THE SURFACE EFFECT SHIP

ADVANCED DESIGN AND TECHNOLOGY

Prepared in Conjunction With

Surface Effect Ships Project (PM-17)

Washington, D.C.

By

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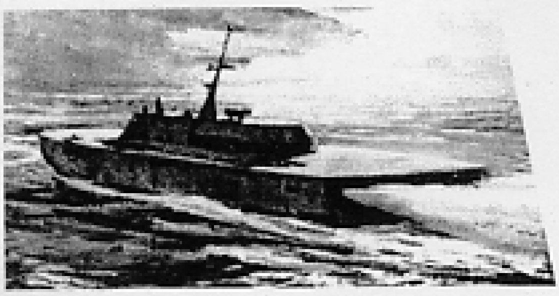
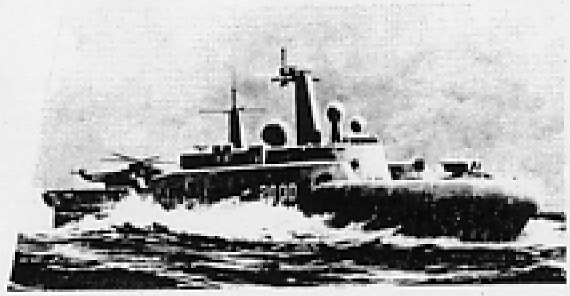
SURFACE EFFECT SHIP PROGRAM





FOREWORD

The surface effect ship (SES) is one of the principal types of high-speed ships in development today, and the text of this book reflects the emergent technology to date. As an advanced development program office, PM-17 has the responsibility for providing a technological base for the design, construction, and operation of fast, ocean going SES's. The program is related to a broad spectrum of national interests through the ultimate goal of utilizing these high performance ships in ocean service at speeds up to 100 miles per hour and with payloads of at least 25 percent of the ship's gross weight.

The material within this book is presented in a manner that is most beneficial to the engineer and the naval architect engaged in advanced design of high-speed surface effect ships. In the marine design field, the ship design sequence, in general, consists of preliminary design, then contract design, and, finally detailed working drawings. In this book advanced design (concept formulation) is considered to be the ship definition phase that precedes preliminary design, and it is the advanced design stage that this book primarily addresses.

A standard methodology for advanced design, however, cannot be described in a textbook manner. Normally, most problems encountered by the designer are presented to him under differing sets of conditions and with two or more unknowns, thus requiring that each problem be resolved in terms of differing variables and end objectives. This implies that the designer must be cognizant of the state of development of various ship systems and the proper projection of critical component development to the time frame of interest. Generally, the progress in technology cannot be derived from a catalog; rather, it is a function of the design engineer's maturity and experience in maintaining his proficiency in consonance with the needs of his profession. Regardless of the nature of the problems, each must be addressed and treated separately, utilizing such complementary fields as mission analysis, cost effectiveness, and cost analysis. Additionally, the designer must understand the relationships among such parameters as lift-drag ratio, propulsor efficiency, specific fuel consumption, and gross to empty weight ratio. In fact, prediction of the ship's drag and determination of the lift-drag ratio and the equivalent lift-drag ratios at various speeds and sea states are prerequisites to solving most design problems. Moveover, the designer must understand the impacts of such factors as hull design, propulsion and lift systems, outfitting, personnel support systems on the payload, ship's operation, and performance.

Generally, the mission of any platform is to deliver a payload over a required distance in a specified time. To relate a mission to specific SES design criteria, the objectives of the mission must be clearly stated and the dominant performance requirements identified. For example, a requirement to match speed with an enemy vessel or to minimize water-transmitted noise may be critical. For a mission involving inshore or coastal operations, a shallow-draft capability may be the overriding design requirement. In other cases cruise speed, range or payload may be the forcing function. The environment in which the SES will perform must also be defined, including both the natural environment, such as the atmosphere and the sea, and the man-made environment, both internal and external to the ship, the latter being the military environment. A number of alternative design configurations, each capable of fulfilling the minimum performance requirements established by the mission and environmental considerations, should be generated in sufficient detail to allow an evaluation of their worth. A preferred design may then be selected.

To assist the designer in pursuing an orderly and rational advanced design approach, the text combines basic theory with numerous design and performance curves, empirical data, and many equations, simplified through acceptable assumptions to enable the user to conduct rapid point or parametric analyses of various SES configurations with appropriate engineering accuracy. The design methodology consolidates tedious and lengthy computations of such parameters as drag, length-beam ratio, cushion pressure, speed, gross weight, and sea state. Using these data, together with additional inputs which are readily derived, the user can perform a rapid series of calculations and arrive at a trial solution. In most cases only two or three iterations or repetitions of the basic calculation are required to converge on a solution. Optimum solutions will, of course, require additional iterations of the same kind.

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Approved	Signed
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Chapter I
THE SES

In recent years a number of different types of high-speed ship concepts have been proposed, and several of these types of ships have been built and demonstrated. The principal ones are the surface effect ship (SES), the air cushion vehicle (ACV), and the hydrofoil. These craft are shown in Figure 1-1 and their characteristics are listed in Table 1-1.

Although all three concepts are interesting innovations and will undoubtedly, in the near future, see much wider use in the military, commercial, and leisure applications, this book concerns itself with the surface effect ship.

SURFACE EFFECT SHIPS PROGRAM

After limited effort during the late 1950's to explore the usefulness of high-speed, captured-air-bubble vehicles, the Navy established a Surface Effect Ships (SES) Project in 1965. The project was proposed to begin in fiscal year 1968. Also in 1965, a study of the SES was initiated by the Center for Naval Analyses. The technical approach subsequently proposed was for seven years of advanced development that included the construction of two 500-ton SES's,

HIGH-SPEED SHIP CONCEPTS

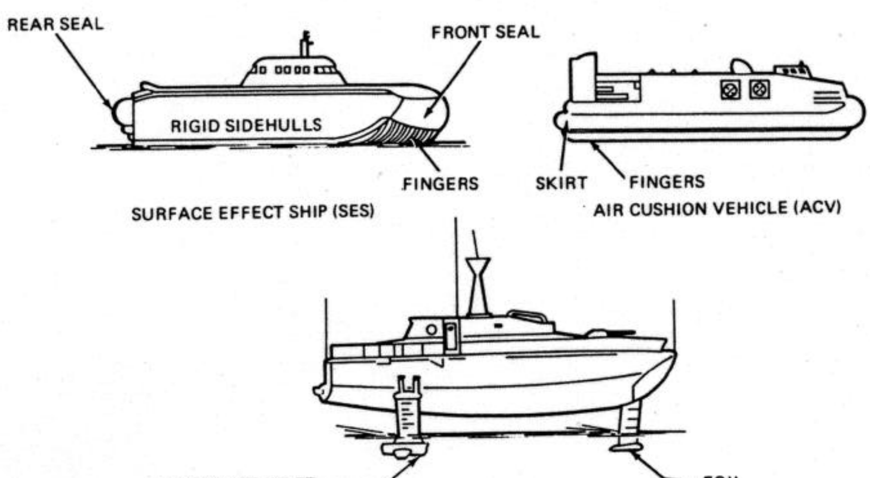


FIGURE 1-1

CHARACTERISTICS OF HIGH-SPEED SHIPS

Surface Effect Ship (SES)	Air Cushion Vehicle (ACV)	Hydrofoil Ship
Hard sidewalls submerged below waterline.	Fully air supported, hovers over the water.	Supported by foils during high-speed operation.
Flexible bow and stern seals only.	Fully flexible skirt around periphery.	Conventional hull with retractable foils.
Nonamphibious.	Amphibious capability.	Nonamphibious.
Can have propeller, waterjets, or air propulsion. Sidewalls provide convenient mounting volume for water propulsor and fuel tanks.	Propulsor always deck mounted and powered by air propulsion.	Propeller, waterjet, or air propulsion.
Sidewalls improve control and maneuverability.	Cross-wind sensitive.	Excellent control and maneuverability.
Sidewalls minimize air cushion loss area. Ratio of lift HP to total HP of approximately 20 percent.	Inherently has greater air cushion loss area. Ratio of lift HP to total HP of approximately 60 percent.	
More efficient at speeds up to 125 knots. Useful operating speeds up to 125 knots.	Less efficient at speeds below 125 knots. Useful operating speeds up to 180 knots.	Useful operating speeds up to 50 to 60 knots.
Directional control obtained from submerged devices, such as keels, skegs and rudders, or waterjet thrust vectoring.	Directional control dependent on air rudders, which are adversely affected by crosswinds.	Directional control obtained from steerable struts.
Lift-drag ratios of 15–25.	Lift-drag ratios of 8–15.	Lift-drag ratios of 8–12.
Pitch-roll stability derived from hydrodynamically shaped sidehulls and appendages.	Pitch-roll stability depends on the air cushion.	Pitch-roll stability depends on the strut location.
High cushion pressure provides inherently smoother ride.	Low air-cushion pressure (~ 45 lbs/ft^2) gives a rough ride.	Proper strut location and damping system provides smooth ride.
Hydrodynamic design permits moderate off-cushion speed.	Off-cushion speed very low.	Off-foil speed comparable to conventional ships.
Water propulsors generate more water noise and little or no air noise.	Air propulsors generate more air noise and little or no water noise.	Dependent on type of propulsion.

TABLE 1-1

one utilizing a waterjet for propulsion and the other using semi-submerged supercavitating propellers.

Also in late 1965, the Commerce Department convened the Surface Effect Ship for Ocean Commerce (SESOC) Study Panel. The panel comprised members of industry, government, the military services, and military laboratories. Panel members studied all the different concepts of SES. As a result of these efforts, the panel identified crucial technical problems and recommended a program to pursue the captured-air-bubble concept because of its comparatively greater efficiency in open-ocean operations. This program included an initial five-year technology development phase prior to building a 500-ton SES.

Reinforced by a Presidential directive in March 1966, the President's Science Advisory Committee called for the Atomic Energy Commission and the Departments of the Navy and Commerce to conduct further investigations into the SES concept. Additionally at that time, the Office of the Secretary of Defense sponsored an Institute for Defense Analysis study to identify SES potential. To conserve funds, the Departments of the Navy and Commerce signed an agreement on 20 June 1966 to pursue jointly a program of SES technological development of mutual interest.

This agreement established a cooperative advanced development program to provide basic technological knowledge and sufficient data to determine the feasibility of building and operating large SES's in the range of 4000 to 5000 tons, capable of 80 knots or higher speeds. The SES program was generally to follow the approach recommended by the SESOC committee and be jointly and approximately equally funded. A unique management structure was specified, including regular Secretarial level review. To initiate the program, Navy funds for fiscal year 1967 were reprogrammed to match Maritime Administration funds. The joint Navy-Commerce Surface Effect Ships Program was officially chartered in early 1967 after program manager selection, and all ongoing Navy SES effort was transferred to program control at that time.

Early in the program, detailed planning studies made it clear that the SESOC approach was not ideal. As a result, a plan was developed for a systematized, two-phase, milestone-oriented program proceeding through increasing confidence levels. This program was technology oriented to solve crucial technical problems through analysis, laboratory effort, and testing of small, manned models. A parallel systems engineering effort combined the emerging technology into overall systems and major testcraft.

This Phase I program extended from program inception during fiscal year 1967 through fiscal year 1972 and was formulated to accomplish four main objectives:

- To establish a technological base through laboratory tests and analytical studies
- To evaluate the operational characteristics of the surface effect ship by operating manned testcraft
- To define appropriate naval missions for surface effect ships' application
- To initiate the development of the 2000-ton prototype ship

In fiscal 1970, following a one-year stretch-out in Phase I due to fiscal restraints, the Commerce Department, while retaining program interest, was forced to make a severe reduction in its SES funding to permit its R & D efforts to be concentrated on more near-term and immediate steps to revitalize the merchant marine. Following the Maritime Administration's decision to reduce its funding, the joint SES program was reevaluated. While the basic approach was retained, the emphasis in the Phase I effort was shifted toward primary Navy objectives.

This shift of Phase I program activity to primary Navy objectives was focused on the two 100-ton testcrafts. Since no adequate laboratory facilities were available for proper testing of the stability, resistance, structural loads, or cavitation characteristics associated with high-speed ships, the testcrafts served two purposes. They not only verified the feasibility of the SES concept but also provided the means for obtaining the necessary performance, man-machine interface, and maintenance data.

Phase II of the Navy's program is planned for fiscal years 1973 through 1977. The main thrust in Phase II is related to the design, construction, and test of an experimental ship of about 2200 tons. The program structure will encompass the same three areas as the Phase I program, analysis, laboratory effort, and operation of manned testcrafts. Technological development will be extended with the basic objectives being improved ship configurations having lower resistance, more efficient structure, more efficient propulsion, and improved handling. In this sense, the technological development of the SES will proceed in a fashion similar to that of aircraft. The hull and propulsor combination of the 2200-ton ship is intended to be adaptable to the maximum extent practical. It will be evaluated first as a test platform during the Phase II test program and evaluated subsequently in conjunction with appropriate mission equipment.

The Phase II SES work represents a substantial research and development effort on the part of the Navy. Two major prerequisites are to be met before undertaking the Phase II effort, namely:

- Sound development of technology, based on successful testing and operation of the 100-ton testcraft
- Definition of priority naval missions for the SES

BASIC PRINCIPLE

The SES is a hybrid ship – a mixture of high-performance ship and low-performance aircraft – marrying proven aircraft practices to ship design and construction in achieving high performance ship characteristics.

In princple, the SES is supported on a cushion of air. This cushion of air raises the ship up in the water, minimizing the contact between the hull and the water. This results in a substantial decrease in resistance (drag). The drag curve resulting from the application of this principle is similar to that shown in Figure 1-2. Lower drag permits higher speed for a given power.

The air cushion is retained by rigid sidehulls that pierce the water surface and by flexible bow and stern seals that follow the action of the waves, thus minimizing leakage from the cushion in all directions. Leaked air is compensated for by air fed into the cushion by gas turbine-powered lift fans.

DRAG CHARACTERISTICS OF THE CONVENTIONAL SHIP AND THE SES

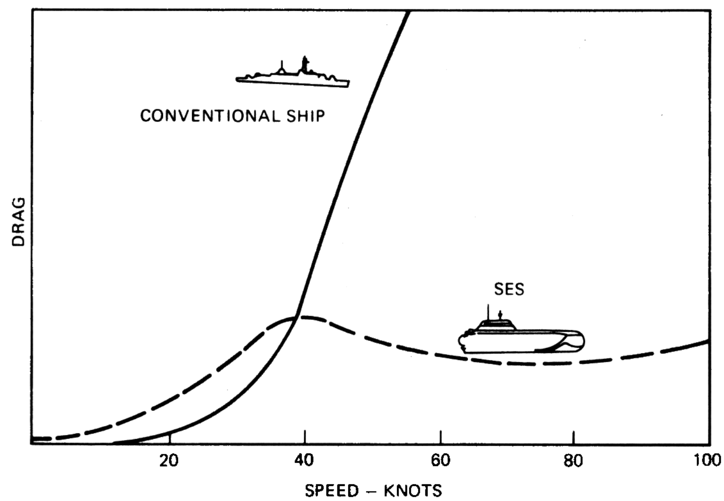


FIGURE 1-2

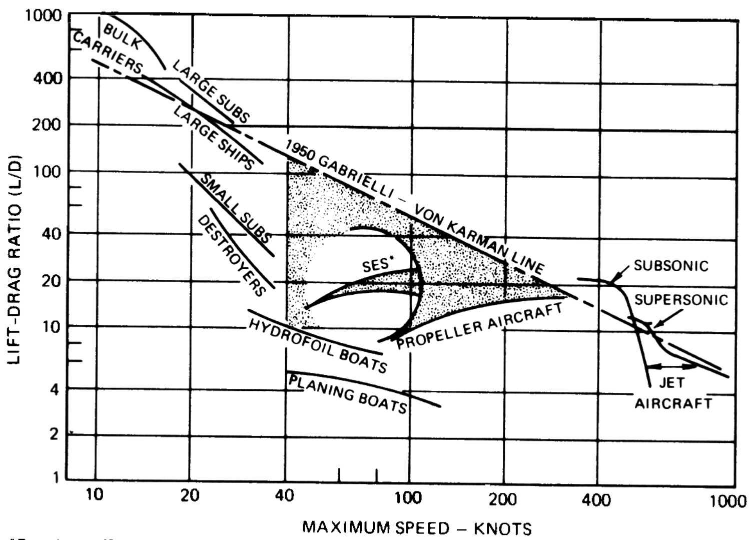
The general geometry is broad and squat. A length-beam ratio of two to three appears to be very close to optimum for a wide range of sizes and missions.

Marine gas turbines drive semi-submerged supercavitating propellers or waterjets through appropriate gears and shafting to drive the ship and to provide power to the lift fans. Gas turbines appear to be the only prime movers having the required combination of efficiency, power and lightness; however, the combined weight of fuel and turbines limit viability of gas turbines as a prime mover to a ship size of approximately 5000 to 8000 tons.

TRANSPORTATION EFFICIENCY OF THE SES

One simple measure of transportation system efficiency is the "Gabrielli-von Karmen" type of presentation shown in Figure 1-3. In this figure, lift-drag ratio which relates the amount of drag incurred to the total vehicle weight is plotted against vehicle speed for a number of air and waterborne vehicles. Reductions in drag allow decreases in propulsion system weight and fuel weight or allow increases in range and speed.

LIFT-DRAG RATIOS OF AIR AND WATERBORNE VEHICLES AS A FUNCTION OF MAXIMUM VELOCITY



*Based on effective lift-drag in calm water, 2000 to 10,000 ton size.

FIGURE 1-3

As shown in the figure, large tonnage bulk carriers have very high lift-drag ratios but can travel only up to speeds of approximately 15 knots. At the other end of the speed spectrum, jet aircraft can achieve speeds up to 1000 knots but with greatly reduced lift-drag ratios. With speed capabilities between 40 and 120 knots, the SES operates at moderate lift-drag ratios in the range of 15 to 25. Since lift-drag ratio is related to the load-carrying or payload capability of a given vehicle, it can be seen that the SES represents the most efficient transportation means between the extremes represented by bulk carriers and jet aircraft. The unique fit of the SES into the overall speed-payload void between conventional ships and jet aircraft provides the potential for a number of commercial and military applications.

It is also worth noting that, although both hydrofoils and planing craft appear within the same speed regime as SES's, each type of ship has certain limitations dictated by the uniqueness of its design. Planing craft are essentially restricted to calm-water operation to achieve their design speed potential. Hydrofoils, although not limited by sea state, are limited in size to less than 2000 tons due to structural considerations. Only the SES appears to offer high speed, high payload, and a clear growth potential.

SPEED, WAVE HEIGHT, AND PERFORMANCE OF THE SES

For military missions, high maximum speed is an asset in combat situations, and a high cruise speed allows rapid, flexible deployment. The ability to maintain these speeds in the weather conditions normally encountered in the open sea is also most desirable. Figure 1-4 indicates the maximum speeds for a number of seagoing vessels and the degradation of these speeds as sea state worsens. Also shown is the percentage of time these sea states will be encountered in the North Atlantic and in other less severe environments. The SES is seen to have a clear speed advantage over other candidate ships for a wide range of sea states.

SPEED AND PERFORMANCE OF SES RELATIVE TO WAVE HEIGHT

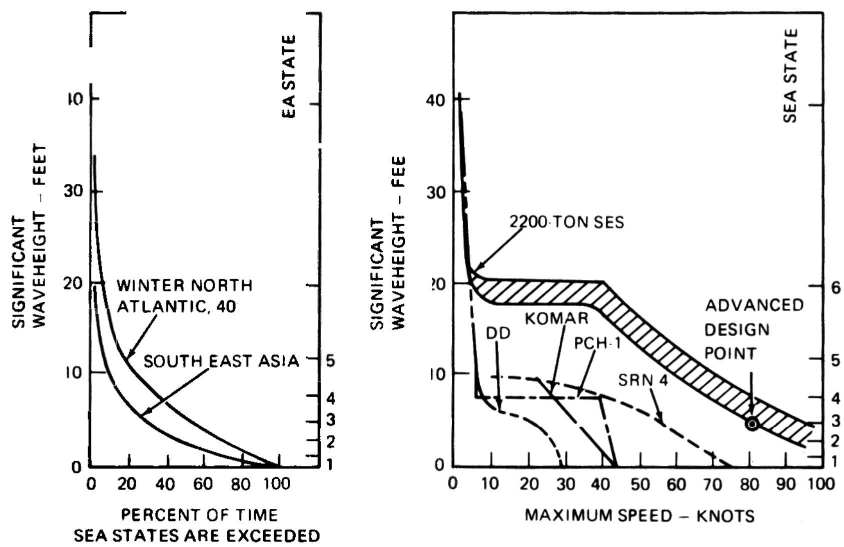


FIGURE -4

A six-foot wave reduces the range of a 100-ton testcraft by about 50 percent, a 1000-ton craft by 25 percent, and a 10,000 ton craft by 10 percent. In general, sensitivity to wave height decreases with SES gross weight. For an SES of approximately 2200 tons, range sensitivity is directly related to performance parameters on approximately a one-for-one basis. The performance sensitivity of the 2200-ton SES at cruise speed is briefly outlined in the following list:

1% decrease in specific fuel consumption gives 1% increase in range

1% decrease in empty weight gives 1.5% increase in range

1% increase in design-speed gives 1.5% decrease in range

1% increase in propulsion efficiency gives 1% increase in range

1% decrease in cushion power gives 1/4% to 1/2% increase in range

1% decrease in drag gives 1% increase in range.

EVOLUTION OF THE SES

If successful, the two phases of the SES program will have established the technological base required to build operational SES's before the turn of the century. Figure 1-5 shows 100-ton ships as being operational in the early seventies, 2200-ton ships in the latter seventies, and 3000-5000-ton ships in the mid-eighties. Such progressively maturing technology will readily provide the capability of designing and building ships in the 10,000-ton range by 1990. Should this capability become a reality, a full operational status of SES's in any size range could be established before the turn of the century.

EVOLUTION OF THE SES

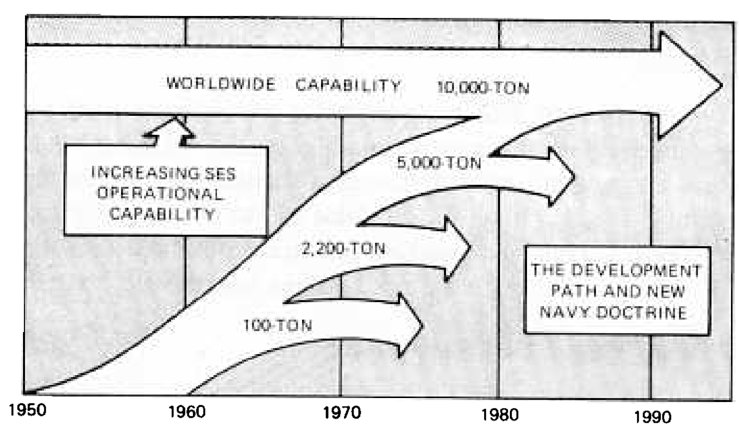


FIGURE 1-5

Chapter II
DRAG

The drag of an SES is the first parameter that must be determined in a design study. Some elements of drag, such as aerodynamic, appendage, and momentum drag can be predicted with relative ease; other elements, such as wavemaking and sidehull frictional drag, are closely interrelated and not easily predicted. In this section of the book attention is focused on the sources and the methods of predicting, the following SES on-cushion drag components: cushion wavemaking drag, sidehull wavemaking drag, sidehull frictional drag, sidehull form drag, aerodynamic drag, momentum or ram drag, bow and stern seal drag, and appendage drag.

In the displacement or off-cushion mode, the SES exhibits characteristics similar to other displacement vessels and is governed by the classic speed-power relationship for this type of vessel, that is that the approximate power required is proportional to the cube of the ship's speed.

CUSHION WAVEMAKING DRAG

Wavemaking drag is the predominant component of SES drag at hump speed. Hump speed is defined as that speed at which wavemaking drag is a maximum. Wavemaking drag is produced by two SES drag subcomponents, one caused primarily by the cushion pressure and the other due to the action of the submerged portion of the sidehull. In reality, the cushion and sidehulls must act as one non-planar pressure surface and produce a single wavemaking drag.

The wavemaking drag of the cushion can be calculated by

$$D_{WM} = 2C_{PC}\,F_N{}^2 f_\ell (P_C S_C) = f_\ell\left(\frac{4}{\rho_w g}\right)\left(\frac{P_C}{L_C}\right)(P_C S_C$$

$$D_{WM} = \frac{f_\ell P_C{}^2 B_C}{16} \qquad (2.1)$$

where

D_{WM} = cushion wavemaking drag – pounds

C_{PC} = cushion pressure coefficient = $P_C/½\rho_w V^2 = P_C/0.995V^2$

F_N = Froude number = $V/\sqrt{gL_C} = V/5.67\sqrt{L_C}$

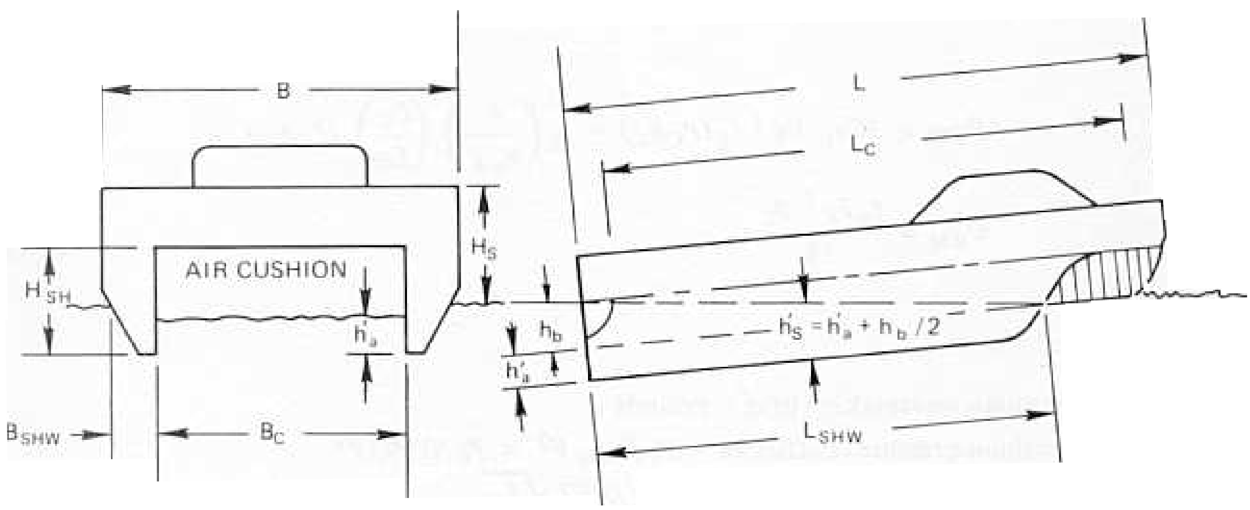
f_ℓ = cushion wavemaking drag coefficient, which is a function of Froude number and length-beam ratio of the cushion (see Figure 2-2)

P_C = cushion pressure (gage) – pounds/foot^2

ρ_w = mass density of water (1.99) – pound-second^2/foot^4
g = gravitational acceleration (32.2) – feet/second^2
L_C = cushion length – feet
V = ship speed – feet per second
S_C = cushion area ($L_C B_C$) – feet^2
B_C = cushion beam – feet.

Figure 12-4 indicates SES dimensions for ships with a cushion length-beam ratio of 2.0 and a cushion pressure-length ratio of 1.5. These dimensions are defined in Figure 2-1. The cushion wavemaking drag coefficient, f_ℓ, is plotted in Figure 2-2 as a function of Froude number and cushion length-beam ratio. These curves are useful for evaluating the true effect of cushion geometry on wavemaking drag. When the SES is operating on the cushion in shallow water, an increase in hump power is required. This requirement is due to an increase in wavemaking drag, which is influenced by the presence of the ocean bottom. Such proximity prevents the formation (and dissipation) of a normal (deep-water) wave shape and thus results in a higher wavemaking drag. At a speed of approximately half the hump speed another wavemaking drag peak occurs. This sub-hump drag is not as great as the primary hump drag, except in shallow water, where it may be twice as great, and may create problems in propulsor design due to difficulties in developing sufficient low speed thrust.

SES GEOMETRY

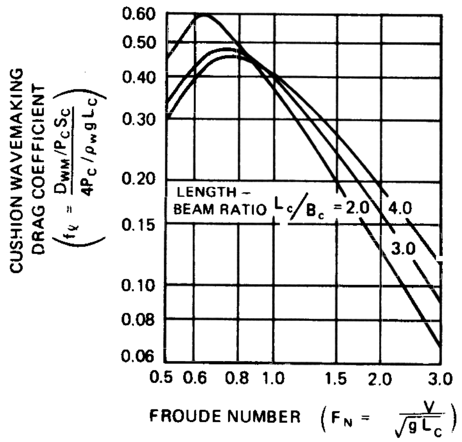


SECTIONAL VIEW

SIDE VIEW

FIGURE 2-

CUSHION WAVEMAKING DRAG COEFFICIENTS



See Reference 1
(References listed in Appendix B)

FIGURE 2-2

SIDEHULL WAVEMAKING DRAG

The following formula can be used to estimate sidehull wavemaking drag:

$$D_{SHW} = C_r \frac{32}{\pi} \left(\frac{B_{SHW}}{L_{SHW}}\right)^2 \left(\frac{h'_s}{L_{SHW}}\right)^2 (\tfrac{1}{2}\rho_w V^2 S_C)\,(L^2_{SHW}) / F_N{}^2 S_C \qquad (2.2)$$

where

D_{SHW} = total sidehull wavemaking drag – pounds
C_r = sidehull wavemaking drag coefficient (see Figure 2-3)
B_{SHW} = maximum wetted sidehull beam – feet
h'_s = average wetted sidehull draft = $h'_a + \dfrac{h_b}{2}$ – feet
L_{SHW} = sidehull wetted length – feet
F_N = Froude number based on sidehull length
h'_a = one-half average wave height (h_w) – feet
h_w = average wave height (3 feet in sea state 3, 9 feet in sea state 6) – feet
h_b = water depression depth due to cushion pressure – feet.

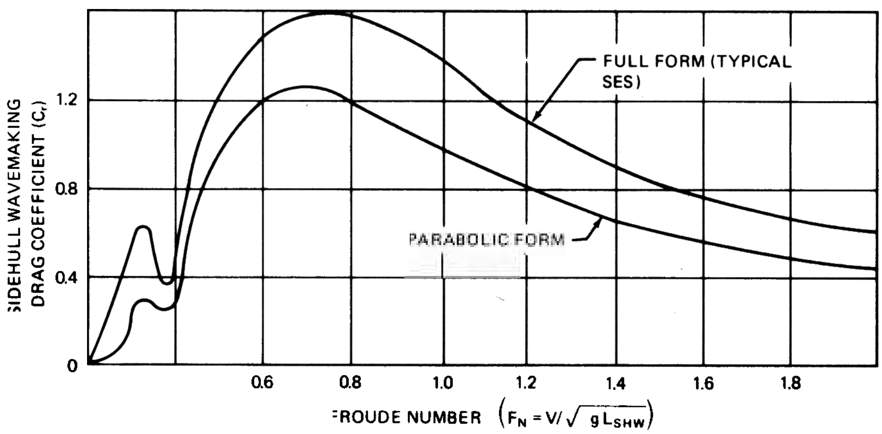
Equation (2.2) can be written much more simply as

$$D_{SHW} = C_r \frac{16}{\pi} \rho_w g\,(B^2_{SHW}\, h'^2_s / L_{SHW}) = 326\, C_r\, (B^2_{SHW}\, h'^2_s / L_{SHW})$$

The sidehull wavemaking drag coefficient, C_r, for parabolic forms and for fuller forms is given in Figure 2-3. For sidehulls with closed sterns, the data for parabolic forms will generally be most appropriate. For sidehulls with transom sterns, the coefficients for the fuller form may be more suitable. Because of the high fineness ratios of the sidehulls and the low immersed depths when on the cushion, sidehull wavemaking drag is usually small.

The water depression depth, h_b, is defined in greater detail in the subsequent discussion of sidehull frictional drag.

THIN SIDEHULL WAVEMAKING DRAG COEFFICIENTS



See Reference 2

FIGURE 2-3

SIDEHULL FRICTIONAL DRAG

Most SES sidehulls will have wetted surfaces on both inner and outer sides. A finite wetted surface is required on the inner sides to minimize air leakage under the sidehulls, particularly during rough-water operation.

Figure 2-1 shows the wetted sidehull geometry for a typical SES. A mean depth, h'_a, is assumed wetted on the inside of the sidehull. The wetted surface is comprised of two areas. The first is the area wetted on both sides of each sidehull by waves. The mean wetted depth, h'_a, is approximately one-half the average wave height. For the two sidehulls the total area is

$$S_1 = 4\, h'_a\, (L_{SHW}) \qquad (2.4)$$

The second wetted area is a triangular area existing only on the outside of the sidehull. The total area for the two sidehulls is

$$S_2 = h_b\, L_{SHW} = L^2_{SHW}\, (h_b/L_{SHW})$$

where

h_b = depth as defined on Figure 2-1

h_b/L_{SHW} = depression angle of the water surface due to cushion.

This angle is given by

$$h_b/L_{SHW} = D_{WM}/(P_C S_C)$$

The total sidehull frictional drag is then given by

$$D_{SHF} = \frac{C_f \rho_w \; V^2}{2} \left[4\, h'_a (L_{SHW}) + \frac{D_{WM} \; L^2_{SHW}}{P_C S_C} \right]$$

$$D_{SHF} = 0.995 \, C_f \, V^2 \, (L_{SHW})^2 \left(\frac{4\, h'_a}{L_{SHW}} + \frac{D_{WM}}{P_C L_C B_C} \right) \tag{2.5}$$

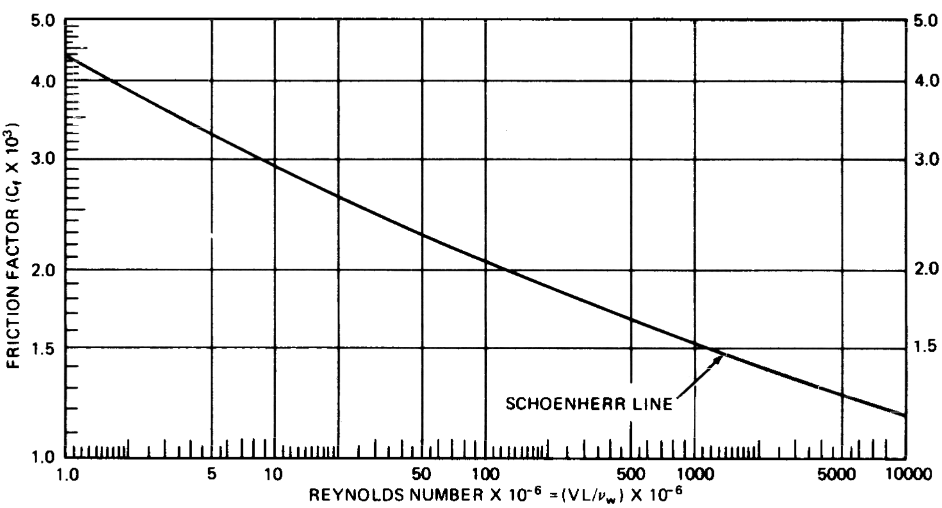
where

D_{SHF} = total sidehull frictional drag – pounds

B_C = cushion beam – feet.

In performing most preliminary SES analyses it is appropriate to base the friction factor, C_f, on the Reynolds number R_e, relative to sidehull wetted length. For smooth surfaces, it is appropriate to use the well-known Schoenherr friction factor shown in Figure 2-4. A suitable roughness factor, assuming a well finished and maintained surface, of approximately 0.1×10^{-3} should be added to the value indicated in Figure 2-4.

FRICTION FACTOR



where ν_w = kinematic viscosity of sea water (1.28×10^{-5}) feet^2/second, at 59°F

FIGURE 2-4

SIDEHULL FORM DRAG

The finite sidehull beam causes a pressure drag and an increase in sidehull frictional drag. These two factors are usually considered together and collectively called form drag. The following formula can be used to estimate form drag:

$$\frac{D_{SP}}{D_{SHF}} = 1.16 \frac{B_{SHW}}{L_{SHW}} \tag{2.6}$$

where

D_{SP} = sidehull form drag pounds.

Equation (2.6) is appropriate for SES sidehulls with closed sterns. For sidehulls with transom sterns, pressure drag, which will be considerably greater than that predictable using Equation (2.6), cannot be readily separated from sidehull wavemaking drag. If a suitable wavemaking drag coefficient is used for transom stern sidehulls, it is probably appropriate to assume D_{SP} is zero. For typical SES's, sidehull form drag will be 5 percent or less of the total sidehull frictional drag.

AERODYNAMIC DRAG

Aerodynamic drag is primarily a form drag caused by the rather blunt bow and stern shape of most SES designs. Therefore, this drag is rather insensitive to variations in Reynolds number. The aerodynamic drag is given by

$$D_A = \frac{C_{DA}\,\rho_a\,(V + V_{HW})^2\,S_r}{2} = 0.0012\,C_{DA}\,(V + V_{HW})^2\,S_r \tag{2.7}$$

wher‹

D_A = aerodynamic drag – pounds

C_{DA} = aerodynamic drag coefficient = $2D_A/\rho_a\,V^2\,S_r$

S_r = reference area for C_{DA} – feet^2

ρ_a = mass density of air (0.0024) – pounds-second^2/foot^4

V = ship speed – feet per second

V_{HW} = headwind velocity (27 in sea state 3, 47 in sea state 6) – feet per second

A drag coefficient based on frontal area is appropriate. Table 2-1 indicates typical values of aerodynamic drag coefficient applicable to the various portions of the SES. Appendages include masts, antennae, and weapons. The frontal area of the hull includes the full beam (B), and the height above waterline (H_S). As a first approximation, the frontal area of the superstructure may be assumed to be one-quarter of the area of the hull, and the drag of appendages may be neglected.

AERODYNAMIC DRAG COEFFICIENTS

	C_{DA}
Poorly streamlined (appendages)	0.70
Average streamlining (hull)	0.35
Well streamlined (superstructure)	0.25

See References 3, 4 and 5

TABLE 2-

Aerodynamic drag will be affected significantly by wind velocity. It will increase rapidly in headwinds; for an 80-knot SES it will be approximately 50 percent higher in a 20-knot headwind than with no headwind. Aerodynamic drag may increase even faster in bow-quartering winds due to higher effective drag coefficients.

MOMENTUM OR RAM DRAG

Momentum drag is the result of reducing the velocity of air brought on board a moving SES to zero with respect to the vehicle. A formula for estimating momentum or ram drag, based on the assumption of complete stagnation of the SES cushion air intake and on a typical contraction coefficient, is as follows:

$$D_m = 2\,D_C \left(\frac{S_g}{S_C}\right) \sqrt{P_C S_C \left(\tfrac{1}{2}\,\rho_a\, V^2\, S_C\right)}$$

$$D_m = 2\,D_C\, S_g\, V \sqrt{\rho_a\, P_C/2} = 0.07\,D_C\, S_G\, V\sqrt{P_C}$$

where

D_m = momentum or ram drag – pounds
S_g = cushion discharge area – feet^2
D_C = air leakage discharge coefficient.

If the total craft lift is provided by the cushion, and a discharge coefficient, D_C, of 0.6 and a discharge area, (S_g/S_C), of approximately 0.0057 based on XR-3 data are used, Equation (2.8) becomes:

$$D_m \cong 0.0002\, S_C\, V\sqrt{P_C}$$

If the quantity of air leakage, Q, in feet^3/second, is known, momentum drag can be written simply as

$$D_m \quad \rho_a QV \quad 0.0024\, QV \tag{2.10}$$

where

$Q \quad 29\, S_g\, D_C \sqrt{P_c}$ (for zero sea state)

$Q \quad B_C\, h_w\, V$ (for non-zero sea state)

h_w average wave height – feet.

The required quantity of air leakage, Q, can be influenced significantly by constraints imposed on the ship's motion, particularly in the heave mode when operating in waves equivalent to about sea state 3 or above. It should also be noted that the discharge coefficient, D_C, is a function of the actual configuration of the equivalent discharge nozzle. A value of $D_C = 0.6$ corresponds approximately to the effect of a sharp-edged orifice; D_C values up to 0.8 may be more appropriate for exit areas with smoother, rounder edges.

Equations (2.8), (2.9) and (2.10) do not take into account the possibility that the momentum of the air leakage provides a net longitudinal force. In reality, there may be considerably greater leakage from the aft seal than from the forward seal, and a net thrust will result.

BOW AND STERN SEAL DRAG

Methods are available for determining bow and stern seal drag based on air cushion vehicle skirt drag. These methods assume flexible seals and finite seal clearance, and under such conditions seal drag can be calculated as follows:

$$D_S \quad \frac{C_{DS}\, \rho_w\, V^2\, S_C\, G}{2} \quad 0.995\, C_{DS}\, V^2\, S_C\, G \tag{2.11}$$

where

D_S total seal drag – pounds

C_{DS} drag coefficient

G ratio of seal length to periphery length $\dfrac{1}{1 + L_C/B_C}$

The drag coefficient is given as

$$C_{DS} = 6.6 \left(\frac{h_w - 2H_C}{L_C} \right)^{1.2} \qquad \text{when } h_w \quad 2H_C \geqslant 0$$

$$C_{DS} \quad 0 \qquad \text{when } h_w \quad 2H_C < 0$$

where

h_w = wave height (trough to crest) – feet
H_C = seal clearance height – feet.

For vehicles with rigid seals, a formula can be derived for total bow and stern seal drag based on data for the XR3A:

$$D_S = 0.0217\, C_{fs}\, \rho_w\, V^2\, S_C\, (B_C/L_C) = 0.043\, C_{fs}\, V^2\, S_C\, (B_C/L_C) \qquad (2.12)$$

where

C_{fs} = the Schoenherr friction factor for the seals based on a Reynolds number calculated from the expression
$R_e = V(0.0433)/\gamma_w$ (see Figure 2-4).

Judgment must be employed in choosing the appropriate equation for seal drag calculation, based on the projected seal characteristics.

APPENDAGE DRAG

SES appendages producing significant drag include control surfaces, such as rudders and ventral fins, and propulsion-mounting appendages. The drag of above-water appendages is assumed to be included in aerodynamic drag. The drag of propulsion appendages may either be included in the calculation of net propulsive efficiency or be treated as a drag component. The specific concern in this section is the calculation of rudder and ventral fin drag, where applicable.

Wedge sections have the most desirable force characteristics. Data for other, lower-drag sections are given in References 6 and 7.

The drag of a control surface with lift (side force) is given by

$$D_{cs} = \frac{C_{Da}\, \rho_w\, V^2\, \ell_a\, h_a}{2} = 0.995\, C_{Da}\, V^2\, \ell_a h_a = D_{fcs} + D_{ccs} + D_{ics} \qquad (2.13)$$

where

D_{cs} = appendage or control surface drag – pounds
C_{Da} = appendage drag coefficient
ℓ_a = appendage length or chord – feet
h_a = appendage depth or span – feet
D_{fcs} = appendage frictional drag – pounds
D_{ccs} = appendage cavity drag (including form drag) – pounds
D_{ics} = appendage induced drag – pounds.

When the control surface is at zero angle of attack, the induced drag is zero.

The appendage friction drag is given by

$$= C_{Dfc}\,\rho_w\ V^2\ \ell_a\ h_a\ =\ 1.99\ C_{Dfc}\ V^2\ \ell_a\ h_a \qquad 2.14)$$

where

C_{Dfc} = the Schoenherr friction factor for the appendage based on appendage Reynolds number, $R_e = V\ \ell_a/\gamma_w$ (see Figure 2-4

The appendage induced drag is given by

$$= \frac{C_{Di}\,\rho_w\ V^2\ \ell_a\ h_a}{2} = 0.995\ C_{Di}\ V^2\ \ell_a\ h_a \qquad (2.15)$$

where

C_{Di} = appendage induced drag coefficient.

The induced drag coefficient can be approximated, for fully ventilated operation, as follows:

$$C_{Di} \cong \frac{\pi}{4}\ \frac{(h_a/\ell_a)}{[3+(h_a/\ell_a)]^2}\ (\alpha_a+\theta)^2 \qquad (2.16)$$

wher

α_a = angle of attack (including side-slip effect) – radians

θ = the half angle of the section = $\tan^{-1}(t_a/2\ \ell_a)$ – radians

t_a = appendage thickness – feet.

As this surface should be fully ventilated, Equation (2.16) is valid when the angle of attack, α_a, exceeds 2 or 3 degrees. For smaller angles, induced drag can usually be neglected.

The cavity drag is given by

$$D_{ccs} = \frac{C_{DC}\,\rho_w\ V^2\ t_a\ h_a}{2} = 0.995\ C_{DC}\ V^2\ t_a\ h_a \qquad (2.\ 7)$$

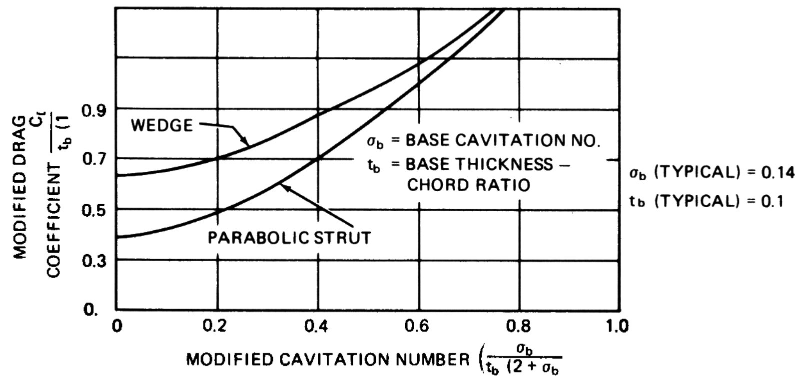
where

C_{DC} = cavity drag coefficient, which is a function of σ_b

σ_b = cavitation number based on base pressure, P_c, and ambient pressure, P_o, at the mean depth of the control surface where $\sigma_b = (P_o - P_c)/1/2\ \rho_w\ V^2$.

Typical values of C_{DC} for zero angle of attack operation may be obtained from Figure 2-5. Typically, a value of 0.125 may be used for C_{DC}. based on an 80-knot speed, a mean depth of control surface of 8 feet, a base thickness-chord ratio of 0.1 and a parabolic strut.

CAVITY DRAG COEFFICIENTS



See Reference 6

FIGURE 2-5

For fully ventilated operation the cavity drag coefficient for sections with wedge or sharp leading edges is given by

$$C_{DC} \cong \frac{\pi}{2} \frac{(h_a/\ell_a)}{(3 + h_a/\ell_a)} \frac{(\alpha_a + \theta)^2}{t_a/\ell_a} \qquad (2.18)$$

As can be seen by comparing equations (2.16) and (2.18), cavity drag will be much larger than induced drag for fully ventilated operation.

TOTAL CRAFT DRAG

Smooth-Water Drag. The total craft drag treated under smooth-water conditions is the sum of the component drag values:

$$(D_{TOT}\ _{SW} \quad D_{WM} + D_{SHW} + D_{SHF} + D_{SP} + D_A + D_m + D_S + D_{cs} \qquad (2.19)$$

where

D_{TOT} total craft drag – pounds
SW smooth-water value

For a craft with transom stern sidehulls, the form drag term, D_{SP}, can usually be omitted. Appendage drag, D_{cs}, will be approximately 7 to 10 percent of the total smooth-water drag. Figure 2-6 shows SES drag composition for a typical SES.

DRAG BUILDUP FOR TYPICAL SES

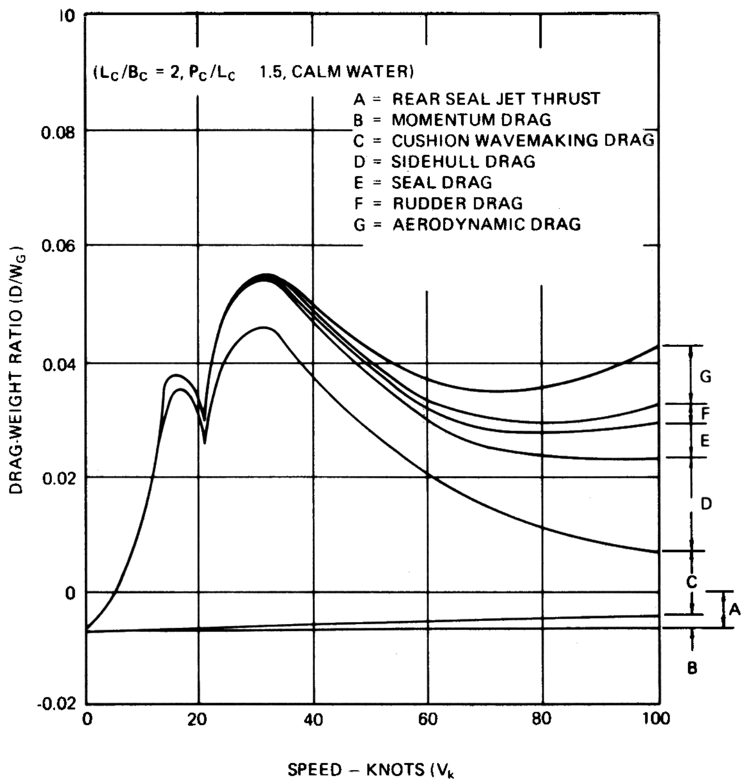


FIGURE 2-6

Rough-Water Drag. In rough-water operation (high sea states) SES drag will be greater, due primarily to increases in seal and sidehull wetted areas and the attendant frictional drag increment, to increases in air leakage rate and momentum drag, and to increases in headwind velocities and aerodynamic drag. Operation in quartering or beam seas can cause increased appendage drag and increased wavemaking (actually pressure) drag. Operation in oblique winds will cause increased aerodynamic drag.

It is difficult to predict accurately the effect of rough water on such critical drag-producing components as sidehulls and cushion seals. Two procedures are presently in general use: in the first, one-half the average wave height is included in the wetted depth, h'_a, of Equation (2.4); in the second, based on test data, the governing parameter for rough-water drag increase is the wave height-cushion length ratio, h_w/L_C. The wave length-cushion length ratio should also be important, but this is determined by the ratio, h_w/L_C, for most sea conditions. Thus, drag increment can be assumed to be the increment in drag-weight ratio as determined from the appropriate test data at the corresponding wave-height ratio and Froude number. The drag increment is then calculated as follows:

$$\Delta D_{RW} = (D_t)_{MRW} - (D_t)_{MSW} \tag{2.20}$$

where

ΔD_{RW} = rough-water drag increment – pounds
D_t = total measured drag – pounds
MRW = measured value of rough-water drag
MSW = measured value of smooth-water drag.

Under either procedure for estimating rough-water drag, it is important to take into account the increased air leakage required to keep heave accelerations to an acceptable level, as well as the increased momentum drag associated with the increased cushion air requirement.

For operation in rough water the total drag is equal to the sum of the total smooth-water drag and the rough-water drag increase. The rough-water drag increment can be calculated either as an overall correction or as incremental corrections of the individual smooth-water drag components.

At 80-knots, the rough-water increment including a headwind allowance will be approximately 50 percent of the smooth-water drag for sea state 3 conditions, and approximately 120 percent for sea state 6 conditions. At 40-knots, the rough-water increment including the headwind allowance will be approximately 10 percent for sea state 3 and approximately 20 percent for sea state 6.

Chapter III
STRUCTURE

Structural loads and the resultant structural weight are second in importance only to the lift-drag ratio for a weight-limited vehicle such as the SES. For all SES's designed in the near future, the weight of the structure will be on the order of 25 to 30 percent of the total gross weight of the ship.

DESIGN CRITERIA

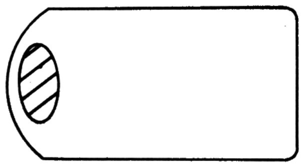
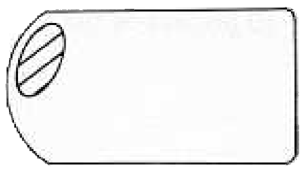
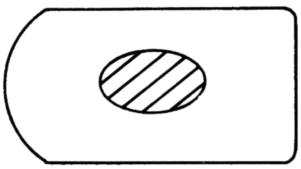
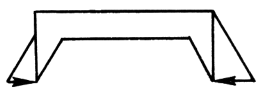
Structural **Loads.** The first step in the structural design process is to predict the loadings to be encountered under the various modes of operation – cushionborne, hullborne, emergency, and ship handling.

In the cushionborne mode the forces generated by wave impact are the most significant and also the most difficult to assess. These forces vary with the attitude of the craft relative to the oncoming wave; the ship velocity relative to the water; the ship characteristics at the point of impact, including the hull configuration and stiffness, and the presence or absence of flexible seals. The design pressures exerted on the exposed surface by the wave impact are currently estimated on the basis of experience with seaplane and planing hulls, together with a limited amount of operational data on existing air cushion vehicles. At the present time the design data given in Table 3-1 are recommended for use in the design of SES's intended for open-ocean operation. Design loads or pressures for components of the hull structure not individually covered in Table 3-1 are given in Table 3-2.

The hullborne mode of the SES is similar to that of a catamaran. The ship is subjected to longitudinal bending in head or following seas, spanwise bending in beam seas, and torsion in quartering seas. Statistical methods developed in recent years are being increasingly used in estimated loading. However, for estimating bending moments and shear loads it is still standard naval architectural practice to use static loads acting on the ship when it is poised in an assumed wave profile. This approach has been employed in the past for SES and ACV designs. The method of deriving the "standard" bending moments and shear loads is well established (see Reference 10). The maximum bending moment can be compared with the bending moment due to bow impact in the cushionborne mode. If the latter is obviously dominant, the hullborne case need not be calculated further. Previous studies (References 3 and 11) indicate that, for small craft, bending moments and shear loads are less critical in the hullborne mode than in the cushionborne. For large craft, however, the situation is reversed. The crossover point is a ship length of about 250 feet, depending on the magnitude of the impact loads used in designing for the cushionborne mode.

For the SES as for surface ships, the design pressure on the forward bottom plating must be increased to counter the effects of slamming. Similar increases in design pressures must be provided in areas subjected to frequent wave impacts. (These considerations are reflected in Tables 3-1 and 3-2) In some cases the structure required to withstand the local impact pressure may also be sufficient for resisting hull girder bending.

CUSHIONBORNE LOADING CONDITIONS

Condition	Description	Wave Slope	Heading	Design Speed	Load Magnitude	Maximum [3] Plating Pressure
Symmetrical bow impact		15°	0°	V	$2W_G$	P = V (psi) (knots)
Unsymmetrical bow impact		15°	30°	V	$2W_G$	P = V (psi) (knots)
Center impact		0°	0°		$0.5W_G$	P = 1500 psi
Side impact		10° impact side 90° resistive side		0	Corresponding to sea level with deck. Half of impact side.	Corresponding to hyrostatic head

See Reference 8

Notes:
1. All loading conditions shall include 1-g cushion loading in addition to water impact load
2. The basic design gross weight shall apply.
3. Related to load magnitude by pressure/area distribution curves.

TABLE 3-

Passengers and crew must be protected from injury that might result from an emergency condition such as sudden sharp maneuvering or failure of the lift system. The main consideration is to ensure that seats, seat belts, shoulder harnesses, and equipment liable to injure personnel if loosened remain secured, and that inflammable liquids remain contained. Design accelerations applicable to emergency conditions in a passenger compartment are summarized in Table 3-3.

The following conditions dictate the criteria for ship-handling loads:

a. Towing. The towing provisions shall be capable of accepting a towing force equal to 0.2 of the design weight.

b. Mooring. The mooring provisions shall be capable of satisfactorily withstanding winds up to 80 knots in any direction.

CUSHIONBORNE DECK LOADS

Location	Design Pressure
Weather Deck	500 pounds/foot^2
Cargo deck	
1) Storage area	150 pounds/foot^2, or actual load x (η_z)
2) Wheel load	2000 pounds on circular wheel area of 25 in^2
Crew rest area	100 pounds/foot^2
Centralized operating station	150 pounds/foot^2
Storeroom	200 pounds/foot^2, or actual load x (η_z)

See Reference 9 where, η_z = local inertia load factor.

TABLE 3-2

DESIGN ACCELERATIONS FOR EMERGENCY CONDITIONS

Direction	Acceleration (g's)
Upward	
Downard	4
Forward	6
Backward	3
Sideward	

See References 3 and 11

TABLE 3-3

c. Handling Impact. The craft shall be capable of satisfactorily withstanding localized handling impacts of 5 feet per second without permanent structural deflection.

d. Drydocking Loads. Provisions shall be made to permit safe ingress, storage, and egress of the craft relative to the drydocking facility.

Structural Safety Factors. The purpose of incorporating a factor of safety is to provide a reasonable margin to account for unknowns in the design process. These unknowns are inherent in the determination of applied loads and may also be a function of imperfections in the analytical methods used to determine stresses, fatigue effects, stress concentrations at discontinuities and hard spots, lack of perfect homogeneity of the material, and effects of the environment on the material.

The magnitude of the factor of safety is further influenced by the importance of the structural member being designed, the consequences of failure, and the limitation placed on structural weight. Inclusion of a large factor of safety may result in unacceptably high structural weight.

The factors given in Table 3-4, based partially on standard ship design practice, are presented for advanced design purposes. In this table, the factor of safety *(FS)* is defined as:

$$FS = \frac{\textit{stress in member at start of failure}}{\textit{stress in member due to applied loads}}$$

STRUCTURAL SAFETY FACTORS
(BASED ON YIELD POINT, BUCKLING STRENGTH OR ENDURANCE LIMIT)

Cause of Failure	Factors for Various Operating Conditions			
	Cushionborne	Hullborne	Emergency	Ship Handling
Yielding by:				
f_1	1.5	1.15	1.0	1.5
$f_1 + f_2$	1.2	1.2	1.0	1.5
$f_1 + f_2 + f_3$	1.0	1.0	1.0	1.5
Instability	2.0	1.75	1.5	1.5
Fatigue	1.5	–	–	–

Note f_1 = primary stress due to hull bending

f_2 = secondary stress due to bending of grid structure between major watertight bulkheads

f_3 = tertiary stress due to bending of plate panel

See References 8 and 9

TABLE 3-4

Structural Stiffness. Adequate strength is assured by the use of proper factors of safety, but their use does not address the entirely different consideration of structural stiffness. The structure must not be subject to excessive elastic deformations which would change geometry and lessen the load-carrying capacity of the structure. Excessive hull deflections could also cause binding and damage to propulsion shafting, piping, and other mechanical components. Structural stiffness can be varied to influence the dynamic response of the hull to wave impact or to propeller excitations.

During the early stages of design, stiffness requirements can usually be met by limiting elastic hull deflections. The maximum allowable deflections for various materials are shown in Table 3-5.

MAXIMUM ALLOWABLE DEFLECTIONS OF HULL MATERIALS

Material	Maximum Deflection
Steel	L/240
Titanium alloys	L/200
Aluminum alloys	L/160
Glass-reinforced plastic (GRP)	L/100

Note: L = length or larger dimension of the structure under consideration. consideration.

See Reference 8

TABLE 3-5

For conventional ships, empirical formulas are available for calculating hull vibration frequencies quickly with reasonable accuracy. The analytical methods described in Reference 10 can be adapted for estimating hull frequencies for an SES.

BENDING MOMENT AND SHEAR

Cushionborne Operation. Various methods can be adapted to analyses of the dynamic response of the SES hull to impact loads. Four types of forces, namely, inertia, elastic, damping, and external forces, are considered in a typical dynamic analysis. Such analysis includes a complicated and time-consuming process for formulating force terms and can be made only by digital computer. It is, therefore, not practical for use in advanced design. A simplified approach for estimating primary bending moments and shear forces is described in the example below.

In this approach, the design external loads are assumed to be balanced by the inertia forces only, the ship being assumed to be a rigid body in determining acceleration. Consider an SES with the following particulars:

W_G, gross weight = 4,000 kips
L, overall length = 220 feet
B, overall beam = 114 feet
P_C, cushion pressure = 250 pounds/foot^2
LCG, longitudinal center of gravity = 110 feet from F.P. (forward perpendicular)
K, mass radius of gyration = 63.5 feet
V_D, design speed = 80 knots in sea state 3.
One kip = 1000 pounds

As a first approximation the LCG can be assumed to be between the ship's mid-length and 5 percent of ship length aft of mid-length, and the radius of gyration can be assumed to be *L/4*, if more precise values are unavailable.

The ship being assumed to be a rigid body, the equations of motion of the craft may be written as

$$P \quad W_G \, a_z \tag{3.1}$$

$$T \quad I\alpha$$

where

- W_G ship gross weight
- $\overline{B}$ resultant upward force due to buoyancy and cushion pressure
- $\underline{P}$ vertical resultant force = $W_G \, \eta_z$
- $\overline{T}$ resultant moment about C.G. = $W_G \, \eta_z \, \ell$
- η_z impact load magnitude in g's
- a_z vertical acceleration of C.G. in g's
- a_x linear acceleration in g's at point x feet from C.G.
- α angular acceleration of the craft – radian/second^2 = $(a_x - a_z)\, g/x$
- mass moment of inertia of the craft about axis passing through the C.G and perpendicular to plane of motion – pounds-second^2-feet – $\dfrac{W_G \, k^2}{g}$
- K radius of gyration about the same axis

Substituting for P, T I, and α in Equations (3 and (3.2)

$$a_z \quad \eta_z$$

$$a_x \quad \eta_z \left(1 + \frac{\ell_x}{k^2}\right) \tag{3.4}$$

A plot of Equation (3.4) is shown in Figure 3-1

IMPACT LOAD AND ACCELERATIONS

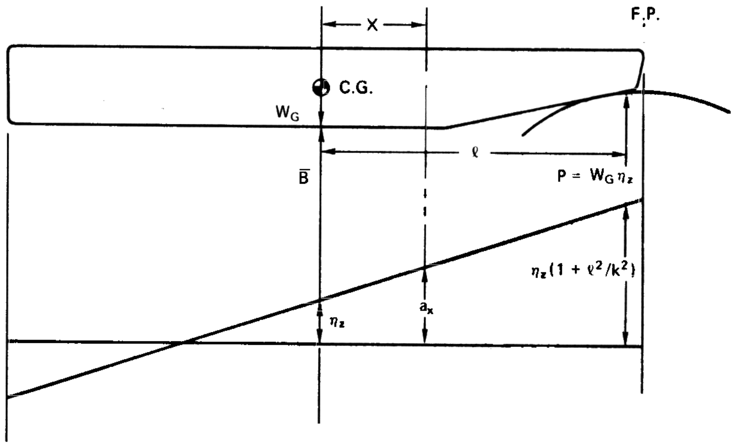


FIGURE 3-1

Based on Table 3-1 for the bow impact case,

$$\eta_z = 2g \text{ and } p = \text{maximum plating pressure} = 80\ psi.$$

Hence

$$P = W_G\ \eta_z = 4{,}000 \times 2 = 8{,}000\ kips.$$

For the sake of simplicity, the following assumptions are made; obviously, these may not be valid for an actual ship:

a. Weight distribution is uniform over the ship length.

b. The acceleration due to gravity is balanced by the buoyant force (including cushion pressure force) such that its effect on the shear force and the bending moment is negligible.

c. The impact force, P, is a concentrated load applied at the F.P., i.e., ℓ = 110 feet.

The ship is divided into ten stations. The acceleration, a_x, at each station is calculated from Equation (3.4) and plotted in Figure 3-2. The problem can now be solved by a tabular calculation, as shown in Table 3-6. The results are plotted in Figure 3-3.

As an alternative, Figures 3-4 through 3-11 may be used. These figures depict the results of a design study using procedures similar to the one just described. In addition to information on the longitudinal bending moment and shear force, these figures include information on the spanwise bending moment and shear and the torsional moment due to unsymmetric impacts. The hull girder design based on longitudinal strength will in general exhibit adequate strength for primary spanwise bending; however, a quick check may be made by using the information contained in these figures, and additional transverse bulkhead cap material provided, if required. For a ship with large openings in the hull envelope, torsion may become critical, and the design must take into account the torsional moments shown in Figure 3-11.

ACCELERATION

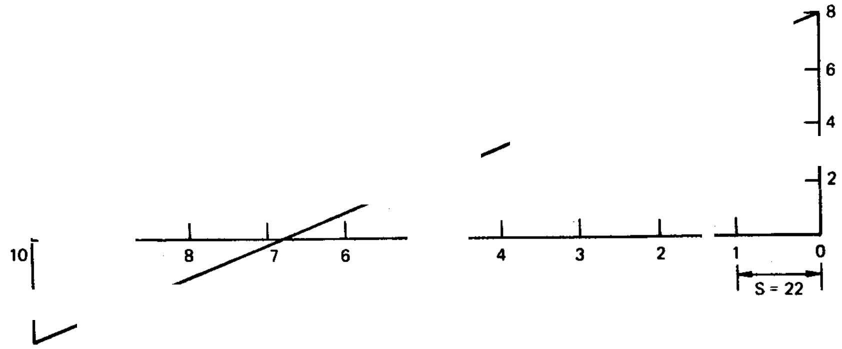


FIGURE 3-2

SHEAR FORCES AND BENDING MOMENTS

							8	9	10
Station (S = spacing = 22 feet)	Weight (kips/foot)	a_x (g) ↑(+)	Load/ft (kips/foot) ↑(+)	Av. load/ft. (kips/foot) ↑(+)	δ (Shear) (kips)	Shear (kips)	Av. Shear (kips)	δ BM (foot/kips)	M_B (foot/ki
Operation			-(2) x (3)		(5) x S	Σ(6)		(8) x S	Σ(9)
					P = +8000				
	18.2	8.0	−145.6	−134.8		+8000	+6568		
	18.2	6.8	−124.0	−113.0	−2965	+5035	+3790	+144,500	+144,500
	18.2	5.6	−102.0	− 91.1	−2490	+2545	+1542	+ 83,500	+228,000
	18.2	4.4	− 80.2	− 69.3	−2005	+ 540	− 223	+ 33,950	+261,950
	18.2	3.2	− 58.3	− 47.4	−1525	− 985	−1505	− 4,910	+257,040
	18.2	2.0	− 36.4	− 25.5	−1040	−2025	−2306	− 33,100	+223,940
	18.2	0.8	− 14.6	− 3.6	− 562	−2587	−2627	− 50,740	+173,200
	18.2	−0.4	+ 7.3	+ 18.3	− 79	−2666	−2464	− 57,800	+115,400
	18.2	−1.6	+ 29.2	+ 40.1	+ 405	−2261	−1819	− 54,300	+ 61,100
	18.2	−2.8	+ 51.0	+ 62.0	+ 883	−1378	− 689	− 40,000	+ 21,100
	18.2	−4.0	+ 73.0		+1365	ERROR −13		− 15,150	ERROR 5950

TABLE 3-6

LOADING, SHEAR, AND BENDING MOMENT

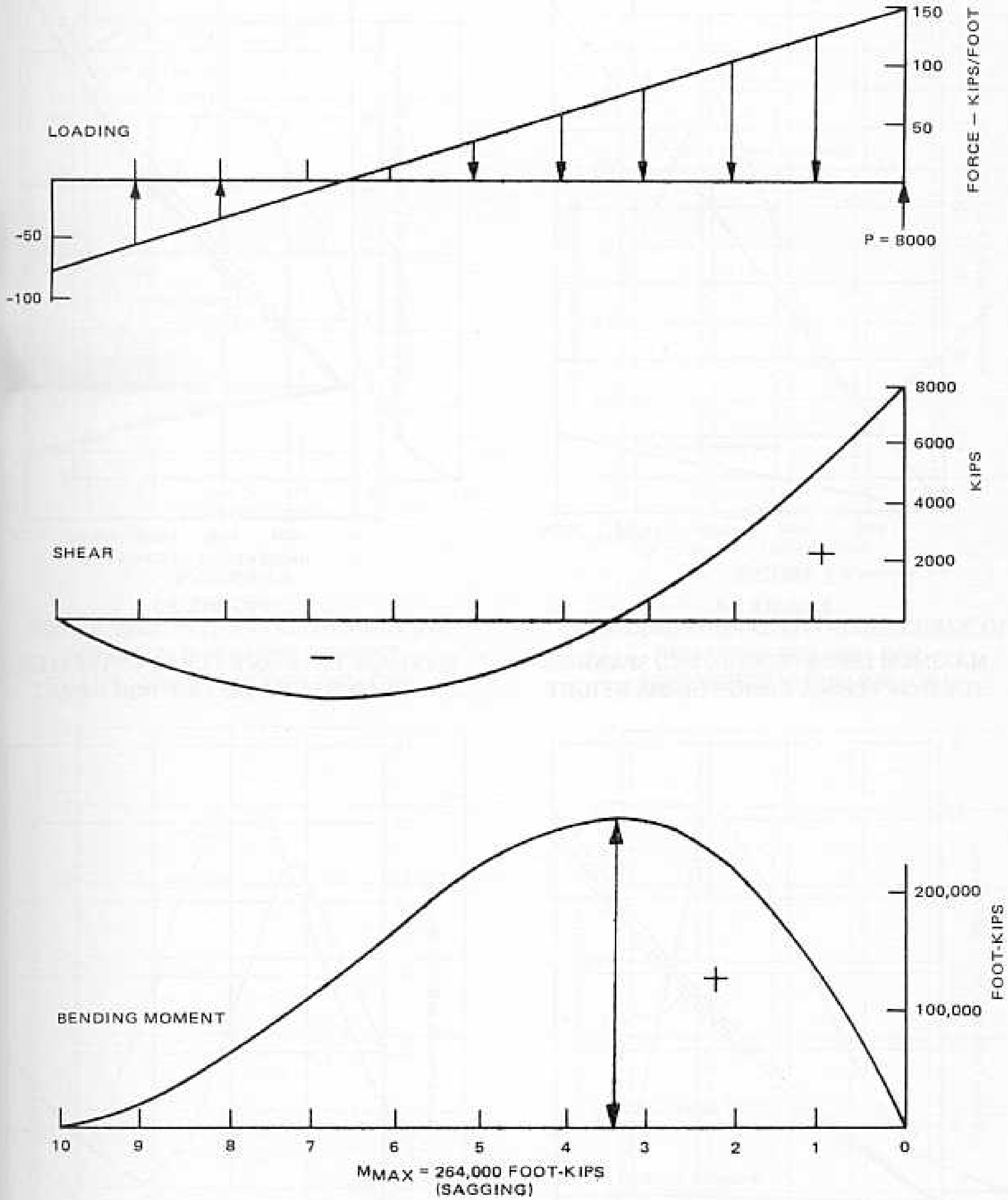


FIGURE 3-3

MAXIMUM LONGITUDINAL AND SPANWISE SHEAR VERSUS DESIGN GROSS WEIGHT

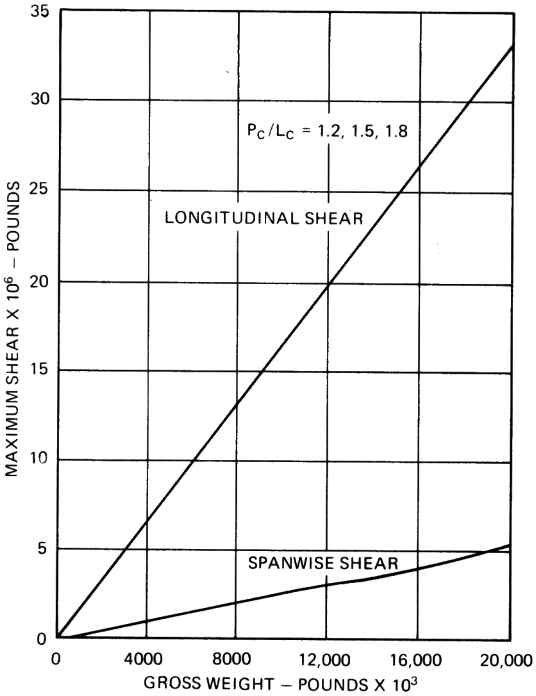


FIGURE 3-4

MAXIMUM LONGITUDINAL AND SPANWISE BENDING MOMENT VERSUS DESIGN GROSS WEIGHT

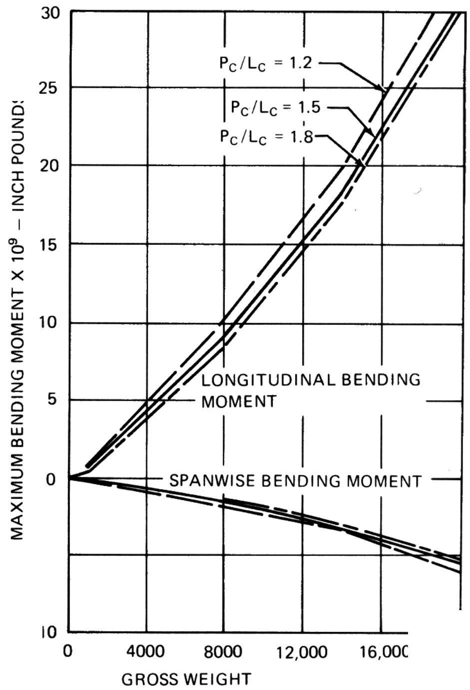
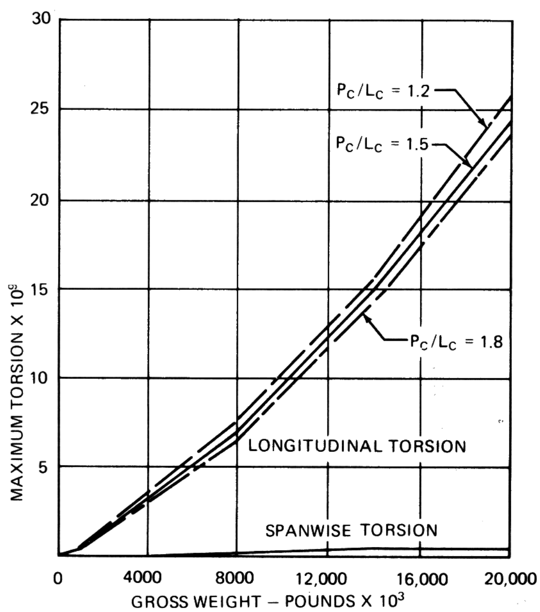


FIGURE 3-5

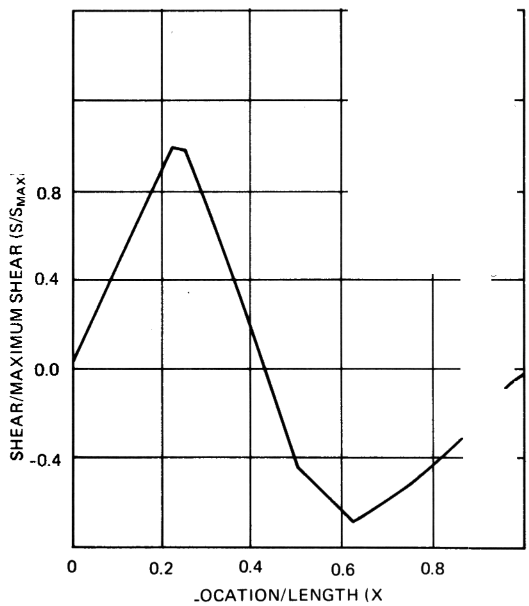
MAXIMUM LONGITUDINAL AND SPANWISE TORSION VERSUS DESIGN GROSS WEIGHT



See Reference 8

FIGURE 3-6

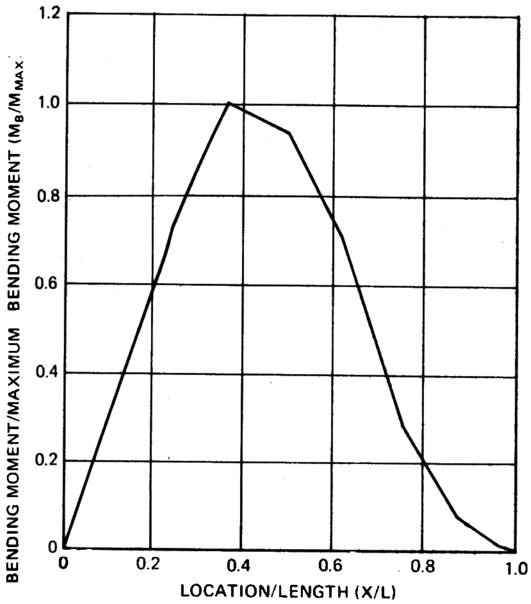
MAXIMUM ENVELOPE CURVE OF IDEALIZED SHEAR FOR SYMMETRIC BOW IMPACT



See Reference 8

FIGURE 3-7

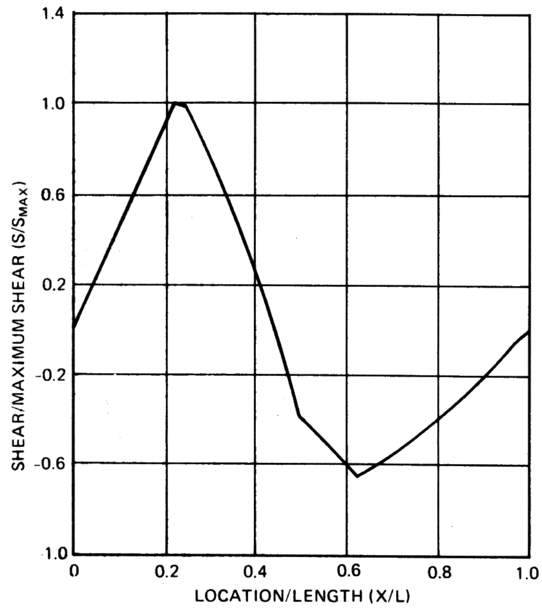
NON-DIMENSIONAL ENVELOPE CURVE OF IDEALIZED BENDING MOMENT FOR UNSYMMETRIC BOW IMPACT



See Reference 8

FIGURE 3-8

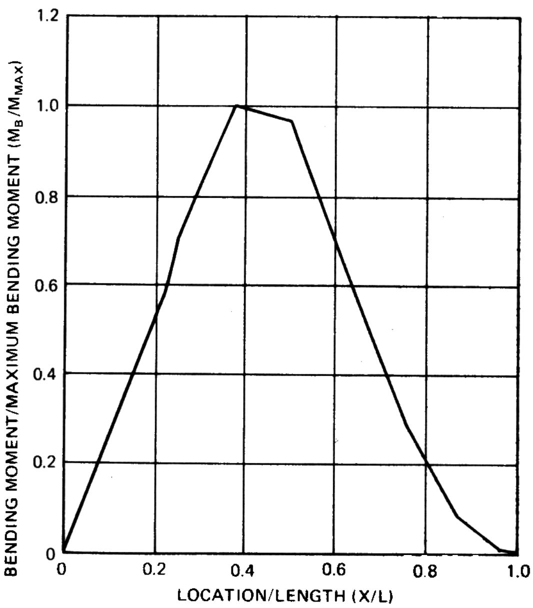
NON-DIMENSIONAL ENVELOPE CURVE OF IDEALIZED SHEAR FOR UNSYMMETRIC BOW IMPACT



See Reference 8

FIGURE 3-9

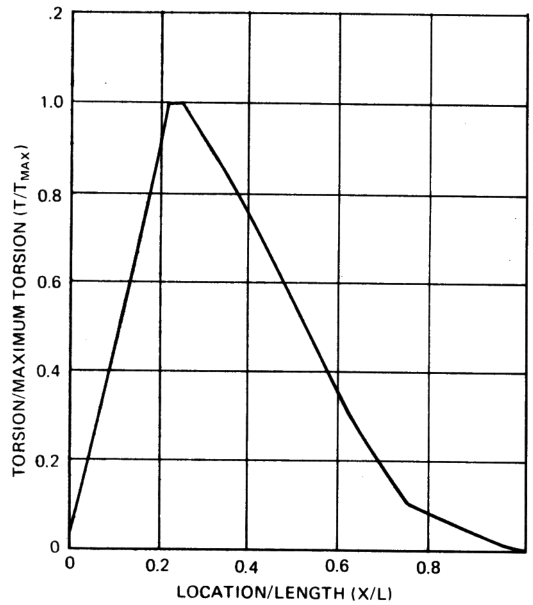
NON-DIMENSIONAL ENVELOPE CURVE OF IDEALIZED BENDING MOMENT FOR SYMMETRIC BOW IMPACT



See Reference 8

FIGURE 3-10

NON-DIMENSIONAL ENVELOPE CURVE OF IDEALIZED TORSION FOR UNSYMMETRIC BOW IMPACT



See Reference 8

FIGURE 3-11

Hullborne Operation. As a first estimate, the maximum longitudinal bending moment may be approximated by

$$M_{MAX} = \frac{W_G \times L}{16}$$

where

M_{MAX} = maximum longitudinal bending moment – foot-kips
W_G = ship gross weight – kips.

For a 4,000-kip gross weight ship with an overall length of 220 feet, the bending moment will be

$$M_{MAX} = \frac{4{,}000 \times 220}{16} = 55{,}000 \textit{ foot-kips.}$$

This is not critical when compared with the previously described (see Figure 3-3) cushionborne impact, which has a bending moment of 264,000 foot-kips.

If the hullborne case becomes the more critical based on this estimate, this finding should be rechecked, using more exact calculations. In the hullborne mode the SES is similar to a displacement ship, and the standard naval architectural strength calculations described in Reference 10 can be used. The commonly used "standard" wave profile is a trochoidal wave of length, λ, equal to the craft length and wave height, h_w, equal to 1.1 $\sqrt{\lambda}$ (see Figure 3-12).

LOADING CONDITION FOR LONGITUDINAL BENDING MOMENT

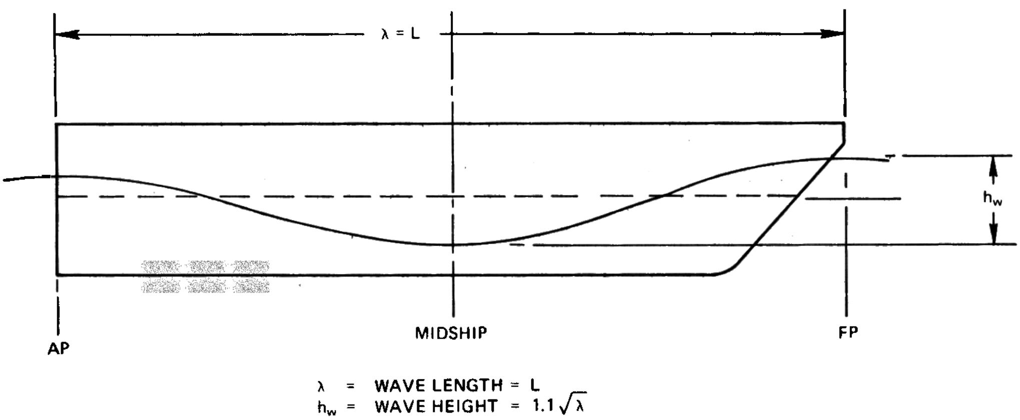


FIGURE 3-12

The maximum spanwise bending moment may be approximated by

$$M_{s_{MAX}} = 30 \times f \times L_c \tag{3.5}$$

where

$M_{s_{MAX}}$ = maximum spanwise bending moment – foot-kips
f = value obtained from Figure 3-13.
L_C = cushion length of the craft – feet.

VALUES FOR "f" IN EQUATION 3.5
(Derived From American Bureau of Shipping Data for Barges)

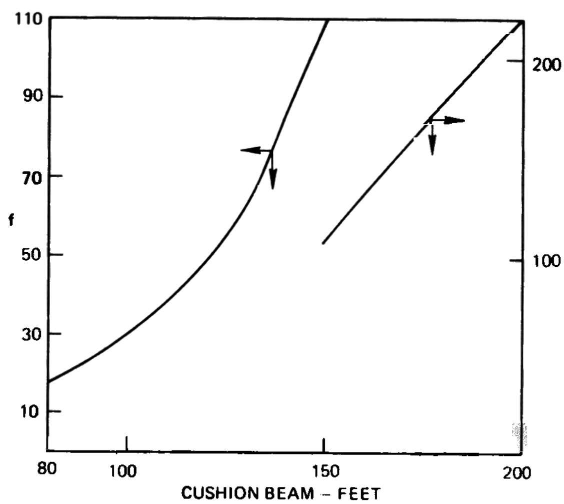


FIGURE 3-13

For the above example, the cushion length and cushion beam of the craft are approximately 175 feet and 82 feet, respectively. From Figure 3-13, an f of 18.6 is obtained. Hence,

$$M_{s_{MAX}} = 30 \times 18.6 \times 175 = 98{,}000 \textit{ foot-kips}$$

This is larger than the value for the cushionborne case of 1.0×10^9 inch-pounds (= 84,000 foot-kips) from Figure 3-5. Since the hullborne case becomes more critical based on the above estimate, this finding should be rechecked, using more exact calculations. For this purpose, based on studies of catamarans, (Reference 12), the craft is poised in a trochoidal beam wave of length, λ, equal to two times the cushion beam and a wave height, h_w, equal to $\lambda/10$ (see Figure 3-14).

