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Bethesda, Md. 20084

DESIGN PRACTICES FOR POWERING PREDICTIONS

by

J. W. GRANT and C. J. WILSON

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DESIGN PRACTICES FOR POWERING PREDICTIONS

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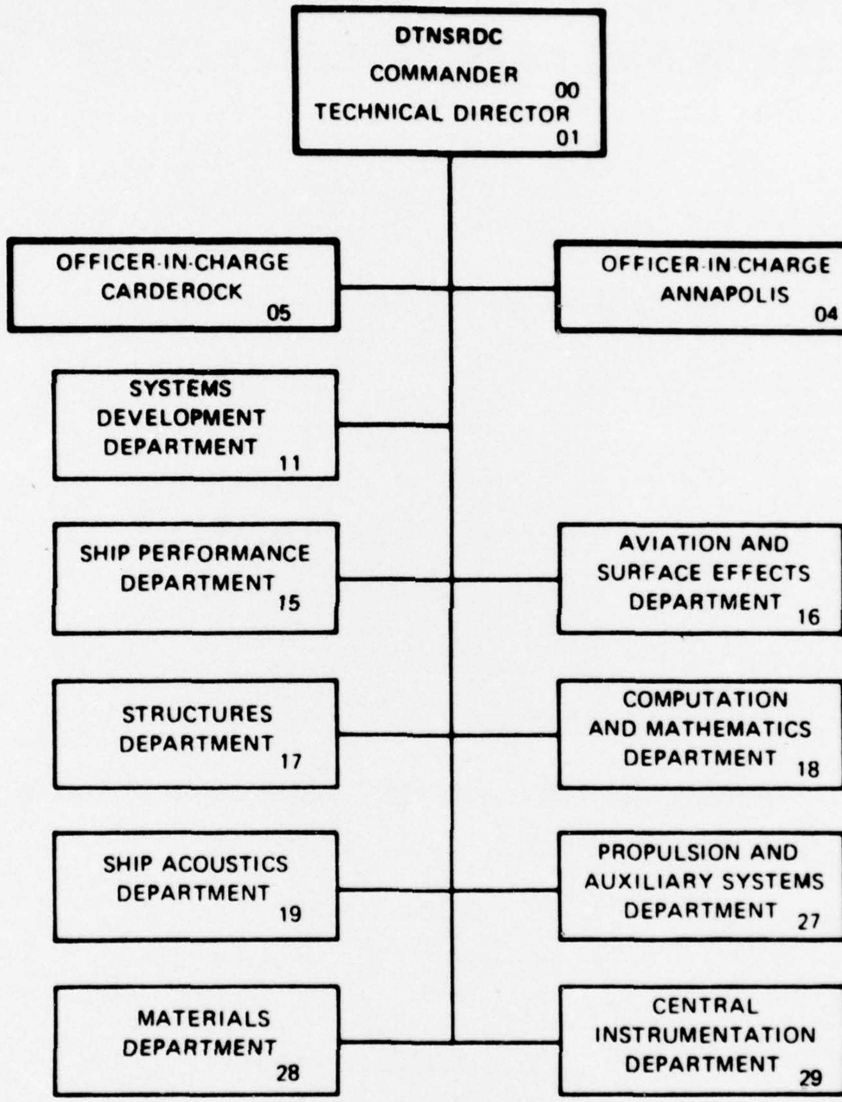

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This work reviews the current procedures, margins, allowances, etc., in light of the most recent literature and documents the design practice procedures for powering predictions in a form suitable for preparing NAVSEC technical practice. This work presents a general discussion of the practice in use for the estimation of powering performance during the various phases of design, the fundamentals of extrapolating model resistance data up to an estimate of the full-scale ship resistance, the techniques for estimating residuary resistance, → next page

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appendage resistance, wind drag, power margins, and shaft power, the techniques for estimating the effects of design alterations and the various aspects of rough water performance. Some supplementary information on the correlation allowance and the interaction coefficients has been included as Appendices.

FOREWORD

The David Taylor Naval Ship Research and Development Center (DTNSRDC) was authorized to assist the Naval Ship Engineering Center (NAVSEC) establish design practice for the prediction of speed and power. This work was funded under NAVSEC Project Order Numbers PO 4-0190 and PO 4-0319, dated 15 May 1974 and 14 June 1974, respectively. The principal objectives of this work were to review the current procedures, margins, allowances, etc., in light of the most recent literature and to document the procedures (updated, where applicable) in a form suitable for preparing NAVSEC technical practice. This report provides the documentation of the work done by NSRDC.

The report has been organized in a manner that will, hopefully, help to make it easy to use. Essentially, the report consists of nine chapters and two appendices. Chapter 1 presents a general discussion of the practice in use for the estimation of powering performance during the various phases of design. Chapter 2 presents the fundamentals of extrapolating model resistance data up to an estimate of the full-scale ship resistance. The techniques for estimating residuary resistance, appendage resistance, wind drag, power margins, and shaft power are discussed in Chapter 3, Chapter 4, Chapter 5, Chapter 6, and Chapter 7, respectively. The techniques for estimating the effects of design alterations are presented in Chapter 8. The various aspects of rough water performance are discussed in Chapter 9.

The nature of this project necessitated a significant amount of background work. The results of these investigations have, for the most part, been incorporated into the nine chapters. Some supplementary information on the correlation allowance and the interaction coefficients has been included as Appendices A and B, respectively, since it could not be readily incorporated into the main text.

The services of specialists were required for this project due to the scope of the background work. Contributions to this report, particularly

those by Mr. Hugh Y. H. Yeh and Dr. Allen G. Hansen, which are included in Chapter 9 and Appendix B, respectively, are greatfully acknowledged.

The authors wish to express their gratitude to Mr. Philip M. Covich of NAVSEC for his skillful coordination of this project. The technical and editorial comments, which he gathered from the staff at NAVSEC, were used to develop a better final draft of this report. The feedback he provided undoubtedly enhanced the utility of this document, since it reflected the opinions of prospective users.

The authors also wish to thank Dr. Arthur M. Reed, Mr. Kenneth D. Remmers, and Mr. William G. Day for their technical comments and Mrs. Elizabeth B. Byrne for her typing of the manuscript and its many revisions. Lastly, the authors wish to thank Dr. Wen-Chin Lin, Mr. Raymond Wermter, and Mr. Vincent J. Monacelia for their constructive review of this report, which has led to a significantly improved final draft.

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NOTATION

An attempt has been made to keep the notation used in this document consistent with the recommendations of the ABC Navies Standardization Program (1969 - Reference 2) and the I.T.T.C. Presentation Committee (1972 - Reference 20). Differences between the notation used herein and these standards are denoted by an asterisk (*). In those cases where the notation is marked by an "*", the notation will not be found in either of the standard lists. When the asterisk appears in the description it denotes supplementary information applicable to this document.

The following notation is subdivided into three categories:

(1) Geometry

(2) Resistance and Propulsion

(3) Rough Water Performance

GEOMETRY

A_{BT}		Area of transverse cross-section of a bulb
A_T		*Area of transom*
A_V		Area exposed to wind (*taken as projected area in a plane normal to the ship centerline)
A_X		Area, maximum transverse section
B		Beam or breadth, moulded of ship
B_X	*	Beam at the waterline at maximum transverse section
C_B		Block coefficient
C_P		Longitudinal prismatic coefficient
C_X		Maximum transverse section coefficient
C_∇		Volumetric coefficient
D		Diameter of a propeller
f_{BT}		Taylor sectional area coefficient for bulbous bow
f_T		Sectional area coefficient for transom stern
$\overline{FB}$		Longitudinal center of buoyancy from forward perpendicular

NOTATION - GEOMETRY

i_B	* Buttock slope
i_E	Angle of entrance, half
i_R	Angle of run, half
L	Length of a ship (*usually refers to L_{WL} or L_{PP})
L_{PP}	Length between perpendiculars
L_{WL}	Length of waterline in general
S	Wetted surface (* S_m - for model; S_s - for ship)
Ⓢ	R. E. Froude's wetted surface coefficient
T	Draft, moulded of the ship
T_W	* Transom width at L_{WL}
T_T	* Depth of transom on the centerline
T_X	Draft at maximum transverse section
Δ	Displacement weight
∇	Displacement volume

NOTATION - RESISTANCE AND PROPULSION

Ⓒ	R. E. Froude's resistance coefficient
C_A	Incremental resistance coefficient for model-ship correlation
C_{AA}	Air or wind resistance coefficient
C_F	Specific frictional resistance or drag coefficient (* C_{F_m} - model; C_{F_s} - ship)
C_R	Specific residuary resistance coefficient
C_T	Specific total resistance coefficient (* C_{T_m} - model; C_{T_s} - ship)
C_γ	* Resistance coefficient for relative wind direction
J	Advance coefficient or advance number of propeller, V_A/nD (* J_o - open water)

NOTATION - RESISTANCE AND PROPULSION

J_Q	* Advance coefficient, determined from K_Q identity (J_{Q_m} - model; J_{Q_s} - ship)
J_T	* Advance coefficient, determined from K_T identity (J_{T_m} - model; J_{T_s} - ship)
J_V	Apparent or ship speed advance coefficient, V/nD
K	R. E. Froude's speed-displacement coefficient
K_Q	Torque coefficient ($*K_{Q_m}$ - model; K_{Q_s} - ship; K_{Q_o} - open water)
K_T	Thrust coefficient ($*K_{T_m}$ - model; K_{T_s} - ship; K_{T_o} - open water)
n	Rate of revolution (*RPM - revolutions per minute)
P_D	Delivered power at propeller
P_E	Effective power
$P_{E_{wind}}$	* Augmentation in effective power due to R_{AA}
P_S	Shaft power
Q	Torque
R	Resistance in general
R_{AA}	Air or wind resistance (* R_{AA} - at a relative wind direction of γ)
R_{AP}	Appendage resistance
R_F	Frictional resistance
R_n	Reynolds number
R_R	Residuary resistance
R_T	Total resistance ($*R_{T_m}$ - model; R_{T_s} - ship)
t	Thrust deduction fraction (*(1-t) = thrust-deduction factor)
T	Thrust
V	Speed of ship ($*V_s$)
$V/\sqrt{L}$	* Speed-length ratio; also $V/\sqrt{L_{WL}}$; (knots $ft^{-1/2}$)

NOTATION - RESISTANCE AND PROPULSION

V_A	Speed of advance of propeller (* When used in the calculation of efficiencies, it is the speed determined from thrust identity)
V_R	Wind velocity, relative
w	Taylor wake fraction in general; $(V - V_A/V)$ (*$(1-w)$ = Taylor wake factor)
w_Q	Taylor wake fraction determined from torque (*coefficient) identity
w_T	Taylor wake fraction determined from thrust (*coefficient) identity
γ_R	Wind direction, relative
ΔC_F	Roughness allowance (obsolete, see C_A)
η	Efficiency in general
η_B	Propeller efficiency behind ship
η_D	Propulsive efficiency
η_H	Hull efficiency
η_O	Propeller efficiency in open water
η_R	Relative rotative efficiency
η_S	Shafting efficiency (* shaft transmission efficiency; assumed to be unity)
ν	Coefficient of kinematic viscosity (*ν_m - model; ν_s - ship)
ρ	Mass identity (*ρ_m - basin water; ρ_s - seawater)

NOTATION - ROUGH WATER PERFORMANCE

g	Acceleration due to gravity
k_{yy}	Real radius of gyration about y - axis (transverse)
P_{AW}	Mean power increase in waves
R_{AW}	Mean resistance increase in waves

NOTATION - ROUGH WATER PERFORMANCE

$R(\omega_E)$	* Response amplitude operator (R_{AW}/ζ^2)
$S_\zeta(\omega_E)$	One dimensional spectral density
T	Wave period
β	* Bulbous bow factor ($\beta = 1$ for a bulb; $\beta = 0$ for no bulb)
ζ_A	Wave amplitude
μ_E	* Nondimensional frequency of encounter
σ_{AW}	* Coefficient of added resistance
ω	Circular wave frequency
ω_E	Frequency of encounter (circular)

Chapter 1.0 POWERING PERFORMANCE, PRACTICES FOR ESTIMATION OF

The prediction of ship performance has traditionally and deliberately been divided into two distinctly separate components, which are the calm-water portion and the rough-water portion. Although the subject of rough-water performance is addressed briefly in Chapter 9.0 of this document, the principal subject matter covered herein deals with calm-water powering performance. The discussion in this chapter centers on the ways in which predictions of powering performance are developed during each of the three phases of ship design. Since the basic methodology used in the estimation of powering performance is the same through all three phases of design, it will be presented in the discussion of the first phase (Conceptual Design). Variations in the estimating procedures which are consistent with the design information available during the later phases of design (Preliminary Design and Contract Design) will be presented in the discussions of these phases.

1.1 Conceptual Design Phase

The conceptual design phase is initiated in response to an operational requirements issued by the Chief of Naval Operations. This design phase is composed of two parts, feasibility studies and conceptual design. The feasibility studies are performed to establish the major characteristics and cost of the ship. During this portion of the design phase, ship concepts are generally defined and evaluated using gross parameters and shipboard experience. The conceptual design portion is used to resolve the technical risks associated with the concept and define the ship in terms of overall geometry, weight, type of propulsion plant, speed and endurance.

Feasibility studies generally utilize the ship synthesis models to define the major ship characteristics and cost. During conceptual design the conventional and more elaborate methods are used to estimate hydrodynamic performance.

The accuracy of the processes used to estimate the powering performance of each alternative hull form during conceptual design is somewhat limited by the lack of ship definition. Inasmuch as the requirements of this phase frequently dictate that several alternative designs be evaluated, it is not practical to develop the fine geometric details of each one. Typically, the known features of each design would consist of at least the principal dimensions (length, draft, and displacement), the type of hull form (e.g., a destroyer, a tanker), an estimate of the maximum sectional area and the wetted surface, and the general propulsion arrangement (single-screw, twin-screw, quadruple-screw, and whether-or-not the shafting will be exposed). Naturally, the selected techniques used in the estimation of powering performance should be the best available that are consistent with the availability of hull-form geometry.

The first step in estimating the powering performance of a ship during this phase of design is to determine the bare-hull effective power for the underwater hull. Following this step the appendage resistance and the wind drag are estimated. Finally, using an estimate of the propulsive efficiency, the required shaft power can be estimated.

1.1.1 Bare-Hull Effective Power

The bare-hull effective power of a ship can be expressed in terms of a total resistance coefficient, which can be further subdivided into three component coefficients.* One of these components, the equivalent flat-plate frictional resistance coefficient (C_F), can be readily determined from an existing formulation. Standard practice dictates the use of the 1957 I.T.T.C. Model-Ship Correlation Line.[1] Utilization of this friction line requires only that the full-scale Reynolds' number be known. Another of these components, the correlation allowance (C_A), is assigned a value of 0.0005 for most naval ships. A complete description of the physical meaning of the correlation allowance and the procedures by which its value has been determined is presented in Section 2.2.3 and Appendix A of this manual. The last of these components, the residuary

*A complete description of the formulation of effective power is given in Section 2.1 of Chapter 2.

1. Hadler, Jacques B., "Coefficients for the I.T.T.C. 1957 Model-Ship Correlation Line," TMB Report No. 1185, 1958.

resistance coefficient (C_R), can be estimated very effectively if a high level of hull-form definition is available.*

Although the techniques used to estimate residuary resistance are discussed at some length in this manual (see Chapter 3.0), some brief comments regarding a few of these procedures are given here. The estimating procedure requiring the lowest level of hull-form definition makes use of series data and worm curve techniques, as described in Sections 3.2 and 3.3, respectively, of Chapter 3.0. The ability to choose an appropriate worm curve, however, depends heavily upon a greater knowledge of the eventual shape of the hull form than would be indicated by the limited parametric hull-form data available during conceptual design. If, indeed, this knowledge does exist and estimates of many of the hull-form parameters can be made, it is possible that one of the analytical techniques described in Section 3.4 of Chapter 3 can be used to estimate the residuary resistance coefficient. Regardless of the technique(s) chosen to estimate the residuary resistance coefficient (C_R), it must be added to the other two component coefficients (C_F and C_A) and expanded to an estimated bare-hull effective power using the formulation described in Section 2.1 of Chapter 2.0.

1.1.2 Fully-Appended Effective Power

A significant increase in the resistance of a hull form generally occurs when appendages are added. The typical complement of appendages for a modern twin-screw military ship, which would normally include shafting and struts, rudders, and stabilizer fins, would lead to a resistance increase in the neighborhood of 25-30% of the bare-hull resistance. Since the resistance of the appendages can be such a significant part of the total resistance, it is quite important that an accurate estimate be made at this phase of design.

Normally, estimates of appendage resistance (R_{AP}) are developed using data for similar ships. While collecting the data on appendage

*The notation used in this document is, wherever possible, consistent with Reference 2.

2.American/British/Canadian Navies Standardization Program, "Symbols for Naval Architecture for Use in Technical Writing," ABC-NAVY-STD-30B, December 1969.

resistance, other data can also be collected, which will be helpful in developing other estimates. Data on similar ships are necessary, for example, in the development of estimates of hull-propulsor interaction coefficients and worm curves. The data from which the estimates will be developed must be selected with great care, since not all ships are configured with similar appendage arrangements. It is better to base the estimates on a few, rather than many sets of data, if the smaller sample of data represents ships which have a complement of appendages nearly identical to that of the conceptual design.

The normal output from this procedure is a ratio of the fully-appended effective power to the bare-hull effective power. This ratio is generally plotted versus speed-length ratio, for each of the similar ships selected. The estimated fully-appended effective power for the conceptual design is then obtained at each speed by multiplication of the bare-hull effective power by the appropriate ratio. A discussion in greater detail of this procedure and others, along with the limitations thereof, is given in Chapter 4.0.

1.1.3 Still-Air Effective Power

The still-air effective power accounts for the wind drag (R_{AA}) on the above-water portion of the ship, including the superstructure. The air drag is determined for a relative wind velocity equal to but opposite in direction from the ship velocity, or, in other words, for a true-wind velocity of zero (still air). Although a complete description and discussion of the procedure is presented in Chapter 5.0, a brief outline of the technique is presented here.

The first requirement is that an estimate of the above water transverse area (A_V) be made. Frequently, a sufficient level of accuracy can be obtained by assuming that this area can be estimated using the following formula.

$$A_V = 0.75 * B_X^2,$$

where B_X = the beam of the ship at the waterline at the station of maximum area.

Although this approximation is sufficiently accurate for most naval combatant and auxiliary ships, it may not be suitable for certain other ships, such as aircraft carriers, heavily laden tankers, and some smaller ships and boats.

The still-air effective power ($P_{E_{wind}}$) may then be determined using the following formulation.

$$P_{E_{wind}} = \frac{C_{AA} A_V V_s^3}{\text{Constant}} ,$$

where, $C_{AA} \equiv$ ahead wind drag coefficient for the ship (dimensionless),

$A_V \equiv$ above-the-water transverse area of the ship (ft^2 or m^2),

$V_s \equiv$ ship speed (knots or m/sec),

and the Constant = 96,500 for the English system of units (A_V in ft^2, V_s in knots, and a mass density for air of 0.00237 lb sec^2/ft^4),

or the Constant = 1638 for the SI system of units (A_V in m^2, V_s in m/sec, and a mass density for air of 1.221 N sec^2/m^4).

It has been determined, per the discussion in Chapter 5.0, that the ahead wind drag coefficient (C_{AA}) is 0.70 for combatant ships, 0.45 for aircraft carriers, and 0.75 for naval auxiliaries. The total effective power of the ship in zero true wind can be determined by merely adding this $P_{E_{wind}}$ to the fully-appended effective power determined by the method described in Section 1.1.2.

1.1.4 Margins of Uncertainty

There are many uncertainties which can affect the estimate of the powering performance of a ship. Since model experiments are seldom conducted during the conceptual design phase, the estimates of powering performance are almost always based upon the analytical procedures described in this chapter. Although satisfactory estimates of performance can be made using these procedures, the lack of specific definition of the hull form, the appendages, the propeller(s), and the superstructure severely limits this accuracy. Furthermore, the estimates of the weights of the hull, machinery, and payload, etc., are, by the nature of the prerequisite assumptions, similarly uncertain and subject to change during the evolution from the conceptual design phase through the preliminary and contract design phases. Occasionally, for example, it is necessary to increase the beam of the ship to accommodate a component of the propulsion system or some other system, or to achieve acceptable transverse stability. Changes of this type obviously can affect both the shape and weight of the hull and, consequently, the total effective power.

Naturally, the lack of confidence in the estimate due to the inherent uncertainties during conceptual design has led to the use of a power margin to cover these uncertainties. The subject of power margins used during each phase of design is discussed at length in Chapter 6.0. The practice during this phase of design is to use a power margin of 11%. This percentage covers not only the uncertainties discussed in this section, but also those associated with the estimation of propulsive efficiency, which is discussed in the next section. The margin is applied directly to the total effective power, which accounts for the resistance of the bare-hull and appendages and the still-air drag, thereby increasing it by 11%, at this phase of design.

1.1.5 Shaft Power

The total estimated shaft power (P_S) for a ship can be obtained by dividing the adjusted (by an 11% margin) total effective power (P_E) by the propulsive efficiency. The propulsive efficiency (η_D) defines the performance of the propulsor when operating in the environment of its particular application. This efficiency (η_D) is generally expressed as the product of four component efficiencies. The first of these, the open-water efficiency (η_0), defines the performance of the propulsor in an undisturbed fluid (uniform inflow). The second component, the hull efficiency (η_H), defines the effects of the propulsor on the resistance of the hull, when the propulsor is operating in the wake of the ship. The third component, the relative rotative efficiency (η_R), defines the difference in performance attributable to the nonuniformity of the inflow velocity. The last component, the shaft transmission efficiency (η_S), which expresses the loss in power through the line-shaft bearings and glands, is defined as the ratio of the actual torque delivered to the propeller to the measured torque during a trial. These brief definitions of the components of the propulsive efficiency are intended to provide the reader with a general understanding, rather than a complete description, of the meaning of each component. A more detailed discussion of these efficiencies is given in Chapter 7.0, with supplementary information in Appendix B.

Although the magnitude of the propulsive efficiency (η_D) is generally determined by estimating the value of each of its component efficiencies, occasionally, when time and/or data availability do not permit an in-depth analysis of the propulsion problem, an estimate of η_D may be made directly. Usually, however, each component is estimated individually. The shaft transmission efficiency (η_S) is almost always assigned a value of unity (1.0), since the torque difference is generally very small (<.5%) due to proximity of the torsion meter to the propeller. This matter, which is primarily a result of trial and towing tank practice, is discussed in Section A.3.6 of Appendix A. The hull efficiency (η_H) is defined as the quotient of the thrust-deduction factor (1-t) over the thrust-wake factor ($1-w_T$). In addition to being necessary for the calculation of η_H, these factors are essential to the selection of suitable propeller geometry. The relative rotative efficiency (η_R) and these two factors (1-t and $1-w_T$) are generally estimated from data accumulated during the search for relevant worm curve and appendage resistance data. The estimate of the open-water propeller efficiency (η_0) is generally developed independently, using the total effective power, the thrust-wake factor, the thrust-deduction factor, the design speed, and other relevant power-plant data as parameters.* Having once obtained estimates of all four component efficiencies (η_S, η_H, η_R, and η_0), the propulsive efficiency (η_D) is determined as the product of these four. The total estimated shaft power (P_S) may then be calculated as the quotient of the total effective power (P_E) over the propulsive efficiency (η_D)**.

1.2 Preliminary Design Phase

The major output from the Conceptual Design Phase is a tentative conceptual baseline (TCBL) study for a ship satisfying the Top Level Requirements (TLR), which were developed during that phase. The objectives of the Preliminary Design Phase are to detail and optimize the TCBL within the constraints imposed by the TLR. The product of this phase is a preliminary design of the ship, complete with its principal dimensions, form coefficients, longitudinal distributions of immersed volume and waterplane area, body plan, appendage configuration, and motion stabilization system. Throughout the process of optimization, it is necessary that estimates of the powering performance of each alternative be determined. Furthermore, at the conclusion of this phase, an estimate must be made for the selected preliminary design.

Essentially, the methods employed should follow the approach outlined in Sections 1.1.1 through 1.1.5. Since greater detail should exist for most of the steps during this phase of design, it is feasible to use the more sophisticated estimating techniques. For example, it should be feasible to use the methods discussed in Section 3.4 of Chapter 3.0 and Section 4.2 of Chapter 4.0, to estimate the fully-appended effective power of the ship. Furthermore, the accuracy of the estimate of wind drag may

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*The practices employed by the designer of marine propellers are discussed in Reference 3.

3."Design Procedure for Surface Ship Propellers," Naval Ship Engineering Center Technical Report No. 6144-76-136, January 1976.

**Since the Shaft transmission efficiency (η_s) is assumed to be unity, the shaft power (P_S) and the delivered power at the propeller (P_D) are equal. Consequently the propulsive efficiency (η_D) may be defined as either P_E/P_S or P_E/P_D.

be increased significantly due to the availability of superstructure detailing, since the above-the-water transverse area may now be calculated, rather than estimated. Regardless of the estimate being made, the sophistication of the technique employed should be at the highest level that is consistent with the availability of data. Whenever possible, an estimate should be made using an alternate technique to supplement the principal estimate.

The estimate(s) of powering performance developed during preliminary design also require that a margin of uncertainty be applied. Although the design is certainly still subject to change, a greater number of the design parameters are now fixed, at least temporarily. As a consequence, the margin of uncertainty is reduced from 11 percent to 9 percent. Frequently, resistance and propulsion experiments are conducted with a model of the preliminary design. A significant portion of the uncertainty associated with the accuracy of powering estimates is eliminated in these cases. The primary uncertainty which remains at this point in the design cycle is due to the fact that design changes can and will, almost certainly, occur. The model, the propeller and/or the experiments conducted therewith will not precisely represent the eventual ship operating at its design condition. Since the propeller used in the propulsion experiments is a stock propeller and, as such, does not represent the final design propeller, and estimate of the differences in performance between the design and stock propellers must be applied as a correction to the propulsion data (after the wind drag estimate has been applied). When this type of propulsion data is available during preliminary design, practice dictates the use of a margin of 6%.

1.3 Contract Design Phase

The last phase of the design process in NAVSEC is the contract design phase. The output of this phase is comprised of detailed drawings and specifications, which define the ship in sufficient detail to permit an intelligent bid by prospective builders on the time and cost to construct the ship(s). Detailed analyses of hydrodynamic performance are performed during this phase, in most cases.

A complete set of model experiments is generally conducted during the contract design phase, using a stock propeller and, eventually, a model of the contract design propeller. A suitable set of experiments can be planned using Chapter 2.0 of Reference 4 as a guide. Although experiments are being used to predict the powering performance, the estimate of the effective power augmentation due to still-air drag must be included in the calculation of total effective power.

Although the margins of uncertainty during this phase of design are somewhat reduced by finality of a contract design and the availability of applicable resistance and propulsion data, they must be used to cover those uncertainties which remain. A margin of 3 percent is applied at the end of contract design, if propulsion data with the contract design propeller exists. This margin covers minor design changes (appendages, propellers, and displacement growth, etc.) and prediction accuracy, in that order of significance. Prior to the availability of propulsion data with the design propeller(s), when experimental data using the stock propeller(s) exists, these data are to be corrected by an estimate of the performance of the design propeller(s), and used with a margin of 6 percent.

