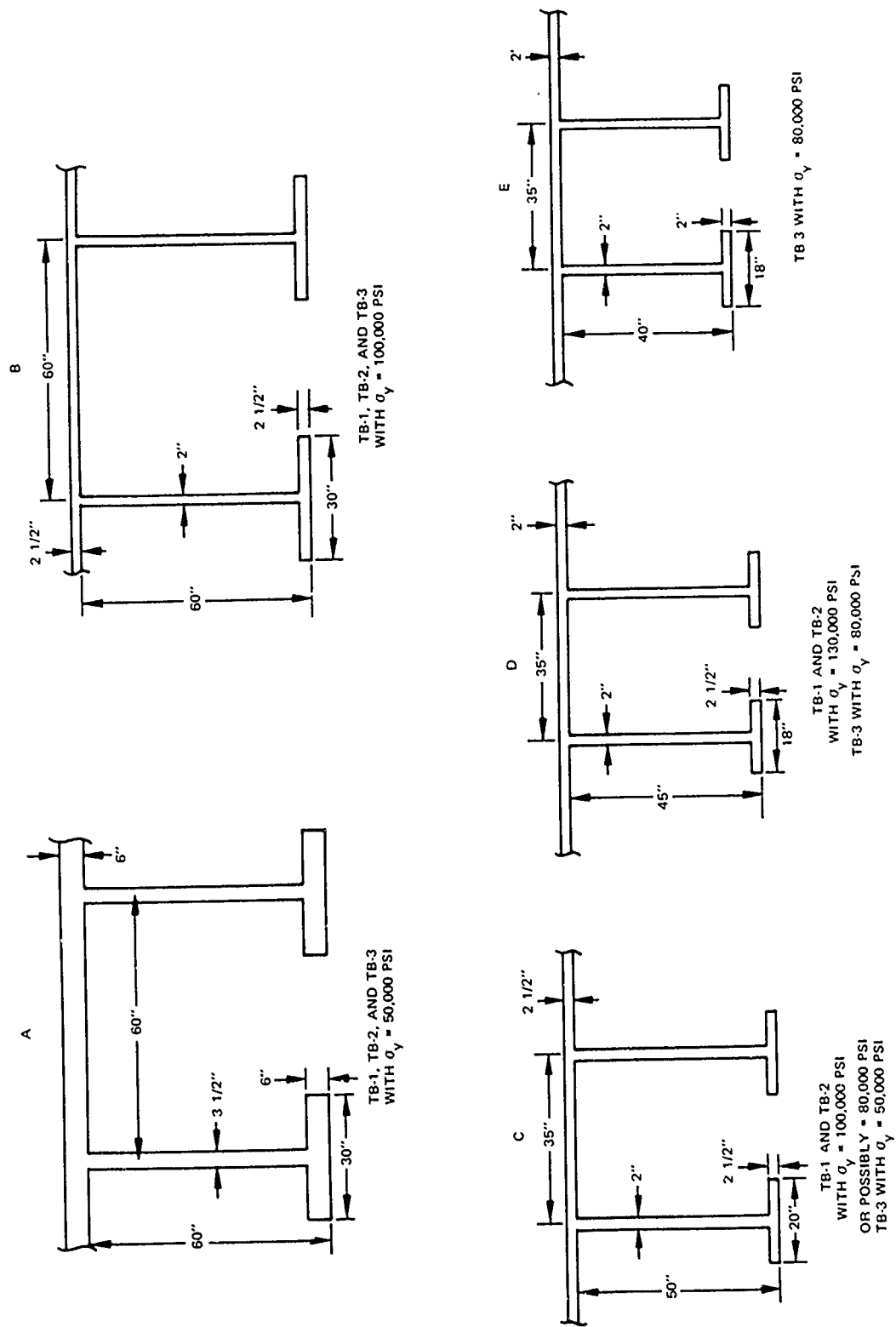


Figure 23 - Five Tentative Hull Designs

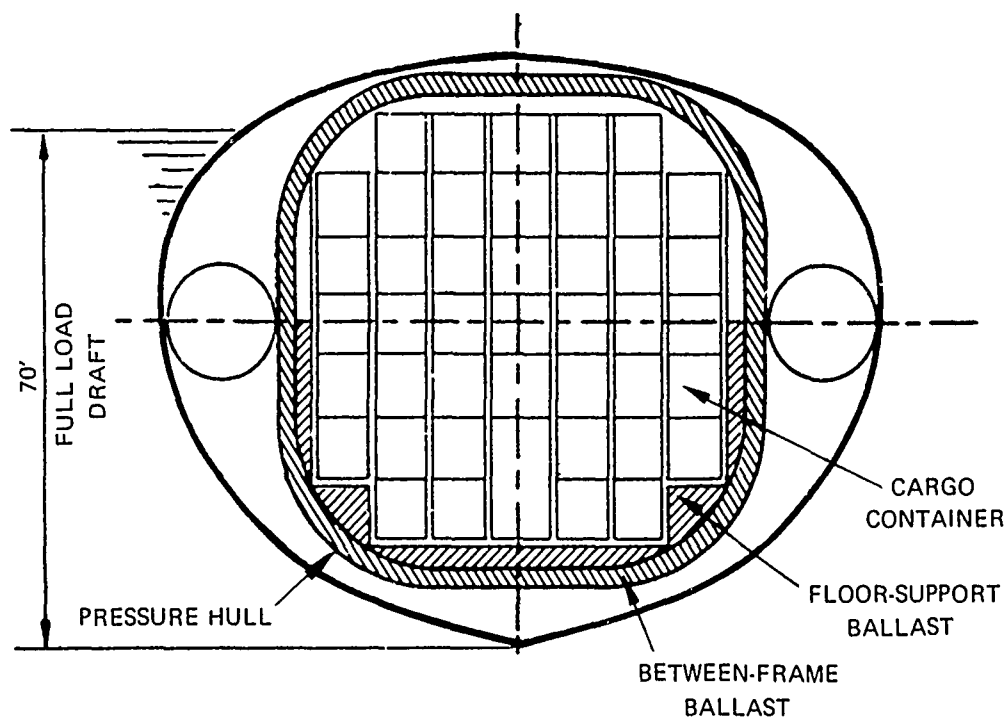


Figure 24 – Location of Permanent Ballast in TB-1 and TB-2

TABLE 1 - PRINCIPAL CHARACTERISTICS FOR PROPOSED SUBMARINE TANKERS

(Measurement is in feet, and tonnage in long tons)

	General Dynamics 170,000 dwt	General Dynamics 250,000 dwt	Saunders Roe 28,000 dwt	MARAD S5-N-M 48a 20,000 dwt	Shigemitsu Kobe 30,000 dwt
Length b.p.	900	1,020	604	565	590.4
Beam, Molded	140	170	72	80	78.7
At Stern Planes	180	220	-	-	-
Depth, Hull at Side	85	90	-	40	-
Hull at Centerline	88	93 1/2	72	40	78.7
Cargo Deadweight	172,200	255,380	28,000	20,000	30,000 (metric)
Displacement					
Full-Load Surfaced	240,100	353,200	45,400	38,000	48,200 (metric)
Light Ship	67,900	94,900	-	-	-
Submerged	254,500	370,000	50,000	41,800	-
Draft, Surface	81	85	59	35 1/2	55.7
Light Ship (mean)	24	24	35	-	-
Minimum Cargo Density	53.6 lb/ft ²	53.6 lb/ft ²	-	-	-
Crew	49	49	-	56	50
Reactor Plant	250 Mw PWR	250 Mw PWR	150 Mw PWR	PWR	180 Mw PWR
Propulsion (by geared steam turbines)					
Shaft Horsepower	75,000	75,000	50,000	35,000	40,000
Propellers (diam)	2(26 ft)	2(26 ft)	1(25 ft)	1(26.1 ft)	2(17.4 ft)
Revolutions per Minute	105	105	120	100	160
Trial Speed (knots)	19.4	17.3	25	20	22
Service Speed	19.0	16.9	-	-	-
Depth Range (ft) Design	1,000	1,000	600	1,000 (approx)	330
Normal Operation	400	400	200-300	-	-