In general, torsion is not critical and need not be considered in the advanced design stage. When it becomes necessary to estimate torsional moment, as for a ship with large openings in the decks, the torsional moment in the cross-structure may be estimated based on studies of catamarans, (Reference 12), by

LOADING CONDITION FOR SPANWISE BENDING MOMENT

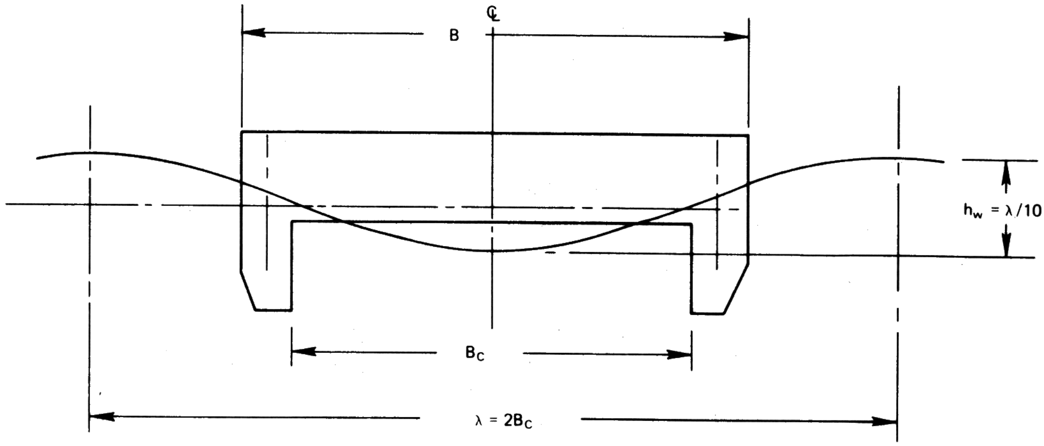


FIGURE 3-14

$$T = 0.04\, W_G\, L$$

where

T = maximum torsional moment – foot-kips.

For a better estimate, the loading condition shown in Figure 3-15 may be used.

LOADING CONDITION OF MAXIMUM TORSIONAL MOMENT

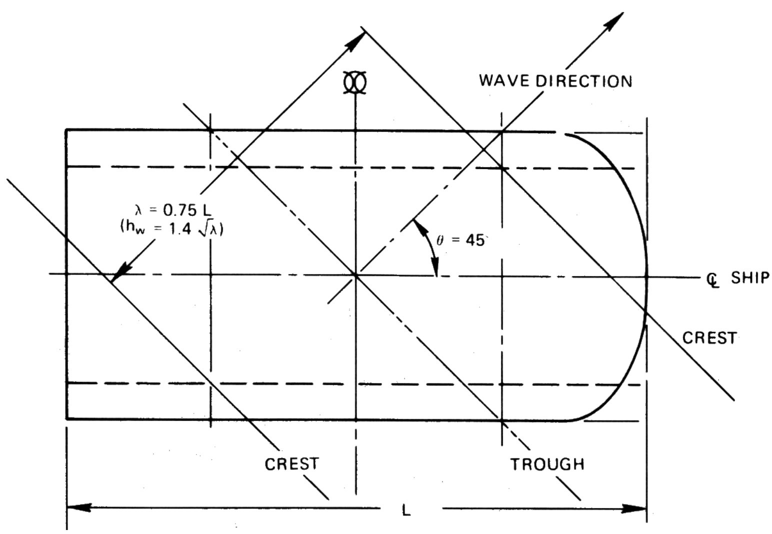


FIGURE 3-15

For advanced design purposes, it can be assumed that structures designed for cushionborne local impact pressure (see Table 3-1) will be adequate for hullborne slamming pressures. The rationale for this assumption is that hullborne speeds are so much less than cushionborne speeds.

STRUCTURAL COMPONENTS

The SES has a semi-monocoque type of hull, which is divided into a number of separate compartments by structural components consisting of plating reinforced by a system of stringers. These compartments are designed to maintain watertightness and structural integrity under the imposed loadings. To achieve a minimum weight-strength ratio, each structural component must be efficiently designed to satisfy its functional requirements. Major structural components are described below.

Shell. The shell is that part of the watertight envelope that keeps the sea out, and it is a principal strength member of the hull girder. The side shell acts as the web of the girder, and the bottom shell acts as the bottom flange in resisting the primary bending and shear loading. The shell may also form part of the torsion boxes resisting torsion loading. The shell must also resist normal pressure loadings from the sea and from liquids stored inside the ship where the shell forms a tank boundary.

Decks. The decks form platforms that carry normal loadings, such as those due to cargo, machinery, and other items on board the ship, and also carry sea loads acting on the topside of the weather deck. The upper decks will act as the top flanges of the hull girder. The weather deck also acts as the uppermost watertight boundary. The decks may also become part of torsion boxes when so designed.

Bulkheads. Bulkheads may be divided into three broad functional groups, namely, "strength" bulkheads, "subdivision" bulkheads, and "tank" bulkheads. Strength bulkheads furnish structural support to the hull. Longitudinal ones act as webs of the hull girder, sharing the shear loading with the side shell. Transverse bulkheads act as strength bulkheads, holding the hull against racking distortions caused by the rolling of the craft and by impact of seas on the side hull and the wet deck; also they act as subdivision bulkheads, dividing the craft into watertight compartments to minimize flooding when the ship is damaged. Both longitudinal and transverse bulkheads also form supports for the decks. Tank bulkheads are those forming the boundaries of water or fuel tanks.

Frames, Pillars, and Girders. Frames, pillars, and girders help to keep the hull girder from deforming under loads. In effect, they transmit and diffuse normal loads. A system of deck girders, for example, will form intermediate supports for the decks and carry the normal loads on these decks to the shell and bulkhead plating.

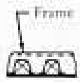
Each major hull girder component – shell, deck or bulkhead – is comprised of stiffened plates with a system of frames, pillars, and girders as the intermediate supports. The main considerations in the design of a hull girder component are:

a. The frames, pillars, and girders should be apportioned to provide effective supports for the stiffened plating. The number and the arrangement of these components are optimized by applying experience and an iterative process to obtain a favorable weight-strength ratio.

b. The material should be efficiently distributed between the plating and the stringers (or stiffeners). The stringer spacing must be so optimized that the plate and stringer combination will minimize the weight-strength ratio. However, when the plating is part of the shear area of hull girder or is an element of a torsion girder, its thickness may be independent of stringer spacing.

Table 3-7 shows various plate and stringer combinations

PLATE AND STRINGER COMBINATIONS

	Aluminum		Steel		Titanium		GRP		Composite	
Alloys	5086	6061	HY-130	ALMAR 362	6A1-2Cb-1Ta-1Mo	6A1-4V ELI*	Polyester Fiberglass Reinforced	Epoxy Fiberglass Reinforced	Metal Core	Balsa Wood or Plastic Core
Temper or Condition	H-111 Extrusion H-34 Sheet H-117 Plate	T6 Sheet Plate Extrusion	O & T Plate Extrusion	O & T Plate Extrusion	Annealed Sheet Plate Rolled-Forms Extrusion	Annealed Sheet Plate Rolled-Forms Extrusion	Mechanical – Hand lay-up	Mechanical – Hand lay-up		
Type of Construction	 Stringers & Plate Integral Extrusion Welded Assy.	 Stringers & Plate Integral Extrusion Riveted Assy.	 Plate & Extrusion Welded Assy.	 Plate & Extrusion Welded Assy.	 Plate & Extrusion Welded Assy.	 Multiple Frames Plate & Rolled Forms Welded Assy.	 Box Core Bonded Assy.	 Box Core Bonded Assy.	 Metal honeycomb Core Metal Skins	 1) Balsa Wood Core 2) Formed Plastic Core Metal Skins
Type No.	1a	1b	1c	1d	1e	1f	4a	4b	6a	6b
Type of Construction	 Formed Stringers & Frame-Sheet Welded Assy.	 Extruded Web & Core Riveted Assy.	 Formed Stringers & Frame-Sheet Welded Assy.	 Multiple Frames Plate & Rolled Forms Welded Assy.	 Formed Stringers & Frame-Sheet Welded Assy.	 Sheet & Rolled Forms Resistance Welded Assy.	 Stringers-Frames & Skin Bonded Assy.	 Stringers-Frames & Skin Bonded Assy.		
Type No.	2a	3a	2b	1g	2c	3b	5a	5b		

*ELI (extra low interstitial) alloy contains reduced amounts of oxygen and iron to provide a high level of notch toughness and ductility.

See Reference 8

TABLE 3-7

Type (1) is the simplest and is essentially that usually employed in displacement ships. In a conventional ship, the stringer spacing is approximately 30 inches, with a minimum plate thickness of about 0.25 inch. For most materials utilized in the SES, however, these proportions will result in far too heavy a structure. For the SES, a smaller stringer spacing, which will result in an acceptable structural weight should be selected. Among Type (1) materials, those suitable for welded assembly are favored because of the ease with which watertight joints can be obtained.

Type (2) is similar to Type (1) except that the frames may be incorporated at the factory. Type (3) is a very efficient section and thus results in a better weight-strength ratio than the previously discussed types. Type (4) is the type of construction suitable for glass-reinforced plastics (GRP). The method of construction and the materials are directly related and must be considered simultaneously. The section properties of this type are similar to those of Type (3). Type (5) is also formed from GRP, with frames incorporated. This type is thus an equivalent of Type (2) in metals. Type (6) is a typical construction for use with composite materials. The weight saving is achieved by using a very light core material.

Foundations. Foundations are fitted to support equipment and to secure it to the main hull structure. The loads transmitted through the foundation include deadweight of equipment; inertia forces due to ship motions; operating loads of equipment, such as torque in the propulsion shafting and the line pull of a winch; dynamic forces due to vibrations; and impact loads caused by sudden accelerations of the ship – for example, those due to wave impact. The foundation should be capable of transmitting and distributing the load into the adjacent structure. The structure to which the foundation is attached may need further reinforcing to take the additional load transmitted by the foundation. In the design of foundations for machinery with moving parts causing pulsating forces, consideration must be given to the prevention of excessive vibration in the foundation and adjacent structure. In general, the material used for a foundation is the same as or similar to that used for the hull so that it can be readily integrated into the hull structure. A unique feature of the SES is the need for foundations at the bow and stern seal attachment points.

MATERIALS

The following subsections provide a brief discussion of the applicability of materials considered to be representative candidate materials for marine use over the next 25 years. The properties of these materials are specified in summary form in Table 3-8.

SUMMARY OF MATERIAL PROPERTIES

Material	Alloy	Temper	Form	Density	Modulus	Mechanical Properties												Remarks
						Base Metal						As-Welded Metal						
						UTS (ksi)		Yield (ksi)		Elong %		UTS (ksi)		Yield (ksi)		Elong %		
						Min.	Typ.	Min.	Typ.	Min.	Typ.	Min.	Typ.	Min.	Typ.	Min.	Typ.	
Aluminum	5086	H-34	Sheet	0.096	10.3×10^6	44	47	34	37	6	10	35*	38	14*	17	–	25	*Minimum values for
		H-117	Plate	0.096	10.3×10^6	40	47	28	47	8	10	35*	38	14*	17	–	25	welded components would
		H-111	Extrusion	0.096	10.3×10^6	37	39	21	25	12	17	35*	38	14*	17	–	25	be annealed properties.
	6061	T-6	Sheet	0.098	10.0×10^6	42	45	35	40	10	12	18*	25	8*	19	–	6	However, values can readily be increased by controlled
		T-6	Plate	0.098	10.0×10^6	42	45	35	40	10	12	18*	23	8*	18	–	6	welding procedures.
		T-6	Extrusion	0.098	10.0×10^6	38	44	35	41	10	15	18*	23	8*	18	–	6	
Steel	HY-130	Quenched and tempered	Plate Shapes	0.280	28×10^6	140		130		18		140		130		18		
	ALMAR 362	Solution treated and aged	Sheet Plate Shapes	0.281	28×10^6		165		155		15		120		105		10	
Titanium	6Al-2Cb 1Ta-1Mo	Annealed	Sheet	0.162	17.0×10^6	130		120		10	13	130		120			6	This weld in the as-welded
			Plate			125		115		10		125		115			8	condition is generally
			Extrusion			120		110		10		120		110			8	stronger than the annealed parent metal.
	6Al-4V ELI	Annealed	Sheet	0.160	16×10^6	130	144	120	136	10	14	130	144	120	136		6	
			Plate			130	136	120	128	10	16	130	136	120	128		8	
			Extrusion			120		110		10		120		110			8	
								Compressive Ultimate										
GRP	Polyester/ glass rein- forced	Box core		0.065	2.2×10^6	30		30										No yield stress. Fatigue limit, 25% UTS
		Hand lay-up		0.062	1.8×10^8	20	23	18	20									Creep limit, 40% UTS
	Epoxy/ glass rein- forced	Box core		0.065	2.8×10^6	36		36										No yield stress. Fatigue limit, 30% UTS
		Hand lay-up		0.060	2.5×10^6	30	38											Creep limit, 40% UTS

See Reference 8.

TABLE 3-8

Aluminum Alloys. The aluminum alloys considered satisfactory for marine use are the heat-treatable 6000-series alloys and the non-heat-treatable 5000-series alloys. Both series require minimum protective coatings, generally only anti-fouling paint for underwater areas, and no coating topside. Protection from cavitation and velocity erosion may also be required.

For a number of years the basic aluminum alloy with good marine corrosion resistance and high mechanical properties was 6061-T6. This alloy is not suitable for welded construction, since the high mechanical properties that are achieved by heat-treatment are lost during the welding process. However, the resistance of this alloy to the marine environment, along with its ability to be extruded into the complex thin-gauge web-core shape (which is not possible with the 5000-series alloys), makes it an excellent selection. It is usually fabricated from integrally extruded shapes and joined by mechanical fastening. The problem of obtaining complete watertightness complicates the fabrication procedure.

The 5000-series alloys, under development for marine use for the past several years, have been improved to give excellent corrosion resistance in the marine environment and to offer superior welding characteristics. The non-heat-treatable alloys have two distinct advantages over the heat-treatable alloys: better ductility in the welds and less loss in strength when welded. Some service problems have occurred with the 5456 alloy; however, with the recent development of new tempers, designated H-116 and H-117, these alloys are expected to be free from the effects of exfoliation corrosion. A comparison between the 5086 and 5456 alloys indicates these materials to be equivalent in most aspects. The mechanical properties of 5456 are slightly higher than those of 5086 in both the unwelded and the welded conditions, but 5456 has a lower ductility in the weld zones, which tends to cause notches in the structure at the welds. Of the two, 5086 has had more previous marine use and exhibits less susceptibility to stress corrosion.

Among materials that may be available in 1975, consideration should be given to alloys in the magnesium-zinc series (7000). These alloys are readily amenable to welding and fabrication and exhibit good resistance to corrosion; in addition, their mechanical properties are higher than those of 5086. Their resistance to stress corrosion is poor, however, and investigations to alleviate this problem are under way.

Glass-Reinforced Plastics. The use of glass-reinforced plastics (GRP) for marine structures has become widespread, and the possibility of fabricating large ship hulls from GRP has been under active consideration for some time. These GRP materials are of composite construction being produced by laying up successive piles of resin-impregnated reinforcement to the desired thickness and then curing this assembly. Complicated double-curvature surfaces can be easily produced at a low cost. A wide range of mechanical properties can be obtained, depending on the resin system, reinforcement materials, and fabrication procedure used.

In the design of primary structures, various effects must be kept in mind when selecting the materials and corresponding design stresses. Immersion of GRP in seawater will cause a reduction in strength unless the surface gel-coat can be prevented from crazing. Also, the creep and fatigue limits of the material represent stress levels far below the ultimate strength of the material. In addition, GRP materials are lacking in elongation properties and have a low elastic modulus. There is no definite yieldpoint, and design must be based on ultimate strength, with careful consideration given to reinforcing areas of stress concentration. Another concern is that some GRP material is combustible, and only fire retardant laminates are permitted in structural applications, for instance, by the U.S. Coast Guard and U.S. Navy. The performance of GRP in the marine environment has nonetheless been very satisfactory. When fabricated into large structures, this material will produce a unitized construction which is easily maintained.

The materials utilized in the fabrication of GRP are reinforcements and resins. The most common types of reinforcements are mat, woven roving, and woven cloth. Many small boats are made entirely of mat construction; the larger craft, however, and particularly those being developed for the U.S. Navy, are made of woven roving. Recently, most laminates are alternate plies of woven roving and mat that yields a reasonable compromise on strength, stiffness, and cost. The mechanical properties of GRP composites are a function of glass content; therefore, woven roving and woven cloth produce the higher-strength materials. In the fabrication of plastic structures for marine use, polyester resins are generally used. The polyester resins are low cost, easy to handle, and generally adequate for low-pressure molding. The epoxy resins will produce higher-strength material but are relatively more expensive and difficult to handle, and are generally used only in aircraft structures.

The method of fabrication and the materials to be employed are directly related and thus must be considered together. In the production of a large hull structure, such as that of the SES, simple, low-cost methods must be utilized. A requirement for vacuum bagging or heat curling would make use of the GRP structure impracticable. Therefore, the hand lay-up or contact-molded process should be used.

In many instances it is advantageous to separate the structural laminates of GRP by introducing a sandwich or core element. The major advantages are improved elastic stability and buckling strength at reduced weights and, often, less lost buoyancy under damaged conditions. Many materials are used as structural cores for stiffeners and sandwich panels, including wood, foamed plastics, and honeycomb. The selected core material should have the following characteristics: good shear strength and rigidity; ability to bond to facings adequately and with a minimum of difficulty, resistance to deterioration caused by water, fungi, and decay; light weight; and sufficient crushing strength to withstand local loading, such as forklift truck tires rolling on a deck.

Titanium Alloys. The most desirable feature of this group of alloys is the high strength-density ratio of titanium, the highest among commercially available metals. Adverse features are the high material fabrication costs. Alloys 6Al-4V and 6Al-2Cb-1Ta-1Mo have high strength, good fracture toughness, good corrosion resistance, and excellent weldability. Both alloys have about the same strength level, but the 6-2-1-1 has superior fracture toughness. Titanium has outstanding overall corrosion resistance. It resists pitting, crevice corrosion, and erosion-corrosion damage and is a recommended candidate material for propulsion pods and struts and other areas where cavitation erosion is likely to occur.

Steel Alloys. The use of high-strength quenched and tempered steels for SES construction is attractive from a strength-weight viewpoint. However, these alloys are susceptive to corrosion and must be protected in all cases. In considering the use of steel for a vehicle that is weight critical, it must be kept in mind that the availability of quenched and tempered steel is limited to plates of a minimum thickness of 0.188 inch. In the welded condition the strength of these materials in the weld zone is equal to the parent-material properties. A comparison of these steel alloys indicates a general similarity in ease of manufacturing with the major differences being related to cost and strength. The fatigue strength of these steels is degraded when stress raisers are present, particularly when exposed to seawater. To meet strength-weight requirements, these alloys should have yield points of 100 ksi or greater. Based on present trends in alloy development and fabrication technology and on known problems with ultra-high-strength steels, alloy with a yield strength of 150 ksi may reliably be expected to be available in the 1975 period.

Stainless steels offer better resistance to corrosion and erosion than do standard shipbuilding steels. Types 304 and 316 stainless steels have been used frequently in marine applications but generally only for fittings and fasteners. Because these materials are heat-treatable, their strength-weight ratios after heat treatment are greater than those of quenched and tempered steels. However, this strength advantage is lost in the welded condition wherein the strength becomes that of the annealed properties of the material. The availability of stainless steels in thin sheet, as well as in plate and shapes, makes it promising for SES application. In a study conducted by the U.S. Naval Applied Science Laboratory, ALMAR 362, a maraging stainless steel, was compared with types 17-4PH, 304, and 410 and found to be similar to 17-4PH. ALMAR 362 can be readily machined in either the annealed or the hardened condition. It can be welded by all conventional processes used for austentic steels and has excellent ductility.

Composite Materials. A composite panel generally consists of two thin metal sheets of such materials as aluminum alloys, which are separated by a very light core. Many types of cores are used, including metal honeycomb, foamed plastics, and wood. The core material must have good shear strength and rigidity. Great care is required in manufacturing to ensure good bonding between the core and the facings. Attaching the composite panels to each other and to the rest of the structure requires special consideration. This step is usually accomplished by mechanical means, and great care must be taken to ensure sufficient strength at the joints. For the present, such materials are not attractive candidates for hull structural use, but they may find use in such components as lift fans and turbine compressor blades.

MAINTENANCE REQUIREMENTS

The exterior of the SES must be maintained to prevent cavitation erosion, corrosion, and biofouling. These damage mechanisms arise because of the high velocity and operation of the craft over seawater. Brief definitions of these surface damage mechanisms are given below.

Cavitation Erosion: loss of material from the exterior surface due to formation and collapse of bubbles or cavities in high-velocity seawater flowing over uneven or rough surfaces.

Corrosion: loss of material caused by chemical or electrochemical action of seawater on metallic materials.

Biofouling: attachment and growth of marine organisms on surfaces that are immersed in seawater. Fouling is especially detrimental in seawater in which hard-shelled animals (barnacles) become attached to the craft resulting in large increases in drag.

The most serious damage would potentially be caused by cavitation erosion. The possibility of rapid surface cavitation erosion places additional stress on controlling the slower damage mechanisms, corrosion and biofouling, to prevent development of rough areas on critical surfaces.

Surfaces subject to immersion in seawater which are not also exposed to high-velocity flow can be maintained according to procedures developed for conventional ships. The lower ends of the sidehulls that are in contact with water at high speeds, however, must be maintained smooth and free from irregularities at all times. Hence, special protective measures involving some use of coatings and special operating procedures must be considered. For those surfaces remaining above water level at all times, no maintenance will be required, providing readily available corrosion resistant materials are used.

Table 3-9 is an evaluation of the resistance to damage of the candidate materials.

STRUCTURAL ARRANGEMENT

The first step in considering ship structural arrangements is to prepare a sketch of a typical cross section showing the arrangement of all major hull girder components, such as the shell, deck, longitudinal bulkheads, and girders. The beam of the ship, height of the sidehulls, and depth of the main hull should conform to the geometry of the ship under consideration. The bulkheads

MATERIALS FOR HULL APPLICATION

Material	Resistance to Damage Mechanisms		
	Cavitation Erosion	Corrosion	Biofouling
Titanium alloys	Excellent**	Excellent	All allow attachment of marine growths
Aluminum			
5000 series	Very poor*	Very good	
6000 series	Very poor*	Good	
Steel			
Stainless	Very good**	Pitting*	
HY series	Good	Good*	
GRP (glass reinforced plastics)	Very poor*	Very good	

See Reference 8

*Should not be exposed to seawater without a protective coating

**The resistance of these materials to cavitation-erosion is dependent on maintaining the intensity of cavitation-erosion at a low level. Even the best materials are damaged by intense cavitation-erosion.

TABLE 3-9

may have to be located to satisfy internal-arrangement requirements rather than strength considerations.

Once tentative structural arrangements have been established, the next step is to determine the amount of material required for each component so that the hull girder will possess adequate primary strength. An example illustrating this procedure follows.

Figure 3-16 represents the cross section intended to resist the bending moment of 264,000 foot-kips previously obtained (see Figure 3-3) in the 4000-kip ship example problem. Item numbers 1 to 10 are assigned to the various structural members. Sectional areas per unit length are then designated for each item, based on good engineering judgment. Hull material and factor of safety are also selected. An example follows:

Hull material: aluminum alloy 5456-H117

Yield point: 26 kips/inch^2 (welded)

Modulus of elasticity: 10×10^3 kips/inch^2

Factor of safety for primary bending stress: 1.5.

Hence, the required section modulus, Z_{REQ}, is

$$Z_{REQ} = \frac{M_{MAX}}{\sigma\ allowable} = \frac{264{,}000}{26/1.5} = 15{,}200\ in^2\ ft.$$

STRUCTURAL ARRANGEMENT OF MIDSHIP SECTION

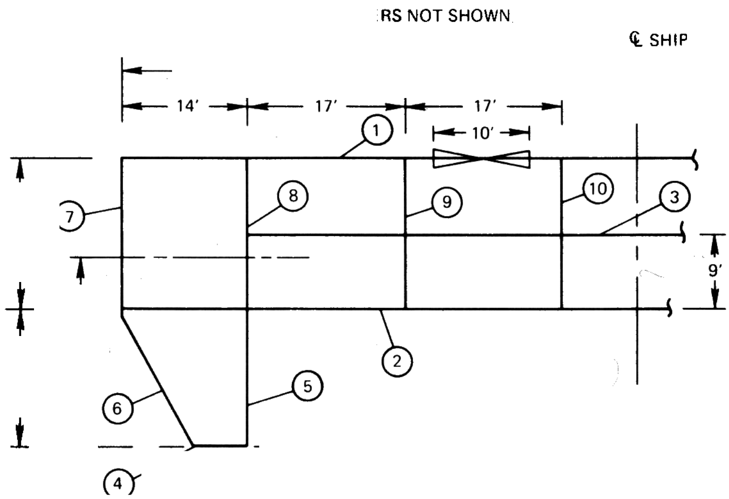


FIGURE 3-16

The calculated results presented in Table 3-10 indicate that the assumed section will have adequate section modulus to resist the bending moment. Similar calculations are performed for other sections along the length of the hull, using in each case the bending moment appropriate for the section being analyzed.

With the sectional area for each component tentatively determined, the next step is to distribute the sectional area efficiently among the structural elements of each component. The various plate and stringer combinations illustrated in Table 3-7 should be considered. An iterative process is required wherein the assumed proportions, width, depth, length, and pitch of each element are analyzed for the in-plane loads due to primary and secondary bending, the buckling strength of plate panels, and the applied loads, including normal pressure. Sufficient iterations should be performed so that a minimum weight structure developed is consistent with reasonable fabrication procedures.

PARAMETRIC WEIGHT DATA

For advanced design purposes, an estimate of structural weight can be obtained from an expression derived empirically from weight data for existing or proposed ACV's and SES's for which the primary structural material is marine aluminum alloy:

$$W_{ST} \quad W \quad [224 \quad '640)\, W_C \quad {}^{.0} \quad {}^{4}\, (P_C/L_C)^{-0.776}] \tag{3.6}$$

where

W_{ST}	structural weight	- pounds
W_G	ship gross weight	- tons
P_C/L_C	cushion density -	pounds/foot

STRUCTURAL PROPERTIES OF MIDSHIP SECTION
(BASED ON STRUCTURAL ARRANGEMENT SHOWN IN FIGURE 3-16)

1	2	3	4	5	6	7	8	9
Item No.	Sect. Area per ft. (in^2/ft)	Total Sect. Area, A (in^2)	VCG,y (ft)	A X y (3)X(4)	$d = (y-\bar{y})$ (ft)	d^2 (ft^2)	Ad^2 (in^2-ft^2)	$i_o = \frac{A}{12}(h^2)$ (in^2-ft^2)
1	7.0	330	34	11,200	13.5	182	60,060	–
2	8.5	485	15	7,270	4.5	20.3	9,846	–
3	4.5	193	24	4,630	3.5	12.3	2,374	–
4	8.5	51	0	0	20.5	420.3	21,435	–
5	7.0	105	7.5	788	13.0	169	17,745	1,970
6	7.5	127	7.5	952	13.0	169	21,463	2,380
7	7.5	142	24.5	3,480	4.0	16	2,272	4,270
8	4.5	86	24.5	2,100	4.0	16	1,376	2,590
9	4.5	86	24.5	2,100	4.0	16	1,376	2,590
10	4.5	86	24.5	2,100	4.0	16	1,376	2,590
TOTAL		1,691	$y = 20.5$	34,620			139,323	16,390

Moment of inertia of total section, $\bar{I} = 2(139{,}323 + 16{,}390) = 311{,}426$ in^2-ft^2

Minimum section modulus, $Z_{MIN} = \frac{\bar{I}}{y} = \frac{311{,}426}{20.5} = 15{,}200$ in^2-ft

Maximum section modulus, $Z_{MAX} = \frac{\bar{I}}{34 - \bar{y}} = \frac{311{,}426}{13.5} = 23{,}000$ in^2-ft

TABLE 3-10

From this equation structural weight is seen to be proportional to ship gross weight and inversely proportional to cushion density. This formulation for structural weight is based on the premise that the SES, viewed as a beam, requires more material to resist a given unit loading as the unsupported span of the beam increases. This may not be precisely the case for small craft of less than 50 tons or so, for which the minimum gauge of material available may be the governing factor in determining structural weight. For purposes of this book, however, the ships of interest will generally be of tonnages larger than this critical value, so the equation is applicable. Figure 3-17 is a plot of structure-gross weight ratio.

As can be seen from Figure 3-17, at $P_C/L_C = 1.5$, Equation (3.6) yields a structure-gross weight ratio of approximately one-quarter.

STRUCTURE-WEIGHT RATIO VERSUS CUSHION DENSITY

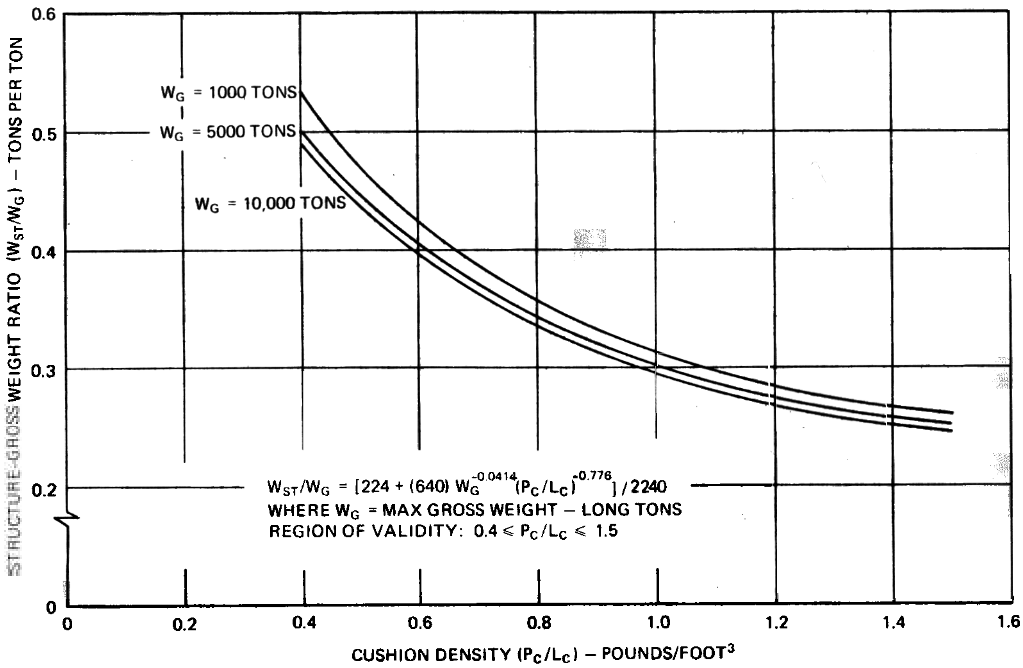


FIGURE 3-17

Chapter IV
PROPULSION

In this chapter the general relationships between the SES propulsion system and the ship's drag and propulsion power requirements are first discussed in terms of propulsion system efficiencies and engine characteristics, such as available power and fuel consumption. Attention is then given to the effect of these characteristics on the ship's endurance and range performance. Detailed descriptions of power plants (fossil-fueled and nuclear-fueled) are also included.

EFFICIENCY AND PROPULSION POWER

The major elements of an SES propulsion system are the prime mover, the transmission, and the propulsor. To establish installed power and fuel requirements for specified performance levels, it is necessary to estimate the individual efficiencies of these elements. Once these have been established, the propulsion system weight and volume requirements can be delineated. Because the relationship between the installed power and the net propulsive force is governed by a number of intervening efficiencies, a brief discussion of these efficiencies follows.

In this handbook, brake horsepower, $(HP)_B$, is defined as the power at the output shaft of the prime mover; shaft horsepower, $(HP)_S$, is the power delivered through the power train to the propulsor. A number of efficiencies reflecting the power train and propulsor system losses can now be defined:

$$\eta_T = (HP)_S/(HP)_B \qquad \textit{(transmission efficiency)}$$

$$NPC = \frac{T_n\ V}{550\ \eta_T\ (HP)_B} \qquad \textit{(net propulsive coefficient)}$$

where

T_n = net propulsor thrust (after accounting for propulsor mounting and hull interaction losses) at speed, V – pounds.

V = ship speed – feet/second.

Typical transmission efficiencies, η_T, are presented in Table 4-1.

Typical net propulsive coefficients, NPC's are presented in Figure 4-1. The range of peak values for the semi-submerged supercavitating propeller is from 0.60 to 0.70, with 0.68 considered to be achievable in the not-too-distant future. Corresponding values for the waterjet are 0.48 to 0.60, with 0.55 recommended for the not-too-distant future. The maximum value of NPC is usually set at cruise speed which results in reduced values at speeds above and below cruise speed.

TRANSMISSION EFFICIENCIES

Transmission Type	Transmission Efficiency
Gears (typical)	0.99 per mesh (0.97 for a typical Z-drive system)
Superconducting electric	0.95
Standard electric	0.90

TABLE 4-1

TYPICAL NET PROPULSIVE COEFFICIENTS VERSUS SPEED

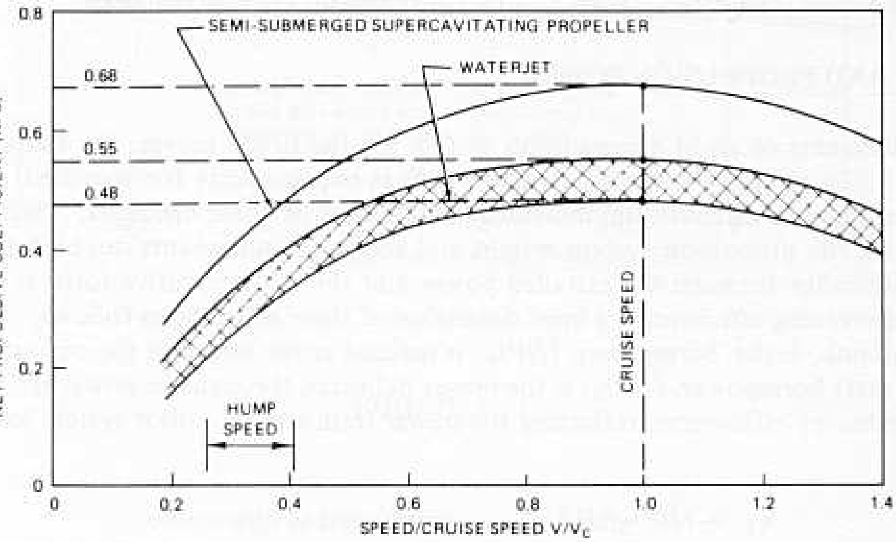


FIGURE 4-1

PROPULSION POWER REQUIREMENTS

Given these considerations, together with the drag data, the propulsion power requirements of the SES can now be developed. The brake horsepower, $(HP)_B$, is of prime interest, since it dictates the propulsion engine selection:

$$(HP)_B = \frac{(1 + K_M)\, D_{TOT}\, V}{550\,(NPC)\,\eta_T} = \frac{T_n\, V}{550\,(NPC)\,\eta_T} \tag{4.3}$$

where

K_M = desired thrust margin (fraction)

D_{TOT} = total drag at speed V – pounds.

To minimize the acceleration time of the ship through hump to cruise or maximum speed, a minimum value for K_M of about 0.20 should be chosen at the hump speed, V_{crit}. At the cruise speed of the ship, V_C, K_M will be zero at the power rating of the engine chosen for the cruise condition. If the brake horsepower indicated by Equation (4.3) at V_{crit} is larger than that indicated at V_C, engine selection is usually based on the maximum intermittent power rating. If $(HP)_B$ is larger at cruise than at hump speed, selection is based on the engine-normal or continuous-power rating. In some cases a satisfactory matching of power (thrust) required and power available will be feasible only by compromising propulsor efficiency, that is, by shifting the peak value of the net propulsive coefficient, NPC, away from the originally selected cruise speed to achieve a better efficiency and an acceptable acceleration margin at hump speed. These matching considerations are shown schematically in Figure 4-2. As an example, the figure indicates that, if the new cruise speed, V'_C, resulting from the rematching of thrust is not tolerable, a propulsion engine of somewhat higher rating is indicated.

PROPULSION SYSTEM MATCHING

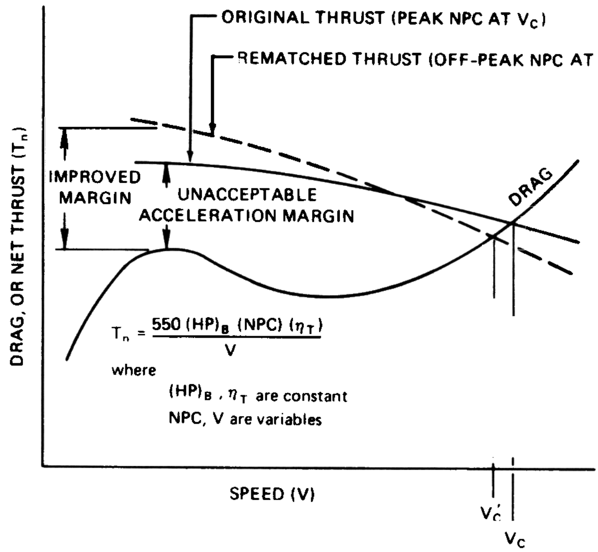


FIGURE 4-2

SPECIFIC FUEL CONSUMPTION

Specific fuel consumption, SFC, reflects the operational efficiency of the ship's power plant. Its determination is basic to accurate estimation of the fuel required for operation in various sea states and at different ranges. For this purpose, curves relating SFC to brake horsepower level, $(HP)_B$, and ship speed, V_k, are required.

The SFC of a gas turbine varies with power-turbine revolution per minute, rpm, and power level. To determine the fuel rate accurately, it is necessary to use the propulsor performance curve to find corresponding values of rpm and power level for each ship speed and gross weight. If such curves are not available in the preliminary stages of design, approximations of propulsor performance curves can be derived by making certain assumptions, as follows. When the propulsor is a waterjet, it can be assumed that turbine speed remains constant with varying power absorption.

The error introduced by using this assumption should not exceed 10 percent. For supercavitating, variable-pitch propellers a more generally applicable assumption can be used in place of the normal cube relationship, that is, that the power absorbed by the propeller is approximately proportional to the 2nd power of the propeller rpm. A non-dimensional curve of this type is shown in Figure 4-3.

POWER VERSUS POWER TURBINE SPEED
(Typical Marine Gas Turbine Engine)

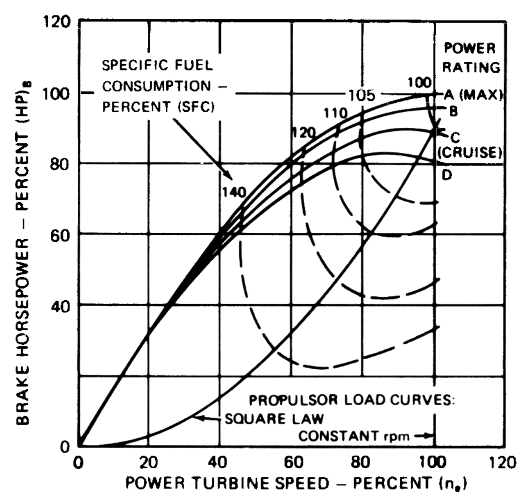


FIGURE 4-3

Data relating SFC to power level can usually be obtained from the manufacturer's performance curves for the engine selected. In the absence of such data, the non-dimensional curve of Figure 4-3 can be used. The power turbine speed is chosen for minimum SFC at the cruise power requirement. The SFC for various power levels can be taken from along the indicated rpm curve. If the actual propulsor rpms, n, for various power levels are known, the curve of n vs. $(HP)_B$ can be superimposed on the turbine performance curve and the SFC read along that curve. In this latter case,

$$n = r_T \, n_e$$

where

n = propulsor rotative speed – rpm

n_e = power turbine speed – rpm

r_T = transmission reduction ratio.

Information relating SFC to $(HP)_B$ can be presented more conveniently as shown in Figure 4-4, again in non-dimensional form. This curve, when used in conjunction with a power-speed curve, relates the specific fuel consumption to the speed of the SES. This information is useful in determining range and endurance under various operating conditions.

SPECIFIC FUEL CONSUMPTION VERSUS BRAKE HORSEPOWER
(Typical Marine Gas Turbine Engine Based On Load Curve from Figure 4-3)

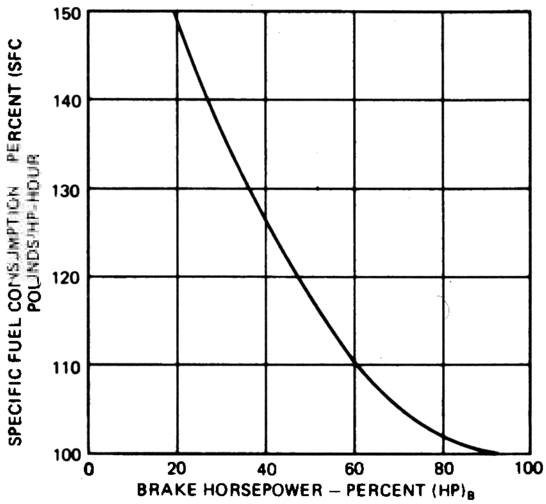


FIGURE 4-4

In the absence of specific manufacturer's data, values of SFC ranging from 0.38 to 0.42 can be assumed for the 100 percent SFC conditions. These values are representative of the performance of the largest modern-technology gas turbine engines suitable for SES propulsion applications in the next several years. If separate, smaller cushion-air fan engines are used, corresponding values of SFC might be on the order of 0.48 to 0.52, since the smaller engines, in general, represent an earlier state of technology. The fan power requirement is usually only about one-quarter that for propulsion.

Engine specific fuel consumption is also related to the rate of flow of the fuel consumed, W_f, as follows.

$$W_f - pounds/hour = (SFC)\ (HP)_B \tag{4.4}$$

MAXIMUM ENDURANCE AND RANGE

SES propulsion system characteristics and, in particular, specific fuel consumption are closely related to SES endurance and range performance. When SES endurance and range can be calculated on the basis of specific data on power available and power required, the approximations resulting from application of the Brequet range equations need to be refined. The calculation of endurance and range under these conditions is discussed below.

Power Available/Power Required Curves. Total power required curves for a particular design of SES can be derived from the basic drag requirements, the fan power requirements, and the accessory power requirements as follows, using the definitions given below:

$$(HP)_{BT} = (HP)_B + (HP)_C + (HP)_{ACC} \tag{4.5}$$

where

$(HP)_{BT}$ = total ship brake horsepower

$$(HP)_B = \frac{D_{TOT}\ V}{550\ NPC\ \eta_T}$$

V = ship speed – feet per second

D_{TOT} = total craft drag – pounds

NPC = net propulsive coefficient at speed, V

η_T = transmission efficiency.

$(HP)_C$ = cushion fan power required

$(HP)_{ACC}$ = accessory (auxiliary) power required

The drag data should be in a form similar to that shown in Figure 4-5, covering the range of interest of gross weights, speeds, sea states, headwinds, and operating modes (on-cushion or displacement mode). The propulsion power required is obtained by applying Equation (4.5) to these drag data. Then, by adding the lift system and accessory power requirements to the propulsion power requirements, curves for total power required are derived, as illustrated in Figure 4-6. The total power available for propulsion and the fan and accessory power requirements are now superimposed on the same figure, thus completing the basic power available/power required curves. The intersection of these curves is the design speed of the SES, as characterized by specified values for gross weight and operating conditions, such as sea state and headwind.

TOTAL DRAG VERSUS SPEED

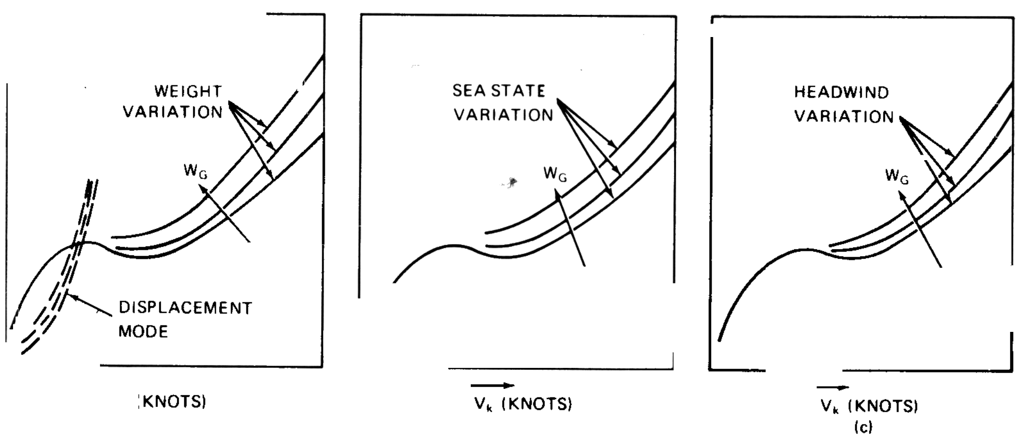


FIGURE 4-5

To simplify the remainder of this discussion, it is assumed that the total available power is derived either from one engine or from several identical engines. In cases where this is not so, the analyses described must be repeated for each of the individual power sources, as required, and the results aggregated to arrive at the desired performance predictions.

POWER AVAILABLE/POWER REQUIRED CURVES
(For Given Weight, Sea State, and Head Wind)

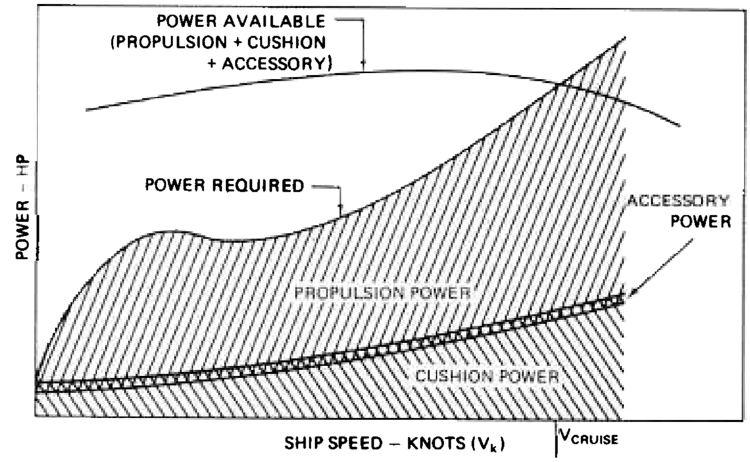


FIGURE 4-6

Endurance **Performance.** The endurance of the SES under stated operating conditions is calculated by using power available/power required curves, such as shown in Figure 4-6, for the appropriate operating conditions: i.e., for maximum gross weight, start at the corresponding design speed, V_D, and calculate the required fuel flow, W_f, in pounds/hour. This is accomplished by using the engine data curves, such as shown in Figures 4-3 and 4-4. For arbitrary reductions in either ship speed or engine power, calculate the corresponding fuel flows, W_f, working down the power required curve to some speed near the hump condition. For a given gross weight, a plot can now be constructed of W_f vs. V_k, and this can be repeated for a number of ship gross weight conditions, working down to the zero fuel weight condition and resulting in a plot similar to that of Figure 4-7. The speeds indicated on this Figure as V_1, V_2, and V_3 correspond to the ship speed for minimum fuel flow for the respective gross weight conditions selected.

FUEL FLOW VERSUS SPEED

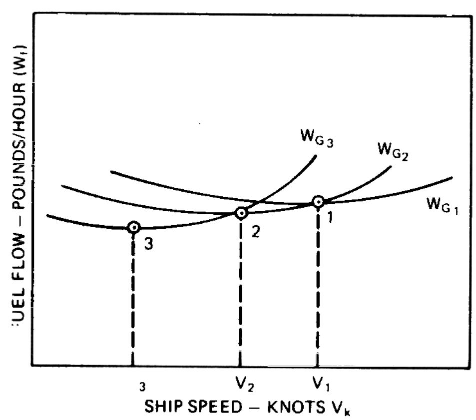


FIGURE 4-7

These points can now be used to plot specific endurance, $1/W_f$ (hours/pound), vs. gross weight, W_G, as in Figure 4-8. The area under this curve between any two weight conditions is the endurance, in hours, of the SES. This area can be found either by graphical integration between the desired weight limits or, if the equation of the indicated curve can be written, by numerical analysis. The maximum endurance of the SES is of course obtained by integrating between the limits of the maximum ship gross weight, $W_{G_{MAX}}$, and the empty weight (zero fuel) condition, W_{EMPTY}. This method of calculating performance is applicable to either the on-cushion or the displacement mode of SES operation.

ENDURANCE INTEGRATION CURVE

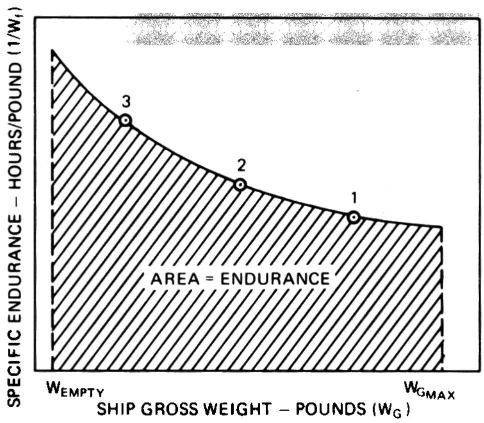


FIGURE 4-8

Range Performance. SES range performance is calculated similarly to endurance performance using the basic power available/power required curves for the appropriate operating conditions. Plots can be constructed of specific range, V_k/W_f (nautical miles/pound) vs. ship weight similar to that in Figure 4-9. The area under this curve between any two weight limits is the range of the SES in nautical miles, and the maximum range is obtained by integrating between the limits of maximum ship gross weight and empty weight.

POWER PLANTS

SES systems to which power must be supplied can generally be defined as propulsion, lift, control, and auxiliary systems. Current practical raw energy sources for primary power are fossil and nuclear fuels.

The primary power plant comprises the equipment employed to release the energy stored in the fuel and convert it into a useful form, such as torque or thrust. The desirable characteristics of this plant are that it be efficient, light in weight, and reliable and that it provide adequate performance at all design conditions. The relative importance of those criteria varies according to the types of fuel being used.

Fossil-fuel weight and volume are very significant considerations in ship design. A fossil-fuel power plant must have a low specific fuel consumption (SFC – pounds/HP-hour) to minimize the total fuel weight carried on board. This implies high overall plant efficiency. In addition, the

RANGE INTEGRATION CURVE

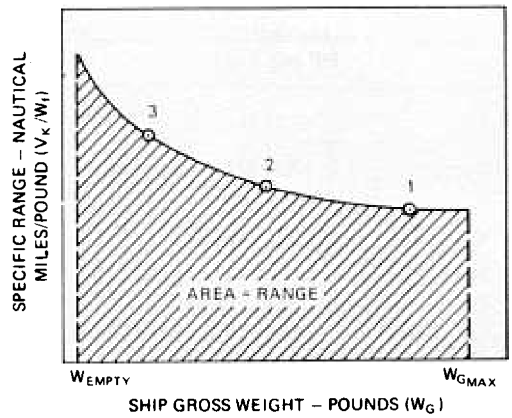


FIGURE 4-9

weight of the plant itself must be minimal, since high-efficiencies, which save on fuel weight, can often be obtained only at the cost of increased plant weight. The measure of this characteristic is the power plant specific weight in pounds/HP.

For nuclear fuel the energy per pound contained in the fuel is very high as compared with fossil fuel; thus, fuel weight is negligible on a nuclear craft, and plant weight becomes the significant factor (if fuel cost is neglected). Efficiency is important only to the extent that it reduces overall plant size and weight. For long range SES's, the cross-over point at which nuclear power appears to be more effective than fossil fuel, is a ship gross weight of about 5000 to 8000 tons.

With either plant, sufficient power must be available to permit the SES to traverse the hump and achieve design speed. Optimum efficiency is desirable at cruise speed. With the fossil fuel plant this high efficiency needs to be maintained from full-load to light-ship conditions (as fuel is burned off). Power must also be available under all off-design conditions.

FOSSIL-FUEL POWER PLANTS

A comparison of available fossil-fuel power plants is given in Table 4-2. Because the aircraft-derivative gas turbine has a competitively low SFC and a minimum weight-horsepower ratio (W/HP) it is the logical choice for the SES. It also has such additional advantages as low volume; ease of automation, installation, and removal for repair; quick starting; and long periods between major overhauls. The engine replacement philosophy as an alternative to on-board repair, reduces craft downtime.

Physical Characteristics. The gas turbine cycle of interest to the SES designer is the simple type rather than one of the various regenerative cycles, which are too heavy to be of interest. The basic components of this cycle for a split-shaft turbine are shown in Figure 4-10.

As indicated in the figure, atmospheric air is drawn into the compressor (1), where it is compressed and driven into the combustion chamber (2). Fuel is there added to the hot pressurized air, which ignites, producing a hotter, higher-pressure gas. This gas is allowed to expand back to atmospheric pressure through the high-pressure turbine (3) (to drive the compressor) and free-power turbine (4) (to drive the load).

POWER PLANT CHARACTERISTICS

Engine Types	Maximum HP per Unit	Minimum SFC pounds/HP-hour	Minimum Weight-HP Ratio* pounds/HP
Diesel low speed < 150 rpm	48,000	.32	130
Diesel medium speed 150 to 750 rpm	18,000	.34	40
Diesel high speed > 750 rpm	3,600	.38	4
Steam turbine	35,000	.4	20
Heavy marine gas turbine	60,000	.5	9
Heavy marine regenerative cycle (gas turbine)	60,000	.4	10
Combined cycle (gas and steam turbine)	45,000	36	6.5
Aircraft-derivative gas turbine	40,000	38	.4

*Reduction gear weights not included.

See References 13-15.

TABLE 4-2

TYPICAL CYCLE – SPLIT-SHAFT TURBINE

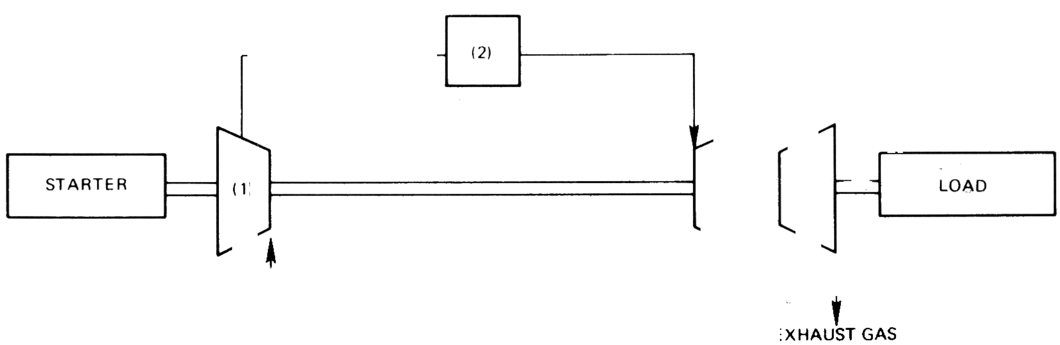


FIGURE 4-10

When the high-pressure turbine (3) is mechanically connected to the power turbine (4), it is known as a solid-shaft design. Such a design is not suited to applications wherein the output speed is not constant, since a 20 percent speed variation will cause the compressor to stall.

The split-shaft design (Figure 4-11) is mechanically more complex, but it permits the compressor to run at a steady speed while the power turbine is free to vary with the load. In addition, the starting effort is less because the compressor-gas generator section can be brought up to speed

LM 2500 ENGINE

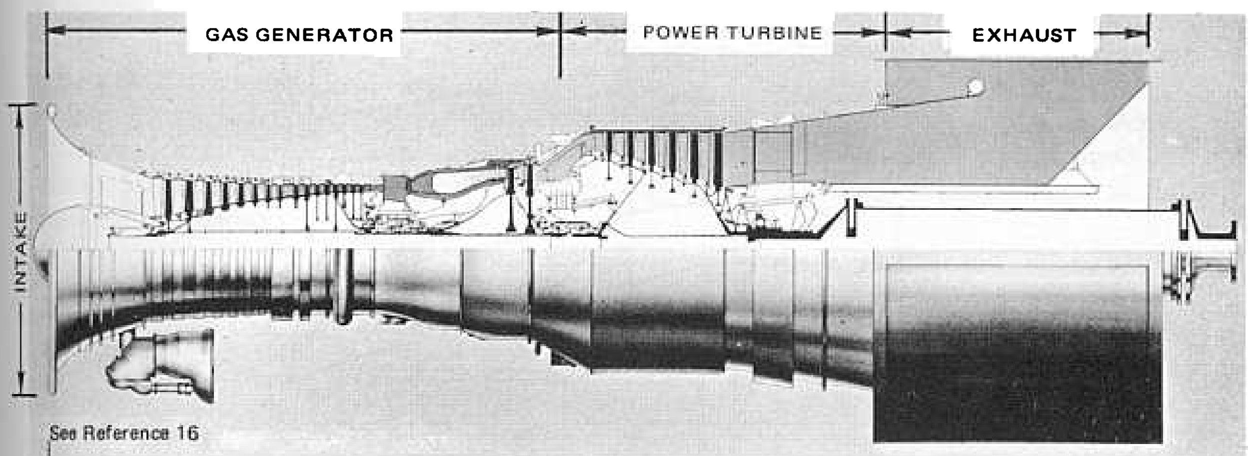


FIGURE 4-11

without rotating the power turbine and its connecting power train. Other variations of the split-shaft design involve variable inlet vanes and twin-spool arrangements, which serve to improve engine stability and control over a wide performance range.

Materials used in aircraft gas turbines are selected for their light weight, strength, and ability to withstand high temperature. The lightweight gas turbine used in the marine environment differs in various respects from the aircraft engine from which it was derived due to the need to accommodate it to the more dense, corrosive marine atmosphere. The vane and blade materials in the marinized engine are high-alloy stainless steel, titanium, and coated aluminum alloys. No magnesium alloys are used. The marinized engine has strengthened engine casings to meet the higher shock requirements and increased bearing capacity, since the loading at sea level is greater due to the increased air density. The combustor equipment is usually altered to permit burning of heavier distillate fuels at prolonged high air densities.

The accessory drive section of the gas turbine takes care of various functions required for operation and control. This unit, which is driven off the compressor shaft, provides drives for the fuel pump, lube oil pump, electric generator, and starter. The fuel, lube oil, electrical, and starting systems will be described later on.

Performance Characteristics. The performance of the gas turbine is affected by its internal design, its installation in the ship, the ambient operating conditions, and the loads imposed on it by the drive train.

The two most important internal design parameters of the gas turbine are pressure ratio, R, and turbine inlet temperature, *TIT*. The effects of varying each of these parameters are shown in Figures 4-12 and 4-13. As is evident from Figure 4-13, simultaneous increases in pressure ratio and turbine inlet temperature result in greater engine efficiency (lower SFC) and, from Figure 4-12, the increase in turbine inlet temperature results in more compact engines (higher specific horsepower, $(HP)_B/m_a$ where m_a is mass flow of air in pounds/second). The curves in these figures are based on equations derived from the basic Brayton Cycle, which describes the gas-turbine cycle. The assumptions used in developing Figures 4-12 and 4-13 are: a simple gas turbine cycle; ambient temperature of 60°F, 0 percent humidity; compressor efficiency of 0.89; turbine efficiency of 0.90; combustor efficiency of 0.98; heating value of fuel of 18,500 BTU per pound.

SPECIFIC HORSEPOWER VERSUS PRESSURE RATIO

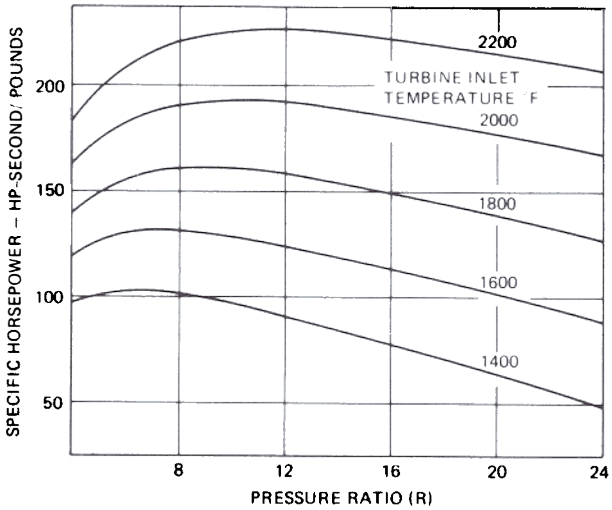


FIGURE 4-12

SPECIFIC FUEL CONSUMPTION VERSUS PRESSURE RATIO

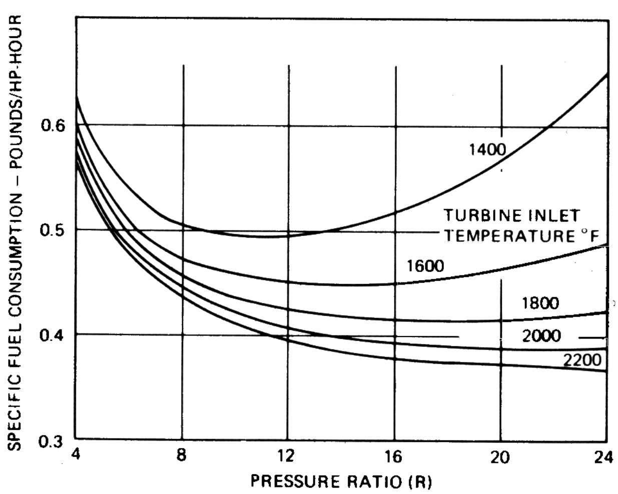


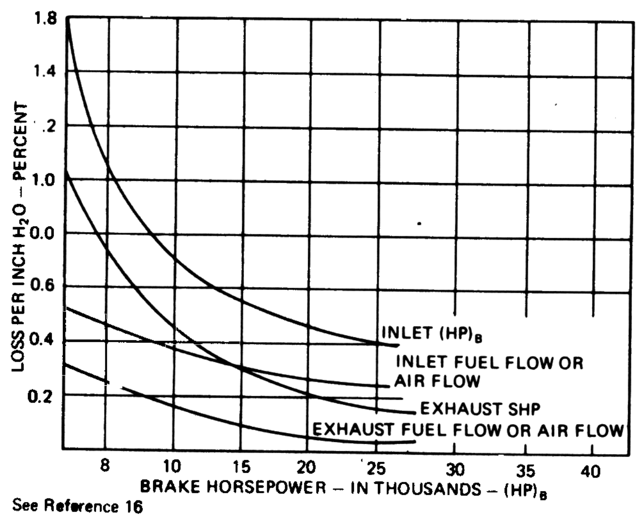
FIGURE 4-13

Design limits are imposed by material strength and resistance to corrosive action at elevated temperatures. Present *TIT* temperatures of 2500°F are being attained by transpiration and by internally air cooling the metal blades down to temperatures around 1500°F. Metal temperatures above 1500°F are avoided because they bring about an increase in the rate of sulfidation occurring on the turbine blades and other hot sections. Sulfidation deposits are due to a combination of salt in the ingested air and impurities in the fuel, such as sulfur and vanadium.

Blade air cooling requires an air flow from the compressor that reduces the overall efficiency of the engine. Further increases in *TIT*, with resultant increases in engine efficiency and power, appear obtainable only with materials that can withstand those temperatures without recourse to air cooling. At present, development work is being conducted with ceramic materials tolerant of 2500°F This suggests that the SES designer can expect gas turbines to be more efficient in the future than at present.

With careful attention given to inlet and exhaust ducting arrangement, pressure losses can be minimized and engine performance maximized. The effect of friction losses in these ducts is shown in Figure 4-14, which may be used to correct any given engine performance to suit the installed conditions.

INLET AND EXHAUST PRESSURE LOSS CORRECTION



See Reference 16

FIGURE 4-14

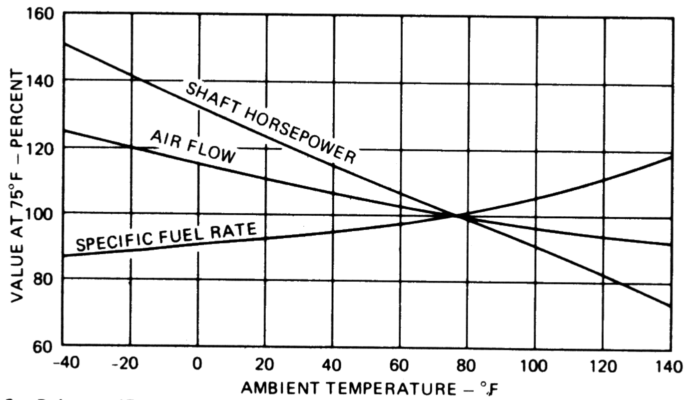
The effects of ambient temperature on turbine performance are shown in Figure 4-15. These curves are obtained from basic cycle relationships under an assumed constant engine rpm and efficiency.

Limitations in the structural strength of the engine dictate a power limit in actual operation. This limit is obtained by power-regulating the turbine through fuel control. Because of this requirement, actual engine curves are held to a maximum power level occurring at an ambient temperature of about 20°F.

Variations in humidity alter the specific heat and density of the air and, in turn, affect the turbine power, $(HP)_B$, and fuel flow, W_f. A correction curve for humidity is given in Figure 4-16. For a given relative humidity a standard psychometric chart can be used to obtain humidity in grains. Fuel flow, W_f, is related to specific fuel consumption, SFC by

$$SCF = \frac{W_f}{(HP)_B}.$$

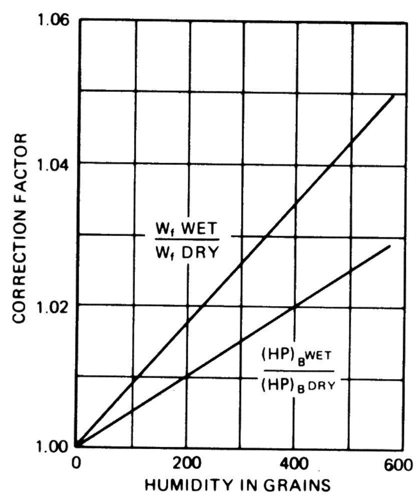
TYPICAL EFFECT OF AMBIENT TEMPERATURE



See Reference 17

FIGURE 4-15

HUMIDITY CORRECTION

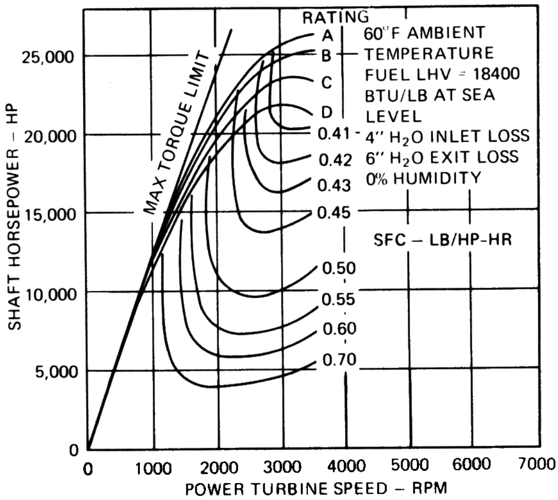


See Reference 16

FIGURE 4-16

The load on the free-turbine output shaft affects the performance of the turbine, as shown in Figure 4-17. This is a performance curve supplied by the manufacturer for a particular engine. Most turbine performance curves have approximately the same form. The two variables of torque and speed of the free turbine determine the operating point of the engine. In the figure the operating region of the power turbine is defined by the selected power rating (A, B, C, or D), the maximum free-turbine speed (on the right side of the figure), and a minimum free-turbine speed (on the left side of the figure).

SHAFT HORSEPOWER VERSUS POWER TURBINE SPEED



See Reference 16

FIGURE 4-17

The maximum power rating is the rating at which minimum acceptable turbine life occurs. Running the engine at lower ratings increases its life expectancy. Estimated life is about 400 hours at the "A" rating while at "D" it is about 8,000 hours. Continuous rating is "C" at 4,000 hours.

Maximum free-turbine speed is designed into the turbine by limitations in centrifugal stress capacity. Minimum free-turbine speed can be as low as 500 rpm. With the compressor section running at idle speed, the free-turbine can be stopped for limited periods of time.

Turbine performance is controlled by regulating the amount of fuel flow to the combustors. Either power or speed control is possible. When propellers are employed, power control is usually preferred on large marine installation, since it prevents cycling of the gas generator caused by propeller load fluctuations occurring in seaways. These fluctuations will probably be absent when the SES is operated at high speed due to the trough created at the stern of the ship. The hump characteristics of the SES drag curve necessitates speed control for stable operation, since power requirements are decreasing as ship speed is increasing in the area just beyond hump.

Engine Availability. A tabulation of marinized gas-turbine engines which currently are or will be available within the next ten years is presented in Table 4-3. Not included therein are engines classified for security reasons.

In the selection of an engine for a particular application, an investigation should be made of the advantages of using multiple smaller engines in parallel (or a large engine with one small engine for boost power) in place of a single large engine. Arrangements of multiple engines offer a potential for lower overall fuel consumption and some degree of system redundancy. With these arrangements it is possible to shut down one or two engines when running at reduced power levels and operate the remaining engines close to optimum efficiency.

Installation Considerations. The numerous details connected with the installation of the gas turbine in the ship are covered quite fully in Reference 18. The purpose of this section is to point out those considerations which significantly affect the arrangement, support interfaces, and weight of the craft. Weight-estimating equations are given at the end of this subsection.

CHARACTERISTICS OF MARINIZED GAS TURBINES

Manufacturer*	Name	$(HP)_B$	SFC (pounds/ HP-hour)	W (pounds)	$W/(HP)_B$ (pounds/HP)	RPM	R	m_a (pounds/ second)	$(HP)_B/m_a$ (HP/pounds/ second)	Max-imum TIT (°F)	Fuel Type**	L Length (feet)	D Envelope Diameter (feet)	Remarks
UACL	ST6J-70	489	0.67	350	0.7	2,200	6	6	80	–	1,2,3,4	5'-2"	1'-7"	
UACL	ST6K-70	489	0.67	317	0.62	6,230	6	6	80	–	1,2,3,4	5'-0"	1'-7"	
UACL	ST6J-77	529	0.663	379	0.69	2,200	6	5.9	90	–	1,2,3,4	5'-2"	1'-7"	
UACL	ST6K-77	529	0.663	350	0.64	6,230	6	5.9	90	–	1,2,3,4	5'-0"	1'-7"	
UACL	ST6L-77	633	0.618	306	0.47	33,000	6.3	6.1	104	–	1,2,3,4	4'-10"	1'-7"	
AL	TF-12	1,090	0.64	920	0.84	18,000	6.1	11	99	–	1,2,3,4	4'-4"	3'-1"	
AL	TF-14	1,390	0.60	920	0.66	19,600	6.5	11	146	–	1,2,3,4	4'-4"	3'-1"	
UACL	ST6T-75	1,289	0.651	730	0.56	6,600	6.3	12.2	105	–	1,2,3,4	5'-6"	3'-8"	Twin engines
AL	TF-25	2,130	0.66	1,020	0.48	15,100	6.1	21	102	–	1,2,3,4,5	4'-1"	3'-1"	
AL	TF-35	2,675	0.57	1,090	0.41	15,350	6.5	23	116	–	1,2,3,4,5	4'-4"	3'-1"	
		3,000												
AL	TF-40	3,440	0.53	1,160	0.34	15,350	7.2	25.5	135	–	1,2,3,4,5	4'-4"	3'-1"	
P & W	FT12	3,620	0.71	1,150	0.36	9,000	7	50	72	1,600	1,2,3,4,5	8'-3"	3'-0"	
GE	LM500	4,500	0.46	1,180	0.26	6,500	14	33	137	2,000	1,2,3,4,5	9'-0"	5'-6"	Proposed
RR	PROTEUS 52M	3,280	0.674	3,531	1.08	5,240	6.9	40.2	123	–	1,2,3,4	9'-4"	3'-10"	
RR	PROTEUS 10M	3,280	0.674	3,669	1.12	1000	6.9	40.2	123	–	1,2,3,4	8'-4"	3'-10"	
	15M					1500								
RR	TYNE RMIA	3,920	0.513	19,000	4.8	3,200	12.5	44.3	88.5	–	1,2,3,4	18'-3"	5'-0"	Enclosed module
RR	TYNE RMIC	5,000	0.492	19,000	3.8	3,425	12.5	44.6	112	–	1,2,3,4	18'-3"	5'-0"	Enclosed module
GA	GTPF 990	5,900	0.423	3,201	0.55	3,600	11.5	39	151	2,000	1,2,3,4	8'-2"	4'-0"	In develop-ment
GE	LM1500	12,400	0.57	7,500	0.61	5,500	12	151	82	1,650	1,2,3,4,5	21'-0"	7'-0"	
GE	LM2500	22,500	0.4	10,600	0.47	3,600	15.45	135	167	2,100	1,2,3,4,5	22'-3"	7'-0"	
RR	OLYMPUS	20,500	0.52	47,000	2.3	5,183	10.3	216.4	94.6	1,817	1,2,3,4	30'-0"	12'x8'	Enclosed module
GE	LM3500	34,950	0.38	13,000	0.37	3,600	21.76	213.5	164	2,100	1,2,3,4,5	25'-3"	8'-4"	Proposed
P & W	FT4C-2	30,600	0.46	16,100	0.53	3,600	12	292	105	1,800	1,2,3,4,5	27'-0"	7'-11"	
P & W	FT9	40,000	0.37	16,500	0.41	3,600	18	220	182	2,000	1,2,3,4,5	22'-6"	7'-11"	Proposed

*The manufacturer key is as follows:

UACL	United Aircraft of Canada Ltd.
AL	Avco Lycoming
RR	Rolls Royce
GA	Garrett Air Research
GE	General Electric
P & W	Pratt & Whitney

**Fuel types are as follows:

1. Kerosene
2. JP-4
3. JP-5
4. Marine Diesel (MIL-F-16884)
5. Navy Std. Distillate (MIL-F-24397)

Characteristics are for the following conditions:

(HP) – "C" continuous rating:	Sea level, static 60°F, 0 humidity, 4" H_2O inlet & 6" H_2O exhaust losses.
Weight engine plus accessories:	Bare engine, plus accessories normally supplied with engine.

TABLE 4-3

a. **Inlet Ducting.** The inlet ducting brings combustion and cooling air to the engine. Air intakes are large, and adequate space allowance must be made for them. An essential consideration in arrangement is to minimize pressure losses by maintaining low duct velocities and smooth runs.

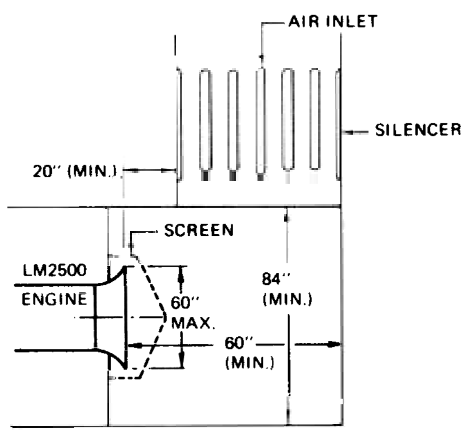
Engine cooling air is usually about 10 percent of the combustion air requirements. Acceptable inlet duct velocities are 50 to 90 feet/second. From these data the duct cross-sectional area is determined. The ducting pressure drop is determined from a standard duct friction chart similar to Figure 8-15 (Chapter VIII). Typical values are between 0.5 and 1.0 inch of water.

Installed in the inlet duct are a bird screen and inlet filter, or demister, which prevents foreign objects and water spray from entering the engine. Typical pressure drop for these two components is 0.5 inch to 2.5 inches of water.

Basically, there are two methods of water removal, inertial demisters and knit mesh demisters. Inertial demisters consist of turning vanes, the water is thrown outward in turning and clings to the surface of the vanes due to the centrifugal effect. The principal advantage of the inertial demister is that it is relatively small and does not require a great deal of attention; however, water removal efficiency at low air flows drops off quite rapidly. The knit mesh demister filters and coalesces the water particles. The efficiency of the knit mesh demister remains high over the entire air flow range and pressure drops are lower than with inertial demisters (1.5 to 3 inches of water versus 2.5 to 3.5 inches of water). The disadvantages of the knit mesh demisters are that they are heavier and require more space and maintenance than inertial demisters. Approximately one square foot of knit mesh demister is required for every 200HP of engine being served.

The internal duct material should be resistant to corrosion caused by the saltwater-laden air. Aluminum and stainless steel are recommended. For weight-estimating purposes, it is appropriate to use figures for insulated duct of 6 pounds/square foot of duct surface area for large turbines and 4 pounds/square foot for turbines below 5000 horsepower. In selecting duct size, consideration should be given to the possibility of removing the gas generator section through the inlet duct. This method is frequently adopted, as it provides quick and easy access from the weather deck to the engine room. The inlet duct should be oriented to prevent the ingestion of salt spray and exhaust gases. Inlets preferably face inboard or aft and are located below the exhaust stacks. Louvres can be used to advantage in reducing ingestion. A flexible joint connects the duct to the turbine inlet and to the chamber around the engine, which directs air over the engine for cooling. This joint provides for engine thermal expansion. Adequate space should be provided for the intake plenum in front of the engine. A typical installation is shown in Figure 4-18. It is desirable to have a separate plenum for each engine because intake flow-pattern distortions result when two engines compete for air.

TYPICAL INTAKE PLENUM



See Reference 16

FIGURE 4-18

Under circumstances where freezing temperatures are anticipated, provisions should be made for deicing and possibly for by-passing the inlet filter-separator by a blow-in door. Hot compressor-bleed air is generally used to provide sufficient head to prevent the formation of ice. An electric heating system applied to the engine inlet fairing can also be used. The electric power requirement is about 1KW per 3000 HP. These features are optional on the engine itself.

b. **Exhaust Ducting.** The exhaust ducting must be designed to minimize duct pressure loss, resist exhaust gas temperatures on the order of 1000°F, and provide attenuation of exhaust noise.

The duct is sized to maintain exhaust gas velocities of 100 to 200 feet/second. The exhaust gas includes both engine exhaust and engine external cooling air. The cooling air is drawn into the exhaust duct by an eductor or a fan. The fan has the advantage of permitting engine cooling after shutdown.

As with the inlet ducting the exhaust ducting is rigidly supported by the ship's structure, and a flexible connection to the engine is required to accommodate engine thermal expansion. The duct itself is usually thermally and acoustically insulated. Silencers can be installed for additional sound attenuation. For weight-estimating purposes, it is appropriate to use figures for insulated duct of 7 pounds/square foot of duct surface area for large turbines and 5 pounds/square foot for turbines below 5000 HP.

c. **Engine Mounting System.** Small turbines are often sufficiently lightweight to be cantilevered off the gear boxes or flanges of the equipment they drive. The mounting system for a larger engine is generally supplied by the engine manufacturer. The foundation the system interfaces must be able to accept the weight and torque reaction of the engine and must conform to the attitude limits set by the manufacturer. Some typical limits are:

Permanent trim	13° up either end
Permanent list	15° either side of vertical
Momentary trim	10° beyond permanent trim for 10 sec
Momentary list	45° beyond permanent list for 10 sec.

d. **Engine Enclosure.** The engine enclosure provides for attenuation of engine noise, thermal containment of engine surface temperature (which can reach 1000°F), and shrouding of cooling air over the engine. It must be designed to permit access for maintenance and removal. Insulation can be similar in type to that of the exhaust ducting.

e. **Fire Protection.** Fire detection and extinguishing equipment should be installed in the engine enclosure. The extinguishing system can employ carbon dioxide, dry chemical, foam, freshwater, or seawater. Each has its advantage. The carbon dioxide system leaves no residue but, as do the dry chemical and foam systems, it does require recharging. Determination of the most suitable system will be influenced by the ship service systems, which can conveniently be interfaced with the engine.

f. **System Interfaces to Engine.** On the larger horsepower engines certain portions of support systems can be mounted outside the engine enclosure and be connected to the remaining portions of the same system that are engine mounted.

For the lube oil system these off-engine components may include lube oil tanks, lube oil/seawater coolers, duplex filters, mist precipitators, control valves, seawater pumps and associated piping.

The electrical system provides sensing, protection, ignition, and control services to the engine. Control power is normally obtained from the ship auxiliary power system. About 1 kilowatt is required for a typical 39,000 horsepower engine.

An engine starter is required to motor the gas generator to a minimum speed before the fuel is ignited. Once the fuel is ignited, it continues to burn of itself, and the engine starter is turned off. Starting time required is about 1 to 2 minutes.

Starting systems can be hydraulic, pneumatic, or electrical. Larger engines, such as the LM 2500 and the FT 4, are restricted to use of hydraulic and pneumatic starters. For these large engines pneumatic starter-system requirements are about 130 pounds/minute of air at 40 psi. This air can be supplied by one auxiliary power unit (APU), which can also be used to drive the ship's emergency generator. Once one engine is operating, a second engine can be started by using bleed air from the first. The weight of the air starter for this system is about 30 pounds. If the APU is part of the emergency generator, only piping and valve weight need be considered in addition to starter weight. If bottled compressed air is required for backup, however, the weight requirement may be as high as 2000 pounds.

The weight of a hydraulic starter system for these large engines is about 2700 pounds. This includes a 100-horsepower, electric motor. It does not include the weight of the seawater in the hydraulic oil cooler. Starters on turbines below 10,000 HP are usually electrical and are included in the engine weight.

Another function of the starter motor is to turn the gas generator during engine washing. For this purpose a freshwater supply to the engine is required. Wash-water requirements for the LM 2500 are 200°F freshwater at 40 psig and 20 gpm. About 120 gallons of water are required for a complete wash, which should be undertaken once every 24 hours or whenever engine performance deteriorates due to salt buildup on the compressor blades.

A vent and drain system is required to provide drainage to the inlet and exhaust ducts, lube oil sumps, and fuel system. The lube oil and the fuel drain to their respective waste tanks.

g. **Fuel Heating Requirements.** Fuel for the gas turbine should have a kinematic viscosity of less than 10 centistokes for cold starting and of 4 centistokes for running to minimize smoking. These figures vary with the engine manufacturer. A JP-5 fuel system, wherein fuel cleanliness is maintained to aircraft standards, is the simplest and lightest, since JP-5 has a kinematic viscosity of 4 centistokes at 28°F. Most of the larger horsepower engines are capable of burning heavier distillates but only after the fuel is heated and purified. Navy distillate has a kinematic viscosity of 65 at 28°F and of 10 at 100°F.

Parametric **Weight Equations.** For preliminary weight estimation of a large gas turbine installation, the following equations may be used:

$$\frac{W_{pp}}{(HP)_B} = 0.92 + \frac{84}{\sqrt{(HP)_B}} \quad - \textit{(pounds/HP) for Navy distillate fuel (per engine)}$$

$$\frac{W_{pp}}{(HP)_B} = 0.69 + \frac{84}{\sqrt{(HP)_B}} \quad - \textit{(pounds/HP) for JP-5 fuel (per engine)}$$

where

W_{pp} = weight of total power plant installation, per engine – pounds.

For smaller engines (below 5000 HP) the last term in Equations (4.6) and (4.7) would be

$$\frac{50}{\sqrt{(HP)_B}} .$$

This relationship is based on the sum of Equations (4.8) through (4.17). Once the approximate power level of the engine is known, a more refined estimate can be obtained by substituting in these equations actual data appropriate for the more precise size range.

$$\frac{W_t}{(HP)_B} = 0.5\ (pound/HP) \tag{4.8}$$

$$\frac{W_d}{(HP)_B} = \frac{12.3\,(\ell_i) + 13.8\,(\ell_e)}{k\,\sqrt{(HP)_B}} \tag{4.9}$$

$(let\ \ell_i = 30\ feet,\ \ell_e = 40\ feet,\ k = 120\ HP/pound/second)$

$$\frac{W_d}{(HP)_B} = \frac{84}{\sqrt{(HP)_B}} \quad pounds/HP) \tag{4.10}$$

$$W_{fe} = LD^2/5.3 \tag{4.11}$$

$$\frac{W_{fe}}{(HP)_B} = 0\ (for\ Equations\ (4.6)\ and\ (4.7)) \tag{4.12}$$

$$\frac{W_e}{(HP)_B} = 0.07\ (pounds/HP) \tag{4.13}$$

$$\frac{W_s}{(HP)_B} = 0.05\ (pounds/HP) \tag{4.14}$$

$$W_e = 14LD \tag{4.15}$$

$$\frac{W_e}{(HP)_B} = 0.05\ (pounds/HP)\ (for\ Equation\ (4.6)\ and\ (4.7)) \tag{4.16}$$

$$\frac{W_{FSYS}}{(HP)_B} = 0.25\ (pounds/HP)\ for\ Navy\ distillate\ fuel$$

$$= 0.02\ (pounds/HP)\ for\ JP\text{-}5\ fuel \tag{4.17}$$

where the subscripts are

t = engine

d = ducting

ℓ = lube oil

fe = fire extinguishing

$FSYS$ = fuel system

s = starting

e = enclosure

and

ℓ_i = inlet duct length

ℓ_e = exhaust duct length

$k = (HP)_B/m_a$

L = engine length – feet

D = engine envelope diameter – feet

NUCLEAR POWER PLANTS

In the nuclear power plant, thermal energy is produced in the reactor through the atomic fission process. The thermal energy is absorbed by a fluid or gas (coolant), which transfers it to the turbine section of the plant for conversion into mechanical energy.