1.4 Margins in Service

The techniques used to estimate the powering performance of a ship, as discussed in this chapter, apply to a new ship operating in a rather ideal environment. It is the exception, rather than the rule, that the ship will operate under such ideal circumstances. The circumstances which tend to affect the powering performance of a ship are frequently sub-divided into Environmental Factors and Deteriorative Factors. A few typical examples of environmental factors, which can significantly affect the powering performance of a ship, are the sea state, the prevailing wind, the current, and the sea-water temperature. The deteriorative factors, which are generally detrimental to the powering performance, are generally sub-divided into two groups, which are either short-term or long-

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4. Covich, P., "Guide for the Preparation of Hydrodynamic and Aerodynamic Model Test Programs in Support of USN Surface Displacement Ship Designs," Naval Ship Engineering Center Technical Report No. 6136-74-17, 15 May 1975.

term in nature. The most well-known short-term deteriorative factor is the fouling of the ship's hull, appendages, and propeller(s). This phenomenon, which is discussed in Section A.3.4 of Appendix A, is controlled and partially remedied by periodic refurbishing of the ship in a drydock. Since routine drydocking procedures do not restore the ship to its original "like-new" condition, the residual short-term deterioration remaining after each drydocking becomes the major component of the long-term deteriorative factor. Some typical examples of the long-term deteriorative factor are the erosion of the hull, appendages, and propeller(s), and a reduction in the maximum deliverable power at the propeller due to machinery system degradation. Although each of these deteriorative factors (fouling, corrosion, etc.) and environmental factors (sea state, current, etc.) can significantly affect the powering performance of a ship, it is impossible to rank their significances at any particular time unless sufficient knowledge of the ship's deteriorative history and operational environment exists.

The consequences of these factors on the powering performance of a ship is that the maximum achievable speed during the ship's life will be reduced from that of a new ship operating in a quasi-ideal environment. The outgrowth of this is a need to specify the speed at some arbitrary period during its lifetime. The response to this need is a specification of a Sustained Sea Speed for a ship. This speed is defined as the speed attainable with the new ship, when operating in an environment corresponding to the condition for which the estimate was made using 80% of the maximum continuous power output. Since it has been determined that this percentage is consistent with the Navys' experience, it is the practice to use 80%, until the effects of all these factors can be addressed analytically and specified statistically.

Chapter 2.0 - BARE-HULL EFFECTIVE POWER, ESTIMATION OF

The effective power (P_E) of a ship is that amount of power which would be required to tow a ship through the water at any particular speed. Although the formulation of the bare-hull effective power is identical to that for the fully-appended ship, this chapter essentially addresses the bare-hull effective power. The bare hull is nominally defined to include such components as skegs, which are a critical part of the ships' supporting structure in a drydock (e.g., a conventional centerline skeg), and bulbous bows and bow sonar domes, which are faired into the ship's lines. It does not, however, include rudders, fins, shafting, struts and bossings.

The bare-hull effective power of a ship can be accurately estimated if a complete geometrical description of the hull form is available. Short of having a complete description, the design displacement (Δ), the waterline length (L_{WL}), the beam (B_X) at the station of maximum area, the draft (T), the wetted surface (S), and the area (A_X) of the maximum station must be known or approximated if an estimate of effective power is to be made. The general underwater shape of the hull form should also be known, e.g. whether it's a cargo ship, a destroyer, a tanker, a carrier, or some other type of ship. Naturally, the accuracy of the powering estimate can be increased if more information regarding the hull geometry is available. The best source for such information would be a complete set of plans for the molded hull form. When these plans are available, it is possible to use experimental and/or sophisticated analytical techniques to estimate the effective power of the bare hull. Furthermore, the existence of these plans facilitates comparisons with simiiar hull-form designs.

2.1 Effective Power, Formulation of

As a consequence of the principles discussed in Sections 1 through 6 of Chapter VII in Reference 5, the following formulation is used to determine the effective power (P_E) of a ship.

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5. Comstock, J.P., editor, "Principles of Naval Architecture," The Society of Naval Architects and Marine Engineers, 1967.

$$P_E = \frac{\rho_s/2 \; S_s \; V_s^{3} \; C_{T_s}}{\text{Constant}}, \qquad (2.1)$$

where ρ_s = density of sea water $(\frac{\text{lb sec}^2}{\text{ft}^4}$ or $\frac{\text{N sec}^2}{\text{m}^4})$,

S_s = wetted surface of ship (ft^2 or m^2),

V_s = ship speed (knots or m/sec),

C_{T_s} = specific total resistance coefficient for the ship (dimensionless),

and the Constant = 114.39 lb-sec^2-knots^3/ft^2-horsepower, for the English system of units,

= 1000.0 N-m/sec-kilowatt, for the SI system of units.

Since estimates of powering performance are generally developed for the standard condition,* equation 2.1 may be re-written, incorporating the standard density of sea water, as follows.

$$P_E = \frac{S_s \; V_s^{3} \; C_{T_s}}{114.94} \text{ (horsepower), for the English system of units,}$$

or

$$P_E = \frac{S_s \; V_s^{3} \; C_{T_s}}{1004.8} \text{ (kilowatts), for the SI system of units.}$$

2.2 Specific Total Resistance Coefficient, Components of

The specific total resistance coefficient for the ship is composed of three component coefficients, as follows.

$$C_{T_s} = C_R + C_{F_s} + C_A, \qquad (2.2)$$

where C_R = specific residuary resistance coefficient,

C_{F_s} = specific frictional resistance coefficient for the ship,

and C_A = incremental resistance coefficient for model-ship correlation (correlation allowance).

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*The standard condition for powering predictions is calm deep sea water at 45° North Latitude having a salinity of 3.5% and a temperature of 59° Fahrenheit (15° Celsius), which has a density of 1.9905 lb sec^2/ft^4 - or - 1025.9 N sec^2/m^4.

2.2.1 Specific Residuary Resistance Coefficient

The specific residuary resistance coefficient (C_R) accounts for the effects of wavemaking resistance and other forms of resistance, such as that due to flow separation, which are not related to the ship's equivalent flat-plate frictional resistance. The basic assumption used in the prediction of the effective power of a ship is that the model and ship have the same specific residuary resistance coefficient at "corresponding" speeds, e.g., same Froude number or speed-length ratio. Since this resistance coefficient is entirely dependent on the shape of the underwater hull form, it can be most accurately determined by resistance experiments with a geosim of the ship. During the earlier phases of design, however, it is frequently impossible to construct a model and conduct experiments, since either the specific hull-form geometry has not been developed or the available time and/or funding do not justify an experimental program. Consequently, other methods to estimate the residuary resistance have been developed using various sources of data on the resistance of ships. Although the credibility of an estimate derived by these methods is somewhat less than that of a prediction derived from a resistance experiment, a fairly accurate estimate can be made if the hull form is well defined and a good sample of data for similar hull forms is available. These estimating techniques are fully described in Chapter 3.0.

2.2.2 Specific Frictional Resistance Coefficient

The specific frictional resistance coefficient (C_F) for both model-scale and full-scale hull forms is required if a prediction of full-scale effective power is to be made using model-scale resistance data. Ever since it was postulated many years ago that the frictional resistance of a hull form is dependent on its Reynolds' number (R_n), the efforts of numerous investigators have been directed toward the establishment of a realistic formulation for hull forms relating frictional resistance to Reynolds' number. Realizing the need to cover the wide range in Reynolds' number dictated by the numerous combinations of length and speed for both ships and models, most investigators have attempted to fit a continuous function of Reynolds' number, based on waterline length, through flat-plate friction data gathered by many experimenters. The result of one such investigation, which related the specific frictional resistance coefficient to the Reynolds' number, was eventually adopted by the American Towing Tank Conference (A.T.T.C.) in 1947.[6] As experience was gained with the A.T.T.C. formulation, it became obvious to experimenters, particularly those who worked with smaller models, that this friction line was not satisfactory for predictions of effective power from resistance measured on the smaller models, i.e., it was frequently necessary to use a negative correlation allowance (C_A) with the "small-model" data to obtain agreement with the prediction of effective power derived from "large-model" data using a correlation allowance of 0.4×10^{-3}. Consequently, in 1957, the International Towing Tank Conference adopted a new, but interim, formulation in an effort to resolve this dilemma. Although this dilemma has not been completely resolved, the U. S. Navy uses the appropriately named I.T.T.C. 1957 Model-Ship Correlation Line as formulated below:

$$C_F = \frac{0.075}{(\log_{10} R_n - 2)^2}, \qquad (2.3)$$

where $R_n = \frac{VL_{WL}}{\nu}$ (Reynolds' number), and

where, V = speed (ft/sec or m/sec),

L_{WL} = waterline length (ft or m), and

ν = kinematic viscosity (ft^2/sec or m^2/sec).

2.2.3 Incremental Resistance Coefficient for Model-Ship Correlation

The incremental resistance coefficient for model-ship correlation, which was formerly called the roughness allowance (ΔC_F) is generally

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6. Todd, F. H., "Tables of Coefficients for A.T.T.C. Model-Ship Correlations and Kinematic Viscosity and Density of Fresh and Salt Water," The Society of Naval Architects and Marine Engineers T&R Bulletin No. 1-25, May 1964.

called the correlation allowance (C_A). Essentially it is a correction factor, which accounts for the effects of many variables that are too small and/or too imprecisely known to be individually determined. Although a significant part of it is attributable to differences in structural and paint roughnesses between the model and the ship, other important components include three-dimensional form factors, flow through scoops and sea chests, and scale effects (e.g. appendages, differences in flow around the ship as compared to the model, difference between model and ship propeller performance, differences in properties of tank water and sea water not compensated for in other corrections, etc.).

Since it is not presently possible to quantify the components of the correlation allowance, its value is determined through correlation experiments. These experiments are designed to duplicate the actual conditions of the full-scale standardization trial. The model, for example, must be fitted with a geosim of the full-scale propeller and the experiments are conducted at conditions corresponding to the trial displacement and trim. The correlation allowance for any ship may then be determined through a comparison of the data from the correlation experiment with that from the standardization trial.

The correlation allowance for most naval combatant and auxiliary ships has been found to be 0.0005, for ships painted with a vinyl resin. The data for some of the larger ships, such as aircraft carriers, indicates that 0.0005 is too large. Since many variables affect the correlation allowance, there remains an uncertainty as to whether the greater length of carriers is responsible. It is the practice, nevertheless, to use 0.0005 for all ships unless otherwise specified. A complete discussion of the correlation allowance is presented as Appendix A.

Chapter 3.0 - RESIDUARY RESISTANCE, ESTIMATION OF

The residuary resistance of a ship can be estimated, per the brief discussion in Section 2.2.1 of Chapter 2.0, using any one of several techniques. Each of these techniques is discussed in this chapter.

3.1 Experimental Technique

Model experiments are generally recognized as being the best source for the residuary resistance of a hull form. There are many other quasi-analytic methods which, using three or more hull-form parameters, interpolate between the experimental data for similar hull forms to arrive at an estimate of the residuary resistance. None of these methods, however, are able to quantify all the effects of shape and anomalies which may evolve in the development of a design.

The scaling laws and other considerations which are the foundation of model experimentation, are discussed in detail in Sections 2.3 and 6.3 of Reference 5. Essentially, it is assumed that the specific residuary resistance coefficient for the ship and the model will be identical at the same Froude number. Therefore the resistance experiments are conducted at corresponding (same Froude number) speeds to determine the total model resistance. These data are reduced to a total resistance coefficient (C_{T_m}), as follows,

$$C_{T_m} = \frac{R_{T_m}}{\frac{1}{2}\rho_m S_m V_m^2}, \tag{3.1}$$

where, R_{T_m} = total model resistance (lb or N)

ρ_m = mass density of basin water (lb sec^2/ft^4 or N sec^2/m^4),

S_m = wetted surface model (ft^2 or m^2), and

V_m = model speed (ft/sec or m/sec).

and further reduced to the residuary resistance coefficient (C_R), as follows:

$$C_R = C_{T_m} - C_{F_m} , \tag{3.2}$$

where, C_{F_m} = the frictional resistance coefficient at the model Reynolds' number per Equation 2.3 in Chapter 2.

3.2 Series Data Technique

Extremely useful sets of resistance data have been collected from experiments with groups of geometrically related model hull-forms. When the group of hull-forms has evolved from a systematic and methodical development program encompassing variations in some of the major hull-form parameters, the group of models is referred to as a series. The resistance data from experiments may then be catalogued with respect to the hull parameters varied in the development of the series. If the form parameters of a new design fall within this catalogue, an estimate of the resistance of the new design may be determined through interpolation.

Numerous sets of series data have been collected from experiments in towing tanks throughout the world. A comprehensive index of the various sets of series data has been compiled by the Society of Naval Architects and Marine Engineers (SNAME).[7] This index on a single page for each series, gives the type of ship, the source of data and a reference, the range of principal dimensions and proportions (parameters), the speed range, the method of presentation, the friction formulation used in data reduction, the model size and other pertinent information.

One of the most well-known sets of series data is the Taylor Standard Series. A reanalysis of this series is used at NAVSEC.[8] These data are presented as contours of the residuary resistance coefficient (C_R) versus speed-length ratio ($V/\sqrt{L}$) for numerous values of the longitudinal prismatic coefficient (C_P), the volumetric coefficient (C_∇), and the beam-draft ratio (B/T) covering the range of hull variations in the series. The residuary resistance coefficient, as published, was determined using the appropriate frictional resistance coefficients from the A.T.T.C. frictional resistance formulation.[6]

7. "Index of Methodical Series Ship Model Resistance Tests," SNAME T&R Bulletin 1-31, July 1973.

8. Gertler, M., "A Reanalysis of the Original Test Data for the Taylor Standard Series," DTMB Report 806, March 1954.

Another well-known set of series data is Series 60. The data from this series[9] are presented as contours of constant residuary resistance per unit displacement (R_R/Δ) versus the block coefficient (C_B) and the length-beam ratio (L/B) for incremental values of the beam-draft ratio (B/T) and the speed-length ratio ($V/\sqrt{L}$). The data are also presented in a form which replaces R_R/Δ and $V/\sqrt{L}$ with the R. E. Froude resistance coefficient C and the R. E. Froude speed-displacement coefficient K, respectively. It should be mentioned here that this series is primarily applicable to merchant-type hull forms.

The available sets of series data can be extremely useful. The only problems encountered are those relating to the method of presentation. Since each investigator has used his own preferred coefficients for resistance and speed, the resulting variations in the presentation technique often make it rather cumbersome to use the data. Furthermore, caution should be exercised in the use of series data, since several different frictional resistance formulations have been used over the years. In many cases, an adjustment will have to be made to the residuary resistance to compensate for the difference between the I.T.T.C. line and whatever friction line had been in use at the time the series data were published.

3.3 Worm Curve Adjustments

When the hull-form parameters of a design fall within the parametric limits of a particular series, but the basic hull shape (e.g., stern shape) and/or the method used to develop the design differ(s) from the series, it is appropriate to use a worm curve to adjust the estimate derived from that series. Essentially a worm curve traces the functional relationship between a resistance correction factor and speed. The appropriate worm curve may be determined by the following technique:

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9. Todd, F. H., "Series 60-Methodical Experiments with Models of Single-Screw Merchant Ships," DTMB Report 1712, July 1963.

(1) Find a hull form and/or hull forms which,

   (a) have substantially the same shape as the design, and

   (b) were developed in a manner similar to that used in the design, and

   (c) whose hull-form parameters fall within the limits of the series.

(2) Determine the ratio of the actual resistance to the resistance estimated from the series data at several speed-length ratios for each of the hull-forms that are applicable.

The information determined in (2) becomes the so called worm curve. There are several schools of thought regarding the resistance quantity that should be used to determine the worm curve. The most widely accepted technique is to use either the ratio of the residuary resistance ($R_R/R_{R_{series}}$) or its coefficient ($C_R/C_{R_{series}}$). When the design itself is parametrically similar to the hull-form(s) used to determine the worm curve, one may use the ratio of the total effective powers at equal displacements ($P_E/P_{E_{series}}$) as the worm curve. This would be permissible since, for hulls that are shape-wise and parametrically similar, the ratio of residuary resistance to frictional resistance for each would be similar.

Occasionally, it is necessary to apply a correction to the worm curve to account for the effects of a parameter that is not addressed by either the series data or the worm curve. If, for example, the design will have a bulbous bow, and neither the series data nor the worm curve expresses the effect of a bulbous bow, a correction must be applied to the worm curve to account for the bulbous bow. Since this type of correction is frequently based on data for hull forms that are not parametrically similar to the designed hull form (e.g., they differ in displacement-length ratio, and/or prismatic coefficient, and/or L/B, and/or B/T, etc.), an effort must be made to determine if the correction should be transferred, either directly or with a modification, to the design in question.

3.4 Analytical Techniques

Numerous sets of data, in addition to the series data, have been collected at towing tanks throughout the world. Several organizations have attempted to determine functional relationships between the hull-form parameters, the speed, and the resistance. The most successful attempts have been made when the data base consisted of only one type hull-form. One example of such an attempt is the regression analysis of resistance data for destroyer-type hull forms done at DTNSRDC.

3.4.1 Regression Analysis for Destroyer Resistance

The data base used in the regression analysis consists of hull-form parameters and bare-hull resistance data for 233 destroyers. The mathematical expression at each speed derived from this analysis expresses the resistance as a polynomial function of the 14 hull-form parameters. The range and mean value of each of the hull-form parameters are given in Table 3.1. The credibility of this analysis has been investigated and it has been demonstrated that the bare-hull ship resistance of 95 percent of the 233 destroyers was estimated to within 5 percent of that predicted from the model experiments. As further proof of the usefulness of this technique, the bare-hull ship resistance has been estimated for some 15 hull forms which were not included in the original data base. These estimates agreed with the predictions from the model test data to the same accuracy as for hull forms within the original data base. One should, however, avoid using this method for hulls having form parameters that are not within the limits of the data base. The accuracy of this method would be degraded considerably when estimating the resistance of hull forms that are not of the destroyer type.

TABLE 3.1

DATA BASE FOR DESTROYER RESISTANCE REGRESSION ANALYSIS

(MEAN VALUE AND RANGE OF HULL-FORM PARAMETERS)

PARAMETER	MEAN VALUE	RANGE
S	7.63	6.56 - 8.67
L_{WL}/B_X	9.54	5.69 - 13.82
B_X/T_X	3.25	2.29 - 4.69
C_P	0.62	0.526 - 0.80
C_X	0.78	0.61 - 0.95
$\frac{\Delta}{(.01L_{WL})^3}$	48.04	34.06 - 89.62
$\overline{FB}/L_{WL}$	0.51	0.49 - 0.55
i_E	8.57°	2.0° - 25.9°
i_R	9.77°	3.0° - 35.7°
i_B	4.40°	0.0° - 9.6°
f_{BT}	0.008	0.0 - 0.062
f_T	0.08	0.0 - 0.525
T_W/B_X	0.43	0.0 - 0.84
T_T/T_X	0.17	0.0 - 0.38

3.4.2 Historic Data Analysis for Ship Resistance

Another analytical technique, which can be applied to a broad data base, is under development at DTNSRDC. This method, which is referred to as the DTNSRDC Historic Data Computer Program, can be used to estimate the bare-hull effective power for a target design, as follows:

(a) It utilizes a data base consisting of eight hull-form parameters and residuary resistance coefficients at several speed-length ratios for many ship designs.

(b) It selects a local neighborhood of 25 hull-forms based on the closeness of the hull-form parameters to those of the target design.

(c) It uses a multi-variable interpolation scheme, at each speed-length ratio, to determine the relationship between the residuary resistance coefficient and the parameters.

(d) It uses this relationship to compute the resistance of the target design and extrapolates it to an estimate of full-scale bare-hull effective power.

Although this technique is in the developmental stage, it has been successfully employed in the estimation of effective power for a few hull forms. The original data base utilized by the program consisted of resistance and hull-form data for naval auxiliary and commercial ships. The ranges covered by this data base for each of the eight hull-form parameters are presented in Table 3.2. It has been determined that the best estimates of effective power can be obtained for hull forms which are parametrically near the center of the data base, where the density of data is the greatest. Consequently, a useable range has been specified to alert the user of the lower level of confidence associated with the edges of the data base, where the data density is at its lowest level. The first documented usage of this technique was its application in the development of a revised hull form for the AO 177 Class.[11] More recently,

11. Robinson, J. H. and J. W. Grant, "Naval Auxiliary Oiler AO 177 Class-Hull Form Design Development," NSRDC Ship Performance Department Report SPD-544-04, June 1974.

after the data base had been expanded to include aircraft carriers, this technique was used in the evaluation of a TCBL in NAVSEC's Continuing CV Conceptual Design study.

TABLE 3.2

HISTORIC DATA PROGRAM - NAVAL AUXILIARY DATA BASE

(Actual and Useable Ranges of Hull-Form Parameters)

Parameter	Actual Range	Useable Range
S	5.690 - 7.939	6.15 - 7.35
L_{WL}/B_X	5.000 - 9.045	6.40 - 7.70
B_X/T_X	2.104 - 5.807	2.45 - 3.80
C_P	0.534 - 0.863	0.55 - 0.67
i_E	2.00 - 47.00	6.50 - 15.00
$\overline{FB}/L_{WL}$	0.468 - 0.572	0.49 - 0.52
f_{BT}	0.0 - 0.27	0.0 - 0.07
$\frac{\Delta}{(.01\ L_{WL})^3}$	53.45 -237.88	80.00 -160.00

3.5 Theoretical Technique

The residuary resistance of a surface ship is composed of numerous components, including wave-making resistance, form resistance, eddy-making resistance and other small components. Because of this complexity, and the complexity involved in forming a mathematical model of how the components interact, no theoretical methods have been developed which accurately estimate the residuary resistance for conventional surface displacement ships. It is possible to estimate the wave-making resistance for simple hull forms with some degree of accuracy. An application of such a technique, as applied to small waterplane area twin-hull forms, is documented in Reference 13.

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13. Lin, W. C. and W. G. Day, Jr., "The Still-Water Resistance and Propulsion Characteristics of Small-Waterplane-Area-Twin-Hull (SWATH) Ships," American Institute of Aeronautics and Astronautics, February 1974.

Chapter 4.0 - APPENDAGE RESISTANCE, ESTIMATION OF

4.1 Experimental Technique

The resistance of a suit of model appendages may be determined through experiments. The resistance of the appendages (R_{AP}) is merely the difference between the resistance of the fully-appended model and that of the bare-hull model. The method used to scale these data up to a prediction of the full-scale effective power (P_E) is as follows:

1. The total resistance coefficient is determined from the total resistance of the fully-appended hull form using a wetted surface which includes,
   - (a) the bare-hull wetted surface, and
   - (b) the surface area added by the rudder(s) and the stabilizer fins and/or the bilge keels, if any.
2. The residuary resistance coefficient is determined according to Equation 3.2 in Section 3.1 of Chapter 3.0.
3. The full-scale effective power is then determined according to Equation 2.1 in Section 2.1 of Chapter 2.0.

The quantity derived from these results that is most critically examined is the ratio of the fully-appended effective power to the bare-hull effective power. This ratio is frequently used in comparing one design with another.

4.2 Computational Technique

The resistance of each element of an appendage suit can be estimated using a computer program developed at DTNSRDC.[14] A description of this method is presented here:

"This method determines the resistance of each appendage (such as exposed shafting, shafting struts, shaft bossings, power transmission pods and struts, bilge keels, and control surfaces) as a function of the Reynolds' numbers appropriate to each appendage. Appendages are grouped in categories which are treated as two-dimensional surfaces, bodies of revolution, or flat panel friction planes. Resistance formulas were derived for each type of appendage, based on appendage physical characteristics and Reynolds' number. The derived formulas were used to develop a computer program, which was

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14. Lasky, M. P., "An Investigation of Appendage Drag," NSRDC Ship Performance Department Test and Evaluation Report 458-H-01, March 1972.

written in FORTRAN IV language, to perform the estimates."

"The estimates derived by this method were correlated with data from model tests for 14 twin-screw, transom-stern hull forms and, for this type of hull form, at least, the mathematical model was found to be an effective means of predicting appendage resistance. The differences between the results from the mathematical model and the model test data appear to be due to two basic resistance phenomena, i.e., induced drag due to misalignment of an appendage with the flow, and a Froude-number-dependent drag due to the interaction between the appendages and the hull."

"The major drawback of this method is that it can only be used when the appendage configuration has been well defined (detailed dimensions of the appendages are required as inputs to the program). At the early stages of a new ship design program, when the appendage configuration is usually undetermined, it would not be feasible to use this method."[4]

4.3 Analytical Technique

This technique is, essentially, the only method available during the early stages of design, when the dimensions of many components of the appendage suit are unknown. Basically it consists of collecting and analyzing sets of data for similar hull forms having similar appendage configurations. The latter should be emphasized since there may be subtle differences in appendage arrangements. A few of these, which can significantly affect the resistance are as follows:

(1) The resistance of the shafting and struts commensurate with a controllable-reversible pitch propeller system is significantly greater than that of a conventional fixed-pitch propeller system.

(2) The resistance of a ship fitted with bilge keels and/or stabilizer fins is measurably greater than that of a ship fitted with neither. In general, however, the resistance of these appendages is primarily frictional. As such, the resistance (R') added by a fin or a bilge keel may be approximated, as follows:

$$R' = \text{Constant}\ \frac{S'}{S}\ R_T, \qquad (4.1$$

where R_T = total bare-hull resistance (lb or N), S' = wetted surface added by the appendage (ft^2 or m^2),

S = wetted surface of the bare hull (ft^2 or m^2),

and in general, the constant may be taken as 1.2.

(3) The resistances of similar bow-mounted and keel-mounted sonar domes are not generally similar but are a significant component of the total appendage resistance.

It is usually necessary to review the data for several hull forms, selecting those having appendage arrangements that are quite similar to the design. After the relevant sets of data are collected, the increase in resistance or effective power due to appendages can be estimated by either averaging or interpolating between two-or-more of these sets of data. Occasionally, when a design happens to be a direct offspring of a particular parent hull form, both parametrically and in its complement of appendages, it is appropriate to use the appendage resistance data of that hull form alone. Despite the fact that this technique might appear to be crude, it can provide a reasonably accurate estimate of appendage resistance when used with care.

Chapter 5.0 - WIND DRAG, ESTIMATION OF

It seems appropriate to review several definitions, i.e., true wind is that wind which is due to natural causes and exists at a point above the sea whether or not the ship is there (zero true wind is still air); relative wind is the vectorial summation of the velocities and directions of the ship and the true wind (zero relative wind is a wind traveling at exactly the same speed and in the same direction as the ship); basic wind drag is that due to the passage of the ship through a zero true wind.

Over the years a number of experiments with superstructure models representing the above water portion of ships have been conducted in water first by Hughes[15,16] and then by others including work done at the Experimental Model Basin. More recent experiments have been conducted in the wind tunnels at the Center. The experiments in water are no longer considered to be as accurate as those conducted in the wind tunnels due to problems with wave-making particularly at angles other than zero. Data have been published for cargo hulls and tankers by a number of investigators at other establishments as listed in the references.

The general experience of investigators has been that Reynolds' number scale effects are not significant. Models built with large cylindrical appendages such as kingposts and stacks are undoubtedly affected to a degree, but small features such as wire rigging are omitted which tends to balance the situation.

Recently there has been an increased interest in the wind drag of ship hulls. The work by Shearer and Lynn[17] brought to attention the fact that part of the wind forces to which a ship is subjected is due to the natural wind in the ambient condition. This natural wind has a velocity gradient which varies with height above the sea and has been the source of considerable discussion by every investigator since Shearer and Lynn. It is questioned how significant the problem of the gradient is for designing ships. Since the wind tunnels in which the models have been tested also have velocity gradients, experimental values of wind

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15. Hughes, G., "Model Experiments on the Wind Resistance of Ships," INA 1930.
16. Hughes, G., "The Air Resistance of Ship's Hulls with Various Types and Distributions of Superstructures," IESS (1932).
17. Shearer, K. and W. Lynn, "Wind Tunnel Tests on Models of Merchant Ships," NECI, 76 (1960).

drag are not as affected as it might appear at first glance. Secondly, if a prediction of wind drag is being made for zero true wind case, the problem has no significance, since the wind drag is due to the passage of the ship through still air which has no gradient. In any case the difference in wind drag using gradient would be about 10 percent of the total wind drag which is normally 2 to 3 percent of the total resistance of the hull. At the preliminary design stage 0.2 to 0.3 percent of the total resistance is presumably not significant.