TABLE 2 - MISSIONS FOR LOGISTICS AND REPLENISHMENT

Mission	Maximum Speed Range knots	Maximum Cargo tons	Maximum Horsepower in 1000 Required	Draft feet
Transport War Materials	15-20	100,000	100-250	(Unlimited)
Transport Troops and Support Materials	15-25	40,000	60-100	45
Replenish Surface Ships at Sea	30-35	30,000	100-250	45
Replenish FBM's and SSN's*	15-25	12,000	20- 40	40
Covertly Supply Friendly Nations	15-25	40,000	10-100	45
*FBM's—Fleet Ballistic Missile Submarine SSN's —Nuclear Powered Attack Submarine				

TABLE 3 - LOADING SPECIFICATIONS FOR STANDARD
DEMOUNTABLE CARGO CONTAINERS

Nominal Size feet	Maximum Gross Weight		Volume in Cubic Feet	
	long tons	pounds	Displacement maximum	Capacity minimum
<u>Group I</u>				
40 x 8 x 8	30	67,200	2,560	2,090
30 x 8 x 8	25	56,000	1,916	1,560
20 x 8 x 8	20	44,800	1,272	1,040
10 x 8 x 8	10	22,400	628	490
<u>Group II</u>				
6-2/3 x 8 x 8	7	15,680	413	329
5 x 8 x 8	5	11,200	307	248

TABLE 4 – PRINCIPAL CHARACTERISTICS OF SUBMARINE TUG-BARGE SYSTEM

	Submarine Tug-Barge			Submarine Tug-Barge	
	TB-1	TB-2	Power Tug	TB-3	Power Tug
Displacement, tons					
Submerged	130,000	106,000	10,000	86,000	6,000
Full Load Surfaced	125,000	96,000	9,150	79,000	5,600
Cargo, Containers	1,080	810		744	
Tons Max/Min	21,600/16,200*	16,200/12,150*		14,880/11,160*	
Length, feet	890	750	240	846	200
Beam, feet	104	104	50	96	40
Depth, feet	80	80	50	54	40
Draft, Full Load Surface, feet	70	70	44	48	35
Shaft Horsepower	60,000	60,000	60,000	40,000	40,000
Speed, Submerged, maximum knots	20	22	36	19	38
Propeller (Power Tug)	1	1	1	1	1
*Based on 20 or 15 tons per container					

TABLE 5 – GENERAL RELATIONSHIPS OF SPEED, POWER, AND SUBMERGENCE

Configuration	TB-1				TB-3			
Hull Form	Squared-Round Corner				Rectangular			
Displacement, Submerged, tons	130,000				86,000			
Speed, knots	20		25		20		24.5	
Power, SHP	60,000	75,000	115,000	215,000	45,000	75,000	80,000	225,000
Submergence, feet	240 (3 diam)	40 (1/2 diam)	240	40	162 (3 diam)	27 (1/2 diam)	162	27
Power Factor, percent*	80	100	54	100	60	100	36	100
Transport Efficiency, ton-mile/SHP	46	37	30	16	38	23	26	9
Cargo Capacity Factor**	1.22				1.24			
*Power Factor of submergence is the ratio of the SHP required for a given speed at the submerged depth to that of half hull diameter from the free surface								
**Cargo Factor is the ratio of the total number of containers accommodated in the designed hull form to that of a circular one								