Although many types of coolants have been proposed and used in nuclear reactors for various applications, helium appears most attractive for the SES. Its ability to be used at high temperatures (1700°F) and pressures (1000 psi) results in more compact, high thermal-efficiency designs. Its inertness eliminates the problem of internal metal corrosion. Its non-radioactive properties permit it to pass through unshielded piping outside the reactor and to be used in single-cycle or direct-loop systems. In contrast to liquid metals (sodium and potassium), which have also been considered for lightweight aircraft reactors, helium requires no startup heat for liquification, and its lower heat-sink capacity reduces the external corrosion problems in heat exchangers.

Helium also has a higher heat transfer coefficient than air and carbon dioxide, which means that comparative heat exchanger sizes can be reduced by one-half to two-thirds (see Reference 19).

Proposed Power Cycles. Of the power cycles proposed for SES's those which are most attractive because of simplicity, light weight, or high efficiency are the "direct closed gas cycle" and the "direct open gas cycle."

For the direct closed gas cycle (Figure 4-19), the upper limit of cycle efficiency, η_t, is dictated by the thermal capabilities of currently available materials. A 54 percent efficiency is obtainable with turbine inlet temperatures of 2000°F. Efficiencies of 36 to 40 percent are currently being achieved (see Reference 20). This efficiency can be maintained for operation down to 30 percent of full load. Control is accomplished by keeping the reactor at constant temperature (by regulating the effect of the control rods) and by transferring helium to and from the cycle via the accumulators. The resultant variations in system pressure maintain the system temperature and efficiency at a constant value.

DIRECT CLOSED GAS-CYCLE NUCLEAR POWER PLANT

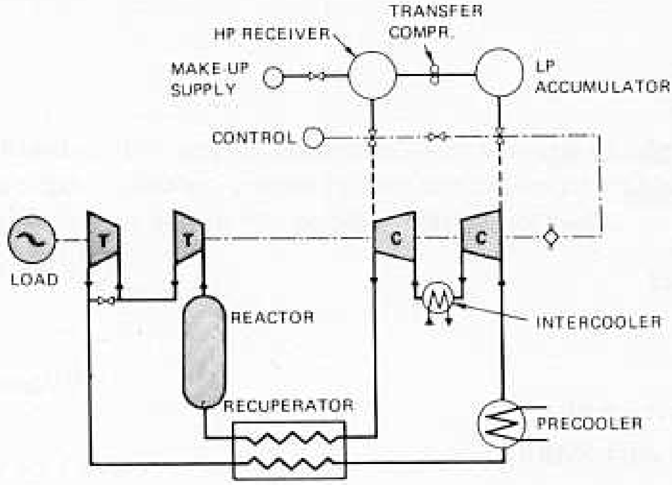


FIGURE 4-19

The increased efficiency the closed cycle plant offers for the SES takes the form of lower operating costs and a reduction in reactor and shield weight. Only a small weight penalty is represented by the recuperator (regenerator) and intercoolers.

The direct open gas cycle (Figure 4-20) is capable of efficiencies in the range of 20 tp 30 percent. Its prime advantage for the SES is its simplicity and light weight. The prime mover in this cycle differs from the fossil-fuel gas turbine in that the air, which is usually heated by combustion in other prime movers, is heated by hot helium via a heat exchanger.

DIRECT OPEN GAS-CYCLE NUCLEAR POWER PLANT

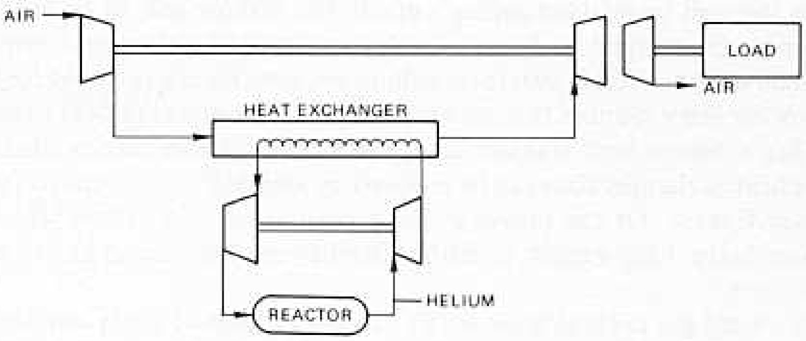


FIGURE 4-20

In both the closed and the open cycle systems, the turbines may be located remote from the shielded reactor. Large ducts on the order of 2 to 3 feet in diameter are required to carry the hot gas to and from the turbine. One means of reducing the weight of this ducting and improving the efficiency of the cycle is to use concentric ducting, so that the high-pressure, high-temperature gas flows through the inner duct and the lower-pressure and lower-temperature gas returns via the outer duct. This arrangement reduces the thermal gradient and stress in the inner duct wall.

For the closed cycle plant only the power turbine section needs to be located near the load. Unlike the open cycle turbine, the closed cycle turbine requires no air ducting, thus simplifying sidehull and transmission design in the vicinity of the propulsor.

In Europe operating experience has been gained with nuclear closed gas cycle turbines in land installation. In the United States lightweight reactors have been studied in connection with aircraft and manned rockets (ROVER). A number of discussions are also available on the adaptability of these concepts to the SES (see References 21, 22 and 23). Since none has actually been built, however, only approximate sizing figures obtained from these sources can be presented. A general layout of a nuclear SES employing an open gas cycle is shown in Figure 4-21. Two cross-connected reactors are installed for reliability. In a closed gas cycle the propulsion and lift power turbines would be located near the driven equipment, and the regenerators, intercoolers, compressors, and compressor drive turbines would be located proximate to the reactors.

Parametric Weight Data. A summary of nuclear power plant characteristics is given in Table 4-4. The engine weights for the closed cycle plant are derived from Reference 24 which gives dimensions for a 250-MW (340,000 HP) turbine. These dimensions are similar to those of the LM 2500 and FT 4 engines. The increase in power per unit volume is thus on the order of ten times. Increased power levels and the resultant higher torques necessitate use of an engine having

NUCLEAR POWER SYSTEM

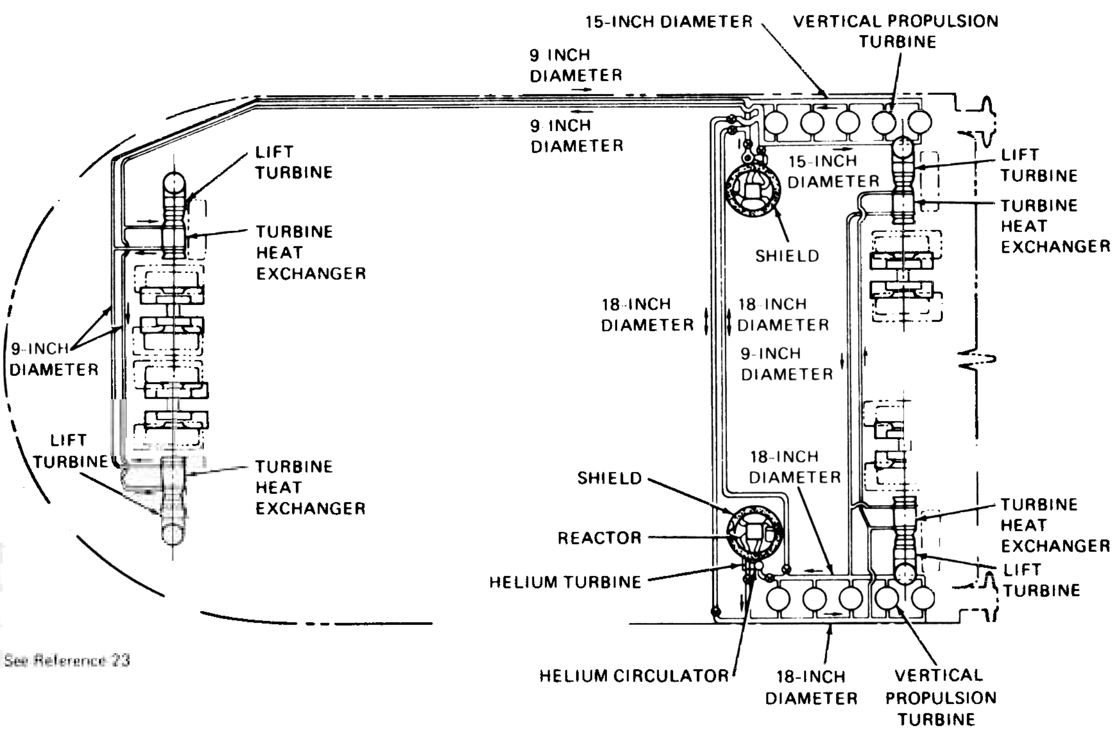


FIGURE 4-21

CHARACTERISTICS OF NUCLEAR SYSTEM

Quantity	Cycle: Open	Cycle: Closed	Units
η_t (cycle efficiency)	0.20 to 0.30	0.30 to 0.54	–
P_{th} (reactor power)	$7.47 \times 10^{-4} \frac{(HP)_{BT}}{\eta_t}$	$7.47 \times 10^{-4} \frac{(HP)_{BT}}{\eta_t}$	MW
W_R (reactor weight)	$\rho_r P_{th}$	$\rho_r P_{th}$	pounds
ρ_r (reactor weight density)	300 (R[illegible]	300 (Re[illegible]	pounds/foot^3
ρ_p (reactor power density	3.5 (Re[illegible]	3.5 (Re	MW/foot^3
W'_R (reactor weight based on 4 & 5)		$85.6\ P_{t}$[illegible]	pounds
W_{SH} (shield weight)	$49.500\ P_{th}^{(0.4}$[illegible]	$49{,}500\ P_{th}$[illegible]	pounds
W_{SH} (shield weight based on 5)	$49.500\ P_{th}^{.4}$	$49{,}500\ P_{th}^{.4}$	pounds
W'_{RS} (weight reactor & shield 6 + 8)	$2.800 \left[\frac{(HP)_{B}}{\eta}\right.$[illegible] $\left[\frac{(HP)_{BT}}{\eta_t}\right]$	$2{,}800 \left[\frac{(HP)_{BT}}{\eta_t}\right]$ $\left[\frac{(HP)_{BT}}{\eta_t}\right]$ 064	
W_{Rc} (recuperator weig[illegible]		$\frac{1.19}{\eta_t}$ $(HP)_{BT}$ 4[illegible]	
11. W_c (intercooler weight)		[illegible] $\left[\frac{1}{\eta_t} - 1\right]$ $(HP)_{BT}$	pounds
12. W_{dc} (coolant ducting weight)	[illegible].6 $(HP)_{BT}$	[illegible].6 $(HP)_{BT}$	pounds
13. W_{da} (air ducting weight)	[illegible]68 $(HP)^{1/2}$[illegible]		pounds
14. W_e (engine weight)	[illegible]	$(HP)_{BT}$ (Refs. 20 and 24)	pounds
15. W_{tp} (propulsor & transmission			
a) Supercavitating propeller	.4 $(HP)_{Bp}$	.4 $(HP)_{Bp}$	pounds
b) Waterjet	.9 $(HP)_{Bp}$	.9 $(HP)_{Bp}$	pounds
16. W_f (fan weight)	[illegible] $(HP)_{Bf}$	[illegible] $(HP)_{Bf}$	pounds
17. First estimation weight, W_{NS}, (9, 10, 11, 12, 13, 14, [illegible] 16)			
a) Supercav[illegible] propeller	$(HP)_{BT}$ + 3.[illegible] [illegible]$P)_{Bp}$ + 4.4 $(HP$[illegible]	$(HP)_{BT}$ [illegible] $(HP)_{Bp}$ [illegible].4 $(HP)_{Bf}$	
b) Waterjet	$(HP)_{BT}$ + 4.[illegible] [illegible]$P)_{Bp}$ + 4.4 $(HP$[illegible]	$(HP)_{BT}$ [illegible] $(HP)_{Bp}$ [illegible].4 $(HP)_{Bf}$	
Supercavitating propeller		5.4 $(HP)_{BT}$ + [illegible] $(HP)_{Bp}$ [illegible].4 $(HP)_{Bf}$	pounds
Waterjet		5.4 $(HP)_{BT}$ + [illegible] $(HP)_{Bp}$ [illegible].4 $(HP)_{Bf}$	pounds

$(HP)_{BT}$ = total ship brake horsepower
$(HP)_{Bp}$ = propulsion brake horsepower
$(HP)_{Bf}$ = fan brake horsepower

TABLE 4-4

sufficiently more material to ensure acceptable stress levels. Since the rotative speeds of the turbines are similar, the weight increase would be proportional to the power increase. The specific weight of the closed cycle turbine is thus the same as that of current fossil-fuel gas turbines.

The size of the turbine in the closed cycle application can be estimated on the basis of existing fossil-fuel turbine dimensions of one tenth the power under consideration. The size of the open cycle turbine is about equal to that of an existing fossil-fuel turbine of comparable power level, except that the diameter will be about 25 percent greater to accommodate the increased air flow. This increased air flow is necessary because heating the air via a heat exchanger results in a lower turbine inlet temperature than is obtainable by combustion.

Chapter V
TRANSMISSIONS

The primary function of the transmission system is to transmit power between two points. A secondary function is to change speed and torque, as well as the direction of motion of the rotating components used to transmit power. Power can be distributed from the prime mover to the lift, propulsion, and auxiliary systems by either an integrated or a non-integrated transmission system. In an integrated system the prime mover drives the propulsion, lift, and auxiliary systems through a common transmission; in a non-integrated system the propulsion, lift, and auxiliary systems have separate prime movers. Nuclear powered, electrical and hydraulic systems are well suited to integration, since their transmission components are simple as compared with those of a mechanical system. In nuclear powered, electrical and hydraulic systems, ducts, wires, or pipes with appropriate valves or switches are all that are necessary to channel power from one area to another, whereas mechanical systems require complex arrangements of clutches, gears, and shafting to accomplish the same end.

The advantage of the integrated system is that it permits installation of a single high-power-level prime mover (which generally has a lower fuel consumption rate than smaller units) and allows it to run at peak efficiency for longer periods of time. Peak efficiency is attained by running the prime mover at maximum power level at all times and by shifting power from the fan system (as its requirements fall off) to the propulsion or auxiliary systems. Since fewer engines are required, the associated support systems are also reduced in number, with resultant savings in weight, space, and maintenance. These advantages can quickly be offset, however, if the machinery arrangement required necessitates locating the driven portion remote from the engine. Moreover, with an integrated system, one prime mover failure conceivably could put the entire SES out of service.

The advantages of a non-integrated system are that it is simpler to design and to control. Each system has its own prime mover(s) so that fewer restraints are imposed on its location in the SES. In the event of failure of one of the systems, the other systems may be able to keep the SES operating. The system is operable under reduced load, with some engines shut off and the remaining operating at maximum efficiency.

On a large SES it may not be feasible to integrate a completely mechanical power system because of the long shafting lengths involved, but integration in localized areas is a possibility. A decision whether to integrate a system totally, partially, or not at all must be based on a trade-off analysis of the various possibilities in terms of weight, maintainability, and economics of installation and operation.

BASIC CHARACTERISTICS

In the selection of a transmission system and components, a low weight-horsepower ratio is of prime importance. Of secondary importance are high efficiency, flexibility, and simplicity. Other desirable characteristics include maintainability, low volume, and low noise levels.

Numerous mechanical, fluid, and electrical power transmission systems are available. Of these particular consideration will be given below to gear, electro-mechanical and superconducting electric systems.

Gear System. Figure 5-1 is a schematic diagram of a typical gear system comprised basically of gear boxes and shafting. The shafting is used to transmit power from one point to another, while the gear boxes are used to change speed, torque, and direction of rotation, as well as to combine parallel and intersecting shafts. Support components include bearings, couplings, clutches, and a lubrication/cooling system. Figure 5-1 indicates right angle gear boxes, however, a helical reduction gear train ("drop-box") would also be suitable.

TYPICAL GEAR SYSTEM

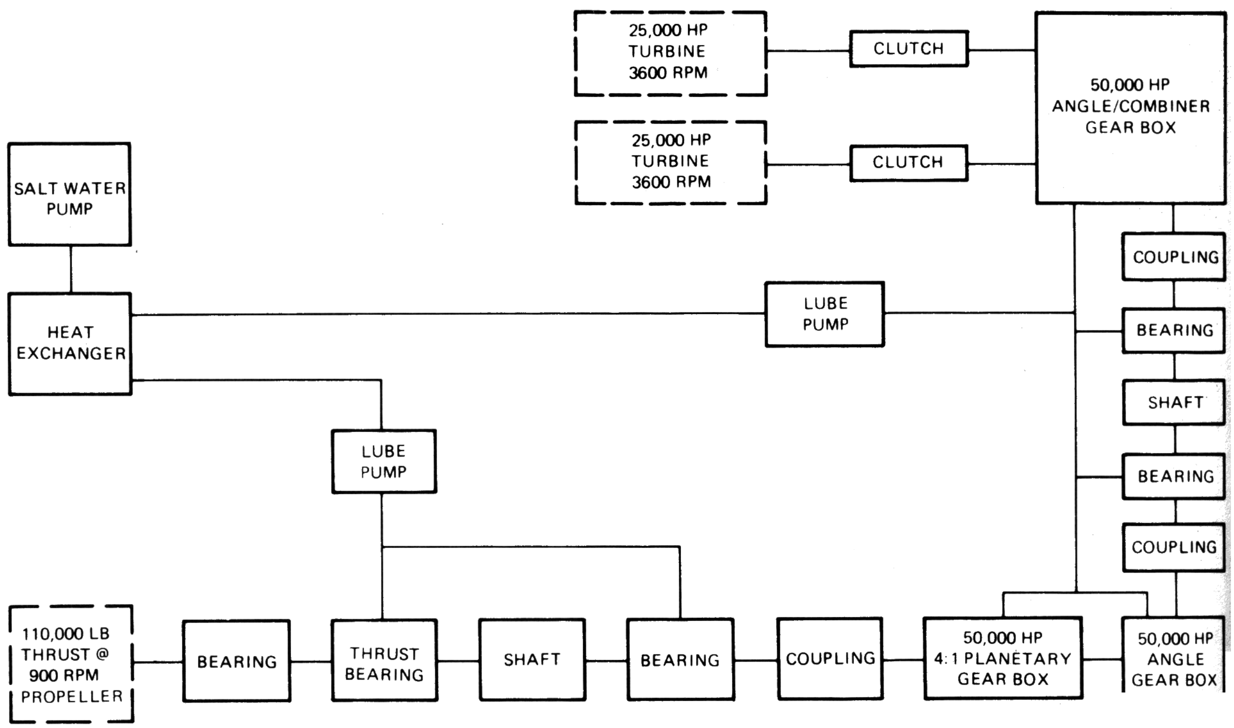


FIGURE 5-1

The major components of a gear system are the gear boxes. For parallel shaft application, gear and pinion or epicyclic arrangements are used (Figure 5-2). Gear and pinion arrangements can vary in the number of branches and number of reductions. Epicyclic arrangements can be either planetary, star, or solar. The planetary arrangement results in the smallest gear box.

Table 5-1 provides basic data on the various epicyclic arrangements. For intersecting shaft applications, angle gear boxes are used. These boxes can accommodate shafts intersecting at any angle but are normally designed for right angle intersections. Sets of angle gears can be ganged to increase power capability.

Use of high-strength materials and precision grinding techniques common to the aircraft and helicopter industry permits gear systems to be the lightest and most compact of the available transmission systems. With proper design, installation, and maintenance, these systems run smoothly and quietly and provide the longest obtainable service life possible within present-day technology.

GEAR BOX ARRANGEMENTS

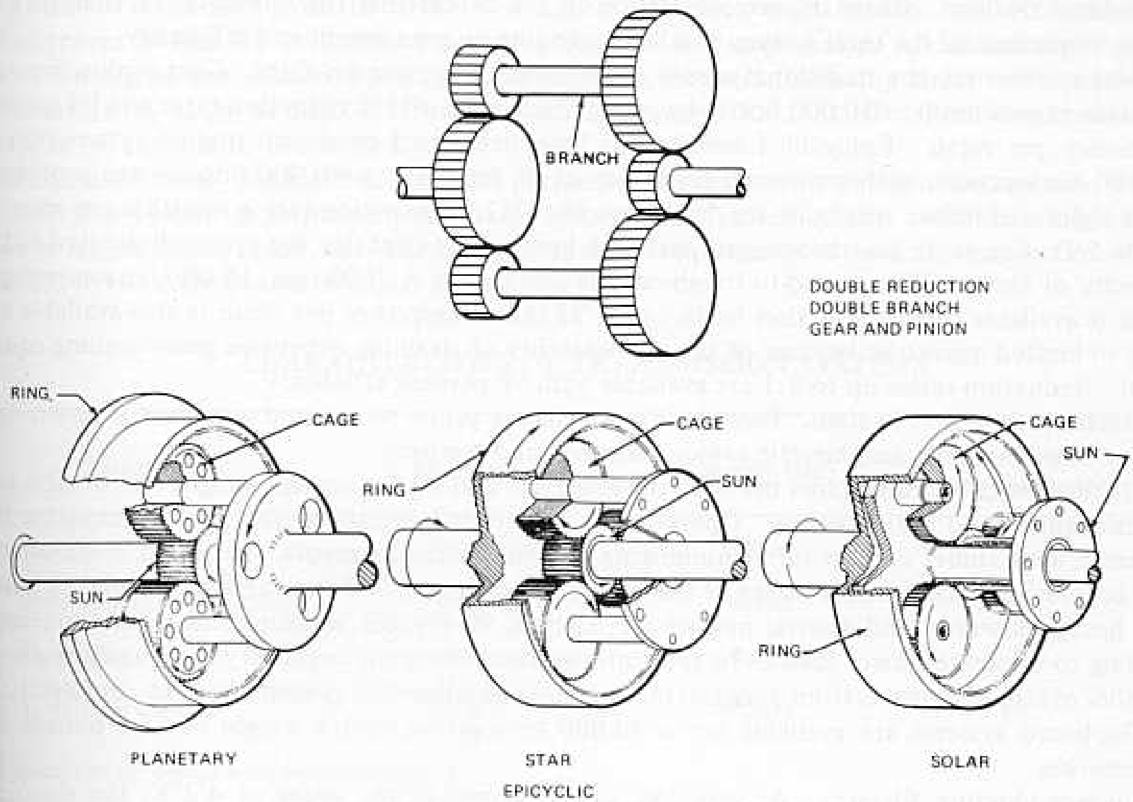


FIGURE 5-2

CHARACTERISTICS OF EPICYCLIC GEARS

Arrangement	Fixed Member	Input	Output	Overall Gear Ratio M_o	Range of Overall Gear Ratios
Planetary	Ring	Sun	Cage	$N_R/N_S + 1$	3:1 – 12:1
Star	Cage	Sun	Ring	N_R/N_S	2:1 – 11:1
Solar	Sun	Ring	Cage	$N_S/N_R + 1$	1.2:1 – 1.7:1

N_R = number of teeth in ring gear
N_S = number of teeth in sun gear

TABLE 5-1

Gears, however, require precise shaft alignment to permit the gear teeth to mesh properly, together with careful balancing to prevent vibration. Such devices as universal joints and special couplings must be employed to absorb flexure between components and prevent noise and vibration and the resultant reduction in life. In addition, because gears lack inherent shock-absorbing characteristics, protective devices must be used to prevent catastrophic breakdowns in the event of sudden overload. Gears require lubrication to the extent that the lubrication system may be 25 to 50 percent of the total system weight, depending on gear weight and efficiency.

Gear systems are the traditional means of transmitting power on ships. Gear pinion arrangements are available up to 30,000 horsepower per mesh, with a 15:1 reduction ratio and 99 percent efficiency per mesh. Epicyclic transmissions have been used in several marine systems up to 16,500 horsepower, with an overall efficiency of 98 percent. A 40,000 horsepower prototype with a 4:1 reduction was built for Navy use in 1967. Reduction ratios available are given in Table 5-1. For angle gear boxes, gear size and horsepower capacity are presently limited to the capacity of the machinery used to rough-cut the gear teeth. A 3600 rpm, 15,000 horsepower per mesh is available on a production basis; up to 25,000 horsepower per mesh is also available but only in limited quantities because of the inadvisability of straining expensive gear-roughing equipment. Reduction ratios up to 9:1 are available with 98 percent efficiency.

Electro-mechanical System. Electric drives utilize a prime mover and a generator powering a motor through connecting electric cables, switches, and controls.

The electric cable connectors between the generator and motor are lightweight and flexible and require almost no maintenance. Operation is relatively lubrication free, thus minimizing the generation of fumes and, in turn, minimizing ventilation requirements. Required speed output can be provided with proper sizing of the motor and generator. Motors and generators are bulky and heavy, however, and electric motors are, at best, 90 percent efficient and require extensive cooling to dissipate power losses. In applications where reversing capability is required or a large number of remote outputs from a central input is desired, electrical systems warrant consideration.

Shipboard systems are available up to 40,000 horsepower with a weight of 15.5 pounds per horsepower.

Superconducting Electric. At very low temperatures, in the order of 4.2°K, the electrical resistivity of certain materials disappears. These materials, called superconductors, have current carrying capacities of up to three orders of magnitude higher than conventionally used copper, thus opening the possibility of designing electrical machinery offering substantial weight and volume reductions over conventional machinery.

For a shipboard installation, a constant-speed prime mover drives a synchronous generator with a superconducting field winding. Cooling of the superconducting field is provided by circulation of liquid helium. A vacuum region surrounding the windings and thermal radiation shield serves as the principal means of insulation. A cycloconvertor, which changes the output frequency, is the primary means of speed reduction and control. A synchronous motor, similar to the generator, completes the system. This motor operates at a synchronous speed for a fixed frequency and, as the output frequency from the cycloconvertor varies, so does the speed of the motor.

An equally important part of the system is the refrigeration plant. This plant provides liquid helium to the generator and motor to maintain the low temperatures required for superconductivity. Other components include a transmission bus to transmit the generated electricity to the cycloconvertor and motor, a braking resistor for dynamic braking, and miscellaneous switches and controls.

The primary advantages of this system are the same as those for an electromagnetic system. The flexibility of the system allows the prime mover to be located anywhere on the ship and to transmit power to any number of motors without significantly affecting the size and complexity of the transmission system. In addition, the system is light, having a total weight of from 0.78 to 0.91 pound per horsepower (Reference 25) and an efficiency of about 95 percent. The refrigeration plant requirements are low, and plant weight is about 10 percent of the total system weight.

At present, the state of the art has not developed sufficiently to permit use of a superconducting electric transmission in the SES. The largest generator built to date is 6700 horsepower (References 26 and 27). It is about 5 feet long by 3 1/2 feet in diameter. Additional development work on higher horsepower equipment is planned in England, Germany and the United States. Generators as large as 50,000 HP (direct current) and 6,700,000 HP (alternating current) are feasible for the mid to late 1980's.

Parametric Data. Table 5-2 summarizes pertinent information on transmission systems.

The gear system is at present, the only lightweight, efficient system available in high horsepower ranges. As a result, the discussion of system components in the following section will be limited to the components of a gear system

CHARACTERISTICS OF TRANSMISSION SYSTEMS ·

System	Efficiency	Capacity (HP)	Weight (pounds/HP)
Gear*	97.0%	40,000	1.0
Superconducting electric	95%	6,700	0.85
Hydraulic	75%	4,000	8.8
Electro-mechanical	90%	40,000	15.5

*Typical system design with planetary gear box.

TABLE 5-2

COMPONENT DESIGN

Gear Boxes. High-speed, heavy-duty applications demand the use of helical or spiral bevel gears. These gears run more quietly and have lower impact loads than other types because the contact surfaces between gears overlap, transferring the load gradually from one tooth to the next. Straight-tooth gears, on the other hand, carry the entire load on only one tooth at a time, generating noise and instantaneous high loads. Double helical gears are most common in high-power parallel and concentric shaft applications. The two opposite sets of helix-tooth forms on each gear produce counterbalancing axial forces and eliminate the need for thrust bearings. Wider face widths are allowable as compared with those of other gears. Spiral bevel gears are used to transmit power between intersecting shafts. They produce thrust loads which must be absorbed by suitable thrust bearings.

Double helical gears are capable of transmitting up to 30,000 horsepower per mesh at linear speeds of up to 30,000 feet per minute. Spiral bevel gears are capable of transmitting up to 25,000 horsepower per mesh at linear speeds of up to 8,000 feet per minute.

For initial estimating purposes the weight of gear boxes can be taken as 0.17 pound per horsepower for right-angle gears and 0.08 pound per horsepower for planetary gears, based on a 10,000-hour life. These numbers will vary greatly, depending on gear box life, service, and construction. A procedure follows for determining the weight and size of the gear box in greater detail.

First, the input horsepower, input rpm, required reduction ratio, and duty cycle must be known. From the input horsepower and duty cycle an equivalent torque for the gear box can be determined using these equations:

$$T \quad \frac{(HP)(63,000)}{rpm} \tag{5-1}$$

$$T_e \quad \left[\frac{\Sigma T^3{}_t}{\Sigma_t}\right]^{1/3} \tag{5.2}$$

whe:

HP = input horsepower at gears

rpm = input revolutions per minute

T = input torque-inch-pounds

= input time-hours

T_e = equivalent torque-inch-pounds.

An approximate sizing of a gear box is based on the load-carrying capacity and the life of the gears. The procedure used is based on information presented in References 28 through 30. Values are based on minimum gear size, utilizing aircraft gear technology and fully hardened teeth.

A "K" factor defining the load-carrying capacity of the gears must also be determined. K factors of 368 for helical gears and 209 for spiral bevel gears are recommended for 10,000 hours of operation at 3600 rpm. Other K factors can be determined from the equation

$$K \quad \frac{S_{ac}^2}{C_K{}^2\, C_D} \tag{5-3}$$

where

S_{ac} allowable compressive stress-pounds/inch^2 (see Table 5-3)

C_K contact stress factor (4677 for helical; 7570 for spiral bevel)

C_D overall derating factor (2.3 for helical; 1.9 for spiral bevel)

K load-carrying capacity-pounds/inch^2.

The allowable compressive stress varies with the number of cycles and can be obtained from Table 5-3. The number of cycles is the product of the speed (rpm) and design life (minutes).

These values of K, ranging from 200 to 370 pounds per square inch, may be compared with values of 100 to 150 pounds per square inch used in present marine gear technology.

ALLOWABLE COMPRESSIVE STRESS OF GEARS

No. of Cycles	S_{ac} Double Helical Gears (pounds/inch^2)	S_{ac} Spiral Bevel Gears (pounds/inch^2)
2.16×10^{10}	121,000	135,000
4.32×10^{9}	132,000	147,000
2.16×10^{9}	136,000	151,000
1.08×10^{9}	140,000	156,000
2.16×10^{8}	157,000	174,000

TABLE 5-3

Figures 5-3 and 5-4 can next be used to select a weight factor and an input gear ratio.

INITIAL GEAR RATIO FOR MINIMUM WEIGHT

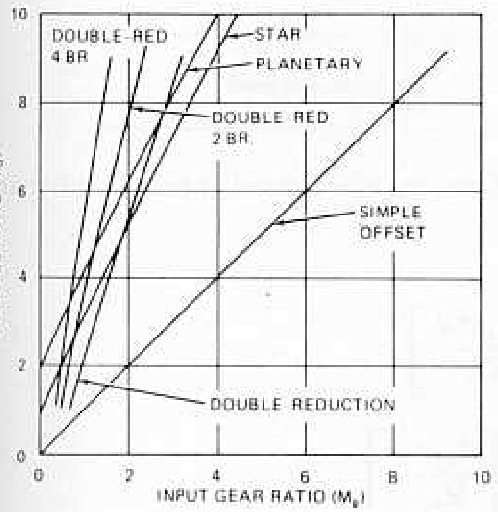


FIGURE 5-3

GEAR WEIGHT FACTOR CURVES

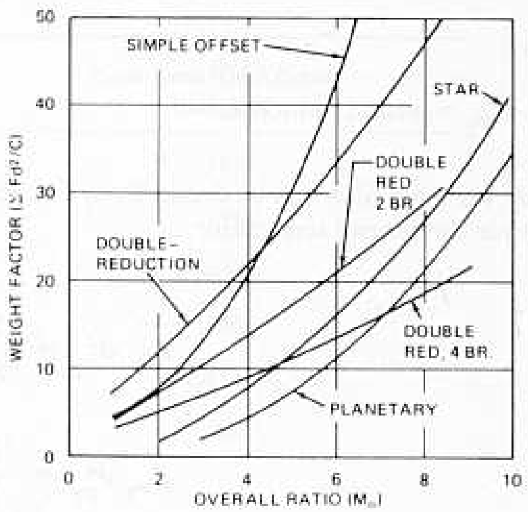


FIGURE 5-4

The weight factor is $\Sigma Fd^2/c$ where

$c = 2T_e/K$

M_o = overall gear ratio

M_g = input gear ratio

F = gear face width-inches

d = gear or pinion diameters-inches.

With the K factor and equivalent torque known, the ΣFd^2 can be found. By multiplying the ΣFd^2 by the aircraft application factor in Table 5-4, an approximate weight is found for the gear box. This weight includes gearing, shafts, bearings, and immediate support structure but does not include accessories, such as lubrication.

WEIGHT APPLICATION FACTORS FOR GEARS

Application	Factor	Typical Construction
Aircraft	0.25 – 0.30	Magnesium and aluminum
Hydrofoil	0.30 – 0.35	Lightweight steel
Commercial	0.60 – 0.625	Cast or fabricated steel

TABLE 5-4

Given the input ratio, M_g, and the overall ratio, M_o, the second reduction ratio is determined from

$$M_{g_2} = \frac{M_o}{M_{g_1}}$$

where

M_{g_2} = second reduction

Fd^2 for the pinion can be found from Equation (5.5) for the first reduction and from Equation (5.6) for the second reduction:

$$F_{p_1} d_{p_1}^2 = \frac{2T}{bK} \left[\frac{M_{g_1} + 1}{M_{g_1}}\right]$$

$$F_{p_2} d_{p_2}^2 = \frac{2TM_{g_1}}{bK} \left[\frac{M_{g_2} + 2}{M_{g_2}}\right] \tag{5-6}$$

where

b = number of branches transmitting torque

d_p = pinion diameter

F_p = pinion face width.

For any system, the greater the number of branches, the smaller the gear box. With an epicyclic system there is a geometric limitation on the maximum number of branches in the system. This value can be obtained from Figure 5-5.

OPTIMUM NUMBER OF BRANCHES FOR AN EPICYCLIC SYSTEM

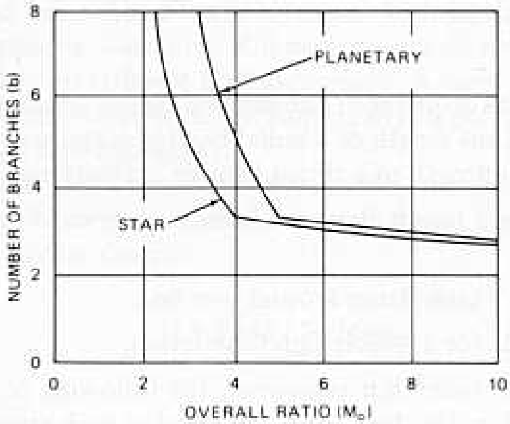


FIGURE 5-5

Once a value of the ratio F/d is selected, values can be found for F_p and d_p. The most compact unit will have the highest ratio of F/d. The recommended maximum value for double helical gears is 14:1. For values for spiral bevel gears, see Table 5-5.

GEAR WIDTH-DIAMETER RATIO

Overall Gear Ratio	Gear Face Width/Gear Diameter
1	.212
2	.335
3	.400
5	.434
7	.454
10	.476

TABLE 5-5

Equations (5.5) and (5.6) and the ratio F/d can be used to obtain values of F_{p1} and d_{p1} and F_{p2} and d_{p2}, respectively. Gear diameters, d_{g1} and d_{g2}, are found using the ratios M_{g1} and M_{g2}.

There is no standard procedure for sizing the gearbox. A simple layout, using known gear sizes, will give an approximate size. The following factors should be included.

a. Shafts: an approximation of shaft sizes, based on use of high-strength steel, can be obtained from the following equation:

$$D = 0.07\,T^{1/3} \tag{5.7}$$

where

D = diameter-inches

T = torque-inch-pounds.

b. Bearings: the outside diameter of the bearing can be approximated as twice the inside diameter, and the length of a radial bearing is approximately equal to the inside diameter. The length of a thrust bearing is about one half the inside diameter.

c. Clearance: allow for a 1-inch clearance between gears and casing with 1/2-inch casing wall.

d. Flange: allow for a 2-inch flange around gear box.

e. Foundation: provide for a 4-inch high foundation.

In cases where a composite reduction is desired, the following procedure should be used to establish the reduction for each gear box type, considering each type individually, as discussed previously. Set up a graph with the initial gear reduction as the abscissa – from $M_{g_1} = 1$ to $M_{g_1} = M_o$ – and with ΣFd^2 as the ordinate, as shown in Figure 5-6.

COMPOSITE GEAR REDUCTIONS

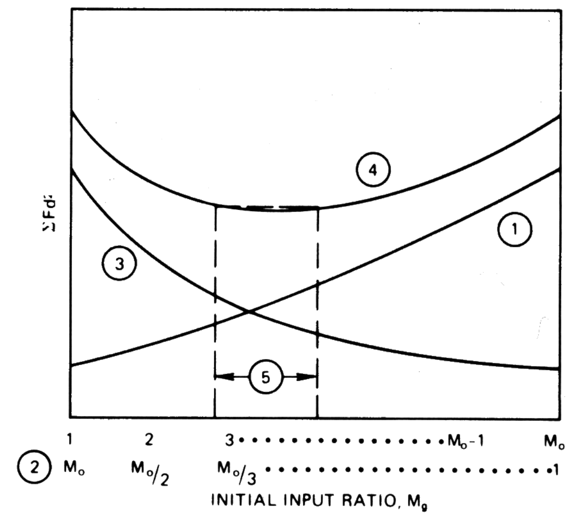


FIGURE 5-6

1. For the first reduction, plot values of M vs. ΣFd^2 from Figure 5-4, remembering to multiply values of $\Sigma Fd^2/c$ by c.
2. Set up an abscissa scale of values of M_{g_2} corresponding to M_{g_1}.
3. Plot values of M_{g_2} vs. ΣFd^2 from Figure 5-4 for the second reduction type.
4. Add curves for each reduction to obtain a curve for the total reduction.
5. Select a combination from the lowest range of ΣFd^2 for the composite system

Lubrication. A lubrication system is required for cooling and for reduction of friction in gear boxes and thrust bearings. The cooling requirements determine the size of the lubrication system, which includes a pump, reservoir, and heat exchanger, plus piping, valves, and filters.

The heat exchanger is the largest component in the system. Either air or seawater can be used for cooling. With an oil-air heat exchanger, fans are required to ensure a sufficient air flow. The ship's lift fans could be used for this purpose by allowing for a small increase in power requirements. The sidewalls could be used as a heat exchanger, with lubricating oil circulating along the sidewalls below the cushionborne waterline. Oil passages would have to be incorporated directly into the hull plating to obtain an efficient heat exchanger. A more conventional approach would be the proven, compact internal oil-seawater exchanger, with a pump-and-piping system providing the supply of seawater to the heat exchanger.

With either system a conservative approximation of the cooling requirements can be found as follows, assuming all power losses in the gear boxes and thrust bearings are in the form of heat and all heat is transferred away by the oil:

$$Q = 2545\,(1-e)\,(hp_i)$$

where

Q = heat loss-Btu/hour

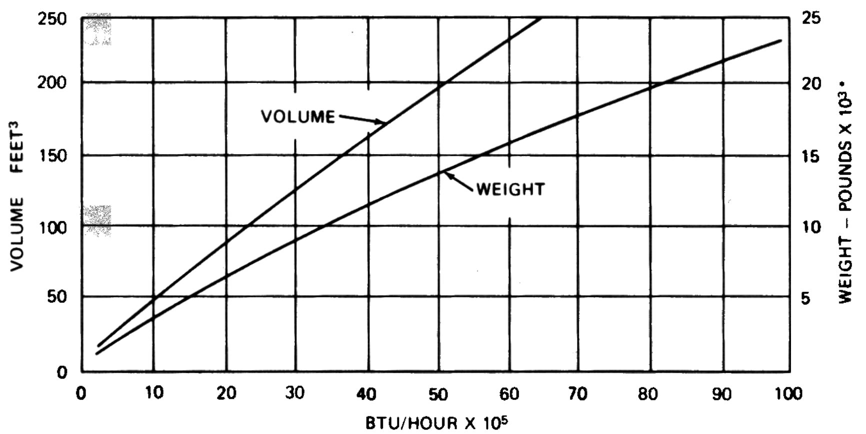
e = overall efficiency of gear box or bearing

hp_i = horsepower input to gear box or bearing.

The horsepower required to run an oil-seawater lubrication system is about 1.4 horsepower per 100,000 BTU per hour. The power required to run an oil-air lubrication system would be about ten times as great because of the fan power requirements.

The size and weight of a lubrication system utilizing an oil-seawater heat exchanger may be estimated from Figure 5-7. Using an aluminum shell in lieu of a steel shell on the exchanger would reduce the system weight by about 10 percent. Use of an oil-air heat exchanger would reduce the system weight by about 20 percent. The system volume would be about the same in both cases, but the oil-air heat exchanger would need to be of a high, wide, and thin configuration such that the maximum possible surface area would be facing the fan. A 400,000 BTU-per-hour unit might be 10 feet high by 16 feet wide by 6 inches thick. An approximation of lubrication system weight for typical SES installations can be obtained from Figure 5-8.

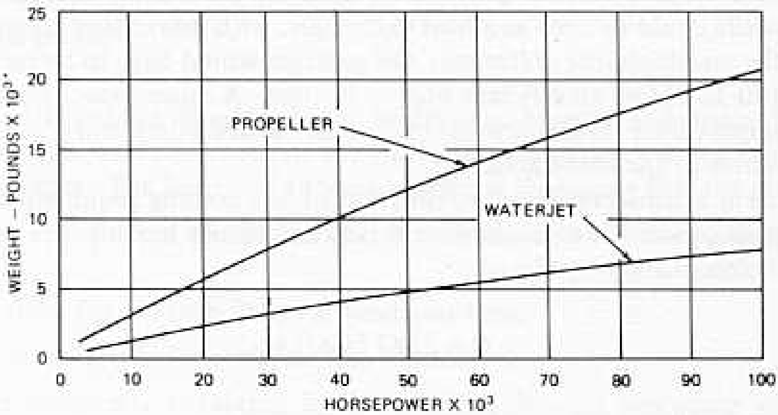
LUBRICATION SYSTEM WEIGHT AND VOLUME



*Includes heat exchanger, seawater, lubricating oil, pump, piping, valves and filters.

FIGURE 5-7

TYPICAL LUBRICATION SYSTEM WEIGHT
(TRANSMISSION ONLY)



*Includes heat exchanger, seawater, lubricating oil, pump, piping, valves, and filters.

FIGURE 5-8

Shafting. Shafting is used to transmit power from the power plant to the propulsor. Currently, solid mild steel shafting is used for commercial vessels. For naval vessels, where weight is critical, hollow, high-strength steel shafting is used, with a ratio of inside to outside diameter of 0.67:1. Thin wall tubing has been proposed for use as inboard shafting to achieve a further weight reduction (see Reference 31). Such tubing could be a centrifugal casting or rolled steel. Solid shafting should be used outboard, since the seawater environment causes fretting and corrosion and reduces the fatigue limit of the shafting. With a solid shaft a smaller percentage of the metal would be in contact with the seawater as compared to a hollow shaft. Solid shafting should also be used at the bearings and couplings to minimize the size of these components.

The size and weight of steel tube shafting can be approximated from Equations (5.9) and (5.10). The size and weight of solid steel shafting can be approximated from Equations (5.11) and (5.12). A weight reduction of about 30 percent can be realized by using titanium shafting rather than steel.

$$D_o = 4.65 \left[\frac{(HP)_s}{n}\right]^{1/3} \tag{5.9}$$

$$W = 0.25\ D_o^{\,2} \tag{5.10}$$

$$D_o = 3.0 \left[\frac{(HP)_s}{n}\right]^{1/3} \tag{5.11}$$

$$W = 2.64\ D_o^{\,2} \tag{5.12}$$

where

D_o = outside diameter of shaft-inches

$(HP)_s$ = shaft horsepower

n = shaft speed-rpm

W = shaft unit weight-pounds/foot.

The weight per foot of shafting can be roughly approximated as 0.002 pound per HP for solid steel shafting and 0.00023 pound per HP for hollow aluminum shafting.

Couplings. Couplings are needed to make semipermanent connections between shafts. The two types are rigid couplings and flexible couplings. Use of rigid couplings should be restricted to accurate, rigid installations. In view of the flexibility of SES structure, flexible couplings would normally be used on such craft. They can accommodate small amounts of lateral and angular misalignment, thus avoiding the stress that would occur if rigid couplings were used. Additionally, flexible couplings are capable of absorbing minor impacts due to fluctuations in shaft torque and speed. General information on some types of flexible couplings is given in Table 5-6. Gear couplings should generally be used for applications over 10,000 horsepower. If large amounts of misalignment or impact loading are caused by vibration, the development of a large, flexible-member coupling is required. The flexibility offered by the gear coupling should be adequate for SES applications.

TYPES OF FLEXIBLE COUPLINGS

Type	Maximum Misalignment	
	Parallel	Angular
Gear	0.02 inches	1 1/2 degrees
Chain	0.04 inches	2 degrees
Flexible	0.08 inches	1 degree

TABLE 5-6

Universal joints are special couplings designed to connect shafts at angles of up to 15 degrees. Presently, no universal joints available are capable of transmitting horsepower at the speeds required for SES operation. Data on two universal joints proposed for the SES by Twin Disc, Inc. are included in Table 5-7. These would be made of aluminum and titanium and would take approximately 24 months to develop.

CHARACTERISTICS OF UNIVERSAL JOINTS

Horsepower	rpm	Efficiency	Length	Diameter	Weight (pounds)
23,400	900	0.99	50.4	26.8	2,000
46,800	900	0.99	68.7	35.3	4,600

TABLE 5-7

Rigid and flexible member couplings require no lubrication; gear and chain couplings require periodic greasing; large universal joints require continuous forced-oil lubrication.

The weight of a coupling may be approximated from

$$W \quad 2.1\,D^{2.5} \tag{5.13}$$

where

W weight of coupling-pounds

D solid shaft diameter-inches.

An approximation of coupling weight is 0.0018 pound per horsepower. The length is approximately 2 times the solid shaft diameter, and the outside diameter is approximately 1.7 times the solid shaft diameter. The sizes and weights of typical universal joints are given in Table 5-7.

Clutches. Clutches are used to start or stop a machine or a rotating element without starting or stopping the prime mover. On the SES the clutch will be located between the prime mover and the gear box, thus allowing multiple turbines to drive one propulsor. Clutches considered for the SES transmit torque by three basic means:

Positive: positive tooth or jaw

Friction: disc or drum

3 Eddy-current: electromagnetic field

These clutches may be actuated electrically, hydraulically, or pneumatically. Specialized variations of these clutches include centrifugal, overrunning, synchro self-shifting, and synchromesh clutches. Centrifugal clutches engage automatically at a predetermined speed. Overrunning clutches drive in one direction only. Synchro self-shifting and synchromesh clutches are described below.

The synchro self-shifting clutch has been used in several marine propulsion drives. It is a positive clutch which automatically engages when the speeds of the two shafts are synchronized. This requires that the driving shaft be brought to the speed of the driven shaft before engagement. It can be operated as an overrunning clutch or can be locked after engagement to operate in either direction.

The synchromesh clutch is a combination positive friction clutch. The friction clutch functions initially, being engaged to bring the two shafts to synchronous speeds. At that time a sensing unit records zero relative motion between the two shafts and engages the positive tooth coupling. The friction clutch then disengages. Since the friction clutch is used only for synchronization, it is relatively small; power is required only during engagement and disengagement of the clutch.

The synchromesh clutch may be most suitable for main propulsion systems. The power required during activation is 150 psi oil (or 100 psi air) for actuation and 50-watt, 115-volt electricity for control. Because of its normally positive engagement, this clutch does not require cooling. Maintenance and lubrication are minimal. Since the size of the clutch is a function of the torque transmitted, the unit should be located on the highest-speed shaft. Sizes and weights of some typical units are given in Table 5-8.

TYPES OF SYNCHROMESH CLUTCHES

Torque (inch-pounds)	Diameter (inches)	Length (inches)	Weight (pounds)
120,000	19.5	19.5	550
475,000	24.0	36.0	1200
745,000	25.0	39.0	2000

TABLE 5-8

An approximation of synchromesh clutch weight is 0.0028 pound per horsepower.

Bearings. Line-shaft bearings are required both to support shafting and to support radial loads applied to the shafting, such as the propeller weight. Thrust bearings are required to absorb thrust from the propeller and transmit it to the hull.

Marine line-shaft bearings are generally of the journal type and depend on a film of oil lubrication (gravity fed or forced) to reduce friction. They are usually conservatively designed and ruggedly constructed. Ball and roller bearings may be used with a resultant decrease in weight and in friction loss; there is also, however, a decrease in reliability and an increase in required maintenance when compared to the standard marine journal type.

Since bearings are likely to have the shortest life of any part of the transmission, ease of replacement must be a prime consideration. This may be achieved by using a special shaft design or split bearings. Bearing spacing must be determined through a complex vibrations analysis (see Reference 17). An approximation of the number of bearings can be obtained by dividing the shaft weight by the bearing capacities. A maximum bearing spacing of 22 times the shaft diameter should be the limit. For light loads ball bearings can be used; for heavier loads roller bearings should be used.

If heavy loads and increased temperatures are encountered, forced-oil lubrication is required. The weight of a line-shaft bearing and housing can be approximated from Equation (5.14). The outside diameter will be about 2 1/2 times the solid shaft diameter, and the length will be about equal to the solid shaft diameter. An approximation of the weight is 0.004 pound per horsepower.

$$W = 5.3\,D^2 \qquad (5.14)$$

where

W = weight of bearing with steel housing – pounds

D = outside diameter of solid shaft – inches.

For propeller-driven ships, two thrust bearings are required per propeller shaft. A main thrust bearing absorbs the propeller's forward thrust, while a second smaller bearing is used to absorb astern thrust. These bearings should be located at the tail shaft to minimize compressive loading on the line shafting.

As are line-shaft bearings, thrust bearings are usually conservatively designed and ruggedly constructed oil bearings. The use of aluminum housings can reduce the weight of these bearings by about 30 percent. The thrust loads and speeds necessary for the SES are too high to permit use of roller thrust bearings. Development of heavy-duty (over 100,000-pound thrust), high-speed (900 rpm) roller bearings does not appear to be practical in the next five years. Development of air bearings of this capacity also appears to be well beyond the state of the art.

The oil bearings support the thrust load on a film of lubrication. The efficiency of these bearings is about 99.7 percent. Forced-oil lubrication is required to maintain the film and dissipate heat.

The weight of the thrust bearing with aluminum housing can be approximated from Equation (5.15). The height and width of the thrust bearing is approximately three times the solid shaft diameter. The length is about 1.5 times the width. An approximation of the weight is 0.17 pound per horsepower.

$$W \quad 0.025\,T \qquad (5$$

·e

W thrust bearing weight - pounds

thrust load on bearing · pounds.

For waterjet-driven ships, the thrust bearing would be an integral part of the pump, and is not considered as a part of the transmission system.

Parametric Weight Data. To summarize the component weight of the transmission system, the gear system shown in Figure 5-1 will be used as an example. Table 5-9 summarizes the type, quantity, weight per horsepower, and efficiency of each component. In a similar system for waterjet-driven ships, the angle (or drop) boxes may not be required and the thrust bearing is integral with the waterjet pump.

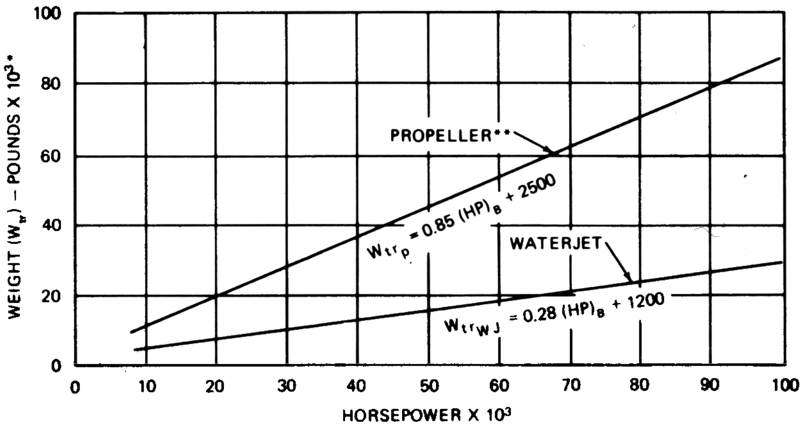
CHARACTERISTICS OF TYPICAL PROPELLER TRANSMISSION SYSTEM

Component	Horsepower	Quantity	Unit Weight/ Horsepower	System Weight/ Horsepower	Overall Efficiency (percent)
Angle	50,000		0.170	0.340	99 ea
Planetary box	50,000		0.080	0.080	99
Coupling	50,000		0.0018	0.005	
Clutch	25,000		0.003	0.003	
Radial bearing	50,000	4	0.004	0.016	
Thrust bearing	50,000		0.17	0.17	99.7
Shaft (line)	50,000	25 ft.	0.0002/ft.	0.005	
Shaft (tail)	50,000	10 ft.	0.002/ft.	0.020	
Lubrication	50,000		0.240	0.240	
			TOTAL	0.879	96.7

TABLE 5-9

It can be concluded that, for the total transmission (gear) system, the weight will be about 1 pound per horsepower and the overall efficiency will be about 97 percent. Figure 5-9 can be used to estimate the total weight of a transmission system.

TYPICAL TRANSMISSION SYSTEM WEIGHT



*Includes gears, coupling, clutches, bearings, shafting, lubricating oil system, saltwater cooling.

**Includes two right-angle drives, or helical reduction gear train ("drop-box").

FIGURE 5-9

Chapter VI
PROPULSORS

WATERJETS

Waterjet propulsors have been employed successfully in high-speed displacement craft and hydrofoil boats and in one of the current 100-ton SES testcraft. As in any propulsive device, the waterjet produces thrust by accelerating a working fluid (water) to achieve a net change in the momentum of the fluid equal to the thrust generated. The basic components of a waterjet propulsion system are (1) an inlet to ingest water and usually an associated inlet diffuser to reduce the velocity and increase the pressure of the fluid; (2) a duct, which transfers the ingested water to the pump and can also further diffuse the flow; (3) a pump, driven by a suitable prime mover, which increases the pressure and velocity of the water; and (4) a discharge nozzle, which further increases the water velocity and can be movable for purposes of steering control. The components of a typical SES waterjet propulsion system are shown schematically in Figure 6-1.

TYPICAL WATERJET PROPULSION SYSTEM

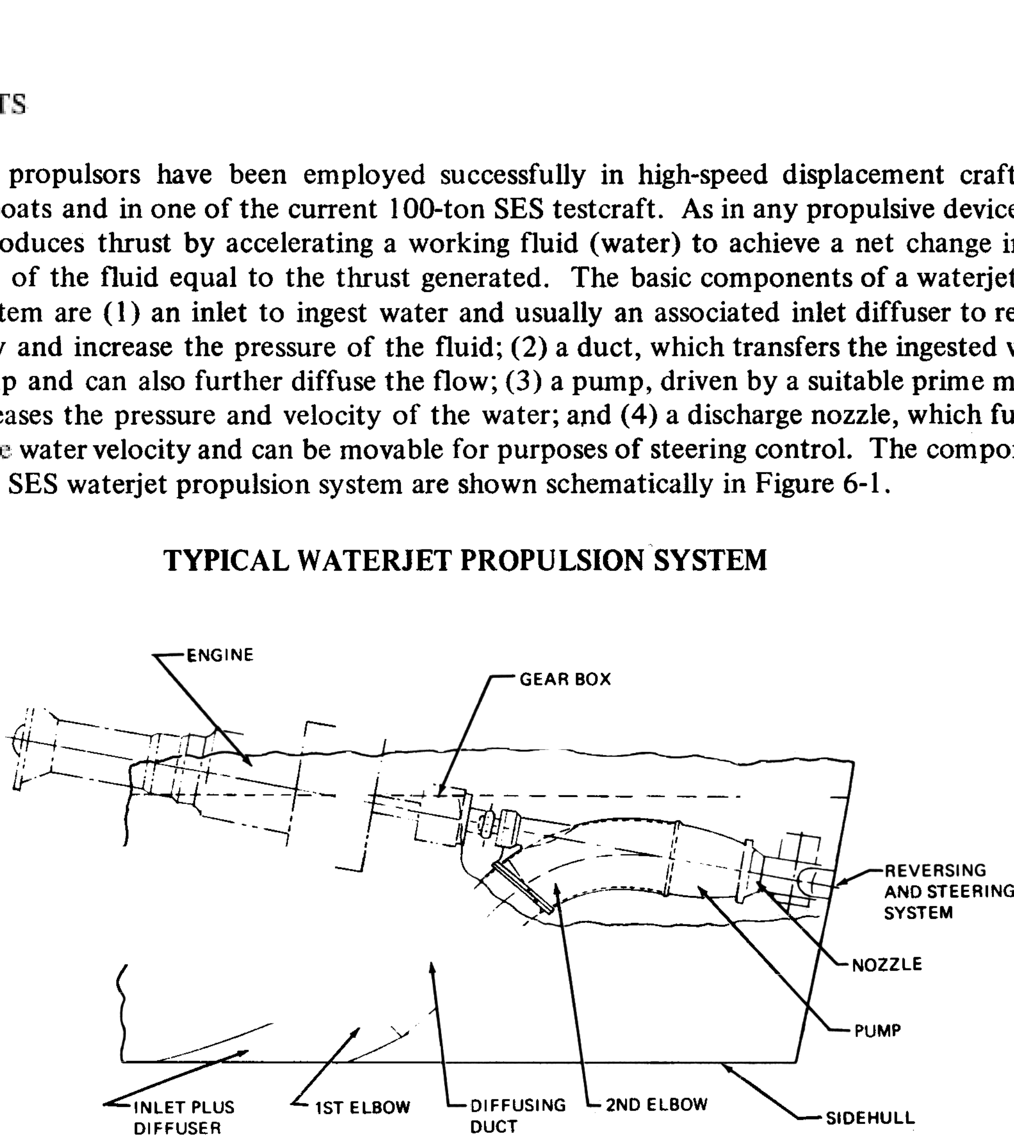


FIGURE 6-

Pump Configuration. Pumps are normally divided into three classes, which are characterized by the direction of flow through the impeller as (1) radial flow (or centrifugal), (2) mixed flow, and (3) axial flow (or propeller). They can also be classified according to the head developed as (1) high head, (2) intermediate head, and (3) low head. These qualitative classifications are somewhat overalpping. For a flow rate, Q, that is relatively low, and a head, (H) that is relatively high, a centrifugal pump is usually used. For a relatively high flow rate and low pump head, an axial inducer pump is used. For higher pump heads, multistage centrifugal or axial pumps can be used. To obtain higher pump speeds at low inlet suction pressures, a double-suction centrifugal pump is sometimes used. An inducer pump will invariably be used as a first stage of a multiple-stage axial pump and inherently has the highest suction performance of any impeller type. Additionally, a two-speed, coaxial-shaft two-stage pump is sometimes used to achieve high suction performance with a low rpm stage and high pressure with a high rpm stage. These various multiple-stage pumps are schematically illustrated in Figure 6-2.

TYPICAL MULTIPLE-STAGE WATERJET PUMP IMPELLERS

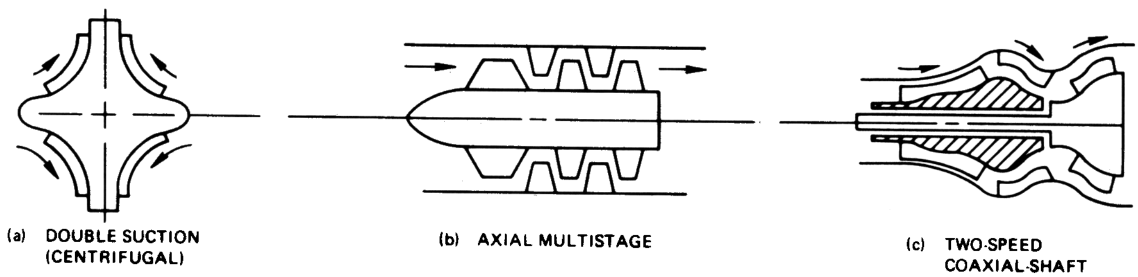


FIGURE 6-2

A more precise classification of pumps is by specific speed, N_S, or basic specific speed, which is defined by

$$N_S = \frac{n\,Q^{1/2}}{H^{3/4}} \tag{6.1}$$

In the most commonly employed form of Equation (6.1), n is in revolutions per minute (rpm), Q is in gallons per minute (gpm), and H is in feet per stage, with Q and H taken at the pump's best efficiency. On this scale (again with overlapping at classification boundaries), typical values of N_S for single-stage pump impellers are shown in Figure 6-3.

TYPICAL SINGLE-STAGE WATERJET PUMP IMPELLERS

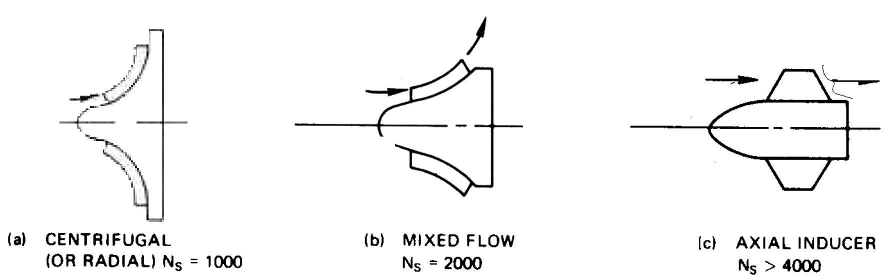


FIGURE 6-3

Types of Inlets. Three general types of inlets are used for a waterjet: (1) flush, (2) semi-flush and (3) ramscoop inlets. The latter can be hull-mounted, strut-mounted, or pod-mounted. The three types are illustrated in Figure 6-4.

WATERJET INLET CONFIGURATIONS

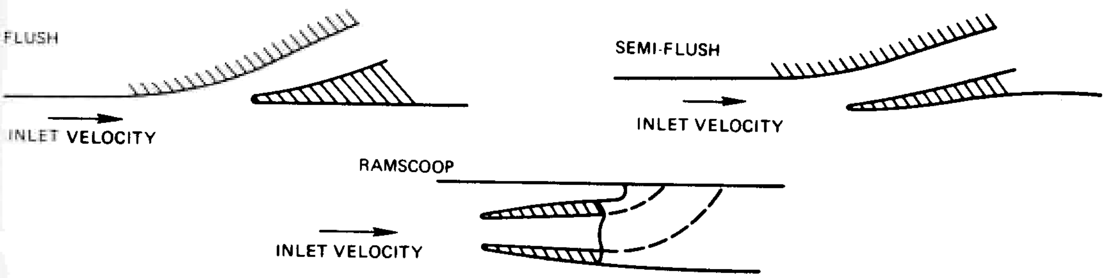


FIGURE 6-4

Because of the large vehicle speed range over which the waterjet inlet must operate in a high speed SES, the variable-area geometry inlet must be used rather than the fixed-area geometry inlet. Internal cavitation at low vehicle forward speed is avoided or minimized by variable geometry (Figure 6-5). The flush-inlet has a lower system drag than a ramscoop inlet, whereas the ramscoop inlet has the better pressure recovery and lesser susceptibility to flow distortion. A low propulsion system drag is essential to high overall propulsive efficiency at high speeds.

VARIABLE-AREA WATERJET INLET CONFIGURATIONS

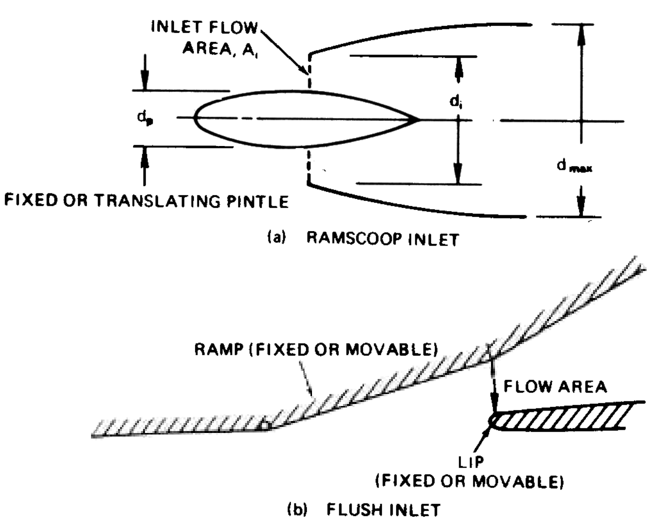


FIGURE 6-5

Performance Characteristics. In addition to the specific speed, N_S, as defined in Equation (6.1), the other important coefficients which relate to waterjet performance are as follows:

Suction specific speed, S, is expressed by

$$S = \frac{n\,Q^{1/2}}{(NPSH)^{3/4}} \tag{6.2}$$

where

n = rotative speed – rpm
Q = flow rate of pump – gpm
$NPSH$ = net positive suction head – feet

The suction specific speed is associated with the inception of cavitation of the pump impeller. Each family of geometrically similar pumps has a critical value of S, above which this family will cavitate. The value of the critical S is nearly independent of the type of pump. The Hydraulic Institute suggests a maximum S value of 8,500 for cavitation-free operation, although a poorly designed pump may cavitate at lower values, and an exceptionally well-designed pump may reach values high as 12,000 to 15,000 before beginning to cavitate. Current technology permits a suction specific speed in the 20,000 to 30,000 range at hump.

Pump efficiency, η_p, can be calculated as follows:

$$\eta_p = \frac{\rho_w\,(\Delta H)\,Q\,g}{550\,(HP)_S} = \frac{0.117\,(\Delta H)\,Q}{(HP)_S} \tag{6.3}$$

where

ΔH = total pump head rise – feet (for a single-stage pump, H in Equation (6.1) is equal to (ΔH)
Q = pump flow rate – feet^3/second
ρ_w = density of water (1.99) – pounds-second^2/feet^4
g = gravitational acceleration (32.2) – feet/second^2
$(HP)_S$ = pump input shaft horsepower – *hp*.

Net propulsive coefficient, NPC, can be expressed as

$$NPC = \frac{T_n\,V}{550\,(HP)_S} \tag{6.4}$$

where

T_n = net thrust – pounds
V = ship speed – feet/second.

Flow rate, Q, is related to net thrust, T_n, by

$$Q = \frac{T_n + D_p}{\rho_w \; V(r-1)} = \frac{T_n + D_p}{1.99 \; V(r-1)} \tag{6.5}$$

where

r = jet velocity ratio (ratio of jet velocity to ship speed – typically, 2.0 at 80 knots, 3.0 at 40 knots, and 4.2 at 25 knots)
D_p = propulsion system drag – pounds.

The propulsion system drag, D_p, is given in terms of ship speed, V, inlet area, A_i, and a drag coefficient, C_D, by the following expression:

$$D_p = 1/2 \, C_D \, \rho_w \, V^2 \, A_i = 0.995 \, C_D \, V^2 \, A_i \tag{6.5a}$$

where

$C_D \cong 0.1 + 0.065 \, (1 - R)^{2.41}$ (typically for flush inlets)
R = inlet velocity ratio (ratio of inlet to ship speed – typically 0.8 – 0.85).

In advanced design stage, D_p, as given by Equation (6.5a), cannot be determined unless the inlet area is known. To facilitate a first-stage prediction, $T_n + D_p$ in Equation (6.5) may be taken as equal to 1.1 to 1.2 times the ship drag for pod mounted inlets, or 1.05 times ship drag for flush mounting. The pump head rise, ΔH, can be expressed as follows

$$\Delta H = \hat{H} \, V^2/2g = \hat{H} \, V^2/64.4 \tag{6.6}$$

where

$\hat{H} = r^2 \, (1 + K_n) - 1 + K_d R^2 + 2Zg/V^2$
K_n = nozzle loss coefficient (typically, 0.02)
K_d = duct loss coefficient (typically, 0.45)
Z = pump centerline elevation head, above water surface – feet.

Then, using the above-suggested values with, say an 80-knot ship speed, $\hat{H}$ can be calculated to be about 3.5, which value can be used for first-step predictions for Z in the order of 10 feet.

The net positive suction head, $NPSH$, which is the head available at the pump inlet, can be expressed as

$$NPSH = \frac{V^2}{2g} \, (1 - K_d) + (H_a - H_v) - Z \tag{6.7}$$

where

H_a = atmospheric head – feet
H_v = water vapor head – feet
$H_a - H_v$ = 32.5 feet under normal conditions at sea level

The net propulsive coefficient *NPC*, and the pump efficiency, η_p, have the following relation:

$$\frac{NPC}{\eta_n} = \frac{2\,(r - 1 - C_D/2R)}{[r^2\,(1 + K_n) - 1 + K_d R^2 + 2Zg/V^2]\ \eta_T} \tag{6.8}$$

The pump efficiency η_p, relative to pump type, specific speed, and flow rate is shown in Figure 6-6.

WATERJET PUMP EFFICIENCY

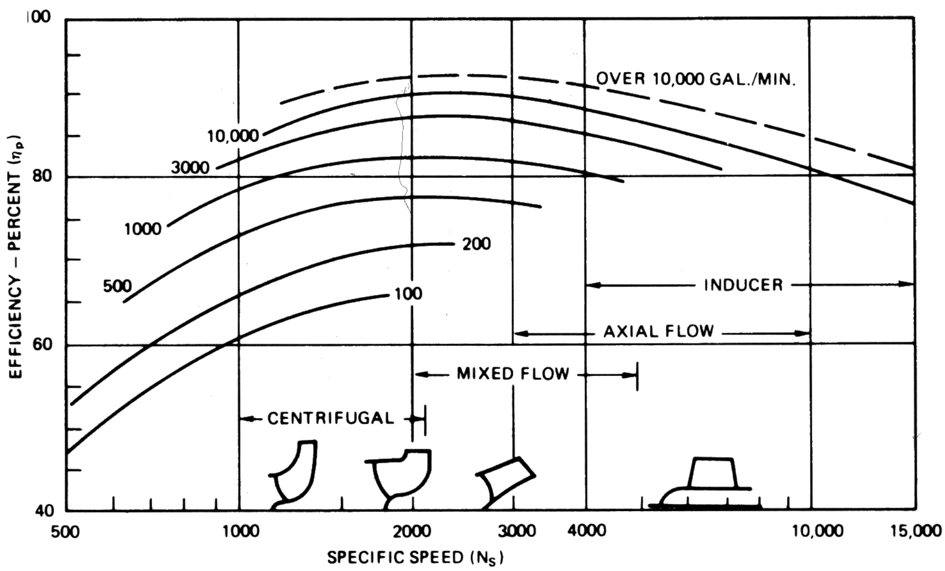


FIGURE 6-6

Design Procedures. Selecting a propulsion system that will perform satisfactorily under the full range of operating conditions that must be met involves a number of compromises among conflicting requirements. The procedure involves an iterative process in which a system is selected to meet one set of conditions – for example, performance at design speed; the system is then evaluated under the other conditions – for example, performance at hump and cruise speeds. The best propulsion system is the one that will enable the ship to achieve the specified range at specified speed with the minimum total weight of propulsion machinery, entrained water, and fuel, subject to power limits. Thus, the problem reduces to an optimization of each component of the propulsion system. Such optimization will not be considered in this book; References 32 and 33, for example, deal specifically with the optimization process. A detailed analysis of propulsive efficiency at hump speed is not necessary for advanced design. In general, the waterjet propulsion system designed to meet the requirements at design condition will be able to produce sufficient thrust to overcome the drag at hump speed with an adequate margin for acceleration.

Emphasis in this discussion is on selection of the pump, using the system parameters for which typical values were suggested in Equations (6.5) through (6.8). The design procedure can be outlined as follows:

1. Obtain total required thrust at design speed and calculate total flow rate by using Equation (6.5).
2. Select the number of pumps. If identical pumps are used in parallel, the required thrust and flow rate for each pump are obtained by dividing those values obtained in Step 1 by the number of pumps.
3. Estimate pump head rise by using Equation (6.6).
4. Select the type and number of stages of the pump. For a multi-thousand-ton SES designed for 80 to 100 knots, large flow rates (200,000 to 300,000 gpm) and high head rise (1,000 to 2,000 feet) will probably be required. A typical pump type suitable for this application is a multi-stage axial flow configuration, which offers a high flow-rate-per-unit frontal area capability (small size) and high efficiency (high specific speed per stage). The pump head per stage can be based on the selected number of stages.

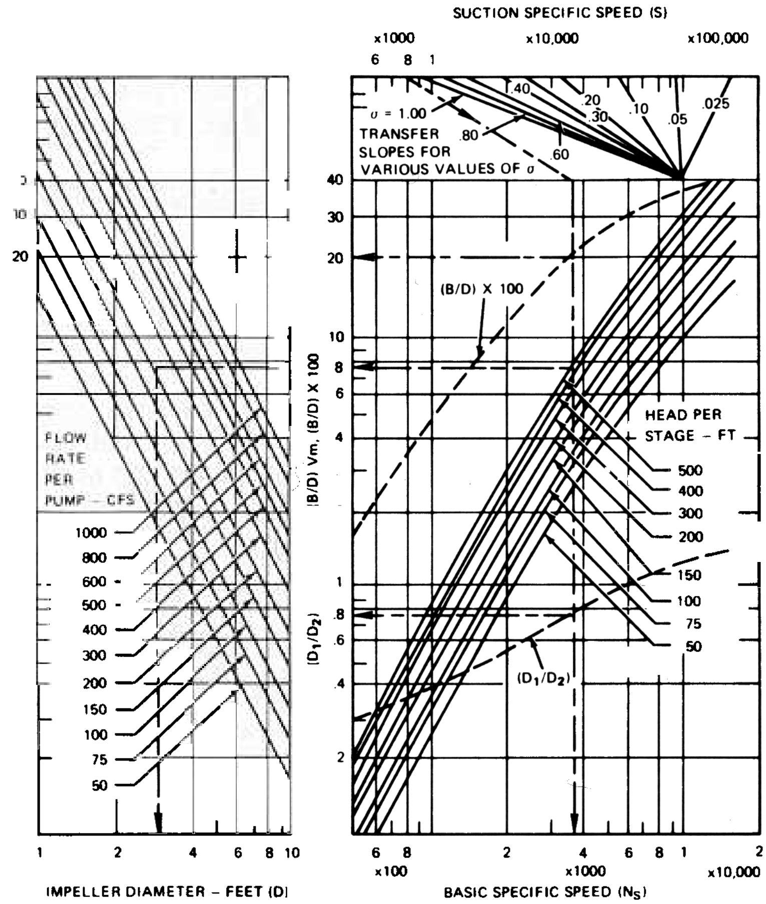
The usual way of preceeding with pump selection from this point is to select a value of rotative speed based on the selected pump type, and obtain the pump characteristics for the appropriate family from pump characteristic curves. However, this process requires the accumulation of data on all types of pumps, since a large number of pumps may have to be considered as candidates. Alternatively, a single diagram Figure 6-7, can be used for advanced design purposes. This chart is applicable to any type and stage of pump, provided the pump is single speed, and is used as explained below.

5. Calculate the net positive suction head, *NPSH*, by using Equation (6.7) and obtain the Thoma cavitation number, σ:

$$\sigma = \frac{NPSH}{H}$$

6. Choose a value of suction specific speed, S. For advanced design, a value of 8,500 (maximum value suggested by Hydraulic Institute) is proposed.
   On the upper scale in Figure 6-7 draw a line at the selected S with a slope of σ across the strip; continue from there vertically down to the lower scale of N_S such that N_S can be read and n can be calculated from Equation (6.1).
7. Draw a horizontal line from the intersection of the vertical line with the dashed curve marked $(B/D) \times 100$ in Figure 6-7 to the middle scale, where (B/D) can be read. This is the ratio of impeller width, B, to impeller mean diameter D, where the impeller width is the distance (in feet) from the outside of the hub to the impeller tip.
   Draw another horizontal line from the intersection of the vertical line with the curve for the appropriate value of H. This line is then carried across to

WATERJET PUMP SIZING CHART



See Reference 32

FIGURE 6-7

the left side to the constant-flow-rate curve for Q. Where this line crosses the middle scale, read $(B/D)\ V_m$, where V_m is the meridional velocity (in feet/second) at impeller discharge. Calculate V_m by using the value of B/D as determined before.

8. Ascertain the ratio of impeller inlet diameter to outside diameter as shown at the intersection of the vertical line (N_S = constant) with the (D_1/D_2) curve.

9. Draw a vertical line downward (from the Q = constant) on the left scale to the impeller-diameter scale, where D can be read. This is the mean impeller diameter. The outside impeller diameter can be determined from the following:

$$D_2{}^2 = \frac{2D^2}{1 + (D_1/D_2)^2}$$

10. Estimate pump efficiency, η_p, by applying the values obtained for N_S and Q and the data given in Figure 6-6. Equation (6.3) thus can be used to obtain pump input horsepower, *SHP.*
11. Determine the net propulsive coefficient, *NPC,* of the system by using the pump efficiency η_p, and Equation (6.4) or (6.8).
12. Establish the casing diameter for axial flow pumps, which can be taken as 1.2 times the outside impeller diameter, D_2, determined in Step 9. For advanced design layout purposes, the overall length, including thrust bearing, inlet elbow, pump, and exit nozzle, can be taken as four times the casing diameter for a two-stage axial flow pump.

A numerical example paralleling the foregoing explanation of the pump selection process follows. Consider an SES with a drag at 80 knots (135 feet/second) in a sea state 3 of 235,000 pounds (including a 10 percent allowance for propulsor drag and margin).

1. Based on Equation (6.5), calculate the total flow rate:

$$\text{Total flow rate} = \frac{235{,}000}{(1.99)(135)(2\text{-}1)} = 870 \text{ cfs} = 390{,}000 \text{ gpm.}$$

2. Select four pumps (two per sidehull). For each pump, obtain

$$T_n = \frac{235{,}000}{4} = 58{,}750 \text{ pounds}$$

$$Q = 217.5 \text{ cfs} = 97{,}500 \text{ gpm.}$$

3. Using Equation (6.6), calculate pump head rise, ΔH:

$$\Delta H = (3.5)(135)^2/64.4 = 1{,}000 \text{ feet.}$$

4. Select a two-stage axial pump (one inducer plus one stage). Thus, the head per stage, H, is 500 feet.

5. From Equation (6.7), calculate

$$NPSH = \frac{(135)^2}{64.4}(1 - 0.45) + 32.5 - 10 = 180 \text{ feet and,}$$

$$\sigma = \frac{NPSH}{H} = \frac{180}{550} = 0.36.$$

6. Choose $S = 8{,}500$; enter S and σ on Figure 6-7 and read $N_S = 3{,}700$. Thus, from Equation (6.1), calculate

$$n = \frac{(3{,}700)(500)^{0.75}}{(97{,}500)^{0.50}} = 1260 \text{ rpm.}$$

From Figure 6-7 read $B/D = 0.2$ and $(B/D)\ V_m = 7.5$.
Therefore, the meridional velocity, V_m, = 37.5 feet/second

8. From 'igure 6-7 read $D\ 'D_2\quad 0.75$.

9. From Figure 6-7 read the impeller mean diameter, $D = 2.95$ *feet* $\quad 35.4$ *inches.*
Therefore, impeller width, B, $= (2.95)(0.2) = 0.59$ *feet.*
For impeller outside diameter, calculate

$$D_2 \quad \sqrt{\frac{2(2.95)^2}{1+(0.75)^2}} \quad 3.34 \text{ feet.}$$

The numerical values read from Figure 6-7 should be viewed as approximate and should be checked for internal consistency. The flow rate, for instance, may be checked by

$$Q = \pi DB\ V_m = \pi(2.95)(0.59)(37.5) = 205 \text{ cfs}$$

which is in good agreement with the *217.5 cfs* of Step 2.

10. From Figure 6-6, read $\eta_p\quad 0.915$. 'herefore, Equation (6.3) yields the SHP as

$$(HP)_S \quad \frac{1.99)(1000)(217.5)(32.2)}{(550)(0.915)} \quad 27{,}700 \text{ hp/pump.}$$

From Equation (6.8) obtain the net propulsive coefficient, $NPC \cong 0.54$, which can also be obtained by using Equation (6.4). Therefore, the total required horsepower estimated at 80-knot speed is 110,800 (4 X 27,700).

Calculate the casing diameter of the axial-flow pump as

$$D_C \cong (1.2)(3.415) = 4.1 \text{ feet}$$

and, calculate the overall length, including thrust bearing, inlet elbow, pump, and exit nozzle as

$$L \quad 4(4.1 \quad 16.4 \text{ feet.}$$

Materials and Operating Life. The factors to consider in selecting materials for the waterjet are seawater corrosion, high-velocity erosion, cavitation damage resistance, and structural strength. Titanium, copper-nickel alloys, and stainless steel are especially resistant to corrosion by seawater and have high strength properties. Either castings or forgings can be used, depending on the design complexity of the part, requisite strength, and cost. Galvanic corrosion occurs when dissimilar metals of widely different electrolytic potentials are placed in contact in seawater. This can be controlled by appropriately placing electrical insulations between certain mating parts. Materials that have exhibited good cavitation damage resistance are titanium and stainless steel.

Through proper materials selection and careful hydraulic design, the pump can be designed for an operating life in excess of 10,000 hours and for operating time in deep cavitation in excess of 100 hours.

Parametric Weight and Data. The weight of a waterjet propulsion system is affected by many design variables, some fixed (for example, the thrust required at a given speed) and others open to design choice. Among the latter are (1) the basic design of the pump, (2) the pump-module arrangement (that is, parallel, series, or both), (3) the design speed, (4) the jet-velocity ratio, and (5) the type of ducting and associated losses.

The best means of estimating weight is to make a preliminary layout of the inlet, elbow, pump, and exit nozzle to obtain the overall envelope length and diameter of the waterjet (Figure 6-8) and then to apply a construction density (specific weight) to the volume. A reasonable approximation of the dry weight is obtained by multiplying the volume of an equivalent cylinder containing the envelope dimensions by a specific weight of 2.0 (0.72 pounds/inches^3). This weight will include the inlet elbow, pump, thrust bearing assembly, and exit nozzle. The wet weight will be approximately 50 percent greater, or a specific weight of 3.0. The typical length-diameter ratio (L/D_C) of two-stage, single-suction, single-speed axial flow pumps is about 4. The diameter, D_C, is the pump case diameter, which is about 1.2 times the pump impeller outside diameter for axial flow pumps. The approximate weight of water in pounds/foot contained in the inlet/diffuser duct per foot of length is given by

$$(Wt)_{H_2O} = 0.348 \; [D_C \; (inches)]^2$$

where the weight of the ducting itself is small compared with the weight of the water and can be ignored in a preliminary estimate. These values have been obtained from design and fabrication experience with pumps for waterjet application.

PRELIMINARY LAYOUT FOR WATERJET WEIGHT ESTIMATION

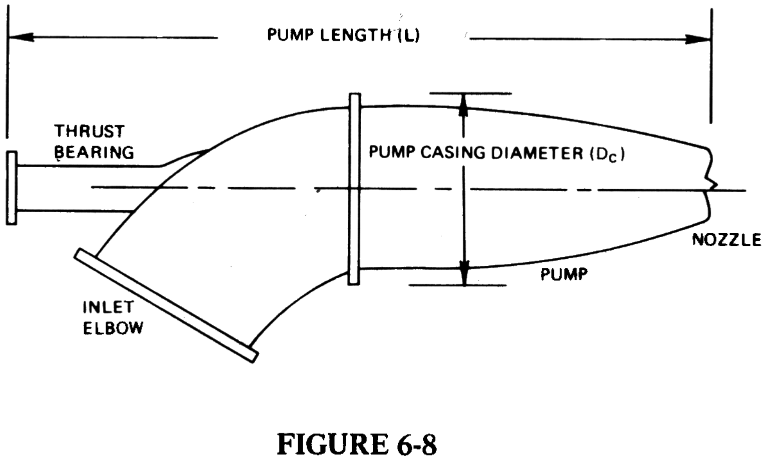


FIGURE 6-8

For advanced design purposes, the data given in Figure 6-9 may be used. The design conditions represented by these data are ship speeds of 80 knots and single-speed, single-suction, two-stage axial pumps (one inducer plus one stage). The weight of the pump has been estimated on the basis of data provided in Reference 32. The SES gross weight noted on Figure 6-9 is for L_C/B_C = 2.0, P_C/L_C = 1.5, 80-knot speed, sea state 3, and four pumps per ship.