5.1 Formulation of Wind Drag

The formulation adopted for the wind drag of the above-water portion of a ship is very straightforward. It has been chosen over the more time consuming scientific methods, since the differences between the results from this formulation and the results from the more sophisticated formulation are within the accuracy of wind tunnel experiments. Although a complete description of this method is presented in this chapter, a more complete discussion of the development of this technique has been published by Wilson and Roddy.[18]

The available wind drag data have been analyzed and reduced to a drag coefficient (C_{AA}) and a heading coefficient (C_γ). The drag coefficient is derived from a wind drag resulting from a relative wind velocity which is opposite in direction from the ship velocity. The relative heading (γ_R), in this case, is defined as zero degrees. The drag at headings other than zero degrees is expressed by the heading coefficient.

5.1.1 The Wind Drag Coefficient

The wind drag coefficient (C_{AA}) for a head wind ($\gamma = 0^\circ$) is expressed by the following formula

$$C_{AA} = \frac{R_{AA_o}}{\frac{1}{2}\,\rho_A\,A_V\,V_R^{\,2}}\,, \tag{5.1}$$

where R_{AA_o} = wind drag for a head ($\gamma = 0^\circ$) wind (lb or N),

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18. Wilson, C. J. and R. F. Roddy Jr., "Estimating the Wind Resistance of Cargo Ships and Tankers," NSRDC Dept. of Hydromechanics Research and Development Report 3355 (May 1970).

A_V = above-the-water transverse area (ft^2 or m^2),

V_R = relative wind velocity (ft/sec or m/sec),

and ρ_A = mass density of air (which is generally taken as $0.00237 \frac{lb\ sec^2}{ft^4}$ or $1.221 \frac{N\ sec^2}{m^4}$).

Representative values for the wind drag coefficient have been determined from analyses of wind drag data using Equation 5.1. An average wind drag coefficient of 0.45 for aircraft carriers has been determined from the analyses of data for CV 9 and CVE 55. Based on the analyses of data for the other combatant ships (CA 139, CL 145, DD 445, DD 692, and LST 1156), the average wind drag coefficient for this type of ship is 0.70. A wind drag coefficient of 0.75 is appropriate for naval auxiliaries at the full-load condition.[18]

5.1.2 The Heading Coefficient

The heading coefficient (C_γ) is expressed by the following formula.

$$C_\gamma = \frac{R_{AA_\gamma}}{R_{AA_o}}, \tag{5.2}$$

where R_{AA_γ} = wind drag at any non-zero relative wind heading (lb or N).

The analysis of the wind drag for all the combatant ships, including the carriers, indicates that the behavior of C_γ, as a function of the relative wind heading (γ), is essentially the same for all these ships. Plots of the heading coefficient (C_γ) versus the realtive wind heading (γ) are presented in Figures 5.1 and 5.2, for naval combatants and auxiliaries, respectively.

It should be noted from the plots of the heading coefficient (C_γ) that the maximum value of C_γ occurs near relative wind headings (γ) of 30 degrees and 150 degrees. Also, there is typically a rather flat spot in the curve at about 80 degrees. Since relative winds from the after quarter (90 degrees through 180 degrees) are not normally very large (unless the true wind speed is very high compared to the ship speed), the area of greater concern is the forward quarter (0 degrees through 90 degrees). The variation in C_γ from the average curve for combatant ships shown in Figure 5.1 is about $\pm$ 10 percent in the range from 0 degrees through 60 degrees. Around the flat spot at 80 degrees the variation is greater than $\pm$ 10 percent. However, since the magnitude of C_γ has decreased substantially from that at 30 degrees, the absolute error in this region is not greater than at 30 degrees.

5.2 Wind Drag and $P_{E_{wind}}$, Estimation of

The wind drag and the change in effective power or speed due to that drag can be determined using the wind drag coefficient (C_{AA}) and the heading coefficient (C_γ), each of which are discussed in Section 5.1 of this chapter.

5.2.1 Generalized Technique

The wind drag (R_{AA}) at any relative wind heading (γ) for the above-water portion of a ship can be calculated using the following formulation,

$$R_{AA_\gamma} = \frac{\rho_A}{2} C_{AA} A_V V_R^2 C_\gamma . \qquad (5.3)$$

When the standard density of air ($.00237 \frac{\text{lb sec}^2}{\text{ft}^4}$) is used and English units (V_R in knots), are used, Equation 5.3 may be re-written as follows:

$$R_{AA_\gamma} = \frac{C_{AA} A_V V_R^2 C_\gamma}{296.2} .$$

The effective power ($P_{E_{wind}}$) required to overcome the wind drag (R_{AA_γ}) may then be written

$$P_{E_{wind}} = \frac{R_{AA_\gamma} V_s}{325.9} \text{ (horsepower)}, \qquad (5.4.1)$$

where R_{AA_γ} is in pounds, V_s is in knots, or $$P_{E_{wind}} = \frac{R_{AA_\gamma} V_s}{1000} \text{ (kilowatts)}, \qquad (5.4.2)$$

where R_{AA_γ} is in newtons (N) and V_s is in metres per second (m/sec).

5.2.2 Specialized Technique for Zero True Wind

Generally, estimates of powering performance are made for zero true wind. Since the relative wind velocity (V_R) is equal to the ship speed (V_s) and the heading coefficient (C_γ) is equal to unity, the still-air effective power may be determined from the following formula.

$$P_{E_{wind}} = \frac{C_{AA} A_V V_s^3}{Constant} \qquad (5.5)$$

where the Constant = 96,500 for horsepower, using English units (A_V in ft^2, V_s in knots, and a mass density of 0.00237 lb sec^2/ft^4),

or where the Constant = 1638 for killowats, using SI units (A_V in m^2, V_s in m/sec, and a mass density of 1.221 N sec^2/m^4).

5.2.3 Supplementary Information

Occasionally it is necessary to convert the effective power due to wind drag ($P_{E_{wind}}$) to a change in ship speed ($\Delta V_{s_{wind}}$). It is necessary in those cases to determine the slope of the relevant speed-power curve at the speed (V_s) of interest. The change in ship speed can then be determined from

$$\Delta V_{s_{wind}} = P_{E_{wind}} / (\Delta P_E : \Delta V_s)$$

where $\frac{\Delta P_E}{\Delta V_s}$ = the slope of the speed-power curve.

When the dimensions necessary to calculate the transverse area (A_V) are not available, it may be estimated for some ships per the discussion in Section 1.1.3 of Chapter 1.

There is one component of ship resistance due to wind which has been largely ignored to date. If there is a strong wind on the beam, the ship will make leeway which must be overcome by using the rudder. The current practice in conducting trials on Navy ships is to try to schedule powering trials to avoid weather conditions which involve average rudder angles greater than 3 degrees. Wagner[19] has proposed a method for correcting the resistance of the ship for the effects of beam winds which should be investigated at greater length. In essence he computes an effective longitudinal force which is composed of the usual resistance force modified by the component due to drift of the hull when subjected to winds at angles of attack other than zero. This component of the force can become quite sizeable for relatively low powered ships or those with large superstructures.

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19. Wagner, B., "Windkrafte an Uberwasserschiffen," Jahrb. STG 61, 1967.

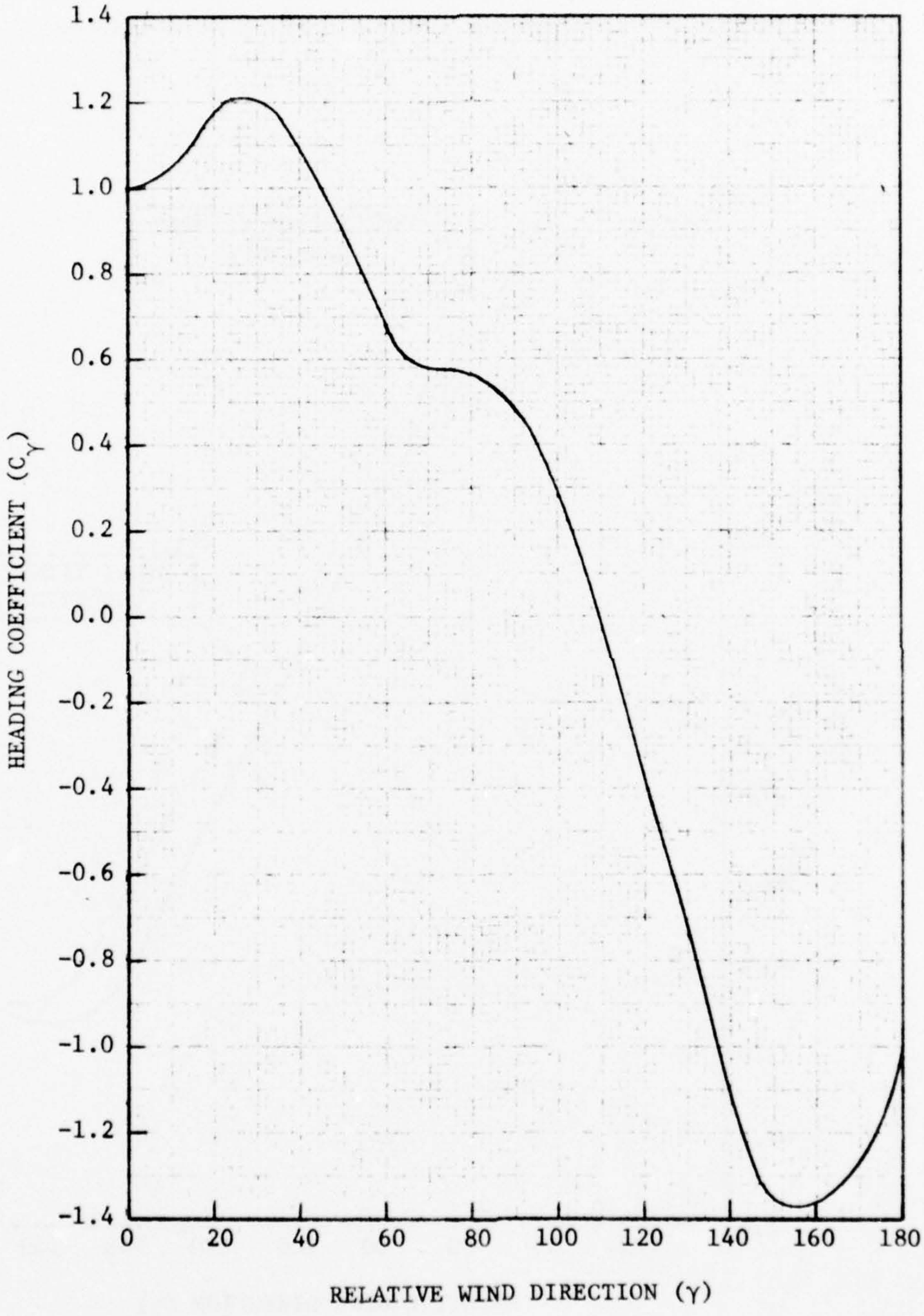

Figure 5.1 - Heading Coefficient (C_γ) versus Relative Wind Heading (γ_R) for Combatant Ships

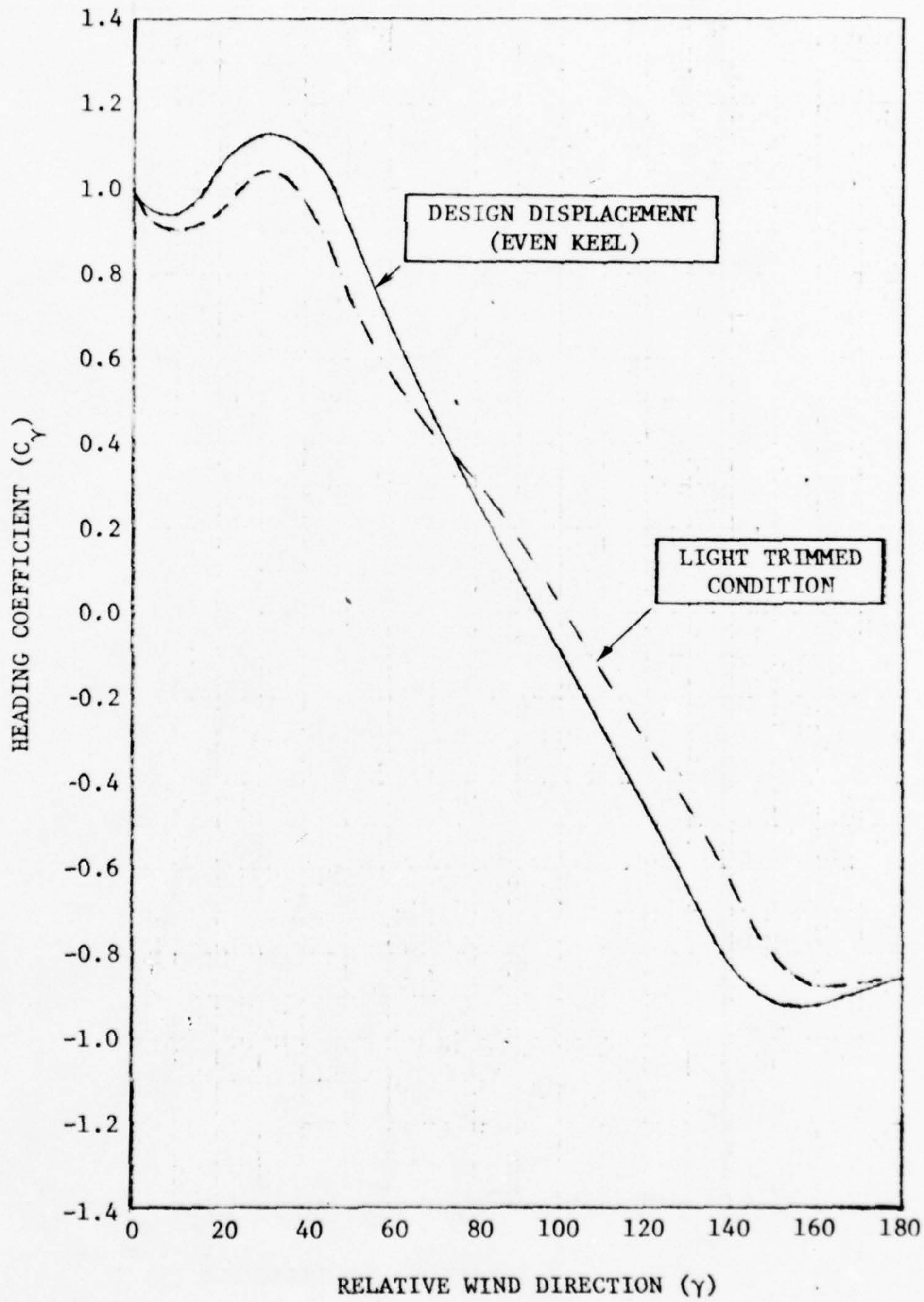

Figure 5.2 - Heading Coefficient (C_γ) versus Relative Wind Heading (γ_R) for Naval Auxiliary Ships

Chapter 6.0 - MARGINS OF UNCERTAINTY, UTILIZATION OF

6.1 Definition

There are many uncertainties which can affect an estimate of the powering performance of a ship. Although accurate estimates of the performance of a ship can be made using analytical techniques during the early phases of design, the lack of specific definition of the hull form, the appendages, the propeller(s), and the superstructure frequently limits the accuracy somewhat. Even after the hull form and its appurtenances have been fully defined, changes of some type are inevitable up through the end of the Contract Design Phase.

The effects of many of the changes (design alterations) that occur during the evolution from Conceptual through Contract Design are estimable after the details of each such change become available. A discussion of some of the applicable techniques is presented in Chapter 8. It is not possible, however, to estimate the effect(s) of a design alteration before the information regarding the change becomes available. The effects of a change in displacement, for example, are inestimable if the direction (increase or decrease) and magnitude of a proposed change are not known.

It could be concluded from the preceeding discussion that margins of uncertainty have been developed to cover only the fact that the ship is never fully defined until the Contract Design Phase has been completed. This however, is not the complete story, as there are several other factors accounted for by these margins.

Many minor alterations are made during the normal ship design processes. It is assumed that many of these will have an immeasurable effect on the powering performance of the ship. Although they may not have a measurable effect individually, collectively they may have. The consequences of this philosophy is that the estimates of powering performance are made for a ship that is not precisely the same as the one eventually constructed. This philosophy is applicable throughout the design process, since the effects of many of the minor alterations, such as a minor change in shaft angle ($\approx$ 1/2 degree), might not be measurable even with a model test. Furthermore, since model tests generally carry an accuracy of about $\pm$ 1%, there is even an uncertainty connected with estimates derived from experimental data. These facts necessitate the use of a margin throughout the design process, even with the prediction of power using data from model tests with the contract design propeller.

6.2 Levels of Uncertainty

During the earliest phase of design (Conceptual Design), very little is known about the final ship characteristics. Although a body plan, and frequently a lines plan, are generally developed for each candidate during this phase of design, it is rather unlikely that the selected design will remain unaltered during its evolution through the Contract Design Phase. It is quite conceiveable, for example, that a principal dimension may change due to subsequent engineering considerations (machinery systems, weaponry systems, etc.). Consequently, the margin of uncertainty associated with this phase is rather large.

Later, as the design moves into the Preliminary Design Phase, the hull-form and appendage definitions are crystallized somewhat. The increased availability and permanence of geometrical details makes it possible to more accurately estimate the ship's powering performance. Eventually, in the Contract Design Phase, a propeller is designed for the ship and evaluated through model experimentation. At this point, the uncertainty has been reduced to its lowest level, where a margin need only account for small differences between the model and the ship and experimental accuracy.

6.3 Selecting an Appropriate Margin

It is necessary that the level of uncertainty during the design process be accounted for by using a margin. Since the slope of a speed/power curve varies significantly with speed, or more precisely, with Froude number, a single power margin is applied through the entire speed range. Naturally this power margin should be commensurate with the actual level of uncertainty and should be applied directly to the estimated power at each phase of design.

An investigation,* which consisted of a comparison of trial data with powering performance predictions, has recently been conducted to determine what the margin policy should be during each phase of design. The recommendations from this investigation regarding a power margin policy, which were issued as a NAVSEC instruction (Ship Engineering and Design Department Instruction 9020.8 of 18 October 1974), are as follows:

(a) 11% at the end of Conceptual Design; prior to model tests

(b) 9% at the end of Preliminary Design; prior to model tests

(c) 6% at the end of Preliminary or Contract Design; based on model test data from propulsion tests using stock propeller(s), results adjusted to reflect the estimated performance of the contract design propeller(s).

(d) 3% at the end of Contract Design; based on model test data from propulsion tests using the design propeller(s).

These margins are applied to the fully-appended effective power which includes the augmentation due to still-air drag.

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*See reference 4, Appendix A entitled "Analysis of Speed-Power Margin-Summary".

Chapter 7.0 - SHAFT POWER, ESTIMATION OF

7.1 Experimental Technique

Propulsion experiments are conducted with ship-models to satisfy either one or both of the following objectives:*

(a) to verify previously made analytical estimates of the powering performance of a ship

(b) to determine the nature of the environment in which the propeller must operate

The method used to conduct these experiments is presented in Section 15 of Chapter 7 in Reference 5. Essentially the model is self-propelled such that the propeller will operate at a loading condition corresponding to that of the ship. This procedure is followed for numerous speeds, covering the entire range of operating ship speeds. Two other types of experiments are necessary to satisfy objectives (a) and (b), above. The results from one of these experiments, the fully-appendaged resistance test, are necessary for the calculation of the fully-appended effective power (P_E), the propulsive efficiency (η_D) and the thrust-deduction factor (1-t), and open-water characterization of the propeller provides the additional data needed to calculate the open-water efficiency (η_O), the relative rotative efficiency (η_R), and the thrust-wake factor ($1-w_T$). The methods by which one determines these coefficients from the experimental data is discussed in Appendix B.

Although it is the practice to determine the value of each of these efficiencies and factors (η_D, η_O, η_R, 1-t, and $1-w_T$), only one of them, the propulsive efficiency (η_D), is needed to calculate the shaft power (P_S), knowing the fully-appended effective power (P_E). The others are merely components of the propulsive efficiency (η_D), as follows:

$$\eta_D = \eta_O \, \eta_H \, \eta_R \, \eta_S \, , \tag{7.1}$$

where η_O = open-water propeller efficiency

η_H = hull efficiency which is $= (1-t)/(1-w_T)$,

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*See Reference 4, Chapter 2.0 entitled "Resistance and Propulsion Model Test Programs."

η_R = relative rotative efficiency, which is

$= \eta_B/\eta_0$,

where η_B = propeller efficiency behind the hull,

and η_S = shaft transmission efficiency.

It is not feasible to accurately determine the actual power delivered to the propeller (P_D) during a full-scale trial. Consequently, the shaft power (P_S) is determined from measurements of torque as far aft as possible on the propeller shaft. The ratio of delivered power to shaft power (P_D/P_S) is referred to as the shaft transmission efficiency (η_S). Since it has been determined that the differences between P_D and P_S are generally very small ($\leq 0.5\%$), it has become the standard practice to assume that $\eta_S = 1.0$.

The standard definition* for the propulsive efficiency is

$$\eta_D = P_E/P_D. \tag{7.2}$$

Since it has been stated that $\eta_S = 1.0$, and therefore that $P_D = P_S$, equation 7.2 may be re-written as

$$\eta_D = P_E/P_S;$$

and consequently,

$$P_S = P_E/\eta_D. \tag{7.3}$$

It should be noted that the fully appended effective power (P_E) used in the calculation of P_S should include an appropriate correlation allowance, a wind drag for still air, and an appropriate margin of uncertainty.

7.2 Analytical Techniques

When experimental data are not available for the design being considered, estimates of the efficiencies and factors can frequently be made using sets of data from similar ships, in a manner similar to that

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*This definition is documented in References 2 and 20. Another propulsive efficiency (η_P), as described in Reference 5, is defined as P_E/P_S. Since η_S is assumed to be unity, $\eta_D = \eta_P$; the notation used herein becomes consistent with all three references.

20. Lackenby, H., "Report of Presentation Committee, 13th International Towing Tank Conference," Berlin/Hamburg, Germany, Sept. 1972.

described for the estimation of appendage resistance (see Section 4.3 of Chapter 4.0). Although the discussion of this procedure in Section 2.4.3.4 of Reference 4 is quite accurate, its implications, that the procedure is "simple" and that "trends" from certain sources can be used to modify these estimates, lead one to believe that these quantities can be predicted quite accurately using a limited amount of data. The procedure is not that good, especially, if one does not consider all those factors which affect these efficiencies. Although this subject is discussed at length in Appendix B, it will be presented briefly here, using an excerpt from Reference 21 as an introduction.

"The best source of hull-propeller interaction coefficient (1-t, 1-w and η_R) data for USN ships are the NSRDC model test reports. Frequently, preliminary estimates of 1-t, 1-w and η_R can be made by . . . inspection of the values of these coefficients (as measured in model tests) for similar hull form/propulsor configurations . . ." It is extremely important that these data be for ship configurations that are very similar to the new (untested) ship design. As emphasized in Section B.3.3. of Appendix B, there are many factors which must be considered. Among these are:

1. The principal hull coefficients,
2. The propeller/hull clearance(s),
3. The height of the propeller centerline relative to the ship's baseline, (and/or the area of the propeller disc relative to the midship section),
4. The size and proximity to the propeller of appurtenant structures (struts and rudders),
5. The diameter of the shafting, bearings, etc., relative to the propeller diameter,
6. The direction of propeller rotation,
7. The rake of the propeller.

Although there are techniques (series data and empirical formulations) which would permit one to adjust estimates of these efficiencies and factors according to observed trends, e.g., variations in $1\text{-}w_T$ versus

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21. Johnson, R. S. and P. A. Gale., "The Navy's Hydrodynamics Problems: A Designers View," NAVSEC Report No. 6114-74-25, July 1975.

a particular hull coefficient, caution should be exercised in the use of such information. For example, the trends which apply to single-screw cargo ships cannot be directly applied to twin-screw destroyers.

After the interaction factors ($1-t$ and $1-w_T$) and efficiency (η_R) have been estimated, the open-water propeller efficiency (η_O) must be estimated. Generally series data or one of the other methods described in Reference 3 are used to estimate η_O. Following the calculation of the propulsive efficiency (η_D) using Equation 7.1, an estimate of the shaft power (P_S) may be made using Equation 7.3.

7.3 Propeller Cavitation, Effects of

"Cavitation is a phenomenon met with in highly loaded propellers in which, beyond certain critical revolutions, there is a progressive breakdown in the flow and a consequent loss of thrust. In its extreme form, it may prevent the ship from reaching the desired speed. Before this stage is reached, however, it manifests itself by noise, vibration and erosion of the propeller blades, struts and rudders. ..."*

Although the consequences of noise, vibration, and erosion can be detrimental to the ship or its mission, the consequence of a thrust loss may mean a significant reduction in the attainable speed for a ship. Therefore, an effort should be made, early in the design process, to determine whether-or-not there is a possibility that cavitation will be a problem. If there is such a possibility, estimates of powering performance must be adjusted to reflect the effect of cavitation.

Prior to experiments with a model of the design propeller, estimates of a possible thrust loss can be made using Reference 22. After these experiments have been conducted, the estimates can be refined based on the results from these experiments.

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*From Section 16, Chapter VII, of Reference 5.

22. Gawn, R. W. L. and L. C. Burrill, "Effect of Cavitation on the Performance of a Series of 16 Inch Model Propellers," Institute of Naval Architects, 1957.

Chapter 8.0 - EFFECTS OF DESIGN ALTERATIONS, ESTIMATION OF

The majority of design alterations involve minor changes in the displacement of a ship. Consequently, the relevancy of the most frequently used estimating technique shall be explored by comparing the measured differences in power with the estimates.

8.1 Effects of Displacement Variations

An empirical formulation frequently used to estimate the shaft power (P_S') at a new displacement (Δ') is as follows:

$$P_S' = P_S\,(\Delta'/\Delta)^M \qquad (8.1)$$

where M = 1 or 7/6, depending on ship type and displacement difference,

and P_S = the shaft power at the original displacement (Δ)

An attempt has been made to demonstrate the usefulness of this formula as applied to some typical military-type ships.

8.1.1 Single-Screw Combatant Ships

Resistance and propulsion experiments were conducted at the design displacement of a typical single-screw combatant-type ship, and at two other displacements, which were $\pm$ 1 foot in draft from the design draft. The higher displacement was 11.9% above the design displacement and the lower displacement was 11.0% below the design displacement. Although design alterations do not always yield such large changes in displacement, this example should be useful in establishing the value of this formulation for anything less than an extreme case.

The experimentally determined values of shaft power (P_{S_x}) are compared with the empirically ($M = 7/6$) estimated shaft power (P_{S_e}) in Figure 8.1. The estimated shaft power (P_{S_e}), expressed as a ratio of the experimental P_{S_x} at the design displacement, is represented by the solid curve. The experimental power, which is also expressed as a ratio of the experimental P_{S_x} at the design displacement, is plotted for several speed-length ratios for both of the off-design displacements. It is apparent from this presentation that the empirical method is quite good for estimating the change in P_S at the higher displacement, especially at the higher speeds. At the lower displacement, however, the empirical method seems to significantly over-estimate the change in P_S due to the decrease in displacement.

This information is also presented in a different manner in Figure 8.2, in which the prediction error is expressed as a percentage of the empirically determined shaft power (P_{S_e}) at each displacement for the entire range of speed-length ratios. It can be seen from these curves that the empirical estimate is as much as 6% below the experimental prediction for the lower displacement at the lower speeds. At the higher displacement, however, it is obvious that the error is less than 2% throughout the speed range.

8.1.2 Twin-Screw Combatant Ships

Resistance and propulsion experiments were conducted at the design displacement of a typical twin-screw combatant-type ship, and at three off-design displacements and three off-design static trim conditions. The three displacements, expressed as a percentage of the design displacement, are 95.0%, 102.6%, and 105.1%.

The results of these experiments are presented in Figure 8.3. The results for all three off-design displacements are presented as the prediction error, expressed as a percentage of the empirically determined ($M = 7/6$) shaft power (P_{S_e}), for the speed range documented by experimental data. The three curves presented, which cover a displacement variation of approximately $\pm$ 5% from the design, indicate that the prediction error is, in general, less than 2%. Specifically, the higher displacements (102.6% and 105.1%) show prediction errors that are within $\pm$ 1 percent, whereas at the lower displacement (95%) the prediction error is between 1 and 2%.