TABLE 6 – GEOMETRIC PROPERTIES AND WEIGHT
DISTRIBUTION CHARACTERISTICS

Length between Perpendiculars, feet	886.0
Beam at Midship, feet	104.0
Draft, feet (Full Load)	70.0
Displacement, long ton (Full Load Surfaced)	125,000
Longitudinal Center of Buoyancy Aft of Forward Perpendicular, feet	409.8
Longitudinal Center of Gravity Aft of Forward Perpendicular, feet	409.8
Vertical Center of Buoyancy above Baseline, feet	39.1
Vertical Center of Gravity above Baseline, feet	40.8
Roll Radius of Gyration about Center of Gravity, feet	33.2
Pitch Radius of Gyration about Center of Gravity, feet	217.2
Yaw Radius of Gyration about Center of Gravity, feet	217.4
Metacentric Height above Baseline, feet	44.6

TABLE 7 – CHARACTERISTICS OF APPENDAGES FOR SUBMARINE
TUG-BARGE TB-1

	Barge Fairwater/Rudder	Barge Sternplanes	Tug Sternplanes and Rudders
Planform Area, feet ² (one surface)	3,800.0	1,959.0	244.8
Location of Quarter Chord Aft of Forward Perpendicular, feet	629.7	686.06	866.8
NACA Section Shape	0020	0018	0015
Steel Weight, tons (one surface)	1,520.0	516.5	12.3
Envelope Volume, feet ³ (one surface)	49,457.0	15,772.9	376.4
Configuration	30 percent flapped	all movable	all movable

TABLE 8 – INDEX OF STABILITY OF SUBMARINE
TUG-BARGE TB-1

Configuration	Horizontal Index of Stability	Vertical Index of Stability
Power Tug Alone	0.23	0.23
Barge with Power Tug—No Fins on Tug	-0.36	0.49
Barge with Power Tug—All Appendages	0.24	0.62

TABLE 9 – MAXIMUM VALUES OF TRANSFER FUNCTIONS FOR DYNAMIC SEA
LOADS FOR SUBMARINE TUG-BARGE TB-1

Heading		Head Seas (180 deg)			Bow Seas (135 deg)			Beam Seas (90 deg)		
Load	Speed knot	Load Function*	Station	λ/l	Load Function*	Station	λ/l	Load Function*	Station	λ/l
Vertical	Shear	0	1,050	15.5	1.8	8	15.5	1.8	680	15.5
		5	970	15.5	2.2	8	15.5	2.0	680	15.5
	Bending	0	18,500	12	1.8	160	7.5	1.1	160	9
		5	20,000	8	1.3	190	7.5	1.3	160	9
Horizontal	Shear	0				10	14.5, 5	0.6	210	15.5
		5				8	14.5, 5	0.6	210	15.5
	Bending	0				280	10	0.6	1700	10
		5				220	10	0.6	3000	9
Axial	Torsion	0				4200	14.5	0.6	1200	15.5
		5				3200	15	0.6	1200	15.5

*Defined in Figure 19

TABLE 10 – GEOMETRIC PROPERTIES OF FIVE HULL STRUCTURE DESIGNS

Design		A	B	C	D	E
Thickness, t, in.		6.0	2.5	2.5	2.0	2.0
Frame Spacing, S, in.		60	60	35	35	35
Cross Sectional Area, A, of Frame and Shell in one Frame-Spacing, in ²		729	340	233	200	182
Circumferential Bending Moment of Inertia, I, of one Frame Space of Hull, in ⁴		493,600	237,900	100,100	68,800	48,500
C/I, to Outer Surface of Shell, in ⁻³		51.85(10) ⁻⁶	103.53(10) ⁻⁶	222.00(10) ⁻⁶	298.40(10) ⁻⁶	347.31(10) ⁻⁶
C/I, to Inner Surface of Frame Flange, in ⁻³		81.88(10) ⁻⁶	159.20(10) ⁻⁶	302.53(10) ⁻⁶	384.74(10) ⁻⁶	518.39(10) ⁻⁶
Shell Buckling Parameter, β^*		1.78	2.84	1.61	1.81	1.81
Weight/Displacement Ratio	TB-1, TB-2	0.418	0.195	0.228	0.196	0.179
	TB-3	0.484	0.226	0.264	0.227	0.207
Required Moment of Inertia of Center Column Cross-Section, Per Frame Space, in ⁴	TB-1, TB-2	19,200	19,200	11,200	11,200	11,200
	TB-3	31,600	31,600	18,400	18,400	18,400
$\beta = \frac{\sqrt{3(1 - \nu^2)}}{\sqrt{P_t}} S$						