ESTIMATED VARIATION OF DIAMETER, WEIGHT, AND ROTATIVE SPEED OF WATERJET SYSTEMS

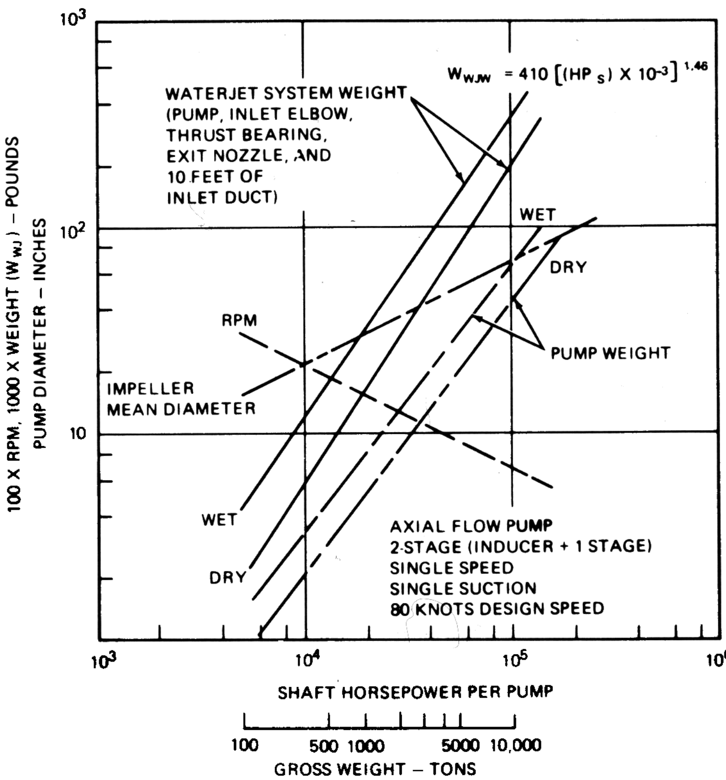


FIGURE 6-9

WATER PROPELLERS

Conventional subcavitating marine propellers are not suitable for high-speed SES propulsion. Operating speeds of 40 knots or more are practical only for very short periods. Sustained operation at such speeds will result in excessive cavitation erosion damage to the propellers and probably also to any ship structure in the propeller wake. By contrast, supercavitating and superventilated propellers are designed to operate with the back or suction side completely enclosed within the vapor cavity and therefore are prime contenders for SES propulsion. For a superventilated propeller, cavitation is ensured by introducing air to the backs of the blades. Supercavitating type propellers can operate fully or partially submerged. Partially submerged propellers are almost always superventilated, while fully submerged propellers can be either supercavitating or superventilated. Partially submerged superventilated propellers, designed by purely "cut and try" methods, have been successfully employed at speeds up to 200 miles per hour on racing hydroplanes. The Bell 100-ton SES test craft is the first large craft to use partially submerged superventilated propellers. Initial tests and model data point to successful operation. The physical and performance characteristics of supercavitating propellers will vary considerably with propeller mounting and submergence. Typical mounting systems are shown in Figure 6-10.

Physical Geometry. Supercavitating propellers can absorb large amounts of power with small diameters. Typical diameters for 80-knot SES propellers are 3 to 4 feet for a 100-ton craft and 10 to 15 feet for a 2000 to 4000-ton craft, respectively. Partially submerged propellers must

SUPERCAVITATING PROPELLER MOUNTING SYSTEMS

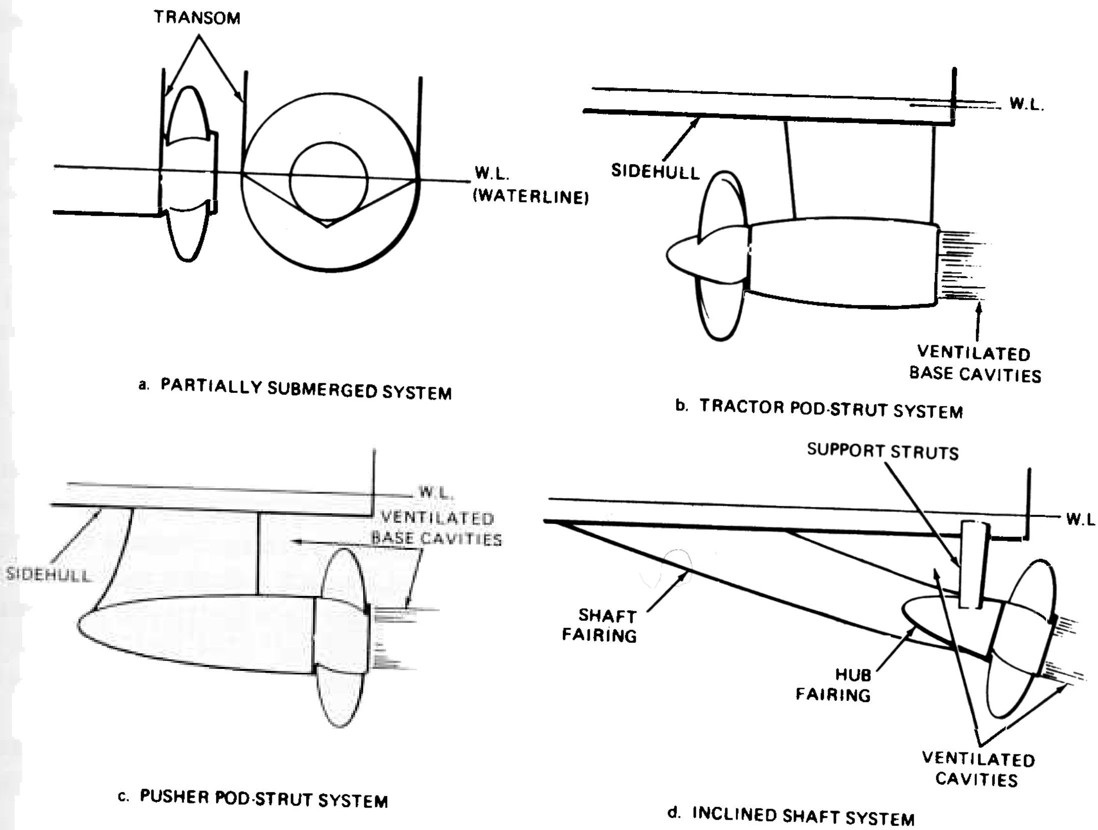


FIGURE 6-10

have larger diameters to compensate for the unwetted (non-thrust) part of the propeller disc. Typical hub-diameter ratios are 0.3 to 0.35 for fixed-pitch propellers and 0.35 to 0.45 for controllable-pitch propellers.

Typical design pitch-diameter ratios for high-speed operation are 1.2 to 1.8. Most SES propellers will have controllable (variable) pitch to maximize off-design operation, for instance, in high sea states, and also, for ship control at low speeds.

Optimum design point and off-design performance will usually be achieved with propellers having expanded-area ratios of 0.2 to 0.4. Even a moderate expanded-area ratio (0.4) can lead to severe flow disturbance and performance loss at hump-speed. Fully submerged propellers will usually have three to five blades (depending on the wake), while partially submerged propellers will usually have six to eight blades. Highly skewed propellers are used to minimize the unsteady thrust and torque due particularly to the blade-water entry and exit. Inclined shaft fully submerged propellers are not feasible at 80 knots due to cyclic changes in blade angle of attack inducing cavitation.

Selection and Advanced Design. A large number of considerations affect the selection and advanced design of a supercavitating propeller. First, the propeller must produce the thrust necessary to meet or exceed craft drag requirements, including rough water allowance and margin for acceleration, with power available at all speeds – and especially at cruise speed and hump speed. The rotative speeds of the propeller must be operationally compatible with available prime movers and transmission/reduction gears. The number and location of the propellers and the maximum propeller diameter must be consistent with sidehull geometry and with machinery arrangements. Additionally, the propeller must operate at acceptable levels of radiated, water-borne noise.

The first step in the design process is to determine the number and location of propellers. For most SES's two propellers will be used, one mounted at the stern of each sidehull; a third, centerline-mounted propeller may also be considered, particularly one that could be used only at low speeds and retracted at high speeds. With fully submerged propellers, pod-strut mounting systems are used. However, there is a strong tendency to use partially submerged propellers; with this kind of system, a small pod-like fairing may be required at the sidehull transom to enclose gears. The primary reasons for preferring partially submerged propellers are twofold: (1) the installation drag is much lower than for a fully submerged propeller, and (2) various means are available, such as flow ramps to effectively match the hump and cruise speed requirements. This is a much more severe problem for fully submerged propellers with constant effective disc area.

In the design of supercavitating propellers, various compromises must be made between high-speed and hump-speed performance requirements. In advanced propeller design it is necessary to select one uppermost design condition or design point, and there is a strong temptation to select hump speed because of the difficulties usually encountered in producing adequate hump-speed thrust. Unfortunately, the low advance coefficients dictated by such a design will result in disastrous high-speed performance in terms of severe blade-pressure face cavitation, poor specific fuel rate and poor efficiency, even if controllable pitch is employed. The best approach is to design for face cavitation-free operation at design speed and to select the corresponding gear ratio and turbine rotative speed to allow increased rotative speed and thrust at hump speed.

Propeller design point efficiency is determined primarily by a set of non-dimensional parameters, including expanded area ratio (EAR), number of blades, blade section characteristics, and the parameters given below in equation form.

Thrust-loading coefficient, C_{TP}:

$$C_{TP} = \frac{T_p}{1/2\, \rho_w A_o V_a^2 \beta} = \frac{T_p}{0.995\, A_o V_a^2 \beta} \tag{6.10}$$

Advance coefficient, J:

$$J = \frac{V_a}{nD} \tag{6.11}$$

Stress coefficient, $\overline{s}$:

$$\overline{s} = \frac{S_d}{1/2\, \rho_w V_a^2} = \frac{S_d}{0.995\, V_a^2} \tag{6.12}$$

Cavity cavitation number, σ:

$$\sigma = \frac{P_o - P_C}{1/2\, \rho_w V_a^2} = \frac{P_o - P_C}{0.995\, V_a^2} \tag{6.13}$$

prime movers and transmission/reduction gears. The number and location of the propellers and the maximum propeller diameter must be consistent with sidehull geometry and with machinery arrangements. Additionally, the propeller must operate at acceptable levels of radiated, water-borne noise.

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Stress coefficient, $\overline{s}$.

$$\overline{s} = \frac{S_d}{1/2\,\rho_w\,V_a^2} = \frac{S_d}{0.995\,V_a} \tag{6.12}$$

Cavity cavitation number, σ

$$\sigma = \frac{P_o - P_C}{1/2\,\rho_w\,V_a^2} = \frac{P_o - P_C}{0.995\,V_a^2} \tag{6.13}$$

Froude number, $F_{N\,0.7}$:

$$F_{N\,0.7} = V_{r0.7}/\sqrt{gD} = V_{r0.7}/5.67\sqrt{D}. \tag{6.14}$$

where

A_o = propeller disc area – feet^2
T_p = propeller open water thrust – pounds
V_a = speed of advance – feet/second (V (1-w))
D = propeller diameter – feet
n = rotative speed – rps
S_d = allowable, average stress – pounds/foot^2
P_o = ambient pressure – pounds/foot^2
P_C = pressure within blade cavities – pounds/foot^2
V = ship forward speed – feet/second
w = wake fraction
ρ_w = mass density of water (1.99) – pounds-second^2/feet^4
β = percentage submergence of propeller
$V_{r0.7}$ = resultant velocity of the blade at 70 percent radius – feet/second.

Froude number does not apply to a fully submerged propeller but must be considered when the propeller operates near or through the free surface. For a partially submerged propeller, however, with the suction side of the propeller blade fully ventilated, cavitation number reduces to zero and Froude number is not important.

Open water propeller performance is usually presented in terms of advance coefficient, J, thrust coefficient, K_T, torque coefficient, K_Q, and open water propeller efficiency, η_{p_o}, as follows:

$$K_T = \frac{T_p}{\rho_w\, n^2\, D^4} = \frac{T_p}{1.99\, n^2 D^4} \tag{6.15}$$

$$K_Q = \frac{Q_p}{\rho_w\, n^2\, D^5} = \frac{Q_p}{1.99\, n^2\, D^5} \tag{6.16}$$

$$\eta_{p_o} = \frac{K_T J}{2\,\pi\, K_Q} = \frac{K_T J}{6.283\, K_Q} \tag{6.17}$$

where

Q_p = propeller torque – pound/foot.

Propeller efficiency is related to the net propulsive coefficient (propulsive efficiency), NPC, by

$$NPC = \eta_{p_o}\,\eta_H\,\eta_R\,\eta_M = \frac{D_{TOT}\,V}{550\,(HP)_B\,\eta_T} = \frac{T_n\,V}{550\,(HP)_B\,\eta_T} \tag{6.18}$$

where

$(HP)_B$ = brake horsepower of the engine – HP

η_H = hull efficiency $(1-t)/(1-w)$

η_M = mounting efficiency = $1 - \dfrac{D_M}{D_{TOT}}$

η_R = relative rotative efficiency

η_T = transmission efficiency

t = thrust deduction, which is related to ship resistance, D_{TOT} by,

$D_{TOT} = (1-t)\,T_p$.

D_M = propulsor mounting drag – pounds.

The brake horsepower can be also calculated by using propeller torque, as follows:

$$(HP)_B = \frac{2\pi\,n\,Q_p}{550\,\eta_R\,\eta_T} = \frac{0.011\,n\,Q_p}{\eta_R\,\eta_T} \tag{6.19}$$

The wake fraction, w, is a function of the velocity distribution in the wake behind the ship, and the thrust deduction is a function of the negative pressure gradient at the ship's stern, which augments the hull resistance. In the SES, which has little structure below the waterline, it can be expected that the value of both w and t will be small and that hull efficiency, η_H, can be assumed to be unity. The relative rotative efficiency, η_R, is the ratio between propeller torque measured in open water and that measured with the propeller mounted to the ship. In general, the value of η_R does not depart materially from unity, being between 0.95 and 1.1 for most twin-screw conventional ships and between 1.0 and 1.1 for single-screw conventional ships. Thus, in the advanced design stage, it can be assumed to be unity. For partially submerged propellers the added drag of the mounting is small, and mounting efficiency, η_M, can be taken as unity for advanced design.

Fully submerged supercavitating propeller design point efficiencies will be typically, 60 to 70 percent, with corresponding net propulsive efficiencies being 45 to 55 percent. Efficiencies are usually relatively higher for partially submerged propellers, the increase resulting from the unwetted condition of the propeller hub. Thus, decreasing the submergence will usually increase the efficiency. For example, peak efficiency of a NSRDC 4281 propeller (at pitch ratio = 1.8) increases from 67 to 75 percent when changing submergence from 50 percent to 30 percent. Typical net propulsive efficiencies of partially submerged propellers at design point will be from 60 to 70 percent. The higher efficiency values can be achieved only when optimum parameters (diameter, rpm, pitch) can be selected. The thrust of a partially submerged propeller is linearly proportional to the wetted propeller area; thus, propeller thrust may be less with relatively little submergence (30 percent, say) than with 50 percent submergence, even though propeller efficiency is higher at the smaller submergence.

The effects of thrust-loading coefficient and stress coefficient on propeller efficiency can be summarized as follows. Efficiency always increases with increasing design stress. Typically, the lowest possible thrust-loading and highest advance coefficient lead to maximum efficiency. Relative to constant design stress, efficiency decreases with increasing blade number and hub diameter ratio (except for partially submerged propellers with unwetted hubs).

The performance of fully submerged propellers is considerably influenced by cavitation number. In the case of partially submerged propellers that are not fully ventilated, thrust can also vary considerably with cavitation number as well as with Froude number.

Those design parameters having the greatest influence on off-design performance are expanded-area ratio and blade camber. Off-design performance always improves with increasing ratios of design lift due to camber and to angle of attack. At hump-speed operating conditions, thrust can be limited by allowable rotative speed or available power. Maximum thrust will be usually obtained when the pitch is selected to give maximum power absorption at maximum rotative speed. Performance at hump speed generally improves with decreasing expanded-area ratio.

The configuration of the propeller mounting has a major influence on propeller selection and performance and on transmission selection. Partially submerged propeller systems usually require a larger sidehull transom area but no external-mounting appendages. To maximize propeller efficiency and minimize unsteady propeller thrust and torque, the propeller hub and the upper part of the propeller disc should be entirely in the shadow of the transom (see Figure 6-10(a)). This implies that the sidehull beam must be at least equal to the propeller diameter, and any such increase in transom area must be considered relative to resultant increases in sidehull drag. Where sidehull geometry is varied to increase low-speed thrust, as in the case of the Bell 100-ton test craft, sidehull drag, thrust deduction, and wake fraction can increase considerably. Partially submerged propeller shafts can be either inclined or horizontal. Comparatively speaking, the primary advantage of an inclined shaft system is that it has a beneficial effect on machinery arrangement. Shaft inclination may degrade propeller efficiency unless the effective rake of the blade, when pointing straight down, is zero at the design shaft angle.

Fully submerged propellers can be mounted on either a pod-strut system or an inclined shaft. The primary advantages of a pod-strut mounting system are the likely improvement in off-design performance realized through prevention of propeller ventilation at low speeds and reduction of blade cyclic or unsteady loads and stresses, particularly in association with tractor-system use. The primary disadvantages are the large pod-strut drag (typically, 20 to 40 percent of total craft drag) and the complexity and questionable reliability of the right-angle transmissions. Using inclined-shaft mounting systems for fully submerged propellers can eliminate the mechanical difficulties associated with the right-angle drive system; but additional brackets may be required to provide rigidity in the long, slender shaft, thus bringing about an increase in unsteady propeller-blade loadings and a reduction in net propeller efficiency. A large propeller inclination may also lead to the early onset of face cavitation, which can cause blade damage, increased blade stresses, and reduced propeller efficiency.

Blade Stresses and Mechanical Design. During the advanced design stage it is impractical to give detailed consideration to blade stresses and propeller mechanical design. Supercavitating propellers with superior theoretical hydrodynamic performance will usually have inadequate blade strength. Under existing methods of blade-stress analysis (for example, the equivalent-beam theory and the Lieberman Shell-Analysis Program with 108 degrees of freedom – a finite element method), the magnitude and distribution of the radial stresses of the airfoil and segment-shaped blade section can be closely predicted, but neither of these methods can be considered entirely satisfactory for stress analysis of the wedge-shaped blade section (supercavitating propeller). Thickening the blade to lower the blade stress will be accompanied by degradation in propeller efficiency

The size of propeller hub has an influence on propeller performance. For controllable-pitch propellers, two facets of mechanical design influence hub-diameter ratio, blade-hub attachment, and internal hub design. For maximum blade strength, individually forged blades are required. Blade attachments can result in larger hub-diameter ratios than desirable for maximum efficiency (about 0.30). Controllable-pitch mechanisms will usually require hub-diameter ratios of from 0.35 (from three- or four-bladed propellers) to 0.45 (for eight-bladed propellers).

Design Procedures. In general, much more comprehensive design procedures apply to supercavitating propellers than to conventional marine propellers. In addition to the large number of important design parameters common to both types, off-design performance, blade strength, and blade supercavitation must be considered from the outset with supercavitating propellers. Accurate means of predicting design point efficiency are not yet available, primarily because of the lack of the necessary theoretical tools. Attention is thus focused on available model test data for partially submerged propellers, as exemplified in Figure 6-11. (The procedure for designing fully submerged propellers is similar, but the cavitation number has to be taken into consideration.)

In the design procedure propeller performance must be taken into account under both design- and hump-speed conditions. It must incorporate means for ensuring that predicted performance corresponds to propeller designs having adequate blade strength, fully supercavitating or superventilating flow, and no face cavitation. For any partially submerged supercavitating propeller, there is always an advance coefficient, or J value, below which the propeller suction side is fully cavitating; when J is below the limit value, performance is independent of Froude number and cavitation number. This limiting J value is determined experimentally; for a NSRDC 4281 propeller with a pitch-diameter ratio equal to 1.8, it is 1.3.

Care must be taken in using data for an existing propeller design to predict the performance of a new design. The two propellers must have identical numbers of blades and very similar (say, within a range of 5 to 10 percent) expanded-area ratios, section cambers, and blade pitches. If test data are available for several pitch angles, performance at intermediate pitch angles can be obtained by cross-plotting data at fixed advance coefficients.

By selecting various combinations of propeller rotative speed, n, and propeller diameter, D, and thus advance coefficient, J, within the allowable range, and by estimating or calculating the various elements of drag and efficiency, Equation (6.18) and (6.19) can be used to determine the horsepower required. The specific procedure used to derive this minimum power combination involves the following steps:

1. Determine ship resistance or drag, including appendage drag, rough water allowance, and margin for acceleration.
2. Select one or more values of propeller diameter and propeller rotative speed. First approximations may be taken from Figure 6-13 for the SES gross weight of interest.
3. Based on the design speed, calculate the advance coefficient from Equation (6.11).
4. From extrapolation of data such as those in Figures 6-11a through 6-11c, estimate thrust and torque coefficients and propeller efficiency for the selected design submergence and pitch-diameter ratio.
5. Calculate propeller thrust and torque, using Equations (6.15) and (6.16).

CHARACTERISTIC EFFICIENCY CURVES FOR SUPERCAVITATING PROPELLER IN SUBMERGED OPERATION

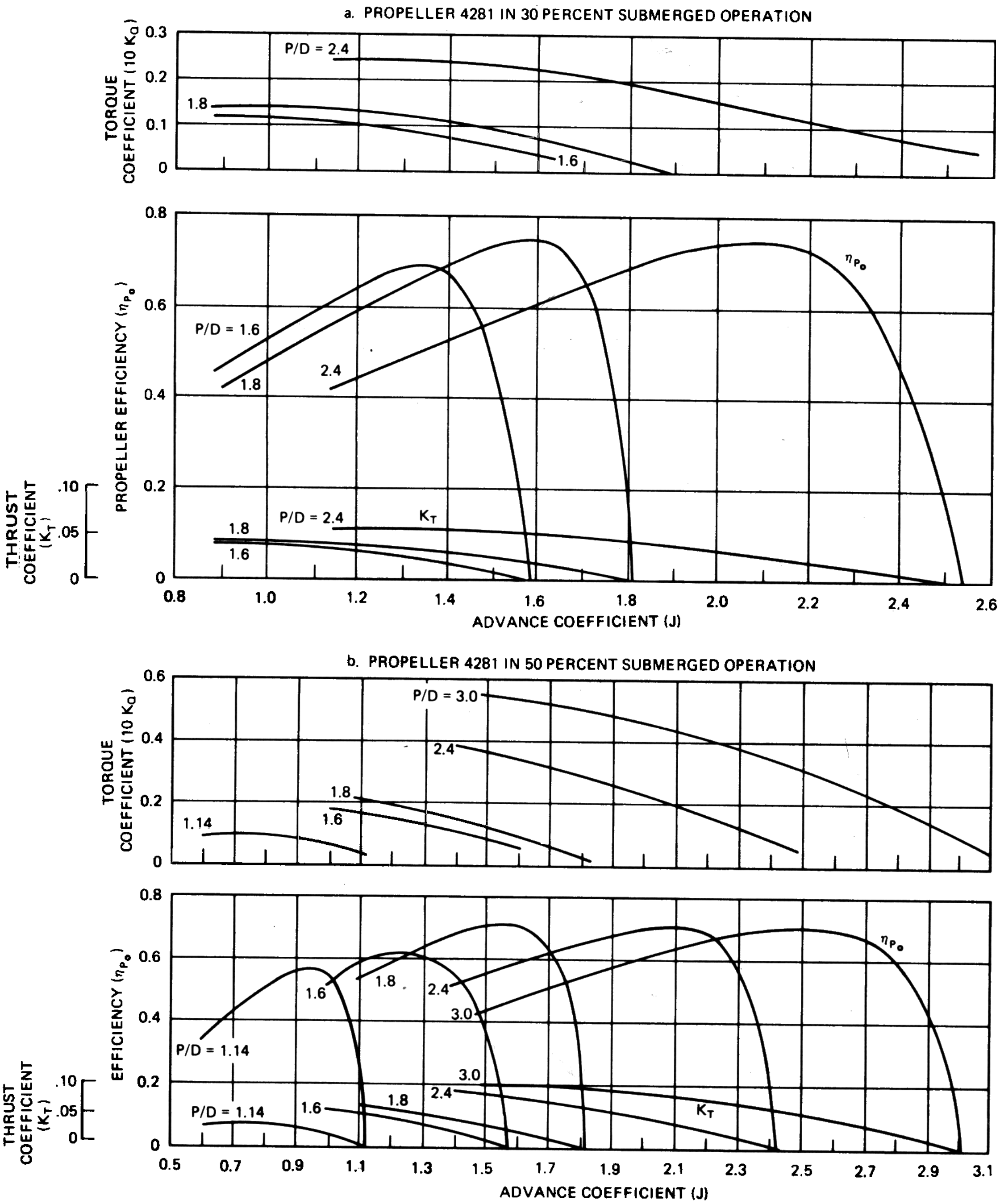
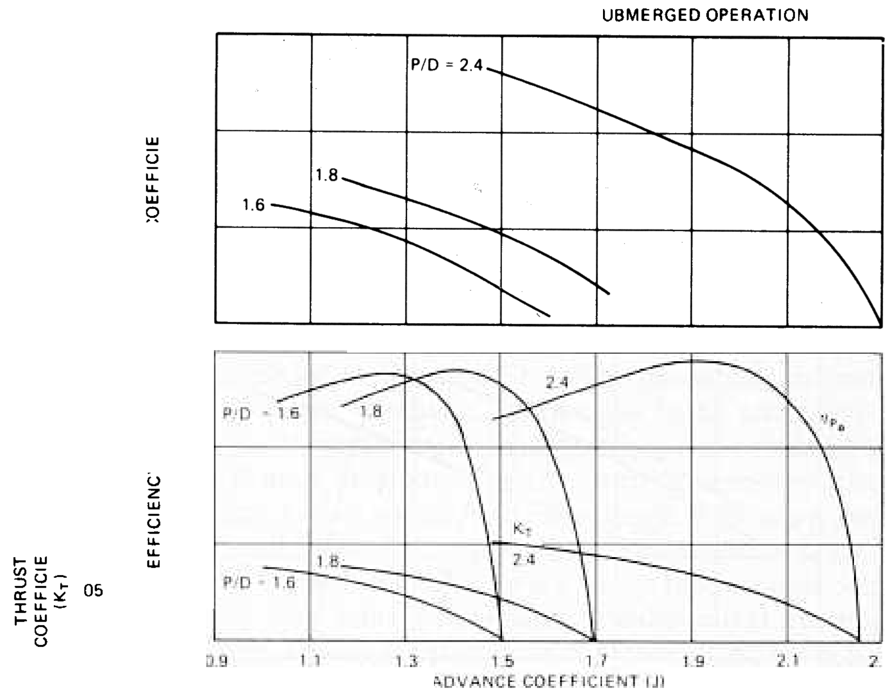


FIGURE 6-11

FIGURE 6-11 (CONTINUED)



6. Estimate hull efficiency, mounting efficiency, relative rotative efficiency, and transmission efficiency and calculate shaft horsepower.
7. Observing all design constraints, select a number of propeller designs meeting the constraints at design speed conditions. Estimate off-design performance for each propeller design and, using suitable engine characteristics, estimate hump speed thrust.
8. Repeat steps 2 through 7 (using propeller diameter based on design conditions) until a propeller can be selected that meets both design-speed and hump-speed conditions.

For advanced design purposes, the above procedure is too elaborate, and some simplifying assumptions are warranted. These assumptions are as follows:

1. A good representative propeller is a NSRDC 4281-series propeller with eight blades, zero skew, zero rake, 0.504 expanded-area ratio, 0.165 mean width ratio, and 0.039 blade thickness fraction.
2. Percent submergence is 50 percent.
3. Pitch-diameter ratio at design speed is 1.8.
4. Propeller selected for design speed conditions will be satisfactory at hump speed with a change in pitch angle, and rpm.
5. Hull efficiency, mounting efficiency, and relative rotative efficiency are unity.
6. Transmission efficiency is 97 percent.
7. Limit value for the advance coefficient is 1.3.

The advanced design procedure is best illustrated by the following numerical example:

Given: (1) design speed = 80 knots = 135 feet/second

(2) total thrust required = 374,000 pounds

(3) number of propellers = 2

(4) brake horsepower available = 66,800/propeller

Required: (1) propeller diameter

(2) rpm.

Trial No.	D (feet)	η (rps)	J	η_{P_o}	K_T	K_Q	Tp (pounds)	Qp (pound-feet)	$(HP)_B$
			Eq. (6.11)		Fig. 6-11b				
1									
2									
3									
4									
5									
6									

Solution the smallest possible propeller diameter = 12 feet and the corresponding rpm ≅ 8.65 X 60 – 525

η_{P_o} ≅ *0.64.* Interpolate between trials 5 and 6 for final solution.

To estimate propeller performance at hump conditions, the following empirical procedure may be used. At low operating advance coefficients, the efficiency of partially submerged propellers can be estimated with good accuracy in this manner, provided that the propeller is operated at a constant-pitch setting and that submergence or hub wetting does not change greatly between the selected and low speed conditions:

$$\eta_j = \eta_d (J/J_d), \text{ for } J < J_d \tag{6.20}$$

where

η_j = efficiency at any advance coefficient, J

η_d = efficiency at selected advance coefficient, J_d.

In Equation (6.20) the advance coefficient must be so selected that the efficiency, η_d, can be read from the straight-line portion of the curve to the left of the point of maximum efficiency. Essentially, in Equation (6.20) this straight-line portion of the curve is being extrapolated downward toward the low operating advance coefficients. Equation (6.20) can be used for any pitch-diameter ratio; it is not necessary to use the ratio selected for the design condition.

Off-design thrust and torque coefficients cannot be estimated by a simple expression such as Equation (6.20). Figure 6-12, which is derived from data for a number of fully and partially submerged propellers, depicts an average non-dimensional behavior of thrust coefficient. This curve is useful for estimating performance when no applicable design data can be obtained.

EFFECT OF ADVANCE COEFFICIENT ON PROPELLER THRUST

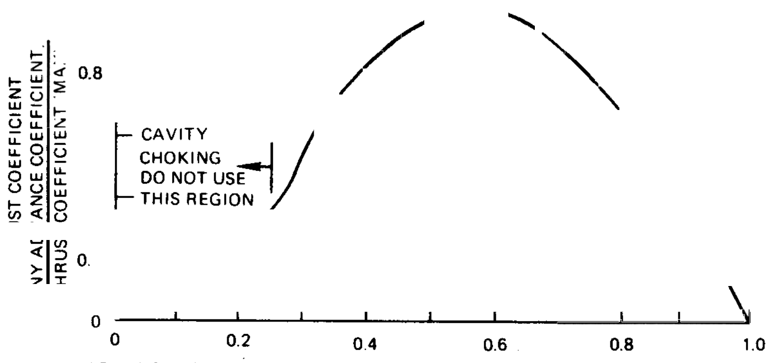


FIGURE 6-12

The thrust coefficient determined from Figure 6-12 can now be used to calculate the thrust produced using Equation (6.15); given η_j, obtained from Equation (6.20) (and previously determined K_T), the torque coefficient, K_Q, can be determined from Equation (6.17). Torque, Q_p, can then be calculated from Equation (6.16), and shaft horsepower from Equation (6.19).

Parametric Weight Data. Figure 6-13 can be used as a guide in estimating propeller weight. The rpm and diameter values shown are averages for a number of propeller designs, together with calculated values. The average line for propeller diameter is very close to Burtner's formula which was originally intended for use in connection with subcavitating propellers for a given shaft horsepower and rpm, as follows:

$$Diameter\ (feet) = 50\ \frac{\left[(HP)_S\right]^{0.2}}{(rpm)^{0.6}}$$

he weight is based on the following equation

$$Weight\ (pounds, \quad (40)\,(D) \quad \left(\frac{material\ weight}{steel\ weight}\right)$$

here

D = propeller diameter – feet

material weight = the weight of material used pounds/foot^3

steel weight = 489.6 pounds/foot^3

titanium (alloy Ti – 4A1-4B) weight = 277 pounds/foot^3

The weight thus calculated includes the weight of the control mechanism in the propeller hub.

The ship gross weight noted on Figure 6-13 is based on a cushion length-cushion beam ratio of 2.0, and a cushion pressure-cushion length ratio of 1.5, plus 80-knot speed, sea state 3, and two propellers per ship.

ESTIMATED VARIATION OF DIAMETER, WEIGHT, AND ROTATIVE SPEED OF OPTIMUM, PARTIALLY SUBMERGED PROPELLERS

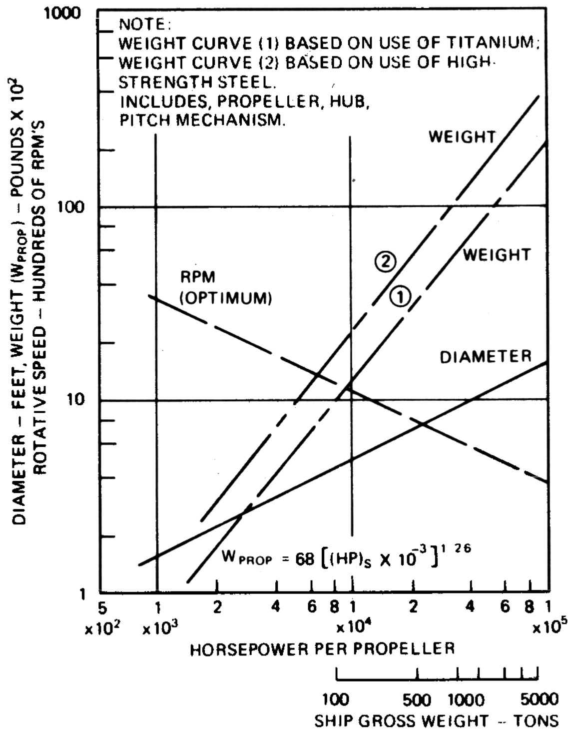


FIGURE 6-13

AIR PROPULSION

Air propulsion has been widely used for hovercraft and other surface effect craft, including the highly successful British SRN series of hovercraft having maximum speeds of up to 75 knots. Airjet propulsion systems using centrifugal and axial flow fans have been successfully used on small hovercraft, such as the CC-4 and SEDAM N102; such systems tend to be very quiet in operation. Fanjets have been successfully used for propulsion of the high-speed hydrofoil testcraft, Fresh 1. While water propulsion has, to date, been favored for surface effect ships, the designer should not overlook the possible advantages of air propulsion, either as a main propulsion device or as a supplementary device for operation through hump speed.

Types of Propulsors. A wide range of air propulsors, including the following, are suitable for high speed SES propulsion:

1. Free air propellers
2. Shrouded air propellers
3. Turbojets and turbofans
4. Q-fans
5. Cruise fans

6. Mistjets
7. Centrifugal and axial flow fans
8. Cushion air propulsion

While each of these propulsors has different physical and performance characteristics, all fall into three basic categories. Types 1 and 2 are propeller systems; types 3 through 6 are turbojet and close-coupled turbojet-fan systems; and types 7 and 8 are mechanically connected gas-turbine-duct-fan systems.

Performance Characteristics. Air propulsors offer several advantages over water propulsors, most specifically, an absence of cavitation problems, coupled with good off-design performance. Only the mistjet, which is a two-phase propulsor, does not share these advantages. On the basis of equal efficiency, air propulsors will be much larger than water propulsors because of the low density of air (about 1/1000 that of water). In many cases craft size will severely limit propulsor size and, thereby attainable propulsor efficiency. Air propulsor performance will also be limited by acceptable levels of radiated, airborne noise. Sketches of typical air propulsors are shown in Figure 6-14.

a. **Free and Shrouded Air Propellers.** Air propellers are characterized by very high potential efficiency (80 percent or more) and by large size. High efficiency correlates with large diameter and very low rotative speed. Propeller diameter is limited by craft size and by allowable blade tip speed and blade strength and weight. For typical SES applications, allowable propeller diameter will limit attainable efficiencies to about 60 percent. For a given propeller diameter, shrouded propellers may have somewhat higher efficiency than free propellers, but the difference will be small.

Four 50-foot diameter free propellers would be required to propel a typical 2000-ton, 80-knot SES. Mounting four propellers of this diameter on a 2000-ton SES would be difficult unless a tandem arrangement were used, but serious performance degradation of the after propeller may occur with this arrangement.

b. **Turbojets and Turbofans.** These propulsor systems have been highly developed by the aircraft industry. They are characterized by very high thrust-weight ratios (up to six), by excellent reliability if adequate demisting is provided, and by small size and weight. Their primary disadvantages are high radiated noise levels (up to 120 db) and poor fuel economy at typical SES operating speeds. At speeds of 100 knots or less, turbofan performance is clearly superior to turbojet performance; thus, of the two, turbofans would be preferable for the SES.

At speeds of 60 to 80 knots, the best turbofan-thrust specific fuel consumption (0.35 to 0.36 for the JT9D engine) is approximately twice that of a typical propeller-turbine or waterjet-turbine system. As an example, a good propeller-turbine system, with a 55 percent net propulsive coefficient and a fuel consumption rate of 0.5 pound/horsepower-hour (which is probably conservative) has thrust specific fuel consumptions of 0.22 and 0.14 pound/pound-hour at 80 knots and 50 knots, respectively. Fuel consumption improves only slightly with decreasing speed. Turbofans are well suited to craft having short range requirements, in which cases the increased fuel weight will be offset by the low propulsor weight.

c. **Q-Fans.** Q-fan systems rely primarily on the fan for thrust and can be categorized somewhere between turbofans and shrouded propellers. They are probably best described as very high bypass-ratio turbofans with elongated shrouds. The primary advantages of the Q-fan are low radiated noise levels and low fuel consumption (when compared with turbofans) and compactness, low weight, and reliability (when compared with air propellers). Q-fans are currently in the early stages of development. Efficiencies of up to 60 percent are obtainable –

TYPICAL AIR PROPULSION SYSTEMS

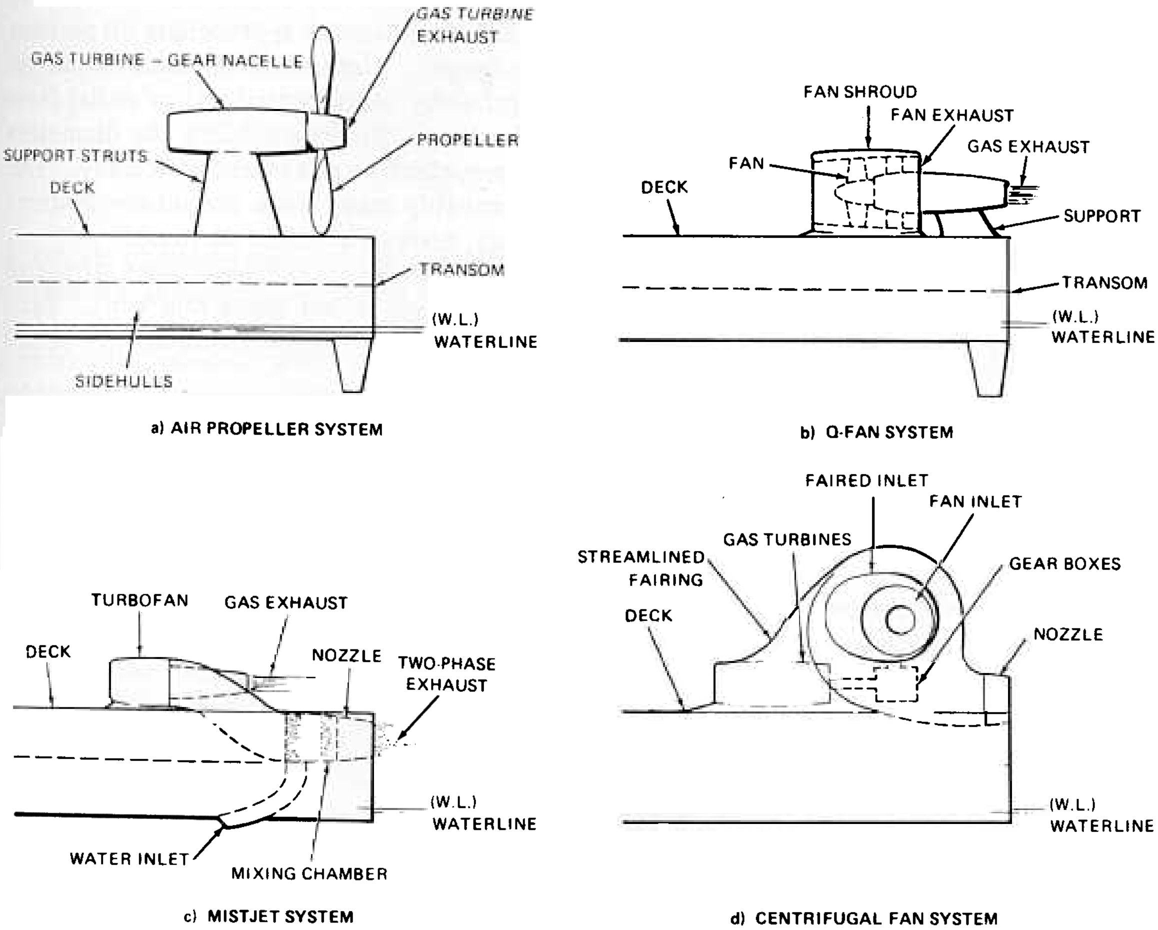


FIGURE 6-14

but only at large advance coefficients and at diameters (or thrust loadings) comparable to those of shrouded propellers of equal efficiency. The compactness of Q-fans may make them attractive as potential SES propulsors.

d. **Cruise Fans.** The cruise fan system consists of a close-coupled turbojet (gas generator), a turbine and a shrouded axial flow fan that function to convert the high-velocity turbojet exhaust to a high-mass, low-velocity jet. The thrust specific fuel consumption of the CJ 610-1 cruise fan (0.37) is comparable to that of the fanjet, while the size and weight are greater than those of the fanjet. Cruise fans must also be considered as being in the early stages of development.

e. **Mistjets.** These water-augmented airjets are comparable in efficiency to waterjets but are more compact and require no transmissions, gears, or cavitation-prone pumps. They do, however, require a water inlet, which is subject to the same cavitation problems as a waterjet inlet. Maximum efficiencies of 46 percent can be achieved if fully mixed flows can be maintained at large mass flow ratios (water mass flow/air mass flow). Mistjets are also in the early stages of development. In theory, very high thrust loadings can be used; thus, very compact and light units can be designed. Variable water mass flow (and perhaps variable water inlet geometry) may be required to obtain acceptable off-design performance.

f. **Centrifugal and Axial Flow Fans.** With these fan systems, very large fan diameters and high mass flows are required to achieve high efficiencies. Efficiencies approaching 60 percent can be obtained only with low jet velocities (and fan heads). Limitations imposed on fan-tip rotative speed to avoid excessive radiated noise will probably make centrifugal or radial flow fans preferable to axial flow fans for most SES applications. For larger SES's the diameters of such fans may be equal to the diameters of air propellers of equivalent efficiency. The large volumes required by the fans and ducting will probably make these propulsion systems unattractive for all but the smallest SES applications (say, craft of less than 50 tons).

g. **Cushion Air Propulsion.** The use of cushion air for primary propulsion is most attractive at high speeds (over 100 knots) at which the SES may begin to act like a ram wing. Such systems do not appear attractive for large SES's because of the volume requirements of the fan systems.

Chapter VII
STABILITY

A vehicle possesses static stability if, when it is disturbed from an equilibrium state, forces and moments are generated tending to oppose the disturbance. For the ship to float, it must have both heave equilibrium and stability. Heave equilibrium is achieved when the water displaced by the sidehulls, cushion, seals, and appendages, is equal in weight to the weight of the ship (Archimedes Principle). The ship is stable in static heave if its displacement increases as it is further immersed in the water. Because the cushion is not a rigid part of the SES, it is possible, but not very likely, that heave instability can occur. For the ship to float upright, it must also have roll and pitch stability, that is, the ability to withstand roll or pitch moments without capsizing. The other motions (sway, yaw, and surge) are unrestored and therefore cause no static stability problems.

Dynamic stability refers to the ability of the ship to be steered and maneuvered. Analysis of these characteristics requires consideration of the so-called transverse motions (sway, roll, and yaw). Dynamic stability is usually achieved by the use of control surfaces. These surfaces are usually wing-like protuberances, which are called rudders if they are steerable or called skegs or fins if they are rigidly attached.

For the SES the most important stability considerations involve the three rotary motions: roll, pitch and yaw. Achieving stability in these modes requires certain restraints on configurations of the cushion, seals, sidehulls, and other lift-system components.

A related problem is the seaway-induced motions of the ship. The equations involved are the same as those required for analysis of dynamic stability. In this case, however, the motions are forced, arising from the interaction of the wavy sea surface and the ship. At high speed these motions may, in fact, be a severe operational limitation.

STATIC STABILITY

The static stability characteristics of the SES are different when it is under way than when it is at rest. When the ship is under way, especially at high speed, the hydrodynamic forces acting on the sidehulls, seals, and appendages dominate the hydrostatic forces. Making an accurate prediction of SES stability in heave, roll, and pitch with forward speed is difficult and for the present purposes of developing a first approximation, stability with forward speed may be estimated as being that without forward speed in calm water. Stability at speed will be discussed, but numerical predictions will be confined to the zero speed, calm water case.

Heave Stability. When the SES is off the cushion, heave equilibrium is the same for the SES as for conventional displacement ships, i.e., the volume of displaced water is equal to the submerged volume of the ship. When the SES is on the cushion, the displaced volume is equal to the volume of the sidehulls, seals, and the cushion below the free surface. The stability of this condition can be characterized in terms of tons per inch immersion. As long as an increase in the ship's weight leads to a proportional increase in draft, the heave equilibrium is stable. The expression for tons per inch immersion has the following form:

$$TPI = \rho_w g A_{wp}/26880 = A_{wp}/420$$

where

TPI = tons per inch immersion – tons/inch
ρ_w = mass density of water (1.99) – pounds-second^2/feet^4
g = gravitational acceleration (32.2) – feet/second^2
A_{wp} = waterplane area corresponding to equilibrium condition – feet^2

When the SES is off the cushion, the choice of waterplane area is determined by the location of the waterline relative to the top of the plenum chamber. The choice is thus between the waterplane area of both sidehulls and the waterplane area of the full beam of the ship, plus the seals.

When the SES is on the cushion, the waterplane area may be taken as the waterplane area of both sidehulls. This approach neglects consideration of the immersion of the seals and the rate of change of cushion pressure with draft, but it provides a satisfactory approximation.

Figure 7-1 illustrates the location of the waterplane when the SES is on the cushion, as well as the terms defined in the subsequent roll and pitch discussions.

STATIC STABILITY NOMENCLATURE

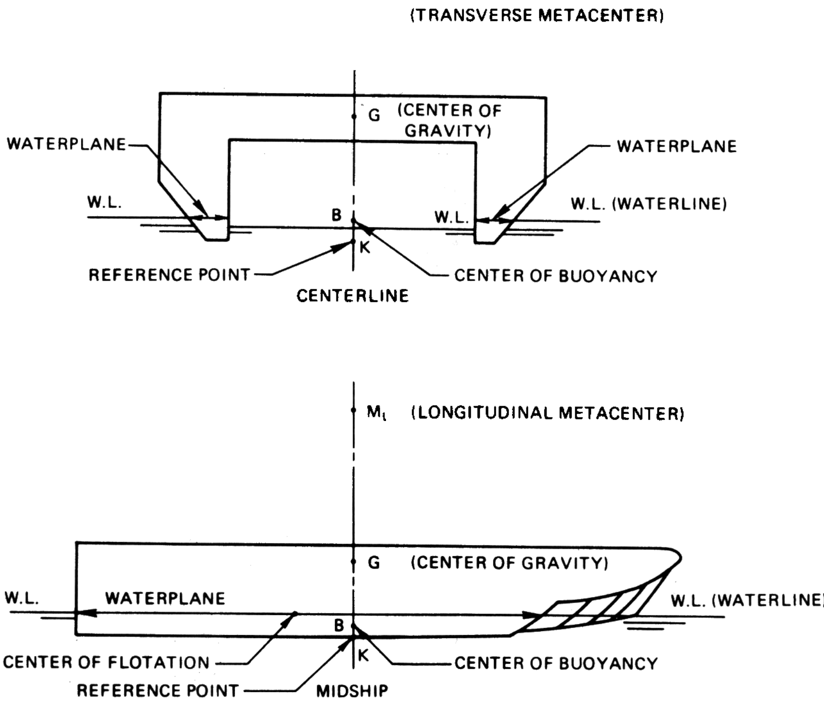


FIGURE 7-

Roll Stability. The most important stability mode of the SES on cushion is roll stability. The character of this stability is different at speed than at zero speed, and these two conditions must be treated separately. At speed, the major problems involve the heel angle resulting from a turn or from wave action or gusts of wind. The principal factors affecting roll stability at zero speed are sidehull geometry and vertical center of gravity.

The major factor determining the sidehull contribution to roll stability at zero speed is the transverse moment of inertia of the two sidehull waterplane areas taken about the vehicle centerline. This inertia increases approximately linearly with the area of an individual sidehull and with the square of the distance separating the sidehulls. Another characteristic of sidehull geometry that is important to the roll stability at zero is the slope of the sidehull on the inboard (cushion) side, which affects the stabilizing moment created by the cushion itself. Generally speaking, if the sidehull slope is such that the beam of the cushion increases with height above the baseline, the cushion contributes positively to static roll stability.

The static roll stability of the SES, either on or off the cushion, can be characterized in terms of the transverse metacentric height. The static roll stability off the cushion is most easily calculated in the manner used for conventional displacement ships. For the SES on the cushion, the following expressions apply

$$\overline{GM}_T = \overline{KB} + \overline{BM}_T - \overline{KG} \quad \text{(see Figure 7-1)}$$

$$\overline{BM}_T = [\rho_w g (I_{Tsh}) + M_{seals}]/W_G = [64(I_{Tsh} + M_{seals}]/W_G$$

where

$\overline{GM}_T$ = transverse metacentric height – feet

$\overline{BM}_T$ = transverse metacentric radius – feet

$\overline{KB}$ = height of center of buoyancy (centroid of displaced volume) above reference line – feet

$\overline{KG}$ = height of center of gravity above reference line – feet

W_G = gross weight of ship – pounds

I_{Tsh} = transverse moment of inertia of the two sidehull waterplane areas, taken about vehicle centerline – feet^4

M_{seals} = roll moment created by seals, – foot-pounds (This term is usually small for flexible seals.)

The effect of vertical center of gravity on static roll stability is obvious, since the value of $\overline{GM}_T$ (see Figure 7-2) decreases linearly with increases in the height of the vertical center of gravity, causing a decrease in righting moment.

The effect of length-beam ratio on static roll stability can be seen from Figure 7-3. With the buoyancy of the sidehulls held constant, the height of the transverse metacenter above the reference line, $\overline{KM}_T = (\overline{KB} + \overline{BM}_T)$, decreases as length-beam ratio, L_C/B_C increases. The height of center of gravity above the reference line, $\overline{KG}$, of a 2000-ton SES with 18-foot high sidehulls and a main hull depth of 18 feet will be approximately 27 feet (sidehull height plus 1/2 main hull depth). With a $L_C/B_C = 2$, the $\overline{GM}_T$ is approximately 47-27 = 20 feet; however, with a L_C/B_C = 4, the $\overline{GM}_T$ is approximately 20-27 = -7 feet or the ship is unstable.

TRANSVERSE AND LONGITUDINAL STABILITY AT ZERO SPEED

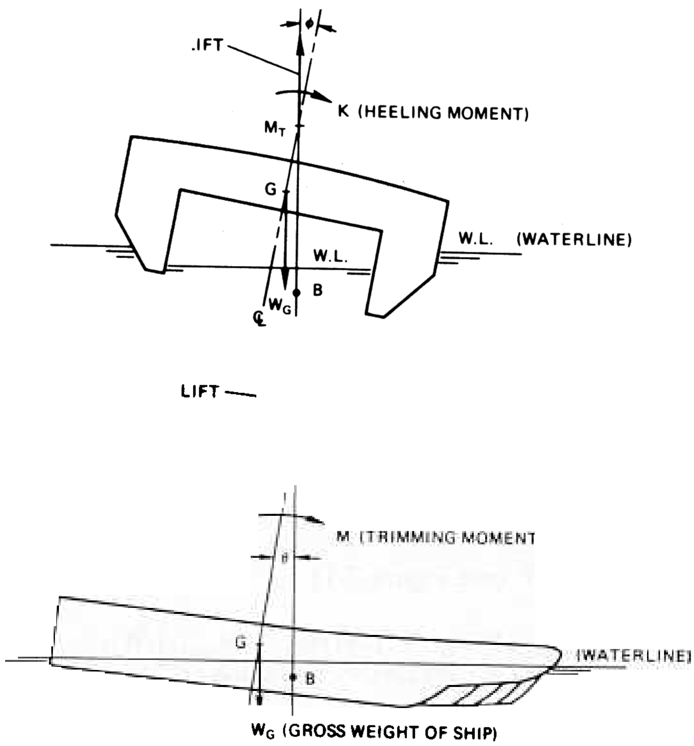


FIGURE 7-2

An initially unstable SES may, however, become stable when it is heeled to an angle that lifts one sidehull clear of the water. The resultant leakage of air will decrease cushion pressure and thus increase the immersion of the other sidehull. The buoyancy of the immersed sidehull will create a restoring moment tending to right the ship. Stability at large angles of inclination can be calculated by standard naval architectural procedures, but for an initial estimation, Figure 7-3 along with an estimate of the height of the center of gravity may be used.

The design criterion most pertinent to roll stability is that a positive righting moment in roll exists up to an angle of ±6 degrees over the entire speed range. This is to ensure that, either at speed or with zero speed, the ship has sufficient righting moment to be unaffected by the amount of roll caused by off-center loads (for instance, personnel movement or fuel usage) and wind gusts. An angle of less than about 0.5 degree would be typical.

Pitch Stability. As applies to roll stability, pitch stability is different in character when the craft is at speed than when it is at zero speed, and these two conditions must again be treated separately. At speed, the hydrodynamic forces acting on the sidehulls, seals, and planing surface (if any) may significantly affect pitch stability. The principal factors involved under speed conditions are sidehull geometry, seal characteristics, vehicle attitude, and control surfaces.

When the craft moves on a straight course, lift force is generated by the sidehulls. Thickening the sidehull at the stern will increase the lift at the stern, resulting in a shift of the center of pressure aft; this affects the restoring moment, counteracting the trim moment.

EFFECT OF LENGTH-BEAM RATIO ON ROLL STABILITY

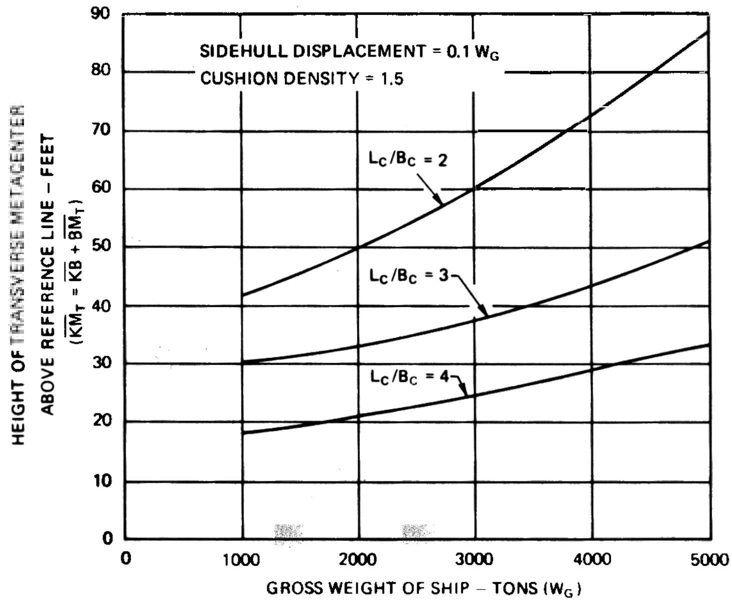


FIGURE 7-3

Pitch stability is significantly affected by the hydrodynamic, as well as the hydrostatic, loadings on the seals. Rigid seals can contribute large restoring forces with pitch, due to hydrodynamic planing effects. In general, flexible seals do not have correspondingly large hydrodynamic effects, but deformation of the seal itself can shift the center of pressure of the cushion and thus contribute large pitch restoring moments.

The attitude of the ship and of the rudders or fixed vertical fins makes only a small contribution to pitch stability. Ventral (canted) fins, however, and especially horizontal foil-like control areas at either the bow or the stern, will have significant effects. Additionally, during acceleration, large bow-up moments can be imposed by submerged propeller configurations. Pitch stability must also be sufficient to prevent this trim condition from becoming excessive.

The principal factors affecting pitch stability at zero speed are sidehull geometry, seal characteristics, and vertical center of gravity. The major factor determining the sidehull contribution to pitch stability is the longitudinal moment of inertia of the sidehull waterplane (taken for both sidehulls about the longitudinal center of flotation). This inertia increases approximately linearly with the sidehull area and with the cube of the sidehull length. With pitch stability as with roll stability, the slope of the inner surface of the sidehull is a contributing factor, but in this case the effect is generally small.

Pitch stability at zero speed is measured by the longitudinal metacentric height, $\overline{GM}_L$ (see Figure 7-2). The static pitch angle, θ, achieved as a result of pitch moment, M, relative to ship gross weight, W_G, is approximated by $\theta = M/(W_G \times \overline{GM}_L)$ for small θ. For stability, GM_L must be larger than zero.

For pitch as for roll, the off-cushion metacentric height is calculated by standard naval engineering procedures. For the SES on-cushion, static pitch stability is related to longitudinal metacentric height, as follows:

$$\overline{GM}_{\mathrm{L}} = \overline{KB} + \overline{BM}_{\mathrm{L}} \qquad \overline{KG} \text{ (see Figure 7-}$$

$$= \frac{\rho_{\mathrm{w}} g}{W_{\mathrm{G}}} \quad [I_{\mathrm{Lsh}} + X_{\mathrm{b}}^2 A_{\mathrm{b}} + X_{\mathrm{s}}^2 A$$

$$\overline{BM}_{\mathrm{L}} = \frac{64}{W_{\mathrm{G}}} \quad [I_{\mathrm{Lsh}} + X_{\mathrm{b}}^2 A_{\mathrm{b}} + X_{\mathrm{s}}^2 A$$

where

$\overline{GM}_{\mathrm{L}}$ = longitudinal metacentric height – feet

$\overline{BM}_{\mathrm{L}}$ = longitudinal metacentric radius – feet

I_{Lsh} = longitudinal moment of inertia of the two sidehull waterplane areas taken about the center of flotation for the ship (centroid of the waterplane area) – feet^4

$X_{\mathrm{b}}, X_{\mathrm{s}}$ = longitudinal distances of the bow and stern seals, respectively (relative to the center of flotation for the ship) – feet

$A_{\mathrm{b}}, A_{\mathrm{s}}$ = bow and stern seal waterplane areas, respectively, at the static pitch angle – feet^2

Longitudinal metacentric height, $\overline{GM}_{\mathrm{L}}$, decreases linearly with increases in the height of the center of gravity. Since the $\overline{GM}_{\mathrm{L}}$ is usually quite large, the location of the vertical center of gravity, within reasonably large bounds, does not usually have a great effect on pitch stability.

The most specific design criterion pertinent to pitch stability is that a positive righting moment in pitch exists up to an angle of ±3 degrees over the entire speed range. The righting moment of the craft at speed should also be sufficiently large enough that expected changes in longitudinal load distribution do not lead to large pitch angles. This is particularly important in preserving yaw stability. A typical maximum static pitch angle would be about 0.2 degree.

Additional Roll and Pitch Stability Considerations. Generally, the least stable roll and pitch conditions occur when the SES is on the cushion. Off-cushion conditions will be similar to those experienced with an ordinary ship-like form, in this case a relatively shallow-draft barge, which will exhibit large roll and pitch stability.

Since SES damage or component failure will generally lead to loss of the cushion, the terminal off-cushion state is of no serious concern. Off cushion, the SES should be designed to have sufficient stability in rolling and reserve buoyancy to withstand a 100-knot beam wind, combined with rolling, or to withstand a shell opening of length equal to 15 percent of overall length of hard structure at any point fore and aft along the sidehull, as well as a 30-knot beam wind combined with rolling.

A particular difficulty in pitch stability can occur in the transition period between on and off-cushion modes as a result of component failure, particularly cushion fan failure. If cushion pressure is lost at high speed, an emergency "plow-in" condition can result if the bow drops faster than the stern. This possibility can be reduced if the sidehulls are so designed that vertical hydrodynamic planing forces will be developed on the sidehull bow in such instances, thus preventing the bow from dropping when cushion pressure is lost.

SES's are high-speed craft and, as a result, the hydrodynamic forces acting on the sidehulls, seals, and appendages dominate the hydrostatic forces. These hydrodynamic aspects are the principal concerns with respect to roll stability. A particularly important design consideration is the roll angle achieved during a turn. It is desirable for the craft to heel inboard during a turn, thus minimizing the apparent transverse acceleration (a coordinated turn). For typical SES designs, it is not possible to achieve perfect coordination; however, it is possible to achieve nearly flat turns, given proper design of the craft. The principal factors affecting roll stability at speed are sidehull geometry, seal characteristics, control surfaces, vehicle attitude, and vertical center of gravity.

When a craft is in a turn, the centrifugal force generated acts in a direction away from the center of the turn. In a steady turn this must be counteracted by hydrodynamic forces on the ship. These forces are created by yawing the craft into the turn. To an observer aboard the craft, it appears as if the craft is sideslipping; the flow under the craft is shown schematically in Figure 7-4. Wave buildup on the outboard sides of the sidehulls causes increased pressure on these structures; separation occurring on the inboard edges causes small pressure changes here. Figure 7-4 shows the relative sizes and directions of the sidehull forces. The principal force arises from the outboard side of the sidehull. If this side is sloped, as shown, allowing the resultant force to pass above the center of gravity, a restoring moment results, tending to heel the craft into the turn. If the outboard side of the sidehull is sloped to a relatively low deadrise angle and, in particular, with the perpendicular to the sloped part of the sidehull at its center lying above the center of gravity, the sidehull will probably create favorable roll moments. Once the SES geometry is known, a quick, approximate check for stability in a turn can be made using the procedure indicated in Figure 7-4.

SIDE SLIPPING FLOW AND FORCES – SCHEMATIC

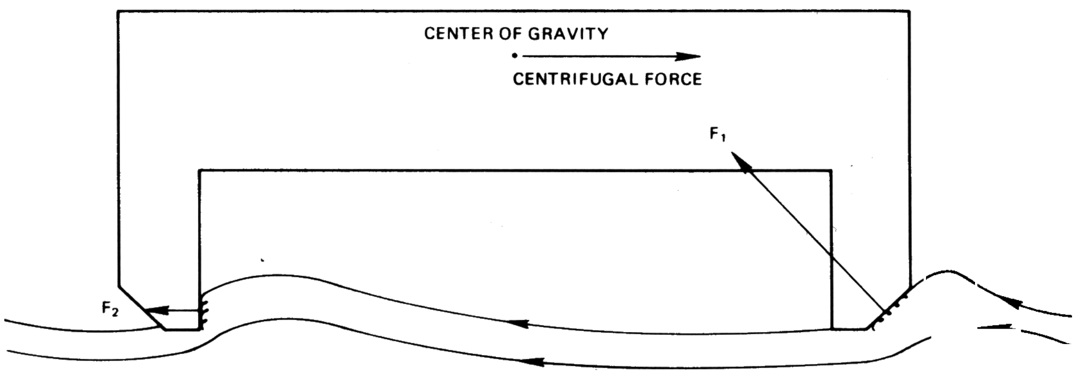


FIGURE 7-4

Flexible seals generally do not contribute much to roll stability, but rigid seals can. The principal effect of rigid seals is the increased hydrodynamic lift of the submerged edge of the seal.

Most control surfaces (rudder, skegs, etc.) also do not contribute to roll stability. However, those with dihedral (non-vertical) surfaces can contribute, that is, outward-sloping surfaces decrease stability and inward-sloping surfaces increase stability. Since forces are created on these surfaces at forward speed, this effect becomes important at high speed.

For most SES's small changes in heave or pitch do not substantially change roll stability.

Increasing the height of the center of gravity will decrease the restoring moment of the hydrodynamic force, thus decreasing the tendency of the ship to heel into the turn. The sidehulls and the control surfaces should be so designed that the roll angle does not affect the performance of the ship, as for example the venting of the air cushion.

DYNAMIC STABILITY

Yaw (Directional) Stability. Unlike stability in the pitch and roll modes, yaw stability is a consideration only when the ship is under way. Yaw stability (usually called directional stability) is particularly important in steering and maneuvering the vehicle. A ship is said to be directionally stable if random perturbations of its yaw motion caused by the seaway or the wind tend to die down without manual intervention. Generally, directional stability is more narrowly defined as the ability to resist yaw rate perturbations, since maintenance of a constant heading requires a rudder control system and a compass. Thus, directional stability usually refers to the straight-line case in which the final path of the ship, after release from a disturbance due to seaway or wind, retains the straight-line attribute of the initial state of equilibrium – but not the original path direction. In terms of hydrodynamic forces and moments, a directionally stable ship has a characteristic yaw moment which serves, without manual intervention, e.g., the use of the rudder, to oppose any increase in sideslip angle.

A ship with directional stability does not require constant steering to maintain a course and is thus relatively undemanding of the helmsman. A ship that is directionally unstable requires constant steering, thus placing an excessive burden on the pilot of a high performance vehicle, such as the SES. On the other hand, if the ship has too much directional stability, it will be hard to make it turn. Thus high directional stability implies poor maneuverability, or lack of ability to steer around obstacles. As a result, the design criterion usually imposed is that the ship be directionally stable – but only slightly so. Such a vehicle affords a good balance between steerability and maneuverability.

Directional stability can be obtained through both the geometry of the ship and the use of automatic or adaptive control systems. A ship is said to have controls-fixed directional stability if it is stable with all rudders, fins, and other control surfaces locked in position. A ship that is not directionally stable with its control surfaces fixed can be made stable by using a yaw-sensing mechanism (usually a gyro) and a feedback control system to steer the rudder. Predictive control techniques can also be used in this case.

The principal geometric factors influencing the desired controls-fixed directional stability are sidehull geometry, control surfaces and appendages, vehicle attitude, and longitudinal center of gravity. The sidehulls of the SES are very long and slender and thus behave similarly to low-aspect-ratio wings. The side force created on the sidehulls by a yaw angle (measured with respect to the ship's path) acts near the bow and causes an unstable yaw moment. This yaw moment increases in direct proportion to the draft at the bow and to the square of the forward speed. In design, this destabilizing moment can be reduced by cutting the sidehulls back from the bow and using a flexible wraparound forward seal. This partial-length type of sidehull design, or careful appendage design, can also reduce the yaw-to-roll coupling characteristics of the ship, which are the primary determinants of directional instability.

To counteract the destabilizing moment due to the sidehulls, it is necessary that rudders, fins, or skegs be installed far aft on the sidehulls. These devices also develop a large side force due to yaw; because of their location, however, the force developed creates a stabilizing moment. In other words, the situation is analogous to adding feathers at the tail of an arrow. These forces are also approximately proportional to the area of the fins and the square of the forward speed. Thus, for a given sidehull immersion at the bow, one rudder-fin-skeg system can provide directional stability at all speeds.

Because immersion of the bow of the sidehull is particularly important in determining the sidehull destabilizing moment, the attitude of the ship is extremely important, particularly the pitch attitude relative to the water surface. If, at certain speeds, the area of immersion of the sidehull bow increases and the area of the sidehull stern immersion decreases, correspondingly, the destabilizing sidehull moment increases and the stabilizing control surface moments decrease. This situation can lead to severe directional instability. The opposite situation, bow-up pitching, can lead to excessive directional stability and, thus, poor capability for maneuvering. As a result, the whole range of vehicle attitudes must be considered in selecting the control surface system.

The farther forward the center of gravity of the SES is, the smaller the destabilizing moment caused by the sidehull and the larger the stabilizing moment attributable to the aft fins and rudders. Because the cushion carries the major portion of the weight, however, not much freedom is possible in choice of location of the center of gravity.

The basic design criterion for yaw stability is that the ship have small, positive controls-fixed directional stability for the range of normal pitch attitudes. The desirable degree of directional stability is given by the requirements for maneuvering.

Yaw, or directional, stability may be lost in going to the off-cushion mode, since the resultant increase in bow immersion area can increase faster than comparable increases in fin or rudder area This is particularly true if the fins or rudders are efficient, high-aspect-ratio foils. Since the SES normally functions off cushion only at low speeds, this loss may be of no great consequence. However, a high-speed transition from on- to off-cushion modes, such as caused by component failure, can result in disastrously large yaw rates if the bow drops into the water first. As mentioned above, choice of bow sidehull design can reduce this possibility.

Maneuverability. Maneuverability generally means the ability of the craft to change course rapidly in response to a need to avoid objects or to perform some special mission task. The ability to change course can be measured in a variety of ways, each with a special significance. The basic parameter governing the maneuver is the rudder angle. For a first approximation, the effect of speed on the actual path is not great. Given any particular ship speed, when the rudder is put in motion the ship begins to turn. The advance is defined as the maximum distance in the direction of the original motion achieved by the ship, and the tactical diameter is the distance between the original straight path of the ship and the path when the ship is 180 degrees to the original motion (see Figure 7-5). The turning radius is the radius of the circular path that might be achieved at a specified speed and rudder angle when all of the transients have died down. In other words, advance and tactical diameter are measures of the ability of the ship to react to a new situation, and turning radius is the measure of the steady-turning capability. In general, the larger the movable rudder and the smaller the directional stability, the smaller (and, accordingly, the better) these measures are. Thus it is that rudder size is selected on the basis of maneuverability.

The tactical diameter, the advance, and the minimum turning radius of the SES must be such as to meet the mission requirements of the craft. Maneuverability, however, is governed by two factors: crew habitability and craft capability. Generally, the SES must be able to operate under a side acceleration of 0.5g for purposes of emergency operation, of 0.3g for 30-minute operations, and of 0.1g for continuous operations.

The maximum rudder angle should be 35 degrees and the rudder rate should be such that this angle is achieved in less than 5 seconds. Since small turning radii lead to large lateral accelerations at high speed, it may be necessary to limit the maximum rudder angle to a much smaller range at the design cruise speed. However, the need for a high rudder rate remains.

TURNING PATH OF A SHIP

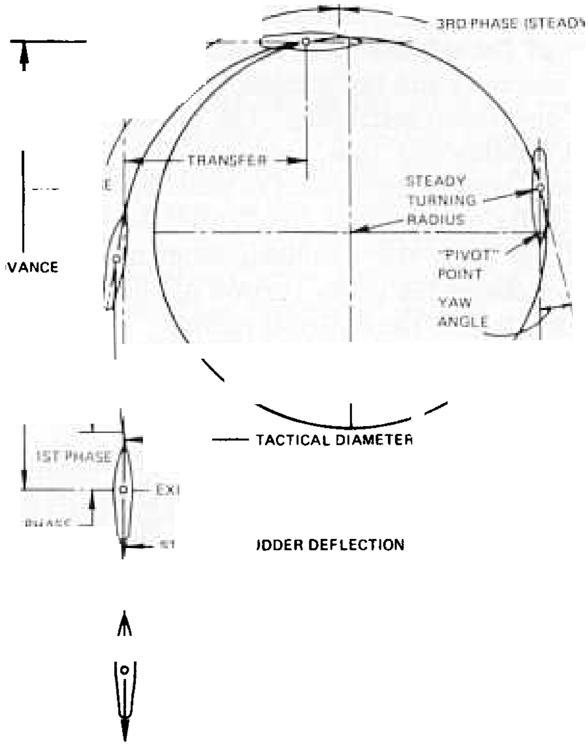


FIGURE 7-5

To prepare for a possible malfunctioning of the steering system, the ship should be so designed that it can be steered by differential thrust between the propulsors in each sidehull. It is anticipated that this type of emergency steering would be required only at low speed.

Stopping is one of the most important ship maneuvers. For conventional ships, two factors are particularly important in stopping. The first is the ship's own resistance, which is dependent on initial speed and may dissipate a substaintial amount of the kinetic energy in the ship at the time of ordering the maneuver. The other factor is the backing or reverse thrust developed by the ship's propulsion system. The inherent stopping ability of a particular ship is thus a combination of its resistance and its backing or reverse-thrust power. At high initial speed, the ship's own resistance is the primary determinant of head reach (stopping distance); at low initial speed, the backing thrust is much more important than the ship's resistance. Because of these considerations and the transient character of the stopping maneuver, it is generally necessary to base estimates of head reach and stopping time on the ship's dynamic properties, installed power, and propulsor performance. The SES has an additional factor which may be even more dominant. As the SES slows down, it may be gradually changed to the off-cushion mode. As a result, the resistance is greatly increased, thus greatly accelerating the energy dissipation.

In general, the head reach and the stopping time of a ship are governed by three non-dimensional parameters (see Reference 10):

$$\textit{dynamic potential} = \frac{2R_oS}{(m - X_{\dot{u}})\,V_o^{\,2}}$$

$$\textit{dynamic impulse} = \frac{R_ot}{(m - X_{\dot{u}})\,V_o}$$

$$\textit{ahead resistance to astern thrust ratio} = \frac{R_o}{T_1}$$

where

V_o = initial steady ahead ship's speed at start of stopping maneuver – feet/second

R_o = total resistance at V_o – pounds

m = mass of ship – pounds-second^2/foot

$X_{\dot{u}}$ = longitudinal added mass – pounds-second^2/foot

S = stopping distance, or head reach – feet

t = stopping time – seconds

T_1 = astern thrust – pounds.

Thus, if two ships are similar in terms of the above non-dimensional parameters, but the head reach and stopping time of one ship are unknown, they can be derived from the data available on the other ship.

In Figure 7-6, general trends are shown for stopping distance and stopping time, based on data extracted from the SESMA 2000-ton SES analysis, to which the similarity laws are assumed to apply. The craft is assumed to be on-cushion throughout the stopping maneuver. If the craft is switched into the off-cushion mode during the maneuver, the results may only be 50 to 70 percent of the values shown, depending on the rate of transition to off-cushion. In preparing Figure 7-6 it is also assumed that 20 seconds elapse between the order to stop and achievement of constant astern thrust. The magnitude of longitudinal added mass is assumed to be – 0.08m.

Some **Stability Related Design Considerations.** The principal contributors to directional stability and maneuverability are the ship's sidehulls, seals, and appendages (control surfaces). Stability and maneuverability are also significantly affected by the ship's speed and attitude.

The principal function of the sidehulls is to seal the air cushion with a minimum of air loss, but this can be accomplished only at the expense of hydrodynamic drag. Additional functions of the sidehulls are normally as follows: (1) to provide the major part of the ship's stability in roll and pitch, (2) to accommodate the main propulsion transmission system and the propulsor, fuel, steering machinery, and void (flotation) spaces, (3) to help in providing the required strength for either the static or the hydrodynamic loads imposed on the ship, (4) to carry the weight of the ship at rest, either in drydock or if grounded, (5) to provide buoyancy for the ship in the off-cushion mode and to provide lift supplementary to that of the air cushion in the on-cushion mode, and (6) to provide directional stability, as well as the required maneuverability characteristics.

Many varying sidehull forms are likely to evolve in response to the need for specific combinations of the foregoing requirements. Figure 7-7 shows a typical shape for a single sidehull. As can

GENERAL TRENDS IN STOPPING DISTANCE AND TIME (SES ON-CUSHION THROUGHOUT STOPPING MANEUVER)

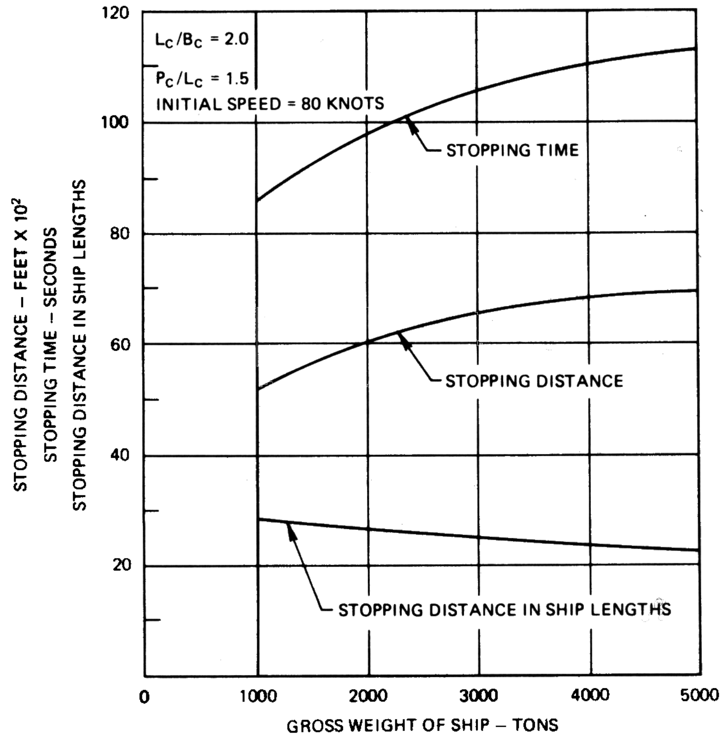


FIGURE 7-6

be seen, the upper waterlines are a modified parabolic shape, thickening aft to accommodate a waterjet propulsion system. In the design of sidehulls, consideration must be given to the interplay among stability, drag, directional stability, seaworthiness, maneuverability, and interior volume. Thus, tradeoff studies should be made to find the optimum sidehull geometry.

For a fixed beam, sidehull depth, and length, a ship having a relatively large sidehull volume experiences greater static stability in roll and pitch but is subject to a larger drag penalty. A partial-length sidehull contributes favorably to directional stability and may thus obviate the need

TYPICAL SIDEHULL FORM

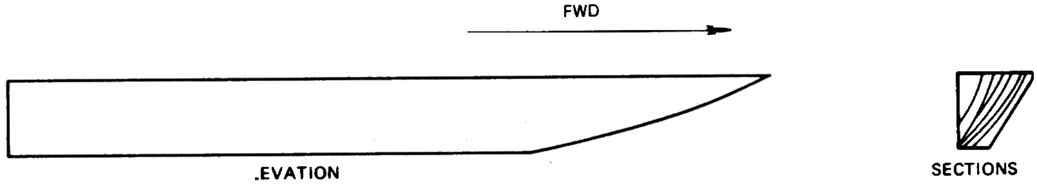


FIGURE 7-7

for large appendages (e.g., ventral fins) at the stern, thereby reducing drag as well. In the maneuvering and rolling and pitching modes, however, such a sidehull has unfavorable characteristics. Furthermore, a partial-length sidehull has less pitch stiffness than a full-length sidehull, and the pitching motion of the ship will be less damped; therefore, larger pitching motions can be expected, excluding seal effects. In this case the coupling of pitch and heave would also tend to increase the heave motions. Another possible effect of utilizing the partial-length sidehull is the increase in wave impact at the center-hull bow due to pitching motions, with the attendant possible increase in structural loadings if seal pitch restoring moment is not as great as that of the replaced sidehull.

The geometric form of the SES (for example the length-beam ratio) directly affects the choice of sidehull geometry. A wide ship will have more stability in roll than a narrow one, either in the static condition or in steady turning. In terms of yaw, the inception of a destabilizing yaw moment will occur at larger yaw angles for a wider ship than for a narrower one. The attitude of the craft significantly affects yaw stability or instability and, therefore, the maneuverability. Adding forward profile area to the sidehulls, i.e., a trim by the bow, results in less stability (or more instability). Changing the forward speed of the ship results in changes in trim, especially if the ship is equipped with dihedral foil-like fins at the stern or bow.

Depending on the specific seal type, both the stern seals and the bow seals help to provide lift and stabilizing forces in roll and pitch and therefore affect the ship's yaw stability and performance in steady turning. A soft fabric bow seal, as for example a semi-circular bow skirt or full finger skirt, has no means of providing auxiliary lift near the bow; thus, although the design is desirable in terms of ship motion considerations, it might be deficient in its roll and pitch stabilization characteristics. Other types of bow seals, which employ air bags in their upper sections, will make favorable contributions to stability under rolling and pitching conditions.

In most SES designs the sidehull draft is small to minimize the hydrodynamic drag penalty when the ship is operating on the cushion. Therefore, directional stability (or instability) is very sensitive to a change in the ship's trim. For example, as reported in Reference 35, the XR-3 testcraft is directionally stable when the trim by the bow is less than 2 degrees, but it becomes unstable when the bow trim is more than 3 degrees. The geometric form of the ship also affects yaw stability, as mentioned previously. As indicated in Reference 36, reducing the length-beam ratio from 3.5 to 2.35 (by increasing the beam by 50%) changes the roll-yaw instability of the XR-1 testcraft to a neutrally stable condition. These data indicate that the SES yaw stability and maneuverability cannot be fully assessed prior to completion of the detailed layout and model test phases.

If the ship is not equipped with rudders, which might perform a similar function, ventral fins at the stern of the sidehulls can make a directionally unstable SES stable, but the fins must be canted inward to achieve an effective dihedral and to minimize any detrimental effect on the roll stability of the ship at speed. However, recent design trends indicate that, with properly tailored partial-length sidehulls, the SES may not need to be equipped with ventral fins to achieve directional stability, although fins at about mid-length of the sidehulls might be necessary to obtain enough side force to generate adequate turning performance. Fins employed for this purpose can be retractable to reduce the drag under other operating conditions, such as straight-course cruising, and are most likely to be used on waterjet-propelled ships where movable jet nozzles obviate the necessity for rudder steering surfaces.

In Figure 7-8, the addition of fixed ventral fins at the stern is diagrammed as a function of gross weight. These data were obtained through extrapolation of XR-1A testcraft data (see Reference 36), based solely on considerations of directional stability. The fin area taken as the

CHARACTERISTICS OF FIXED FINS AT STERN FOR FULL-LENGTH SIDEHULLS

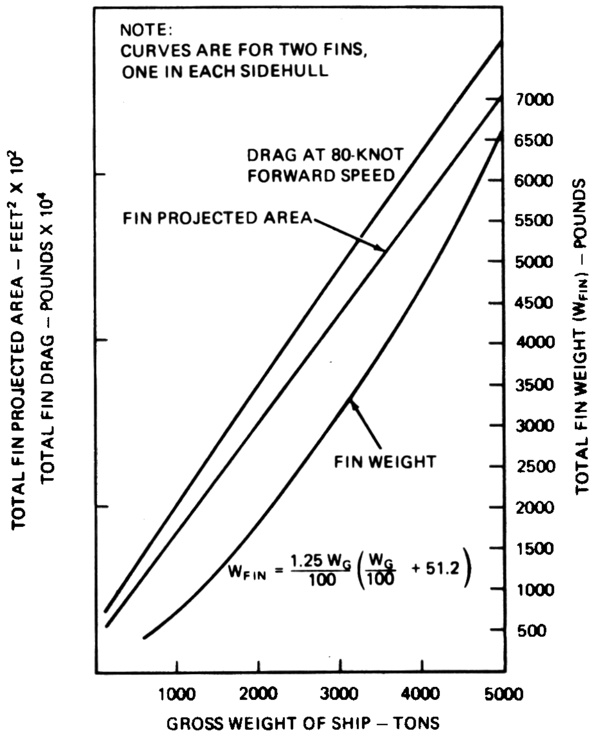


FIGURE 7-8

basis for the weight estimation is the projected area on the vertical plane. The drag due to the fixed fin consists of frictional and form drag components; a zero yaw condition is assumed at 80 knot speed, and the fins are assumed to be symmetrical foil sections with a thickness-chord ratio of 0.037. These data cannot be generally applied unless the ship design under consideration is geometrically and dynamically similar to the XR-1A including the seal and sidehull (full-length) characteristics; Figure 7-8 shows only general trends for ventral fin drag and weight.

Rudders, found principally on propeller-driven SES's, have two primary functions in controlling ship motions. First, rudders at the stern have a yaw stability effect and will stabilize a neutrally stable, rudder-free craft. This is the case with the XR-1 design. However, once the air cushion is vented, as initiated by a severe roll condition, the rudder is no longer able to produce a moment sufficient to stabilize the yaw effect on the craft. The second primary function of the rudder is to change the ship's course. Outwardly canted rudders, when activated in a turn will roll the craft into the turn. Such roll-control devices have the potential of eliminating or at least delaying the major yaw instability that follows cushion venting. Furthermore, as shown in Reference 36, outwardly canted rudders have the potential to roll the ship into a turn independently of the roll-stabilizing function of the bow and stern seals. This potential is significant in that the focus in seal design can then be on alleviating sea state effects, unconstrained by seal-roll-stabilizing requirements.

A precise estimation of the required rudder area is impractical at the advanced design stage, as the maneuverability of the SES depends greatly on its hydrodynamic characteristics. Consider, for example, a ship in a steady turn such that (1) the centrifugal force acts through the center of gravity of the ship outward of the turn, (2) the hydrodynamic forces act on the sidehulls in a direction perpendicular to the path of the turn and into it, and (3) the rudders exert various forces on the ship. In a steady-state condition these forces must be in balance. Thus, although the rudder size and, consequently, the rudder force can be arbitrarily chosen, the maneuverability is limited by (1) the maximum side force, i.e., the hydrodynamic force on the sidehulls that can be generated to sustain the side acceleration and (2) the maximum heeling angle that is allowed without venting the cushion.

As a first estimation, the inherent limitation on turn maneuverability can be deduced from the following:

$$\frac{\textit{side acceleration}}{g} = F_N{}^2 \left(\frac{L_{SH}}{R}\right)$$

$$\textit{side acceleration} = \frac{V^2}{R}$$

where

g = gravitational acceleration – feet/second^2
F_N = Froude number
L_{SH} = sidehull length – feet
R = turning radius – feet
V = ship speed – feet/second.