8.1.3 Summary

It would appear that the empirical formulation is moderately good for estimating the changes in shaft power due to small changes in displacement for both the single- and the twin-screw combatant-type ships using an exponent (M) of either 7/6 or 1. Certainly the data presented indicate that the method could be used to estimate the changes in P_S due to an increase in displacement of up to about 5% without any measurable error being introduced. If the method were to be used to estimate the change in P_S due to a decrease in displacement, the same limit of 5% could be used, bearing in mind that the method tends, according to the data presented herein, to be slightly less accurate with decreases in displacement. In general however, when a change in displacement greater than 5% is being considered, it may be desirable to reanalyze the entire design problem. The result might be a revised hull form optimized for the new design displacement.

8.2 Effects of Changes in Initial Static Trim

Resistance and propulsion experiments were conducted at the design even-keel displacement of a typical twin-screw combatant-type ship, and at three non-zero initial static trim angles. The shaft power (P_{S_x}) data derived from the experiments at the non-zero trim angles have been compared to the data for the even-keel (trim angle = 0 degrees) condition. These comparisons are presented as a percentage change in P_S for each of the trimmed conditions in Figure 8.4. These curves reveal an insignificant (less than 1%) change in P_S occurs with an initial static trim of 0.11 degrees by the bow. The experiments conducted with trim by the stern, however, do reveal significant differences. For example, at a speed-length ratio of 1.05, the ship having an initial static trim of 0.22 degrees (0.4% of L_{WL}) by the stern would absorb nearly 4.5% more shaft power than its counterpart having an initial static trim of 0.0 degrees. There was a large change in transom immersion in this case.

Certainly the evidence presented in Figure 8.4 indicates a strong dependence of the shaft power upon the initial static trim angle.

Certainly, from such a small sample of data it would be unreasonable to attempt to develop any tool for use in estimating the effects of initial static trim. It does, however, point out that this parameter may be critical in ship operations, and should not be overlooked. Normally, small variations in trim (less than 1 1/2% L_{WL}) do not cause any significant variations at deep drafts but may be significant if the propeller tips are close to the water surface or if there is a significant change in transom immersion.

8.3 Effects of Propeller Changes (Diameter, Pitch, etc.)

The effects of these types of changes on shaft power are, in general, accounted for by first considering the magnitude of the resultant change in open-water propeller efficiency. The more subtle effects, such as changes in the interaction coefficients, are frequently small when compared to the effect on the open water efficiency. If the propeller diameter or location is changed there will probably be a change in the thrust-wake factor ($1-w_T$). As propeller diameter increases, w_T will normally decrease as a larger area of comparatively undisturbed water is being swept. As the propeller is moved away from the hull surface to increase clearance it is normally anticipated that w_T will decrease. At the present there is some evidence that skewed propellers of the latest design usually result in a change in value of w_T compared to more conventional designs. This is presumably due to the fact that the high skew induces a rake in the propeller blades which changes the location of the propeller with respect the the hull surface. Very recent propeller design practice has been to correct for this movement by raking the blades forward a certain amount.

It is believed that variations in propeller load distribution and in rake are responsible to some degree for the differences in the thrust-wake fraction (w_T) measured with different propellers (see Figures 4 and 5 in Reference 21). This is an area which needs to be investigated because it is one more complication in an already complicated situation. At the present time it is standard practice at NSRDC to repeat the propulsion experiment with the stock propeller to check minor changes in instrument performance before proceeding with the design propeller. It is unfortunate that many times minor changes are made to shafts, struts and skegs after the original stock propeller propulsion experiment which tend to complicate the picture. This is particularly true when the original appendages were sized for a fixed-pitch propeller and then increased for a controllable-pitch propeller.

8.4 Effects of Appendage Changes (Rudders, Bilge Keels, Stabilizer Fins, etc.)

An increase or decrease in the size of any of these appendages may be accounted for by considering the change in wetted area. For example, a wetted surface increase of 1% would result in an increase in P_E of about 1% and a P_S increase of about 1%. If a new appendage were to be added to a hull, it is recommended that a slightly more conservative method be used to estimate the change in effective power. It is suggested that a factor of 1.2 be used in conjunction with the wetted surface to account for the residuary resistance of the new appendage. If, for example, a set of bilge keels, which would increase the ship's wetted surface by 3%, were added to the ship, a crude, but conservative estimate of 3.6% (1.2 * 3.0%) would be added to the previously determined effective power. The only case where this procedure might cause a significant error is when the appendage is in close proximity to the propeller.

In the case of a rudder, located behind a propeller, certain changes will most likely affect the thrust deduction, while not likely affecting the wake fraction. If the thickness, fore-and-aft clearance, or the span are changed, the thrust deduction factor (1-t) will change. Although it is not currently possible to estimate the magnitude of such a change, qualitatively speaking, the thrust deduction fraction (t) tends to increase with either increasing rudder size or decreasing propeller-rudder clearance.

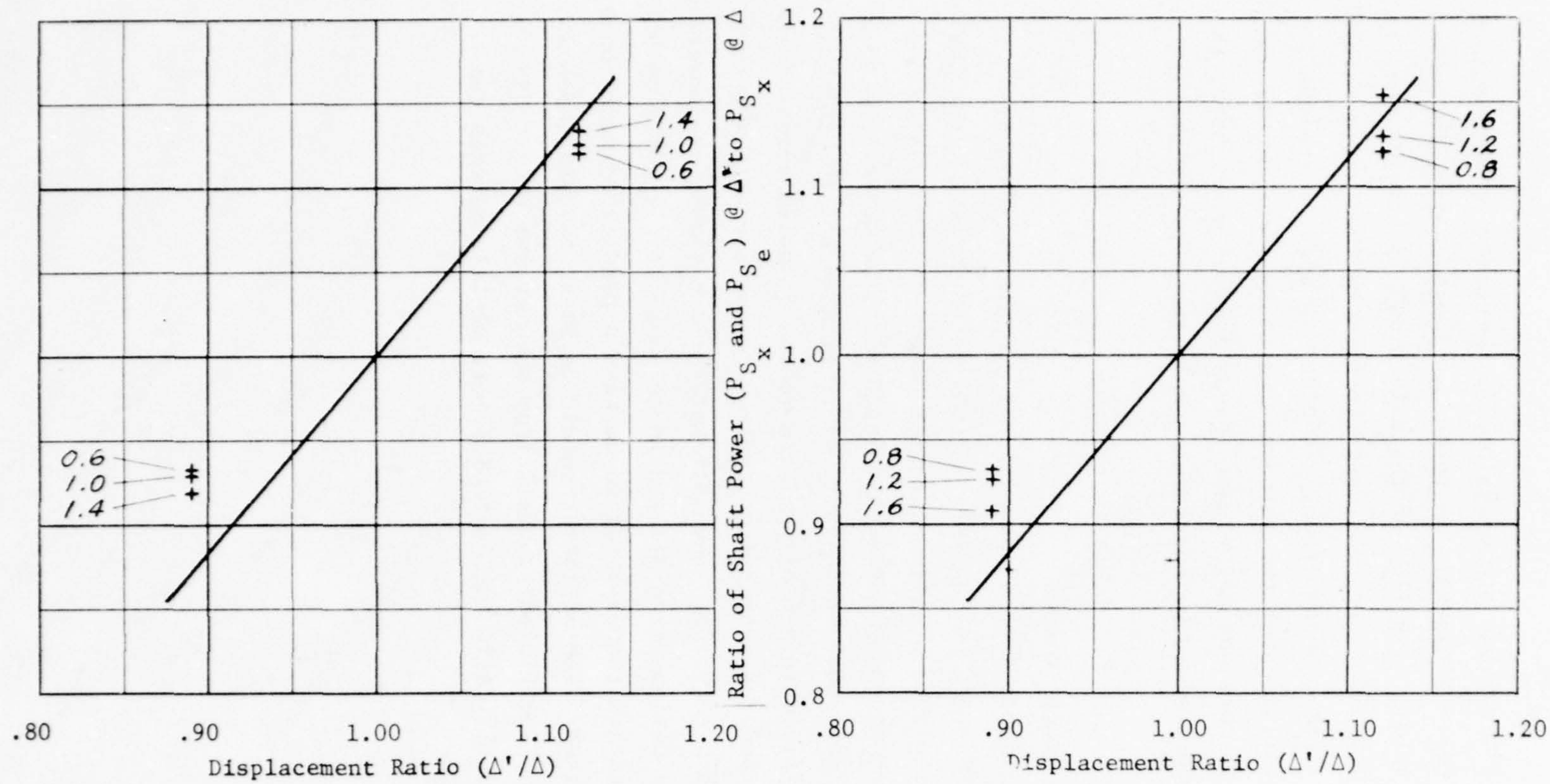
8.5 Effects of Hull Form Changes

A major hull form change should be defined as one which is perceivable in the sectional area curve and/or which significantly affects the shape and/or smoothness of the waterlines and buttocks in the area being altered.

These types of alterations are occasionally necessary to accommodate a change in internal arrangements or to obtain improved flow patterns in a particular region of the hull. If the alteration is a major one, it is suggested that its effect be considered to be more than would result from displacement or trim changes alone. In other words, the residuary resistance characteristics of the ship may change by a measurable amount. If the alterations are not major, per the definition given here, the effects may be determined by considering only the displacement and trim changes.

8.6 Effects of Other Changes

Although a change from a conventional propulsion system to a CRP propulsion system would not usually be considered to be a minor design alteration, it shall be briefly discussed here. The penalty of the greatly increased appendage drag associated with the conversion to a CRP system is a well documented fact. This penalty and the changes in interaction coefficients are all related to the relative sizes of the shafting, hubs, and the supporting struts commensurate with the control mechanism for such a system. When attempting to estimate the powering performance of a design being altered in this manner, it would be better to start anew, rather than attempt to correct a prediction made for a conventional system. The estimates should be based on data (appendage drag, interaction coefficients, etc.) from some of the recent CRP applications, specifically the one which most closely matches the particular design.



Note: 1-The solid curve represents the Empirical Formula: $P_{S_e} = P_{S_x} (\Delta'/\Delta)^{7/6}$

2-The symbol (+) represents the experimental shaft power (P_{S_x}) at displacement ratios (Δ'/Δ) of 0.89 and 1.12, at the speed length ratios indicated.

Figure 8.1 - Comparison of Experimentally Determined Shaft Power (P'_{S_x}) to Empirically Estimated Shaft Power (P'_{S_e}) at Six Speed-Length Ratios for Two Off-Design Displacements for a Single-Screw Combatant-Type-Ship

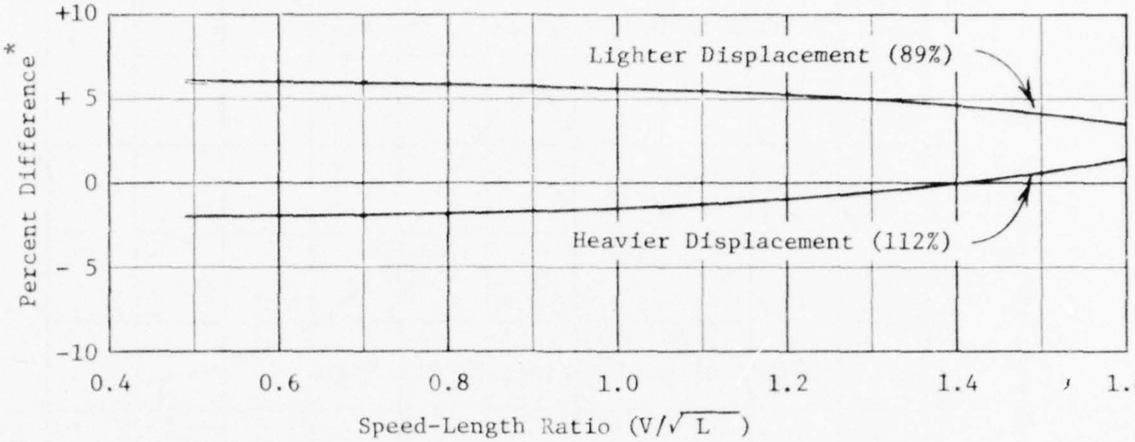


Figure 8.2 - Prediction Error versus Speed-Length Ratio for a Single-Screw Combatant-Type Ship at Two Off-Design Displacements

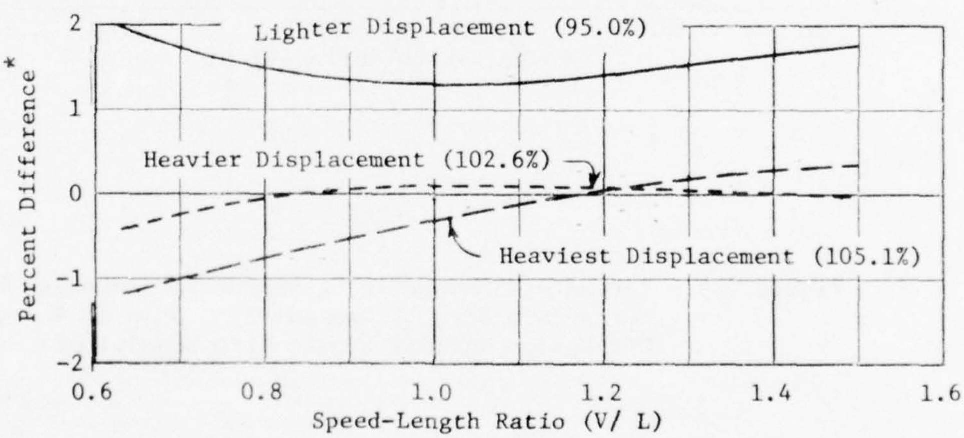


Figure 8.3 - Prediction Error versus Speed-Length Ratio for a Twin-Screw Combatant-Type Ship at Three Off-Design Displacements

*The percentage differences reflect the differences between the experimentally determined shaft power (P_{S_x}) and the empirically estimated shaft power (P_{S_e}), according to the formula:

$$\% \text{ Difference} = \frac{100\,(P_{S_x} - P_{S_e})}{P_{S_e}}$$

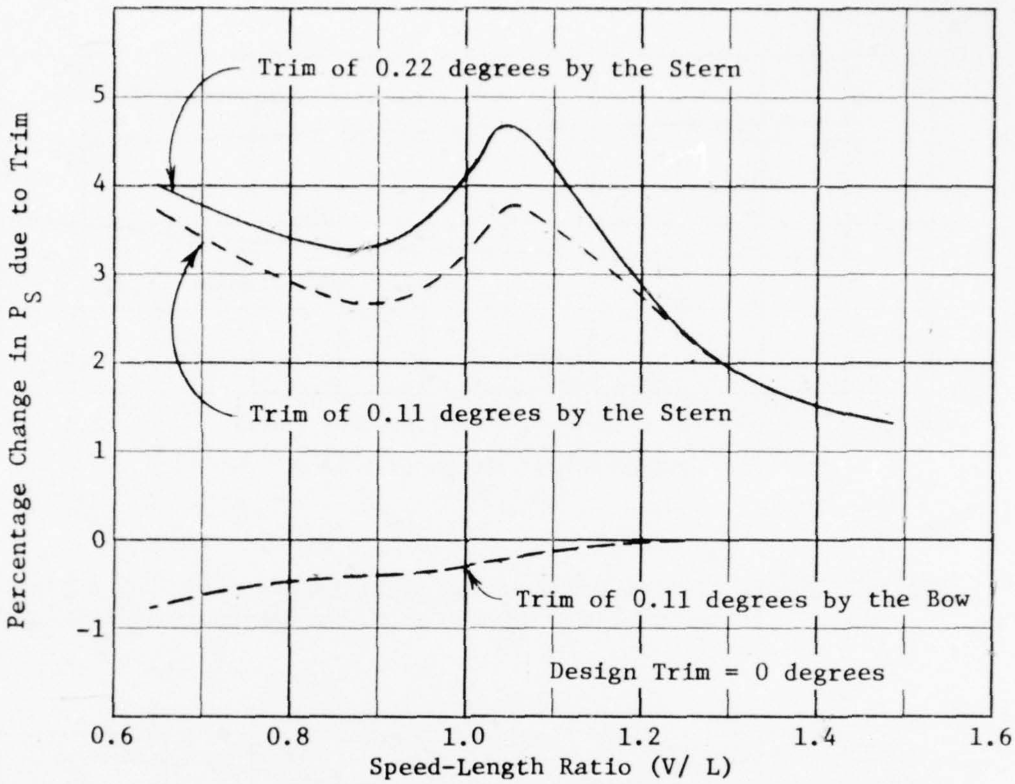


Figure 8.4 - Percentage Change in P_S versus Speed-Length Ratio for a Twin-Screw Combatant-Type Ship at Three Off-Design Initial Static Trim Conditions

Chapter 9.0 - ROUGH WATER PERFORMANCE, ESTIMATION OF

The prediction of ship performance has traditionally and deliberately been divided into two distinctly separate parts, one for calm water and one for rough water. Although there is much room for improvement in the art of estimating the calm-water performance of a ship, the Naval Architect is readily able to estimate calm-water performance and to compare that performance with the performance of any other ship. The art of estimating and describing rough-water performance is not nearly so well developed.

9.1 State-of-the-Art

The characteristics that are needed to describe the rough-water performance of a ship are the added resistance, the motions (local velocities and accelerations due to pitch, heave, etc.), and the consequences of these motions (slamming, deck-wetness, habitability, etc.). Since this manual documents only the powering performance prediction practice, this chapter will focus on the speed limiting aspects of rough water performance, rather than other design criteria, such as freeboard, strength, etc.

The added resistance in rough water may result in an involuntary speed reduction if there is not a sufficient amount of power to maintain speed. The effects of ship motions on personnel and/or ship systems may lead to a voluntary reduction in speed and/or a change in heading. Although it is within the state-of-the-art to estimate those performance characteristics which lead to voluntary or involuntary speed reductions, there are some shortcomings in the most widely used techniques, a few of which are as follows:

(1) The estimates of performance are for unidirectional long-crested head seas only.

(2) The one-parameter energy spectrum (Pierson-Moskowitz) represents only:

  (a) a fully-developed sea, not a developing or decaying sea condition, and

  (b) a single sample of a fully-developed sea having a particular significant wave height, e.g., there are other spectra which represent a fully-developed sea having the same significant wave height (Newmann, measured sea spectra, etc.).

These limitations and others may be reduced or eliminated as a result of some of the ongoing work in this field, as briefly discussed in Section 9.3.

9.2 The Estimating Techniques

Although the discussion in Section 9.1 indicates that there are limitations on the usefulness of the estimating techniques, many techniques are available for making estimates in a fully-developed sea spectrum. The techniques which may be used to estimate ship motions, and the consequences of motions, such as slamming and deck wetness, etc., are discussed briefly in Reference 23. The techniques which may be used to estimate the added resistance in waves are discussed in the following sections.

9.2.1 Analytical Methods

Techniques have been developed to estimate the added resistance in head seas based on (1) a set of experimental data and (2) a theoretical hypothesis. The first of these resulted from a regression analysis of model experimental data for single-screw ships, as documented in References 24 and 25. The formulation derived therefrom is

$$P_{AW} = \frac{L^{3.5}}{\text{Constant}} \left[A_0 + A_1 \; (C_B - .5)^5 + A_2 \frac{L}{B} + A_3 \frac{L}{T} + A_4 \frac{\overline{FB}}{L} + \right.$$

$$\left. A_5 \frac{k_{yy}}{L} + \beta(A_6 + A_7 \frac{L}{T}) + A_8 \frac{V}{\sqrt{L}} \right], \qquad (9.1)$$

where $A_0, A_1, \cdots A_8$ are the regression coefficients given in Table 9.1;

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23. Hubble, E. N. and J. B. Hadler, "Prediction of Ship Motion in Regular and Irregular Head Waves," DTNSRDC Ship Performance Department Report SPD-623-01, April 1975.
24. Moor, D. I. and D. C. Murdey, "Motions and Propulsion of Single Screw Models in Head Seas," Transactions, Royal Institution of Naval Architects, Vol. 110, No. 4, October 1968.
25. Moor, D. I. and D. C. Murdey, "Motions and Propulsion of SIngle Screw Models in Head Seas, Part II," Transactions, Royal Institution of Naval Architects, Vol. 112, No. 2, April 1970.

$k_{yy}/L = 0.25$, unless k_{yy} is known;

Constant $= 5.74 \times 10^4$, for P_{AW} in horsepower

$= 4.28 \times 10^4$, for P_{AW} in kilowatts;

length (L) is in feet;

and all ratios and coefficients are per the definitions in the list of notation.

Although this method can be an extremely valuable process by which one may evaluate conceptual design variations, one must be cognizant of the fact that it is based entirely on model data for single-screw ships.* Caution must be exercised in its usage since it has not, as yet, been verified by full-scale data. The regression coefficients for Equation 9.1 are given in Table 9.1.

The second type of technique is based on Maruo's hypothesis. This hypothesis states that the added resistance of ships in a seaway is proportional to the square of the wave height and to the ship motions and their phase relationships to the wave field. Furthermore, the added resistance is independent of the calm-water resistance. Several techniques of this type have been developed in recent years and are discussed in Reference 27. Since, with any of these techniques the accuracy of the estimate of motions is extremely important, it can be said that the estimate of added resistance will be no better than the estimate of ship motions. Consequently, it is recommended that these techniques be used only to develop the data used in comparisons between different designs. Experimental procedures should be used for the numerical evaluation of any particular design.

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* Some data for twin-screw ships is available in Reference 26. These data may be helpful in estimating the effects of variations in a few of the principal hull-form parameters for twin-screw naval auxiliaries.

26. Moor, D. I., "Effects on Performance in Still Water and Waves of Some Geometric Changes to the Form of a Large Twin-Screw Ship," Transactions, Society of Naval Architects and Marine Engineers, Vol. 78, 1970.

27. Strom-Tejsen, J., H. Y. H. Yeh, and D. D. Moran, "Added Resistance in Waves," Transactions, Society of Naval Architects and Marine Engineers, Vol. 81, 1973.

TABLE 9.1 -- COEFFICIENTS FOR EQUATION 9.1

Beaufort Number	LBP	A_0	A_1	A_2	A_3	A_4	A_5	A_6	A_7	A_8	Standard Error
5	400	-0.615	41.4	-0.115	0.0091	0.040	2.91	0.209	-0.0073	1.528	0.07
	500	-0.408	28.5	-0.063	0.0086	0.026	1.68	0.109	-0.0037	0.891	0.05
	600	-0.194	16.5	-0.034	0.0074	0.015	0.70	0.060	-0.0021	0.454	0.03
	800	-0.044	6.6	-0.011	0.0024	0.005	(0.11)	0.023	-0.0009	0.128	0.01
	1000	-0.015	2.8	-0.004	0.0014	0.002	(0.03)	0.009	-0.0004	0.041	0.01
6	400	-1.044	80.0	-0.247	0.0179	0.076	5.44	0.470	-0.0172	3.111	0.16
	500	-0.919	63.2	-0.166	0.0149	0.059	4.22	0.297	-0.0104	2.235	0.11
	600	-0.626	45.0	-0.101	0.0140	0.042	2.59	0.175	-0.0060	1.400	0.07
	800	-0.185	19.0	-0.037	0.0092	0.017	0.61	0.069	-0.0026	0.472	0.03
	1000	-0.066	9.4	-0.015	0.0047	0.007	(0.17)	0.032	-0.0013	0.179	0.02
7	400	-1.358	126.0	-0.394	0.0285	0.110	7.77	0.753	-0.0283	4.761	0.26
	500	-1.308	97.1	-0.288	0.0223	0.092	6.59	0.539	-0.0195	3.675	0.18
	600	-1.113	78.1	-0.201	0.0191	0.073	5.08	0.357	-0.0125	2.702	0.13
	800	-0.496	39.9	-0.085	0.0159	0.037	1.89	0.153	-0.0054	1.156	0.07
	1000	-0.179	20.4	-0.039	0.0098	0.017	0.58	0.075	-0.0029	0.482	0.03
8	400	-1.630	191.8	-0.585	0.0413	0.150	10.58	1.061	-0.0405	6.721	0.44
	500	-1.586	140.0	-0.431	0.0326	0.125	8.76	0.820	-0.0306	5.253	0.28
	600	-1.481	112.2	-0.326	0.0264	0.105	7.41	0.605	-0.0219	4.157	0.21
	800	-0.984	72.4	-0.171	0.0211	0.067	4.23	0.301	-0.0105	2.320	0.12
	1000	-0.459	39.5	-0.084	0.0166	0.036	1.73	0.152	-0.0055	1.113	0.07

Values in brackets are statistically non-significant.

A computer program (YF17GERI) *is available to calculate the added* resistance response curves using the YF17 motion program[28] modified to calculate added resistance using the Gerritsma method. The added resistance response curves thus obtained can then be applied to any desired sea spectrum to obtain the average added resistance ($\overline{R}_{AW}$) at that long-crested head-sea condition. Since the details of the calculation method will not be discussed here, the reader is referred to Reference 27 for a more detailed description. The input for YF17GERI are the same as for YF17. The output quantities (the added resistance coefficient (σ_{AW}) and the nondimensional frequency of encounter (μ_e), which are calculated at each desired Froude number) are dimensionless and are defined as follows:

$$\sigma_{AW} = \frac{R_{AW}}{\rho g \frac{B^2}{L} \zeta_A} , \tag{9.2}$$

and

$$\mu_e = \omega_e \sqrt{L/g} , \tag{9.3}$$

where

R_{AW} = added resistance in regular waves at a wave encounter frequency, ω_e,

ρ = mass density of seawater,

g = gravitational acceleration,

B = ship beam,

L = ship length,

ζ_A = wave amplitude,

ω_e = circular frequency of wave encounter, $\omega\ (1 + \omega\ V/g)$,

ω = circular wave frequency, $2\ \pi/T$,

and T = wave period.

The average added resistance ($\overline{R}_{AW}$) in a seaway is then obtained using the following equation:

$$\overline{R_{AW}} = \int_0^\infty R(\omega_e)\ S_\zeta(\omega_e)\ d\omega_e , \tag{9.4}$$

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28. Frank, W., and N. Salveson, "The Frank Close-Fit Ship Motion Computer Program," DTNSRDC Department of Hydromechanics Research and Development Report 3289, June 1970.

where $R(\omega_e)$ = response amplitude operator (R_{AW}/ζ^2), the response to a sinusoidal excitation of unit amplitude,

and $S_\zeta(\omega_e)$ = one dimensional spectral density of the seaway.

9.2.2 Experimental Methods

Model experiments can be conducted in regular or irregular waves. Regular wave (sinusoidal) experiments are conducted to determine the response amplitude operator (R_{ω_e}) at many frequencies of encounter (ω_e). These results can be applied to any known sea spectrum $(S_\zeta(\omega_e))$. The experiments in irregular waves, having a particular sea spectrum, yield the average added resistance $(\overline{R}_{AW})$ in that spectrum. Although the irregular wave experiments take considerably less time to conduct and are hence less expensive, the results are currently applicable only to the experimental sea spectrum. However, an analytic technique (cross-bi-spectral analysis), which is currently being developed, should enable one to determine the response amplitude operator (R_{ω_e}) from an irregular wave experiment. If this technique proves to be a viable tool, the irregular wave experiment may replace rather than just serve as a check of the regular wave experiments, as it currently does.

9.3 Recent Progress

The limitations mentioned in Section 9.1 have been addressed by numerous authors in recent years. Certainly those working in the field are well aware of the shortcomings of applying the Pierson-Moskowitz spectral formulation to describe the overall sea environment. This formulation is weak in that it represents only a mean fully-developed sea having a particular significant wave height. Chryssostomidis conducted an investigation for NAVSEC [29] in which he addresses this problem and suggested that a two-parameter spectral formulation be used to represent the seaway. This suggestion is consistent with the recommendation (that a two-parameter wave spectrum of the general Bretschneider

29. Chryssostomidis, C., "Impact on Ship Speed of Installing Fins," George Sharpe, Inc., Report, September 1974.

form be used) adopted by the 12th International Towing Tank Conference (I.T.T.C) in Rome, Italy in 1969. Furthermore, Chryssostomidis recommends that existing data be used to determine the frequencies of occurrence of various significant wave heights and periods in a real sea environment. Other investigators feel that the two-parameter spectral formulation is inadequate, since it is an idealized spectra that is not always representative of the real sea environment. Some investigators recommend the use of the 323 actual sea point spectra measured at Station INDIA in the North Atlantic.[30] Ochi of DTNSRDC is currently working on a six-parameter spectral formulation which uses a band width concept to obtain families of spectra having the same significant wave height and average period. Furthermore, he uses a second frequency range to formulate a spectra for swell.