TABLE 11 - STRUCTURAL CHARACTERISTICS OF PRESSURE HULL DESIGN

Ship System		TB-1 and TB-2			TB-3					
Hull Configuration	Midship*	Flat Section Top and Side	Circular Section	Midship*	Flat Section Side	Flat Section Top	Circular Section	Flat Section Side	Flat Section Top	Circular Section
Max. Cir. Unit Bending Moment, lb/in/psi	I IV	-14,000 -17,000	+12,000 +13,000	II IV	- 4,000 - 5,000	- -	+ 3,000 + 3,000			
Hull Structure**		Design C			Design C			Design E		
Mat. Yield Strength, psi		100,000			50,000			80,000		
Direct Stress, psi		28,100	32,100		28,100	21,700	28,700	35,900	27,700	36,600
All. Cir. Unit Bending Moment, lb/in/psi		-20,800	+14,400		- 6,300	- 8,200	+ 4,500	- 8,100	- 9,700	+ 5,400
Max. Long. Stress due to Bending of Entire Hull by Surface Wave Loads, psi	I	28,000								

*Midship section configuration as Figure 19

**Hull structure design as Figure 23

*Midship section configuration as Figure 19

**Hull structure design as Figure 23

TABLE 12 - WEIGHTS OF PRESSURE HULLS

Design	TB-1		TB-2		TB-3	
	A	C	A	C	C	E
Weight of pressure hull structure, tons	31,480	17,660	24,050	13,600	16,740	13,770
Weight of displaced water, tons	75,830	75,830	58,030	58,030	53,430	53,430
Weight/weight of displaced water	0.415	0.233	0.414	0.234	0.313	0.237
Weight of cargo, tons Max/Min	21,600/16,200	21,600/16,200	16,200/12,150	16,200/12,150	14,880/11,160	14,880/11,160
Cargo weight/weight of displaced water Max/Min	0.284/0.214	0.284/0.214	0.279/0.209	0.279/0.209	0.279/0.208	0.279/0.208
Weight of concrete* frame support/ballast, if space between frames is completely filled, tons		33,070		24,740	28,650	23,020
Weight of frame support ballast/weight of displaced water		0.436		0.426	0.536	0.431
Weight of concrete* floor-support ballast, tons		11,120		8,320	12,800	15,720
Weight of floor support ballast/weight of displaced water		0.147		0.143	0.240	0.294
*Weight of concrete is taken as 145 lb/ft ³						

TABLE 13 – WEIGHT ESTIMATE FOR TB-1

Weight Item	Loading Conditions		
	Minimum	Maximum	No Load
Fixed Weights, tons			
Structure	62,000	62,000	62,000
Machinery	9,300	9,300	9,300
Outfit, at 10 Percent of Structure	6,200	6,200	6,200
Subtotal	77,500	77,500	77,500
Variable Weights, tons			
Cargo (1080 Containers)	16,200	21,600	—
No Load Ballast (1/3 Cargo Helds at 35 ft ³ /ton	—	—	17,200
MBT	9,700	8,150	8,700
VCT	3,850	0	3,850
VBT	5,640	5,640	5,640
Subtotal	35,390	35,390	35,390
Margin, tons	7,110	7,110	7,110
Total, tons	120,000	120,000	120,000
Note: MBT—Main ballast tank VCT—Variable cargo tank VBT—Variable ballast tank			

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