To establish a turning radius it is necessary only to determine the ratio of side acceleration to gravitational acceleration that can be accepted for a given ship design and stated speed. The design criteria for maneuverability in side acceleration are 0.3g to 0.5g for emergency operations, 0.1g to 0.3g for 30-minute operations, and less than 0.1g for continuous operations. In Figure 7-9 data are plotted for the required area and drag of vertical rudders, as estimated for SES's of a length-beam ratio of 2.5 on the basis of the foregoing equation, wherein the side acceleration is taken to be 0.2g, the length, L_{SH}, refers to the sidehull length at the waterline, and the speed is 80 knots. Factors not taken into account are the augmented resistance of the ship and the change of propeller performancce in turning.

Under the above assumptions a constant turning radius of 2850 feet is derived for all gross weights. In this derivation the effect of heeling in the steady turn was not considered; the rudder area was estimated on the basis of the following assumptions: (1) the hydrodynamic force, which equals the sum of the centrifugal force and the rudder force, acts at a point one-fourth the length of the sidehull behind leading edge, (2) the center of gravity of the ship is the same as the centroid of the cushion footprint, (3) the moment of the rudder force is equal to and opposite to the moment caused by the hydrodynamic force on the sidehulls, (4) the aspect ratio of each rudder, i.e., the depth-mean chord ratio is 1.5, and the thickness-chord ratio is 0.08, (5) the cavitation of the rudder is assumed to occur at a 5-degree rudder angle, and the lift at this angle determines the rudder moment about the center of gravity, (6) the rudder drag consists of solely of frictional and form drag components, and (7) the interaction between the rudders and the

GENERAL TRENDS IN RUDDER CHARACTERISTICS

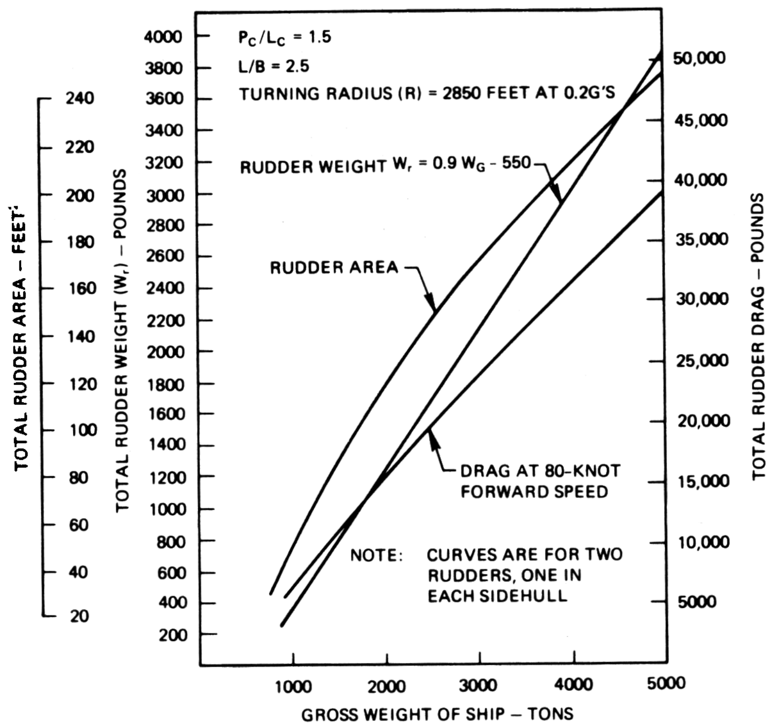


FIGURE 7-9

sidehulls or between the rudders and the propulsors is small. The latter assumption implies that the rudders are located ahead of partially submerged propellers. Again, Figure 7-9 provides only a first estimation, but is suitable for advanced design purposes.

Habitability. As do all rigid bodies, SES's have six degrees of freedom, and all six motions – surge, sway, heave, roll, pitch and yaw – are excited in a seaway. The importance of the severity of these motions is twofold: first, human performance can be degraded; second, the mechanical or hydrodynamic performance of the ship can be impaired. Usually, the requirement of habitability will predominate.

The SES travels at high speeds and thus encounters ocean waves at a high frequency. The high-frequency rate of this encounter implies that even small amplitude motions lead to large accelerations. Thus, the habitability requirement is essentially an acceleration limitation. This means that the dynamics of the ship must be decoupled from the forcing input of the waves. The decoupling can be achieved in two main ways. The first involves providing the ship with dynamic characteristics that will keep the natural period of the motions well above the encounter period. This type of inertial decoupling is very effective in eliminating motions. The second method, to be employed when inertial decoupling is not possible, involves choosing a damping sufficiently high to avoid resonance.

Because the roll stability of the SES at speed is usually small, the roll period tends to be quite large; and, as a result, the ship is well decoupled from the high-frequency excitations that usually cause habitability problems. Furthermore, SES roll motions are usually reasonably well damped, and resonance problems are not severe. Roll motions can cause some difficulty if a sidehull becomes exposed thereby resulting in significant cushion leakage. Sway motions are dynamically coupled to roll motions, but these are first-order motions, that is, they exhibit no resonance and are characterized by a time constant. In the cushion mode this time constant is quite long, and, thus, inertial decoupling also occurs.

Pitch and heave are both restored motions and thus exhibit resonances. The geometry of the SES obviates any practical means of providing for inertial decoupling of the heave motions. Thus, these motions must be controlled by providing sufficient damping. Of the six possible motions, the heave motions are the most objectionable from a habitability viewpoint. While these motions may be greatly abated by proper venting of the cushion, there is often an offsetting expenditure of fan power. Pitch motions are less related to cushion performance, and the craft is normally designed so that the pitch period is relatively high (a direct consequence of thin sidehulls). Thus, pitch motions are more subject to inertial decoupling than are heave motions.

Yaw and surge motions have relatively long time constants and are quite well decoupled from the objectional high-frequency wave-forcing function.

For the SES, the most critical motion is heave; and in considering habitability criteria for advanced design purposes, it is usually sufficient to examine this criterion independently of other habitability considerations. Root-mean-square heave accelerations of 0.1g or less in any sustained operational environment and of 0.25g or less in any short-term situation would seem to be appropriate design values.

Figure 7-10 shows the vertical acceleration levels specified by the Surface Effect Ships Program Office. This figure shows maximum levels of sinosoidal accelerations (single amplitude) as a function of encounter frequency under two opposite conditions of operation, 30-minute and 24-hour (or continuous) exposure. For random excitation, the root-mean-square spectral level (in g's per cps) for the appropriate time of exposure will be less than half the amounts shown in Figure 7-10.

LIMIT OF VALUE OF VERTICAL ACCELERATION

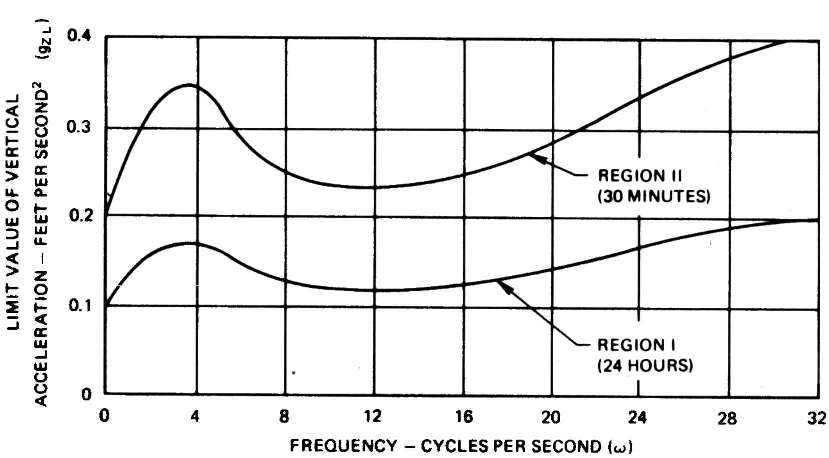


FIGURE 7-10

Air Cushion **Power**. The primary purpose of the air cushion is to support the ship, that is, to minimize the wetted surface area of the sidehulls. It thus has an effect on the attitude and motions of the SES in either a calm sea or in a seaway. When the craft is in calm water, such that the motion is steady, the mass flow rate of the air provided by the fan must equal the flow out of the cushion due to leakage. In this case,

$$Q = S_g D_C \sqrt{2 P_C / \rho_a} = 21.65\, S_g \sqrt{P_C}$$

where

S_g = cushion discharge area – feet^2
D_C = air leakage discharge coefficient (typically, 0.75)
ρ_a = mass density of air (0.0024) – pounds-second^2/feet^4
P_C = cushion pressure – pounds/feet^2
Q = mass flow rate of air – feet^3/second.

When the SES is in a seaway and oscillating in all degrees of freedom, the vertical acceleration may exceed the tolerance limit set by habitability considerations. To preclude excessive accelerations, large leakage areas must be provided for the cushion air. To estimate the required mass flow rate of cushion air, or the required leakage area, assume that the ship is supported entirely by the cushion and consider the heave motion over a sinusoidal wave train. For this case, the heave acceleration of the craft can be expressed as

magnitude of heave acceleration =

$$B_w\, \omega_e^{\,2} \left[\left(1 - \frac{\omega_e^{\,2}\, W_G}{C_1 S_C\, g}\right)^2 + \omega_e^{\,2} \left(\frac{W_G\, C_2}{C_1 S_C\, g}\right)^2 \right]^{-1/2} \qquad (7.1)$$

where

B_w = wave excitation – feet

$$= \frac{h_w}{2} \; \frac{\sin(\pi L_C/\lambda)}{\dfrac{\pi L_C}{\lambda}}$$

ω_e = frequency of encounter – radians/second
$= \omega - \omega^2\, V/g$
$C_1 = k\,(P_C + P_{atm})\, S_C/\overline{V}_C$
$C_2 = C_1 S_g D_C / (S_C \sqrt{2\, \rho_a P_C})$
g = gravitational acceleration – feet/second^2
h_w = wave double amplitude – feet
λ = wave length – feet
ω = wave frequency – radians/second
S_C = cushion area – feet^2
W_G = gross ship weight – pounds
L_C = cushion length – feet
$\overline{V}_C$ = cushion volume – feet^3

V = ship speed – feet/second

k = gas constant – 1.4 for air

P_{atm} = atmospheric pressure – pounds/feet^2.

For a given ship, where the length, pressure, and volume of the cushion are specified, the leakage area and, thus, the flow rate can be calculated from the previous equation for a specified wave condition, subject to a specified heave acceleration limitation. Under realistic operating conditions the root-mean-square value of the heave acceleration of the SES in a seaway can be obtained by employing the above equation and a wave height spectrum. In this case, the leakage area provided must be such that the root-mean-square value of the acceleration is less than half the limit value for a sinusoidal wave train. Calculations show that, to meet standard habitability requirements for sea states higher than 3 combined with ship speeds in excess of 40 knots, the leakage area required is sufficiently large to force consideration of some sort of active control system, such as controllable louvers. The effectiveness of louvers can be illustrated by considering a pressure-sensitive system, such as shown in Reference 37, which is half open at normal cushion pressure, fully open at a pressure 10 percent higher than the normal condition, and fully closed at a pressure 90 percent of the normal value. Such a louver system is eleven times as effective as one with only passive leakage control. Other active control systems, such as heave acceleration-sensitive and heave velocity-sensitive louver systems or controllable pitch lift fan systems can also be used to reduce the required leakage area. An effective means might be a combination of an active louver system and a controllable pitch lift fan.

An alternate method is to calculate cushion power on the basis of rate of air flow into the cushion, which must match the rate at which waves sweep (wave-pumping action) through the cushion volume with cushion pressure maintained. In this case, the heave accelerations of the ship and the use of control devices are not taken into account. For a single sinusoidal wave train, the wave excitation per unit width is given by Equation (7.1). The limiting condition for cushion power is the highest combination of ship speed and sea state, i.e., the situation wherein wave length is twice ship length. Therefore, cushion power is

$$(HP)_C = \frac{Q\,(P_C + \Delta P_C)}{33{,}000\,\eta_F\,\eta_{TC}\,\eta_D} = \frac{Q\,(1.05\,P_C)}{33{,}000\,\eta_{LS}} \tag{7.2}$$

where

$Q = 60\,B_C\,h_w\,V$ (see below) – feet^3/minute

B_C = cushion beam – feet

h_w = wave height – feet

V = ship speed – feet/second

ΔP_C = variation of cushion pressure, typically, $0.05 - 0.1\,P_C$ – pounds/foot^2

η_F = cushion fan efficiency

η_{TC} = lifting system transmission efficiency

η_D = duct efficiency

η_{LS} = lift system efficiency = $\eta_F \; \eta_{TC} \; \eta_D$

As an example, apply the above equation to a 2200-long ton SES operating at a cushion pressure of 300 psf under the following conditions:

$\eta_{LS} = 0.70$

$L_C/B_C = 2.0$

L_C = 170 feet

B_C = 85 feet

$\Delta P_C = 0.05\, P_C$

Q = 2,060,000 feet^3/minute for 80 knots and 3-foot waves.

$$HP)_C = \frac{(2{,}060{,}000)(1 + 0.05)(300)}{(33{,}000)(0.70)} = 28{,}200. \tag{7.3}$$

If the ship is designed to operate in 9-foot waves at 40 knots, the required cushion power will be 42,500 horsepower, i.e., 1.50 times that required for operating in 3-foot waves at 80 knots.

Compared to a precise analysis, Equation (7.3) will yield somewhat conservative values for the cushion air flow rate.

Chapter VIII
LIFT

The function of the lift system is to maintain a cushion of air beneath the SES to support it (a) at a pressure sufficient to keep the sidehulls submerged at a favorable depth to minimize drag and (b) at a flow rate adequate to provide a habitable ride over the ocean surface. A theoretical explanation of cushion air flow requirements has been previously presented. For purposes of this discussion, cushion air flow rate, Q, can be approximated by

$$Q = 60B_C\ h_w\ V\,(feet^3/min) \tag{8.1}$$

where

B_C = cushion beam-feet

h_w = average wave height-feet

V = ship speed-feet/second.

The major components of the lift system include the inlet, where the air is ingested; the fan, which adds velocity and pressure to the air; the diffuser, which converts the air velocity energy to pressure energy; and, the ducting, which transports the air to the cushion chamber or plenum. Closures to isolate individual fans from the duct system and controls to vary fan performance or system characteristics are also necessary to permit operation over the ship's entire performance envelope.

One important consideration is the variation in lift power required as the weight of the SES changes with fuel consumption. Another consideration is the increase in air flow rates required for higher sea states. The fans and ducting are necessarily large to provide adequate flow at reasonable pressure losses. Although the SES is not volume critical, large volumes often reflect large weights, which are critical.

System weight minimization can be accomplished by careful materials selection and system arrangement. The materials not only should be light-weight (e.g., fiberglass, aluminum) but also should be impervious to the effects of salt spray. In system arrangement efforts are generally made to place the fans in locations that minimize duct length. However, such other considerations as redundancy, maintenance, and power plant penalties should also be taken into account.

Desirable Characteristics. The ideal fan performance curve for the SES is depicted in Figure 8-1. The first characteristic of the ideal system is that it supplies an almost constant pressure at any flow rate. This means it is an inherently passive control system which maintains the cushion air at a constant pressure regardless of leakage or the frequency of wave encounter.

The second characteristic indicated is that the static pressure is almost equal to the total pressure. The difference between static pressure and total pressure is the dynamic pressure. If the dynamic pressure is not converted into static pressure before entering the cushion, energy is

IDEAL CUSHION FAN CHARACTERISTIC CURVE

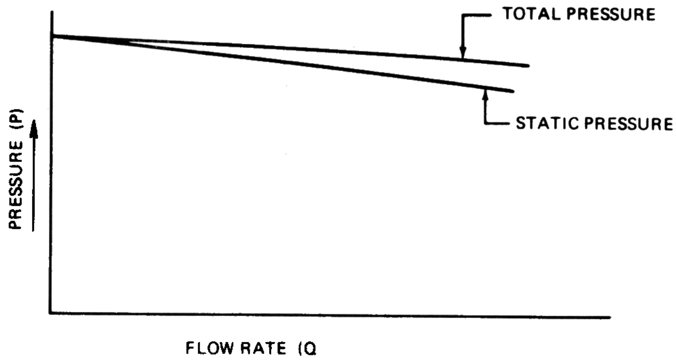


FIGURE 8-1

wasted. This conversion can be accomplished with diffusers, which are transition ducts of increasing area. However, the conversion occurs with some loss in total head such that overall system efficiency is reduced.

The third characteristic of the ideal fan is that it is operable from zero flow. Thus, no fan stall occurs when the craft attempts to lift off the water surface.

Since no existing fan design has these characteristics, attempts are made to approach this ideal by varying the physical characteristics of the fan or the fan system. These variations may be controlled either by an automatic sensing system that monitors cushion pressure or ship motions or by an operator who adjusts the system to suit sea conditions or SES gross weight.

The general effect of varying the fan blade angle (controllable pitch), the inlet vane angle, or the rotative speed is shown in Figure 8-2. Curve "B" is the characteristic curve of the fan for a

CUSHION FAN CHARACTERISTIC MAP

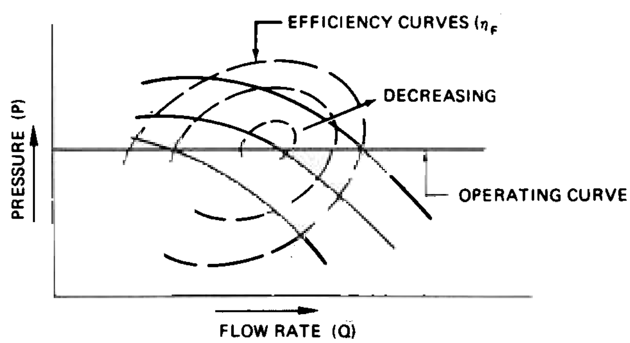


FIGURE 8-2

particular blade setting, vane setting, or speed. The pressure drops as flow demand increases. To maintain the same pressure, the speed, blade angle, or vane angle is changed to permit operation on curve "C". These curves indicate a loss in fan efficiency as the operation shifts from one curve to the next. The reduction in efficiency is greatest with variable inlet vanes and least with controllable-pitch fan blades.

Speed control via engine control is the simplest of the three control possibilities, but there is the associated disadvantage of far greater reduced drive-turbine efficiency at off-design conditions. Installing a slip coupling between the engine and the fan increases the complexity of the system but permits running the fan at variable speed, thereby minimizing overall inefficiencies.

An advantage unique to the controllable-pitch fan blade installation is that it permits the fan engines to be run at constant rpm while operating at close to peak fan efficiency for various power levels. Operation at peak efficiency means minimum fan-generated noise. Additionally, when the propulsion and lift systems are integrated with a single power supply, controllable-pitch permits independent control of each system.

The above-mentioned methods are attempts to fit the fan performance curve to the system. Still other methods involve altering the system characteristics to fit the fan, such as employing dampers to vary the resistance in the ducting; a fixed leakage area to prevent the fan from operating in the stall region and to flatten the *P-Q* curve for the pressure apparent to the plenum; and variable leakage areas, such as relief valves or louvers, to vent off excess pressure in the cushion. These alternatives do not utilize all of the energy available from the fan, their advantage is simplicity in design and control.

In fan selection, the influence of the fan on the other components of the system must be carefully considered. In some instances a fan of reduced size and weight may generate unacceptable penalties in other parts of the system. For example, fans with high dynamic heads (low static pressure relative to total pressure) are usually of reduced size and weight. Their use necessitates the installation of a diffuser or transition piece to convert the dynamic head to a static head. Inherent in such an installation is the additional weight of the diffuser and the total pressure loss due to expansion of the air through the transition piece. Evidently, a weight-power tradeoff analysis is warranted in considering fans of equivalent total pressures but different static pressures.

Noise generated by the fan is of considerable importance. Fans that operate at close to optimum efficiency will be the most quiet. Low tip speeds are also conducive to quiet operation.

Fan Types. Fans may be divided into three categories, by type: axial fans, which move the air in a direction parallel to the fan axis; centrifugal fans, which rotate the air flow in a radial direction; and mixed flow fans, which combine axial and radial flow.

a. **Axial Fans.** Axial-type fans include propeller, tube axial, and vane axial fans. The vane axial (Figure 8-3) is a propeller fan in a tube, with guide vanes employed to improve its pressure and performance characteristics.

A typical axial fan performance curve derived from test data is given in Figure 8-4. For the SES, operation of the fan would be to the right of line A-A. Alterations to the total pressure curve can be accomplished by speed control, inlet vane angle variation, or blade pitch variation. Some alteration is necessary, since its characteristics curve is very steep.

The axial fan lends itself most readily to controllable blade pitch (Figure 8-5). Controls for the variable-pitch mechanism can, for example, sense the cushion air pressure. Hydraulic or mechanical means of control are possible. Variable inlet vanes can also be used to modify fan performance. Although this means is less efficient than controllable pitch, it is simpler and adds less weight (20 percent less).

CASING DESIGN FOR FULLY STREAMLINED VANE AXIAL FANS

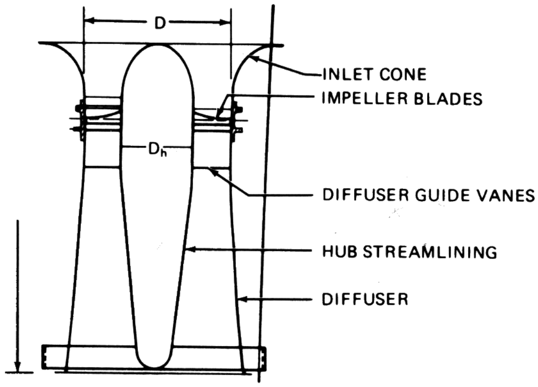


FIGURE 8-3

PERCENTAGE PERFORMANCE CURVES OF AN AXIAL-FLOW FAN

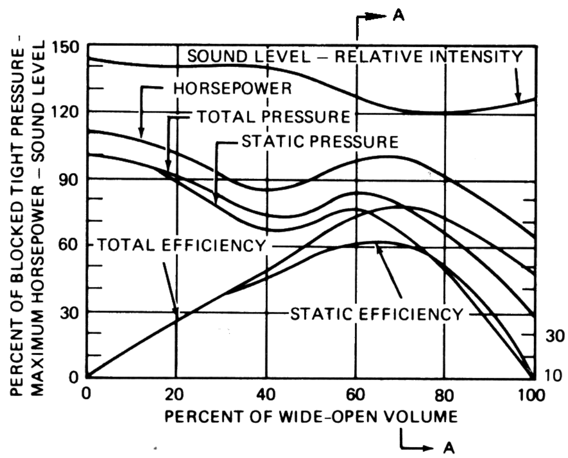


FIGURE 8-4

As a single unit, the axial fan is a high-flow, low-pressure device. Increased pressure capacity is obtained by adding stages in series, but this gain is at the expense of reduced overall efficiency.

An undistorted intake velocity profile across the entire fan area is important to prevent vibration and to maintain high efficiency. Turning vanes are necessary in all bends located in the inlet ducting to the fan to prevent air from building up on one side of the fan.

AXIAL FAN WITH CONTROLLABLE-PITCH BLADES

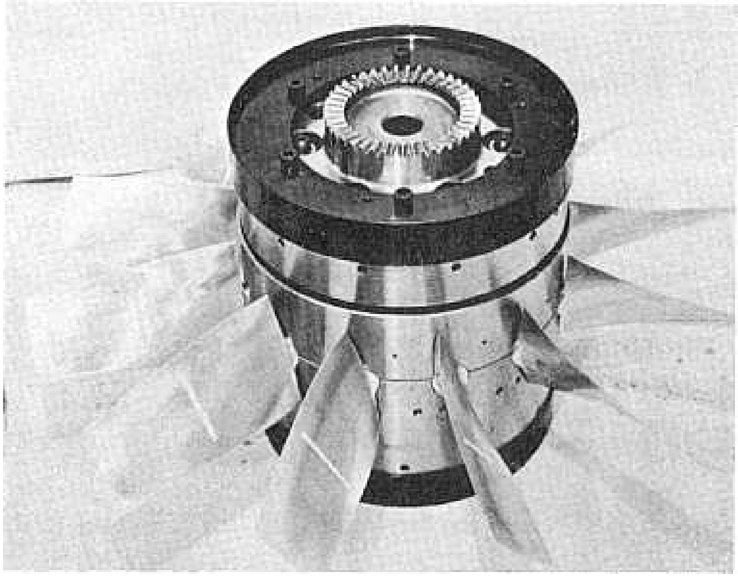


FIGURE 8-5

The overall proportions of the axial fan can be obtained from these relations (refer to Figure 8-3):

D = (see Fan System Design)

$$L_{f_1} = 1.2D$$

$$L_{f_2} = 1.4D$$

$$\frac{D_h}{D} = 0.5, \textit{ maximum}$$

where

L_f = fan length-feet
D = fan wheel diameter-feet
1 = one-stage
2 = two-stage
D_h = fan hub diameter-feet.

Means for determining the wheel diameter are presented later in the section on Fan System Design.

Based on use of lightweight materials, such as aluminum, the weight of a controllable-pitch vane axial fan, including housing and right-angle gear box, can be obtained from the following relations.

Weight of fan, housing, vanes, single stage, W_{A_1} (in pounds)

$$W_{A_1} = 40D^2$$

Weight of fan, housing, vanes, two stage, W_{A_2} (in pounds):

$$47D^2$$

Weight of drive mechanism, W_{AM} (in pounds)
(20 percent less for vane control)

$$= 1.5\,(HP).$$

For initial estimation (before the fans are sized), the following relations can be used.

Lift fan total weight, controllable-pitch, W_f (in pounds)

$W_f = 2\ lb/(HP)$ (refer to Equation (8.2)).

Fan total efficiency, per stage, $\eta_{f_{TOT}}$

$$\eta_{f_{TOT}} = 0.90.$$

Fan static efficiency, per stage, $\eta_{f_{STATIC}}$

$$\eta_{f_{STATIC}} = 0.76.$$

It should be borne in mind that two stages of axial fans will usually be required to develop the 200 to 300 pound/foot^2 cushion pressures required in a large SES. Thus, the efficiencies for a dual stage fan will be $(\eta_{f_{TOT}})^2 = 0.81$ and $(\eta_{f_{STATIC}})^2 = 0.58$.

b. Centrifugal Fans. This category of fans (Figure 8-6) includes forward-curved, radial, backward-curved, and backward-curved airfoil blade types. Typical characteristic curves for forward-curved and backward-curved airfoil types are given in Figure 8-7 and 8-8, respectively. These curves are obtained from test data; characteristic curves for the other fan types lie between the two shown.

The comparative advantages of the backward-curved-airfoil type blade are: (a) its total and static efficiencies are higher; (b) it has a relatively flat pressure vs. flow characteristic, with no dip (unstable range) at reduced flow; (c) it is quieter; and (d) its power curve dips at high flow rates. This last characteristic is particularly advantageous in SES application when, as a result of fuel burn-off, a lower cushion pressure is required. By venting part of the air flow, this can be accomplished at a reduced power level. Additionally, becuase this type of blade has a high static pressure efficiency, it can be used to produce high pressures without resorting to multistaging. It thus offers a distinct advantage over axial types, since it represents a simpler installation with higher overall efficiencies.

CENTRIFUGAL IMPELLER COMPONENTS

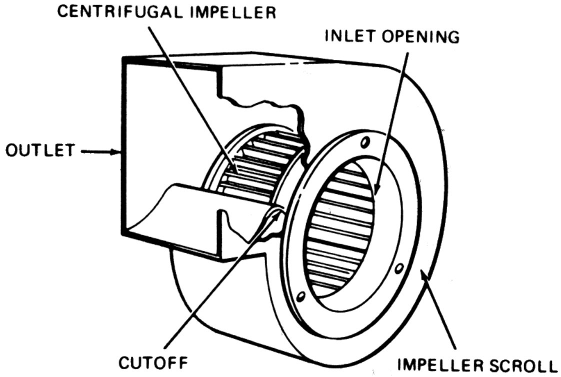


FIGURE 8-6

PERCENTAGE PERFORMANCE CURVES OF A FORWARD-CURVED AIRFOIL BLADE CENTRIFUGAL FAN

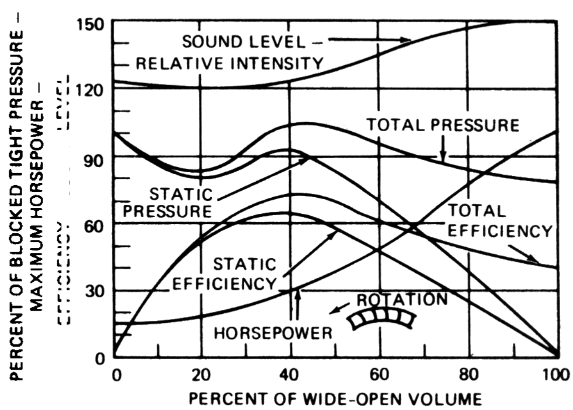


FIGURE 8-7

Again comparatively speaking, the centrifugal fan is less sensitive to distorted inlet flow patterns than the axial fan; thus, fewer inlet and ducting problems are encountered. The system, does, however, require a bulky diffuser to convert dynamic pressure to static pressure. This diffuser can be a vaneless volute type (Figure 8-6) or a vaned diffuser which delivers the air either parallel to the shaft (Figure 8-9) or radial to the shaft (Figure 8-10).

Modifications to the centrifugal fan curve have been achieved through use of variable inlet vanes, variable speed, and dampers. Controllable-pitch fan blades have been proposed, but to date no hardware of this type has been developed.

Overall proportions of the centrifugal fan with backward-curved airfoil blades may be obtained from the following equations (refer to Figure 8-11).

PERCENTAGE PERFORMANCE CURVES OF A BACKWARD-CURVED AIRFOIL BLADE CENTRIFUGAL FAN

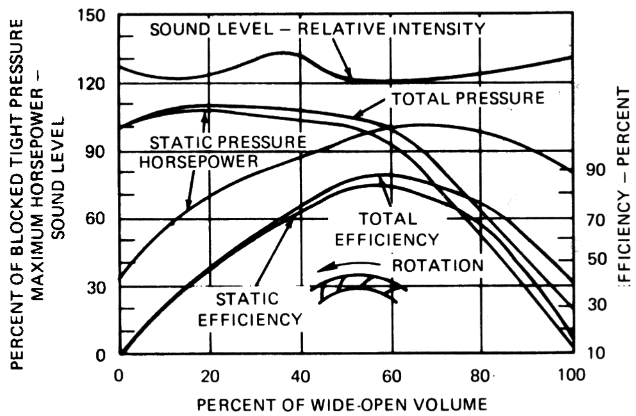


FIGURE 8-8

DIFFUSER FLOW PARALLEL TO SHAFT

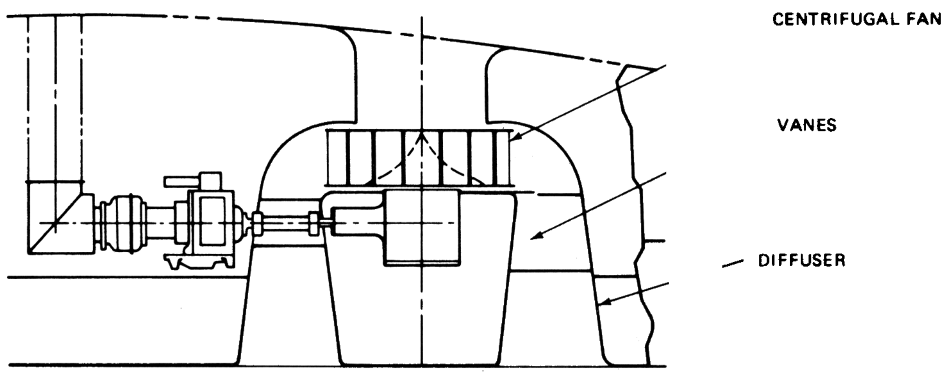


FIGURE 8-9

DIFFUSER FLOW RADIAL TO SHAFT

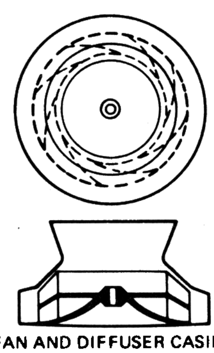


FIGURE 8-10

CENTRIFUGAL FAN DIMENSIONS

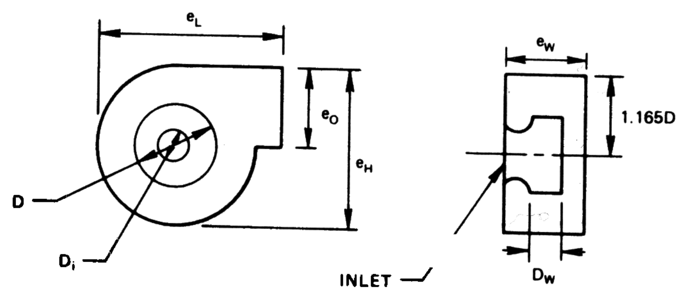


FIGURE 8-11

$$D = \text{(see Fan System Design)}$$
$$D_W = 0.26D$$
$$D_i = 0.75D$$
$$\text{No. of blades} = 8 - 16$$
$$e_H = 2.0D$$
$$e_L = 1.6D$$
$$e_w = 0.42D$$
$$e_o = 1.5D$$

where

D = fan wheel diameter-feet
D_W = fan wheel width-feet
D_i = fan inlet diameter-feet
e_H = volute maximum height-feet
e_L = volute maximum length-feet
e_w = volute maximum width-feet
e_o = volute outlet height-feet.

These dimensions are for a single inlet fan. For a double inlet fan (having inlets on right and left sides), volute maximum width would be $2e_w$. Means for determining the wheel diameter are presented later in the section on Fan System Design.

Based on the use of aluminum for the fan wheel and housing, fan weight can be approximated from the following relations.

Weight drive shaft, bearings, and supports, W_{CM} (in pounds)

$$W_{CM} = 1.0\,(HP).$$

Weight single inlet fan wheel and enclosure, W_{C_1} (in pounds)

$$W_{C_1} = 70D^2$$

Weight double inlet fan wheel and enclosure, W_{C_2} (in pounds)

$$W_{C_2} = 100D^2$$

For initial estimations (before the fans are sized) the following relations can be used for fans with backward-curved airfoil blades.

Lift fan total weight, W_f (in pounds)

$W = 2\ lb/(HP)$ (refer to Equation (8.2)).

Fan total efficiency per stage, $\eta_{f_{TOT}}$

$$\eta_{f_{TOT}} = 0.9.$$

Fan static efficiency per stage, $\eta_{f_{STATIC}}$

$$\eta_{f_{STATIC}} = 0.86.$$

c. **Mixed-Flow Fans.** The mixed-flow fan wheel (Figure 8-12) consists of contoured blades mounted between a curved inlet ring and a backplate. The blades are axially shaped on the intake side and centrifugally shaped on the discharge end.

TYPICAL MIXED-FLOW BLOWER WHEEL

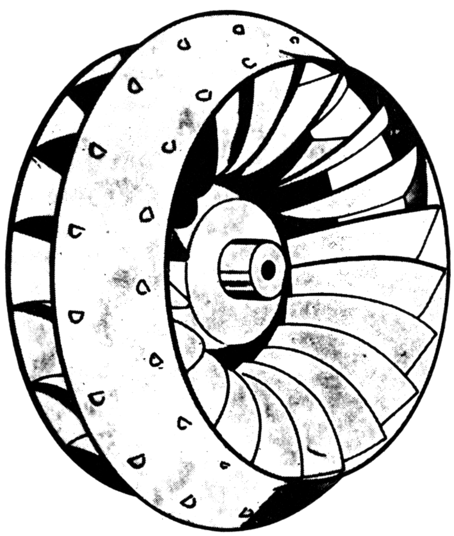


FIGURE 8-12

The performance characteristic of this fan is similar to that of the forward-curved centrifugal blade type except that it has no unstable range (0 to 40 percent wide open volume as shown on Figure 8-7). It is also generally installed in a housing similar to that of the centrifugal fan. The mixed-flow fan has a particular advantage for SES application, however, in that it can be used without a housing to produce a fairly flat static pressure curve (Figure 8-13).

COMPARISON OF PRESSURE-FLOW CHARACTERISTICS OF MIXED-FLOW AND CENTRIFUGAL BLOWERS

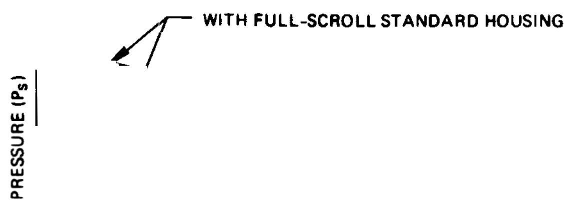


FIGURE 8-13

Dimensions and weights can be estimated under the procedures used for centrifugal fans. However, efficiencies will be different.

Mixed-flow fan static efficiency per stage, $\eta_{f_{TOT}}$:

$$\eta_{f_{TOT}} = 0.72.$$

Mixed-flow fan static efficiency per stage, $\eta_{f_{STATIC}}$:

$$\eta_{f_{STATIC}} = 0.62.$$

Fan Materials. The British have extensive experience in lightweight centrifugal fan manufacture for air cushion vehicles. Materials employed include aluminum, glass reinforced plastic (GRP), and stainless steel (References 3, 4 and 38).

In the United States there is extensive experience with controllable-pitch fans developed for aircraft use. Lightweight materials in current use include aluminum and titanium. Development work is being carried out with hollow blades, using composite materials of boron epoxy and aluminum. Although the use of these materials may not significantly affect the weight of the fan system, they will increase system life and resistance to damage from ingestion of foreign matter (Reference 39).

Due to material limitations, tip speeds for centrifugal and axial fans are limited by stress considerations. For example, a forged aluminum 8.5-foot-diameter fan is limited to a top speed of about 650 feet/second. Further increases in permissible tip speeds achieved through use of improved materials would reduce the fan size for a given power level.

Fan System Design. Fan system design is influenced by the cushion air requirements, the ship layout, and the characteristics of the fan system itself. The first two factors are largely apparent to the designer.

Ship motions determine the cushion pressure and flow requirements. Because of some overriding considerations, such as weapon systems requirements, compromises will often have to be made in fan location resulting in non-optimum lift system performance. Tradeoff studies are then required to minimize system degradation. In dealing with the last factor, the internal characteristics of the system, an iterative process must be used to obtain an optimum design.

The first step in this process is to translate the cushion air requirements into rough fan requirements. Basic data required are the cushion pressures for the fully loaded ship, P_{C_L}, and for the zero fuel condition, P_{C_E}, and the cushion air flow rate, Q. The cushion horsepower $(HP)_C$ is given by (see Equation (7.2)):

$$(HP)_C = \frac{Q(P_C + \Delta P_C)}{33{,}000\, \eta_F\, \eta_{TC}\, \eta_D} = \frac{Q(P_C + \Delta P_C)}{33{,}000\, \eta_{LS}}$$

where

Q = cushion air flow rate – feet^3/minute (refer to Equation (8.1))
P_C = cushion pressure – pounds/foot^2.

The brake horsepower required is obtained by estimating the fan, duct, and mechanical efficiency of the system. Reasonable initial lift system efficiency η_{LS} is 0.70. ΔP_C may be assumed to be 0.05 P_C.

Thus,

$$(HP)_C = 4.55 \times 10^{-5}\, Q\, P_C \tag{8.2}$$

This total power requirement is obtained for both the P_{C_L} and P_{C_E} values (for a fossil fuel ship). Some idea of the number of fans can now be obtained by considering the available engines and the difference in power levels between fan power for the fully loaded ship $(HP)_{C_L}$, and for the zero fuel condition, $(HP)_{C_E}$.

Since the efficiency of the gas turbine is greatest at maximum power level, it is advantageous to shut down some engines and keep the others running at the highest allowable power as the total power requirement drops off. Selection of the number of fans, then, is based on employment of engines with the lowest specific fuel consumption which are capable of satisfying the overall power requirement while permitting one or two engines to be shut down as fuel is consumed. In making this selection it is well to keep in mind that one engine can drive two or more fans and that more than one engine can drive a single fan. For reliability purposes, a minimum of three separately driven fans is desirable, since the loss of one can usually be compensated for by the remaining two operating at higher rpms.

The next step is to determine the rpm, n, for each fan. The air flow requirement, Q_f, for each fan will be proportional to the power supplied by its driving engine. Thus,

$$Q_f = Q\, \frac{(HP)_{C/\text{fan}}}{(HP)_C}$$

Then,

$$n = 6.95 \frac{P_f^{3/4} N_S}{Q_f^{1/2}} \quad (in\ rpm)$$

where

P_f = fan discharge total pressure $= \dfrac{0.1924\, P_C}{\eta_D}$ (in inches of H_2O)

N_S = specific speed (refer to Figure 8-14).

η_D = duct efficiency, typically 0.80.

The specific speed, N_S, is selected at the optimum efficiency for the selected fan configuration. The corresponding specific diameter, D_S' is also obtained from Figure 8-14, and the fan wheel diameter, D, is computed from

$$D = \frac{0.0437\, D_s'\, Q_f^{1/2}}{P_f^{1/4}} \quad (in\ feet).$$

The tip speed of the fan, V_T, is obtained from

$$V_T = \frac{\pi D n}{60} \quad (in\ feet/second)$$

or

$$V_T = 0.159\, D_S'\, N_S\, \sqrt{P_f} \quad (in\ feet/second).$$

FAN DESIGN CHARACTERISTICS

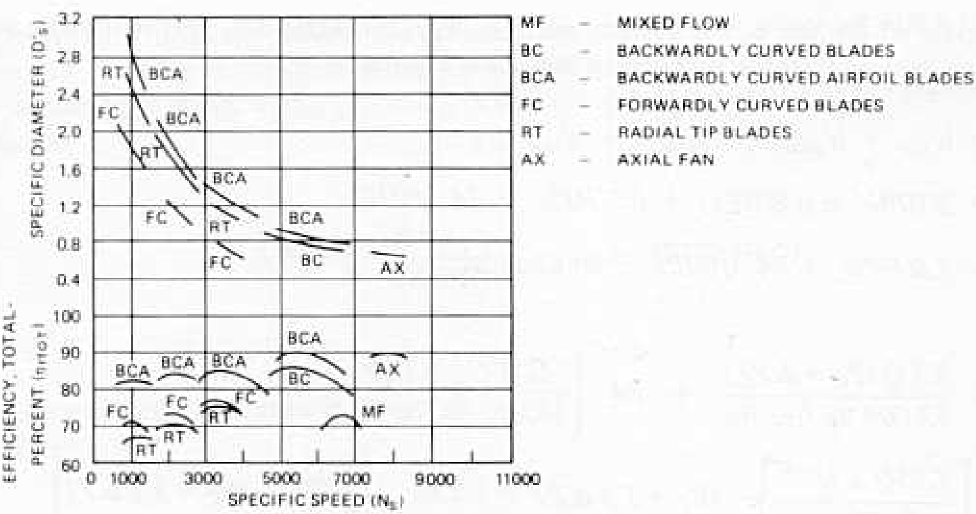


FIGURE 8-14

Limitations on tip speed will be imposed by stress and noise considerations. A new specific speed should be selected and the fan resized if the tip speed is too high or if the fan speed or fan diameter do not interface properly with other components.

With the fan diameter known, the fan dimensions and weight can be determined from the applicable formulas previously presented. The ducting system can now be designed and the actual ducting losses determined. The points to be considered in designing the ducting are to minimize duct losses, maximize the static pressure of the air delivered to the cushion, and minimize the weight of the system.

Total pressure losses – over and above those due to friction in straight sections – occur in duct bends, inlets, outlets, and areas of contraction and expansion. These losses are minimized by designing the runs as straight as possible, with gradual changes in area where necessary, and by maintaining low duct velocities.

The static pressure of the air delivered to the cushion can be maximized by reducing the velocity of the air entering the cushion area.

Although gradual changes in duct area and reduction in duct velocities will improve system efficiency, this will be gained only at the expense of increased duct weight. A tradeoff analysis is necessary to choose between this reduced power requirement weight and the increased duct weight.

Duct size is obtained from a duct friction chart, such as Figure 8-15. Round ducts are recommended because of their higher strength and lighter weight as compared with equivalent-area rectangular ducts.

Duct weight, W_D, is approximately 4 pounds/foot^2 of duct area, including acoustic insulation. Thus,

$$W_D = 1.047 d\, L \text{ (in pounds)}$$

where

L = duct length-feet

d = duct diameter-inches.

The weight of the fan power system, per fan, W_{PS}, including fan, gear reduction, gas turbine, and turbine ducting (fan system ducting excluded) is given by

$$W_{PS} = W_{fan} + W_{gear} + W_{turbine} + W_{turbine\ duct}$$

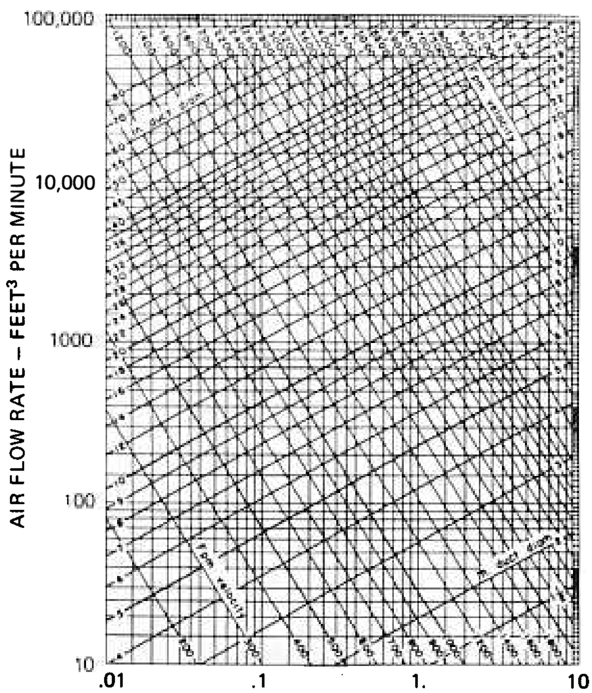
$$W_{PS} = 2(HP)_C + 0.8(HP)_C + 0.5(HP)_C + 84\sqrt{(HP)_C}$$

$$W_{PS} = 3.3(HP)_C + 84\sqrt{(HP)_C} \quad \text{(in pounds)}$$

$$W_{PS} = \frac{3.3\, Q\,(P_C + \Delta P_C)}{33{,}000\, \eta_F\, \eta_{TC}\, \eta_D} + 84 \left[\frac{Q\,(P_C + \Delta P_C)}{33{,}000\, \eta_F\, \eta_{TC}\, \eta_D}\right]^{1/2}$$

$$\frac{1.05Q \times 10^{-4}}{\eta_F\, \eta_{TC}}\,(P_C + 5.2\,\Delta P) + 0.475 \left[\frac{Q}{\eta_F\, \eta_{TC}}\,(P_C + 5.2\,\Delta P)\right]^{1/2}$$

DUCT FRICTION CHART



See Reference 40

FIGURE 8-15

where

ΔP = total pressure duct loss (in inches of H_2O).

On high velocity ducting systems maximum speeds are on the order of 6000 feet/minute (Reference 41). This figure can serve as a guide for initial approximations.

Losses occurring at inlets, exits, and changes in areas, ΔP_V, are related to the dynamic pressure by the relation

$$\Delta P_V = c\left(\frac{V_d}{4005}\right)^2 \qquad \textit{(in inches of } H_2O\textit{)}$$

where

V_d = duct velocity-feet/minute

Values of loss coefficient, c, are obtained from Table 8-1

LOSS COEFFICIENTS FOR DUCT AREA CHANGES

TYPE	ILLUSTRATION	CONDITIONS	LOSS COEFFICIENT	
ABRUPT EXPANSION	A_1, A_2	A_1/A_2	C_1	C_2
		0.1	0.81	81
		0.2	0.64	16
		0.3	0.49	5
		0.4	0.36	2.25
		0.5	0.25	1.00
		0.6	0.16	0.45
		0.7	0.09	0.18
		0.8	0.04	0.06
		0.9	0.01	0.01
GRADUAL EXPANSION	A_1, A_2, θ	θ	C_r	
		5°	0.17	
		7°	0.22	
		10°	0.28	
		20°	0.45	
		30°	0.59	
		40°	0.73	
ABRUPT EXIT	A_1, A_2 ($A_2 = \infty$)	$A_1/A_2 = 0.0$	1.00	

TYPE	ILLUSTRATION	CONDITIONS	LOSS COEFFICIENT
ABRUPT CONTRACTION SQUARE EDGE	A_1, A_2	A_2/A_1	C_2
		0.0	0.34
		0.2	0.32
		0.4	0.25
		0.6	0.16
		0.8	0.06
GRADUAL CONTRACTION	A_1, A_2, θ	θ	
		30°	0.02
		45°	0.04
		60°	0.07
EQUAL AREA TRANSFORMATION	A_1, A_2	$A_1 = A_2$, $\theta \leq 14°$	C
			0.15
FLANGED ENTRANCE		$A = \infty$	C
			0.34
DUCT ENTRANCE		$A = \infty$	C
			0.85
FORMED ENTRANCE		$A = \infty$	C
			0.03

TABLE 8-1

Losses occurring at bends can be estimated from Figure 8-16. When the diameter of the duct has been estimated, the effect of the bend is obtained in terms of an equivalent length of ducting, which is added to the measured ducting length (excluding the bend).

LOSS IN 90-DEGREE ELBOWS OF ROUND DUCTS

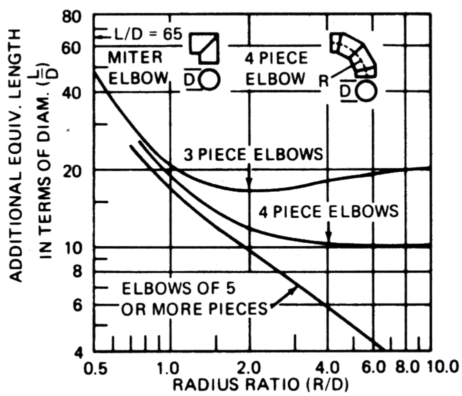


FIGURE 8-16

Diffuser sections can be used to advantage in regaining a good portion of the dynamic head. Figure 8-17 is used to determine the amount of total head lost and the amount of static head gained. These quantities are obtained from the following relations, where η_c = effectiveness for conical diffuser

Static head gain, ΔP_s (in inches of H_2O).

$$\Delta P_s = \eta_c \Delta P_V$$

Total head loss, ΔP (in inches of H_2O)

$$\Delta P = (1 - \eta_c) \Delta P_V$$

Dynamic pressure loss, ΔP_V (in inches of H_2O)

$$\Delta P_V = \left[\frac{V_{d_1} - V_{d_2}}{4005}\right]^2$$

It is preferable to use a diffuser section at the entrance to the plenum rather than to discharge directly into it at high velocity, since the latter procedure wastes all the dynamic energy.

The fan inlets are best located so as to reduce the ingestion of green water. Placing horizontal inlets flush on an upper deck is one possibility, but inboard-facing vertical inlets are more effective in that they also protect against rain ingestion.

In typical SES lift systems there is little aerodynamic advantage afforded by forward-facing intakes. Well-designed flush intakes can serve equally as well (Reference 38). This means the use of well-formed entrances that tend to supress separation by having gradual curvatures in the direction of flow. The laminar flow of the free-stream air moving by the ship into the fan reduces the amount of power the fan must deliver, since the air requires no acceleration. The theoretical amount of power saved increases with ship speed. In practice, laminar flow may not always be achievable under all conditions – and inlet losses, coupled with reduced fan efficiency, will tend to offset any gains. A typical flush intake configuration is shown in Figure 8-18.

The size of the inlet is related to the design speed of the ship. It is preferable that the air not be decelerated as it enters the duct, since this situation is more prone to cause flow separation (non-laminar flow) than acceleration at point of intake. The inlet is therefore sized so that

$$V_o / V_d \leqslant 1$$

where

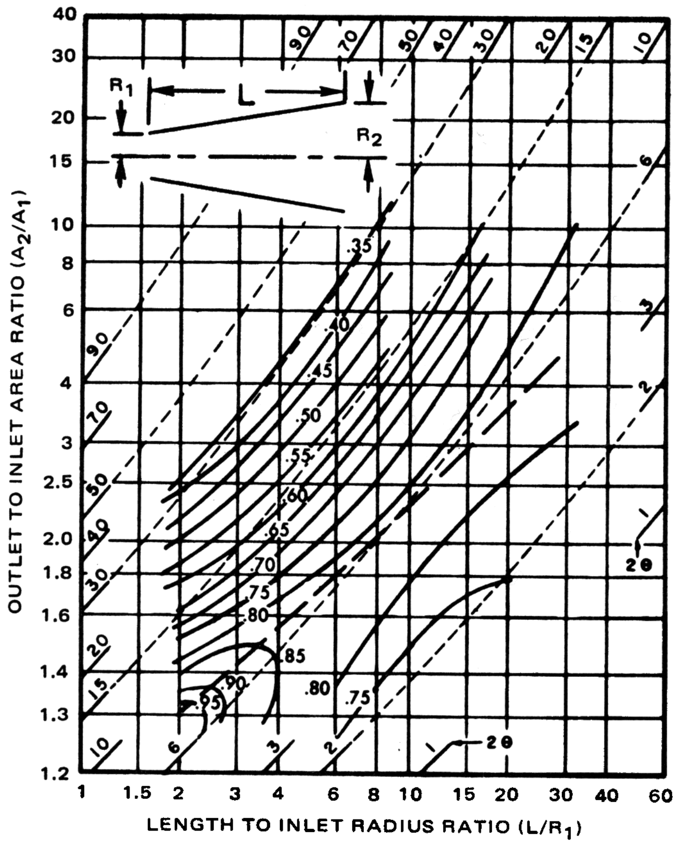
V_o = craft velocity relative to air-feet/second

V_d = duct inlet velocity-feet/second.

Intake losses based on laminar flow are obtained from Equation (8.3), and the loss coefficient factors for entrance losses are given in Table 8-1.

All the losses can now be summed and the derived duct efficiency substituted for the assumed one in Equation (8.2). A more accurate estimation of cushion power requirements is thus obtained; the fan is also resized to suit the adjusted fan pressure.

EFFECTIVENESS VALUES FOR CONICAL DUCT DIFFUSERS



See Reference 42

FIGURE 8-17

TYPICAL FLUSH DUCT INLET

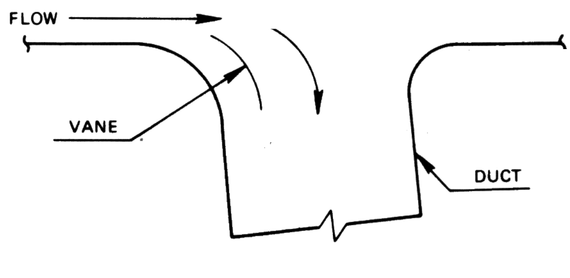


FIGURE 8-18

Fan performance over the operating range can be estimated by using the appropriate typical performance curve in Figure 8-7 or 8-8. The maximum efficiency, $\eta_{f_{TOT}}$, obtained from Figure 8-14, will correspond to the maximum total efficiency on the performance curves and the flow and pressure to be used to calculate N_S, specific speed, will be determined therefrom. The absolute values of flow and pressure at other than maximum efficiencies can then be calculated and a fan characteristic curve constructed. From this curve a plot can be made of the performance of fans acting as a total system, as in Figure 8-19. Such a plot is useful in assessing the excursion limits of the system and the necessity for accessory control devices.

FAN PERFORMANCE AS A SYSTEM

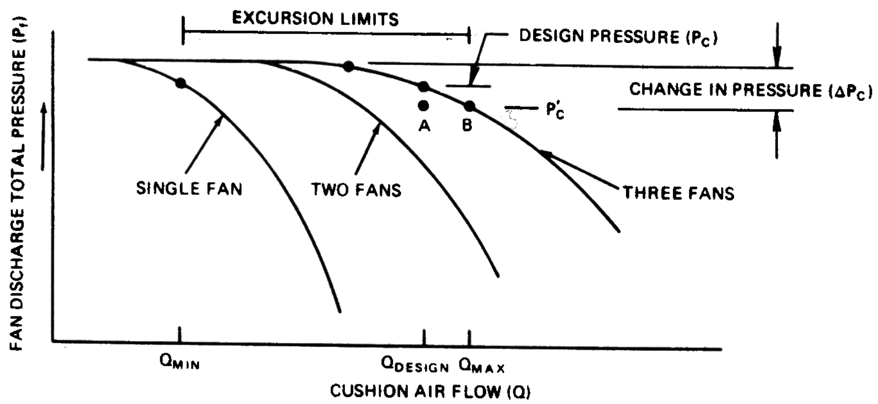


FIGURE 8-19

If the variation in cushion pressure, ΔP_C, is large enough to produce unacceptable vertical accelerations of the ship, consideration should be given to incorporating a controlled leakage or relief flow area. Such an area is so sized that, while the fan always operates at Point B (Figure 8-19), the cushion pressure, P'_C, can be maintained constant by opening the louvers to respond to a rise in cushion pressure (shift to Point A) and by closing them to respond to a drop in cushion pressure (shift back to Point B).

In the event a controlled leakage area is required, the fans will have to be resized so that cushion pressure, P'_C, (see Figure 8-19) is equal to the original cushion design pressure, P_C. This is accomplished by using in Equation (8.2) a cushion air flow rate of Q_{max} instead of Q_{design}.

Accessories. Closures are necessary to prevent the escape of cushion air through fans that may not be operating. These closures can be similar to butterfly, flapper, or gate valves which permit virtually free passage of air when open.

Relief valves, vents, or controlled leakage areas can be of louvered design to permit rapid opening and closing. Operation by hydraulic, pneumatic, or mechanical springs is feasible.

Drains should be provided at the low points of all ducts to remove ingested seawater and rainwater. Screening the ducting upstream of the fan will protect it from ingestion of foreign objects (ice, birds, etc.).

More detailed design considerations for ducting and fans may be obtained from References 41, 43, and 44.

Parametric Data. For initial estimations the following equations may be used to determine fan system weight:

$$W'_{fs} = 3.3\,(HP)'_C + 84\,\sqrt{(HP)'_C} + 256\left[\frac{(HP)'_C}{P_C\,V_d}\right]^{1/2} L$$

where

W'_{fs} = fan system weight – pounds per fan (fan, engine, cushion, and engine ducting)

L = cushion ducting length per fan – feet

P_C = cushion pressure – pounds/foot^2

V_d = duct velocity – feet/minute.

Then,

$$W_{fs} = 3.3\,(HP)_C + 84\sqrt{(HP)_C\,N} + 256\left[\frac{(HP)_C\,N}{P_C\,V_d}\right]^{1/2}$$

where

N = number of fans

$P)_C = N\,(HP)'_C$

$W_{fs} = N\,W'_{fs}$

Then, if $N = 4,\ L = 25$ *feet*, $P_C = 250$ *pounds/foot*^{2} and $V_d = 6000$ *feet/minute,*

$$3.3\,(HP)_C + 178.5\sqrt{(HP)_C}.$$

Figure 8-20 indicates the relationship between the fan system weight and horsepower and the ship gross weight for ships with cushion length-beam ratios of 2.0 and cushion densities of 1.5. It is interesting to note that, for a constant habitability standard, the power requirements for operating in sea state 6 (9-foot wave) at 40 knots are approximately 1.5 times those required for sea state 3 (3-foot wave) at 80 knots.

FAN SYSTEM WEIGHT AND HORSEPOWER VERSUS SHIP GROSS WEIGHT

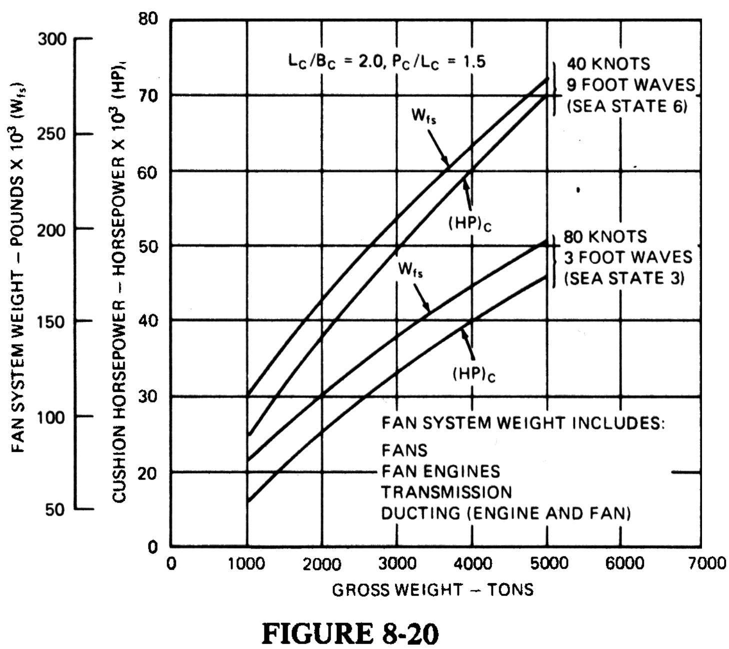


FIGURE 8-20

Chapter IX
SEALS

In SES design the bow seal and the stern seal function to contain the air cushion throughout the varied range of dynamic and sea-state conditions in which the ship operates. The air cushion should be contained without any adverse effect on the performance of the craft. Therefore, the seals should be highly responsive and, at the same time, provide as little drag as possible under all operating conditions. A general seal configuration is shown in Figure 9–1.

GENERAL SEAL CONFIGURATION

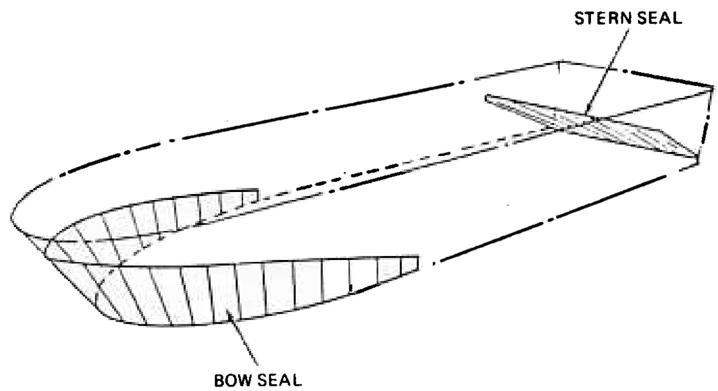


FIGURE 9-1

Many bow and stern seal designs have and are actively being considered during the evolution of the SES concept. Analysis of the proper functioning of the seals is, however, a highly unique and complex matter and, thus, the dynamic characteristics of various designs are difficult to evaluate.

BOW SEAL CHARACTERISTICS

Formed Membrane Seal. One of the simplest bow seals is shown in Figure 9–2A. This seal consists of a single membrane of fabric, attached as a unit along its upper edge. The seal is held in position by the force of gravity and the cushion pressure of the vehicle, which also returns the seal to its normal position when deflected by waves or obstacles.

Full Finger Seal. The full finger seal (see Figure 9–2B) consists of a series of individual fabric-folds, each of which is attached along the tip. Cushion pressure forms the folds into fingers and forces the edges of the folds together to provide an air seal. Because of the small radius of the fingers, this type of seal offers minimum resistance to vertical, aft, and side loads.

Partial-Height Finger Seal. By reducing the full finger seal in vertical length and inserting an air bag, formed to the bow contour, between the finger and rigid bow, the partial-height configuration shown in Figure 9-2C is formed. The air bag thus constitutes a variable-stiffness impact attenuator with varying inflation pressure and provides a righting force in the event of plow-in.

The fingers are attached to the air bag individually and offer minimal resistance to wave passage; thus being shorter, they will provide a faster response than the full finger skirt.

Partial-Height Membrane Seal. The partial-height membrane bow seal concept is shown in Figure 9-2D. As does the partial-height finger seal, this concept utilizes an air bag formed to the bow contour. Because the seal is a single membrane, it presents relatively few attachment problems. This type of seal has slightly greater wave passage resistance than the partial-height finger seal due to its higher pressure stresses.

Partial-Height Stayed Seal. The stiffened membrane, or stayed, bow seal concept is shown in Figure 9-2E. In this design a flexible fabric membrane is stiffened by metal ribs spaced at intervals around the bow. The seal extends a short distance aft of the point of tangency to form a non-rigid part of the sidehull; and a curtain of the fabric continues aft, inboard of the rigid sidehull, to provide a cushion seal between fabric and rigid sidehulls. A pressurized air chamber between the seal membrane and rigid bow structure forces the seal membrane against the water surface to retain the cushion pressure.

All of the bow seal concepts described have general application for use with partial-length rigid sidehull configurations. Particular requirements, however, such as for a lift-producing bow seal and high-speed performance, limit the application of some of the described concepts.

The basic characteristics of these five concepts are summarized and compared in Table 9-1, with each concept being ranked from 1 to 10 on the basis of each particular characteristic. The indication of seal response given is a qualitative evaluation only, based on general observations throughout the industry. Quantitative evaluation of frequency response requires a detailed dynamic analysis of each particular design, including such detailed considerations as fan performance, ducting size and arrangement, and wave environment. Because the seal designs have not been analyzed to this extent, and because insufficient data are available on seal response and characteristics listed in Table 9-1, the ratings shown should be considered as subjective.

STERN SEAL CHARACTERISTICS

Air Bag Seal. An extremely simple stern seal design is the air bag concept, as shown in Figure 9-3A. This concept consists solely of a piece of fabric with a semi-rigid planing surface on the lower surface. For a given arc length of fabric and fixed attachment points, the shape of the bag will change with varying internal pressure and/or external forces. This concept is highly flexible in that many end closures (seals between the bag and the sidehull) can be adapted to the bag. Typical of these are flat and contoured end closures.

Two-Lobe Bag Seal. Figure 9-3B shows a two-lobe bag seal consisting of two interconnected, inflated air bags. As compared with the single air bag concept, this configuration offers a significant reduction in the size and required length of seal-to-hull attachment. As with the single air bag, various end closures may be used. For more detailed information on this concept, refer to Reference 45.

Hinge Beam Seal. The hinge beam seal design is shown in Figure 9-3C. In the AIRMAT* design, the beam is an air inflated mat. The cushion is vented through large area ducts to the aft

*TM, Goodyear Aerospace Corporation – Akron, Ohio.

BOW SEAL DESIGNS

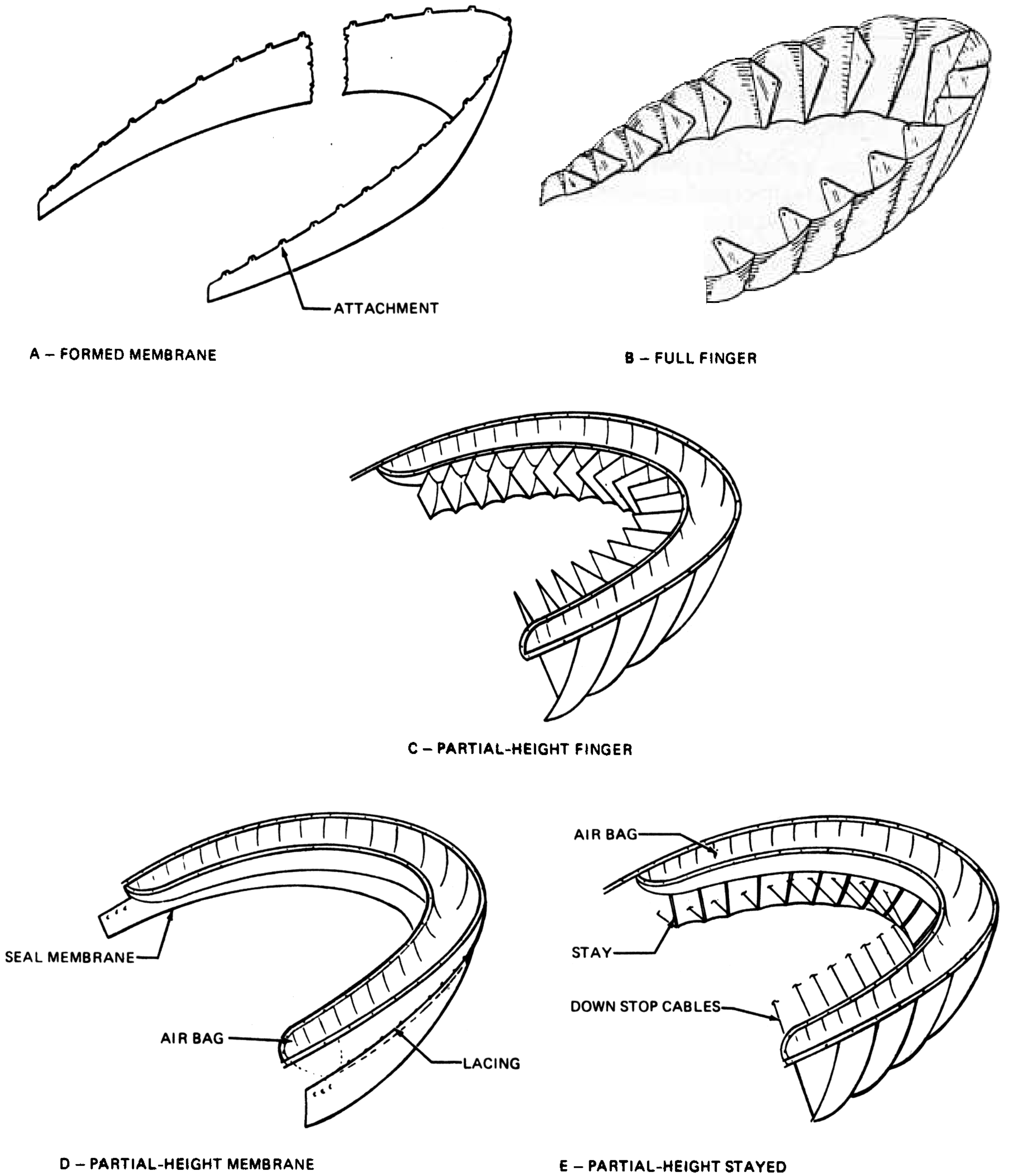


FIGURE 9–2

COMPARISON OF BOW SEAL DESIGNS*

Type of Seal	Characteristics	Low Wave Passage Resistance	Fast Response	Ease of Fabrication	Simplicity of Attachment	Seal Weight
Formed membrane	Simple to fabricate and attach; good for lower speed applications.	5	6	3	3	3
Full finger	Low wave passage resistance; good for low speed applications; complex to attach.	3	7	4	6	4
Partial-height finger	Low wave passage resistance; intermediate to high speed application; complex to fabricate and attach.	4	5	6	4	6
Partial-height membrane	Intermediate to high speed application; complex to fabricate and attach.	6	4	5	5	5
Partial-height stayed	High response rate; high speed application; complex to fabricate and attach.	7	3	7	7	7

*The designs considered are given a relative ranking for the particular characteristic considered. A number 1 would indicate an excellent rating and the number 10 an unsatisfactory rating.

TABLE 9-1

portion of the seal to minimize the spring force required to place the seal in static equilibrium against cushion forces. This arrangement also minimizes the structural requirements of the seal beam, minimizes the cushion pumping effect due to seal motion, and virtually eliminates the large damping effects that are detrimental to seal response. The spring force is provided by a small, high-pressure air spring located near the hinge point. With this system, the seal preload force is a function of the initial preload spring pressure and of the time rate of change of volume and mass flow from the preload spring to the spring plenum chamber.

Double-Hinge Beam Seal. A two-section-hinged rigid beam seal is shown in Figure 9-3D. A lobed air bag spring controls the upper beam section, while a smaller air bag spring controls the flexure of the lower beam section with respect to the upper beam section. The operating pressures in each air bag spring may be varied as required.

Stiffened Membrane Seal. The stiffened membrane seal, one of the basic stern seal concepts to be investigated and developed, is shown in Figure 9-3E. This concept utilizes a piano hinge and a flexible membrane of neoprene-coated cloth, stiffened at intervals by metal ribs in the fore and aft direction. Vertical adjustable cables are used to provide some variation in the shape of the seal membrane. Single or multiple air chambers, constructed of two or more interconnected lobed compartments, are located between the seal membrane and the vehicle structure.

STERN SEAL DESIGNS

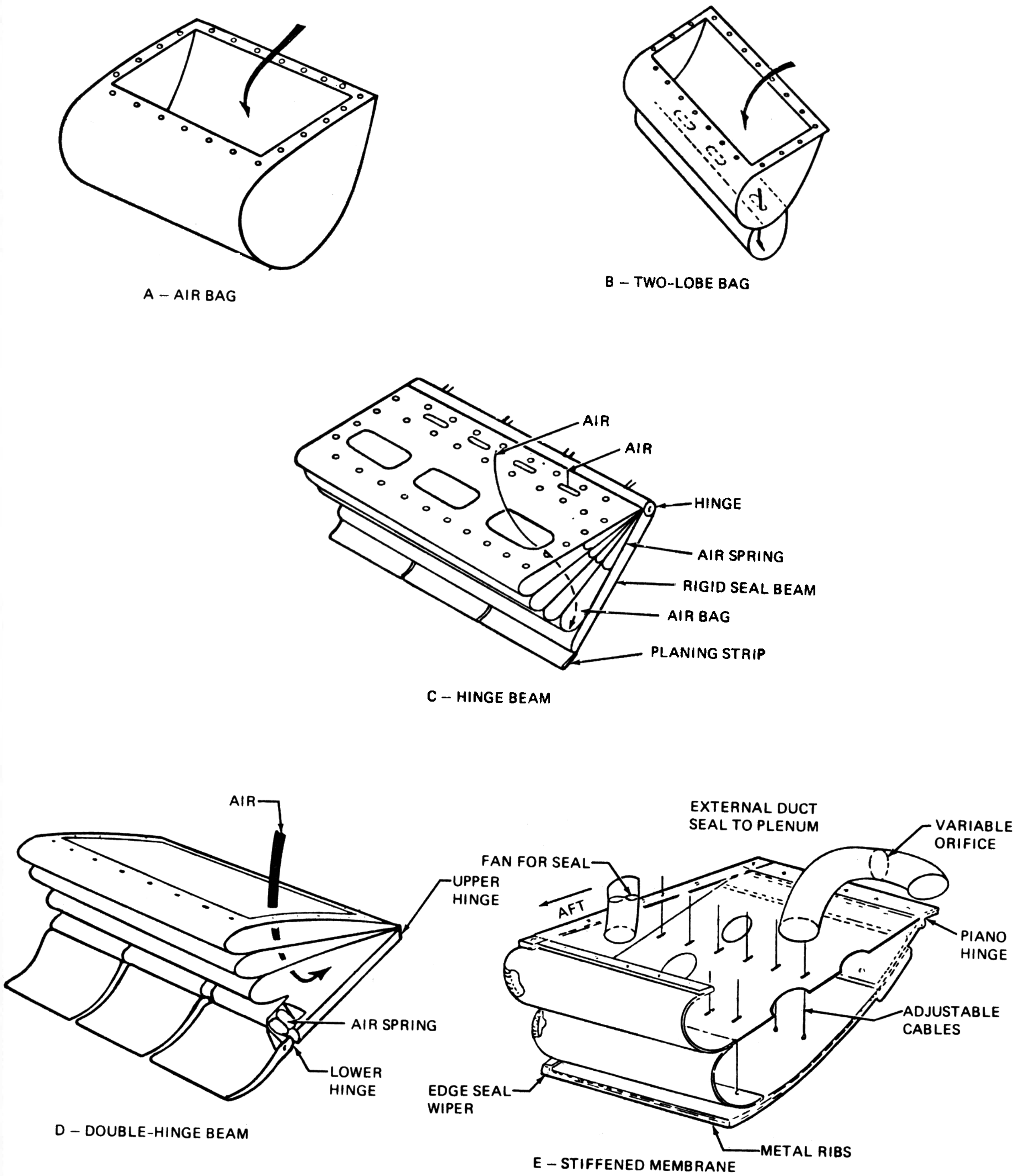


FIGURE 9-3

This air chamber is pressurized by a fan fed from the atmosphere, and the pressure is regulated by a variable orifice in the venting duct between the air chamber and the vehicle plenum.

A comparison of the general characteristics of each of these five concepts is presented in Table 9–2. For the stern seals as for the bow seals (Table 9–1), frequency response can be evaluated only qualitatively owing to the lack of detailed design analysis and the unavailability of appropriate data.

COMPARISON OF STERN SEAL DESIGNS*

Type of Seal	Characteristics	Low Wave Passage Resistance	Fast Response	Ease of Fabrication	Simplicity of Attachment	Seal Weight
Air bag	Simple to fabricate and attach; light weight; relatively high hydrodynamic drag.	7	7	3	3	4
Two-lobe bag	Light weight; simple to attach; relatively high hydrodynamic drag.	6	6	4	4	3
Hinge beam	Good hydrodynamic performance over wide speed range; high response rate; complex to fabricate and attach.		5	5	5	7
Double-hinge beam	Good hydrodynamic performance over wide speed range; high response rate; complex to fabricate and attach.	3	4	6	6	6
Stiffened membrane	Good hydrodynamic performance and wave contouring ability; complex to fabricate; relatively low susceptibility to damage.	4	3	7		5

*The designs considered are given a relative ranking for the particular characteristic considered. A number would indicate an excellent rating and the number 10 an unsatisfactory rating.

TABLE 9–2

DESIGN PROCEDURE

The overall selection of the types of bow and stern seals to be used should be based on the vehicle operational requirements and the fabrication facilities available. If the vehicle and seals are being designed on the basis of model tests, scaling may be important with respect to data on physical dimensions, pressures, forces, and response times. In the following sections scaling techniques are discussed, along with such advanced design considerations as seal wave–encounter frequencies, bow and stern seal selection, basic material stresses, seal attachment, and dynamic response. This discussion should provide the seal designer with sufficient information to generate a conceptual seal design.

Scaling. Model scaling becomes important when basing the design criteria for a vehicle on the performance of a dimensionally similar vehicle. For dynamic similarity to exist between the prototype and the model, the Froude modeling ratios are set equal to each other:

$$\frac{V_p}{\sqrt{L_p \times g}} = \frac{V_m}{\sqrt{L_m \times g}}$$

where

V = velocity

L = physical dimension

g = gravitational acceleration (32.2) – feet/second^2

and the subscripts are

m = model

p = prototype.

Based on this relationship, an equation for the linear velocity of the prototype, V_p, can be written as follows, where the scaling ratio, SR, is equal to L_p/L_m:

$$V_p = V_m \sqrt{SR}$$

Wave heights, h_w, and wave lengths, λ, as well as vehicle dimensions, are scaled directly by the scaling ratio. Wave height and wave length relationships are as follows:

$$h_{wp} = h_{wm}(SR)$$

$$\lambda_p = \lambda_m(SR).$$

The vehicle size relationship for length, width, and height is

$$L_p = L_m(SR).$$

Time, t, also can be scaled to relate times observed in the model to the prototype vehicle. Because velocity is defined as a distance traveled per unit time, the velocity expression given previously can be used to determine the desired time relationship:

$$V_p = L_p/t_p.$$

Therefore,

$$t_p = \frac{(L_p)(t_m)}{L_m \sqrt{SR}}$$

Thus,

$$t_p = \sqrt{SR}\, t_m .$$

For a hinged seal, the angular rotation, θ, will be the same for both the model and prototype vehicles, as both wave and seal dimensions are scaled by the scaling ratio. The angular velocity, $\dot{\theta}$, and angular acceleration, $\ddot{\theta}$, applicable to the prototype are related to the corresponding variable in the model by the following relationships:

$$\dot{\theta}_p = \frac{1}{\sqrt{SR}}\, \dot{\theta}_m$$

$$\ddot{\theta}_p = \frac{1}{SR}\, \ddot{\theta}_m$$

Tangential acceleration, A_T, and normal acceleration, A_N, of a hinged seal are defined by the following equations:

$$A_T = radius \times \ddot{\theta}$$

$$A_N = radius \times \dot{\theta}^2$$

When dimensional similarity exists, the tangential and normal accelerations in the prototype and the model are identical.

Other pertinent relationships for dimensionally similar situations are given below:

Weight (or mass), W (in pounds):

$$W_p = (SR)^3 \times W_m$$

Force, F, (in pounds):

$$F_p = (SR)^3 \times F_m$$

Torque, T (in inch-pounds):

$$T_p = (SR)^4 \times T_m$$

Mass moment of inertia, I (in pounds-second^2-inch):

$$I_p = (SR)^5 \times I_m$$

Pressure, P (in pounds/inch^2):

$$P_p = (SR) \times P_m .$$

Seal Wave-Encounter Frequency. The wave height versus velocity curve shown as a dashed line in Figure 9–4 is a typical operating envelope for the SES. The curve indicates the speed up to which the vehicle can be operated for the given wave height. Seal wave–encounter frequency for unity wave height is also plotted in Figure 9–4 as a function of craft velocity. Various wave height–wave length ratios are plotted ranging from 1/20 to the theoretically limiting value of 1/7. The wave encounter frequency indicated for the 1/7 wave height-wave length ratio represents the highest possible cyclic frequency for a one–foot–high wave. Wave heights greater than one foot, up to and including the limits of the wave height envelope, represent reduced encounter frequencies as defined by the frequency equation below:

$$f = FR/h_w$$

where

f = frequency – cycles/second
FR = frequency factor – cycles/second/foot
h_w = wave height – feet.

No operational data are available on which to base a decision on what constitutes a high or a low frequency. However, a wave encounter frequency of 10–Hz or above is arbitrarily considered high.

As Figure 9–4 indicates, a short choppy wave represented by the 1/7 wave height–wave length ratio curve will exceed the 10–Hz value at relatively low craft velocities; these types of waves are thus the hardest to follow.

Bow **Seal Design.** In the following subsection specific attention is given to some of the more important considerations governing bow seal design.

a. **Air Bag-Seal Height Ratio.** Several of the bow seal designs previously described utilize an air bag with a fabric finger or membrane extension. The purpose of the air bag is to improve seal response while providing a positive–lift force on the bow in rough sea conditions. Such operational requirements as sea state and craft pitch stability must therefore be considered in selecting the initial air bag–skirt height ratio for a particular vehicle. Table 9–3 summarizes the effect of air bag size on response time for both open and closed pneumatic systems (see Figure 9–5).