Chryssostomidis also presents a scheme by which one could characterize all the aspects of a ship's performance in omnidirectional seas.[29] The performance of the ship for all headings (relative to the waves) could be graphed in polar coordinates. It is important to remember, however, that the techniques for estimating ship performance in oblique seas have not been fully developed and validated. It would appear that the recently developed method of Lin and Reed will prove to be a viable estimating technique.[31] It is hoped that this technique can be fully evaluated in the very near future.

9.4 Service Margin

"A service margin is a margin of performance (speed and power) provided in the design of a ship which will enable it to move between two or more points within a given time during some arbitrarily specified future period during its life. This implies being able to achieve a certain average speed over a certain time period. It is measured in relation to

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30. Miles, M., "Wave Spectra Estimated from a Stratified Sample of 323 North Atlantic Wave Records," N.R.C. Report LTR-SH-118A, May 1972.
31. Lin, W. C. and A. M. Reed, "The Second-Order Steady Force and Moment on a Ship Moving in an Oblique Seaway," Office of Naval Research, Eleventh Symposium/Naval Hydrodynamics, London, April 1976.

a specified trial condition (displacement and trim) on a new, clean ship in standard sea water with no wind and a calm sea....."*

The service margin, which is accounted for by specifying a "sustained speed", may be very logically split into two distinctly different parts. One of these may be referred to as the deteriorative part. It encompasses the fouling and corrosion of the hull and propeller and the general breakdown (wear) of the entire propulsion system. Many of these factors can be controlled through systematic maintenance schedules. The subject of fouling is discussed at some length in Appendix A. The other part of the service margin is due to environmental factors. It includes the effects of sea state, true wind, sea water temperature, etc., which generally cannot be avoided and certainly cannot be controlled.

9.4.1 State-of-the-Art

The study presented in Reference 32 indicates that a service margin must account for a rather large variety of factors. Although it is possible to make engineering estimates of the effects of many of these factors, some of the estimating procedures are crude since very little work has actually been done to quantify the components of the service margin. For example, the procedures used to estimate the speed limiting effects of rough water (power and motion limits), are not currently adequate for estimates of speed reduction for a specified operational profile. The deficiencies which lead to this inadequacy are discussed in the preceding sections of this chapter.

9.4.2 Recommended Procedure

Since the state-of-the-art does not permit accurate estimates of the components of the service margin, it is recommended that the NAVSEC practice be maintained.** The service margin is accounted for by specifying

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*This definition was taken from Reference 32.

32. Levine, G. H. and S. Hawkins, "Comments on Service Margins for Ships," Panel H-2, Hydrodynamics Committee, S.N.A.M.E., March 1970.

**The state-of-the-art is fairly well documented in Reference 33.

33. Giblon, R. P., "Service Margins and Power Plant Selection," Transactions, STAR-ALPHA Symposium, Washington, D. C., 1975

a "sustained speed". This sustained speed is defined as being that speed which can be maintained after a certain time at sea in a certain seaway. That speed is taken to be the attainable speed using 80% of the maximum continuous power for a new ship, with a clean bottom, at the designed displacement, in calm water, where the true wind velocity is zero knots. Until such time as better methods are developed or data indicate that a figure of 80% is not a good choice for all, or a subset of, the ships of the Navy, it is recommended that NAVSEC continue with its current practice. A trial speed is also specified as 95% of maximum continuous power before the standardization trial, and 100%, thereafter.

APPENDIX A
THE CORRELATION ALLOWANCE

A.1 Definition

The correlation allowance, C_A, used in the prediction of powering of ships is the naval architect's version of the engineering (correction) factor common to practically all branches of engineering. It is normally a correction to the total resistance coefficient estimated from experiments with a model which will enable the towing tank to predict the shaft power, P_S, of the ship at a given speed. It compensates for a rather large number of variables which influence the flow of water around the surface of a ship, the magnitudes of which are too small, or too imprecisely known to be determined individually. At one time [34] the term roughness allowance, ΔC_f, was used and was applied to the frictional resistance coefficient. This designation is no longer considered appropriate because it became obvious as ship construction improved and hulls became smoother that the allowance was due in part to scale effects between the ship and its model rather than being totally a compensation for surface roughness and protuberances of various sorts. The major items covered in the correlation allowance are usually considered to be: structural roughness, anti-fouling paint roughness, three dimensional form factors, flow through scoops, sea chests, scale effects (e.g. appendages, differences in flow over ship as compared to model, difference between ship and model propeller performance, difference in properties of sea water and tank water not compensated for in other corrections), etc. Obviously the list of differences between ship and model can be extremely lengthy. From the financial and technical viewpoints it is not practical, even if it were possible, to construct the model to exactly represent the ship in every detail, and even if this were done there would still be corrections. Hence, the correlation allowance is used as a standard device to account for these differences.

34. Gertler, M., "The Prediction of the Effective Horsepower of Ships by Methods in Use at the David Taylor Model Basin," DTMB Report No. 576, December 1947.

It has been previously stated that the normal practice was to correlate on shaft power. It is also possibe, of course, to correlate on the basis of thrust which is usually of interest to the propeller designer. One of the primary reasons for selecting shaft power for the correlation is that torsionmeters are fitted on all standardization (powering) trials of U. S. Navy ships, while thrustmeters are not fitted on all ships, usually because of the added expense of buying, installing and removing the thrustmeters during the trial availability, which can be considerable. It is also considered at DTNSRDC that the type of torsionmeter currently in use in the U. S. Navy is more accurate than the present-day thrustmeter used on trials. It is agreed that the preceding statement is still open to debate.

A.2 Standard Correlation Practice at DTNSRDC

It is the normal practice at the Center to conduct powering experiments using a geosim of the actual propeller design on a model of the ship hull at the trial displacement and trim condition. Adjustments in C_A are made such that the shaft power (P_S) predicted from the model experiments agrees with that from ship trials in the upper speed range. It is usually found that the agreement is not as good in the lower speed range due to the typical instrument inaccuracies at low levels of measurement. On occasion, where model data are already available at the proper displacment, but at a different correlation allowance, it is assumed that the propulsion coefficients from the model experiments will not change significantly for minor changes in correlation allowance. This assumption has been verified on numerous occasions. When the correlation is made on the basis of P_S, the assumptions are that the residuary resistance coefficient, (C_R), and the propulsive efficiency (η_D) predicted from the model experiments apply to the full-scale ship without any corrections for scale effects.

The correlation allowance, C_A, can then be computed as follows:

$$P_{S_s} \times P_{E_m} / P_{S_m} = P_{E_s} \qquad (A.1)$$

$$C_{T_s} = P_{E_s} \div \left((\rho_s/2)\ S_s\ V_w^{\ 3}\right) \qquad (A.2)$$

$$C_A = C_{T_s} - (C_{F_s} + C_{R_m}) \qquad (A.3)$$

where the subscript s is applied to full-scale trial values and the subscript m applied to values predicted from model experiments. V_w is the trial speed corrected to the zero relative wind conditions (see Chapter 5 for method of correcting wind). C_{F_s} is the frictional resistance coefficient from the I.T.T.C. frictional formulation at the Reynolds' number for the ship appropriate to the particular value of V_w.

It is also possible to correlate on the basis of thrust assuming that the residuary resistance coefficient (C_R) and the thrust deduction fraction (t) are the same for ship and model at corresponding Froude numbers.

$$T_s \times (1-t)_m = R_{T_s} \qquad (A.4)$$

$$C_{T_s} = R_{T_s} \div \left((\rho_s/2)\ S_s\ V_w^{\ 2}\right) \qquad (A.5)$$

$$C_A = C_{T_s} - (C_{F_s} + C_{R_m}) \qquad (A.6)$$

Typically the correlation allowances derived from shaft power and thrust do not agree. More often than not the thrust correlation allowance is lower than that from P_S.[35] A direct correlation is not made on RPM; a comparison is made usually at the correlation allowance derived from P_S.

Most towing tanks also compare propulsion coefficients from trials with predicted values from model experiments. These procedures vary substantially from one tank to another because of the problems of scaling between the model and full-scale hulls, propellers, and the interaction

35. Hadler, J. B., C. J. Wilson and A. L. Beal, "Ship Standardization Trial Performance and Correlation with Model Prediction," SNAME Vol. 70, 1962.

between hulls and propellers. It is stipulated, therefore, that the following assumptions and procedure are most probably in error to some degree. They do seem to give moderately reasonable answers for a variety of hull forms, presumably because the model size used at the Center is large enough to accommodate model propellers of reasonable diameter. The propeller Reynolds' number is correspondingly large.

Assuming that the model propeller open-water characteristics apply to the full-scale propeller, then J_{T_s} and J_{Q_s} can be derived from the open-water K_{T_m} and K_{Q_m} curves using the values of K_{T_s} and K_{Q_s} determined on trials. J_{V_s} can also be determined from trial data. Therefore the wake factors ($1-w_{T_s}$ and $1-w_{Q_s}$) and the relative rotative efficiency (η_{R_s}) can then be obtained from the trial coefficients and compared with predicted values.

A.3 Variations in Correlation Allowance

A.3.1 Structural Roughness

Unless ship construction methods are watched carefully to minimize obstruction to the flow, the hull of a ship can be far from a hydrodynamically smooth surface. It is possible to walk under some ships in drydock and see a substantial number of clips, padeyes, and other impedimenta which may make drydock operations more efficient, but certainly add a small increment to the total resistance. In any case there are roughnesses such as zincs, plate laps, and weld beads which are nearly unavoidable with the present standards of construction.

A.3.2 Anti-Fouling Paint Roughness

Paint roughness is probably the largest contributor to the correlation allowance where U. S. Navy ships are involved. For many years the Navy has conducted a considerable amount of research on protective coatings for hulls. There are the inevitable trade-offs between cost of materials and application, degree of smoothness, anti-fouling and anti-corrosion qualities. Prior to World War II the U. S. Navy used an anti-fouling paint designated 15 RC which was very smooth initially but fouled comparatively rapidly. During

World War II hot plastic anti-fouling paint came into use and proved to be superior in preventing fouling. This paint is applied hot, and in cold weather the paint surface can be extremely rough. There is also a cold plastic anti-fouling paint which is sprayed on without heating and is usually smoother than the hot plastic paint. The paint system which is normally used today is vinyl resin which is initially much smoother than hot plastic although it does foul somewhat more rapidly[35] (Figure A-1).

The correlation allowance must vary to compensate for the initial roughness of the paint system in question. Fouling is compensated for in the service margin. The current values of C_A used for the various paints are: hot plastic 0.0008 and vinyl resin 0.0005, when the I.T.T.C. friction formula is used. At the present time vinyl resin is used predominantly, so the usual correlation allowance specified for clean hull U. S. Navy ships is 0.0005.

It is interesting to note some experimental results from tests with the 20-foot friction plane at DTNSRDC. The plane itself is described in Reference 36. For the experiments with Navy paints the side panels of the plane were sprayed with several of the various Navy paint systems previously discussed. Three hot plastic paint systems were evaluated at the time. In addition the paint system used on models at DTNSRDC was sprayed on one set of panels. The plane was then run at various speeds up to as high a speed as permissible with Carriage 2 at DTNSRDC. The results are presented in Figure A-2. It is usual practice to qualify the results by stating that the results may not be typical of shipyard painting practice; that the friction plane is not shaped like a ship's hull, etc. It is interesting to note that in spite of these qualifications, the incremental resistance coefficient values for the various paints are very nearly the same as deduced from standardization trials with those paints.

36. Couch, R. B., "Preliminary Report of Friction Plane Resistance Tests of Anti-Fouling Ship Bottom Paints," DTMB Report 789, August 1951.

A.3.3 Length or Smoother Construction

There is an apparent decrease in correlation allowance with either length or chronology, or possibly a combination of both as the effects are difficult to separate. The more recent ships are smoother that those built prior to 1950, which is probably attributable to the increase of welding in ships, and in particular with the increase in use of butt welds. It may also be noted that ship length has increased in recent years, particularly commercial ships and Navy auxiliaries. It is quite possible that the decrease in correlation allowance noted is a combination of length, better shipbuilding practice, and the fact that excrescences are proportionately smaller on a large ship than a small one. A recent investigation was conducted to determine whether the correlation allowance for carriers should be significantly less than that for destroyers. One of the products of this investigation is Figure A-3[37] which illustrates the proposed variation in correlation allowance (C_A) with length. As discussed in a subsequent section, the empirical data supporting this type of correction are rather scanty. In spite of the last comment the trend seems to exist and the main question is what should be the recommended correlation allowance for a given length of ship. At this point it seems appropriate to point out the obvious; better shipbuilding practice which ends in a smoother hull will reduce the power required to propel the ship, and thusly the correlation allowance will be reduced.

A.3.4 Fouling

There is very little precise information available on the effects of fouling on the powering of ships and, consequently, the correlation allowance. The problem of fouling is extremely complex, depending on the geographic location of the ship, the season of the year, the type of paint, how recently the ship has been painted, and what the activity record of the ship has been, among other variables. Probably the best study of the

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37. Covich, P., "Variation in Correlation Allowance with Ship Size," Naval Ship Engineering Center Report 6136-74-20, December 1974.

effects of fouling on powering was conducted by DTNSRDC on four destroyers. Two of the hulls were painted with the hot plastic anti-fouling system and two with vinyl resin paint. Figure A-4 shows the results in unclassified form with regard to increase in correlation allowance. Figure A-5[38] shows the percentage increase in shaft power.

It should be noted that even this rather extensive series of trials does not begin to cover all of the many variables which affect fouling. It remains, however, the most complete set of data available to date. There are other experimental data available which are unclassified.[38] The usual problem with these data are that torque and RPM were measured but speed was not, so it is not possible to compute precisely the change in correlation allowance.

For the most part, with the exception of aircraft carriers, most U. S. Navy standardization trials are conducted with a clean hull less than 4 months out of dock. From Figure A-4 it may be seen that any corrections for this length of time out of dock is not really warranted. Aircraft carriers are a distinct problem because the docking and painting costs are quite large, thus it is rather rare for a carrier to have a clean hull when trials are run.

A.3.5 Friction Formula

As pointed out in a previous section the correlation allowance is dependent on the friction formula used since $C_R = C_{T_m} - C_{F_m}$. Obviously, if C_{F_m} changes due to friction formula, then C_R will change. Since $C_A = C_{T_s} - (C_R + C_{F_s})$, it is apparent that C_A is dependent on the friction formula.[5]

The A.T.T.C. (Schoenherr) Friction Line, first used at DTNSRDC in 1947, was derived from resistance experiments with flat plates by a number of experimenters as described on page 297 of Reference 4 (see Figure A-6). It was the successor to a number of frictional resistance formulae which were used as the state-of-the-art progressed starting with Froude. The

38. Stenson, R., "Hull Fouling," NSRDC Report 2509, July 1967.

A.T.T.C. line had the disadvantage that it was not steep enough in the low Reynolds' number range to suit the smaller towing tanks. They found they were getting negative correlation allowances. The formula currently in use at DTNSRDC and at many other towing tanks, for computing the frictional resistance coefficient is the I.T.T.C. 1957 Ship-Model Correlation Line.[1] It is identical within 4 significant figures to the previous formula, the A.T.T.C. friction line, at Reynolds' numbers above 1×10^9 but deviates significantly from the A.T.T.C. Friction Line in the range of Reynolds' numbers common to the smaller towing tanks using models 5 to 10 feet in length. The deviation in the 20- to 30-foot model range is small; about 0.00005 is an average value for a typical Navy ship as shown in Figure A-7. The deviation decreases as Reynolds' number increases, i.e., model length, speed or water temperature increases.

As the I.T.T.C. line is now standard for U. S. Navy use, the biggest concern is in making comparisons of performance of new ship designs with those where older friction formulae were used. The A.T.T.C. data are readily available for computation of the difference in C_F; data prior to 1947 were worked up using the Gebers friction formula.

A.3.6 Towing-Tank Practice

It is not advisable to assume that correlation allowance will be the same for different towing tanks. There are variations between the tanks which are by no means insignificant for a variety of reasons.

There are a number of methods of analysis used by the towing tanks throughout the world as described in the I.T.T.C. Proceedings. The smaller tanks use larger models than desirable to reduce scale effects. In doing so they encounter blockage effects which are compensated for in the analysis of resistance data by using one of several blockage correction methods. Another correction that is recommended by the I.T.T.C.[39] is the use of a transmission efficiency coefficient η_S, where $\eta_S = \frac{P_D}{P_S} = 0.98$ for

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39. Proceedings of the Ninth International Towing Tank Conference, Paris France, (1960).

ships with machinery aft and η_S = 0.97 for ships with machinery amidships. The transmission efficiency coefficient is necessary to compensate for the losses in bearings and seals from the point where the shaft torque is measured to the propeller; therefore, P_D is the horsepower delivered to the propeller. DTNSRDC recommends the use of a transmission efficiency of unity (1.0) because experimental data have been obtained by the U. S. Navy[40] which indicate that the percentage losses used by the I.T.T.C. are too high. Figure A-8 is a summary of pertinent data from Reference 40. In addition, it should be noted that it is standard Navy practice to place the torsionmeters as close to the stern tube bearing as possible rather than in engine rooms as foreign practice normally dictates.

A number of years ago the A.T.T.C. sponsored a standard model for resistance testing in the hope that the various towing tanks could conduct resistance experiments under identical conditions and exchange data.[41] The variations from the mean reported by the tanks involved were on the order of $\pm$ 2 percent. Thus even the basic experimental data can be so different that large corrections may be required to obtain full-scale correlation. In summary, do not mix correlation data or predictions from different tanks unless it has been determined previously that they are compatible.

A.4 The Influence of Scale Effects

There has been a considerable amount of effort expended by the I.T.T.C. Performance Committee in studying how scale effects affect the prediction of powering performance from experiments with models. This work is documented in the I.T.T.C. Proceedings each time they are issued. The major problems encountered in recent years have been with the very full-form tankers which are so prevalent today. As the U. S. Navy has not shown any particular interest so far in this type of hull form, most of the material is not really pertinent to this manual.

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40. Pitre, Lt. Cdr. A. S., "Propulsion Problems of a Destroyer," Experimental Model Basin Report No. 390, (Oct 1934).
41. Gertler, M. and C. H. Hancock, "Comparative Resistance Tests with the ATTC Standard Model," David Taylor Model Basin Report 1357, (Jul 1959).

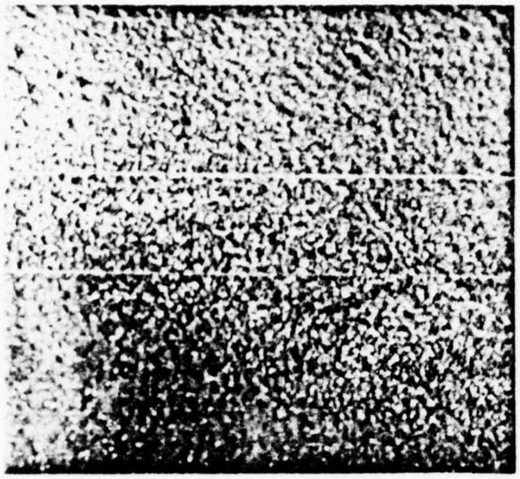
One item that should be noted is a correction to the predicted RPM which should be made after the self propulsion experiments are conducted at the desired correlation allowance. The reason why this correction is necessary is not clearly understood at this time. It is quite certain that the boundary layer around the model is not the same as that around the ship which undoubtedly affects the wake and the RPM.

It is usually the case that the RPM is over-predicted from the model experiments at DTNSRDC for rough hulls (high correlation allowance), while for very smooth hulls the correction is negligible.

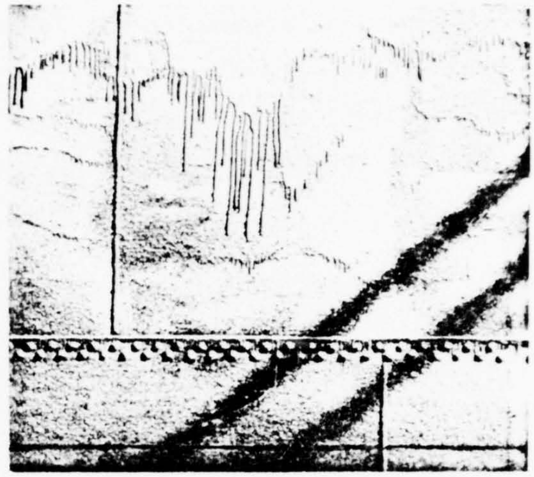
Prior to the compilation of these data, the relationship of ship RPM to model RPM was considered to be essentially a constant. For those correlations available at the time, a value of 0.98 was a good average. As more ships were added to the study, it became apparent that a constant value was not a particularly good approximation.

This ratio was plotted against a number of variables, including C_B and C_P. The only plot which seemed to be comparatively consistent was RPM_s/RPM_m against C_A. This plot is shown in Figure A-9. It may be seen that there is a definite tendency for the ratio to decrease as C_A increases. A least-square line has been drawn through the data points indicating a decrease in this ratio of approximately 3 percent in the range of C_A from 0 to 0.0010. Admittedly there is a large deviation from this line for a number of the ships tested. Attempts have been made to further isolate ships as to number of propellers and paints to see if any trends could be established. The least-square lines derived for each of these attempts have been invariably so close to the line drawn for all the data that the difference was negligible.

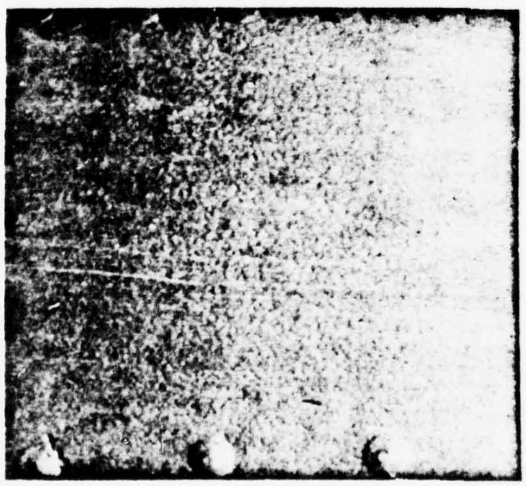
The hypothesis that this RPM relationship varies with correlation factor due to significant changes in wake with growth in boundary-layer thickness can be advanced. Therefore it would appear that there would be a difference in the relationship for single-screw ships as compared to multiple-screw ships. This is not the case. Further reasoning could be applied to rationalize why this anomaly exists, but it does not appear fruitful to do so. It appears, therefore, that the ratio RPM_s/RPM_m varies with C_A within certain limits, and the data must be viewed from an empirical standpoint without a rigorous explanation. Other than the correction for RPM, the state-of-the-art is such that other scale-effect corrections are considered to be so conjectural at this time that they are not recommended for U. S. Navy designs.



Good hot plastic surface



Hot plastic surface as typical of Ship 19



Good vinyl resin surface

Good cold plastic surface

Figure A-1 - Typical U.S. Navy Anti-Fouling Paints

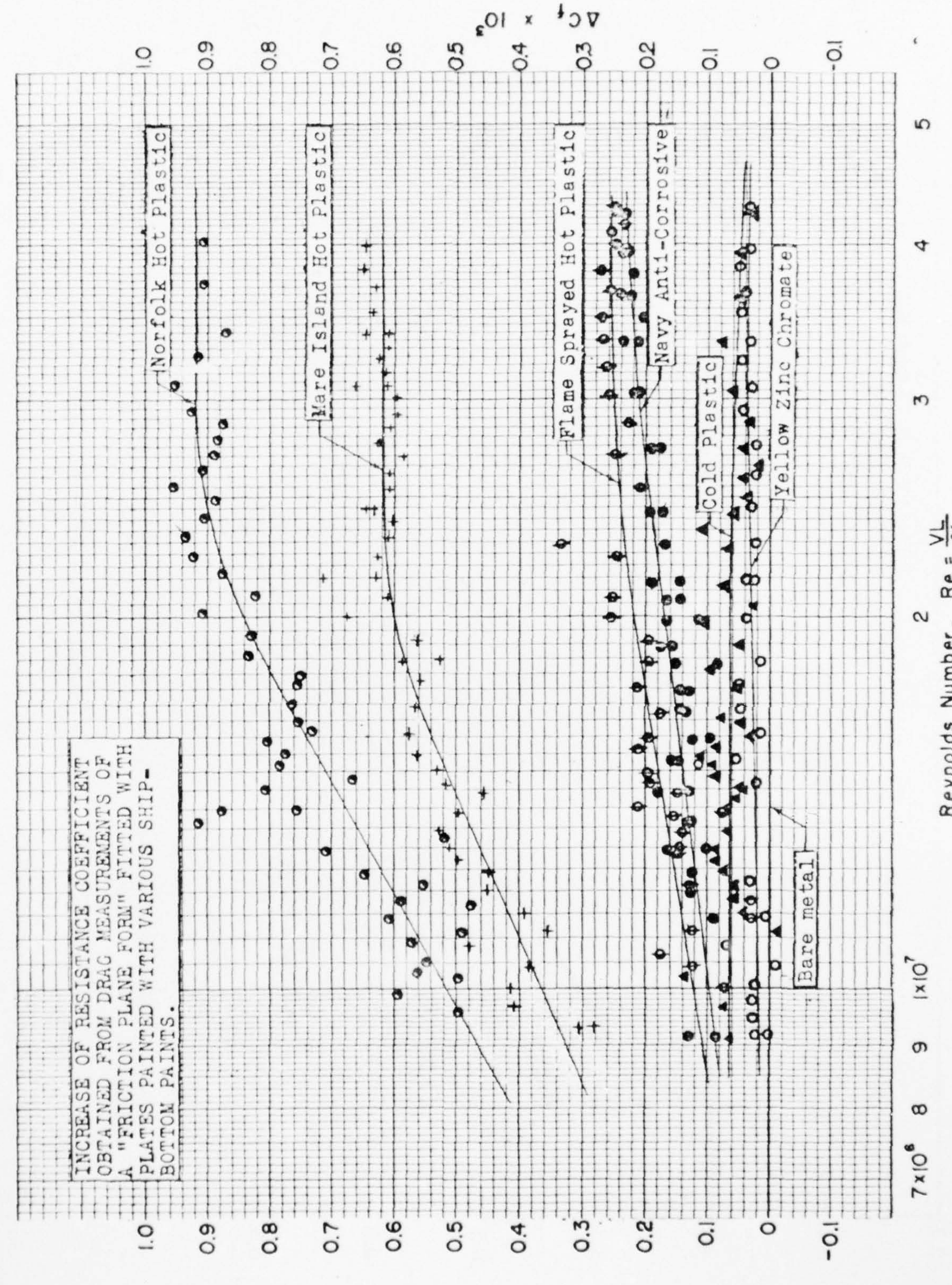


Figure A-2 - Friction Plane Data for Navy Paints

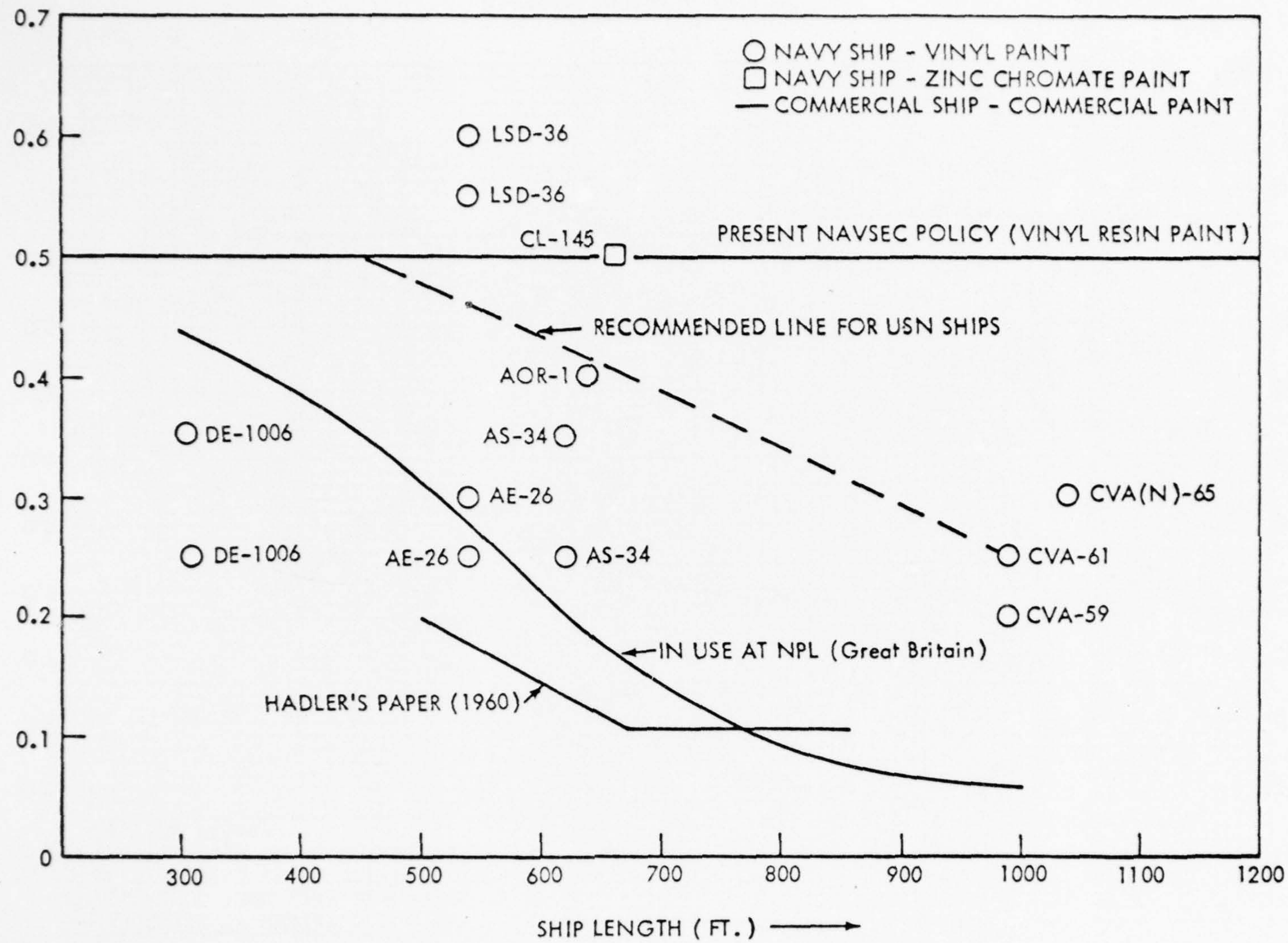


Figure A-3 - Correlation Allowance (C_A) for U. S. Navy and Commercial Ships as a Function of Ship Length

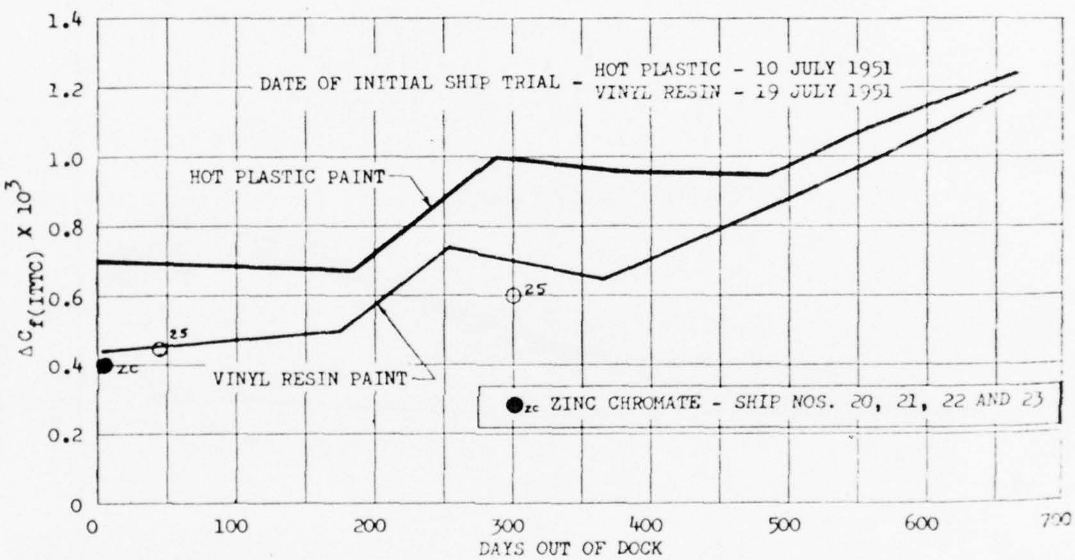


Figure A-4 - Effect of Correlation Allowance (C_A) of Time Out of Dock for Hot Plastic and Vinyl-Resin Paints from Correlations of Ship Nos. 20, 21, 22 and 23

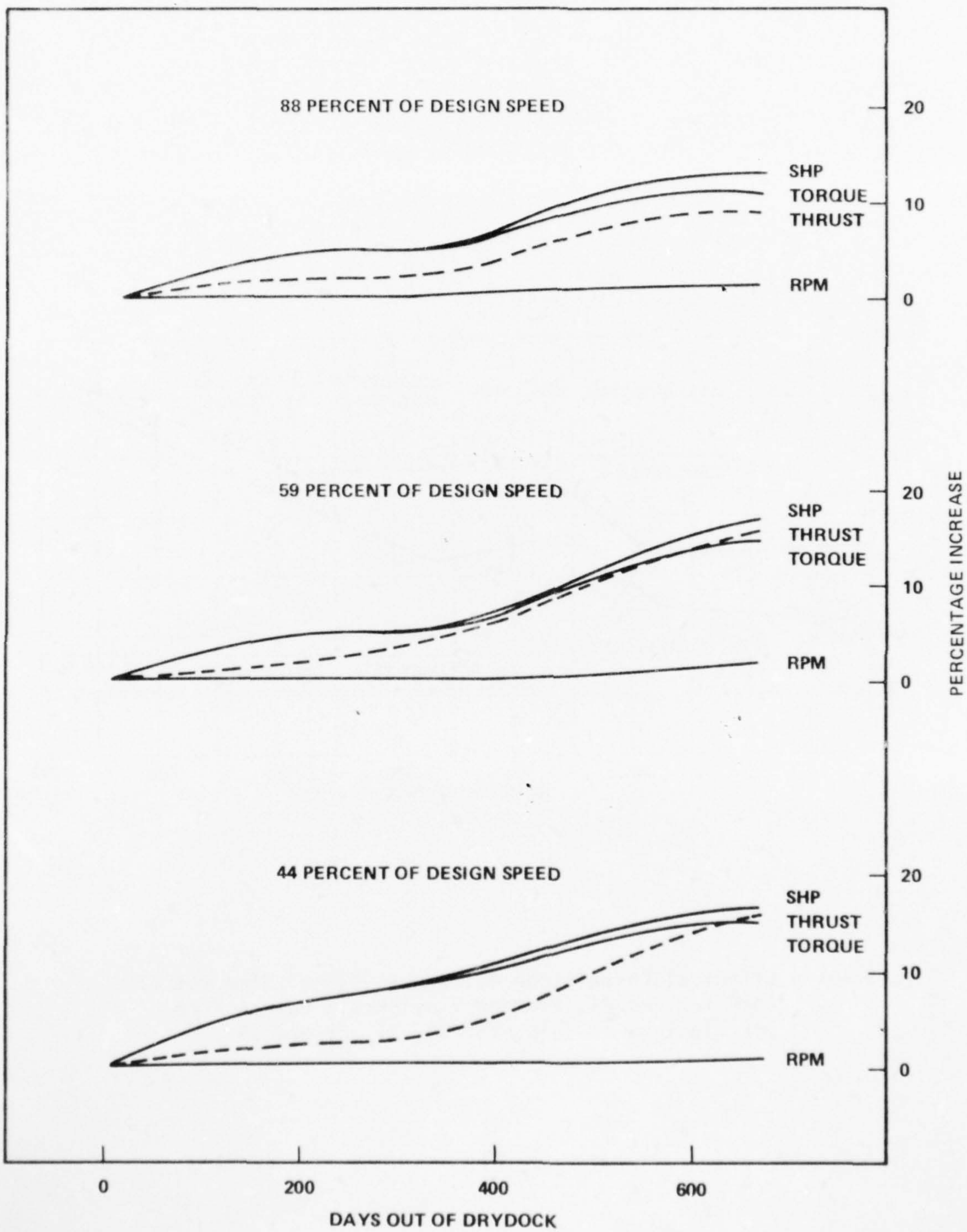


Figure A-5 - Increased Powering Requirements versus Time Out of Drydock for Destroyer Hull Painted with Vinyl Resin

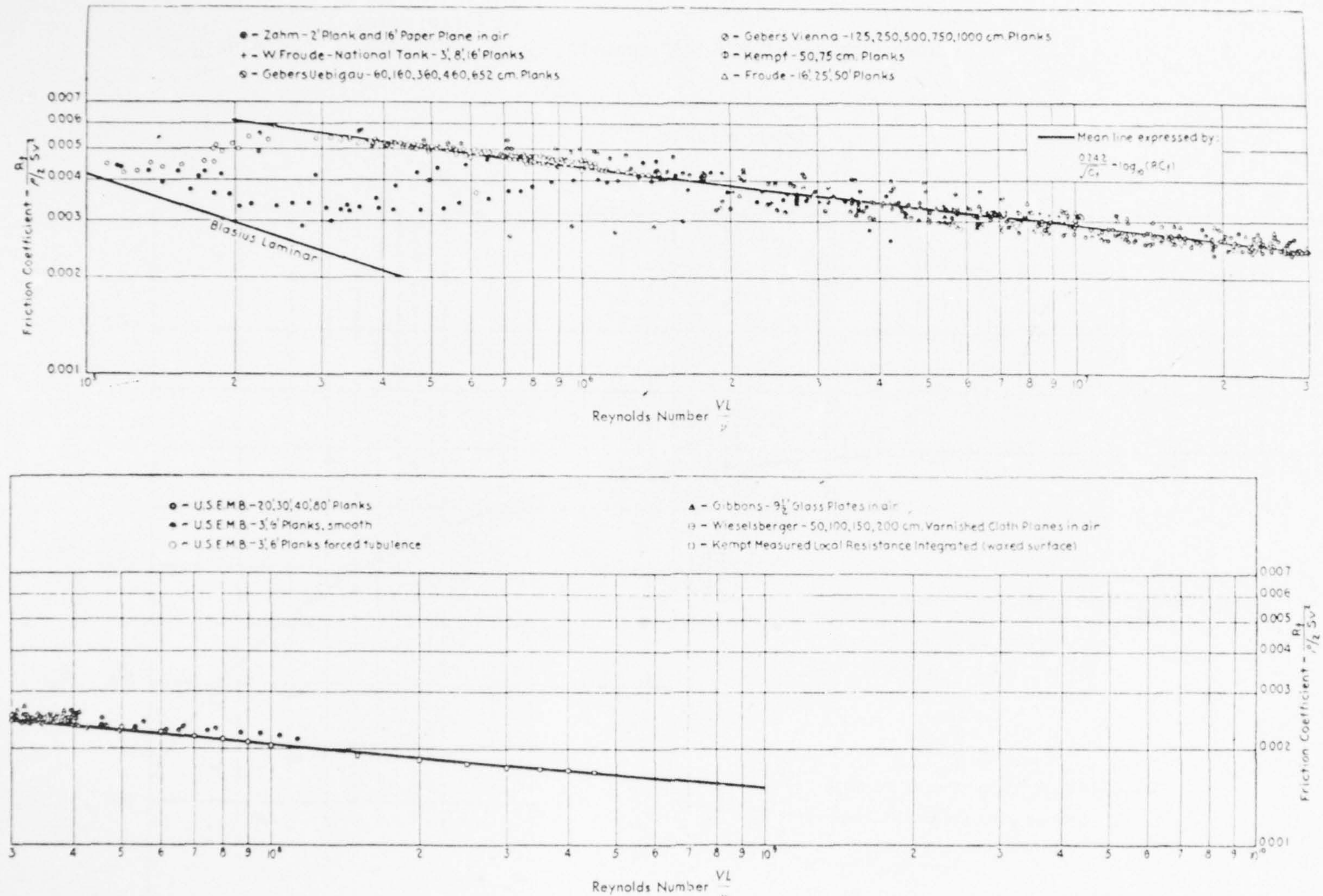


Figure A-6 - Schoenherr's Log-Log Chart for Friction Formulation

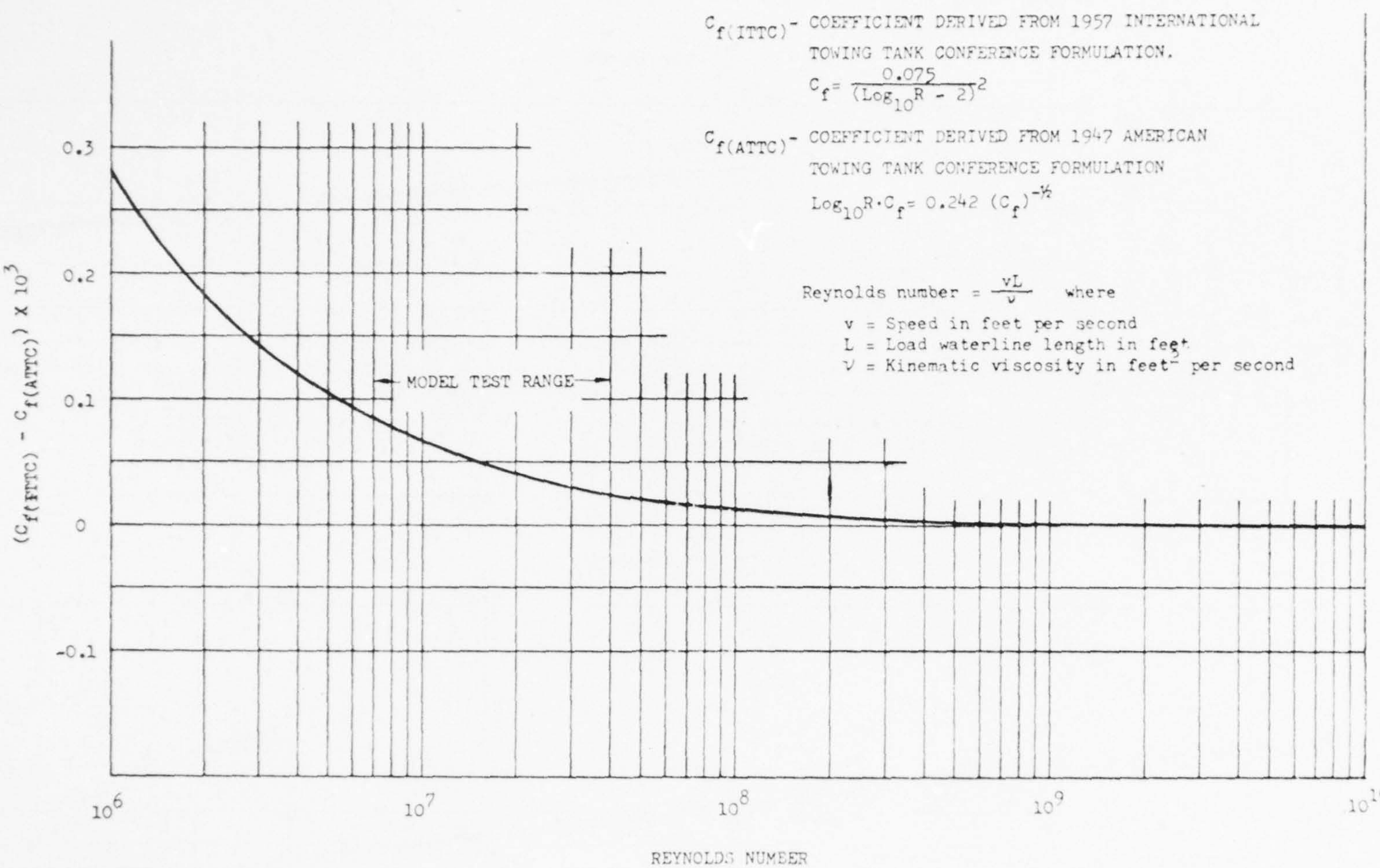


Figure A-7 - Comparison of Frictional Resistance Coefficient Formulations

USS HAMILTON. FRICTIONAL TRANSMISSION LOSSES FOR STARBOARD SHAFT ABAFT TORSIONMETER, EXPRESSED IN SHAFT HORSEPOWER

FROM MODEL TESTS AND CALCULATION

RPM	BEARINGS AND STUFFING BOX	DUMMY HUB AND CAP	TOTAL
100	5.47	0.23	5.70
200	5.55	1.40	6.95
300	9.23	4.02	13.25
400	15.14	9.12	24.26
500	22.62	17.07	39.69

FROM FULL SCALE SHIP TESTS

RPM	TOTAL
36	4.5
50	4.9
75	5.4
100	5.7
201	8.2
300	13.9
400	23.5
492	39.4

The total loss in power at 100 RPM is approximately 0.5 per cent of the delivered power, and at 400 RPM approximately 0.2 per cent of the delivered power.

Figure A-8 - Transmission Losses (Frictional) for a Typical Naval Ship

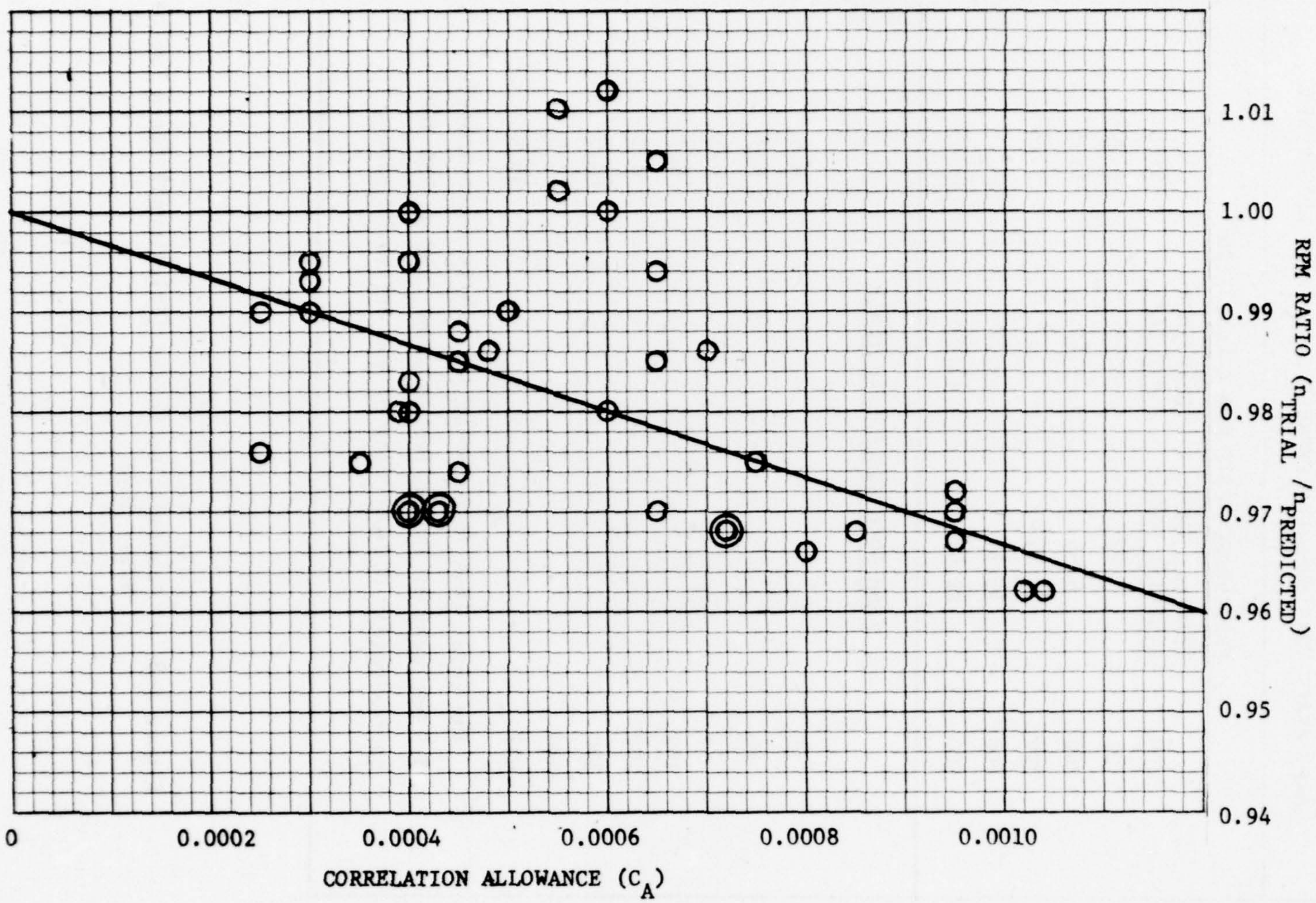


Figure A-9 Correlation of Revolutions per Minute (RPM) Versus Correlation Allowance (C_A)

APPENDIX B

THE PROPULSION COEFFICIENTS

B.1 General Discussion

The propulsion coefficients used at DTNSRDC and NAVSEC are the propulsive-efficiency (η_D), the thrust-deduction factor (1-t), the thrust-wake factor ($1-w_T$), the propeller efficiency (η_0), the hull efficiency (η_H), and the relative rotative efficiency (η_R). Three types of experiments are required to derive the above coefficients: the resistance experiment, the self-propulsion experiment and the open-water experiment. The propulsive efficiency and the thrust-deduction factor are derived from the resistance and propulsion experiments; the thrust-wake factor and the relative rotative efficiency are derived from the propulsion and open-water experiments; the propeller efficiency is derived from the open-water experiment; and the hull efficiency is derived from all three experiments.

During the past several decades a rather large assortment of notation has been used to describe the propulsion coefficients. This has certainly been an impediment to the understanding and usage of these coefficients. Fortunately, although different names and/or symbols have been used, the definitions of the coefficients have remained the same. An effort has been made in this document to standardize the symbols used to describe the powering performance of ships. Wherever possible, the symbols and definitions cited in Reference 20 have been used. Since many readers may not be familiar with the other notation used in the past, a brief discussion of each coefficient is presented:

(1) propulsive efficiency (η_D)
other names: quasi-propulsive coefficient
other symbols: P_E/P_S, EHP/SHP, P.C., η_S

(2) relative rotative efficiency (η_R)
other names: (none)
other symbols: e_{rr}

(3) open-water propeller efficiency (η_0)
other names: (none)
other symbols: e_p, e_o, η_P

(4) shaft transmission efficiency (η_S)

not formerly used - η_S assumed to be 1.0 for all naval ships (see Chapter 7.0, Section 7.1)

(5) hull efficiency (η_H)

other names: (none)

other symbols: e_h

(6) thrust-deduction factor (1-t), and thrust-deduction fraction (t)

other names: (general) thrust deduction may have been used to describe either the fraction or the factor.

other symbols: (none)

(7) thrust-wake factor ($1-w_T$), and thrust-wake fraction (w_T)

w' is defined as the Taylor wake fraction, in general

w_T is defined as the Taylor wake fraction determined from thrust identity; hence, the name thrust-wake fraction

other names: effective wake fraction
wake fraction
wake
thrust wake

other symbols: (none)

B.2 The Propulsive Efficiency, Components of

The propulsive efficiency (η_D) is derived from the resistance and self-propulsion experiments. In its simplest form

$$\eta_D = \frac{RV}{2\pi\ Qn} = \frac{P_E}{P_D}, \tag{B.1}$$

where R is the resistance at a given speed (V), Q is the torque, and n is the rotational speed from the propulsion experiment.

There are several popular misconceptions about propulsive efficiency. One is that if a change is made in a design which results in a higher propulsive efficiency, the change must be beneficial with regard to powering performance. It may be noted that if P_E increases during the change, and P_D remains constant, η_D will increase. Yet a decrease in P_D is what the designer really wants. The U. S. Navy does not buy P_E to put in its ships; it buys P_D.

Another popular misconception is that a re-fairing of open-water propeller efficiency (η_0) must affect propulsive efficiency. This idea stems from the usual breakdown of η_D into components:

$$\eta_D = \eta_H \, \eta_0 \, \eta_R \, \eta_S \quad \text{(See Equation 7.1 and the discussion; assumption: } \eta_S = 1.0\text{)}$$

$$= \frac{R\ V}{T\ V_A} \times \frac{T\ V_A}{2\pi\ Q_o n} \times \frac{Q_o}{Q} \tag{B.2}$$

where the quantities with the zero subscript denote the values measured when the propeller is advancing through undisturbed water and those without the subscript are the corresponding values measured when the propeller is driving the model. It should be noted that V_A is the speed of advance determined from thrust identity.

As pointed out previously, the propulsive efficiency can be determined from the combination of the resistance and propulsion experiments. Where a breakdown of η_D into its components is desired, the addition of the data from the open-water propeller experiment is needed. If the open-water curves were refaired the changed values would affect not just propeller efficiency, but the hull and relative rotative efficiencies as well. The value of η_D would not be affected unless the resistance and/or propulsion data were altered. It is agreed that in the early stages of preliminary design, where stock propeller tests are not available, that if η_0 only is changed obviously η_D must change.

With the accumulation of a considerable amount of experience on tankery methods at DTNSRDC it has been found that, in cross fairing the propulsion coefficients from the experimental data, the propulsive efficiency should be faired first as the results are usually less scattered than thrust-deduction and thrust-wake factors. Thrust deduction, in particular tends to be rather erratic as it is the ratio of two variables whose magnitudes are usually very similar; R/T for a destroyer might be 0.98, for example. If the measured thrust is one percent higher and the measured resistance one percent lower, the shift in thrust deduction would be very nearly two full percentage points. As discussed previously wake is derived from the open-water and self-propulsion experiments. A shift in thrust of one percent higher during the open-water experiments and one percent lower in the propulsion experiments will change wake by about one percentage point. The same type of problem is encountered in the sensitivity of relative rotative efficiency to comparatively small changes in the measurements. On this point of accuracy of measurements it should be noted that a change of one percentage point in the propulsion coefficients may not be significant.

Typical values of η_D for several types of Navy ships are as follows:

DESTROYERS (Twin-Screw)	0.62 - 0.66
DESTROYER ESCORTS (Single-Screw)	0.61 - 0.65
CARRIERS (Quadruple-Screw)	0.62 - 0.66

As stated by Johnson and Gale in Reference 21, the problems with predictions using η_D are those involved with w_T, t, and η_R, which will be discussed individually.

B.2.1 Thrust-Wake Fraction (w_T)

The thrust-wake fraction is the velocity defect in way of the propeller deduced from the experimental data from an open-water test and a propulsion test with the same propeller. It is determined by computing K_T and J_V from the propulsion test, and by using the open-water curve at the experimental value of K_T to get a value of J_T. The thrust-wake fraction is then determined from

$$w_T = 1 - (J_T/J_V) . \qquad (B.3)$$

It has been the practice for many years to separate wake into frictional, potential and wave components. The frictional component is greatest at the surface and in the center of the stern and decreases downward and outward. The potential component is due to the velocity of the water closing in around the stern. It will also be greatest at the center and the surface of the water and decrease downward and outward. The component of wake due to wavemaking will be greatest when there is a wave crest at the stern and least when there is a wave hollow at the stern. For further details see Section 149 of Reference 42.