With an open system, the smaller the air bag, the faster its response rate will be under conditions of either identical pressure or seal stiffness. This relationship as stated does not take into account the inertial effect of the seals and stays and, as such, is applicable only as long as the inertial effect remains small in comparison with the bag restoring force. Bow seals have been designed with air bags of heights ranging between 50 and 70 percent of cushion height. Tests have indicated highly satisfactory performance for the vehicles involved.

b. **Pressure Stresses.** Figure 9–6 shows some of the construction detail of the partial height finger and the hoop and longitudinal stresses for both the finger and the air bag. Selection of a suitable seal material is dependent on the pressure stresses in the finger and the air bag. The

SEAL WAVE-ENCOUNTER FREQUENCY

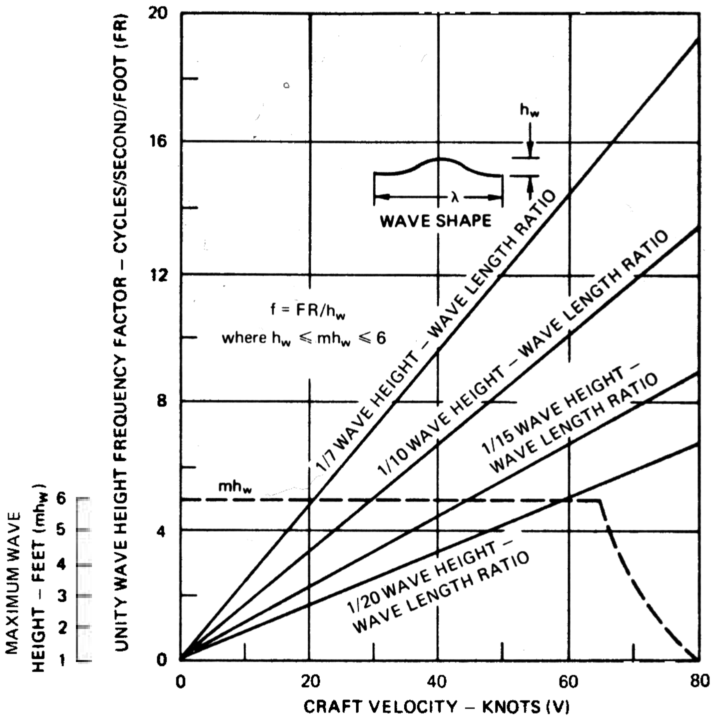


FIGURE 9–4

OPEN OR CLOSED AIR BAG DESIGN

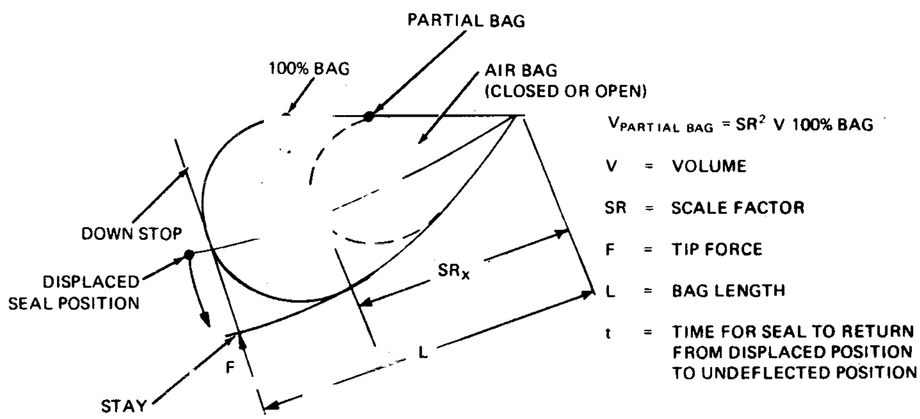


FIGURE 9–5

EFFECT OF AIR BAG SIZE ON RESPONSE TIME

Condition	Closed Pneumatic System		Open Pneumatic System	
	Effect on Response Time	Remarks	Effect on Response Time	Remarks
Identical seal pressure	$t_{partial\ bag} = t_{100\%\ bag} = P(pressure)$	Identical response time irrespective of scale factor, SR	$t_{partial\ bag} \sim SR^2 t_{100\%\ bag}$	Smaller bag responds faster, or $1/SR^2$ times faster than the 100% bag (fan operates at wide-open condition)
Identical seal stiffness (tip force, F)	$t_{partial\ bag} = SR^{1.5} t_{100\%\ bag}$	Smaller bag responds faster, or $1/SR^{3/2}$ times faster than a 100% bag	$t_{partial\ bag} \leqslant SR^2 t_{100\%\ bag}$	Smaller bag responds faster (fan operates at wide-open condition)

TABLE 9-3

PARTIAL HEIGHT FINGER DESIGN

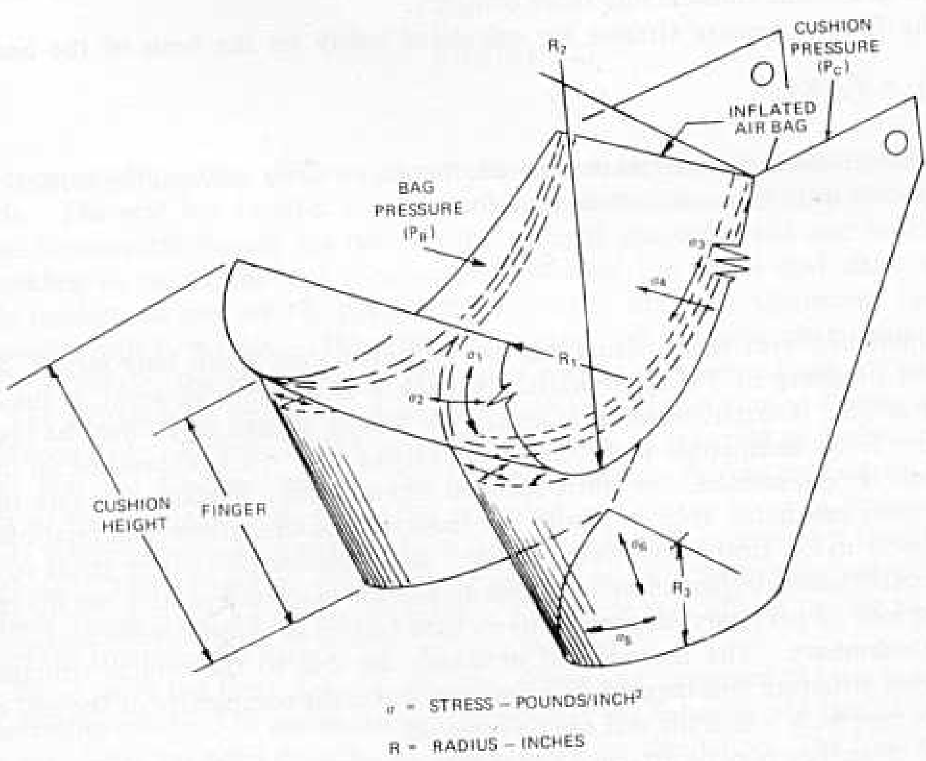


FIGURE 9-6

following equations can be used to determine these pressure stresses, first for the air bag and then for the finger.

For the air bag the hoop stress is determined by

$$\sigma_1 = p_B R_1$$

$$\sigma_3 = (p_B - P_C) R_2$$

and the longitudinal stress, for a straight section, is given by

$$\sigma_2 = \sigma_4 = 0$$

where

σ = stress – pounds/inch^2
R = radius – inches
p_B = bag pressure – pounds/inch^2
P_C = cushion pressure – pounds/inch^2

The subscripts for σ and R are as indicated in Figure 9–6.

If the air bag is curved rather than being a straight section, as shown in Figure 9–6, the stress equations become considerably more complex.

For the finger, pressure stresses are calculated solely on the basis of the hoop stress:

$$\sigma_5 = P_C R_3$$

There is no longitudinal stress to be considered, that is, no stress exists in the vertical direction as there is no curvature in this direction; therefore,

$$\sigma_6 = 0.$$

Past experience with fabric structures has indicated that, when long service life is desired, a factor of safety of 5 is appropriate. Not taken in account, however, is the effect of fatigue on seal life. Current research indicates the factors as high as 10 may be required to maintain stress levels compatible with the desired service life. Thus, depending on the extent of exposure to fatigue stresses, the fabric selected should have a breaking strength of 5 to 10 times the stresses calculated above. Figure 9–7 indicates the strength-weight relationship of a typical uncoated nylon cloth and "Kevlar" (Fiber B). Elastomeric coating equal to or more than the uncoated cloth weight must be added to seal the cloth and protect the fibers. Additional coating may be necessary, depending upon such factors as abrasion resistance.

c. **Seal Attachments.** The means used to attach the seal to the vehicle structure or to another inflated structure will depend to a large extent on the complexity of the seal configuration and the ease with which the seal can be replaced.

The seal or air bag may be attached to the vehicle structure by a clamp plate attachment. A finger seal may be attached to an air bag by a woven or sewn Y-joint. The finger seal can be laced if it is to be replaced independently of the upper bag.

STRENGTH-WEIGHT RATIO OF TYPICAL SEAL FABRICS

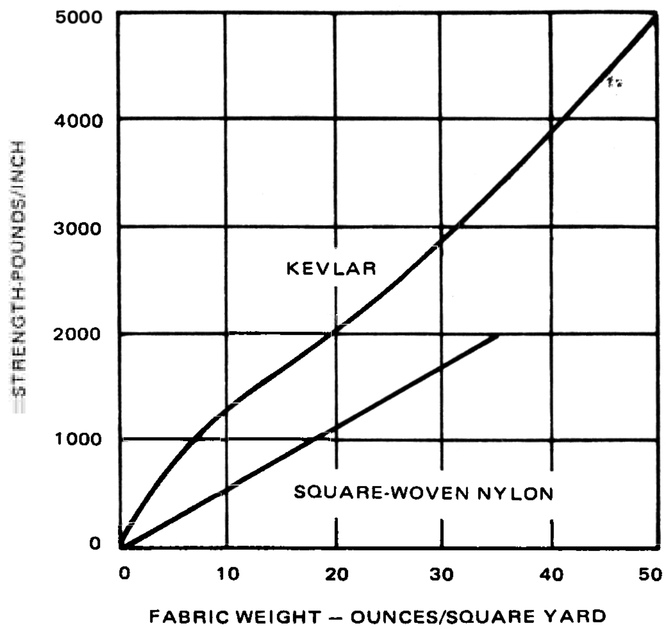
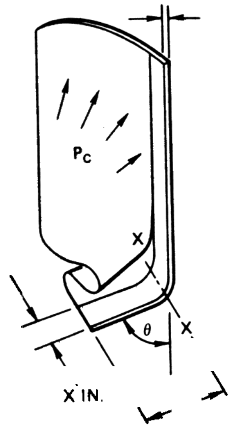


FIGURE 9-7

d. **Dynamic Response.** SES seals are being flexed continually as waves pass underneath the vehicle. The seal is a flexible membrane structure having many natural frequencies. These natural frequencies depend not only on the material properties but also on the size and type of buckling to which the seal is subjected. Because the waves that cause the buckling are highly random, so also are the natural frequencies of the seal. Obviously, then some natural frequencies will be excited. However, due to the high damping characteristics of the water surface interface, the mode shapes associated with these resonant conditions are suppressed.

The British have actually observed the seal-wave interaction at the water interface on a film taken underneath an SRN 4 air cushion vehicle, as reported in Reference 46. This film shows that the leading edges of the fingers undergo a flexing motion in passing over the crests of small waves. The frequency recorded was on the order of hundreds of cycles per minute (several cycles per second), which is consistent with wave impact frequencies previously mentioned. Thus, based on the observations reported by the British, and taking into account the high damping nature of the sea system, the cyclic rate of seal flexing is considered to be that of the frequency of wave interactions on the seal.

Some work has been done in the area of dynamic response of bow seals; because of the complexities involved in mathematically describing the behavior of a flexible membrane, however, very little has been done in the way of dynamic simulation. The following equation can be used to obtain a rough approximation of seal response time, or the time for the seal to return to its initial position after being deflected, as shown in the accompanying sketch:



$$= \sqrt{\frac{40 I_{xx}}{x P_C \ell^2}}$$

where

θ = deflection angle – radians
t = response time – seconds
ℓ = deflected length – inches
P_C = cushion pressure – pounds/inch^2
x = unit width of fabric along the X axis – inches
I_{xx} = mass moment of inertia about the X axis – pounds-second^2-inch

$$I_{xx} = \frac{m\,(d^2 + \ell^2)}{12} + \frac{m\ell^2}{4}$$

where

m = mass of moving fabric ℓ inches long, x inches wide, and d inches thick – pounds-second^2/inch
d = fabric thickness – inches.

Some simplifying assumptions are made in deriving the equation, such as considering the seal material to be a thin membrane and thereby neglecting the effect that the bending stiffness of the material has on response time. However, the results can still be meaningful in evaluating seal response time for various-size obstacles.

e. **Seal Deflections.** When passing over a wave the SES bow seal buckles inward to form a series of triangular-shaped folds. The size of these folds depends to a large extent on the type of seal. Maximum flexural stress will occur at the apex of the buckle, while the tensile stresses induced by inertia and concentrated attachments (such as stays) are greatest along the lower edge of the sample.

Stern Seal Design

a. **Dynamic Analysis.** The primary purpose of conducting a dynamic analysis of a stern seal is to determine the effect of various seal design parameters on seal performance. To this end it is necessary to (1) develop a mathematical model and computer simulation of the base-line seal design, (2) establish the desired performance characteristics, and (3) evaluate the effect of seal parameters on system performance.

The base-line seal concept to be investigated must first be defined in sufficient detail that the equations of motion can be written. Vehicle and wave motion also must be considered early in the math modeling process, as the complexity of the problem is vastly affected by these decisions. For a complete description of vehicle motion (not including seal), six equations would be required, one for each of the six degrees of freedom of the vehicle. The mathematical description of the wave environment can also be simple or complex, depending on the objectives of the program. In a sophisticated program the random nature of waves might be considered; this is not always necessary, however, because the wave environment as represented by a simple sine-wave equation can provide much useful performance information about the seal.

The following dynamic analysis was performed on the rigid hinge beam design shown in Figure 9–3C and is presented here as an example of the type of analysis that can be performed (Reference 47). The analysis of this rigid-beam stern seal is broken down into two parts – on and off the wave. The wave form is assumed to be sinusoidal, with a fixed wave height-wave length ratio. The tip of the seal is considered to ride on top of the wave and not to be submerged beneath the water surface at any time. Under these assumptions, and considering the platform of the craft to have only one degree of freedom in the horizontal direction, the angular position and angular velocity of the seal can be determined from the geometrical relationships, as shown in Figure 9–8. Subscripts 1 and 2 will be used to designate parameter values at the beginning and end of a given time interval.

SEAL WAVE-FOLLOWING GEOMETRY

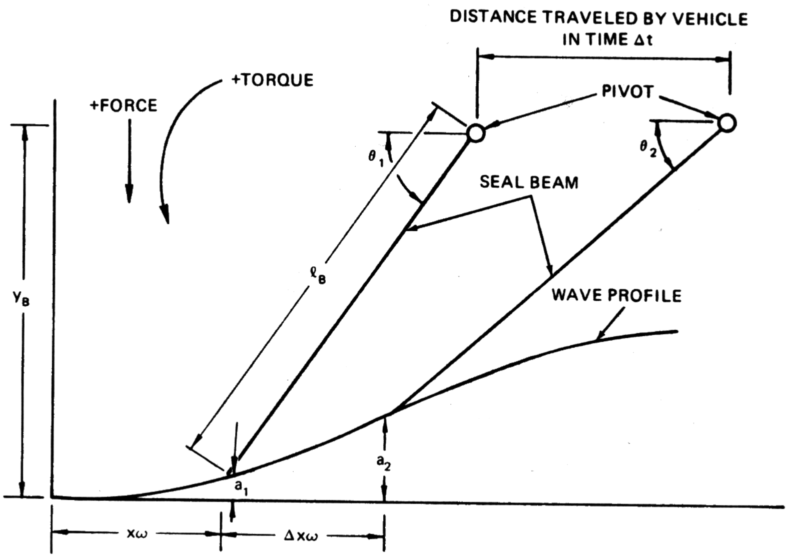


FIGURE 9–8

Newton's second law of motion, when applied to rotational systems can be expressed as

$$\Sigma \text{ applied moments} \approx \text{mass moments of inertia} \times \text{angular acceleration.}$$

This equation is expressed mathematically as:

$$\frac{I(\omega_2 - \omega_1)}{\Delta t} = \Sigma \textit{ applied moments}$$

where

I = seal mass moment of inertia about pivot – pound-second^2-inch
ω = angular velocity of seal – radians/second
t = time – seconds.

The forces and moments applied to the seal beam are indicated in Figure 9-9. The hydrodynamic tip force, T, created in passing over the waves can be expressed mathematically in terms of the remaining external moment, M_{EX}, and the inertia moment of the seal:

$$T = \frac{\left[\dfrac{I(\omega_2 - \omega_1}{\Delta t} + M_{EX}\right]}{\ell_B}$$

The M_{EX} term can be expanded mathematically as follows:

$$M_{EX} = B\int_0^{\ell_1} (P_H - P_C)\,dx + B\int_{\ell_1}^{\ell_2} (P_S - P_C)\,dx +$$

$$B\int_{\ell_2}^{\ell_3} (P_H - P_C)\,dx - B\int_{\ell_3}^{\ell_B} P_C\,dx +$$

$$B\int_0^{\ell_B} W\cos\theta\,dx$$

RIGID-BEAM SEAL SIMULATION LOADING

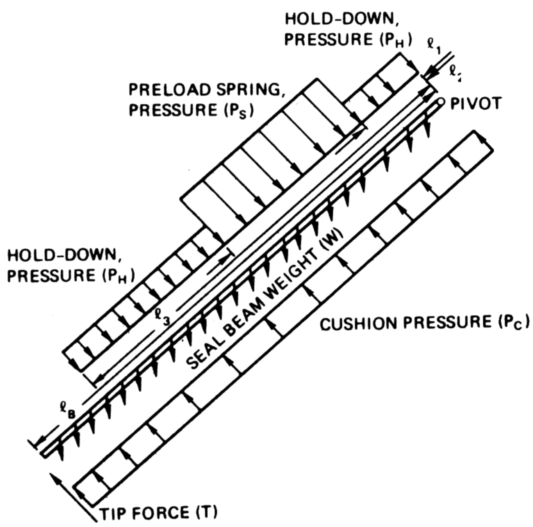


FIGURE 9–9

where

P_H = hold-down pressure – pounds/inch^2
P_S = preload spring pressure – pounds/inch^2
P_C = cushion pressure – pounds/inch^2
W = weight per unit width of seal – pounds
B = beam width – inches
ℓ = spring contact length – inches.

To evaluate the external pressure moments, additional equations are needed relating air bag geometry and volumes to seal position. These equations, while relatively complex, are basically geometrical expressions and are not presented here.

Pressure rises and flow rates also must be determined. Pressure rise, dp/dt, is determined as follows:

$$\frac{dp}{dt} = \frac{WR}{V}\frac{dT}{dt} + \frac{RT}{V}\frac{dW}{dt} - \frac{WRT}{V^2}\frac{dV}{dt}$$

where

P = pressure – pounds/inch^2
V = volume – feet^3
W = weight of gas – pounds
R = gas constant (53.3 for air) – foot-pound/pound-°R
T = temperature – °R
t = time – seconds.

The flow rates can be evaluated by using sonic and subsonic gas-flow equations.

These equations are then numerically integrated to determine the forces and displacement of the seal. The calculated tip force shown in Figure 9-9 is assumed to be the normal component of the resultant tip force. To determine the value of the resultant tip loads, a lift-drag ratio needs to be determined. As long as the seal is assumed to be in contact with the wave, only the normal component of the tip force need be known to determine required forces and displacements.

If the seal leaves the water in passing over the crest of a wave, the tip force is set equal to zero and a new seal position is predicted, based on previous inertia and moment data. New inertia and moment data are determined and the numerical routine continued as applicable to the entire period the seal remains above the wave.

b. **Pneumatic System Requirements.** Off-to-on cushion time, seal volume, and percent seal overpressure must be considered in establishing the pneumatic system requirements for a particular vehicle. Operational pressures can best be selected through computer simulation. Past experience has indicated, however, that seal pressure should exceed cushion pressure by 10 to 20 percent to provide the necessary seal response.

c. **Structural Analysis.** The results of the dynamic simulation provide the seal load information necessary to conducting a stress analysis of the seal system that includes such items as the air bag, seal beam, hinge and attachment structure, and down–stop provisions. Basic air bag fabric strength can be determined by the means previously discussed. As fabric stress is a function of both pressure and radius, maximum pressure stresses may not occur at maximum bag pressure because the bag radius decreases as the bag is compressed.

The seal beam analysis must take into account the maximum hydrodynamic, pressure, and the inertia loads resulting from the operational conditions under investigation. The method of analysis will depend to a large extent on the complexity and composition of the seal structure. Shear and moment diagrams should be generated for the critical loading conditions.

Loads transmitted to the structure will depend on the method of seal attachment and the contact area between the structure and the seal air bag. Intermittent hinge attachments to the structure will introduce concentrated loads, while continuous attachment of the seal membrane will induce a uniform loading to the structure.

Hinge and hard structure attachments to the vehicle should be analyzed to ensure proper design and a long service life. Fatigue and wear of the hinge should not be overlooked, as the loads applied to the structure could occur at a sufficiently high cyclic rate that the fatigue life of the material is rapidly approached. Conservative design in this area is felt to be warranted.

d. **Down–Stop and Retraction System.** Down stops are used to provide a mechanical limit to seal motion; they also provide a means of retracting the seal for emergency stopping and/or non–cushion operation. Power requirements for emergency retraction will therefore be dependent on seal size, operating pressure, and retraction time. Consideration must be given to such details as cable or strap load and to the effect that these concentrated loads have on the seal structure. If these loads are excessive for the structure, they can be distributed through an intermediate structure, or alternatively, the number of cables can be increased. Increasing the number of cables, however, requires a corresponding increase in retraction system hardware. There is thus a tradeoff between (a) adding structure to the seal and, (b) increasing the number of cables and, in turn, the complexity of the retraction system.

MATERIALS

Selection of Coated Fabrics. Since only a few SES's have been built to date, and most of these have been relatively small, low–speed units, little effort has been expended in developing materials for bow and stern seals. Of all materials tested thus far in service, nylon fabrics coated with a nitrile–PVC compound to a total weight of approximately 70 to 90 ounces/square yard have shown the best performance for SES's in the 100 ton size range.

A number of coated fabrics are commercially available, representing an almost unlimited number of different combinations of fabrics and rubber coatings. The relationships between coated fabric physical properties and expected seal life are currently being evaluated on laboratory flagellator test equipment. Certain properties of both the fabric and the coating compound tend to increase seal life. With respect to the rubber coating, seal life has generally been found to be directly related to flex life, dynamic resilience, and dynamic modulus and indirectly related to hardness. Similarly, with respect to the coated fabric, seal life appears to be directly related to tear strength, coating thickness, coating adhesion, total weight, and flexural stiffness.

At this stage of development, the material that appears to be best for seals is some type of a nylon fabric coated with a modified natural rubber compound. Oxypropylene, ethylene-propylene terpolymer, and neoprene compounds also appear to be attractive coating materials. There has not yet been a sufficient amount of evaluation of different configurations of fabric weaves to predict which might be most suitable for seals. Fabrics made from nylon appear to be best choice for the reinforcing fabric because nylon has a high strength–weight ratio and relatively high elongation that is related to increased coating adhesion retention during flexing, and because adhesive systems are available for achieving maximum adhesion between nylon and a variety of coating compounds.

The two most important environmental parameters affecting the service life of seal materials are the expected speed of the vehicle being designed and the temperature range in which it is expected to be operational. Little information is available on the service life to be expected from materials operating in cold environments. There is, however, some limited amount of laboratory data available on the effect of vehicle speed on the life of seal materials at normal operating temperatures. Figure 9–10 is a plot of service life versus simulated vehicle velocity for three different rubberized fabric materials, as determined with a laboratory flagellator. These data were obtained by subjecting small samples of rubberized fabric to a waterjet at three impingement velocities. All of the samples were approximately 0.090–inch thick. The square–woven nylon fabric used with the natural rubber and nitrile–PVC coated materials weighed 18 ounces/yard^2, while that used in the neoprene samples weighed 12 ounces/yard^2. Since few SES vehicles have been operated in this country, there is not enough actual service life data available to make a correlation between actual and flagellator performance. Therefore, any reporting of service life performance other than the relative performance reported in Figure 9–10 would be misleading.

RELATIVE SERVICE LIFE OF COATED FABRICS FOR SEALS

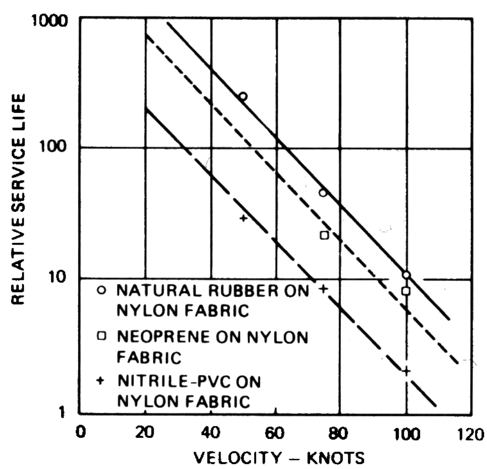


FIGURE 9–10

Other environmental factors might be considered in selecting seal materials if they appear to be severe enough to limit seal life. Such factors might be unusually high temperatures and the presence of foreign materials in the service environment, such as oil or debris resulting from unusual mission requirements. In most cases, seal materials will have to be selected on the basis of expected vehicle speed and temperature of operation.

Manufacture and Repair. Coated fabrics for seals are processed in the factory under conventional rubber industry practices. Coating compounds are mixed on a rubber mill or in an internal mixer, and fabrics are pretreated with dip–coated adhesives. The adhesive–dipped fabric is then coated, usually with the selected elastomer by a process called calendering. In this process the mixed coating compound is sheeted out between two metal rolls and then squeezed into the fabric as it passes the second and third rolls of the calender unit.

The resultant coating fabric can be cured in either of two ways: it can be cured in large rolls prior to seal fabrication or the seal components can be cut from uncured material and cured individually prior to final seal fabrication. The former process is the more common, but seal components are sometimes cut from uncured fabric and the cut edge sealed with a rubber strip before the section is cured. The seal components are then assembled into the finished seal, using cemented and sewn seams as well as lacings through grommets and riveted attachment plates.

Damaged SES seals can be emergency–repaired in the field, but drydocking will probably be required for more permanent repairs. In some designs employed with smaller craft, simple attachment methods may permit dockside in–the–water removal for repair or replacement.

Emergency repairs may be made by rough sewing, lacing, riveting, or bolting (with or without patches) as the particular condition requires. When major tears in the reinforcing fabric of the seal make drydocking necessary, it will probably be more economical in the long run to replace rather than repair a damaged component. Small permanent repairs can be made by using hot-patch techniques, but such repairs should be made by experienced personnel to ensure high adhesion levels. Surface preparation includes cleaning the repair area, followed by buffing both the patch and the repair area. Adhesive cements are then applied to both surfaces and the repair area clamped between heated plates for curing purposes.

PARAMETRIC DATA

The weight of SES seals depends significantly on (1) ship configuration – for example, cushion beam, plenum chamber height, sidehull length, and ship planform geometric shape; (2) type of seals, (3) seal materials, and (4) method of attachment. Factors having indirect effects on seal weight are (1) cushion pressure and bag pressure, which in turn affect SES performance relative to both calm–water and seaway conditions and (2) ship operational speed, which affects the service life of the seals. At the present time, a simple method for predicting seal weight is not available. For advanced design guidance, Figure 9–11 indicates bow and stern seal projected area and weight. The data presented in this figure are calculated on the basis of the following assumptions:

Cushion length – cushion beam ratio	= 2.0
Cushion pressure – cushion length ratio	= 1.5
Height of plenum chamber	= 14 feet for ships of 1,000 tons
	= 20 feet for ships > 2,000 tons
Semi-circle planform shape at bow, with 50° bow seal angle.	
Seal material	= 100 ounce/yard^2 coated fabrics.

PARAMETRIC WEIGHT OF SEALS

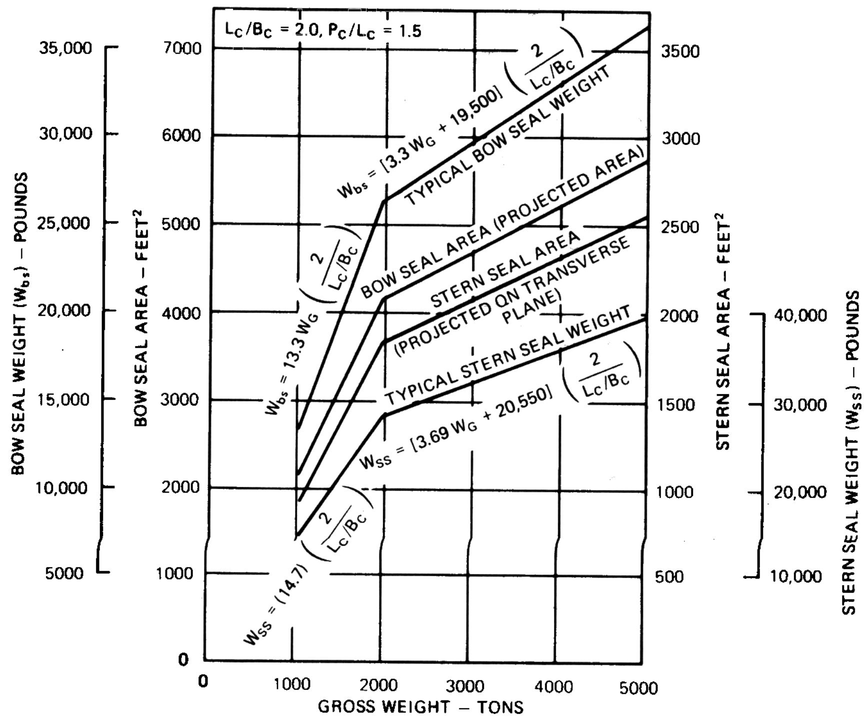


FIGURE 9-11

Chapter X
AUXILIARIES

The primary auxiliary systems on an SES are the control station, fuel, personnel, electrical, hydraulics, bilge and ballast, fire protection, countermeasure washdown, deicing, aircraft facilities, stores handling and refueling at sea, boat handling and lifesaving, anchoring, towing and mooring and damage control systems. The auxiliary systems–gross weight ratio is approximately 5 percent.

CONTROL STATION (Navy Weight Group 4)

In the deisgn of the SES control station (bridge), the basic consideration is the speed capability of the SES. Ship control must be fast and efficient as opposed to the slower methods of control traditionally used. The bridge is therefore designed to provide for direct helm control by the officer of the deck (OOD), relief helm control by the junior officer of the deck (JOOD), centralization of ship's status information at a commanding officer's (CO) console, and a central engineering operating station.

Two views of a typical arrangement of the bridge are shown in Figures 10–1 and 10–2. Descriptions of the functions and equipment of the various stations follow.

Helm (Officer of the Deck's Station). The OOD is the officer on watch who has been designated by the commanding officer to be in charge of the ship. His primary responsibility is to ensure the safe and proper operation of the ship, including safe steering, speed, and course keeping; control of propulsion and lift systems, stability of the craft; and monitoring of engineer's functions during start up to ensure that all systems are ready. The equipment needs essential to performing these functions are as follows: (1) lift and propulsion controls mounted between helm and relief helm stations; (2) a helm panel containing speed and direction, trim, steering control, and rudder angle indicators, depth finder, and autopilot; (3) communications equipment, and (4) an engineering status board for monitoring the performance of vital systems.

Relief Helm/Assistant Navigator (Junior Officer of the Deck's Station). The JOOD is primarily responsible for collision avoidance, visual navigation, ship identification, and relay of navigational information from the combat information center. He also can assume the helm during emergency conditions and back up the OOD in course-keeping operations. The equipment needed for these functions is as follows: (1) lift and propulsion controls shared with the helm; (2) a helm panel identical to the one at the primary helm; (3) communications equipment independent of that at the helm; (4) a collision avoidance panel that includes radar repeater and computerized calculation and read–out of ship and target course, speed, and closest point of approach; and (5) a navigation panel equipped with navigational–lighting array switches.

Commanding Officer's Console. This console is provided to permit the CO to oversee the operation of the ship. Manning this station does not require the performance of any discrete function, and, accordingly, this station need not be manned. The items of basic equipment

BRIDGE DESIGN FORWARD/STARBOARD VIEW

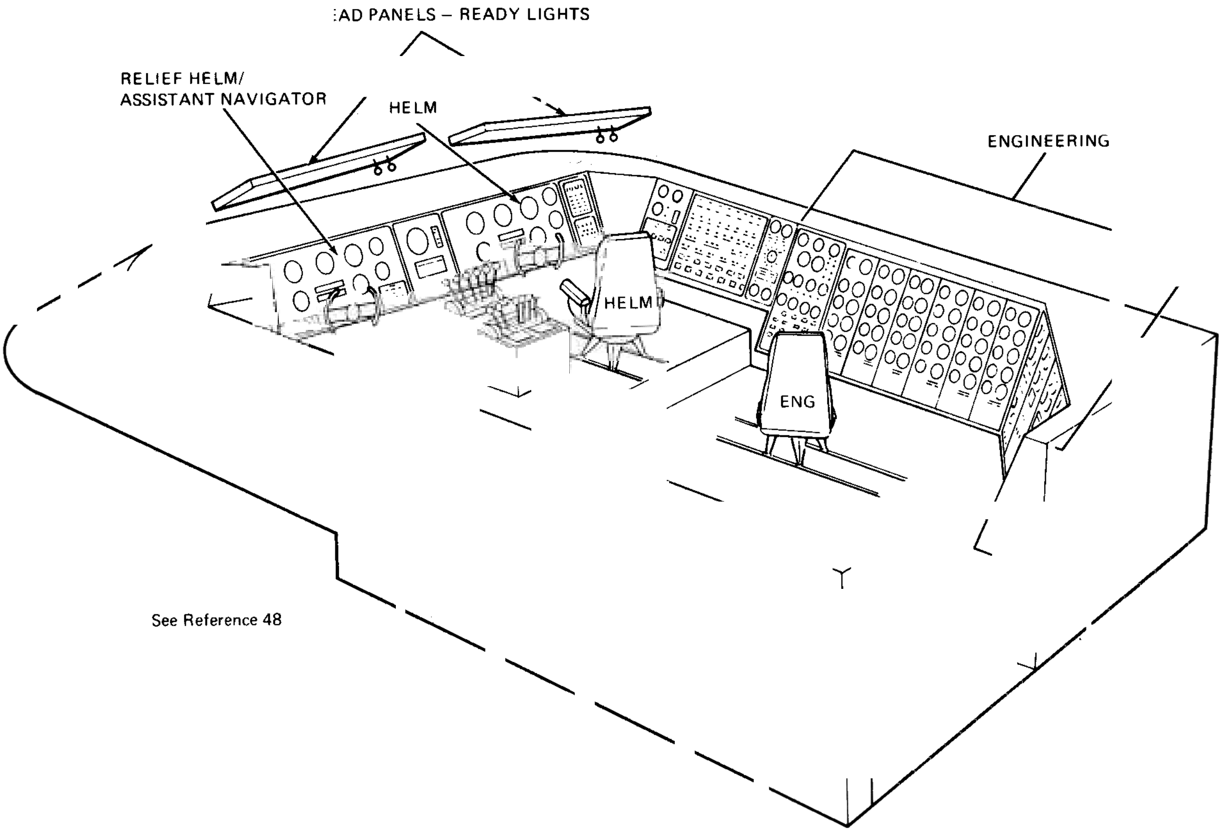


FIGURE 10-1

provided at this station are: (1) radar/sonar display, indicating target course and speed; (2) damage control display, indicating nature, location, and extent of damage; (3) navigational data display, indicating time/distance to destination and latitude/longitude of destination and of own ship; (4) environmental display, indicating wind speed and direction and outside temperature; and (5) communications equipment.

Engineer's Station. The engineer is responsible for the ship's mechanical plant, including the propulsion and lift systems and auxiliary systems, the ship–service electrical plant, and the damage control system. The equipment in the engineer's station should be designed to provide for command inputs, monitoring, and malfunction detection. Through the use of readily programmable computers controlling both system functioning and monitoring, automatic engine start–up, and normal operations over the whole speed range (both on and off–cushion), provision can be made in the station for close–in maneuvering in port, plant standby, and automatic shutdown with minimum operator and engineering attention and with optimum power splits between lift and propulsion, trim angles, and lift system louver control. A data logger can also be employed to collect the information monitored by the various ship systems, which can then be printed out as an on–line display of the ship systems' condition or stored for future use in a shore–based maintenance analyzer to predict and schedule maintenance overhauls for the systems involved.

BRIDGE DESIGN: FORWARD/PORT VIEW

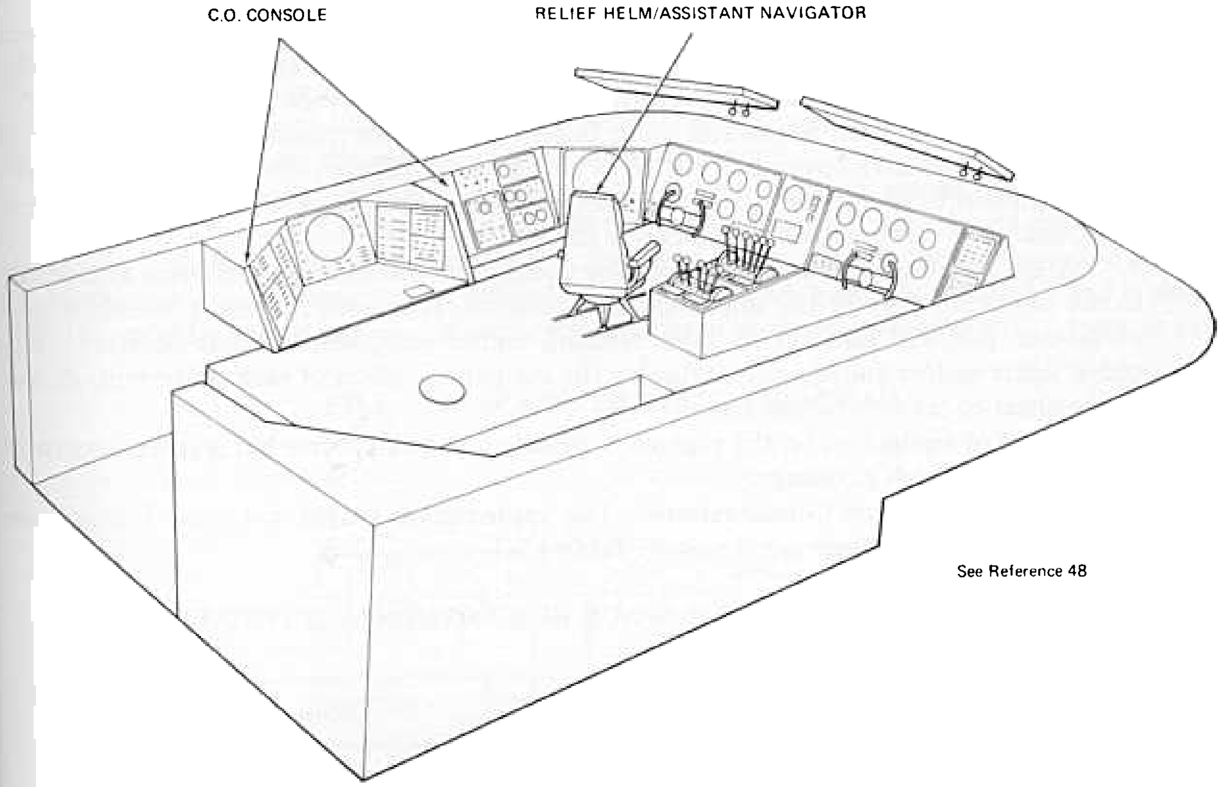


FIGURE 10–2

To provide these engineering controls, the following minimum equipment panels should be in the engineer's station. Each panel should include a facsimile, or mimic, of the ship system it controls and monitors and the appropriate malfunction alert lights and buzzers such that the engineer can scan the individual panels and, by noticing color-coded signal lights, quickly determine the ship system's status and take corrective action as needed. The equipment panels in point are as follows:

1. An automatic Engine Start-up Command and Monitoring Panel that provides master switches for engine starting, condition signals, and, through a central control computer, sequences starting, including the bypassing of engines exhibiting faulty starting.
2. A Propulsion Mimic and Fault Isolation Panel that controls and monitors propulsion system parameters, such as RPM, thrust/torque, fuel flow, turbine inlet and exhaust temperature, lube oil pressure, and air inlet filter pressure drop. The central computer interrogates each operating engine and periodically displays the proper color-coded light signal on the panel.
3. A Lift System Panel that provides the same basic information as the propulsion panel for the functioning of the lift system prime movers and also controls and monitors such lift system elements as fan operation, cushion pressure, leakage rates, seal conditions, and cushion air louver operation.
4. An Auxiliary Mimic and Fault Isolation Panel that controls and monitors auxiliary parameters of the ship's power plant, such as fuel level and delivery pressure, propulsor shaft

speed, propeller pitch, waterjet nozzle position, vibrations, ship control surface position, actuator angle, and transmission component temperatures.

5. A Ship Service Panel that controls and monitors the status of the various ship's service subsystems, such as freshwater, bilge, anchoring and mooring, and ventilating/air conditioning, including such components as motors, fans, compressors, pumps, and valves.
6. An Electric System Mimic and Fault Isolation Panel that monitors the ship's electrical plant and the auxiliary power unit (APU) as well if one is installed aboard ship. The following electrical subsystems are included: generators, lights, navigation equipment, communications, and hotel power system for dockside use.
7. A Damage Control Panel that provides a mimic of the various subsystems appropriate to the inherent safety of the ship, such as ventilation, firemains, fire doors, flotation compartments, fuel and lube system lines, steering control subsystems. Fault-isolation color-coded lights and/or audible signals display the current condition of each subsystem, alerting the engineer to take corrective action.

The above list of equipment on the engineer's console is not exhaustive but is representative of the basic elements in such a concept.

Overall Weight and Space Considerations. The approximate weight and space requirements for the control station (bridge) are shown in Table 10-1.

BRIDGE WEIGHT AND SPACE REQUIREMENTS (TYPICAL

Item	Weight (pounds)
Helm	400
Relief Helm/Assistant Navigator	900
Commanding Officer's Console	300
Engineer's Station	,100
Cable	400
Total	3,100

Item	Dimensions (feet)
Overall Deck Space, including Consoles	12 x 15
Deck Space, between Consoles	9 x 11

See Reference 48

TABLE 10-1

FUEL (Navy Weight Group 5)

The fuel system provides for the storage, handling, and purification of distillate fuel for the propulsion, lift, and auxiliary system prime movers.

Fill and Transfer. The fill and transfer portion of the fuel system consists of the piping, valves, pumps, and tanks required to bring the fuel on board and store it in such a way as to maintain the SES in proper trim. The prime candidate location for fuel storage is the relatively narrow, deep sidehulls, which do not easily lend themselves to other uses. As fuel is usually taken on board on only one side of the SES, the piping must be sized to transfer fuel from one side of the ship to the other at high replenishment rates (180,000 gallons per hour through a 7-inch hose). The large fuel requirements necessitate high fueling rates.

A basic diagrammatic arrangement of a fuel oil fill and transfer system is shown in Figure 10-3. The cross-connect piping is used to fill the port tanks from the starboard side and should be at least 6 inches in diameter. In a high purity JP-5 supply system, this piping would have to be of stainless steel or copper nickel material. PVC piping is adequate for the service and is about half the weight of metal piping, but no approval has been given for its use for this purpose as yet.

FILL AND TRANSFER SYSTEM DIAGRAM

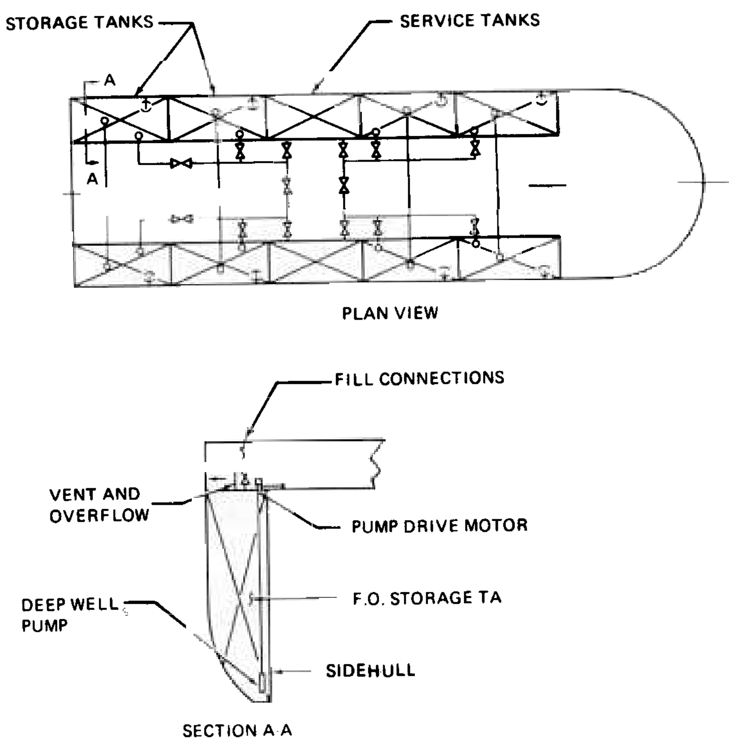


FIGURE 10-3

Vents and overflows are required on each tank. The vents extend up to about the weather deck, with a U-turn and a check valve incorporated to prevent ingestion of water into the tank. To comply with the latest anti-pollution requirements, an overflow tank should be provided. The pumps used to transfer fuel from the storage tanks to the service tanks can be submersible types located in the deep tanks, deep-well types with the impellers in the tanks and the drives installed on top, or centrifugal pumps located in trunks within the sidehulls. The trunks permit the suction lift of the centrifugal pump to be maintained below its 30-foot limit. Another possible transfer system design incorporates air pressure to move the fuel from one tank to another. In this case the tanks would have to be capable of pressurization. System pressure would be a function of the height of the fuel tank and of the pressure drop through the system. Gas turbine bleed air could be the air supply source.

Each tank also requires an indicating system with high- and low-level alarms. Lightweight electrical systems operating on pressure differentials are available as standard marine equipment.

The operational sequence is to pump fuel from the forward and aft tanks into the service tanks, located at midship. A series of valves permits pumping from port to starboard, and the addition of a few bypass valves would permit the system to shift fuel forward and aft as well. In this way the fuel system also serves to maintain the SES trim and heel and reduces the size of the seawater ballast system.

Stripping. Tank stripping may be accomplished by either a portable or a fixed system. The choice between systems depends on the frequency of refueling. In a portable system submersible pumps are employed periodically to clean the tank bottoms of sediment-loaded fuel. The weight difference between a fixed and portable system is small.

Service. The service portion of the system, as illustrated in Figure 10–4, consists of the piping, valves, pumps, heaters, purifying equipment, and tanks necessary to remove any accumulation of water or sediment that enters the fuel during transfer or while on board to heat the fuel to proper operating viscosity, and to deliver it under pressure to the engine.

SERVICE SYSTEM DIAGRAM

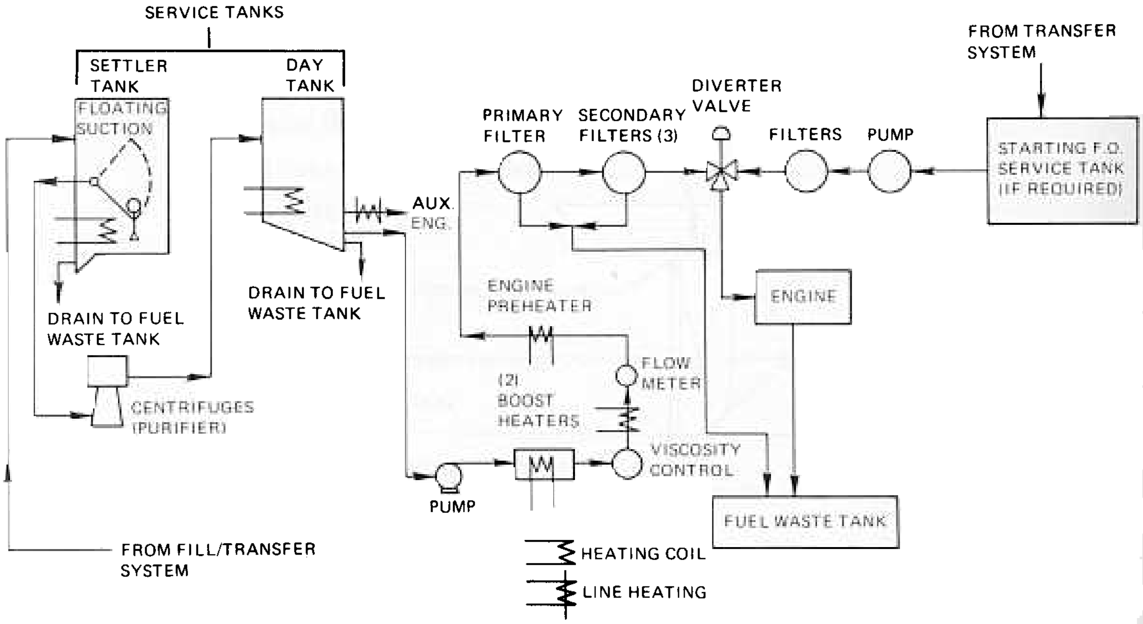


FIGURE 10–4

Both a starting and a running subsystem are necessary for those engines requiring a light distillate fuel for start-up and shutdown. If the engine is capable of starting on a heated medium distillate, however, a starting subsystem is not required. If aircraft-purity JP-5 can be delivered, a simple system similar to the starting subsystem becomes the entire service system.

Maximum recommended viscosity of the fuel at the engine is 10 centistokes for cold starting and 4 centistokes for running (to minimize smoking). JP-5 has a kinematic viscosity of 10 centistokes at about -20°F. It is possible that the sidehulls above the waterline and the fuel in them could be exposed to temperatures as low as -40°F. Figure 10-5 shows the effect of temperature on various fuels used in the marine environment.

EFFECT OF TEMPERATURE ON FUEL VISCOSITY

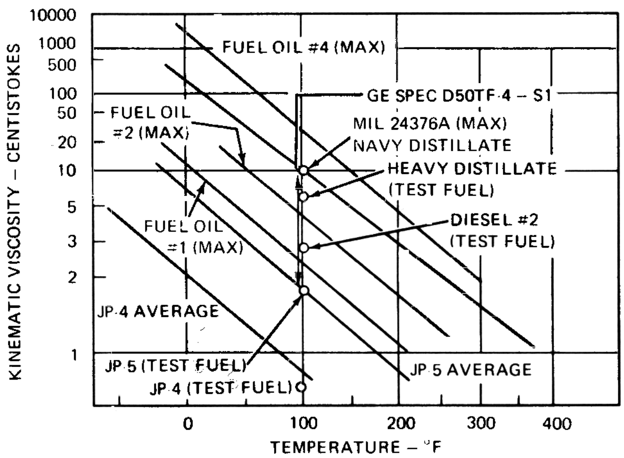


FIGURE 10-5

Overall Weight and Power Considerations. The heaviest fuel that will probably be considered for the SES is Navy Distillate (ND) MIL-24376A, which has a viscosity of 200 centistokes at 0°F. This is well below the pumping limit of 1500 centistokes; thus, no heating will be required in the storage tanks unless it becomes more economical from a weight standpoint to have the fuel partially heated in the storage tanks instead of adding all the heat as it flows out of the tanks to the filters and turbines.

The amount of power required to pump fuel from the tanks can be determined from the following relationship (which assumes laminar flow):

$$(HP)_p = 89 \times 10^{-3} \frac{\nu S Q^2 L}{d^4} + \frac{Q \rho h}{33{,}000}$$

where

$(HP)_p$ = pumping power – hp

Q = flow rate – foot^3/minute

ν = kinematic viscosity – centistokes

S = specific gravity

ρ = density – pounds/foot^3

d = internal diameter of pipe – inches

L = length of piping

h = head – feet of fuel oil (bottom of tank to discharge level)

Specific gravity and density for two typical fuels are shown in Table 10-2.

SPECIFIC GRAVITY AND DENSITY OF TWO TYPICAL FUELS

	JP-5	Navy Distillate
Specific gravity, S	0.62	0.87
Density, (pounds/foot^3), ρ	39	54

TABLE 10–2

Velocity limits in fuel oil transfer are 6 feet per second (fps) for suction piping and 15 fps for discharge piping. For the service fuel piping, these values are 4 fps and 6 fps, respectively.

Fuel is pumped from the storage tanks to the settling tank, which is equipped with a floating suction system and a drainage system to remove accumulated water and sediment. The storage tank may be designed to act as a settling tank as well. Some heating by steam or electric coils may take place here to bring the fuel above the cloud point prior to centrifuging. After centrifuging, the purified fuel (99.5% water removal by volume) is pumped into the day tank, which is best located above the engines to provide a gravity supply in case of an emergency. Heat is also applied here, if required, to bring the fuel to proper engine inlet viscosity, and a booster pump provides the fuel with a positive head (12 psig min.) to the engine. A series of filters removes impurities down to 20 microns before the fuel enters the engine-mounted system.

The maximum temperature the fuel is permitted to reach in any of these heating phases is 150°F, which is close to the flash point. The electrical requirements for heaters in this system are given in Table 10–3.

FUEL SYSTEM ELECTRICAL HEATING REQUIREMENTS

Heater	Power (kilowatts)	Flow Rate (gallons per minute)	Temperature, Δ T (°F)
Tank	15	35	35 to 120
Booster	15	35	90 to 150
Preheater	100	35	60 to 100

TABLE 10–3

The fuel flow requirement for two typical marine gas turbines, the LM 2500 and the FT 4, is about 25 to 35 gallons per minute. System pressure should not exceed 75 pounds per square inch. To maintain the purity of the fuel, it is important to use such materials as stainless steel and copper nickel (maximum 70% copper) for the piping and stainless steel, aluminum, and coated steel for the tanks.

For initial estimating purposes the specific weight of a Navy Distillate system consisting of piping, heaters, purifiers, filters, and pumps can be taken to be about 0.40 pound per horsepower. If aircraft purity JP-5 can be delivered there is no requirement for purifiers and filters or for a heating system, and the specific weight would be about 0.15 pound per horsepower. If the JP-5 system requires that purifiers and filters be installed, the specific weight would be about 0.25 pound per horsepower. For advanced design purposes assume the 0.25 pound per horsepower system installation.

PERSONNEL

A high standard of personnel services consistent with reliability, low power consumption, and low system weight is especially desirable in the SES. For an advanced design, it is important to have a clear idea of a concept to be followed as well as the implications of that concept in terms of weight, space and power requirements. Described below are recommended concepts for the various personnel systems, with their attendant rationales.

Accommodations (Navy Weight Group 6). In the area of crew accommodations it is desirable to afford the crewmen some degree of privacy and comfort, as well as the opportunity for group activities. This need might be met by providing bunks arranged one above the other equipped with individual curtains, lighting, ventilation, and stereo entertainment. The bunks can be arranged head to head, for simplification of connections, in carpeted compartments to reduce noise. Ship issue of personal supplies and clothing would reduce stowage space requirements as well as help control the generation of such solid refuse as paper products and containers. Further reduction in personal stowage space could be achieved by central stowage of dress uniforms and ship issue of jump-suit-style duty uniforms of synthetic permapress fabric to reduce laundry requirements. Provision should also be made for an equipped exercise room and a lounge with audio visual facilities and games.

Food Service (Navy Weight Group 6 and Variable Load). Of the many possible food service plans, the three most likely to be used are: conventional or fresh food provisioning; frozen pre-prepared meals; and freeze-dried foods. A reduction in manning and weight can be achieved by incorporating a degree of self-service and the use of disposable service items. An unknown factor is crew acceptance of either the frozen, pre-prepared or the freeze-dried foods. Although the freeze-dried foods are the lightest in weight, and require the smallest storage volume, the approach may be too extreme to be acceptable. Frozen, pre-prepared meals offer a significant weight-saving over conventional practice and should be acceptable if the meals are well prepared and palatable. A mix of frozen and freeze-dried food might be tried along with crew participation in menu selection. This could be easily accomplished through computerized inventory control. Dining facilities should be designed to be of minimum weight consistent with accessibility, comfort, and multiple use.

Table 10-4 gives a comparison of the storage and preparation requirements of the three plans. This table does not include the weight savings possible with the incorporation of self-service features and disposable service items. The weight-savings attributable to elimination of the need for mess personnel (possible with the use of frozen or freeze-dried foods) has also not been included.

Medical Service (Navy Weight Group 6). Because of the capability for medical evacuation, shipboard medical service requirements can be reduced to treatment of minor illness and injury, pain relief, and stabilization of more serious cases prior to evacuation. Thus, medical personnel requirements can be reduced to one Independent Hospital Corpsman. Under this

ALTERNATIVE FOOD SERVICE PLANS: WEIGHT AND SPACE REQUIREMENTS
(Based on 40-Day Supply, 15% Reserve, 85-Man Crew)

	Storage (per man)			Preparation		
	Weight (pounds)	Volume (foot^3)	Power (kilowatts)	Weight (pounds)	Volume (foot^3)	Power (kilowatts)
Freeze dry	77.2	0.895	0	477	12	18.2
Frozen	255.0	22.42	0.040	1500	48	35.0
Fresh	323.0	27.6	0.035	4432	264	54.5

See Reference 49

TABLE 10-4

arrangement his duties would include hygenic inspection, water testing, and medical inventory control. In emergencies he could be assisted by crewmen trained in first aid. He could also make use of the shipboard computer system as programmed for medical inventory and records and a medical diagnostics program.

Facilities necessary would be a dispensary and several battle-dressing stations, each equipped to meet expected medical requirements.

Hygenic Service (Navy Weight Group 5). The collection system employed with the hygenic facilities can be gravity, positive pressure, or vacuum. The vacuum collection system is recommended, as it avoids the piping-slope requirements of a gravity system and the pump requirements of the positive pressure system. A sufficient number of traps and check valves must be included to maintain waste flow. The facilities themselves should include lightweight, low water-usage fixtures. Selection of a particular unit should be based on unit impact on the water and waste management system. For example, recirculating toilets require less water than low-flush vacuum toilets, but their discharge is not suitable for aerobic digestion. Although vacuum toilets require more water and discharge more waste than recirculating toilets, they require only some 3 1/2 pints of water per flush rather than the 3 to 5 gallons required in a conventional toilet. Vacuum toilets presently have vitreous china bowls, but lighter weight can be obtained by using steel or plastic bowls.

In the shower and lavatory areas, in addition to using lightweight fixtures, dispensable bath and hand towels can be provided. This innovation, plus laundry pickup or disposal in those areas will reduce crew storage requirements, decrease the number and weight of these items, and help provide efficient laundry operation.

Laundry Service (Navy Weight Group 6). Efficient laundry operation can significantly reduce the requirements of the water and waste systems and reduce power usage as well. For larger crew sizes, laundry needs can be best met by having the laundry facility operated by crew members who could also maintain ship's stores and issue supplies. A separate laundry for the officers might also be desirable.

Under this arrangement laundry items could be picked up and delivered by the crewmen. This would allow the laundry equipment to operate on complete loads, as well as centralize the supply operation. Ship issue of jump-suit uniforms of permapress fabric would reduce hot water needs, eliminate ironing, and reduce laundry time and stowage requirements. The weight of the system and the water requirements involved would depend on the type of units selected. Presently available laundry equipment can be redesigned to reduce weight, particularly if 400 Hz electrical power is utilized.

Information Systems (Navy Weight Group 4). Information requirements for the personnel-related subsystems fall into three categories: monitoring, storage and retrieval, and specialized functions. Direct monitoring is necessary in all cases where systems normally operate unattended and their continued operation is important. These include environmental control, water, waste management, fire detection, and watertight door status. These functions should be monitored from the bridge, using a visual display to indicate system status.

Information storage and retrieval should be incorporated to achieve weight and space economy in record-keeping. Access would probably be required only once a day for updating inventories related to food and medical services and ship's stores and for maintenance records. Information could be entered by utilizing either pre-punched cards or a teletype console when the equipment is not being used for other purposes. An additional terminal facility should be included to give the hospital corpsman ready access to patient records or use of the diagnostics program. Either this facility or a bridge facility could be used for updating other records.

Specialized functions include interior communications and crew entertainment. To save weight, the interior communication system can be designed on a multiplex basis, and the crew entertainment system can be similar to that used on commercial aircraft.

Environmental Control (Navy Weight Group 5). An air conditioning system consisting of an electric-motor-driven vapor cycle system with a chilled water distribution system can provide a shirt-sleeve environment with low weight and minimum fuel penalties. For the air conditioning plant, light-weight aircraft-type components can provide sufficient reliability and capacity to meet requirements. An effort should be made to discharge electronic waste heat directly to ambient air, cabin exhaust air, or seawater, as this arrangement considerably reduces the size and power requirements of the mechanical refrigeration system.

An alternative to employing a "fan-coil" type distribution system is to distribute self-contained packaged units throughout the ship. While self-contained units require no chilled-water distribution system, they do require a saltwater supply. However, if a firemain system is incorporated in the ship design, the weight penalty associated with the additional saltwater piping needed for the air conditioning system will be small. The saltwater supply can in each case be taken from the nearest firemain.

Water Service (Navy Weight Group 5). Desalination of seawater is the best method of meeting the freshwater requirement of 30 gallons per man day. Of the possible processes, reverse osmosis appears the most promising. Reverse osmosis technology (see Reference 49) is advancing at a pace sufficient to warrant its consideration. It is a demand system that can respond quickly to changes in need.

Two complete systems per ship are recommended, each capable of producing the ship's total water requirement each day. Thus, in case of shutdown of one unit through some malfunction or for servicing, the remaining system can meet total water demands. Provision should be made for the alternate unit to initiate operation automatically in the event of shutdown of the other unit.

As compared with distillation or electrodialysis, a reverse osmosis system is lighter, requires less power and less tankage, and is inherently simpler to maintain or repair.

Waste Management (Navy Weight Group 5). Recent increased awareness of pollution in the total environment has imposed strict performance criteria on waste management systems. Raw liquid waste streams formerly discharged directly to the ocean must either be treated to a high degree of purity before discharge to the sea or be stored or incinerated aboard ship. Similarly, the discharge of solid wastes from a ship is no longer acceptable other than in emergencies. Incinerator stack gases must be clean, odorless, and smoke-free.

Equipment design for waste management practices meeting environmental criteria is particularly challenging in view of SES constraints on weight and volume of installed components and stored materials. Disposal of the many forms and types of waste generated aboard the SES thus presents the most difficult problem of all in providing personnel service systems. Solid waste generation complicates the problem of disposal. The most effective way to deal with this complication is to prevent materials that will present a later disposal problem from being brought aboard at the start of the mission.

Waste water can be treated and the purified liquid returned safely to the sea. The waste solid materials from the waste water treatment process become additive to the solid waste disposal problem in that they cannot be disposed of at sea. Sewage wastes present an equal problem in that liquids may be treated and returned to the sea but solids cannot. It is recommended that the waste streams be mixed to provide a liquid stream content including: (1) waste water, (2) liquid human waste, and (3) solid waste pulping water and a solid waste content of (a) waste water treatment residue, (b) solid human waste, and pulped solid waste. The liquid stream may be treated through a reverse osmosis system to a condition safe for disposal at sea. The residual sludge can be incinerated. The residual concentrated brine can be re-introduced into the waste stream or incinerated.

Stores (Navy Weight Group 6 and Variable Load). Stores should be brought aboard in containers designed to minimize solid waste generation. Inventory records should be computerized to simplify storeskeeping and to allow the storeskeepers to manage the laundry operation as well.

Emergencies. Major emergencies aboard the SES will probably occur with little warning, making it essential that the protective systems provided be not only adequate but also readily accessible for immediate use. The configuration and arrangement of the eventual SES as ultimately designed will dictate the quantity, type, and deployment of emergency equipment necessary for fire suppression and for emergency egress from the ship.

Commercially available fire suppression materials, such as Halon 1301, dry chemicals, carbon dioxide, and water with an added foaming agent should be installed in an appropriate system incorporating bridge alarms; small portable fire extinguishers should be readily available throughout the ship. Design of escape routes and facilities should take into account ship dynamics as well as the possibility that the ship may be in the hullborne mode. Lightweight liferafts with appropriate survival gear should also be installed. The weights associated with emergency equipment have been included in the appropriate subsystems.

Overall Weight and Power Considerations. Figure 10-6 indicates weight and power requirements for the lightweight personnel systems described above, including the necessary plumbing, ducting, and thermal and acoustic insulation. Weight also includes stores for 40 days with a 15% reserve.

ELECTRICAL (Navy Weight Group 3)

The electrical system includes generating equipment, switchboards for generator control and power distribution, distribution panels, transformers, and bus transfer equipment for providing proper power and protection relative to electric loads. Electrical power is vital to all shipboard operations and to the safety and comfort of the crew.

Power Generation. Electrical power continuity is the primary concern in electrical system design, and careful consideration must thus be given in equipment selection to the number,

PERSONNEL SYSTEM: WEIGHT AND POWER REQUIREMENTS

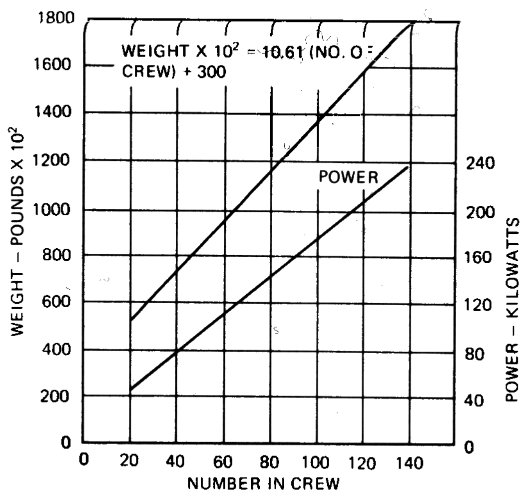


FIGURE 10–6

size, and location of generators and switchboards to provide adequate redundancy and isolation in case of damage. As a minimum, 50% generator–switchgear redundancy should be provided. Minimum weight is also an essential factor in equipment selection.

The prime mover for the generator may be chosen from among lightweight, medium weight and standard marine diesel engines; regenerative gas turbines; aircraft derivative gas turbines; and aircraft derivative gas turbines with waste heat utilization. The generator may also be driven by the propulsion or lift–fan engines if this approach is compatible with the ship operating mode and the level of required electrical power.

The standard diesel will have the heaviest installed weight; and, although it has a low specific fuel consumption (0.35 pound per hp–hour), it will probably not be competitive for missions of less than 20 days without refueling. The regenerative gas turbine becomes competitive with the lightweight diesel for short missions of about 3 or 4 days. The aircraft derivative gas turbines in the low power ranges will not be competitive because of the associated high specific fuel consumption (0.62 pound per hp–hour). The aircraft derivative gas turbine with waste heat utilization will probably not be competitive because of its high installed weight.

For most missions the choice will be between the lightweight and medium weight diesels. Lightweight diesels, however, are characterized by a short mean time between overhauls (1500 to 3000 hours), while a medium weight diesel will offer at least 10,000 hours of operation between overhauls. Consideration should be given to the possibility of selecting a diesel engine of sufficient capacity to provide direct drive for the main hydraulic pumps as well as the electrical generators.

Standard marine electrical power generation is at 60 Hz, 450 volts. Any system frequency higher than the standard 60 Hz increases speed of rotation and thus reduces size and weight. Since interior communication, electronic, and ordnance systems require 400 Hz power supply, and since considerable progress has been made on 400 Hz systems in the aircraft industry, this frequency is considered a logical choice in the interests of weight reduction. One of the areas of concern in substituting aircraft electrical for marine electrical equipment is the lack of knowledge of performance in the marine environment. Solutions thus need to be developed to such potential problems as sulfidation and seawater corrosion, as well as performance degradation due to shipboard shock and vibration. Until this is done, and until motors (and matching pumps) are made available in a variety of ratings, the 60 Hz marine equipment will continue to be used.

Increasing the voltage to greater than 450 volts will have a negligible effect on weight. Generator and motor weights do not change significantly with voltage. Increasing voltage will reduce cable size, but on an SES the minimum cable size will be governed by mechanical strength requirements rather than by the electrical load-carrying capacity at the increased voltage. A 450-volt system represents a good balance between load-carrying and mechanical requirements for ships in the SES size range.

The electric power plant is controlled under normal operating conditions by an electric control console located in an engineering control center. All the controls, visual indicators, and alarms necessary for operation and monitoring of the entire electric plant are contained in this console. A mimic bus with indicator and control switches provides a continuous pictorial presentation of system setup conditions. Audible and visual indications of system operating conditions should be provided to the engineering station in the bridge. The instrumentation and controls associated with each generator switchboard should be sufficient to permit local operation during emergency conditions or maintenance periods.

Switchboards are located in the compartments with their associated generators and are of the dead-front type, with aluminum structure for lightness. Switchboards provide the means for controlling, protecting, and paralleling the generators and for controlling and protecting the ship's service and emergency power distribution system.

Power Distribution. Ship loads are supplied either individually or in groups by feeder cables from the switchboards. To supply large loads, individual feeders are normally used; grouped loads are supplied by feeders through distribution panels. Distribution loads, vital or non-vital, are balanced among the switchboards of the system. Loads requiring power continuity are provided with alternate sources of power and can be transferred from one power source to the other either manually or automatically. A casualty power distribution system is available for bridging damaged sections of the ship's service power system. This system consists of through-bulkhead and through-deck terminals and portable cables located strategically throughout the ship.

The ship's lighting system is a 60 Hz, 120-volt system comprising lighting distribution, 120-volt ship's service, emergency lighting, and isolated receptacle circuits. The lighting distribution system is similar to the power distribution system in consisting of feeders connecting ship service/emergency switchboards to distribution panels or fuse boxes, located at central distribution points, from which power is then distributed to local lighting circuits. The isolated receptacle circuit serves to limit the leakage current that might cause possible personnel shock to acceptable values and is a safety requirement. General lighting is provided by fluorescent units because of their higher lumen output relative to that of incandescent lights.

Investigations have been made of the potential weight savings introduced by using aluminum conductor cables instead of standard copper conductor cables. For equivalent current-carrying capacity, however, the conductor area of aluminum cables is on the average 1.6 times that of copper cables, and the larger conductor diameter equates with greater weight of insulating material. Aluminum conductor cables themselves are not lighter than copper cables below cable ratings of 39 amps (approximately 30 horsepower). There are very few loads of this magnitude on the SES.

Overall Weight and Power Considerations. Figure 10-7 indicates electrical system weight and rating in kilowatts as a function of ship gross weight. Only the equipments required to operate the basic ship and support the operating crew are included. All equipments associated with

ELECTRICAL SYSTEM: WEIGHT AND POWER REQUIREMENTS

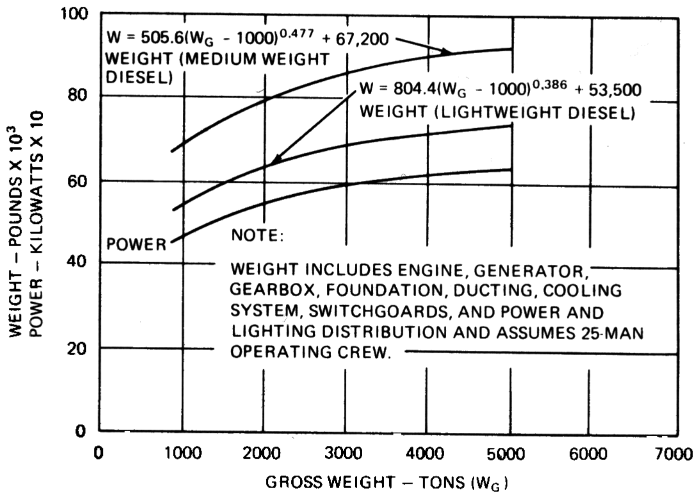


FIGURE 10-7

payload functions have been excluded. The weights have been based on 60 Hz, 450-volt power distribution and on 60 Hz, 120-volt lighting distribution.

HYDRAULICS (Navy Weight Group 5)

The SES systems and components that may utilize hydraulics include steering, mooring, anchoring, cargo handling, and weapon systems; door closures; elevators; auxiliary thrusters; remote valve operation; turbine starting; and rotating machinery, such as pumps. Three basic types of hydraulic systems can be considered for the SES: constant flow, constant pressure, and demand. The demand system, shown in Figure 10-8, is of prime interest in that it incorporates within its own design all the elements of the others and can perform all the same functions.

In the demand system, a variable pump and/or motor serves to regulate the pressure and flow in the system which, in turn, varies the system output force, torque, and speed. All of this can be accomplished with a constant rpm input. This arrangement is especially useful for winches, hoists, and mooring and anchoring equipment and for turbine starting. An accumulator permits storage of energy for reuse, as employed in the operation of an elevator, which stores energy going down and uses energy going up, or of hatch doors that must be closed quickly. The accumulator is also a source of emergency power in the event of a power failure, as it can maintain a system at constant pressure for long periods of time.

A centralized system incorporating the features just described is light in weight and economical to install, operate, and maintain. In this system one or more pumps supply a number of components through a common manifold, with supply and return tubing for each component. On the SES, where this arrangement may not prove feasible because of the remote location of components, individual electro- or turbo-hydraulic systems can be installed locally. Since long tubing runs add to the weight and power requirements of a centralized system, a weight/power trade-off analysis should be performed to determine which locations are sufficiently remote to warrant use of individual systems.

BASIC HYDRAULIC DEMAND SYSTEM DIAGRAM

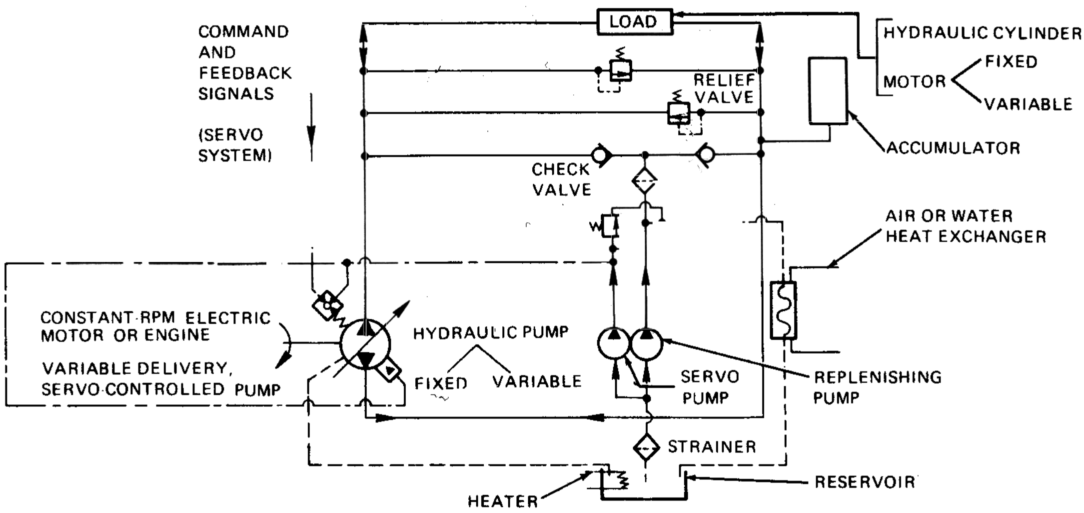


FIGURE 10–8

Whether a centralized system is incorporated or not, a separate hydraulic system should be used to operate the steering gear for rudders or waterjet nozzle/diverters. In many instances such a system is driven by one of the propulsion turbines or the auxiliary power unit.

Generally, hydraulic systems in aircraft and marine service operate at a pressure of 3000 pounds per square inch (psi). This high pressure permits use of smaller components. Even higher pressures (5000 psi) are possible, with resultant weight and volume savings.

Hydraulic fluid temperature must be maintained below about 160°F during operation. With a system efficiency of 75%, heat exchangers are often necessary, especially in miniaturized systems wherein surface radiation area has been minimized. In cold environments heaters may be necessary for start–up.

The materials that can be used in hydraulic systems are steel, stainless steel, or copper-nickel 70–30, with preference given to the second two materials because of their resistance to environmental corrosion.

Figure 10-9 indicates the weight of the hydraulic system, based on the requirements of high-speed maneuvering. High–speed maneuvering particularly requires steering, and will also require ventral fin actuation if a retractable fin is involved. In the weight calculations it has been assumed that the hydraulic pumps are directly driven by an auxiliary power diesel engine, and the prorated portion of the diesel installation weight is thus included.

BILGE AND BALLAST (Navy Weight Group 5)

The bilge and ballast systems function to maintain the safety, stability, list, and trim of the SES. The fluid utilized in both the bilge and the ballast system will be essentially seawater. System pressures will depend on the height the liquid is to be raised and the piping lengths involved; generally, 40 psig is considered ample. To provide corrosion resistance, materials such as copper–nickel, and PVC can be used for piping, while bronze, monel, titanium, and PVC are suitable for pumps, valves, and fittings.

HYDRAULIC SYSTEM: WEIGHT REQUIREMENTS

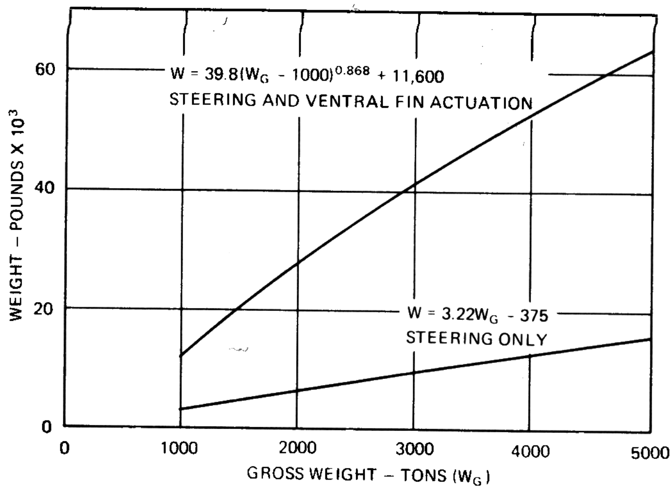


FIGURE 10–9

Bilge. A unique feature of the surface effect ship is its high degree of compartmentation; and, because of this feature, its bilge system is less important in damage control than is the system on a displacement-type ship. Thus, the primary function of the SES bilge system is to remove accumulated liquids (condensation, leakage, washwater) from watertight compartments below the weather deck to maintain the stability and the weight of the ship rather than to ensure survival if one of the buoyancy compartments is ruptured. On the SES the bilges (the low points of each compartment) are located between the sidehulls and in the sidehull areas not occupied by fuel or ballast tanks. Accumulated liquids in the bilges between the sidehulls can be drained by gravity when the SES is on the cushion or by using portable, submersible pumps or permanent rotary pumps when it is off the cushion. The sidehull bilges can be drained only by using pumps. Figure 10–10 depicts a typical bilge system arrangement.

TYPICAL BILGE SYSTEM DIAGRAM

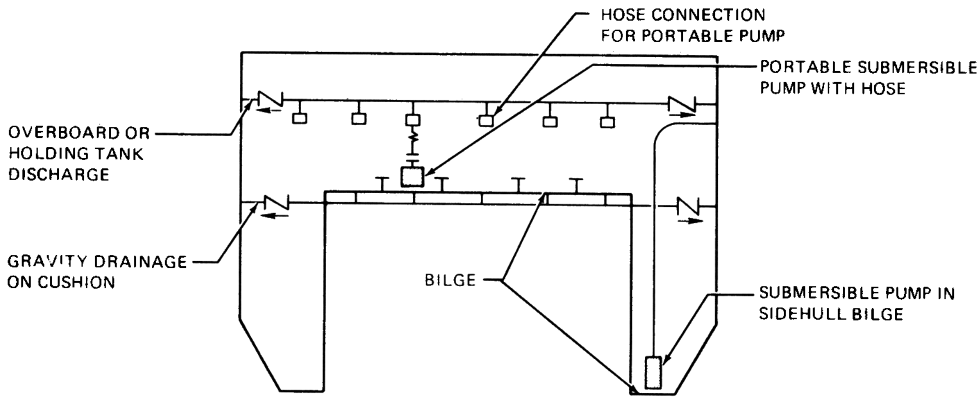


FIGURE 10–10

Should the hull of the SES be ruptured, water would enter only the compartment adjacent to the rupture, and the buoyancy of the remaining non-flooded compartments would ensure survival. Once the hull were cushion-borne, the water would most probably drain out by gravity. Hence, the weight penalty for a bilge system that might provide total dewatering capability is not warranted. Of the alternatives, the submersible electric pump system, with a 20-pound, 100-gpm pump (or connections for a portable pump) in each compartment (Figure 10-10), is the lightest of the systems available. Other available choices would be to employ manifolding, eductors, or compressed air.

Ballast. The purpose of the ballast system is to maintain trim and stability by providing balancing weight (seawater) about the transverse and longitudinal centerline. A fuel system in which fuel can be transferred to and from tanks located forward, aft, port, and starboard can serve as the primary system for maintaining trim and heel when loads are added or removed. For those instances in which little fuel remains and ballasting is necessary, a seawater ballast system should be available. Such a system consists basically of tanks located at extreme forward, aft, port, and starboard positions (for a minimal weight penalty and maximum moment effect) and a means of flooding and emptying them.

A typical ballast system is diagrammed in Figure 10-11. Gravity filling and drainage can be accomplished by using sea-connected piping and valves to flood the tanks when the SES is off the cushion and to drain them when on the cushion.

TYPICAL BALLAST SYSTEM DIAGRAM

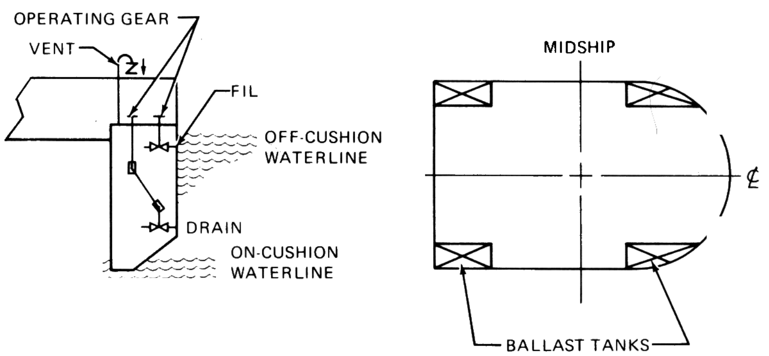


FIGURE 10-11

An auxiliary fill and drainage capability may be necessary to compensate for fuel burnoff while the SES is operating on the cushion. The capacity (gpm) of this system will be a little smaller than that of the fuel system. Fill water can be obtained either from the sea adjacent to the ballast tank by pumping or from the lube-oil cooling system.

Overall Weight Considerations. Figure 10-12 indicates the weight of a bilge and ballast system consisting of four ballast tanks, each with its own pump for filling, and a portable pump bilge system. All piping has been assumed to be PVC.

FIRE PROTECTION AND COUNTERMEASURE WASHDOWN (Navy Weight Group 5)

Fire Protection. This function involves three basic elements: detection, containment, and extinguishing. The means of protection vary with the types of combustibles involved (solid-Class A, liquid-Class B, and electrical-Class C).

BILGE AND BALLAST SYSTEM: WEIGHT REQUIREMENTS

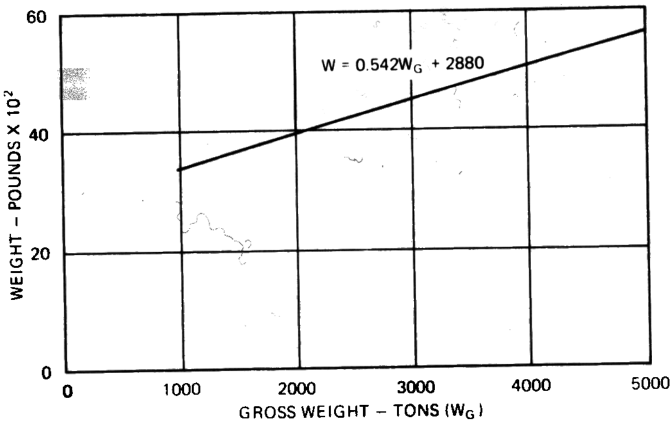


FIGURE 10-12

Automatic detection of fires can be provided by installing product-of-combustion and/or thermal detectors. Lightweight installations are available which employ remote detectors wired to a main control circuit board. Ideally, ionization detectors are best suited to Class C fires whereas smoke detectors can be effectively used in Class A and B fires. However, ionization types are relatively new and unproven in the marine environment, where unusual vibration and atmospheric conditions exist.

Thermal detection devices employing either pneumatic tubes for rate of rise or thermal elements for fixed temperature limit or rate of rise are already proven in the marine field. In these systems the sensing elements, which are 0.08-inch diameter tubes containing air or thermistors, are strategically located in the space to be protected to monitor abnormal elevations in temperature. These detectors, each covering about 250 square feet of floor, active audible or visual alarms at a central control station.

Once the fire has been located, means can be taken to contain and extinguish it. Means of containment can be implemented in the design stage with the specification of fire boundary bulkheads adequately insulated to resist conflagration temperatures for minimal periods of time up to 60 minutes. Also in the design stage, closures can be provided in the ventilation system to permit containment of the fire and of the extinguishing agent as well, if necessary.

The extinguishing agent serves to smother and cool the fire below combustion temperature. Since seawater is readily available, it is usually the prime extinguishing agent on a marine vessel. The standard installation is a water-filled firemain running throughout the ship and connecting to fireplugs and sprinklers. The weight of such a system may be prohibitive for SES application, however, and a more suitable installation may therefore be a dry pipe seawater system of thin wall (Schedule 5) piping acting as a backup to a primary liquid gas system. With this arrangement, water will not be introduced into the piping until a fire occurs.

Those areas in which seawater sprinklers are normally used because of the high cooling capacity of water include munition storage and dry cargo spaces. Pipe sizing for these areas is based on 8 gpm/square foot of area. All other parts of the SES should be serviced by 1 1/2- to 2 1/2-ips fireplugs with 100- to 250-gpm supply through 50- and 100-foot hoses. Water supply for the system can be obtained from the lube-oil seawater cooling pumps or from pumps installed at the seawater ballast tanks especially for that purpose.

Large Class B (fuel, oil, etc.) fires such as may occur in the helicopter handling and hangar areas, require a foam system for control and extinguishing. The central unit in such a system is a portable electric or gasoline-motor driven generator which mixes seawater with appropriate chemicals to produce foam. A typical 7-hp unit (3 1/2 by 3 1/2 by 5 feet, and weighing 210 pounds) can generate foam at the rate of 6,000 cubic feet per minute (cfm). This capability is obtained using a mixture ratio of 45 gpm of water to 0.7 gpm of foam concentrate. Concentrate weight is about 8 pounds per gallon. A permanently installed water-driven generator (4 1/2 by 5 feet in diameter and weighing 550 pounds) can generate 15,000 cfm of foam with a ratio of 150 gpm of water to 2.2 gpm of concentrate. The water can be supplied from the dry firemain and the foam distributed by portable fabric ducts.

In enclosed spaces, where the extinguishing agent can be confined to the fire area, the use of Halon 1301 (a liquified gas) provides a minimal weight system for all classes of fires. This agent is non-conducting, colorless, and odorless and leaves no residue, thus permitting its use on machinery and electrical equipment without attendant damage. Its use also allows personnel to see and breathe in the fire area, permitting them to remain to help combat the fire. This latter characteristic makes Halon preferable to the conventional total-flooding carbon dioxide system. The Halon system is also lighter and requires less space than carbon dioxide systems, the ratio being on the order of 1 to 5. It is similar in installation to a high-pressure carbon dioxide system in that it is stored in banks of cylinders and distributed through fixed nozzles in an overhead location. Each nozzle covers an area of 900 to 1200 square feet. System pressures are on the order of 600 psi with pipe fittings of galvanized steel. The cylinders weigh from 250 to 300 pounds (capacity, 200 pounds) and are 4 to 6 feet high and about 14 inches in diameter. One pound of Halon can service up to 100 cubic feet of space.

The installation of identical portable fire extinguishers throughout the enclosed areas of the SES will simplify both the operation and the servicing of the fire protection system. Dry chemical units presently available can be used to combat all three classes of fires. A typical installation would be one 9-pound cylinder per 3,000 square feet of area, with a 75-foot travel distance between stowages.