Wake is affected by a rather sizeable number of variables such as: hull shape (particularly just forward of the propeller location), propeller geometry (including diameter, pitch, rake, loading, tip clearance between hull and propeller, distance of propeller tips below the surface of the water, size, shape and location of appendages with respect to the propeller), and roughness of the hull surface.

In fairing wake data from a propulsion test, it is normal to find that there is a good deal of scatter at the very low speeds; this data should not be given much weight. Wake does not vary much with speed for fine hulls until a $V/\sqrt{L}$ of 1.0 is reached at which point w_T, the wake fraction, normally decreases, and then levels off again when $V/\sqrt{L} = 1.2$ is exceeded. Since the design speed of most naval auxiliary hulls will not exceed $V/\sqrt{L} = 1.0$, w_T will be very nearly constant through the speed range tested. For very full forms ($C_B \geq 0.75$) a variation above the $V/\sqrt{L} = 0.85$ hump is usually found.

B.2.2 Thrust-Deduction Fraction (t)

Thrust-deduction fraction (t) can also be separated into frictional, potential and wave components. Thrust deduction is due to the action of the propeller on the hull in that a suction (decrease in pressure) is created by the propeller. Even the term thrust deduction is controversial. The British have used the term "resistance augmentation" since it can be viewed as an addition to resistance instead of a deduction from thrust. One of the sayings of naval architecture is that the entire hull generates the wake but the thrust deduction is primarily affected by the hull and appendages within 1 to 2 propeller diameters of the propeller. It is unfortunate that most changes in hull form and propeller location that

42. Taylor, D. W., "The Speed and Power of Ships," United States Maritime Commission, 1943.

affect wake fraction beneficially (higher) also affect thrust deduction fraction in the same direction such that hull efficiency, $\frac{1-t}{1-w_T}$, does not change a great deal.

The variables which affect thrust deduction are the same, generally speaking, as those listed for wake. In addition, the size and shape of the rudder and its proximity to the propeller may have a decided effect on thrust deduction. Usually wake is expected to change more than thrust deduction when displacement is decreased a significant amount, e.g., from full load to ballast condition. Consequently hull efficiency is normally higher in the ballast condition. For single-screw ships an increase in propeller diameter normally increases the thrust-deduction fraction. Since the thrust-wake fraction normally decreases, hull efficiency will be expected to decrease in this case. With twin screws both wake (w_T) and thrust deduction (t) decrease if the distance of the propellers from the hull is increased.

Thrust deduction fraction is much more sensitive to variations in test data than wake as it is the small difference between two large variables $\frac{T-R}{T}$. To repeat a point made previously, if a thrust measurement was one percent high and the resistance measurement one percent low for a given data set the variation in t would be about two percentage points. A fair scatter of data is normally encountered at the lower speeds. Again, as for wake, the lower speed points should be given little weight in fairing the curve. Unless there is an unmistakable trend below $V/\sqrt{L} = 0.6$, it is usually sufficient to consider t as a constant.

Amost all combatant ships and a number of the auxiliaries in the U. S. Navy are built with exposed shaft(s) and employ shaft struts to keep vibration at acceptable levels. The shaft and struts in front of the propeller are major contributors to the wake and to the thrust deduction to a lesser degree. The use of the controllable and reversible pitch propeller in recent designs has been a problem as far as thrust deduction is concerned because the method to date of controlling pitch requires a comparatively large shaft to contain part of the pitch control mechanism. Consequently, the struts, bearings, etc., will be more massive because the size and weight of the shaft have increased. It is anticipated, therefore, that the use of a controllable pitch propeller over a fixed pitch propeller will generally result in an increase in thrust deduction even though the increase may not be large. It should be pointed out that the effect on P_D from the change in appendage resistance will usually be much larger than the effect from the change in thrust deduction.

B.2.3 Relative Rotative Efficiency (η_R)

The relative rotative efficiency (η_R) for a propulsor/hull combination is the ratio of the propeller's efficiency behind the hull to its efficiency in open water. Essentially it is a measure of how well-suited the propulsor is to the velocity field in which it must operate.

As discussed in Section B.2, the open-water efficiency is defined by the expression

$$\eta_0 = \frac{T\,V_A}{2\pi Q_o n}, \tag{B.4}$$

where all the measured quantities are determined in a uniform inflow velocity (V_A). Since the current practice at DTNSRDC does not dictate that the self-propulsion and open-water experiments be conducted for precisely the same range of speed and revolutions per minute, the open-water data are generally expressed in coefficient form, as follows,

$$\eta_0 = \frac{J_o\,K_{T_o}}{2\pi K_{Q_o}}, \tag{B.5}$$

where the subscript "o" means "determined in open-water".

The efficiency of a propeller behind the hull (η_B) is generally expressed, in a manner similar to Equation B.4, as follows,

$$\eta_B = \frac{T\,V_A}{2\pi Q n}, \tag{B.6}$$

where the torque is the only factor which makes η_B differ from η_0. The thrust and torque, delivered and absorbed, respectively, during the self-propulsion experiment may be expressed as the thrust and torque coefficients (K_T and K_Q), respectively. One would then determine the advance coefficient

(J_T) which would correspond to the thrust coefficient (K_T), using the open-water curve of J_o versus K_{T_o}. Then, one could determine the K_{Q_o} that corresponds to J_T in the open-water curve of K_{Q_o} versus J_o. Considering the foregoing, it is apparent that the expression for relative rotative efficiency

$$\eta_R = \eta_B/\eta_O, \tag{B.7}$$

can be reduced to

$$\eta_R = K_{Q_o}/K_Q. \tag{B.8}$$

There are essentially two factors which affect the relative rotative efficiency (η_R), each of which alter the ability of the propeller to deliver a particular thrust with a particular torque as input at the same speed and revolutions per minute. One of these is the amount of homogeneity in the inflow into the propeller. Not only is the axial component of the wake non-uniform, but tangential components also exist behind a ship. The other factor is the increased degree of turbulence in the fluid. Both of these items materially affect the relative rotative efficiency.

The relative rotative efficiency generally has a value near unity. It is between 0.95 and 1.0 for most single-and twin-screw ships with shafts and struts. There are some data to indicate that the struts are the primary factor tending to reduce η_R. A value in excess of 1.0, and sometimes as great as 1.10, is common for single-screw merchant ships.

B.3 The Interaction Coefficients, Estimation of

The best method for estimating the thrust-wake fraction, the thrust-deduction fraction, and the relative rotative efficiency is to conduct resistance, propulsion, and open-water experiments, and to analyze the data as discussed in Section B.2. This is not, however, always practical, and the naval architect must often resort to another estimating technique.

B.3.1 Single-Screw Cargo Hull Forms

Several techniques have been developed for estimating the thrust-wake fraction and thrust-deduction fraction of single-screw cargo ships. In 1950, Harvald[43] discussed such techniques in a comprehensive study of wake fraction and thrust deduction. He evaluated 21 different methods used for conventional single-screw cargo ships, and determined that the methods of Taylor and Schoenherr were the best available at that time. He then presented his own method, and concluded that his method was as accurate as the Schoenherr method, but easier to use. His final recommendation was to use the Taylor method if a simple technique was satisfactory, and to use his own method if more accuracy was needed.

Harvald also studied the thrust-deduction fraction, and presented his own estimation method for conventional single-screw cargo ships. The Harvald diagrams are shown in Figures B-1 and B-2, and the Taylor estimation equations are

$$w_T = -0.05 + 0.50\, C_B, \text{ and} \tag{B.9}$$

$$t = (1.1)(-0.20 + 0.55\, C_B),^{*} \tag{B.10}$$

where C_B is the block coefficient.

<u>Sample Estimates</u>

The Taylor formulae (Equations B.9 and B.10) and the Harvald diagrams (Figures B-1 and B-2) were used to estimate the thrust-deduction fraction and thrust-wake fraction for 150 models of single-screw cargo ships tested at DTMB and DTNSRDC, and the results are shown in Figures B-3 and B-4. The scatter on all the diagrams reveals the poor quality of the methods. To further illustrate this poor quality, Table B-1 was prepared. The average error and standard deviation of the errors were calculated for the Taylor and Harvald methods, and also for the simple method of assuming that every

43. Harvald, S.A., "Wake of Merchant Ships," Danish Technical Press, Copenhagen, 1950.

* The factor "1.1" is an estimate based on a qualitative statement.

value was equal to the mean value determined from the 150 model studies. In the case of thrust-deduction fraction, choosing the mean value was better than using either of the published prediction methods, while for thrust-wake fraction the mean value was better than the complex Harvald method, but not as good as the simple Taylor equation.

B.3.2 Multi-Screw Military Hull Forms

Unfortunately, similar techniques do not exist for the estimation of thrust-wake fraction and thrust-deduction fraction for multi-screw military hull forms. Schoenherr[5], however, gives a formula for twin-screw propellers supported by struts ($w = 2\, C_B^{\,5}\,(1-C_B) + 0.04$), but states that this is for merchant ships of normal form operating at speed-length ratios below unity. He states that the wake fraction for destroyer forms lies between -0.02 and +0.02 when the ship is equipped with struts.

B.3.3 Other Useful Techniques

Since the aforementioned techniques are frequently inadequate, naval architects must rely on other methods. Although there is a lot of valuable information in some of the series data such as Series 60,[9] it is seldom useful in the design of naval ships. Consequently, the naval architect is forced to do some research each time he wishes to estimate the thrust-wake fraction, the thrust-deduction fraction, and the relative rotative efficiency for a ship.

This research will consist of collecting and analyzing sets of data for similar ships. If the new design happens to be a direct offspring of a particular ship, e.g., in its hull form and in its complement of appendages, it would be appropriate to use the data from that hull form alone. In most cases, however, it is necessary to review several sets of data, and to interpolate between two or more of the relevant sets of data. Although this method can provide accurate results, one may misjudge the evidence and, consequently, develop poor estimates.

Since this analytical process does not lend itself to precise step-by-step instructions, some basic guidelines are presented here to help prevent these types of errors. It is rather important that the naval architect know the precise ship geometry and the nature of the appendages of the hull for which he has data. In addition to the importance of the principal hull coefficients, there are other items, such as the propeller tip clearance, the location of the propeller certerline relative to the ship's baseline, the size of the propeller hub compared to its diameter, the size of the rudder(s), the propeller-to-rudder clearance, and other factors that significantly affect the thrust-wake fraction, the thrust-deduction fraction and the relative rotative efficiency. Unfortunately, however, the state-of-the-art has not provided any guidelines as to how these factors might rank in their level of importance. All that can be said is that the sets of data should be chosen with these factors in mind.

As an aid to the naval architect, Figure B-5 and B-6 and Tables B-2 and B-3 have been prepared which show the value of thrust-wake fraction and thrust deduction fraction determined from a selection of DTMB and DTNSRDC model tests. Mean, plus-or-minus one standard deviation from the mean, and maximum and minimum values are given for 10 classes of ships. A quick estimate for a new ship can be made by choosing the mean value of all previous ships in that class. If a more accurate figure is desired, the value from the ship in the data base which most resembles the new design can be used. If several ships in the data base closely resemble the new design, then average values from those ships could be used. Until new prediction methods are developed, this seems to be the best procedure to follow to estimate t and w_T in the early stages of design. Since there are so many parameters involved and their relative influences are ill defined, it is unlikely that the estimates will be highly accurate for all new ships.

B.4 The Interaction Coefficients, Scale Effects

The thrust-wake fraction is different for model and full-scale due in part to the difference in Reynolds' number and hull roughness. Due to the increase in Reynolds' number, the boundary layer (compared to ship length) should be thinner for the full-scale ship, so the propeller should operate in less of a wake, thereby reducing the value of w_T. However, the roughness substantially affects the boundary layer, and these roughness effects are the subject of a continuing international controversy. Thus, nothing definite can be stated about these effects at this time. For a "smooth" full-scale ship, current DTNSRDC practice is to use the model value. For a rough ship hull, the value is generally increased.

Effective wake can be computed from full-scale trial results in conjunction with the model propeller open-water experimental data. There is still a sizeable controversy with regard to this procedure since the magnitudes of the scale effects on both the model-propeller data and the interaction between model hulls and propeller are still very much in question. In its simplest form, where scale effects are ignored,

$$K_{Q_s} = \frac{Q_s}{\rho_s n^2 D^5} .$$

Entering the model open water curves at specific values of K_{Q_s} it is possible to determine values of J_{Q_s}, and to determine the torque-wake factor

$$1-w_{Q_s} = \frac{J_{Q_s}}{J_{V_s}}, \quad \text{where } J_{V_s} = \frac{101.27\ V_s}{n\ D}.$$

If thrustmeters have been installed in the ship, w_{T_s} can be computed.

$$K_{T_s} = \frac{3600\ T_s}{\rho n^2\ D^4}$$

J_{T_s} is determined from the model open-water (K_T versus J) curve.

$$\text{Then } 1-w_{T_s} = \frac{J_{T_s}}{J_{V_s}}$$

Relative rotative efficiency for the trials can also be computed if both torque and thrust values are available, since

$$\eta_{R_s} = \frac{K_{Q_{open}}}{K_{Q_s}} ,$$

where $K_{Q_{open}}$ is determined from the model open water curve of K_Q at the value of J_{T_s} as previously computed.

Please note again that the use of the model open-water curves in conjunction with trial values of speed, torque, thrust and RPM to determine "full-scale" propulsion coefficients is highly controversial.

B.5 Summary of the State-of-the-Art

It would appear that the experimental method is the only accurate way to estimate the propulsive coefficients for most naval ships. Reasonably accurate estimates can be made, however, for some naval auxiliaries using Series 60[9] and for those ships which are a direct offspring from another known ship. The common deficiency of all the existing techniques is that they fail to account for the effects of all the significant variables.

Since the thrust-wake fraction and thrust-deduction fraction are dependent on so many factors, it is extremely difficult to determine which are truly significant. Figures B-7 through B-11 are included for twin-screw destroyer forms to demonstrate the variations with some of the factors which are presumed to have some influence on these fractions. Each point represents the fractions at the design speed for one ship model of a twin-screw destroyer hull form. Some of the trends are obvious (e.g., with $V/\sqrt{L}$), where others are not. Certainly some of the trends may be masked by the effects of some of the other factors. It would appear that the significance of these parameters could be determined through regression analysis techniques. It is also likely that such a procedure would lead to a useful estimating method.

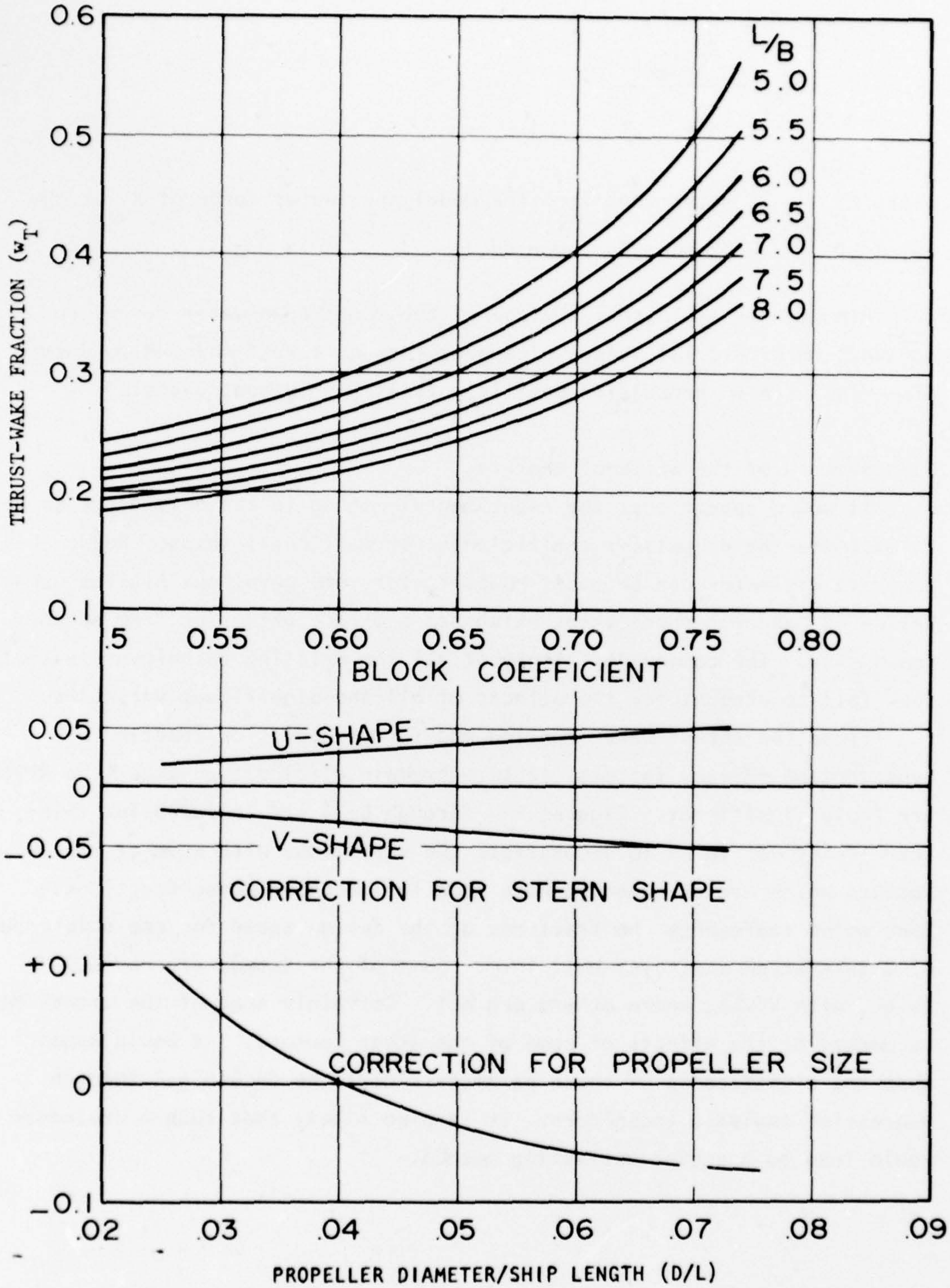


Figure B-1 - Harvald Diagram for Prediction of Thrust-Wake Fraction (w_T) for Conventional Single-Screw Cargo Ships

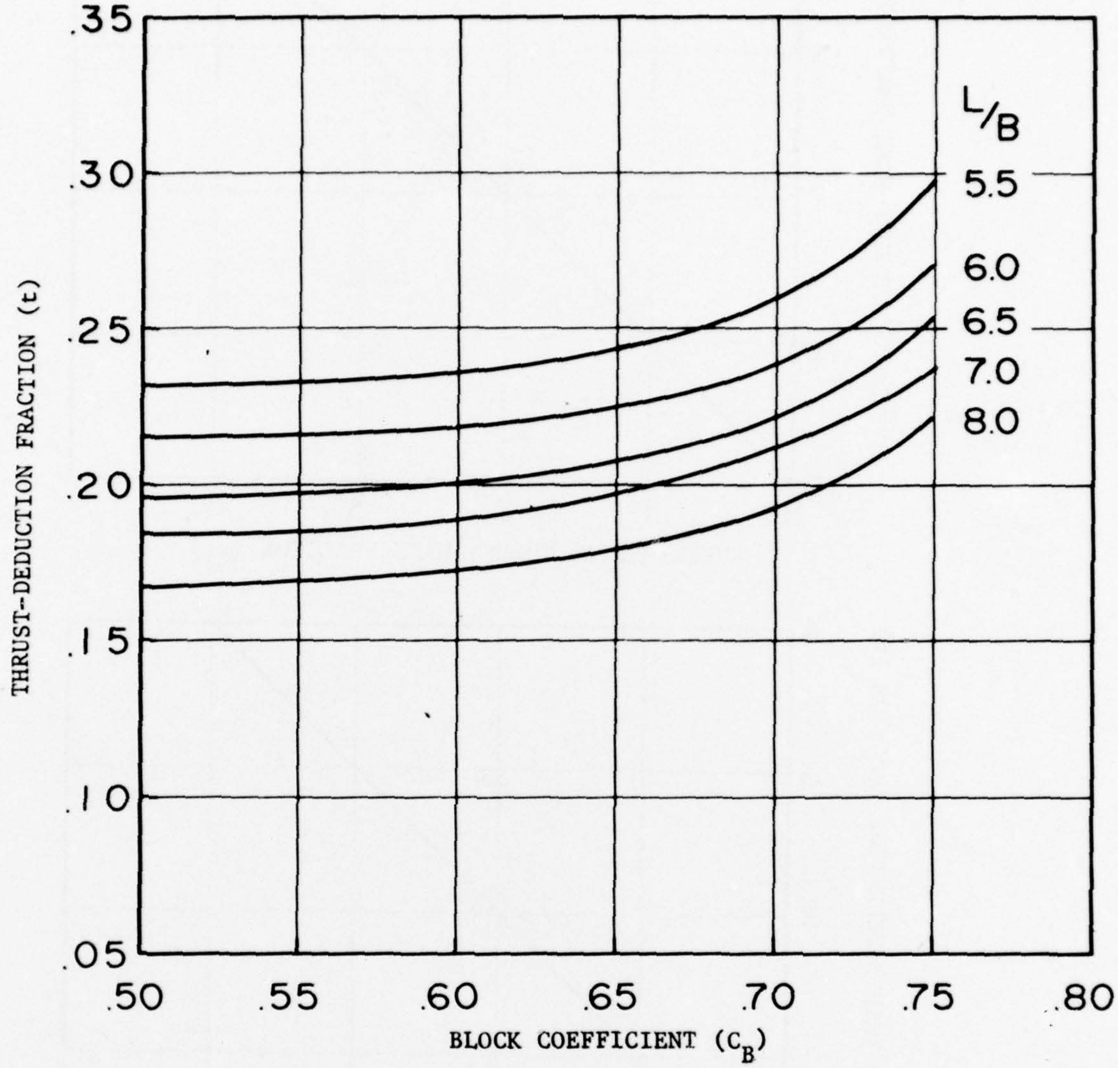


Figure B-2 - Harvald Diagram for Prediction of Thrust-Deduction Fraction (t) for Conventional Single-Screw Cargo Ships

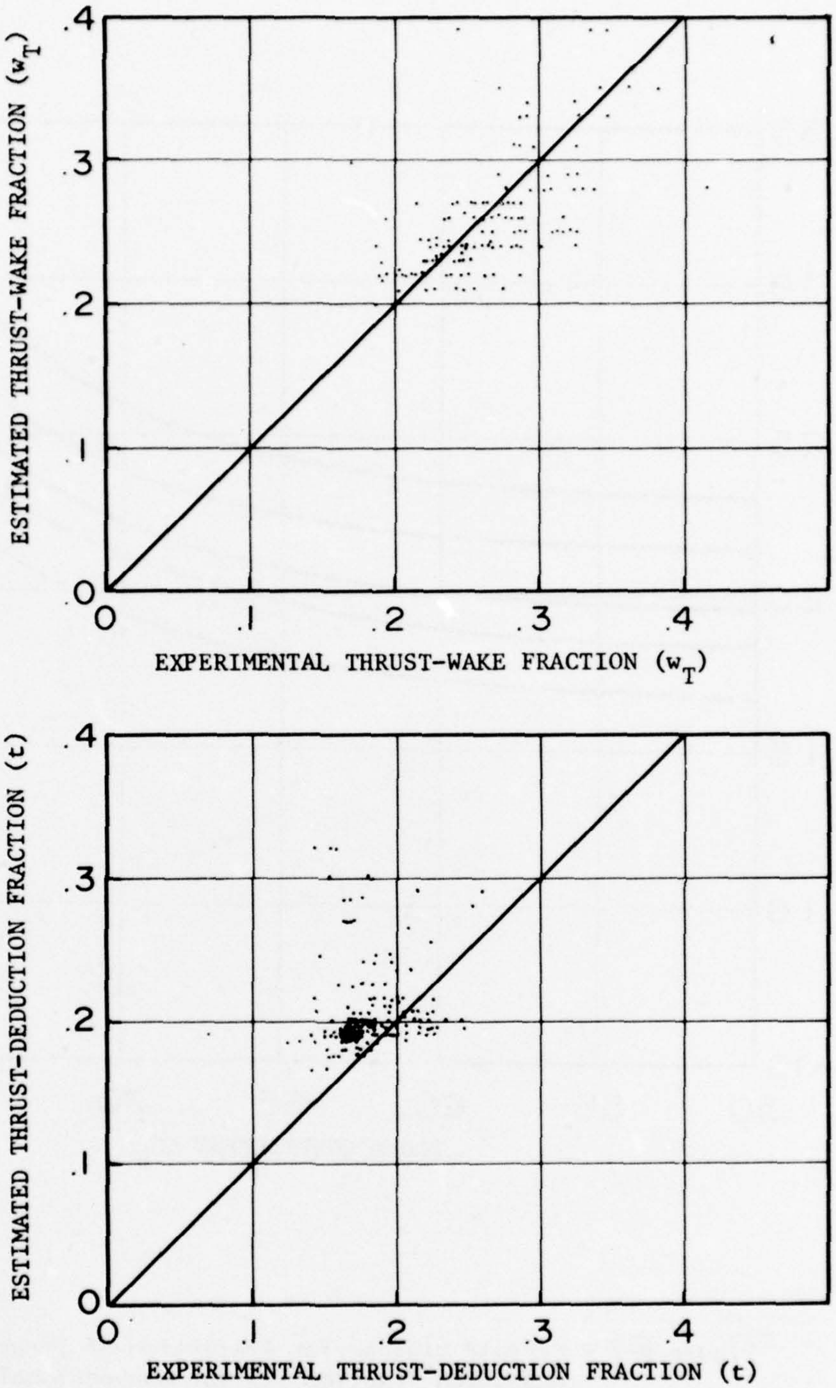


Figure B-3 - Comparison of Experimentally Determined Thrust-Wake Fraction (w_T) and Thrust-Deduction Fraction (t) with Empirical Estimates (Harvald Diagrams: not corrected for stern shape or propeller size) for 150 Conventional Single-Screw Cargo Forms.

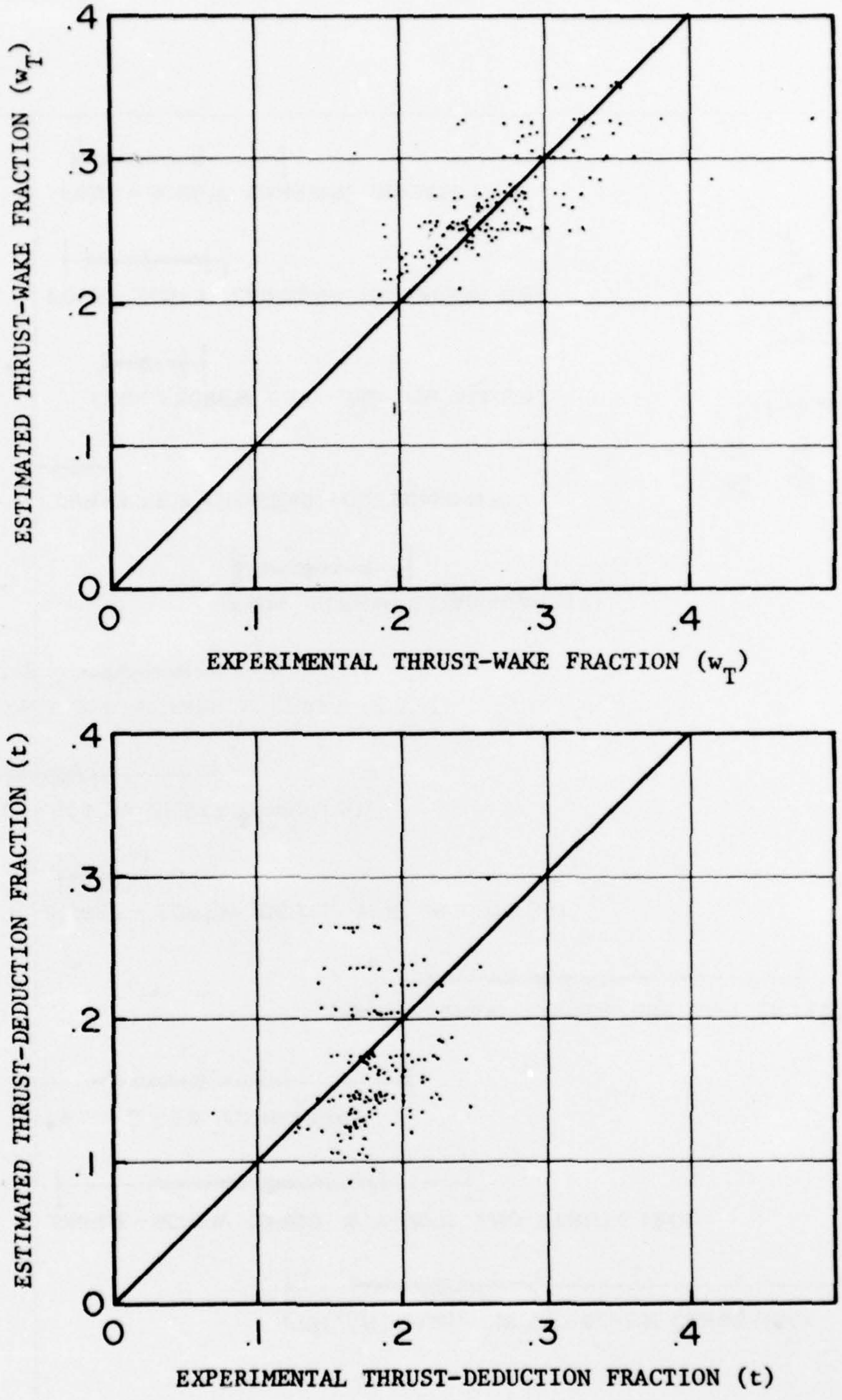


Figure B-4 - Comparison of Experimentally Determined Thrust-Wake Fraction (w_T) and Thrust-Deduction Fraction (t) with Empirical Estimates (Taylor Equations) for 150 Conventional Single-Screw Cargo Forms.