Provision must also be made for the storage of fire-fighting equipment intended for use by the crew. This equipment includes proximity fire-fighting suits, oxygen-cylinder breathing apparatus, and pack-type oxacetylene emergency cutting gear. This equipment should be stored close to high-hazard areas but not directly in them.

Countermeasure Washdown. The countermeasure washdown system distributes seawater over the weather surfaces to prevent adherence of contamination particles (such as result from nuclear fallout). It consists of forward-facing (windward) spray nozzles, positioned along the forward edge of the SES and athwartships at intervals farther aft, which make use of surface airflow to distribute the washdown spray. The supply water can be obtained from the same source as that for the dry firemain. Operating pressures are usually in the range of 20 to 80 psi, depending on the type of nozzle employed. Because of the highly exposed conditions pertaining, nozzle and piping material should be PVC, titanium, monel, or copper-nickel 70-30.

A drainage system can be provided in the weather deck area (including inlet ducts, etc.) to remove water accumulated from such sources as rain, spray, or the washdown system. Drainage overboard is above the off-cushion waterline by gravity flow. Drain sizes are generally 2 ips, with PVC being the recommended piping and fitting material.

Overall Weight Considerations. Figure 10-13 indicates the weight of the fire protection and countermeasure washdown systems.

FIRE PROTECTION AND COUNTERMEASURE WASHDOWN SYSTEMS WEIGHT REQUIREMENTS

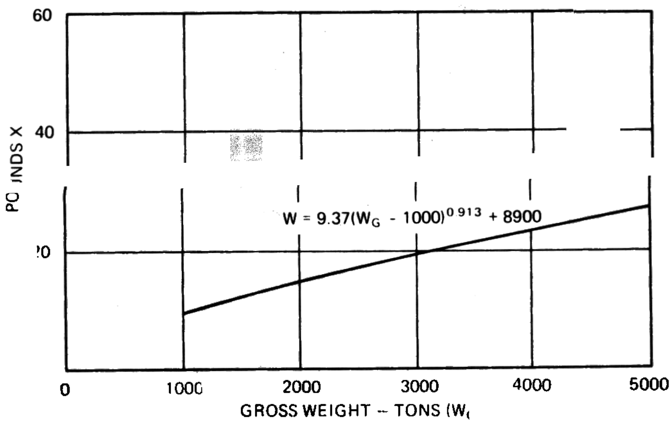


FIGURE 10–13

DEICING (Navy Weight Group 5)

Icing phenomena have been the subject of extensive study and analysis in the case of airborne vehicles, such as airplanes and helicopters, but little similar data have been developed regarding icing on ocean going ships. Thus, the specific conditions of temperature, wind, relative humidity, water droplet size, and spray pattern under which icing can be expected are not well defined at this time. It is well known, however, that ships encounter icing conditions in northern latitudes and that accumulations of up to 5 inches or more in thickness occur frequently. It is therefore necessary to consider the effect of icing on the SES and to incorporate appropriate deicing means.

The accumulation of ice on SES surfaces exposed to ambient air and sea spray can affect the weight, trim, and stability of the ship to a marked degree. For example, it has been calculated that a typical SES in the 2000–ton class might accumulate over 100 tons of ice and that this tonnage could cause incremental trim moment changes of approximately 6000 ft–tons. Additionally, such accumulations can eliminate visibility from the bridge and impair the functioning of such equipment as aircraft landing pads, air intakes, electronic antennae, and externally mounted weapons.

A number of deicing and anti–icing systems are available for possible SES application, many of them developed for aircraft application and brought to a high degree of effectiveness and reliability by extensive use. In general, deicing systems are defined as those which remove ice after its formation, while anti–icing systems prevent the formation and buildup of ice. Various distinct types of deicing and anti–icing systems are described below.

Chemicals. These systems utilize ethylene glycol and various alcohols as freezing–point depressants. A typical system consists of storage tanks, pumps, valves, controls, piping, and spray nozzles or bars. The amount of fluid applied determines the system characteristic; i.e., whether it is an anti–icing or a deicing system. The utility of chemical systems on the SES is probably limited to specific, relatively small areas, such as windows and windshields, as attempts to employ them over large surface areas leads to prohibitively high fluid–storage requirements.

Electric Heat. These systems employ grids of thin wires either flush–mounted on the surface to be protected or, in some cases, as in windshields, imbedded just below the exterior surface. If operated continuously at temperatures above freezing, they are classified as anti–icing. Where cyclic operation is employed to remove ice periodically, as in deicing applications, provision should be made to remove the loosened ice if the airstream is not capable of doing so. Electric systems require, besides the obvious electric power source, wiring, switches, and other control and protective devices. Heating from these systems is about 60% efficient in terms of the input power supplied to the system. For typical temperature conditions and ice removal rates of 1 inch per hour, an electrical anti–icing system can be expected to require approximately 0.75 kilowatt of power per square foot, while a deicing system will require power on the order of one-tenth of this amount.

As can be deduced from these figures, electric heating systems are probably not practical when large surface areas are involved because of the large total power requirements. For example, complete anti–icing of all exposed surfaces on a typical 2000– to 3000–ton SES design might require on the order of 30,000 kilowatts; even deicing would require some 3,000 kilowatts, which is probably considerably in excess of the total electric power that would be installed for other purposes.

Hot–Air Heating. These systems employ heated air bled from gas turbine compressors or turbine exhaust gases. As practicable, the hot air or gas is either circulated through the double-wall structure of the outer surface to be protected or, in some cases, conducted to the surface directly by transpiration. Such a system basically requires the incorporation of extensive double-wall structure capable of resisting moderate pressures, plus attendant ducting, insulation, and controls. Fuel consumption penalties would be imposed on the prime movers due to the bleed-air requirements and the possible increase in exhaust back–pressure.

The heating efficiency of such systems ranges from 50% to 80%, depending on the working pressure of the circulating gas. Due to the relatively large amount of energy available in the exhaust stream of the gas turbine engine, only a fraction of the exhaust gas from the total installed powerplant would be required to operate a typical gas heating system. However, because of the large weight penalty associated with such a system, application is probably limited to relatively small areas of the ship. If anti–icing in the cushion volume is required, a portion of the exhaust gas could be mixed with the cushion–fan air to increase that air temperature to a point above freezing.

Pneumatics. These systems employ surface–mounted rubber boots that are cyclically inflated by compressed air to break up ice forming on surfaces to be protected. Additional provisions must be made to remove the broken ice if the airstream is unable to do so. System components include an air supply, valves, switches, piping, and the inflatable boots. The typical system weighs about 0.8 pound per square foot of boot area, and approximately 1.0 cfm of air (at 15–22 psig) is required per square foot of boot area. Because of these requirements, application to large surfaces is impractical and must be limited to special, relatively small areas.

Mechanical, Electrical Impulses. A recent Russian development, based on the transmission of an electrical impulse to an ice–bearing structure, causes a momentary deformation of the structure below the permanent set level but above that level for the attached ice. The ice is broken into relatively small pieces, and tests have been successful on ice thickness of up to 2 inches. No operational system of this type is presently known. Power requirements are about 3.5 watts per square foot of surface area, and the system weighs about 0.6 pound per square foot, including the required electrical impulse generating units, programming switches, inductors and thyristors. Both the weight and the required power characteristics of this system make it attractive for applications to large surfaces, such as decks and sidehulls, if further development of the system is successful. Its use is obviously restricted in areas near electronic equipment, which may be adversely affected by the electrical pulses.

Overall Weight and Power Considerations. Table 10-5 indicates the preferred deicing or anti-icing system for the critical items that might be required to be kept ice free on an SES. It is impractical, however, in terms of weight and power requirements to incorporate anti-icing or even deicing systems on every surface of large SES's subject to icing. The exception to this might be employment of the electric impulse system, although operational use may be some years in the future.

PREFERRED DEICING/ANTI-ICING SYSTEMS

Critical Item	Preferred System				
	Chemical	Electrical	Hot-Air	Pneumatic	Impulse
Pilot House Windows		X			
Turbine Intakes			X		
Antennae		X		X	
Helicopter Landing Area			X		
Large Exposed Surfaces					X

TABLE 10-5

Since the added weight of ice will result in degraded performance only in the full load condition; only a limited capability to deice equipment and critical areas need be included in SES weight requirements. Electrical thermal anti-icing of pilot house windows is included in the electrical system weight. Turbine intakes are provided with manifolding and valves for bleeding compressor discharge air, and these weights are included with the engine weights. Antennae will be equipped with electric thermal or pneumatic boots for deicing, and these weights are included with the equipment. Furthermore, helicopter landing and other localized areas can be deiced manually by using hot turbine bleed air supplied by insulated pipe and hose; for advanced design, assume an additional weight allowance of approximately 500 pounds for these pipes and hoses.

AIRCRAFT FACILITIES (Navy Weight Group 6)

An SES can be configured to accommodate vertical and short takeoff and landing aircraft as well as helicopters; for the immediate future, however, the helicopter is the most likely choice. Thus, this discussion will be limited to the requirements for helicopter operations.

Capabilities provided can vary from complete maintenance facilities, with hangars and day/night, all-weather launch/land capabilities to a simple replenishment hover area for visual-flight-rule-only operations. The size of the ship and its mission will determine the level and classes of facilities. A simple VERTREP (vertical replenishment) drop area can be incorporated on a small SES for use in daylight under visual flight rules, with the only weight penalty being some additional fire–fighting equipment.

If a landing area is required, the effect of helicopter weight on the trim and stability of the ship must be analyzed, and deck structure in the landing area must be designed for the dynamic landing loads. The gross weight of helicopters presently used by the U.S. Navy varies from 10,000 pounds for the H1 helicopter to 42,000 pounds for the H53 helicopter. The clear area required for landing and launching a helicopter is closely related to rotor diameter. The minimum length of open deck required aft of the deckhouse varies from the approximately 74 feet needed for a H2 helicopter to approximately 90 feet for the H53 helicopter. Landing areas should not be located forward of the deckhouse. Helicopter facility requirements are discussed in detail in Reference 50.

For initial weight estimations only a drop area for replenishment purposes is included. Fire-fighting equipment weights are incorporated with those for the fire protection systems. More elaborate helicopter facilities are considered to be part of the payload.

STORES HANDLING AND REFUELING AT SEA (Navy Weight Group 5)

Stores handling operations can be divided into two categories: pier–side handling and replenishment at sea. While the purpose of the operation is the same in each case, the tempo and working conditions are very different.

At the pier, the operation is facilitated by using shore–based cranes to deliver the load to the hatch. This technique eliminates, or greatly reduces, the horizontal movement on the main deck otherwise associated with stores replenishment. Refueling at pier side is relatively straightforward.

At sea, alongside–connected replenishment should be considered for fuel oil transfer only. Stores replenishment should be by helicopter. In alongside-connected replenishment at sea, the sending and receiving ships are separated by some distance and the stores are transferred by a system of king posts, wires, and blocks. The receiving ship is thus required to carry such additional gear as blocks, lines, winches, portable king posts, outriggers, or other reinforced structure to withstand the loads of the replenishment gear. This gear and its associated weight can be eliminated by replenishing by helicopter.

The pallets received in vertical (helicopter) replenishment average 1000 pounds for dry stores and 2000 pounds for ammunition. These stores must be transported by the ship's force from the receiving area to the stowage area on board. The design of the handling system is influenced by the quantities and types of stores to be received, the frequency of resupply, the general arrangement of the ship, and the manpower available. For a smaller ship that will receive only one or two pallets in a replenishment operation, a chute installed in an existing hatchway may suffice. For a larger ship that may receive 50,000 pounds of stores over a span of several hours, some mechanical means of stores handling must be provided. Hand or fork–lift trucks may be used to move the stores on the main deck. Where weight permits, hoists or elevators may be used to lower the stores. A lighter weight but slower alternative to a mechanical hoist is a portable davit. This davit might also be used for boat handling. Below deck stores may be moved horizontally with relative ease on gravity conveyors.

Refueling at sea may be accomplished by the probe–fueling method, which is the most efficient and lightest of the conventional systems. In a refueling–at–sea operation the delivery ship and SES maneuver to parallel courses and maintain a separation distance of 100 to 200 feet. A wire rope is then passed from the delivery ship to the SES, and a probe installed at the end of the fueling hose and mounted on a trolly is run along that wire to the SES. On the SES, the fueling hose is connected to a bell-mouth receiver that incorporates a quick-release device. The receiver is connected to the fuel oil fill piping by hose, and the receiver may be mounted on a portable tripod to reduce topside equipment. However, a series of padeyes must be mounted on the side of the SES deckhouse to secure the span wire, and the deckhouse must thus be reinforced to withstand the loads imposed by the span wire. For larger ships, a double probe system can be considered; in this way, 1000 tons of fuel can be transferred in an hour.

A weight allowance of about 2500 pounds should be made for a system consisting of the following: (1) helicopter replenishment of stores, using hand trucks for movement on the main deck, a jib crane for lowering below deck, and aluminum roller conveyors for movement below deck and (2) refueling at sea by the single-probe method, using a portable tripod.

BOAT HANDLING AND LIFESAVING (Navy Weight Group 5)

The traditional system of lifesaving at sea, using hard-hulled lifeboats installed under gravity davits, is too heavy and introduces too great an aerodynamic drag penalty to be considered for the SES. For this traditional approach, the weight per lifeboat seat is 450 pounds. Alternatively, using inflatable 15-man CO_2 life rafts, complete with survival equipment and installed in hard covers, the weight per lifeboat seat can be reduced to 38 pounds. In addition, the frontal area of the inflatable life rafts is only 5 square feet as compared with 50 square feet for a conventional lifeboat.

A work, shore-party, or rescue boat may be required aboard the SES, and an inflatable raft will not be suitable for these functions. A standard 12-foot to 16-foot boat equipped with a small outboard motor might be suitable; if intensive helicopter operations can be expected, however, a more substantial utility boat should be provided to support those operations. To save weight, the small crane needed for boat handling can also be used to handle stores during replenishment operations.

Given an operating crew of 25 men, an appropriate system would include two 15-man inflatable life rafts and miscellaneous life jackets, representing a weight allowance of approximately 1200 pounds.

ANCHORING, TOWING AND MOORING (Navy Weight Group 5)

If the design of the SES anchoring, towing, and mooring systems is based on conventional criteria for winds and currents, water depths, and safety factors and on use of standard equipment, an unacceptably large weight will result. For an SES a much more critical assessment must be made.

Anchoring. Reasonable design requirements for anchoring are a 70-knot wind, a 4-knot current, and 20 fathoms (120 feet) of water. These requirements are as cited in Reference 51 except for the water depth, for which Reference 51 suggests 40 fathoms. U.S. Navy requirements are 100-knot wind and 60 fathoms of water. Standard practice is to design the anchor line to a factor of safety of 5; however, this can be reduced to 3 for the SES, providing the lower factor of safety is supported by a higher level of maintenance and more frequent replacement than occur in standard practice.

A Navy lightweight (Danforth) anchor is a good choice. Explosive embedment anchors are difficult to stow and handle and are still experimental, with low reliability.

The anchor line is normally steel chain. Chain is heavy as compared with wire and synthetic rope, but it is more resistant to wear and deterioration. A synthetic rope system would be the lightest in weight, but the large diameters required make it extremely cumbersome to handle and store. Wire rope represents the best compromise among the conflicting requirements of light weight, ease of handling, and reliability. A modified conventional hydraulic windlass/capstan arrangement can be used to handle the anchor cable.

Towing. This system is light in weight, since the wire-rope anchor cable can be used for towing. Closed chocks installed port and starboard are the only additional components required on the SES. This system will permit towing at speed up to approximately 10 knots in a sea state 3.

Panama Canal towing cables, chocks, and bitts need not be incorporated. They can be added on a temporary basis when the canal is to be transited.

Mooring. The mooring system should be designed to withstand a 100-knot wind from any direction at dockside. The system itself consists of nylon hawsers and of aluminum bitts and chocks installed along the perimeter of the ship. Warping is accomplished using the anchor capstan, forward, and a capstan on centerline, aft. To save weight, the hawsers and their stowage reels are kept dockside rather than on the ship. In an emergency, the wire rope anchor cables and an arrangement of portable fairleads (carried on board) can be used for mooring.

Overall Weight Considerations. Figure 10-14 indicates the weight of the anchoring, towing, and mooring systems.

ANCHORING, TOWING, MOORING SYSTEMS WEIGHT REQUIREMENTS

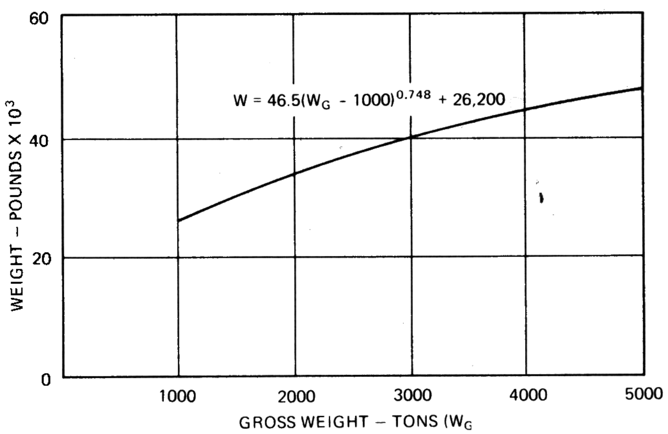


FIGURE 10-14

DAMAGE CONTROL (Navy Weight Group 6)

The standard naval architectural rules for preservation of watertight integrity and prevention of progressive or unsymmetrical flooding must be observed. The typical SES structural configuration, with the transverse and longitudinal bulkheads relatively closely spaced to ensure low structural weight, is also an excellent configuration with respect to basic damage control requirements.

Watertight doors and closures and remote valve operating gear should be installed where necessary to ensure that accesses and subsystems can be closed off to maintain the integrity of watertight boundaries. Fire dampers should be installed in ventilation and air conditioning ductwork to prevent the spread of fire.

Damage control shoring normally consists of various lengths of 4–inch by 4–inch timbers. To save weight, a system of telescoping aluminum tubes would be more appropriate.

An allowance of approximately 500 pounds should be made for damage control stations consisting of a table and rack for damage control diagrams, hand lanterns, and telescoping aluminum shoring.

Chapter XI
WEIGHTS

In this chapter the SES weight classifications as based on the weight groupings listed in the U.S. Navy Ship Work Breakdown Structure are noted, and advanced design estimating relationships are given. Additionally, specific definitions are given for design and maximum gross weight, empty weight, weight margin, and variable or disposable load.

WEIGHT CLASSIFICATIONS

As part of the General Specification for Ships, the U.S. Navy has established a procedure whereby all phases of ship design, construction, or conversion are identified, correlated, and categorized, resulting in a standardized system of grouping materials and components. The basis of the ten classification groups in the Ship Work Breakdown Structure (SWBS), Reference 52, is basic function, and seven are concerned with the weight breakdown of the ship, as shown in Table 11-1.

WEIGHT GROUPINGS AS CLASSIFIED BY BASIC FUNCTION

Major Group	Function
Group 100	Hull structure
Group 200	Propulsion plant
Group 300	Electric plant
Group 400	Communication and surveillance
Group 500	Auxiliary systems
Group 600	Outfit and furnishings
Group 700	Armament

TABLE 11-1

Table 11-2, taken from the SWBS, summarizes the subgroups (three-digit numbers ending in a single zero) and elements (all three-digit numbers not ending in a zero), included in each of the seven major weight groups. Detailed descriptions of the elements in each subgroup can be found in the SWBS.

An examination of the SWBS indicates that provision has been made for items peculiar to the SES. Element 234 includes the propulsion gas turbines. Subgroup 240 includes the propulsion transmission and propulsor. Propellers are included in Element 245, and waterjets, in Element 247. Element 567 is for the complete lift system, including fans, fan engines, and seals. If the propulsion and lift system engines are integrated, they would be included in Element 234. The sidehulls are included in Subgroup 110.

U.S. NAVY SHIP WORK BREAKDOWN STRUCTURE

GROUP 100 HULL STRUCTURE

100	Hull structure, general
101	General arrangement-structural drawings
110	Shell and supporting structure
111	Shell plating, surf. ship and submarine press. hull
112	Shell plating, submarine non-pressure hull
113	Inner bottom
114	Shell appendages
115	Stanchions
116	Longit. framing, surf. ship and submarine press. hull
117	Transv. framing, surf. ship and submarine press. hull
118	Longit. and transv. submarine non-press. hull framing
120	Hull structural bulkheads
121	Longitudinal structural bulkheads
122	Transverse structural bulkheads
123	Trunks and enclosures
124	Bulkheads in torpedo protection system
125	Submarine hard tanks
126	Submarine soft tanks
130	Hull decks
131	Main deck
132	2nd deck
133	3rd deck
134	4th deck
135	5th deck and decks below
136	01 hull deck (Forecastle and Poop decks)
137	02 hull deck
138	03 hull deck
139	04 hull deck and hull decks above
140	Hull platforms and flats
141	1st platform
142	2nd platform
143	3rd platform
144	4th platform
145	5th platform
149	Flats
150	Deck house structure
151	Deckhouse structure to first level
152	1st deckhouse level
153	2nd deckhouse level
154	3rd deckhouse level
155	4th deckhouse level
156	5th deckhouse level
157	6th deckhouse level
158	7th deckhouse level

TABLE 11–2

TABLE 11-2 (Continued)

159	8th deckhouse level and above
160	Special structures
161	Structural castings, forgings, and equiv. weldments
162	Stacks and Macks (combined stack and mast)
163	Sea chests
164	Ballistic plating
165	Sonar domes
166	Sponsons
167	Hull structural closures
168	Deckhouse structural closures
169	Special purpose closures and structures
170	Masts, kingposts, and service platforms
171	Masts, towers, tetrapods
172	Kingposts and support frames
179	Service platforms
180	Foundations
181	Hull structure foundations
182	Propulsion plant foundations
183	Electric plant foundations
184	Command and surveillance foundations
185	Auxiliary systems foundations
186	Outfit and furnishings foundations
187	Armament foundations
190	Special purpose systems
191	Ballast, fixed or fluid, and buoyancy units
192	Compartment testing
195	Erection of sub sections (progress report only)
198	Free flooding liquids
199	Hull repair parts and special tools

GROUP 200 PROPULSION PLANT

200	Propulsion plant, general
201	General arrangement – propulsion drawings
202	Automated ship control systems
210	Energy generating system (nuclear)
211	(Reserved)
212	Nuclear steam generator
213	Reactors
214	Reactor coolant system
215	Reactor coolant service system
216	Reactor plant auxiliary systems
217	Nuclear power control and instrumentation
218	Radiation shielding (primary)
219	Radiation shielding (secondary)
220	Energy generating system (non-nuclear)
221	Propulsion boilers
222	Gas generators

TABLE 11-2 (Continued)

223	Main propulsion batteries
224	Main propulsion fuel cells
230	Propulsion units
231	Propulsion steam turbines
232	Propulsion steam engines
233	Propulsion internal combustion engines
234	Propulsion gas turbines
235	Electric propulsion
236	Self-contained propulsion systems
237	Auxiliary propulsion devices
238	Secondary propulsion (submarines)
239	Emergency propulsion (submarines)
240	Transmission and propulsor systems
241	Propulsion reduction gears
242	Propulsion clutches and couplings
243	Propulsion shafting
244	Propulsion shaft bearings
245	Propulsors
246	Propulsor shrouds and ducts
247	Water jet propulsors
250	Propulsion support sys. (except fuel and lube oil)
251	Combustion air system
252	Propulsion control system
253	Main steam piping system
254	Condensers and air ejectors
255	Feed and condensate system
256	Circulating and cooling sea water system
259	Uptakes (inner casing)
260	Propulsion support systems (fuel and lube oil)
261	Fuel service system
262	Main propulsion lube oil system
263	Shaft lube oil system (submarines)
264	Lube oil fill, transfer, and purification
290	Special purpose systems
298	Propulsion plant operating fluids
299	Propulsion plant repair parts and special tools
	GROUP 300 ELECTRIC PLANT
300	Electric Plant, General
301	General arrangement-electrical drawings
302	Motors and associated equipment
303	Protective devices
304	Electric cables
305	Electrical designating and marking
310	Electric power generation
311	Ship service power generation
312	Emergency generators

TABLE 11-2 (Continued)

313	Batteries and service facilities
314	Power conversion equipment
320	Power distribution systems
321	Ship service power cable
322	Emergency power cable system
323	Casualty power cable system
324	Switchgear and panels
330	Lighting system
331	Lighting distribution
332	Lighting fixtures
340	Power generation support systems
341	SSTG lube oil
342	Diesel support systems
343	Turbine support systems
390	Special purpose systems
398	Electric plant operating fluids
399	Electric plant repair parts and special tools
	GROUP 400 COMMAND AND SURVEILLANCE
400	Command and surveillance, general
401	General arrangement – command and surveillance
402	Security requirements
403	Personnel safety
404	Radio frequency transmission lines
405	Antenna requirements
406	Grounding and bonding
407	Electromagnetic interference reduction (EMI)
408	System test requirements
410	Command and control systems
411	Data display group
412	Data processing group
413	Digital data switchboards
414	Interface equipment
415	Digital data communications
416	Command and control testing
417	Command and control analog switchboards
420	Navigation systems
421	Non-electrical/electronic navigation aids
422	Electrical navigation aids (incl navig. lights)
423	Electronic navigation systems, radio
424	Electronic navigation systems, acoustical
425	Periscopes
426	Electrical navigation systems
427	Inertial navigation systems
430	Interior communications
431	Switchboards for I.C. systems
432	Telephone systems

TABLE 11-2 (Continued)

433	Announcing systems
434	Entertainment and training systems
435	Voice tubes and message passing systems
436	Alarm, safety, and warning systems
437	Indicating, order, and metering systems
438	Integrated control systems
439	Recording and television systems
440	Exterior communications
441	Radio systems
442	Underwater systems
443	Visual and audible systems
444	Telemetry systems
445	TTY and facsimile systems
446	Security equipment
450	Surveillance systems (surface)
451	Surface search radar
452	Air search radar (2D)
453	Air search radar (3D)
454	Aircraft control approach radar
455	Identification systems (IFF)
459	Space vehicle electronic tracking
460	Surveillance systems (underwater)
461	Active sonar
462	Passive sonar
463	Active/passive (multiple mode) sonar
464	Classification sonar
465	Bathythermograph
470	Countermeasures
471	Active ECM (incl combination active/passive)
472	passive ECM
473	Torpedo decoys
474	Decoys (other)
475	Degaussing
476	Mine countermeasures
480	Fire control systems
481	Gun fire control systems
482	Fire control systems (non-sonar data base)
483	Fire control systems (sonar data base)
489	Fire control systems (switchboards)
490	Special purpose systems
491	Electronic test, checkout, and monitoring equipment
492	Flight control and instrument landing systems
493	Non combat data processing systems
494	Meteorological systems
495	Integrated operational intelligence systems
498	Command and surveillance operating fluids
499	Command and surv. repair parts and special tools

TABLE 11-2 (Continued)

GROUP 500 AUXILIARY SYSTEMS

500	Auxiliary systems, general
501	General arrangement-auxiliary systems drawings
502	Auxiliary machinery
503	Pumps
504	Instruments and instrument boards
505	General piping requirements
506	Overflows, air escapes, and sounding tubes
510	Climate control
511	Compartment heating system
512	Ventilation system
513	Machinery space ventilation system
514	Air conditioning system
515	Air revitalization systems (submarines)
516	Refrigeration system
517	Auxiliary boilers and other heat sources
520	Sea water systems
521	Firemain and flushing (sea water) system
522	Sprinkler system
523	Washdown system
524	Auxiliary sea water system
526	Scuppers and deck drains
527	Firemain actuated services – other
528	Plumbing drainage
529	Drainage and ballasting system
530	Fresh water systems
531	Distilling plant
532	Cooling water
533	Potable water
534	Aux. steam and drains within machinery box
535	Aux. steam and drain outside machinery box
536	Auxiliary fresh water cooling
540	Fuels and lubricants, handling and storage
541	Ship fuel and fuel compensating system
542	Aviation and general purpose fuels
543	Aviation and general purpose lubricating oil
544	Liquid cargo
545	Tank heating
550	Air, gas, and misc. fluid systems
551	Compressed air systems
552	compressed gases
553	O2 N2 system
554	LP blow
555	Fire extinguishing systems
556	Hydraulic fluid system
557	Liquid gases, cargo

TABLE 11-2 (Continued)

558	Special piping systems
560	Ship control systems
561	Steering and diving control systems
562	Rudder
563	Buoyancy and hovering (submarines)
564	Trim system (submarines)
565	Trim and heel (roll stabilization)
566	Diving planes and stabilizing fins
567	Lift systems
568	Maneuvering systems
570	Underway replenishment systems
571	Replenishment-at-sea
572	Ship stores and personnel and equip. handling
573	Cargo handling
580	Mechanical handling system
581	Anchor handling and stowage systems
582	Mooring and towing systems
583	Boat handling and stowage systems
584	Mechanically operated door, gate, ramp, turntable sys.
585	Elevating and retracting gear
586	Aircraft recovery support systems
587	Aircraft launch support systems
588	Aircraft handling, servicing and stowage
589	Miscellaneous mechanical handling systems
590	Special purpose systems
591	Scientific and ocean engineering systems
592	Swimmer and diver support and protection systems
593	Environmental pollution control systems
594	Submarine rescue, salvage, and survival systems
595	Towing, launching and handling for underwater sys.
596	Handling sys. for diver and submersible vehicles
597	Salvage support systems
598	Auxiliary systems operating fluids
599	Auxiliary systems repair parts and tools
	GROUP 600 OUTFIT AND FURNIHSINGS
600	Outfit and furnishings, general
601	General arrangement – outfit and furn. drawings
602	Hull designating and marking
603	Draft marks
604	Locks, keys, and tags
605	Rodent and vermin proofing
610	Ship fittings
611	Hull fittings
612	Rails, stanchions, and lifelines
613	Rigging and canvas
620	Hull compartmentation

TABLE 11-2 (Continued)

621	Non-structural bulkheads
622	Floor plates and gratings
623	Ladders
624	Non-structural closures
625	Airports, fixed portlights, and windows
630	Preservatives and coverings
631	Painting
632	Zinc coating
633	Cathodic protection
634	Deck covering
635	Hull insulation
636	Hull damping
637	Sheathing
638	Refrigerated spaces
639	Radiation shielding
640	Living spaces
641	Officer berthing and messing spaces
642	Noncommissioned officer berthing and messing spaces
643	Enlisted personnel berthing and messing spaces
644	Sanitary spaces and fixtures
645	Leisure and community spaces
650	Service spaces
651	Commissary spaces
652	Medical spaces
653	Dental spaces
654	Utility spaces
655	Laundry spaces
656	Trash disposal spaces
660	Working spaces
661	Offices
662	Machinery control centers furnishings
663	Electronics control centers furnishings
664	Damage control stations
665	Workshops, labs, test areas (incl portable tools, equip)
670	Stowage spaces
671	Lockers and special stowage
672	Storerooms and issue rooms
673	Cargo stowage
690	Special purpose systems
698	Outfit and furnishings operating fluids
699	Outfit and furnish. repair parts and special tools
	GROUP 700 ARMAMENT
700	Armament, general
701	General arrangement – weaponry systems
702	Armament installations
703	Weapons handling and stowage, general

TABLE 11-2 (Continued)

710	Guns and ammunition
711	Guns
712	Ammunition handling
713	Ammunition stowage
720	Missiles and rockets
721	Launching devices (missiles and rockets)
722	Missile, rocket, and guidance capsule handling sys.
723	Missile and rocket stowage
724	Missile hydraulics
725	Missile gas
726	Missile compensating
727	Missile environmental monitoring and launcher contr.
728	Missile heating, cooling, temperature control
730	Mines
731	Mine launching devices
732	Mine handling
733	Mine stowage
740	Depth charges
741	Depth charge launching devices
742	Depth charge handling
743	Depth charge stowage
750	Torpedoes
751	Torpedo tubes
752	Torpedo handling
753	Torpedo stowage
754	Submarine torpedo ejection
760	Small arms and pyrotechnics
761	Small arms and pyrotechnic launching devices
762	Small arms and pyrotechnic handling
763	Small arms and pyrotechnic stowage
770	Cargo munitions
772	Cargo munitions handling
773	Cargo munitions stowage
780	Aircraft related weapons
782	Aircraft related weapons handling
783	Aircraft related weapons stowage
790	Special purpose systems
792	Special weapons handling
793	Special weapons stowage
797	Misc. ordnance spaces
798	Armament operating fluids
799	Armament repair parts and special tools

WEIGHT ESTIMATION

For advanced design purposes it is appropriate to consider the light ship weight of the SES as consisting of the sum of a few major weight categories. A separate weight estimate can be made for each major group by using theoretical or empirical relationships, and the light ship weight can then be estimated by summing the individual group estimates. The following major weight categories constituting the light ship weight represent the minimum number recommended for advanced design. The equations needed to calculate the weight of the individual groups are summarized in Table 11-3. Each of these equations has been discussed previously in the appropriate chapter.

Structure. This category includes the primary hard structure of the ship, which consists of such structural elements as keel members, frames, stringers, bulkheads, hull plating, decking, and the basic superstructure. Most hard secondary structure, if integral with the primary structure or if serving a purpose other than aesthetic or cosmetic, should be included in this weight category. Also assumed to be included are structural elements integral with the primary structure, such as fuel, ballast, trim, and flotation tanks; air and fluid ducting; and machinery mounts. As defined here, the structure weight group does not include the bow and stern seals (except for such integral attachments as may be provided for such seals).

Power Plant. The second major weight group for which estimates are to be made in the initial design phase comprises the power plant components. For a fossil-fuel system the principal components are the propulsion turbines and the inlet and exhaust ducting. Also included are thermal and acoustical ducting insulation; filters and demisters; engine closures; engine starting system; and those portions of the lubricating oil, electrical control, anti-icing, and engine mounting systems that are integral with the engine. Machinery mounts integral with the main structure are included in the structural weight category. If the lift fans are driven by the propulsion turbines, the turbine weight is included here; but if separate turbines are used to drive the lift fans, those are included with the lift system.

Included in the weight of the nuclear power system are the reactor, shield, heat exchangers, ducting, propulsion and lift turbines, transmissions, and propulsors. The lift system and the transmissions and propulsors have been included because the nuclear power system is so closely integrated that these components cannot be readily separated.

Transmissions. The transmission system constitutes the third weight group as comprised of the following: gears; clutches; shafting; couplings; bearings; lubricating oil system, including pumps, reservoirs, valves, filters, piping; and saltwater cooling system, including pumps and heat exchangers.

For ships with waterjet propulsion, the transmission system is lighter than for ships with propeller systems, since right-angle gearboxes, or "drop boxes," may not be required and the thrust bearing is integral with the waterjet pump.

Propulsors. The propulsor may be either a waterjet or a semi-submerged supercavitating propeller. For weight-estimating purposes the waterjet system includes the pump, inlet diffuser, inlet duct, thrust bearing, exit nozzle, thrust vectoring and reversing assembly, and entrapped water. The hydraulic actuation of the thrust vectoring and reversing assembly is not considered to be part of the propulsor weight and is thus included with the weight of auxiliaries.

Propeller weight includes the weight of the blades, the hub, and the control mechanism in the hub.

Lift. Lift system weight is taken to include the weight of lift fans, fan engines (if not driven directly by the propulsion engines), transmission, and ducting for fan engines and fans.

Seals. Incorporated in seal system weight estimates are the seals, down-stop cables, attachments, and other hardware, except that attachments integral with the hull are included in the weight of the structure.

Appendages. The weight of appendages is specifically the weight of rudders for a propeller-driven ship or of ventral fins for a waterjet-propelled ship. The weight of the steering assembly for the waterjet is integral with the propulsor weight. The weight of hydraulic actuation for the steering system for both propeller and waterjet systems is included in the hydraulic system weight. If the fin is made retractable, the weight of the retraction system is included as an option in the hydraulic system weight.

Auxiliaries. The auxiliary systems comprise the mechanical, piping, and electrical systems that perform the support functions for the ship's major elements. The auxiliary systems include the control station (bridge), and the fuel, personnel, electrical, hydraulic, bilge and ballast, fire protection, deicing, stores handling and refueling, boat handling, lifesaving, and anchoring, towing and mooring systems.

The personnel system weight includes the weight of accommodations, food service, water, plumbing, heating, ventilating, and air conditioning necessary to support the basic ship operating crew; that weight required to support crew members associated with operation and maintenance of payload functions, such as weapons, sonar, and fire control radar, is considered to be part of the payload weight.

The weights assigned to the electrical and hydraulic systems are also only those occasioned by basic ship functions. Increases in system weight to accommodate payload functions are considered to be part of the payload weight.

Light Ship Weight Estimate. The summation of the advanced design weight categories – structure, power plant, transmissions, propulsors, lift, seals, appendages, and auxiliaries – gives a first approximation of the light ship weight (W_{LS}), except for an allowance for weight margin. Summing the advanced design weight categories, or the U.S. Navy SWBS groups 100 through 700, results in defining the light ship weight of the SES.

ADDITIONAL DEFINITIONS

Weight Margin. Throughout the design phases, light ship weight calculations for the SES, as for conventional ships, should include an allowance for contingent differences that may develop between the initial design estimates and the actual weight realized after construction. This allowance is called the weight margin. Unforeseen or unavoidable increases in light ship weight can occur for a number of reasons, including predicted but unrealized advances in the component state of the art and changes in the performance specifications during the design stage. The practice of allowing for the almost-invariable increase in the light ship weight estimated in the initial design stage is therefore a sound one and is recommended as an essential part of a conservative design procedure.

The penalties incurred by an unplanned increase in light ship weight of as little as 5% on an SES may result in a 20% reduction in payload or a 10% reduction in range for typical design parameters. With an allowance made for such unplanned increases during initial design, performance degradations of this kind can largely be avoided if weight increases in fact do occur. In those cases in which a weight margin is assumed but no increase is experienced, the resulting increases in the performance of the ship are obtained at a relatively small expenditure of design effort and resources.

For advanced design purposes the weight margin can be accounted for in at least three ways in the weight-estimation process. First, the weight-estimating relationships for the groups making up the light ship weight can be considered as conservative estimates and as including, therefore, a weight margin allowance. A second method is to incorporate a suitable decimal fraction of each weight group as a margin allowance. Alternatively, a percentage of light ship weight can be assumed as a margin. For most advanced designs a weight-margin value of 5% of the light ship weight should be adequate and is recommended for use as part of the design procedure.

Variable Load. The variable load is the sum of those weight items aboard the ship which are consumable or subject to frequent change. The term "disposable load" is sometimes applied to this weight category. The variable load can be defined as the sum of the following items:

1. Ship's officers, crew, and personal effects
2. Marines, troops, passengers, and personal effects
3. Ship ammunition
4. Aviation ammunition
5. Aircraft
6. Provisions and personal stores
7. General stores
8. Ship marine complement stores
9. Aeronautical stores
10. Ship ordnance stores
11. Aviation ordnance stores
12. Fuel, water, oil, and miscellaneous liquids
13. Cargo

This list agrees with the definition of variable load given in the SWBS. For advanced design weight-estimating purposes, the variable load can be considered as consisting of fuel and payload, as follows:

$$W_{VAR} = W_F + W_{PL}$$

where

W_F = weight of fuel – pounds
W_{PL} = weight of payload – pounds

The payload term, W_{PL}, includes all the variable-load items 1–13, above with the exception of fuel and further includes hardware items, electronic and electrical power equipment, hydraulics, foundations and associated supporting structure, and crew and personnel systems essential to the particular assigned mission of the SES. As defined here, then, the payload comprises items from Group 100 (Hull Structure), Group 300 (Electric Plant), Group 400 (Communications and Surveillance), Group 500 (Auxiliary Systems), Group 600 (Outfit and Furnishings) and the entire Group 700 (Armament) of the SWBS. For advanced design purposes this is necessary to facilitate weight accounting when examining alternate concepts.

Design, Maximum Gross, and Empty Weight. The design gross weight of the SES, or full-load condition, is defined as the sum of the light ship weight (W_{LS}) and the variable load (W_{VAR}):

$$W_G = W_{LS} + W_{VAR}$$

where the variable load includes the fuel and payload required for a specified mission profile and ship performance. If the ship is designed for this specified mission – that is, the fuel and payload capacities are specifically equivalent to those required, the design gross weight is then identical to the maximum gross weight of the ship, which is defined as the weight condition with full fuel and maximum payload capacity.

The empty weight (W_{EMPTY}) is defined as the sum of the light ship weight (including margin allowance) and the payload.

$$W_{EMPTY} = W_{LS} + W_{PL}$$

ADVANCED DESIGN WEIGHT EQUATIONS

Eq. No.	Navy Weight Group	Equation	Definition of Terms
		$W_{ST} = W_G\ [224 + (640)\ W_G^{-0.0414}\ (P_C/L_C)^{-0.776}]$ (also see Figure 3-17)	
		$W_{pp} = (HP)_{Bp}\left[0.69 + \frac{84}{\sqrt{(HP)_{Bp}}}\right]$	
		$W_{tr_p} = 0.85\ (HP)_B + 2500$	
4		$W_{tr_{WJ}} = 0.28\ (HP)_B + 1200$	
		$W_{PROP} = 68\ [\ (HP)_S \times 10^{-3}]^{1.26}$	
6		$W_{WJW} = [\ (HP)_S \times 10^{-3}]^{1.46}$	
		$W_{fs} = 3.3\ (HP)_C' + 89.2\ \sqrt{(HP)_C'}$	
8		$W_{bs} = [3.3\ W_G + 19500]\left(\frac{2}{Lc/Bc}\right)$ where $W_G \geqslant 2000$ tons	
8a		$W_{bs} = 13.3\ W_G\left(\frac{2}{Lc/Bc}\right)$ where $W_G < 2000$ tons	
		$W_{ss} = [3.69\ W_G + 20550]\left(\frac{2}{Lc/Bc}\right)$ where $W_G \geqslant 2000$ tons	W_{ss} = stern seal weight -- pounds
9a		$W_{ss} = (14.7\ W_G)\left(\frac{2}{Lc/Bc}\right)$ where $W_G < 2000$ tons	
10		$W_r = 0.9 W_G - 550$	W_r = weight of two rudders (used with propellers) -- pounds

TABLE 11-3

TABLE 11-3 (Continued)

Eq. No.	Navy Weight Group	Equation	Definition of Terms
11		$W_{fin} = \frac{1.25 W_G}{100} \left(\frac{W_G}{100} + 51.2 \right)$	W_{fin} = weight of two fins (used with waterjets) – pounds
	3,4,5,6	$W_{aux} = 5500\, W_G^{0.4} + 1060\, (N_{cr}) + 0.25\, [\, (HP)_{Bp} + (HP)_C\,] + 61{,}000$ (without ventral fin actuation) $W_{aux} = 3500\, W_G^{0.485} + 1060\, (N_{cr}) + 0.25\, [\, (HP)_{Bp} + (HP)_C\,] + 57{,}000$ (with ventral fin actuation)	W_{aux} = weight of auxiliary systems – pounds N_{cr} = number of crew (25 typically) $(HP)_{Bp}$ = total brake horsepower, propulsion $(HP)_C$ = total cushion horsepower
		$W_{NS} = 5.6\, (HP)_{BT} + 3.8\, (HP)_{Bp} + 4.4\, (HP)_{Bf}$ (open cycle, supercavitating propeller)	W_{NS} = weight of nuclear system, including complete power plant, transmission, propulsor, lift fan system – pounds
	2,5	$W_{NS} = 5.6\, (HP)_{BT} + 4.3\, (HP)_{Bp} + 4.4\, (HP)_{Bf}$ (open cycle, waterjet)	$(HP)_{BT}$ = total ship brake horsepower $(HP)_{Bp}$ = propulsion brake horsepower $(HP)_{Bf}$ = fan brake horsepower
3c	2,5	$W_{NS} = 5.4\, (HP)_{BT} + 1.8\, (HP)_{Bp} + 2.4\, (HP)_{Bf}$ (closed cycle, supercavitating propeller)	
13d		$W_{NS} = 5.4\, (HP)_{BT} + 2.3\, (HP)_{Bp} + 2.4\, (HP)_{Bf}$ *(closed cycle, waterjet)*	
14		$W_{MARG} = 0.05\, W_{LS}$	W_{MARG} = weight of margin – pounds
15		$W_G = 1.05\, W_{LS} + W_{PL} + W_F$	W_{LS} = light ship weight W_{PL} = payload weight W_F = useful fuel weight

Chapter XII
PARAMETRICS

In advanced design it is often necessary to make quick, accurate performance estimates of conceptual SES designs and, in particular, to provide rapid solutions to the general range-payload problem, namely, for a stated payload and range, what would be the gross weight and physical dimensions of the SES? Based on certain simplifying assumptions concerning SES geometry, and assuming one specific design speed and design sea state, a method can be formulated for obtaining ship characteristics and performance. Such a method is presented here.

STANDARD PARAMETRIC RELATIONSHIPS

The following principal assumptions apply to this method:

(1) Cushion length-beam ratio $(L_C/B_C) = 2.0$

(2) Cushion density $(P_C/L_C) = 1.5$ pounds/foot^3

(3) Cushion pressure $(P_C) = 0.9\ W_G/S_C$ pounds/foot^2

(4) Cruise speed $(V_C) = 80$ knots

(5) Average wave height $(h_w) = 3.0$ feet.

It was established by Stoiko in the SESMA study that $L_C/B_C = 2.0$ and $P_C/L_C = 1.5$ provided a good compromise between design for maximum range and the carrying of maximum payload and fuel. Reductions in these L_C/B_C and P_C/L_C values were found to cause fairly rapid reductions in payload and fuel-carrying capacity, whereas increases up to twofold had little effect.

By applying the above conditions and assumptions, one can readily write the drag equations in a simplified coefficient form, develop the drag and lift-drag ratio curves in zero sea state and waves for a ship of any gross weight, and determine the lift and propulsion requirements. Additionally, with the cushion length-beam ratio and cushion density known, the cushion pressure, cushion length and overall length, cushion width (beam) and overall beam, cushion area, cushion height, and propulsive and cushion power can be determined and reduced to a function of SES gross weight. These simplified expressions yield results that are reasonably accurate and acceptable from an advanced design standpoint, that is, within a few percent.

Drag Components. The significant components of SES drag are wavemaking drag, sidehull drag, aerodynamic drag, appendage drag, and the increment of drag due to wind and waves. The momentum drag of the cushion air and the jet thrust of the lift air flowing under the stern seal are assumed to balance each other out and, therefore, are neglected in this analysis.

a. **Wavemaking Drag.** At speeds above hump, the cushion wavemaking drag may be estimated from the relationship

$$= k \frac{W_G{}^2}{q_w \, B_C{}^2} \tag{12.1}$$

where

D_{WM} = wavemaking drag – pounds
W_G = gross weight of ship – pounds
q_w = dynamic pressure of water – pounds/foot^2
B_C = cushion beam – feet
k = constant.

This equation can also be expressed as

$$= q_w \, S_C (C_D)_{WM} \tag{12.1a}$$

where

S_C = cushion area – feet^2
$(C_D)_{WM} = 0.385(L_C/B_C)\,C_L{}^2 = 0.77\,C_L{}^2$
L_C/B_C = length-beam ratio
$C_L = P_C/q_w$ or $W_G/(S_C q_w) = W_G/2.85\,S_C\,V_k^2$
P_C = cushion pressure – pounds/foot^2
$q_w = 1/2\,\rho_w\,V^2 = 2.85\,V_k^2$
V_k = vehicle speed – knots.

b. **Sidehull Drag.** In principle, the sidehull drag consists of frictional drag, form drag, wavemaking drag, and seal and spray drag. For purposes of this analysis, sidehull drag may be considered to result from the immersion of the sidehulls due to craft pitch and heave and cushion wave slope and then may be expressed by

$$q_w \, S_C (C_D)_{SH} \tag{12.2}$$

where

D_{SH} = sidehull drag – pounds

$$(C_D)_{SH} = 4C_f \left(\frac{L_{SH}{}^2}{S_C}\right) \frac{z_{imm}}{L_{SH}} + \frac{\theta - \theta_w}{2} + \frac{\theta_w}{4} \tag{12.2a}$$

where

C_f = coefficient of frictional drag = 0.002 for high Reynolds number, typically.
L_{SH} = length of sidehull – feet, $L_{SH}{}^2/S_C \cong 2.0$, usually

z_{imm} = immersion of sidehull, z_{imm}/L_{SH} of the order of 0.005 – feet
θ = craft trim angle – radians
θ_w = wave slope = $0.385\,(L_C/B_C)\,C_L$ – radians.

In advanced design, Equation (12.2a) may be approximated by

$$(C_D)_{SH} = C_f\,(L_{SH}^2/S_C)\,[4(z_{imm}/L_{SH}) + 1.385\,(L_C/B_C)\,C_L] \tag{12.2b}$$

$$(C_D)_{SH} = 0.00008 + 0.011\,C_L \tag{12.2c}$$

c. **Aerodynamic Drag.** Aerodynamic drag (D_A), in pounds, can be expressed as:

$$D_A = q_a\,S_r\,C_{DA} = q_w\,S_C\,(C_D)_A$$

$$(C_D)_A = C_{DA}\left(\frac{S_r}{S_C}\right)\left(\frac{q_a}{q_w}\right) = C_{DA}\left(\frac{S_r}{S_C}\right)\left(\frac{\rho_a}{\rho_w}\right) \tag{12.3a}$$

$$(C_D)_A = 0.0012\,C_{DA}\left(\frac{S_r}{S_C}\right) \tag{12.3b}$$

where

D_A = aerodynamic drag – pounds
C_{DA} = aerodynamic drag coefficient (based on S_r)
q_a = dynamic pressure of air – pounds/foot^2
S_r = reference area for aerodynamic drag – (typically, frontal area) – feet^2
S_C = cushion area – feet^2
ρ_a = mass density of air (0.0024) – pound-second^2/foot^4
ρ_w = mass density of water (1.99) – pound-second^2/foot^4

With a typical value of 0.33 for C_{DA} (using frontal area as S_r) and S_r/S_C equal to 0.25, we have

$$(C_D)_A = 0.00010 \tag{12.3c}$$

d. **Appendage Drag.** The two basic components of appendage drag to be considered are drag due to ventral fins or rudders and drag due to the propulsor and its mounting system.

1. Fins or Rudder. The drag of this component consists of form drag and frictional drag. For a parabolic cross section, the drag may be approximated as

$$D_{cs} = q_w\,S_C(C_D)_{app}$$

where

D_{cs} = control surface drag – pounds

$$(C_D)_{app} = \left(\frac{S_{app}}{S_C}\right)\left[\frac{\pi}{8}\left(\frac{t_a}{\ell_a}\right)^2 + 2C_f\right] \tag{12.4a}$$

where

S_{app} = appendage or rudder area
t_a/ℓ_a = thickness-chord ratio = 0.10, typically
C_f = coefficient of frictional drag = 0.002 for high Reynolds number, typically

For a first approximation, S_{app}/S_C = 0.00685 may be used.

With these typical values, we have

$$(C_D)_{app} = 0.00005. \tag{12.4b}$$

2 Propulsor. Propulsor drag depends largely on the type and the size of the propulsor. For a first estimation, it may be expressed as

$$D_{prop} = q_w \, S_C \, (C_D)_{prop} \tag{12.5}$$

D_{prop} = propulsor drag – pounds

$$(C_D)_{prop} = 0.00003. \tag{12.5a}$$

e. **Increment of Drag Due to Wind.** When the ship is operating in a non-zero sea state, the effect of wind should be taken into consideration. A head wind increases the drag of the craft, whereas an astern wind increases propulsive thrust. The increment of drag due to wind can be expressed similarly to aerodynamic drag, that is,

$$D_{wind} = \pm \, q_w \, S_C \, (C_D)_{wind} \tag{12.6}$$

D_{wind} = drag due to wind – pounds

$$(C_D)_{wind} = \frac{2V_w}{V_k} + (V_w/V_k)^2 \quad (C_D)_A \tag{12.6a}$$

where

V_w = component of wind velocity parallel to the ship's course – knots
V_k = vehicle speed – knots

The positive sign in Equation (12.6) denotes a head wind; a negative sign indicates an astern wind.

f. **Increment of Drag Due to Waves.** The wave drag in non-zero sea states may be expressed as

$$D_{RW} = q_w S_C (C_D)_{ss} \tag{12.7}$$

where

D_{RW} = drag increment due to waves – pounds
$(C_D)_{ss} = (h_w/L_C)^2$
h_w = average wave height – feet
L_C = cushion length.

g. **Advanced Design Drag Equation (Above Hump).** If the length-beam ratio, L_C/B_C, equals 2.0, using SESMA developed Equations (12.1) to (12.7) and the typical values as suggested, the drag is reduced to:

$$D_{TOT} = q_w S_C C_D = 2.85 V_k^2 S_C C_D \tag{12.8}$$

and

$$C_D = (C_D)_{WM} + (C_D)_{SH} + (C_D)_A + (C_D)_{app} + (C_D)_{prop} + (C_D)_{wind} + (C_D)_{ss}$$

$$C_D = 0.00026 + 0.011 C_L + 0.77 C_L^2 + 0.0001 \left[\frac{2V_w}{V_k} + (V_w/V_k)^2 \right] + (h_w/L_C)^2 \tag{12.8a}$$

Based on the simplified drag expression explained above, Figure 12-1 depicts a family of SES drag curves expressed as lift-drag ratios *(L/D)*, above hump. It is significant to note that as the gross weight of an SES increases, an increase is realized in both maximum efficiency, as measured by *L/D*, and the speed at which this efficiency occurs, attaining a maximum *L/D* for the 10,000-ton craft at the 80-knot cruise speed.

h. **Advanced Design Drag Equation (Below Hump).** The drag coefficient, C_D, of Equation (12.8a) must be modified for hump speed and below. The cushion wavemaking drag coefficient, $(C_D)_{WM} = 0.77 C_L^2$ is only satisfactory above hump. If the wavemaking drag based on Equation (2.1) is substituted the expressions will be satisfactory for advanced design purposes. The drag is then:

$$D_{TOT}\ (below\ hump) = 2.85 V_k^2 S_C (C_D - 0.77 C_L^2) + \frac{f_\ell P_C^2 B_C}{16} \tag{12.9}$$

LIFT-DRAG RATIO ABOVE HUMP
(3 Foot Average Wave)

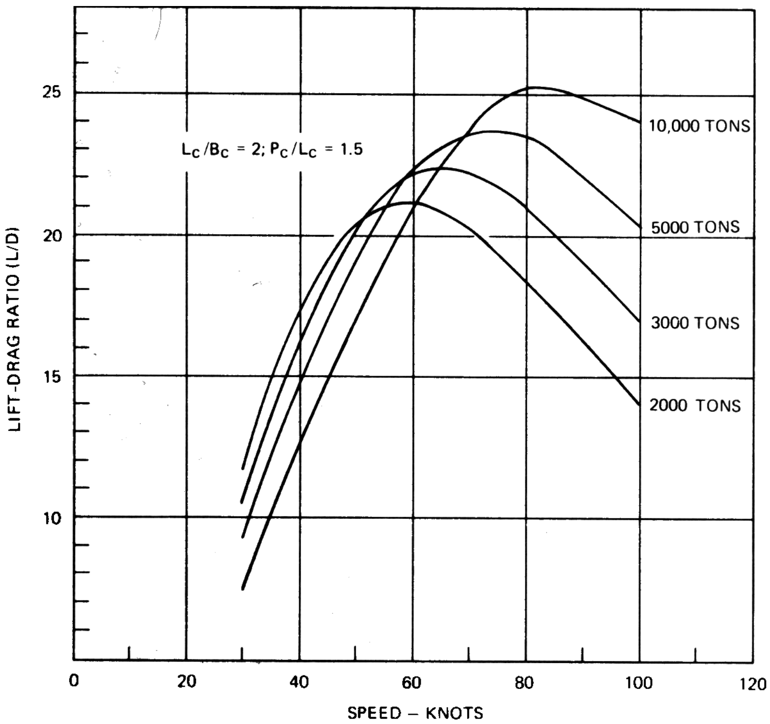


FIGURE 12–1

where

f_ℓ = cushion wavemaking drag coefficient (see Figure 2-2)

i. **Advanced Design Drag Procedure (Off-Cushion).** The approximate drag per ton of displacement (displacement = W_G) has been estimated based on barge data and is shown as a function of speed-length ratio in Figure 12–2. In computing the speed-length ratio, the speed in knots and the overall length should be used.

The horsepower can then be computed by

$$HP = \frac{(drag/ton)\,(W_G)\,(V_k)}{325\,(NPC)\,\eta_T}$$

If the high speed propulsion system is used to provide the off-cushion power, the value of (NPC) (η_T) would be approximately 0.1 to 0.2. If a separate hullborne propulsion system designed specifically for low speed operation is used, the value of (NPC) (η_T) would be approximately 0.65.

Effective Lift-Drag Ratio. In comparing vehicular systems that use power for both lift and propulsion the concept of "effective lift–drag ratio," or $(L/D)_e$, is used.

DRAG PER TON OF DISPLACEMENT (OFF-CUSHION)

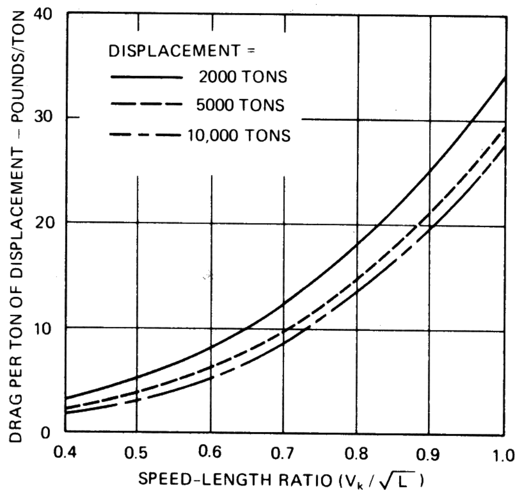


FIGURE 12-2

For the SES, the $(L/D)_e$ for range calculation is derived as follows:

$$\text{Range } (R) = (Speed)\,(Time)$$

$$R = \frac{(Speed\text{-}knots)\,(Fuel\ used\text{-}pounds)}{(Total\ fuel\ flow\text{-}pounds/hour)}$$

$$R = \frac{V_k\, W_F}{W_{fT}}$$

where

W_F = weight of fuel – pounds
$W_{fT} = (W_{fp} + W_{fc})$
W_{fp} = fuel consumption of the propulsion system – pounds/hour
W_{fc} = fuel consumption of the cushion or lift system – pounds/hour

therefore

$$\frac{R}{W_F} = \frac{V_k}{W_{fp} + W_{fc}}$$

or $$\frac{R}{W_F} = \frac{V_k}{W_{fp}\,(1 + W_{fc}/W_{fp})} \quad \text{(nautical miles/pound of fuel)}$$

where

$W_{fp} = SFC_p (HP)_{Bp}$
$W_{fc} = SFC_C (HP)_C$

SFC_p = specific fuel consumption of the propulsion system – pounds/hp-hour
$(HP)_C$ = cushion system horsepower
SFC_C = specific fuel consumption of the cushion system – pounds/hp-hour
$(HP)_{Bp}$ = propulsion brake horsepower

$$(HP)_{Bp} = \frac{(\textit{Drag-pounds})\,(\textit{Speed-knots})}{325\,(\textit{Propulsor eff.})\,(\textit{Transmission eff.})}$$

therefore,

$$\frac{R}{W_F} = \frac{V_k\ 325\ \eta_p\ \eta_T}{SFC_p\ D_{TOT}\ V_k\ [1 + (W_{fc}/W_{fp})]}$$

$$\frac{R}{W_F} \quad \left(\frac{325\ \eta_p\ \eta_T}{SFC_p\ W_G}\right) \quad \frac{W_G}{D_{TOT}\ [1 + (W_{fc}/W_{fp})]}$$

where

η_p = propulsor efficiency
η_T = transmission efficiency

In this expression the "effective" drag = D_{TOT} + $(W_{fc}/W_{fp})]$, and the effective lift–drag ratio, $(L/D)_e$, is

$$(L/D)_e = \frac{(L/D)}{[1 + (W_{fc}/W_{fp})]} \tag{12.10}$$

Expressing the above equation in a more convenient computational form results in

$$(L/D)_e = \frac{1}{\dfrac{D_{TOT}}{W_G} + 1.575\left(\dfrac{SFC_c}{SFC_p}\right)\left(\dfrac{\eta_p\ \eta_T}{\eta_{LS}}\right)\dfrac{1}{B_C}} \tag{12.11}$$

where

η_{LS} = cushion or lift system efficiency

Figure 12–3 depicts a family of curves based on solving Equation (12.11) for the standard conditions.

SES Geometry and Cushion Pressure. Figure 12–4 shows important design parameters such as cushion pressure, cushion and overall length, cushion and overall beam, cushion area, among others, as a function of gross weight.

Propulsion Horsepower. Propulsion horsepower is derived directly from the drag characteristics of the SES:

EFFECTIVE LIFT-DRAG RATIO ABOVE HUMP
(3 Foot Average Wave)

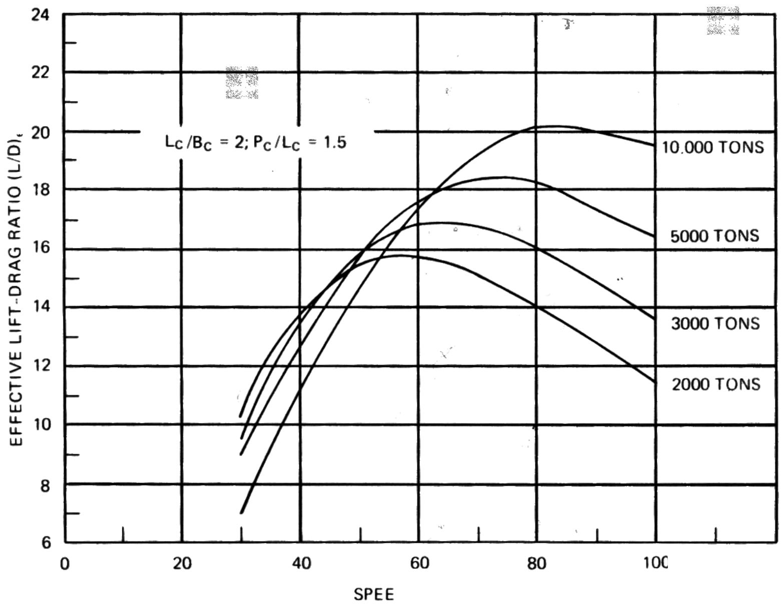


FIGURE 12–3

$$(HP)_{Bp} = \frac{(\textit{Drag-pounds})\ (\textit{Speed-knots})}{325\ (\textit{Propulsor eff.})\ (\textit{Transmission eff.})}$$

$$(HP)_{Bp} = \frac{D_{TOT}\ V_k}{325\ \eta_p\ \eta_T}$$

$$(HP)_{Bp} = \left(\frac{W_G}{(L/D)}\right)\left(\frac{V_k}{325\ \eta_p\ \eta_T}\right)$$

The critical areas for power performance are the design high–speed condition, and the hump condition. That is, the power required at hump, with no margin and with a 20% margin, in a sea state 6 (9 foot average wave height), and the power required at 80 knots in a sea state 3 (3 foot average wave height).

As a first approximation of horsepower required, the cruise conditions are assumed. Figure 12–5 shows the horsepower requirements for the standard design conditions (80 knots cruise and 3 foot average wave) as a function of SES gross weight.

Cushion Horsepower. Under calm sea conditions, the cushion horsepower is directly proportional to the leakage area and to the cushion pressure to the 3/2 power. As a maximum the leakage area may be assumed to be in scale with cushion size (i.e., proportional to cushion area), or, as a minimum, the leakage gap may be considered to remain constant (leakage area proportional to cushion length or width).

SES CHARACTERISTICS

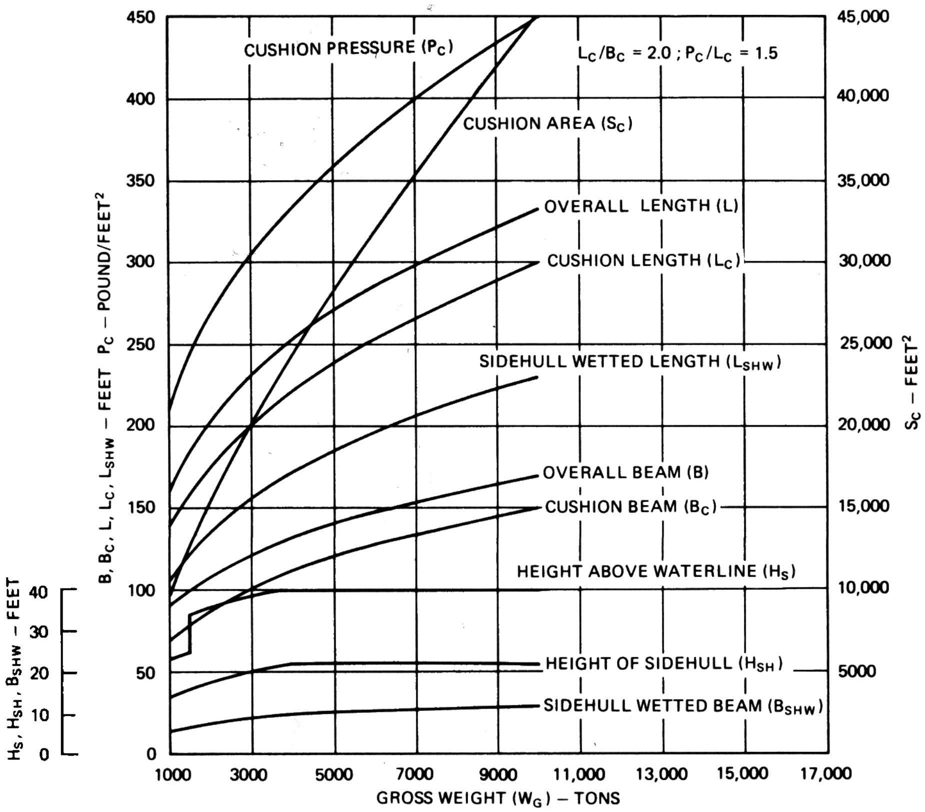


FIGURE 12-4

To maintain pressure in rough water, the air flow into the cushion must match the rate at which waves sweep through the cushion (wave pumping action). In this case, the cushion power required is proportional to the average wave height, the forward speed, the cushion beam, and the cushion pressure. The limiting condition for cushion power is evidently the highest combination of sea state and speed; therefore, since

$$\frac{(HP)_C}{W_G} = \frac{Q(P_C + \Delta P_C)}{550\, \eta_{LS}\, W_G}$$

and, for wave pumping,

$$Q = B_C\, h_w\, V$$

then,

$$\frac{(HP)_C}{W_G} = \frac{B_C\, h_w\, (P_C + \Delta P_C)\, V}{550\, \eta_{LS}\, W_G}$$

If it is assumed realistically that ΔP_C = constant (P_C), then

PROPULSION HORSEPOWER VERSUS GROSS WEIGHT

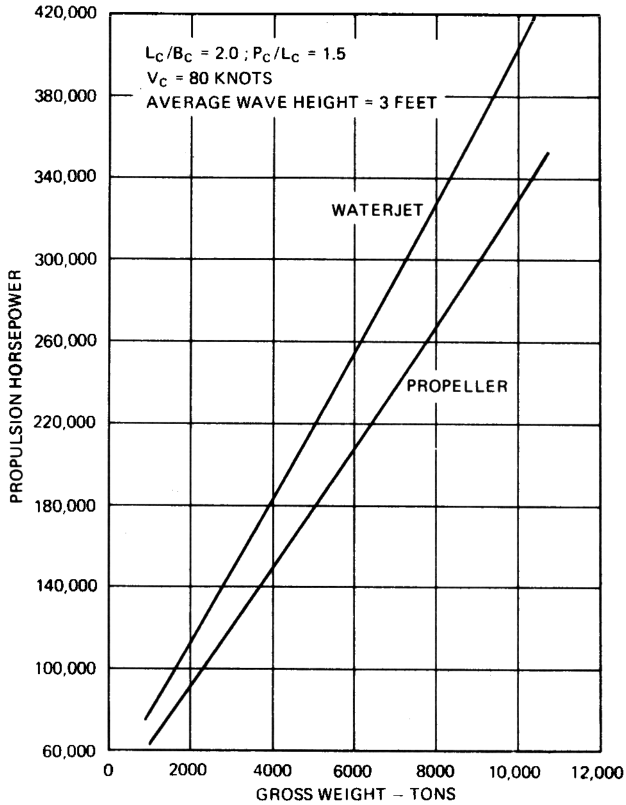


FIGURE 12-5

$$\frac{(HP)_C}{W_G} \; \alpha \; \frac{B_C \, P_C \, V \, h_w}{W_G}$$

$$\frac{(HP)_C}{W_G} \; \alpha \; \frac{L_C \, L_C}{L_C{}^3} \; h_w \; V, \; (since\; B_C \; \alpha \; L_C, \; P_C \; \alpha \; L_C, \; W_G \; \alpha \; L_C{}^3).$$

therefore,

$$\frac{(HP)_C}{W_G} \; \alpha \; \frac{h_w}{L_C} \; V.$$

Consequently, if the same design speed is chosen,

$$\frac{(HP)_C}{W_G} \; \alpha \; \frac{h_w}{L_C}.$$

Finally, if it is considered that SES capability to operate in increasing sea states is proportional to SES size, the wave height relative to cushion length (h_w/L_C) is constant, and the cushion power required relative to ship gross weight [$(HP)_C/W_G$] is constant regardless of size.

Given the following values relative to a 2200-ton SES,

B_C = 90 feet based on $L_C/B_C = 2.0$
V = 80 knots
P_C = 274 psf
$\Delta P_C = 1.05\ P_C$
η_{LS} = 0.7
h_w = 3 feet

then,

$$\frac{(HP)_C}{W_G} = \frac{90 \times 3 \times 80 \times 274 \times 1.05}{325 \times 0.70 \times 2200} = 12.4$$

Practical considerations indicate that the cushion length-beam ratio may increase beyond 2.0 to about 2.4, thus reducing the cushion width and reducing the cushion power required-ship gross weight ratio to 10.3. As a check it was estimated that, for a 2200-ton SES, a maximum flow of 35,000 cubic feet per second could be expected through the cushion, at a pressure of approximately 274 psf with a lift-system efficiency of 0.70; the result was a ratio of 11.2. Consequently, it is assumed that a reasonable relationship is an 11-horsepower lift requirement per ton of ship gross weight. It is further assumed that this ratio will hold true as fuel is consumed. This relationship is illustrated generally in Figure 12.6.

CUSHION HORSEPOWER PER TON VERSUS GROSS WEIGHT

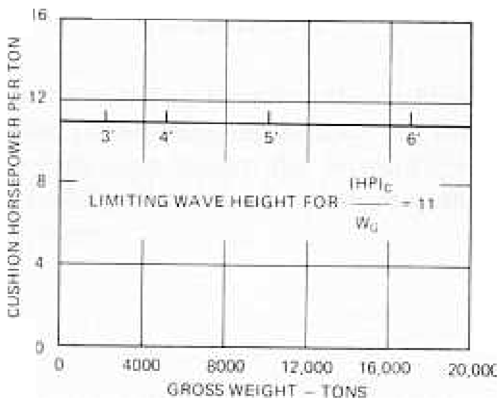


FIGURE 12-6

Figure 12-7 indicates lift horsepower requirements for sea states 3 and 6 as a function of gross weight.

Weight Distribution. The component weight distributions for the waterjet-powered and the propeller-powered SES's, as derived as a function of gross weight using the expressions of Table 11-3, are shown graphically in Figures 12-8 and 12-9, respectively.

CUSHION HORSEPOWER VERSUS GROSS WEIGHT

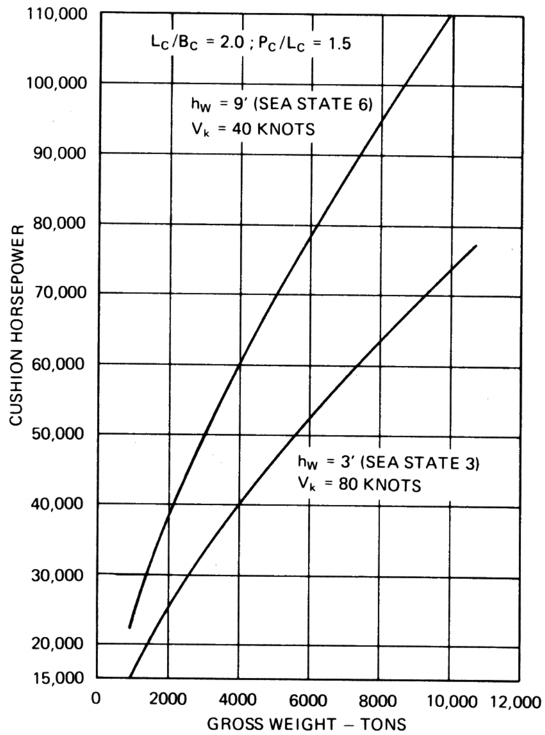


FIGURE 12-7

The propulsion system is based on the 80-knot, sea state 3 condition, but the weight of the lift system is based on satisfying the 40-knot, sea state 6 requirements.

Payload-Range – Gross Weight. The ultimate goal of advanced design is to determine the gross weight, physical dimensions, and horsepower of the SES for a stated payload and range. To arrive at this goal, the light ship-gross weight ratio (W_{LS}/W_G) for a given gross weight can be determined from Figure 12-8 or 12-9. The portion of gross weight available for payload and fuel is then

$$\frac{W_{PL}}{W_G} + \frac{W_F}{W_G} = 1 - \frac{W_{LS}}{W_G}$$

For a given payload, the payload-gross weight ratio (W_{PL}/W_G) is determined, and the fuel-gross weight ratio (W_F/W_G) is then

$$\frac{W_F}{W_G} = 1 - \frac{W_{LS}}{W_G} + \frac{W_{PL}}{W_G}$$

Range can be determined for a given fuel-gross weight ratio by the Brequet range equation expressed in the following form

WEIGHT DISTRIBUTION WITH WATERJET

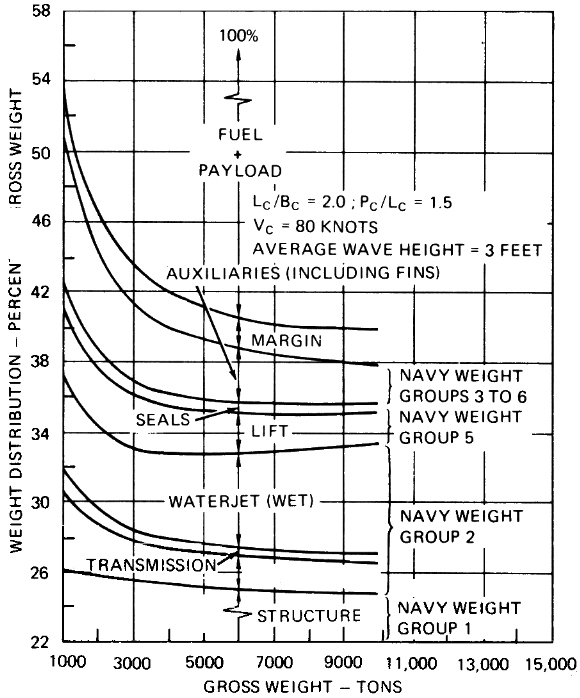


FIGURE 12-8

WEIGHT DISTRIBUTION WITH SEMI-SUBMERGED SUPERCAVITATING PROPELLER

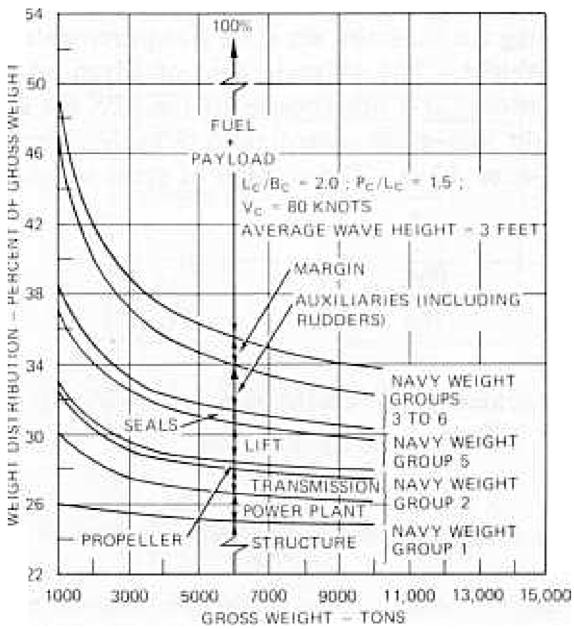


FIGURE 12-9

$$\frac{W_F}{W_G} = 1 - e^{-k}$$

where

$$k = \frac{(R)\,(SFC)}{V_C}\left[\frac{(HP)_{REQ}}{W_G}\right]$$

e = base of natural system of logarithms = 2.7183.

Figures 12–10 and 12–11 indicate the possible range and payload combinations as a function of gross weight for waterjet–powered and propeller–powered SES's, respectively. In developing Figures 12–10 and 12–11, the horsepower required has been determined from Figures 12–5 and 12–7, and the light ship–gross weight ratio from Figure 12–8 or 12–9. An average *SFC* of 0.42 was used based on a minimum *SFC* of 0.40 at maximum continuous rating increasing to a maximum *SFC* of approximately 0.44 as horsepower is reduced to keep speed constant as ship weight decreases due to fuel burn–off.

For the selected gross weight, SES dimensions can be determined from Figure 12–4.

PAYLOAD-RANGE-GROSS WEIGHT CHARACTERISTICS (WATERJET)

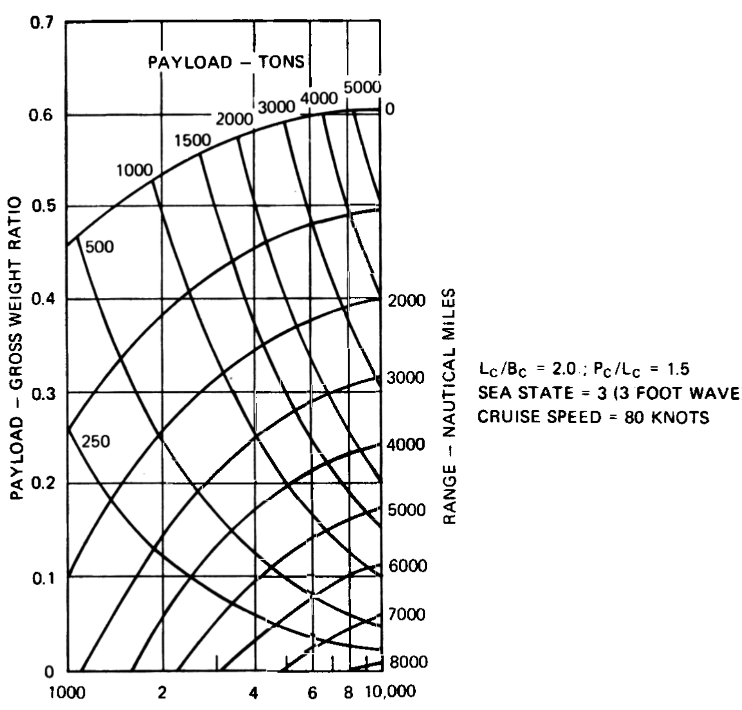


FIGURE 12–10

PAYLOAD-RANGE-GROSS WEIGHT CHARACTERISTICS
(Semi-Submerged Supercavitating Propeller)

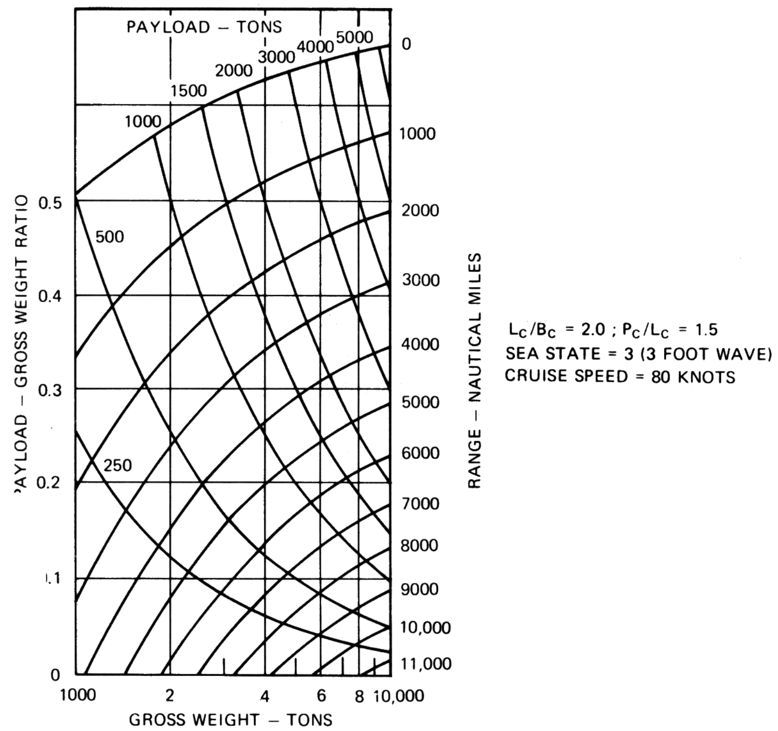


FIGURE 12-11

GENERAL PARAMETRIC RELATIONSHIPS

The previous information was predicated on a point design condition. The purpose of this section is to demonstrate the relationships between SES design parameters, such as cushion length-beam ratio, cushion pressure, and cruise speed, and SES physical and performance characteristics, such as range, payload, and overall length and beam. The objective in demonstrating these relationships is twofold: first, to provide an awareness of the interrelationships between those design parameters largely under the designer's control and the resulting SES physical and performance characteristics from a qualitative viewpoint and, second, to demonstrate the effects of design input variation on the final SES configuration in an absolute or quantitative fashion. Implementation of the second objective may also serve to assist the designer in manipulating the design data, at least to the extent of providing additional useful information when interpolation or extrapolation of design data is deemed necessary.

Table 12-1 shows the general relationships between the design parameters and the physical or performance attributes of the SES. The design parameters considered, while not constituting an exhaustive listing by any means, are nonetheless representative of the major design inputs over which the SES designer has at least some control. Included are geometric properties of the SES (cushion length-beam ratio, cushion pressure), operational parameters (wave height, cruise speed), mass properties (gross weight, light ship weight), and power characteristics (specific weight, fuel

RELATIONSHIPS BETWEEN DESIGN PARAMETERS AND PHYSICAL AND PERFORMANCE CHARACTERISTICS

Design Parameters		Physical Characteristics					Performance Characteristics			
		Geometry	Propulsive Power Req'd	Cushion Power Req'd	Light Ship Weight	Stability	Max Speed	Range	Payload	Acceleration
L_C/B_C	Cushion Length-Beam Ratio	X	X	X		X	X			X
P_C	Cushion Pressure	X	X	X		X	X			X
h_w	Average Wave Height	X	X	X			X	X		X
W_G	Gross Weight	X	X	X	X			X	X	
V_C	Cruise Speed		X				X	X	X	X
P_C/L_C	Cushion Density	X	X	X	X		X		X	X
W_{pp}/HP	Power Plant Specific Weight				X			X	X	
SFC	Specific Fuel Consumption							X	X	
NPC	Net Propulsive Coefficient							X	X	

TABLE 12–1

consumption, propulsive coefficient). Those SES physical characteristics and performance items affected by a change in a listed design parameter are noted in Table 12-1 in a self-explanatory format.

To demonstrate the specific effects of design parameter variation on SES characteristics and performance, each of the input, or design, parameters listed in Table 12-1 was varied through a range of representative values while holding the other input parameters at certain designated fixed values. In this manner the effect of each design input on the overall SES configuration was analyzed. Values assigned to the design parameters in this analysis are shown in Table 12-2.

Effect of Design Parameters on Drag. Figures 12-12 through 12-17 show the effect of cushion length-beam ratio (L_C/B_C), lift coefficient (C_L), sidehull length²-cushion area ratio ($L_{SH}{}^2/S_C$), ship frontal area – cushion area ratio (S_{front}/S_C), and appendage planform area – cushion area ratio (S_{app}/S_C) on the drag calculated by equations (12.1) – (12.8).

Effect of Cushion Length-Beam Ratio and Cushion Pressure. Figures 12-18 to 12-22 show the effect of variations in cushion length-beam ratio (L_C/B_C) and cushion pressure (P_C) on overall and cushion length, overall and cushion beam, cushion area, and sidehull length. In these curves it is assumed that 10% of the gross weight is carried by sidehull buoyancy. For a given gross weight, higher values of cushion pressure will result in less cushion area (S_C) and thus reduced length and beam if their ratio is held constant.