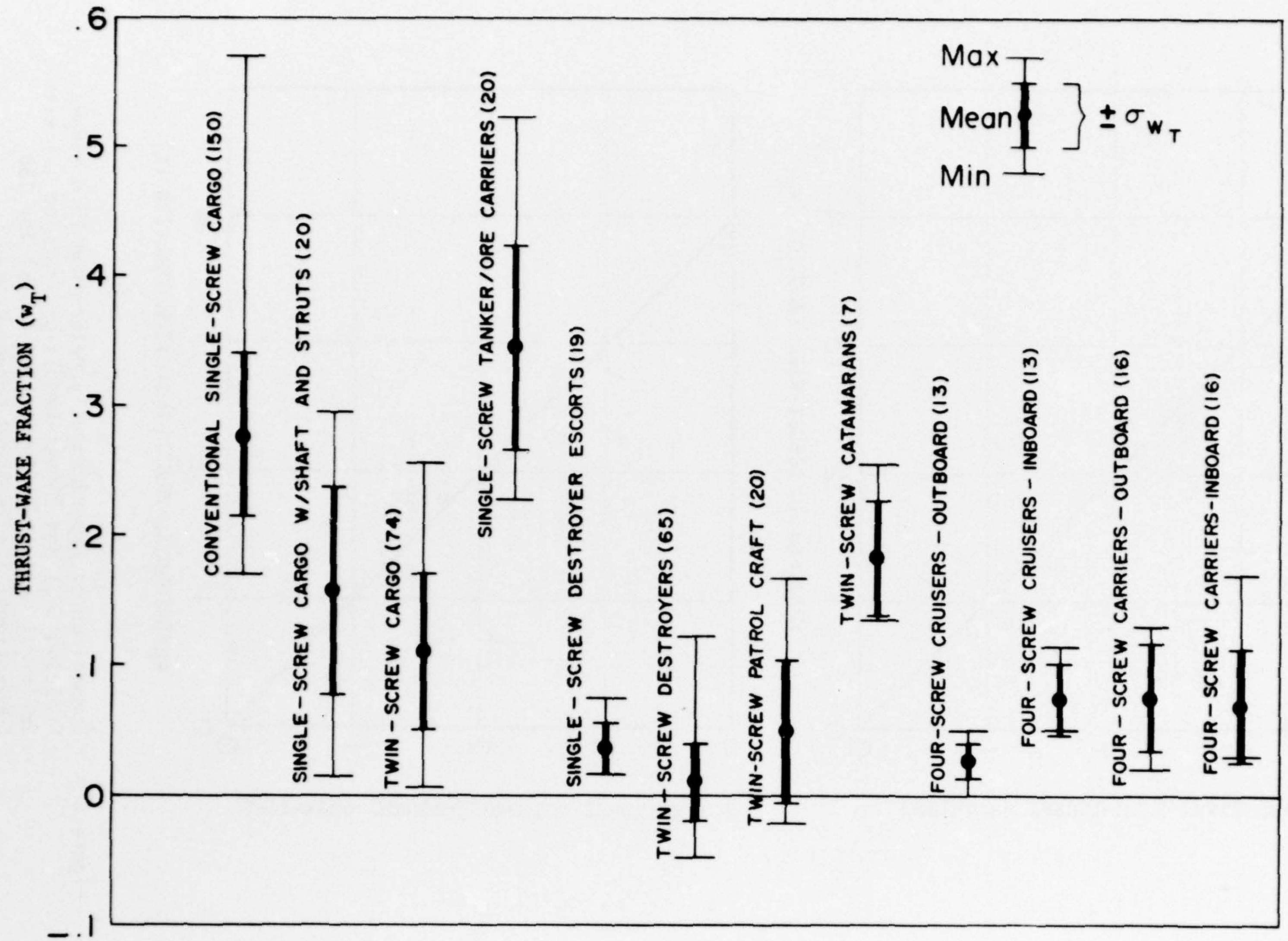


Figure B-5 Thrust-Wake Fraction (w_T) for 10 Types of Ships Tested at DTNSRDC

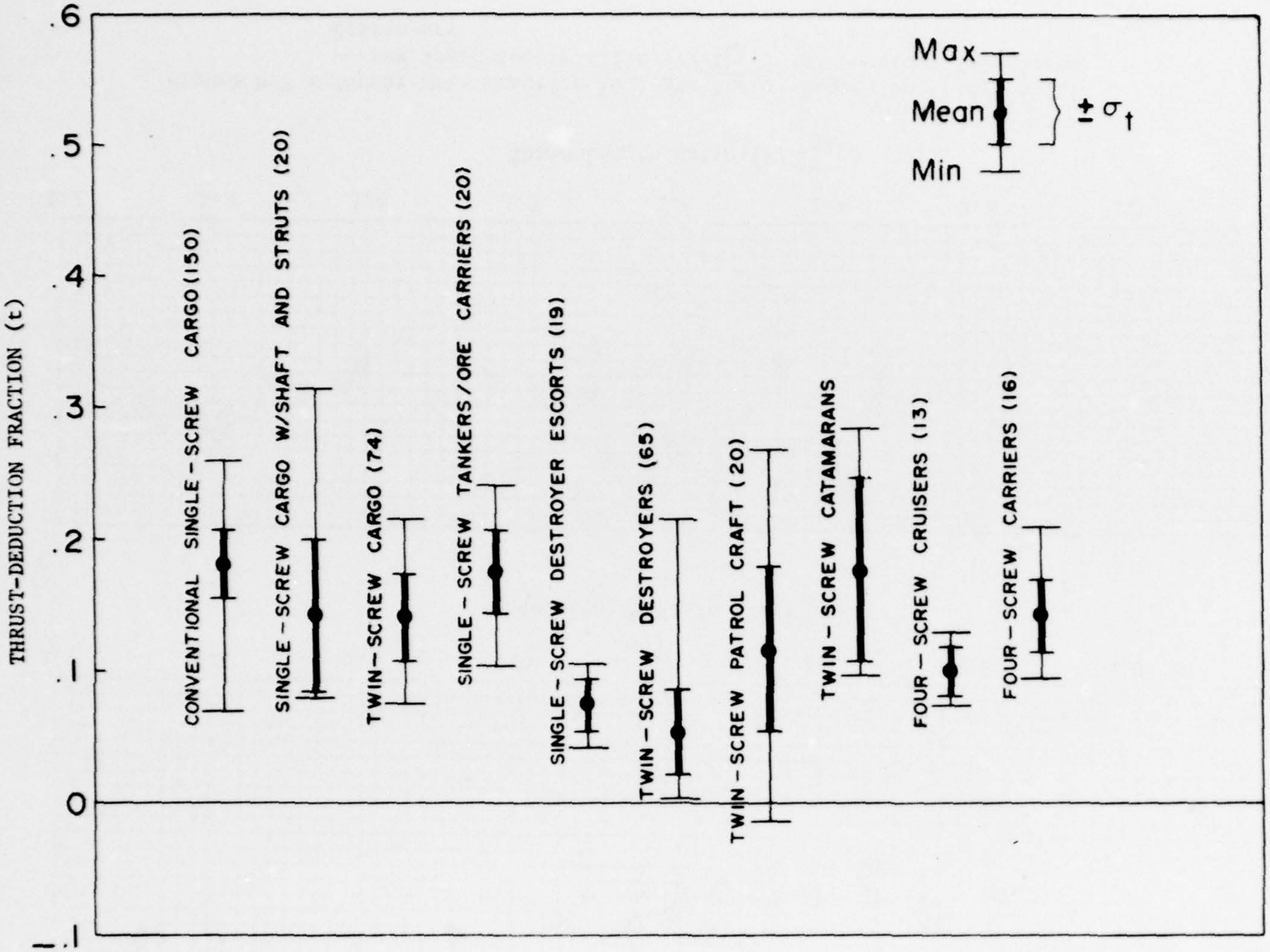


Figure B-6 - Thrust-Deduction Fraction (t) for 10 Types of Ships Tested at DTNSRDC

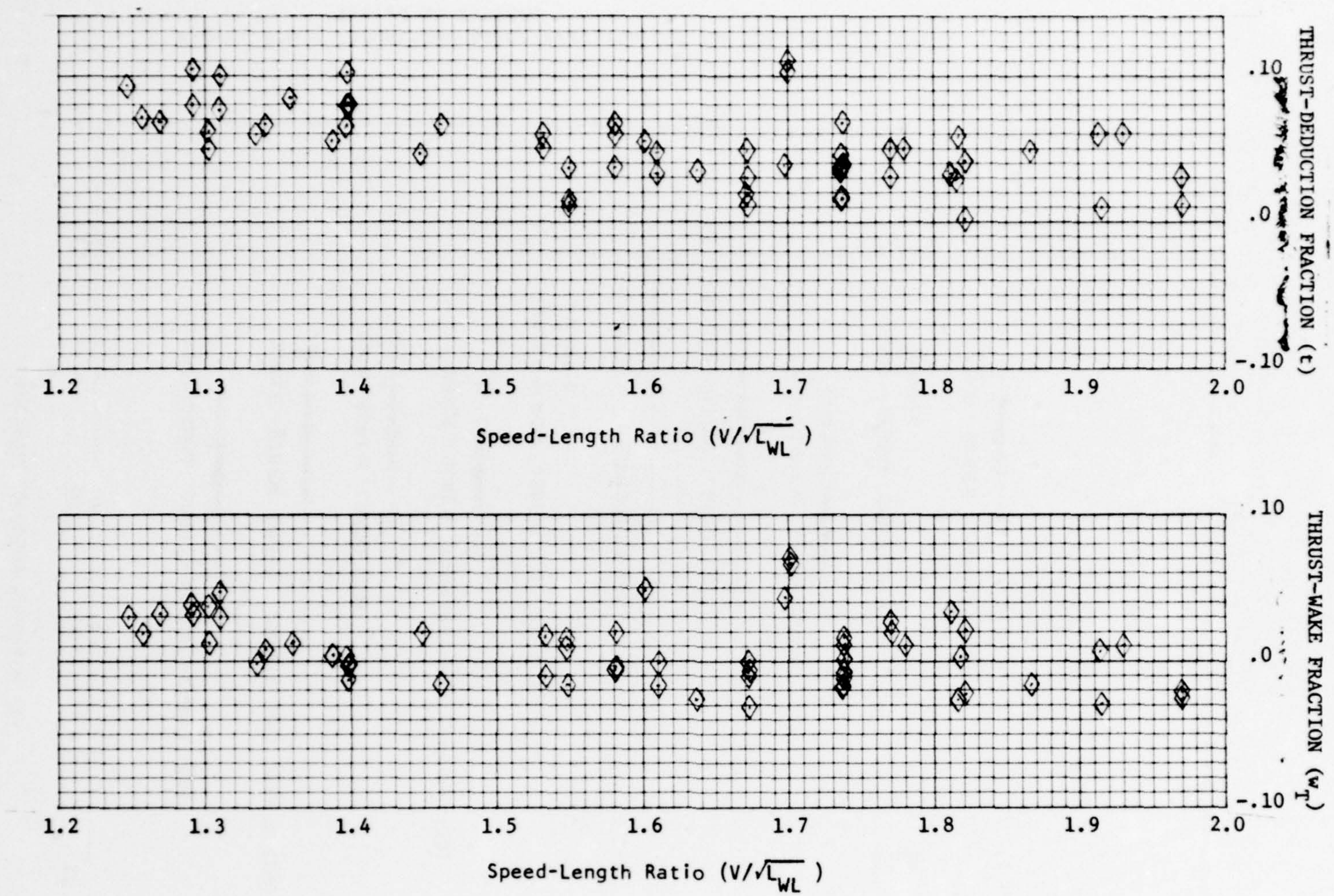


Figure B-7 - Thrust-Wake Fraction (w_T) and Thrust-Deduction Fraction (t) versus Speed-Length Ratio ($V/\sqrt{L_{WL}}$) for Numerous Twin-Screw Destroyers

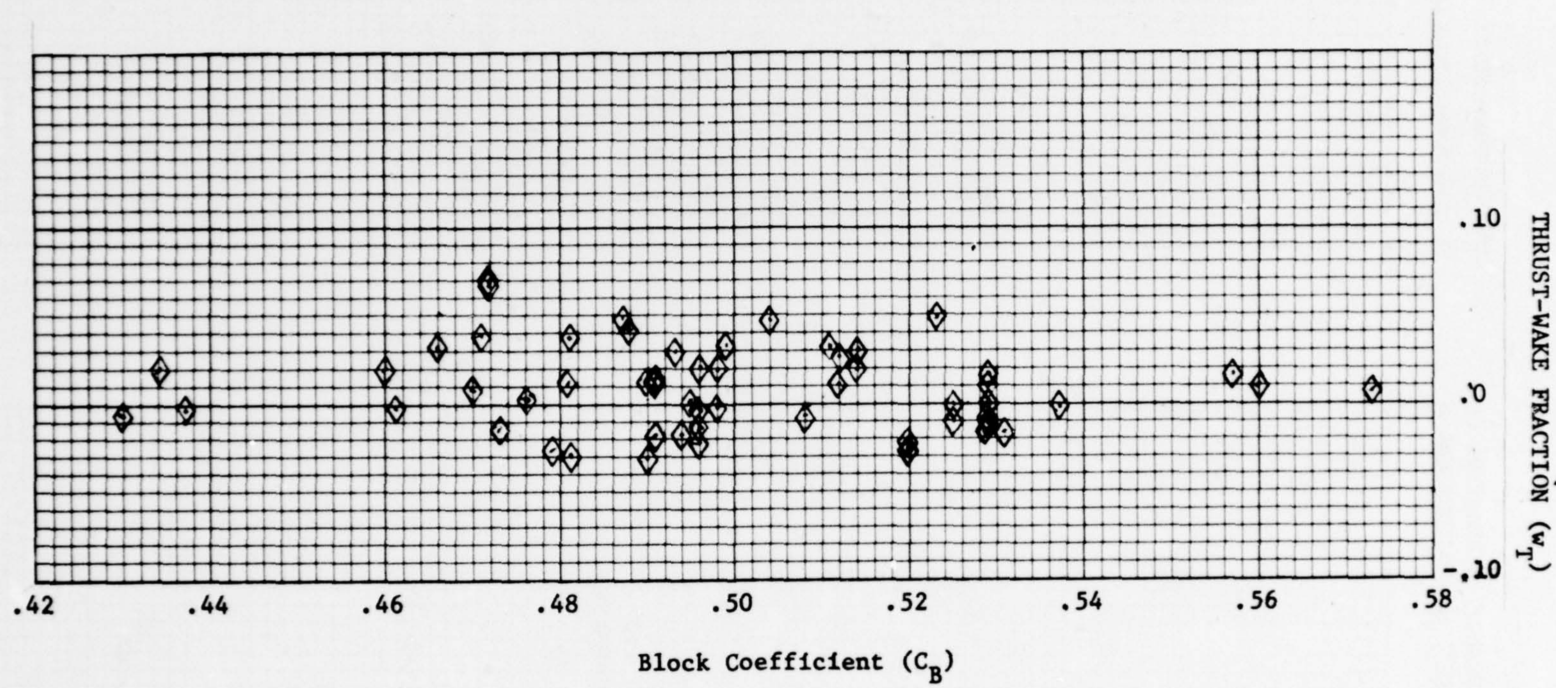


Figure B-8 - Thrust-Wake Fraction (w_T) versus Block Coefficient (C_B) for Numerous Twin-Screw Destroyers

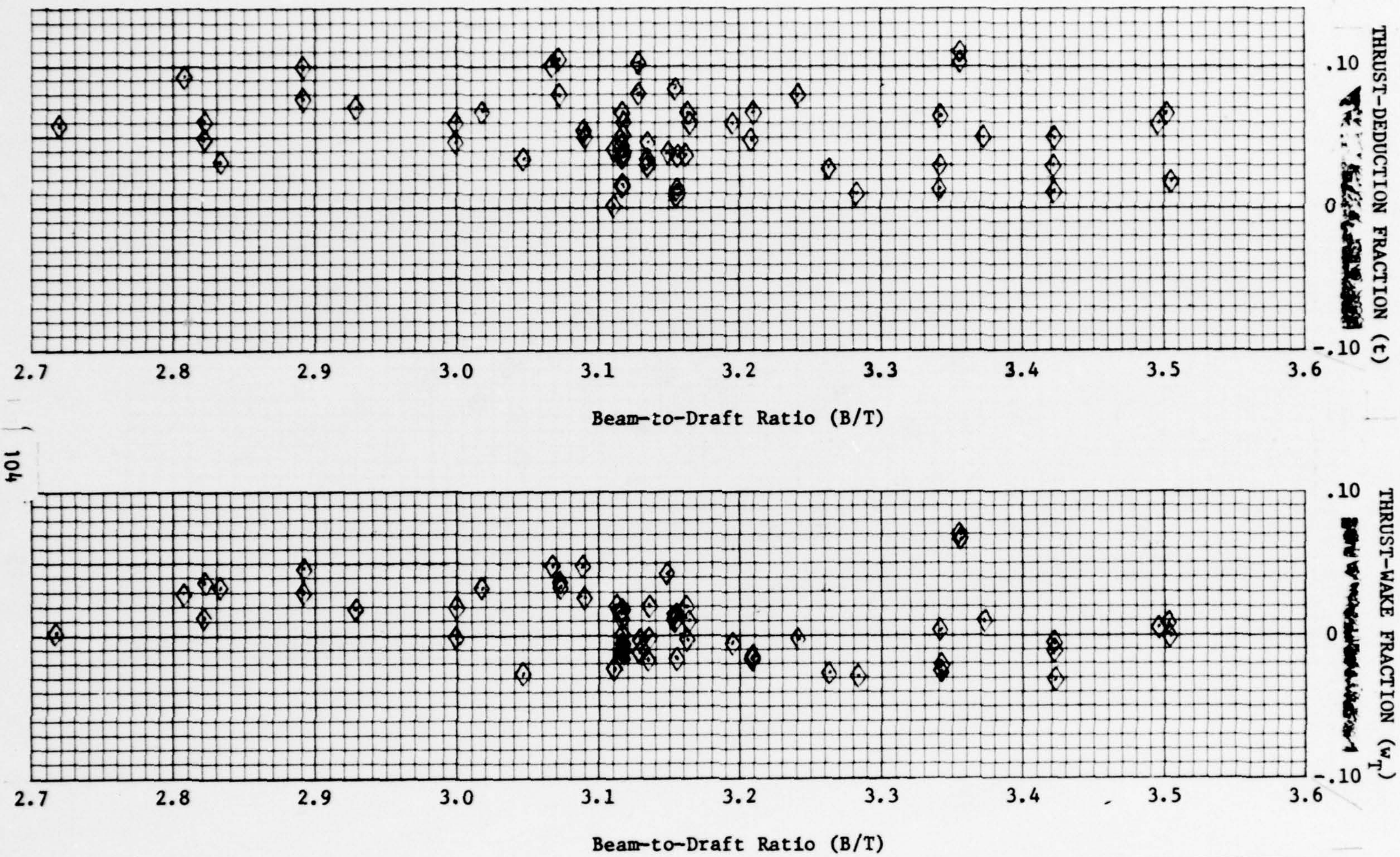


Figure B-9 - Thrust-Wake Fraction (w_T) and Thrust-Deduction Fraction (t) versus Beam-to-Draft Ratio (B/T) for Numerous Twin-Screw Destroyers

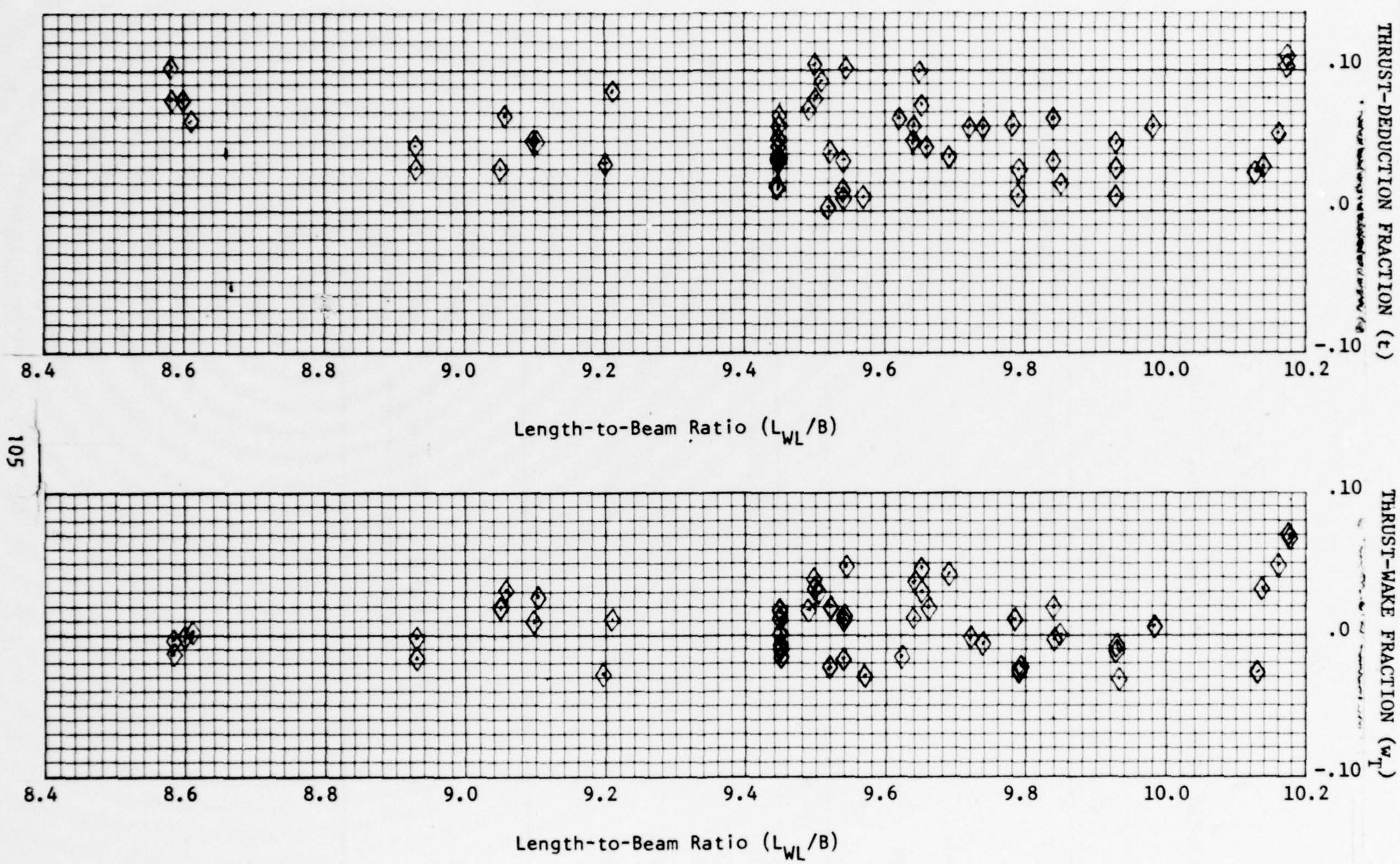


Figure B-10 - Thrust-Wake Fraction (w_T) and Thrust-Deduction Fraction (t) versus Length-to-Beam Ratio (L_{WL}/B) for Numerous Twin-Screw Destroyers

TABLE B-1

SUMMARY OF PREDICTIONS FOR 150 CONVENTIONAL SINGLE-SCREW CARGO SHIPS

THRUST-DEDUCTION FRACTION (t) $\bar{t}$ = .182		
METHOD	AVERAGE ERROR	STANDARD DEVIATION σ_t
Harvald	-.026	.040
Taylor	+.011	.047
$\bar{t}$	0	.026
THRUST-WAKE FRACTION (w_T) $\bar{w}_T$ = .277		
METHOD	AVERAGE ERROR	STANDARD DEVIATION σ_{w_T}
Harvald	-.010	.070
Taylor	+.004	.046
$\bar{w}_T$	0	.063

where

$$\sigma_x = \sum_{i=1}^{n} \frac{(x_{exp_i} - x_{method_i})^2}{n-1}$$

TABLE B-2

w_T FOR 10 CLASSES OF SHIPS TESTED AT DTMB AND NSRDC

CLASS OF SHIP/NUMBER OF EXPERIMENTS	AVERAGE	MAXIMUM	MINIMUM	σ_{w_T}
Conventional Single-Screw Cargo (150)	.277	.571	.170	.063
Single-Screw Cargo with Shafts and Struts (20)	.158	.295	.014	.080
Twin-Screw Cargo (74)	.110	.256	.007	.060
Single-Screw Tankers and Ore Carriers (20)	.345	.523	.229	.079
Single-Screw Destroyer Escorts (19)	.036	.074	.017	.019
Twin-Screw Destroyers (65)	.010	.121	-.049	.030
Twin-Screw Patrol Craft (20)	.049	.167	-.021	.055
Twin-Screw Catamarans (7)	.183	.255	.135	.045
Four-Screw Cruisers (13)				
Outboard Pair	.026	.050	.000	.014
Inboard Pair	.074	.140	.050	.027
Four-Screw Carriers (16)				
Outboard Pair	.075	.130	.020	.041
Inboard Pair	.069	.170	.030	.043

TABLE B-3

t FOR 10 CLASSES OF SHIPS TESTED AT DTMB AND NSRDC

CLASS OF SHIP/NUMBER OF EXPERIMENTS	AVERAGE	MAXIMUM	MINIMUM	σ_t
Conventional Single-Screw Cargo (150)	.182	.260	.070	.026
Single-Screw Cargo with/ Shafts and Struts (20)	.142	.314	.080	.058
Twin-Screw Cargo (74)	.141	.216	.075	.033
Single-Screw Tankers and Ore Carriers (20)	.176	.241	.104	.032
Single-Screw Destroyer Escorts (19)	.075	.107	.042	.020
Twin-Screw Destroyers (65)	.054	.215	.003	.033
Twin-Screw Patrol Craft (20)	.117	.269	-.013	.063
Twin-Screw Catamarans (7)	.178	.285	.098	.070
Four-Screw Cruisers (13)	.100	.130	.075	.019
Four-Screw Carriers (16)	.142	.210	.095	.028

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