PRIMARY PARAMETRIC DESIGN INPUTS

Design Parameters		Units	Range of Values
L_C/B_C	Cushion length-beam ratio		2, 3, 4
P_C	Cushion pressure	pounds/foot^2	200, 250, 300, 400, 500
h_W	Average wave height	feet	0, 3, 6, 9
W_G	Gross weight	tons	1000 to 5000
V_C	Cruise speed	knots	60, 80, 100
P_C/L_C	Cushion density	pounds/foot^3	As calculated; not specified
W_{pp}/HP	Power plant specific weight	pounds/hp	1, 2, 3, 4
SFC	Specific fuel consumption	pounds/hp-hour	0.3, 0.4, 0.5
NPC	Net propulsive coefficient		0.5, 0.6, 0.7

TABLE 12-2

SIDEHULL DRAG COEFFICIENT
(L_C/B_C = 2.0)

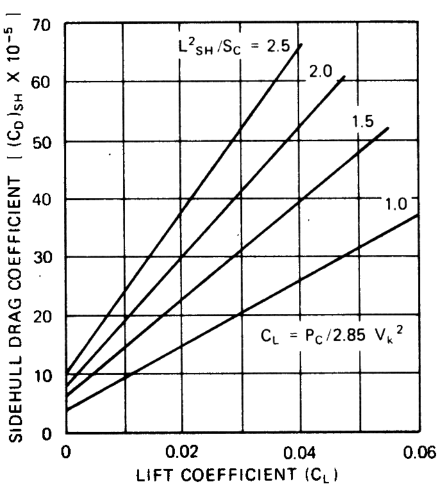


FIGURE 12-12

SIDEHULL DRAG COEFFICIENT
(L_C/B_C = 3.0)

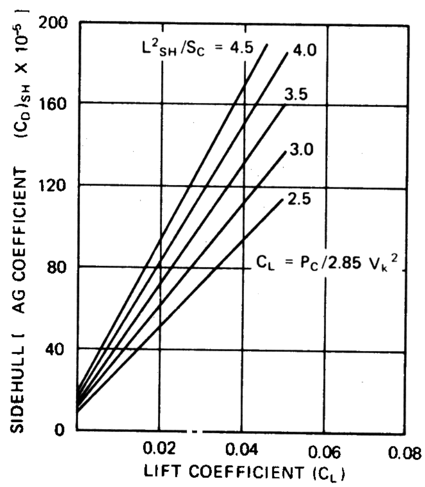


FIGURE 12-13

SIDEHULL DRAG COEFFICIENT
(L_C/B_C = 4.0)

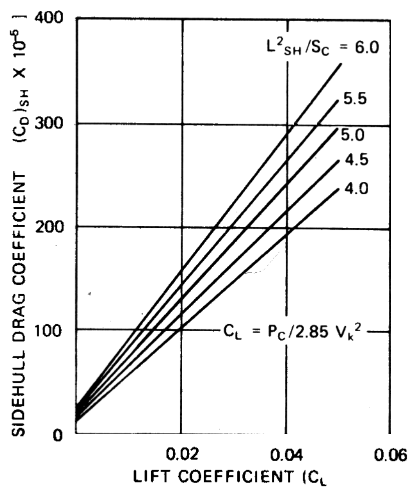


FIGURE 12-14

WAVEMAKING DRAG COEFFICIENT

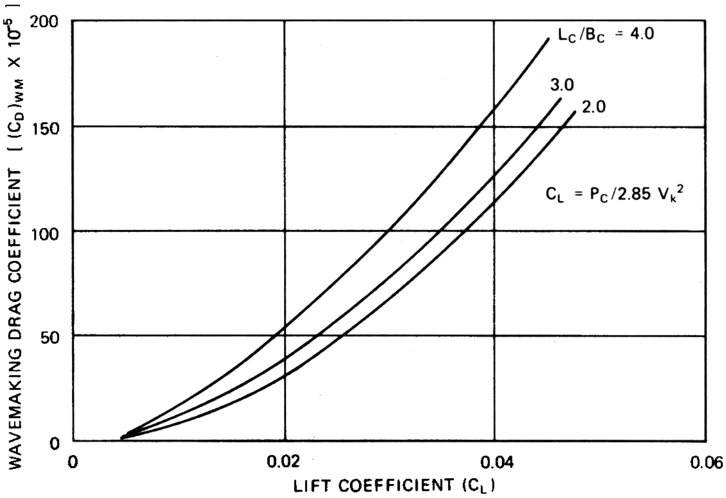


FIGURE 12-15

AERODYNAMIC DRAG COEFFICIENT

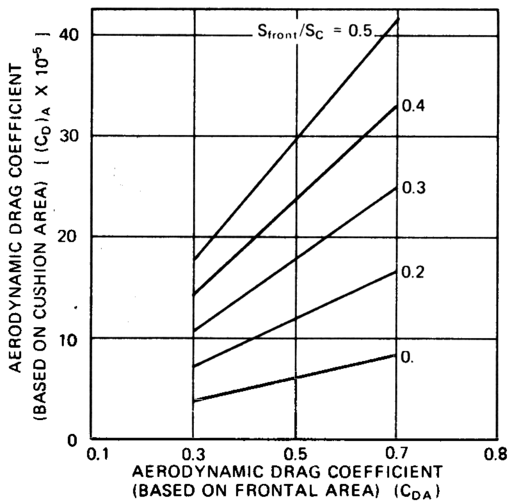


FIGURE 12-16

APPENDAGE DRAG COEFFICIENT

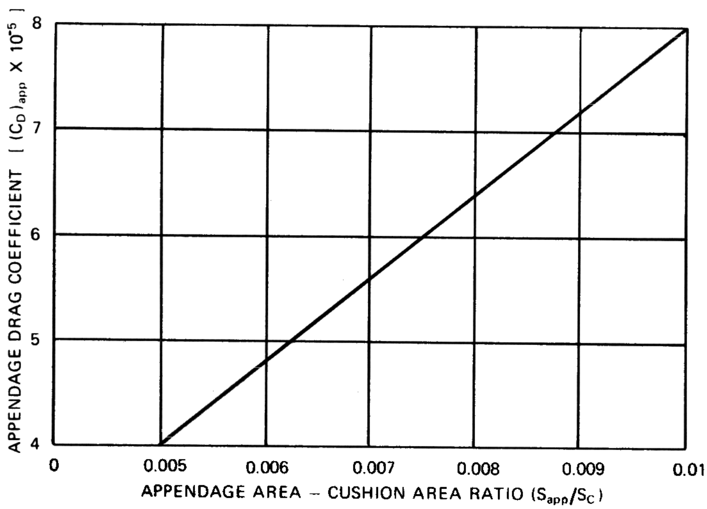


FIGURE 12-17

OVERALL AND CUSHION LENGTH, AND OVERALL AND CUSHION BEAM (L_C/B_C = 2.0)

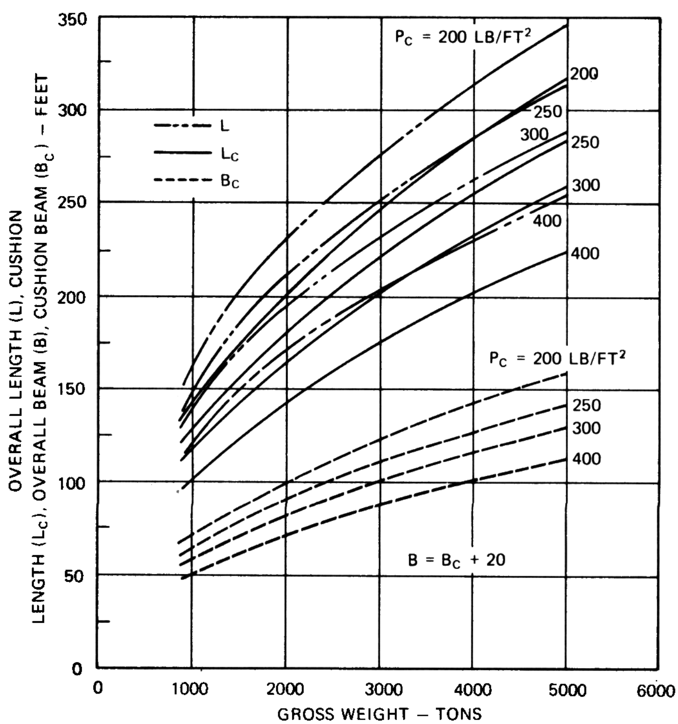


FIGURE 12-18

OVERALL AND CUSHION LENGTH, AND OVERALL AND CUSHION BEAM ($L_C/B_C = 3.0$)

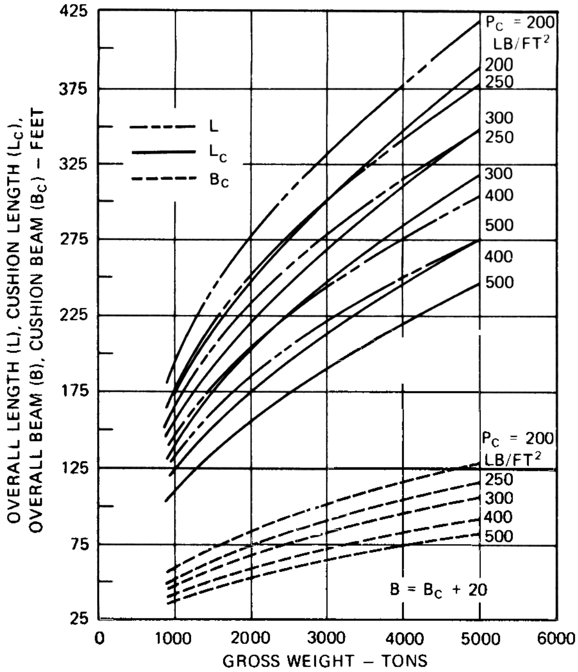


FIGURE 12-19

OVERALL AND CUSHION LENGTH, AND OVERALL AND CUSHION BEAM ($L_C/B_C = 4.0$)

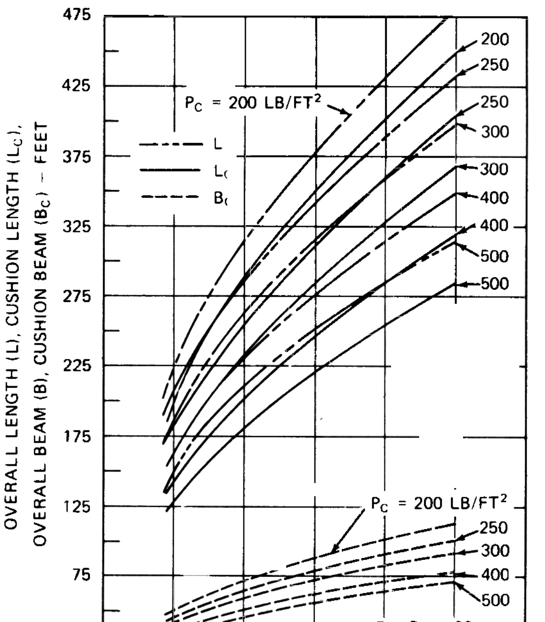


FIGURE 12-20

CUSHION AREA

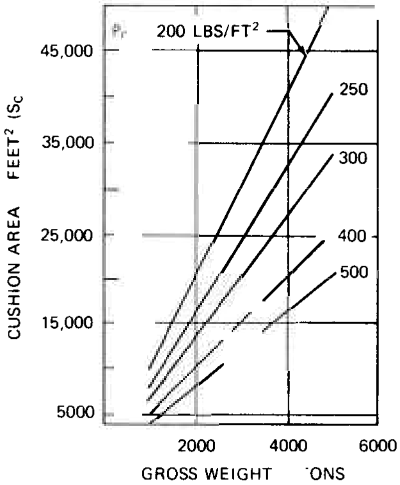


FIGURE 12–21

SIDEHULL LENGTH

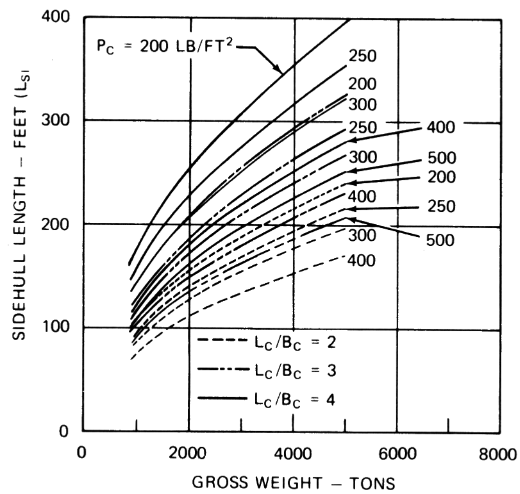


FIGURE 12–22

Figure 12–23 shows the effect of variations in cushion length–beam ratio on the specific power (horsepower required per pound of gross weight) as a function of SES speed. The power shown is the total power required for propulsion, lift and auxiliary power systems. For the cushion pressure assumed in drawing the figure, it can be seen that low values of cushion length–beam ratio are desirable.

The effect of cushion pressure variation relative to a fixed cushion length–beam ratio is shown in Figure 12–24, where it can be noticed that high values of cushion pressure produce relatively high power requirements at the hump speed. At speeds of 80 knots and above a cushion pressure–cushion length ratio of approximately 1.33 (P_C = 250 pounds per square foot) results in the lowest power requirements.

EFFECT OF LENGTH-BEAM RATIO ON POWER REQUIRED

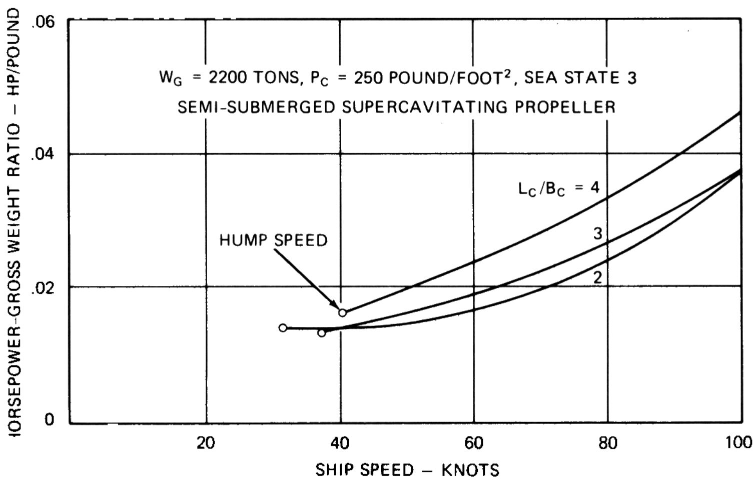


FIGURE 12-23

EFFECT OF CUSHION PRESSURE ON POWER REQUIRED

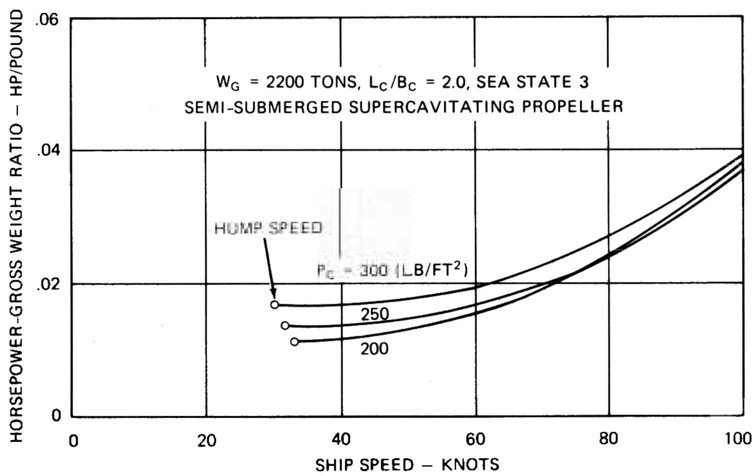


FIGURE 12-24

Figures 12-25 and 12-26 show the effect of cushion length-beam ratio and cushion pressure, respectively, on the payload-range curve for a typical SES design configuration. In these figures payload plus fuel is considered to constitute the disposable load, where the light ship weight plus disposable load equals the maximum gross weight of the ship. Therefore, at zero range the entire disposable load is payload, and at zero payload-gross weight ratio, it is all fuel. A low cushion length-beam ratio and a cushion pressure-cushion length ratio of approximately 1.33 are required to achieve high payload-gross weight ratios and favorable payload-range tradeoffs.

EFFECT OF LENGTH-BEAM RATIO ON PAYLOAD AND RANGE

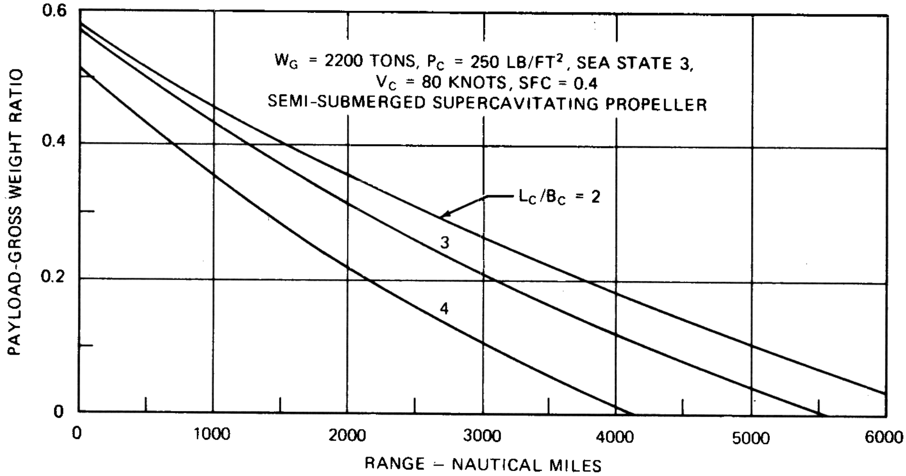


FIGURE 12–25

EFFECT OF CUSHION PRESSURE ON PAYLOAD AND RANGE

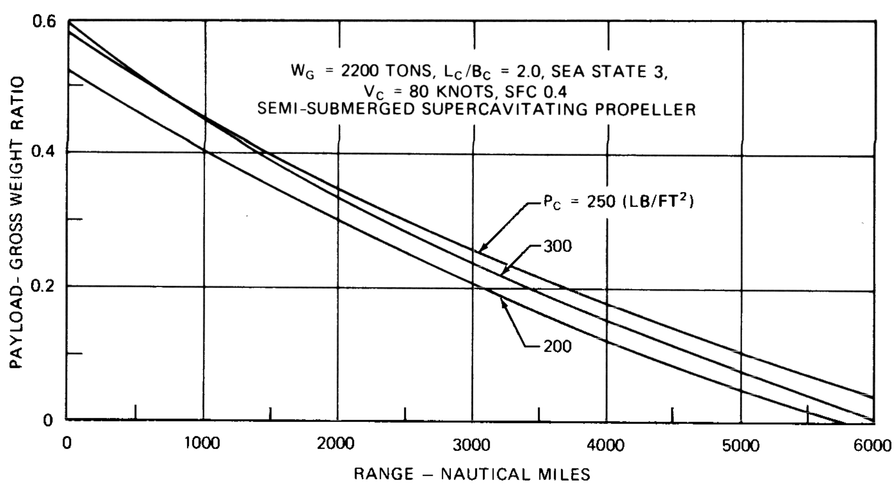


FIGURE 12–26

Figures 12–27 and 12–28 show the respective effects of cushion length–beam ratio and cushion pressure on the maximum range of a typical SES configuration, where the data are shown as a function of cruise speed. The cruise speed is defined as the high speed determining the propulsion and lift system sizes, providing some specified acceleration criterion is satisfied at hump. As in the case of the payload–range tradeoff curves, relatively low cushion length–beam ratios and P_C/L_C ratios of approximately 1.33 are evidently required to achieve high maximum range values. The effect of cruise speed on maximum range can also be noted in these two figures where the

EFFECT OF LENGTH-BEAM RATIO ON MAXIMUM RANGE

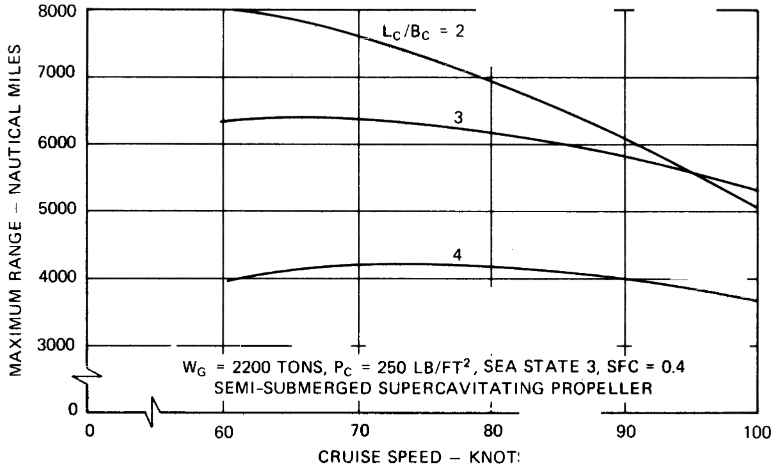


FIGURE 12–27

EFFECT OF CUSHION PRESSURE ON MAXIMUM RANGE

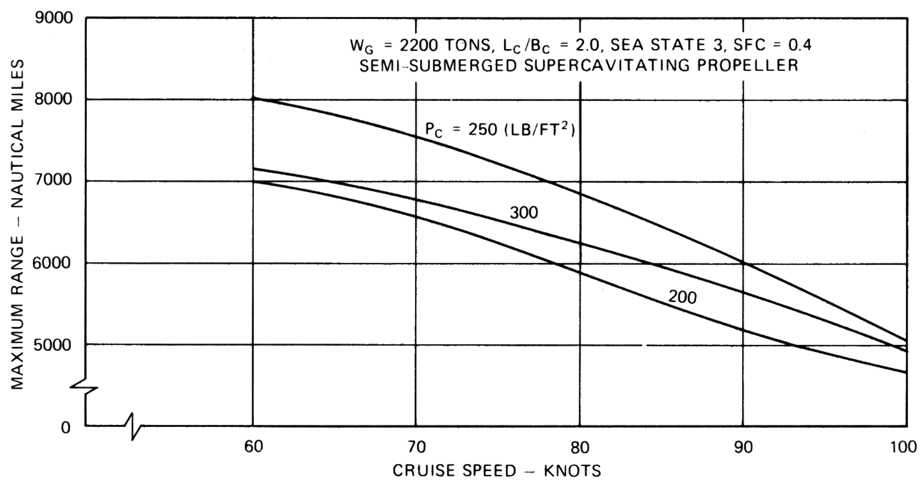


FIGURE 12–28

achievable maximum range at a design speed of 100 knots is shown to be on the order of only 60% to 65% of the range attainable at a design speed of 60 knots, for example. At 100 knots, a ship with a L_C/B_C = 3 has a greater range than one with a L_C/B_C = 2 due to the lower aerodynamic drag created by its reduced frontal area.

The effect of cushion length–beam ratio on another significant SES parameter can be noted in Figures 12–29 and 12–30, where the static roll stability margin, $\overline{GM}_T$, is shown as a function of SES gross weight. For the smaller ships, high length–beam ratios and high cushion pressures will result in ships of marginal stability.

EFFECT OF LENGTH-BEAM RATIO ON STATIC ROLL STABILITY MARGIN

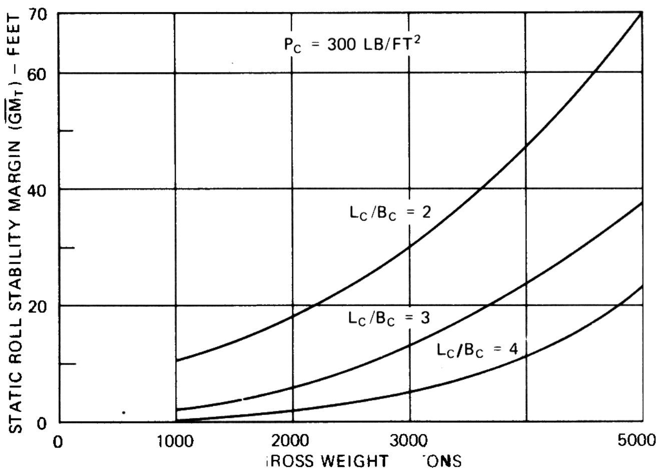


FIGURE 12-29

EFFECT OF CUSHION PRESSURE ON STATIC ROLL STABILITY MARGIN

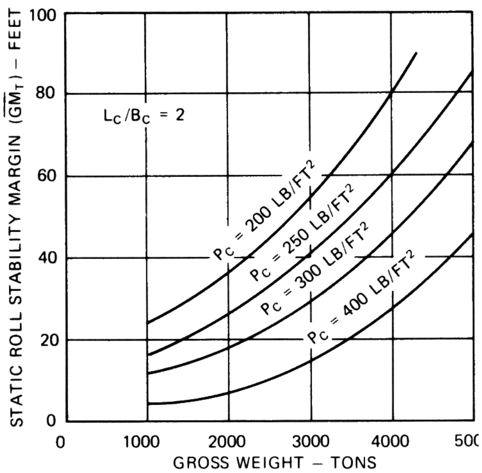


FIGURE 12-30

Effect of Operating Wave Height (Sea State) on Maximum Range. The effect of varying sea states, as described by the average wave height, on maximum range are shown as a function of design gross weight of the ship in Figure 12-31. The maximum range value is based on all weight available for payload being used for fuel weight.

Effect of Cushion Density. The cushion density, P_C/L_C (pounds/foot³), is not a separable input in generating SES parametric design data but is rather the result of various combinations of cushion pressure, cushion length-beam ratio, and gross weight inputs to the design. Typical cushion densities resulting from the designated inputs of cushion pressure, cushion length-beam ratio, and gross weight are shown in Figure 12-32. The principal design parameter affected by

EFFECT OF OPERATING WAVE HEIGHT ON MAXIMUM RANGE

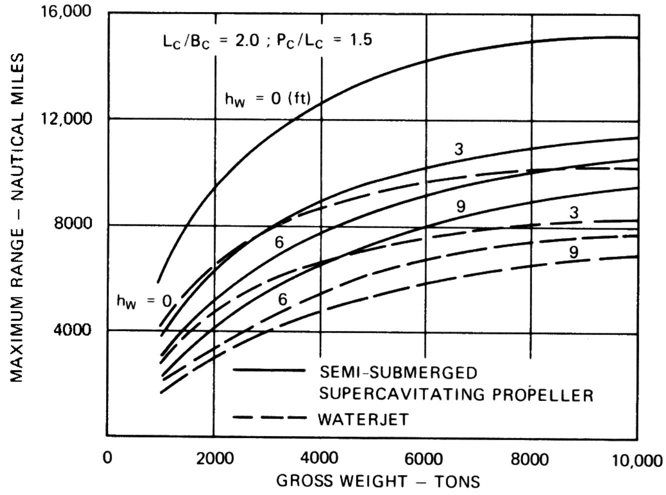


FIGURE 12-31

EFFECT OF CUSHION PRESSURE ON CUSHION DENSITY

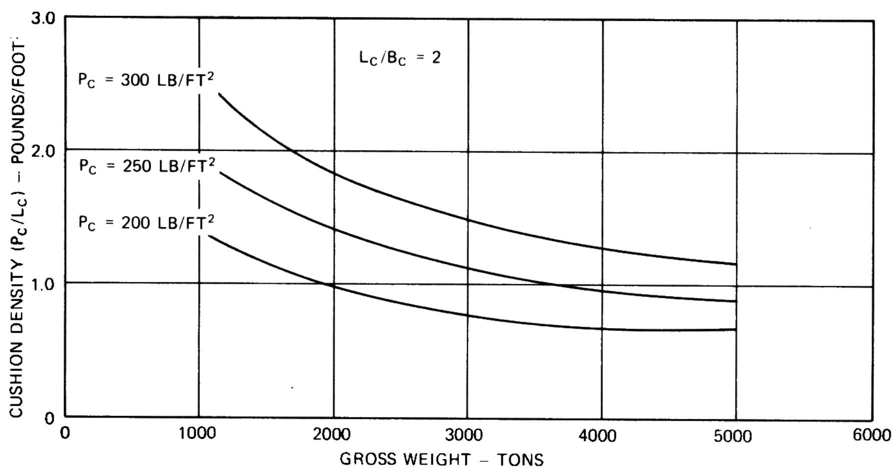


FIGURE 12-32

cushion density (besides those obviously affected by cushion pressure and cushion length-beam ratio, for example) is the structural weight of the SES. This relationship can be seen in its simplest form by visualizing the SES structure as a beam supported by a unit-loading cushion pressure. The longer the beam (cushion length, L_C) is, the greater the section modulus and the stiffness and, consequently, the weight required to ensure adequate strength and adequate resistance to deflection. Therefore, cushion density should be a principal determinant of structural weight, and this is borne out empirically by statistical analysis of the structural weight of known ACV's and SES's. Figure 12-33 shows the effect of cushion density on structural weight relative to the design gross weight of the ship.

EFFECT OF CUSHION DENSITY ON STRUCTURAL WEIGHT

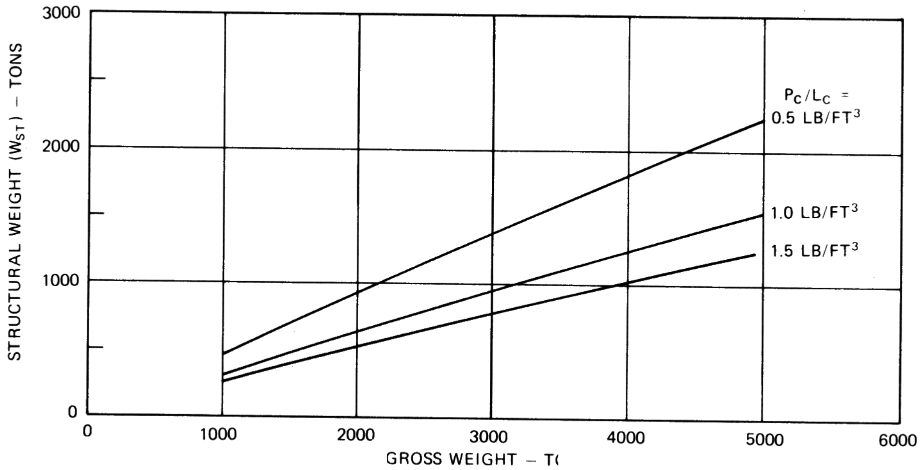


FIGURE 12-33

Effect of Power Plant Specific Weight, Specific Fuel Consumption and Net Propulsive Coefficient. Power plant specific weight, W_{pp}/HP, and specific fuel consumption, *SFC*, and net propulsive coefficient, *NPC*, have a direct relationship to SES range performance, as shown in Figures 12-34, 12-35 and 12-36. From these figures it can be deduced that the payload-range characteristics are more sensitive to variations in specific fuel consumption and net propulsive coefficient than to variations in specific weight; given equivalent percentage changes in specific fuel consumption, net propulsive coefficient and specific weight, the variation in specific fuel consumption and net propulsive coefficient result in the larger range change.

EFFECT OF POWER PLANT SPECIFIC WEIGHT ON PAYLOAD AND RANGE

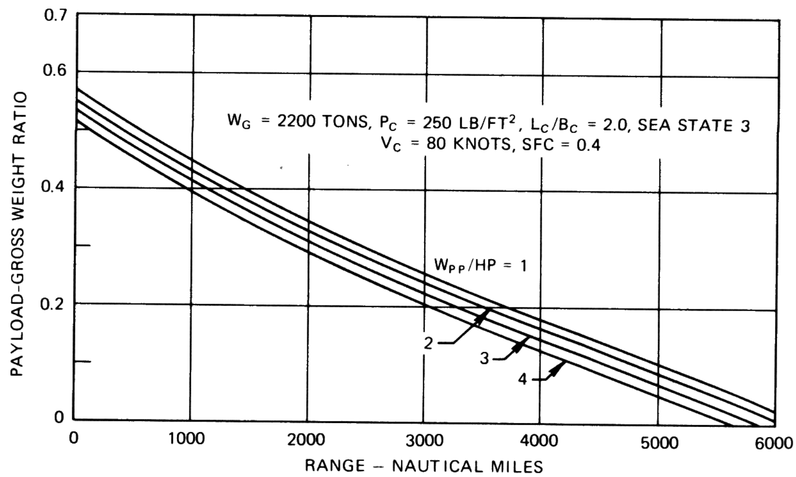


FIGURE 12-34

EFFECT OF POWER PLANT SPECIFIC FUEL CONSUMPTION ON PAYLOAD AND RANGE

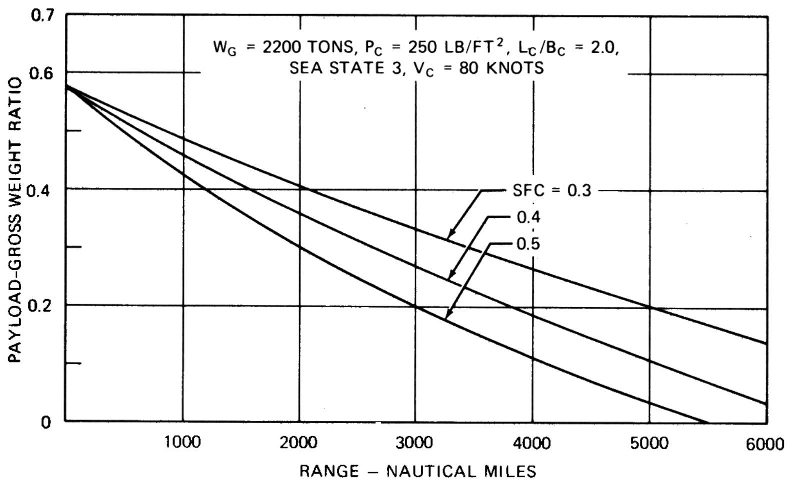


FIGURE 12-35

EFFECT OF NET PROPULSIVE COEFFICIENT ON PAYLOAD AND RANGE

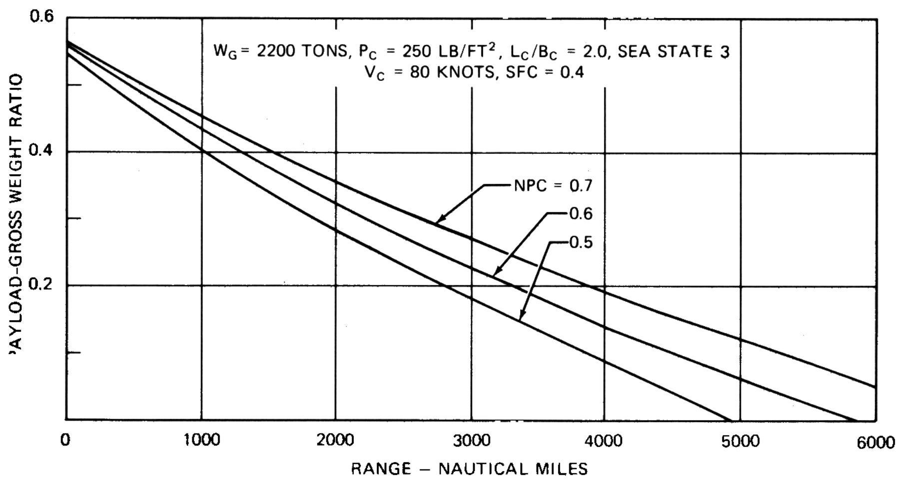


FIGURE 12-36

Chapter XIII
ANALYTICS

PERFORMANCE CHARACTERISTICS

The basic objective in the advanced design process is to examine and select concepts and configurations leading to minimum fuel loads and maximum ranges and payloads. It is evident at this point that the payload requirement is related to the solution of the general-weight equation, while the desired speed is related directly to the general-power-required expression. The range and endurance performance of the SES are related to several design characteristics, including speed, and, indirectly, to payload. An analytical approach to the derivation of general range and endurance equations is shown below.

Brequet Range and Endurance Equations. Define the following terms:

D_E = equivalent drag (including lift power equivalent) – pounds

E = endurance – hours

$(HP)_{REQ}$ = total horsepower required – hp

L = total lift – pounds

NPC = net propulsive coefficient

R = range – nautical miles

SFC = specific fuel consumption – pounds/hp-hour

T = total equivalent thrust required (including lift power equivalent) pounds

t = time – hours

V = ship speed – knots

W = instantaneous ship weight – pounds

$W_{EMPTY} = W_G - W_F$

W_F = fuel weight – pounds

W_G = ship gross weight – pounds

η_T = transmission efficiency.

As the SES consumes fuel, the time rate of weight change, dW/dt, is

$$dW/dt = -(SFC)(HP)_{REQ}$$

where the right-hand side of this equation is negative because the weight of the SES decreases as fuel is burned. Equation (13.1) can be written as

$$\int_{W_G}^{W_{EMPTY}} \frac{W}{(HP)_{REQ}} \frac{dW}{W} \qquad \int_{O}^{E} -(SFC)dt \tag{13.2}$$

$$dR = V\,dt \text{ and } R = \int_{O}^{E} V\,dt. \tag{13.3}$$

Using Equations (13.2) and (13.3) and performing the indicated integrations, assuming that V, SFC, and $W/(HP)_{REQ}$ are constants, results in

$$E = \frac{1}{\quad} \frac{W}{\quad} \ell n \left(\frac{1}{1 - W_F/W_G}\right) \tag{13.4}$$

$$R = \frac{V}{SFC} \frac{W}{(HP)_{REQ}} \ell n \left(\frac{1}{1 - W_F/W_G}\right) \tag{13.5}$$

Substituting the following identities,

$T = D_E$

$L = W$

$$(HP)_{REQ} = \frac{TV}{325\,NPC\,(\eta_T)}, \text{ or } \frac{D_E\,V}{325\,NPC\,(\eta_T)} \tag{13.6}$$

and assuming that NPC and the equivalent lift-drag ratio, $(L/D)_e$ are constants, Equations (13.4) and (13.5) can be rewritten as

$$E = 325(L/D)_e \left(\frac{NPC\,(\eta_T)}{SFC(V)}\right) \ell n \left(\frac{W_G}{W_{EMPTY}}\right) \tag{13.7}$$

$$R = 325(L/D)_e \left(\frac{NPC\,(\eta_T)}{SFC}\right) \ell n \left(\frac{W_G}{W_{EMPTY}}\right) \tag{13.8}$$

These are the basic Brequet endurance and range equations. It should be noted that

$$\frac{1}{1 - W_F/W_G} = \frac{W_G}{\quad} \tag{13.9}$$

Use of the Brequet Equations. The basic Brequet equations developed for the SES are similar to those used in calculating aircraft range and endurance. These basic equations, while not yielding as accurate an answer as would be obtained from an integration of the specific range or miles per pound of fuel curves, are adequate to estimate the endurance and range of an SES during the advanced design stages, and the convenience of use is significant. In this connection it should be noted that in Equations (13.7) and (13.8)

$$\frac{V}{(SFC)\,(HP)_{\mathrm{REQ}}} = \frac{325\;NPC\,(\eta_{\mathrm{T}})}{(SFC)\,(D)_{\mathrm{E}}} = \text{nautical miles/pound of fuel,}$$

respectively; thus, the Brequet equation accuracy can be improved if plots of miles per pound of fuel versus gross weight have been developed and if average values of the specific range are used. On the other hand, once such plots are available, only a fairly simple calculation is needed to integrate graphically, for example, and solve for range directly. In the final analysis, the assumptions made in deriving the Brequet equations are not overly restrictive for advanced design purposes, since the weight-to-power ratio for representative cruise conditions, engine throttle settings, and total fuel loads, for example, changes relatively slowly as fuel is burned.

GENERAL ADVANCED DESIGN METHOD

If greater flexibility in the selection of key design input parameters is desired, over and above that allowed by employment of the standard parametric procedure (where $L_C/B_C = 2.0, P_C/L_C = 2.5, V_C = 80$ knots, sea state = 3), a more general approach may be used.

The following sequential steps will provide solutions to the general advanced design problem.

1. For an assumed trial gross weight, W_G, and for assumed values of such design parameters as speed, cushion length-beam ratio, average wave height (sea state), cushion pressure and net propulsive coefficient, calculate the propulsive horsepower required, $(HP)_{\mathrm{Bp}}$, by summing the individual drag terms in the following manner to obtain total craft drag, D_{TOT}:

$$D_{\mathrm{TOT}} = q_{\mathrm{w}} S_{\mathrm{C}} \,[\,(C_{\mathrm{D}})_{\mathrm{WM}} + (C_{\mathrm{D}})_{\mathrm{SH}} + (C_{\mathrm{D}})_{\mathrm{A}} + (C_{\mathrm{D}})_{\mathrm{app}} + (C_{\mathrm{D}})_{\mathrm{prop}} + (C_{\mathrm{D}})_{\mathrm{wind}} + (C_{\mathrm{D}})_{\mathrm{ss}}\,]$$

where

q_{w} = dynamic pressure = $2.85\,V_{\mathrm{k}}^{2}$ – pounds/foot^2

V_{k} = ship speed – knots

S_{C} = cushion area – feet^2

$(C_{\mathrm{D}})_{\mathrm{prop}}$ = propulsor drag coefficient = 3×10^{-5}

$(C_{\mathrm{D}})_{\mathrm{wind}} = \left[\dfrac{2V_{\mathrm{w}}}{V_{\mathrm{k}}} + (V_{\mathrm{w}}/V_{\mathrm{k}})^2\right](C_{\mathrm{D}})_{\mathrm{A}}$

V_{w} = head wind velocity – knots

$(C_{\mathrm{D}})_{\mathrm{ss}} = (h_{\mathrm{w}}/L_{\mathrm{C}})^2$

h_{w} = average wave height – feet.

The drag coefficients for speeds above hump for wavemaking, $(C_D)_{WM}$, sidehull, $(C_D)_{SH}$, aerodynamic, $(C_D)_A$, and appendage, $(C_D)_{app}$, are displayed in Figures 12-12 through 12-17, wherein the frictional drag coefficient, C_f, is taken as 0.002. Momentum drag and stern thrust are assumed to be compensative, and seal drag is included in the sidehull drag and the drag increment due to waves.

Aerodynamic drag coefficients are shown in Figure 12-16 as a function of frontal-cushion area ratio (S_{front}/S_C). For a ship of length-beam ratio = 2, a typical value of S_{front}/S_C is 0.25. For narrower ships, the value of S_{front}/S_C will most likely be less than 0.25. To estimate frontal area, assume total frontal area (including superstructure) is 1.25 times the frontal area of the basic hull.

Geometric properties and ratios required as inputs to the data provided in these drag charts can be obtained from Figures 12-18 through 12-22. Drag can be converted to power by applying the following formula:

$$(HP)_{Bp} = \frac{D_{TOT}\ V}{550\ NPC\ \eta_T}$$

Estimate the required cushion power, $(HP)_C$, as follows:

$$\frac{Q(P_C + \Delta P_C)}{550\ \eta_f\ \eta_{TC}\ \eta_D} = 2.7 \times 10^{-3}\ Q\ P_C$$

where

$$h_w\ V - \text{feet}^3/\text{second.}$$

3 Obtain the total required horsepower, $(HP)_{REQ}$, by summing $(HP)_{Bp} + (HP)_C$ If the specific fuel consumption, SFC, is not identical for the propulsion and lift-fan engines, the total ship SFC can be estimated from the following expression:

$$SFC = \frac{(SFC)_{prop}\ (HP)_{Bp} + (SFC)_C\ (HP)_C}{(HP)_{REQ}}$$

where

$(SFC)_{prop}$ = specific fuel consumption for propulsion engines pounds/hp-hour

$(SFC)_C$ = specific fuel consumption for lift fan engines – pounds/hp-hour.

4. From the equations in Table 11-3, determine light ship weight, W_{LS}.
5. For the desired range, R, or endurance, E, speed, V; the value of specific fuel consumption, SFC, estimated from step 3; the desired fuel or range reserve

factors; and the total required horsepower, $(HP)_{REQ} = (HP)_{Bp} + (HP)_C$ from steps 1 and 2; calculate the fuel-gross weight ratio, W_F/W_G, using the applicable expression following.

Fuel Reserve Case:

$$\frac{W_F}{W_G} = \frac{1 - e^{-k}}{1 - a} \tag{13.10}$$

$$k = \frac{(R)(SFC)(HP)_{REQ}/W_G}{V_C} \quad \text{– range case}$$

$$k = (E)(SFC)(HP)_{REQ}/W_G \quad \text{– endurance case}$$

where

a = decimal fraction of total fuel assigned to reserve
e = base of natural system of logarithms = 2.7183
V_C = ship speed at cruise – knots.

Range, Endurance Reserve Case:

$$\frac{W_F}{W_G} = 2 - e^{-k} - e^{-k'} \tag{13.11}$$

$$k = \frac{(1 - b)(R')(SFC)(HP)_{REQ}/W_G}{V_C} \quad \text{– range case}$$

$$k' = \frac{b(R')(SFC)(HP)_{REQ}/W_G}{V_C} \quad \text{– range case}$$

$$k = (1 - b)(E')(SFC)(HP)_{REQ}/W_G \quad \text{– endurance case}$$

$$k' = (b)(E')(SFC)(HP)_{REQ}/W_G \quad \text{– endurance case}$$

where

b = decimal fraction to total (all-out) range (R') or endurance (E') assigned to reserve.

Depending on the accuracy desired, the SFC and horsepower-weight ratio at W_{EMPTY} can be determined and values that represent an average between gross and empty weight used when solving for k. This will require determining the $(HP)_{REQ}$ at W_{EMPTY} (holding speed constant) and determining the SFC at the reduced horsepower (SFC increases as horsepower decreases).

In the above, the fuel consumption associated with electrical power generation for basic ship functions has been considered to be negligible. If payload functions, however, require considerable electric power, the associated fuel requirements should be taken into account by appropriately increasing the previously determined $(HP)_{REQ}$.

To facilitate solution of the above expressions for fuel-gross weight ratio, a graph of e^{-k} versus various values of k is included as Figure 13-1.

RANGE AND ENDURANCE FUNCTION

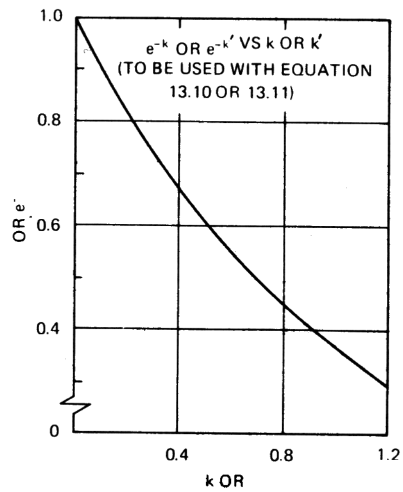


FIGURE 13-1

6. From the results of steps 2-5 above, calculate the available payload-gross weight ratio:

$$\left(\frac{W_{PL}'}{W_G}\right)_{AVAIL} = 1 - \frac{W_F}{W_G} + \frac{W_{LS}}{W_G} \qquad (13.12)$$

7 From the desired or specified value of payload and the gross weight assumed in step 1, calculate the required payload gross weight ratio:

$$\left(\frac{W_{PL}'}{W_G}\right)_{REQ} = \frac{W_{PL\ DESIRED}}{W_{G\ ASSUMED}} \qquad (13.13)$$

8. Compare steps 6 and 7. Repeat steps 1 through 7, if necessary, until the available and the required payload-gross weight ratios are in agreement consistent with the accuracy desired.
9. Using the methods outlined in Figures 7-3 and 7-4, check the static and turning roll stability to ensure adequate stability margins. If the margins prove unsatisfactory, repeat steps 1 through 8 above, assuming new values for the pertinent parameters, such as cushion pressure and length-beam ratio until satisfactory stability margins are attained.

Steps 1 through 9 above constitute the basic advanced design procedure. Having gone through these nine steps, it may be desirable to make such additional checks and iterations as the following:

a) Vary the assumed cushion pressure to determine an optimum value, that is, the design cushion pressure that results in the lowest gross weight or least total power for given performance specifications.
b) Calculate the power required and available at the hump, or critical speed of the ship and the corresponding acceleration performance in this speed regime. The hump speed occurs at a Froude number $(F_N = V/\sqrt{gL_C})$ of 0.7 when $L_C/B_C = 2.0$, and a F_N of 0.75 for $L_C/B_C = 3$ or 4.

The drag at hump speed can be determined by substituting the wavemaking drag determined from Equation (2.1) for the wavemaking drag calculated by using the $(C_D)_{WM}$ of Figure 12-15. A thrust margin of approximately 20% should be provided to ensure sufficient acceleration through the hump condition. The net propulsive coefficient at hump can be determined from Figure 4-1, entering the curve at the ratio of V_{hump}/V_C. The thrust available at hump speed may be determined using the intermittent rather than the continuous rating of the engine. The intermittent rating is approximately 20% greater than the continuous rating.

APPLICATION OF GENERAL ADVANCED DESIGN METHOD TO SAMPLE DESIGNS

In this section the calculations involved in the advanced design process are applied to three typical examples of basic SES design:

1 – SES with waterjet propulsion

2 – SES with semi-submerged supercavitating propeller propulsion

3 – SES with nuclear reactor and waterjet propulsion

The purpose here is to show the advanced design calculations in detail. In each case the rotating machinery, that is, the prime mover for propulsion and lift, is assumed to be of the gas turbine type. In the first two examples energy is assumed to be derived from the combustion of conventional hydrocarbon fuel, while in the last example a nuclear reactor is used as a heat source.

All three examples are designed to satisfy common specifications of range, speed, and payload, as stated in the individual example calculations. These are representative specifications to which no special mission implications should be attached. Further, no attempt has been made to optimize the example designs, as for example by searching for the particular values of length-beam ratio or cushion pressure resulting in a minimum gross weight design. For this reason, no conclusions as to the relative appropriateness of the various propulsor systems or heat sources should be drawn from these examples.

All the calculations shown were accomplished using only a slide rule and the design charts of Chapter XII. Therefore, accuracy only to three significant figures should be expected.

For a desired payload-range combination the first estimation of gross weight may be obtained from Figure 12-10 or 12-11.

Waterjet Design

Objectives

1 – Range: 4000 nautical miles at 80 knots and sea state 3 (average wave height, 3.0 feet). Zero fuel reserve.

2 – Payload: 1000 tons.

3 – Dimensional limitations: None except cushion length-beam ratio = 2.0.

4 – Power: Gas turbines for propulsion and lift engines buring JP-5-type fuel.

Assumptions

1 – Propulsion engine specific fuel consumption:

$$(SFC)_{\text{prop}} = 0.4 \text{ pound/horsepower-hour.}$$

2 – Lift engine specific fuel consumption:

$$(SFC)_{C} = 0.5 \text{ pound/horsepower-hour.}$$

Calculations

For a first trial, assume that gross weight, W_G, = 4000 tons and cushion pressure, P_C, = 300 pounds/feet^2. From Figures 12-18, 12-21, 12-22 obtain overall length, L, = 260 feet; cushion length, L_C, = 232 feet; cushion beam, B_C, = 116 feet; overall beam, B, = 136 feet; cushion area, S_C, = 26,800 feet^2; and wetted sidehull length, L_{SH}, = 178 feet. Assume that net propulsive coefficient, NPC, = 0.55; transmission efficiency, η_T, = 0.98; frontal area ratio, S_{front}/S_C, = 0.25; appendage area ratio, S_{app}/S_C, = 0.00685; and aerodynamic drag coefficient based on frontal area, C_{DA}, = 0.33

(a) Drag (D_{TOT}):

$$C_L = P_C/q_w = \frac{300}{} = 0.0165$$

$(C_D)_{WM}$ (Figure 12-15) = 0.000220

$$L_{SH}{}^2/S_C = \frac{(178)^2}{26{,}800} = 1.18$$

$(C_D)_{SH}$ (Figure 12-12) = 0.000154

$(C_D)_A$ (Figure 12-16) = 0.000095

$(C_D)_{\text{app}}$ (Figure 12-17) = 0.000055

$(C_D)_{prop}$ (assumed) = 0.000030

$$(C_D)_{wind} = \left[\frac{2(16)}{80} + (16/80)^2\right] (0.000095) = 0.000042$$

$(C_D)_{ss} = (3/232)^2 = 0.000167$

$$D_{TOT} = 2.85(80)^2 (26{,}800) [0.00022 + 0.000154 + 0.000095 + 0.000055 + 0.000030 + 0.000042 + 0.000167]$$

D_{TOT} = 373,000 pounds

(b) Propulsive horsepower $[(HP)_{Bp}]$

$$(HP)_{Bp} = \frac{373{,}000\,(80 \times 1.689)}{550\,(0.55)\,(0.98)} = 70{,}000 \text{ hp.}$$

2. Cushion horsepower $[(HP)_C$

(a) 80 knots, sea state 3 (use for range calculations):

$Q = (116)(3)(80 \times 1.689) = 47{,}000$ feet^3/second

$(HP)_C = 2.7 \times 10^{-3} (47{,}000)(300) = 38{,}000$ hp.

(b) 40 knots, sea state 6 (use for weight calculations)

$Q = (116)(9)(40 \times 1.689) = 70{,}500$ feet^3/second

$(HP)_C = 2.7 \times 10^{-3} (70{,}500)(300) = 57{,}200$ hp

3. Specific fuel consumption (SFC):

$$SFC = \frac{0.4\,(170{,}000) + 0.5\,(38{,}000)}{170{,}000 + 38{,}000} = 0.418$$

4. Light ship weight (W_{LS}) (see Table .-3):

(a) $P_C/L_C = (300/232) =$ .293

W_{ST} (also see Figure 3-17) = $0.269\,(4000)\,2{,}240$ = 2,410,000 pounds.

(b) Assume 4 engines:

$$W_{pp} = 4\left(\frac{170{,}000}{4}\right)\left(0.69 + \frac{84}{\sqrt{42{,}500}}\right) = 186{,}000 \text{ pounds.}$$

$$W_{tr_{WJ}} = 4\ [0.28\,(42{,}500) + 1200] = 52{,}500 \text{ pounds.}$$

Assume 4 pumps:

$$W_{WJW} = 4\,(410)\left[\frac{(170{,}000)\,(0.98)}{4} \times 10^{-3}\right]^{.46} = 378{,}000 \text{ pounds.}$$

Assume 4 fans:

$$W_{fs} = 4\left[3.3\left(\frac{57{,}200}{4}\right) + 89.2\ \sqrt{14{,}300}\right] = 232{,}000 \text{ pounds.}$$

(f) $W_{bs} = [3.3\,(4000) + 19{,}500]\ \frac{2}{2} = 32{,}700$ pounds.

(g) $W_{ss} = [3.69\,(4000) + 20{,}550]\ \frac{2}{2} = 35{,}330$ pounds.

(h) $W_{fin} = \frac{1.25\,(4000)}{100}\left(\frac{4000}{100} + 51.2\right) = 4{,}550$ pounds.

(i) $W_{aux} = 3500\,(4000)^{0.485} + 1060\,(25) + 0.25\ [170{,}000 + 57{,}200]$
$+ 57{,}000 = 335{,}700$ pounds.

(j) W_{LS} (less margin) $[\Sigma\ (a)\ to\ (i)] = 3{,}666{,}780$ pounds.

(k) $W_{MARG} = 0.05\,(3{,}666{,}780) = 183{,}300$ pounds.

(l) $W_{LS}\ [\Sigma\ (j) + (k)] = 3{,}850{,}080$ pounds.

(m) $W_{LS}/W_G = \frac{3{,}850{,}080}{2{,}240\,(4000)} = 0.430.$

5. Fuel-gross weight ratio (W_F/W_G):

Considering Equation (13.10), with a = 0 since no fuel reserves are required, calculate k:

$$k = \frac{(4000)\,(0.418 \times 1.05)\,(170{,}000 + 38{,}000)}{(80)\,(4000 \times 2240)} = 0.510.$$

(Above assumes horsepower weight ratio remains constant, but *SFC* increases an average of 5 percent as horsepower and weight decrease).

From Figure 13-1, read $e^{-0.510} = 0.602$. Equation (13.10) yields the following fuel-gross weight ratio:

$$\frac{W_F}{W_G} = 1 - 0.602 = 0.398$$

6. Available payload-gross weight ratio $(W_{PL}/W_G)_{AVAIL}$:

   Using Equation (13.12), the available payload-gross weight ratio can now be calculated:

$$\left(\frac{W_{PL}}{W_G}\right)_{AVAIL} = 1 - [0.398 + 0.430] = 0.172.$$

7. Required payload-gross weight ratio $(W_{PL}/W_G)_{REQ}$:

$$\left(\frac{W_{PL}}{W_G}\right)_{REQ} = \frac{1000}{4000} = 0.250.$$

8. A comparison of steps 6 and 7 shows the payload ratio available for the assumed W_G = 4000 to be less than that required. Therefore, another iteration through steps 1–7 is indicated, assuming a larger initial gross weight.

Semi-submerged Supercavitating Propeller Design

Objectives

1 – Range: 4000 nautical miles at 80 knots and sea state 3 (average wave height, 3.0 feet). Zero fuel reserve.

2 – Payload: 1000 tons

3 – Dimensional limitations: None except cushion length-beam ratio = 2.0

4 – Power: Gas turbines for propulsion and lift engines burning JP-5-type fuel.

Assumptions

1 – Propulsion engine specific fuel consumption:

$$(SFC)_{prop} = 0.4 \text{ pound/horsepower-hour.}$$

2 – Lift engine specific fuel consumption:

$$(SFC)_C = 0.5 \text{ pound/horsepower-hour.}$$

Calculations

For a first trial, assume that gross weight, W_G = 4000 tons and cushion pressure, P_C, = 300 pounds/foot^2. Use the same assumptions and Figures as in the preceding example, except assume that net propulsive coefficient, NPC, = 0.68 and transmission efficiency, η_T, = 0.97 for a propeller.

1. (a) Drag (see Waterjet Design):

= 373,000 pounds.

(b) Propulsive horsepower $(HP)_{Bp}$:

$$(HP)_{Bp} = \frac{373{,}000\,(80 \times 1.689)}{550\,(0.68)\,(0.97)} = 140{,}000 \text{ horsepower.}$$

2. Cushion horsepower (see Waterjet Design)

80 knots, sea state 3:

$(HP)_C$ = 38,000 horsepower.

(b) 40 knots, sea state 6:

$(HP)_C$ = 57,200 horsepower.

3. Specific fuel consumption (SFC):

$$SFC = \frac{0.4\,(140{,}000) + 0.5\,(38{,}000)}{140{,}000 + 38{,}000} = 0.421.$$

4. Light ship weight (W_{LS}) (see Table 11-3 and Waterjet Design):

(a) W_{ST} = 2,410,000 pounds

(b) Assume 4 engines:

$$W_{pp} = 4\left(\frac{140{,}000}{4}\right)\left(0.69 + \frac{84}{\sqrt{35{,}000}}\right) = 160{,}000 \text{ pounds}$$

(c) $W_{tr_p} = 2\,[0.85\,(70{,}000) + 2{,}500]$ = 124,000 pounds.

(d) Assume 2 propellers:

$$W_{prop} = 2\,(68)\left[\frac{140{,}000\,(0.97)}{2} \times 10^{-3}\right]^{1.26} = 27{,}600 \text{ pounds}$$

(e) W_{fs} = 232,000 pounds.

(f) W_{bs} = 32,700 pounds.

(g) W_{ss} = 35,330 pounds.

(h) $W_r = 0.9\,(4000) - 550$ = 3,050 pounds.

(i) $W_{aux} = 5{,}500\,(4000)^{0.4} + 1060\,(25) + 0.25\,[140{,}000 + 57{,}200] + 61{,}000$ = 288,700 pounds.

(j) W_{LS} (less margin) $[\Sigma\ (a)\ to\ (i)]$ = 3,313,380 pounds.

(k) $W_{MARG} = 0.05\,(3{,}313{,}380) = 165{,}800$ pounds.

(l) $W_{LS}\ [\Sigma\ (j) + (k)]$ = 3,479,180 pounds.

(m) $W_{LS}/W_G = \dfrac{3{,}479{,}180}{2240\,(4000)} = 0.387.$

5. Fuel-gross weight ratio (see Waterjet Design):

$$k = \frac{(4000)\,(0.421 \times 1.05)\,(140{,}000 + 38{,}000)}{(80)\,(4000 \times 2240)} = 0.439$$

$$e^{-0.439} = 0.646$$

$$\frac{W_F}{W_G} = 1 - 0.646 = 0.354$$

6. Available payload-gross weight ratio (see Waterjet Design):

$$\left(\frac{W_{PL}}{W_G}\right)_{AVAIL} = 1 - [0.354 + 0.387] = 0.259.$$

7. Required payload-gross weight ratio $(W_{PL}/W_G)_{REQ}$:

$$\left(\frac{W_{PL}}{W_G}\right)_{REQ} = \frac{1000}{4000} = 0.25.$$

8. Comparing steps 6 and 7 shows the payload ratio available for the assumed $W_G = 4000$ to be approximately equal to that required. Therefore, no further iterations are indicated.

A note of caution is interjected here: no direct comparison of the propeller and waterjet design examples in terms of "which is better" should be made on the basis of the limited information developed here. For example, neither of the two designs has been optimized in terms of cushion length-beam ratio or cushion pressure variation to determine a minimum gross weight. Therefore, these designs, which are presented solely for the purpose of explaining the advanced design procedure, should not be used to infer, for example, differences between the two assumed propulsion systems.

Nuclear Reactor and Waterjet Propulsion

Objectives

1 – Range: Limited only by requirement of nuclear fuel.
Cruise at 80 knots and sea state 3 (average wave height, 3.0 feet).

2 – Payload: 1000 tons.

3 – Dimension limitations: None except cushion length-beam ratio = 2.0.

4 – Power: Closed-cycle, gas-cooled reactor. Gas turbine propulsion and lift engines.

Calculations

For a first trial, assume that gross weight, W_G, = 4000 tons and cushion pressure, P_C, = 300 pounds/foot^2. Use the same assumptions and Figures as in the first example.

(a) Drag (see Waterjet Design):

373,000 pounds.

(b) Propulsive horsepower (see Waterjet Design):

= 170,000 horsepower.

2. Cushion horsepower (see Waterjet Design)

(a) 80 knots, sea state 3

38,000 horsepower

(b) 40 knots, sea state 6

$(HP)_C$ = 57,200 horsepower.

3. Specific fuel consumption (not applicable).

4. Light ship weight (W_{LS}) (see Table .-3 and Waterjet Design):

(a) W_{ST} = 2,410,000 pounds.

(b) $W_{NS} = 5.4\,(170{,}000 + 57{,}200) + 2.3\,(170{,}000) + 2.4\,(57{,}200)$ = 1,755,000 pounds.

(c) W_{bs} = 32,700 pounds.

(d) W_{ss} = 35,330 pounds.

(e) W_{fin} = 4,550 pounds.

(f) $W_{aux} = 3500\,(4000)^{0.485} + 1060\,(25) + 57{,}000$ = 278,950 pounds.
(weight of fuel system not applicable)

(g) W_{LS} (less margin) [Σ *(a) to (f)*] = 4,517,390 pounds.

(h) $W_{MARG} = 0.05\,(4{,}517{,}390)$ = 225,870 pounds.

(i) $W_{LS} = [\Sigma\,(g) + (h)] = 4{,}743{,}260$ pounds.

(j) $W_{LS}/W_G = \dfrac{4{,}743{,}260}{2240\,(4000)} = 0.528$

5. Fuel-gross weight ratio (not applicable)

6. Available payload-gross weight ratio (see Waterjet Design)

$$\left(\frac{W_{PL}}{W_G}\right)_{AVAIL} = 1 - [0.528] = 0.472$$

7. Required payload-gross weight ratio $(W_{PL}/W_G)_{REQ}$

$$\left(\frac{W_{PL}}{W_G}\right)_{REQ} = \frac{1000}{4000} = 0.250$$

8. A comparison of steps 6 and 7 shows the payload ratio available for the assumed $W_G = 4000$ to be larger than that required. Therefore, another iteration through steps 1–7 is indicated assuming a smaller initial gross weight.

APPLICATION OF GENERAL ADVANCED DESIGN METHOD TO OTHER TYPICAL DESIGN AND PERFORMANCE CHARACTERISTICS

In addition to the calculation procedures employed in the sample designs, the design procedure can be the basis for other calculations related to the advanced design process and to the prediction of performance for a desired ship configuration. To demonstrate this and to show the utility and versatility of the approach, three typical calculation procedures or situations are analyzed below in outline form.

Operation in High Sea States at Reduced Speed. In the sample designs, the total installed power is determined by the power required for propulsion at 80 knots in a sea state 3 and for lift at 40 knots in a sea state 6. As an alternate approach, the propulsion and lift systems can both be designed for 80 knots, sea state 3. With this total installed power the ship should be able to operate in a sea state somewhat higher than in the sample designs but at a speed less than the specified speed. The calculation to be made thus involves finding the speed and sea state conditions where the specific power is the same as the specific power installed in the ship. This relationship can be found by estimating the horsepower required for higher operating wave heights – sea state 5, for example. At this point in the calculations, and particularly if separate lift and propulsion engines have been assumed, the split between propulsion and lift power must be determined and a check made to see whether the installed cushion (lift) power is adequate. If it is not, another combination of speed and sea state requiring a cushion power equal to that on board must be chosen or additional cushion power must be assumed, with a corresponding adjustment made in the light ship weight. As can be surmised from this discussion, the assumption of an integrated propulsion and lift system obviates the need for the cushion power check if the addition of extra power is not desired.

Optimum Cruising Speed. Determination of the optimum cruising speed – the speed resulting in maximum range – for a given configuration of SES can be facilitated by using the sample design methods and assumed engine characteristics and the range-performance method presented in Chapter IV. In most cases the analysis will indicate a schedule of speed as fuel is burned which maximizes range performance. This variation of speed may not be large and, in any event, an indication of the average speed for maximum range can be determined from a plot of specific power versus speed. Once such a plot is made, the point of tangency of a ray from the origin to the specific power curve determines the speed at which $(V)\ [(HP)_{REQ}/W_G]^{-1}$ is a maximum. From the Brequet Range Equation, it can be seen that range will be a maximum when this quantity is maximum; thus other than for the influence of variations in specific fuel consumption with speed and power level, the speed derived in the manner just described is a good approximation of the optimum cruise speed.

Integrated Versus Separate Propulsion and Lift Power. In the advanced design process, either integrated or separate propulsion and lift power sources can be assumed. In the sample designs separate engines were included as can be seen by the assumption of different values of specific fuel consumption for the propulsion and lift engines. If an integrated system is assumed, a common value of fuel consumption can be used. Some of the other implications of integrated versus separate power systems have been mentioned above in the discussion of operations at sea states higher than design.

Appendix A
SYMBOLS

NOTE Dimensional units are assigned to each symbol. Dimensionless quantities are indicated by "nd."

A — Area, general – in^2 or ft^2

A_b — Bow seal waterplane area at static pitch angle – ft^2

A_i — Inlet flow area – ft^2

A_N — Normal acceleration – ft/sec^2

A_o — Propeller disc area – ft^2

A_s — Stern seal waterplane area at static pitch angle – ft^2

A_T — Tangential acceleration – ft/sec^2

A_{wp} — Waterplane area corresponding to equilibrium condition – ft^2

a_x — Linear acceleration in g's at point x feet from center of gravity of ship – nd

a_z — Vertical acceleration of center of gravity in g's – nd

B — Overall beam of ship – ft
Pump impeller width – ft

$\overline{B}$ — Resultant upward force due to buoyancy and cushion pressure – lbs

b — Number of branches transmitting torque – nd

B_C — Cushion beam – ft

$\overline{BM}_L$ — Longitudinal metacentric radius – ft

$\overline{BM}_T$ — Transverse metacentric radius – ft

B_{SH} — Sidehull beam – ft

B_{SHW} — Sidehull wetted beam – ft

BTU — British thermal units – BTU

B_w — Wave excitation – ft

c — Loss coefficient – nd

C_D — Drag coefficient – nd
Overall derating factor – nd

C_{DA} — Aerodynamic drag coefficient – nd

$(C_D)_A$ — Aerodynamic drag coefficient based on cushion area – nd

C_{Da}, $(C_D)_{app}$ — Appendage drag coefficient – nd

C_{DC} — Cavity drag coefficient – nd

C_{Dfc} — The Schoenherr appendage friction factor, based on appendage Reynolds number – nd

C_{Di} — Appendage induced drag coefficient – nd

$(C_D)_{prop}$ — Propulsor drag coefficient – nd

C_{DS} — Seal drag coefficient – nd

$(C_D)_{SH}$ — Sidehull drag coefficient – nd

$(C_D)_{ss}$ — Drag coefficient due to sea state (waves) – nd

$(C_D)_{wind}$ — Drag coefficient due to wind – nd

$(C_D)_{WM}$ — Wavemaking drag coefficient – nd

C_f — Frictional drag coefficient – nd

C_{fs} — Schoenherr friction factor – nd

$C.G$ — Center of gravity

C_K — Contact stress factor – nd

℄ — Centerline of ship

C_L — Lift coefficient – nd

C_{PC} — Cushion pressure coefficient – nd

C_r — Sidehull wavemaking drag coefficient – nd
Friction factor – nd

C_{TP}	Thrust loading coefficient – nd
D	Diameter, general – ft or in Drag, general – lbs
d	Thickness – ft Gear or pinion diameter – in Duct diameter – in Internal diameter of pipe – in
D_A	Aerodynamic drag – lbs
D_C	Casing diameter, waterjet pump – ft Air leakage discharge coefficient – nd
D_{ccs}	Appendage cavity drag – lbs
D_{cs}	Appendage or control surface drag – lbs
D_E	Equivalent drag (including lift power equivalent) – lbs
D_{fcs}	Appendage frictional drag – lbs
d_g	Gear diameter – in
D_h	Fan hub diameter – ft
D_i	Fan inlet diameter – ft
D_{ics}	Appendage induced drag – lbs
D_M	Propulsor mounting drag – lbs
D_m	Momentum or ram drag – lbs
D_o	Outside diameter of shaft – in
D_p	Waterjet propulsion system drag – lbs
d_p	Pinion diameter – in
D_{prop}	Propulsor drag – lbs
D_{RW}	Drag increment due to waves – lbs Rough-water drag increment – lbs
D_S	Total seal drag – lbs
D'_s	Specific diameter – ft
D_{SH}	Sidehull drag – lbs
D_{SHF}	Total sidehull frictional drag – lbs Total sidehull wavemaking drag – lbs
D_{SP}	Sidehull form drag – lbs
D_t	Total measured drag – lbs
D_{TOT}	Total craft drag – lbs
D_W	Fan wheel width – ft
D_{wind}	Drag due to wind – lbs
D_{WM}	Cushion wavemaking drag – lbs Wavemaking drag – lbs
dW/dt	Time rate of weight change
E	Endurance – hrs
e	Overall efficiency of gear box or bearing – nd Base of natural system of logarithms
EAR	Expanded area ratio – nd
e_H	Volute maximum height – ft
e_L	Volute maximum length – ft
$env.$	Envelope
e_o	Volute outlet height – ft
e_w	Volute maximum width – ft
F	Gear face width – in Force, general – lbs
f	Frequency, general – Hz ABS bending moment factor – nd
f_ℓ	Cushion wavemaking drag coefficient – nd
F_N	Froude number – nd
$F_{N0.7}$	Froude number based on velocity at 70% radius of propeller blade – nd
F_p	Pinion face width – in
FR	Frequency factor – cycles/second/foot
FS	Safety factor – nd
G	Ratio of seal length to periphery length – nd
g	Gravitational acceleration – ft/sec^2
$\overline{GM}_L$	Longitudinal metacentric height – ft
$\overline{GM}_T$	Transverse metacentric height (static roll stability margin) – ft
gpm	Gallons per minute
GRP	Glass reinforced plastic
g_{ZL}	Limit value of vertical acceleration – ft/sec^2
H	Pump head per stage – ft
h	Head – feet of fuel oil (bottom of tank to discharge level) – ft

ΔH	Total pump head rise – ft
H_a	Atmospheric head – ft
h_a	Appendage depth or span – ft
h'_a	Mean wetted depth – ft
	One-half average wave height (h_w) – ft
h_b	Water depression due to cushion pressure – ft
H_C	Seal clearance height – ft
HP	Horsepower, general – hp
$(HP)_{ACC}$	Accessory (auxiliary) power required – hp
$(HP)_B$	Brake horsepower – hp
$(HP)_{Bf}$	Fan brake horsepower – hp
$(HP)_{Bp}$	Propulsion brake horsepower – hp
$(HP)_{BT}$	Total ship brake horsepower – hp
$(HP)_C$	Cushion fan horsepower – hp
$(HP)'_C$	Cushion horsepower, per fan – hp
hp_i	Horsepower input to gear box or bearing – hp
$(HP)_P$	Pumping power – hp
$(HP)_{REQ}$	Total horsepower required – hp
$(HP)_S$	Shaft horsepower – hp
hr	Hours
H_S	Ship depth or height above waterline – ft
h'_s	Average wetted sidehull draft – ft
H_{SH}	Height of sidehull – ft
H_v	Water vapor head – ft
h_w	Wave double amplitude – ft
	Average wave height – ft
I	Mass moment of inertia – lb-sec^2-ft
$\bar{I}$	Moment of inertia of total section – in^2-ft^2
I_{Lsh}	Longitudinal moment of inertia of the two sidehull waterplane areas taken about the center of flotation (centroid of waterplane area) – ft^4
i_o	Moment of inertia about center of member – in^2-ft^2
I_{Tsh}	Transverse moment of inertia of the two sidehull waterplane areas taken about vehicle centerline – ft^4
I_{xx}	Mass moment of inertia about X-axis – lb-sec^2-ft
J	Advance coefficient – nd
J_d	Selected advance coefficient – nd
K	Heeling moment – ft-lbs
	Load carrying capacity – lbs-in^2
	Radius of gyration – ft
k	Gas constant, ratio of specific heats – nd
	Constant
	Range or endurance function
$\overline{KB}$	Height of center of buoyancy (centroid of displaced volume) above reference line – ft
k_d	Waterjet duct loss coefficient – nd
$\overline{KG}$	Height of center of gravity above reference line – ft
kip	1000 pounds
K_M	Desired thrust margin – nd
$\overline{KM}_T$	Height of transverse metacenter – ft
K_n	Waterjet nozzle loss coefficient – nd
K_Q	Torque coefficient – nd
K_T	Thrust coefficient – nd
L	Length, general – ft
	Overall ship length – ft
	Total lift – lbs
ℓ	Deflected length – in
	Spring contact length – in
ℓ_a	Appendage length or chord – ft
lb	Pound
L_C	Cushion length – ft

	Cushion length-beam ratio – nd
	Longitudinal center of gravity – ft
L/D	Lift-drag ratio – nd
$(L/D)_e$	Effective lift-drag ratio – nd
L_f	Fan length – ft
L_{SH}	Sidehull length – ft
L_{SHW}	Sidehull wetted length – ft
M	Pitch moment – ft-lbs
	Trimming moment – ft-lbs
	Bending moment – ft-lbs
m	Mass of the ship – lb-sec^2-ft
m_a	Mass flow of air – lbs/sec
MAX, max	Maximum
M_{EX}	External moment – lb-ft
M_g	Input gear ratio – nd
mh_w	Maximum wave height – ft
M_{MAX}	Maximum bending moment – ft-kips
	Overall gear ratio – nd
	Measured value of rough-water drag – lbs
	Roll moment created by seals – ft-lbs
$M_{s_{MAX}}$	Maximum spanwise bending moment – ft-kips
	Measured value of smooth-water drag – lbs
	Propeller blade mean width ratio – nd
N	Number, general – nd
n	Rotative speed – rpm
N_{cr}	Number of crew
n_e	Power turbine speed – rpm
NPC	Net propulsive coefficient – nd
$NPSH$	Net positive suction head – ft
N_R	Number of teeth in ring gear – nd
N_S	Number of teeth in sun gear – nd
	Specific speed
P	Pressure, general – lbs/in^2
ΔP	Total pressure duct loss – in of H_2O
	Total head loss – in of H_2O
	Atmospheric pressure – lbs/ft^2

P_B	Bag pressure – lbs/in^2
P_c	Pressure within propeller blade cavities – lbs/in^2
	base pressure – lbs/in^2
P_C	Design pressure – lbs/ft^2
	Cushion pressure – lbs/ft^2
P'_C	Cushion pressure variation relative to average pressure – lbs/ft^2
ΔP_C	Variation in cushion pressure – lbs/ft^2
P_{CE}	Cushion pressure for zero fuel condition – lbs/ft^2
P_{CL}	Cushion pressure for fully loaded ship – lbs/ft^2
P_C/L_C	Cushion density – lbs/ft^3
P_f	Fan discharge total pressure – lbs/in^2
P_H	Hold-down pressure – lbs/in^2
P_o	Ambient pressure – lbs/ft^2
P_S	Static pressure – lbs/in^2
	Preload spring pressure – lbs/in^2
ΔP_s	Static head gain – in of H_2O
P_{th}	Reactor power – megawatts
ΔP_V	Pressure loss at duct inlets, exits, and changes in area (dynamic pressure loss) – in of H_2O
Q	Cushion leakage, or cushion air flow rate – ft^3/min or ft^3/sec
	Heat loss – BTU/hr
	Pump flow rate – ft^3/sec
q_a	Dynamic pressure of air – lbs/ft^2
Q_f	Air flow requirement for each fan – ft^3/sec
Q_p	Propeller torque – lb-ft
q_w	Dynamic pressure of water – lbs/ft^2
R	Radius, general – ft
	Turning radius – ft
	Inlet velocity ratio – nd
	Range – nau. miles
	Pressure ratio – nd
	Gas constant (53.3 for air) – ft-lb/lb-°R

r	Waterjet velocity ratio – nd
R_e	Reynolds number – nd
R_o	Total resistance at initial steady ahead ship's speed at start of stopping maneuver – lbs
rpm	Revolutions per minute
rps	Revolutions per second
r_T	Transmission reduction ratio – nd
S	Stopping distance, or head reach – ft
	Suction specific speed
	Specific gravity – nd
$\overline{s}$	Stress coefficient – nd
S_{ac}	Allowable compressive stress – lbs/in^2
S_{app}	Appendage plan area – ft^2
	Appendage or rudder area – ft^2
S_C	Cushion area – ft^2
S_d	Allowable, average stress – lbs/in^2
sec	Seconds
SFC	Specific fuel consumption – lbs/hp-hr
SFC_C	Specific fuel consumption of cushion system – lbs/hp-hr
SFC_p	Specific fuel consumption of propulsion system – lbs/hp-hr
S_{front}	Ship frontal area – ft^2
S_g	Cushion discharge area – ft^2
SR	Scaling ratio – nd
S_r	Reference area for aerodynamic drag – ft^2
SW	Smooth-water value – nd
	Saltwater
T	Thrust, general – lbs
	Temperature, general – °F or °R
	Torque – lbs-ft in-lbs
	Tip force at bow seal – lbs
	Maximum torsional moment – ft-kips
t	Time, general – sec or hrs
	Thrust deduction – nd
$\overline{T}$	Resultant moment about center of gravity of ship – lb-ft

ΔT	Change in temperature – °F or °R
t_a	Appendage thickness – ft
t_a/ℓ_a	Thickness-chord ratio – nd
T_e	Equivalent torque – in-lbs
TIT	Turbine inlet temperature – °F
T_n	Net thrust – lbs
ton	2,240 pounds
T_p	Propeller open water thrust – lbs
TPI	Tons per inch immersion – tons/in
T_1	Astern thrust – lbs
V	Ship speed – ft/sec
V_a	Advance speed – ft/sec
V_C	Cruise speed – knots
$\overline{V_C}$	Cushion volume or average value of cushion volume – ft^3
VCG	Vertical center of gravity – ft
V_{crit}	Critical or hump speed – knots
V_D	Design speed – knots
V_d	Duct velocity, duct inlet velocity – ft/sec
V_{hump}	Hump speed – knots
V_{HW}	Head wind velocity (27 in sea state 3, 47 in sea state 6) – ft/sec
V_k	Ship speed – knots
V_m	Pump meridional velocity – ft/sec
V_o	Initial steady ahead ship's speed at start of stopping maneuver – ft/sec
	Craft velocity relative to air – ft/sec
$V_{r_{0.7}}$	Resultant velocity of propeller blade at 70% radius – ft/sec
V_T	Fan tip speed – ft/sec
V_w	Component of wind velocity parallel to ship's course – knots
W	Weight, general – lbs
	Weight of coupling – lbs
	Unit weight of shaft – lbs/ft
	Instantaneous ship weight – lbs
w	Wake fraction – nd
W_A	Weight of lifting fan housing – lbs

W_{AM}	Weight of lifting fan drive mechanism – lbs
W_{aux}	Weight of auxiliary systems – lbs
W_{bs}	Weight of bow seal – lbs
W_c	Weight of intercooler – lbs
W_C	Weight of fan wheel and enclosure – lbs
W_{CM}	Weight of fan drive shaft, bearings and supports – lbs
W_D	Weight of duct – lbs
W_d	Weight of ducting for turbine installation – lbs
W_{da}	Weight of air ducting – lbs
W_{dc}	Weight of coolant ducting – lbs
W_e	Weight of turbine enclosure – lbs
	Weight of engine – lbs
W_{EMPTY}	Empty weight (zero fuel condition) – lbs
W_F	Weight of fuel – lbs
W_f	Total weight of lift fan – lbs
	Fuel flow – lbs/hr
W_{fc}	Fuel consumption of the cushion or lift system – lbs/hr
W_{fe}	Weight of fire extinguishing equipment – lbs
W_{fin}	Weight of fin(s) – lbs
W_{fp}	Fuel consumption of the propulsion system – lbs/hr
W_{fs}	Total weight of fan system – lbs
W'_{fs}	Weight of fan system, per fan – lbs
W_{FSYS}	Weight of fuel system – lbs
W_{fT}	Total fuel consumption – lbs/hr
W_G	Gross weight of ship – lbs or tons
$W_{G_{MAX}}$	Maximum gross weight of ship – lbs or tons
W_{ℓ}	Weight of lube oil system – lbs
$W.L.$	Waterline
W_{LS}	Lightship weight – lbs
W_{MARG}	Weight of margin – lbs
W_{NS}	Weight of nuclear system – lbs
W_{PL}	Weight of payload – lbs
W_{pp}	Weight of power plant, per engine – lbs
W_{pp}/HP	Specific weight of power plant – lbs/hp
W_{PROP}	Weight per propeller – lbs
W_{PS}	Weight of fan power system – lbs
W_r	Weight of rudder – lbs
W_R	Weight of reactor – lbs
W'_R	Weight of reactor based on reactor weight density and reactor power density – lbs
W_{Re}	Weight of recuperator – lbs
W'_{RS}	Weight of reactor plus shield – lbs
W_s	Weight of engine starting system – lbs
W_{SH}	Weight of shield – lbs
W'_{SH}	Weight of shield based on reactor power density – lbs
W_{ss}	Weight of stern seal – lbs
W_{ST}	Structural weight – lbs
W_t	Weight of gas turbine installation – lbs
W_{tp}	Weight of propulsor and transmission – lbs
W_{tr}	Weight of transmission system, per propeller – lbs
W_{VAR}	Variable load – lbs
W_{WJW}	Weight of wet waterjet – lbs
x	Unit width of seal fabric – nd
X_b	Longitudinal distance of bow seal from center of flotation – ft
X_s	Longitudinal distance of stern seal from center of flotation – ft
$X_{\dot{u}}$	Longitudinal added mass – lb-sec^2/ft
Z	Section modulus – in^2-ft
	Waterjet pump centerline elevation head, above water surface – ft
z_{imm}	Immersion of sidehull – ft

Z_{REQ}	Required section modulus – in^2-ft
α (alpha)	Angular acceleration of craft – rad/sec^2
α_a	Appendage angle of attack – rad
β (beta)	Percentage submergence of propeller – nd
η_c (eta)	Conical diffuser effectiveness – nd
η_D	Duct efficiency – nd
η_d	Efficiency at selected advance coefficient J_d – nd
η_F	Cushion fan efficiency – nd
η_f	Fan efficiency, per stage – nd
$\eta_{f_{STATIC}}$	Fan static efficiency, per stage – nd
$\eta_{f_{TOT}}$	Fan total efficiency, per stage – nd
η_H	Hull efficiency – nd
η_j	Efficiency at advance coefficient J – nd
η_{LS}	Cushion or lift system efficiency – nd
η_M	Mounting efficiency – nd
η_p	propeller or pump efficiency – nd
	Propulsor efficiency – nd
η_{p_o}	Open water propeller efficiency – nd
η_R	Relative rotative efficiency – nd
η_T	Transmission efficiency – nd
η_t	Cycle efficiency – nd
η_{TC}	Lifting system transmission efficiency – nd
η_z	Impact load magnitude, in g's – nd
θ (theta)	Angular rotation – rad
	One-half angle of appendage section – rad
	Trim or pitch angle – rad
	Deflection angle – rad
θ_w	Wave slope – rad
$\dot{\theta}$	Angular velocity – rad/sec
$\ddot{\theta}$	Angular acceleration – rad/sec^2
λ (lambda)	Wave length – ft
ν (nu)	Kinematic viscosity – centistokes
ν_w	Kinematic viscosity of seawater (1.28×10^{-5}) – ft^2/sec
ρ (rho)	Density – lb/ft^3
ρ_a	Mass density of air (0.0024) – $lb\text{-}sec^2/ft^4$
ρ_p	Reactor power density – megawatts/ft^3
ρ_r	Reactor weight density – lbs/ft^3
ρ_w	Mass density of water (1.99) – $lb\text{-}sec^2/ft^4$
σ (sigma)	Stress, general – lbs/in^2
	Cavitation index, or cavity cavitation number – nd
	Thoma cavitation number – nd
$\sigma_{allowable}$	Allowable stress – lbs/in^2
σ_b	Base cavitation number – nd
ω (omega)	Frequency, general or frequency of waves – rad per unit time
ω_e	Frequency of encounter – rad per unit time

Appendix B
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