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HYDRODYNAMIC DEVELOPMENT OF A HIGH SPEED PLANING HULL FOR ROUGH WATER

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ABSTRACT

The hydrodynamic development of a new planing craft intended for sustained high-speed operation in a seaway is discussed. The design philosophy is presented and then implemented to achieve optimum hull form and loading for both smooth and rough water operation of the craft. The resultant hull form is a high length-beam ratio, highly loaded, double chine configuration which provides greatly improved seakeeping, high speed and high maneuverability.

Extensive model tests were conducted to predict the SHP ; EHP ; seakeeping ; course-keeping stability ; and turning characteristics of the design. The extensiveness of the model test program and data analysis are unique for a planing craft. These results are presented in a form which should be of general interest to the designers of a high-speed planing craft.

INTRODUCTION

Although world navies are traditionally considered to be high sea state fleets with ocean-spanning capabilities, the small, high-speed craft is an essential complement to this fleet. These small craft are called into service to assist in ASW operations ; to patrol in coastal and riverine environments ; and to act in concert with larger naval units. Although ubiquitous in numbers, small craft are not individually a high portion of total naval capability, cost, or personnel. In fact, on an equal cost basis, it appears that more units can be procured which can cover more areas than can larger ships. These features have made the small, high-speed craft attractive as the principal naval force for many small countries with large coastlines.

There are basically four different types of small, high-speed craft, i.e., round-bottom boats, hard-chine planing craft, hydrofoil boats, and various forms of air-supported vehicles. The most numerous of these craft is by far the hard-chine planing hull-especially when considering speed-length ratios in excess of approximately 2.0 where dynamic lifting forces are significant. Because they are equipped with large power, lightweight engines, it is not uncommon for planing craft to operate at speed-length ratios in excess of 5.0. Further, it is also not uncommon for these craft to operate sufficiently removed from the coastline so that moderate to high sea states are their normal environment. Thus, the small boat designer is faced with the formidable task of producing craft whose high speed potential is not seriously compromised in rough seas.

The purpose of the present paper is to describe the hydrodynamic development of a new planing craft intended for sustained high-speed operation in a seaway. The design philosophy is presented and then implemented to achieve optimum hull form and loading for both smooth and rough water operation of the craft. The resultant hull form is a high length-beam ratio (6.5), highly loaded (beam loading = 0.75), moderate deadrise ($\beta = 20°$), double chine configuration which provides good seakeeping, high speed, and large maneuverability. Extensive model tests were conducted to predict the SHP ; EHP ; seakeeping ; course-keeping stability ; and turning characteristics of the design. The extensiveness of the model test program and data analysis are unique for a planing craft. The present paper presents these results in a form which should be of general interest to designers of high-speed planing hulls.

POSTULATED PERFORMANCE OBJECTIVES

The design of any marine craft is based upon specifications which have been prescribed to achieve desired performance objectives. Among the more significant requirements which have a pronounced influence on hull form are : operational speeds ; dimensions and weight of payload components ; sea state; tolerable "g" loadings in that sea state ; restrictions on draft, length, and beam, metacentric stability ; and maneuvering qualities. Certainly, special purpose craft have additional restrictions but these are not included in this study.

For the present paper, the following set of performance specifications are prescribed.

1. Full Load Displacement : 150,000 lbs.

A designer's experience is invaluable in making the first engineering estimate of full load displacement. Nevertheless, design studies and possible model tests are carried out at additional displacements of approximately 80% and 120% of the initial estimate.

2. Maximum Speed : In excess of 40 knots.

Interception and attack missions require high speeds. For the present study, a nominal speed of 45 knots is assumed.

3. Cruise Speed : Approximately 12 knots.

A patrol mission requires long endurance at slow search speeds. A cruise speed of 12 knots is selected.

4. Maximum Hull Draft : 3.5 ft.

5. Operational Sea State : 3.

It is desired that the craft operate at 45 knots in a state 3 head sea having a significant wave height of 4.6 ft.

6. Average Center-of-Gravity Impact Acceleration : $(\eta_{CG})_{avg} = 0.4g$

It is specified that the average center-of-gravity impact acceleration should not exceed 0.4g while running in a state 3 head sea at 45 knots.

7. Metacentric Stability : GM = 3.0 ft.

A GM of 3.0 ft. was selected to provide metacentric stability under conditions of high wind and severe super-structure icing.

The above set of requirements do not pre-specify the length or beam of the craft. There may, of course, be operations where either of these dimensions must be fixed in advance. The design procedure developed in subsequent sections of this paper can be equally applied to those cases and will show the extent to which the pre-specified length, for example, may inhibit attainment of the performance requirements.

DESIGN PROCEDURES FOR BASIC HULL DIMENSIONS

In this section of the paper a methodology is developed for rationally selecting the length, beam, longitudinal center-of-gravity, nominal deadrise, and effective horsepower of the basic hull form which will satisfy the performance requirements previously specified. No attempt is made to optimize the hull design to attain, say, minimum resistance while satisfying the seakeeping requirements. This can be developed as a subsequent study using the basic design procedures developed herein.

The design procedure is primarily based upon a combination of smooth water prediction techniques such as given in References 1 and 2, and rough water prediction techniques such as given in Reference 3 . While both studies are concerned with prismatic planing hulls (constant beam, constant deadrise, buttocks parallel to the keel) these techniques have been successfully applied to actual hull forms by proper selection of an effective constant deadrise and beam.

Separate considerations are first given to relating hull dimensions to the following hydrodynamic characteristics

1. Hydrodynamic Impact in a Seaway
2. Hydrostatic Displacement
3. Smooth Water Planing (High Speed)
4. Smooth Water Operation (Low Speed)
5. Metacentric Stability

The results of these elemental studies are then combined to specify a hull form, overall dimensions, center of gravity, and effective horse-power to achieve the operational objectives.

Hydrodynamic Impact in a Seaway

In Reference 3 , Fridsma presents the results of a systematic study of the effects of deadrise, trim, loading, length-beam ratio, bow section shape, speed, and sea state on the performance of a series of prismatic planing models operating in irregular waves. A statistical analysis of the measurements of added resistance, heave motions, pitch motions and impact accelerations resulted in quantitative relations between these measured quantities and hull dimensions and operating conditions. The results of those parametric studies are summarized in design charts which enable full-scale performance predictions for planing craft. Using the procedures described by Fridsma, the average center of gravity acceleration is computed for a range of combinations of beam, length-beam ratio, deadrise and trim angle for a maximum speed of 45 knots in a state 3 head sea. To enter the design charts of Reference 3 , the following coefficients are evaluated :

C_Δ	= Beam loading = $\dfrac{\Delta}{wB^3}$
$V_k/\sqrt{L}$	= Speed-length ratio
L/B	= Length-beam ratio
$H_{1/3}/B$	= Significant wave height/beam = $H_{1/3}/B$
C_v	= Speed coefficient = $V/\sqrt{gB}$

where

Δ	= displacement = 150,000 lbs.
V_k	= maximum speed = 45 knots
V	= maximum speed = 76 ft/sec.
L	= length between perpendiculars, ft.
B	= average beam over aft 80% of hull, ft.
$H_{1/3}$	= significant wave in state 3 sea = 4.6 ft.
w	= weight density of water = 64 lbs/ft^3

ρ = mass density of water = 2 $lb\text{-}sec^2/ft^4$

and

β = deadrise angle at station 5, degrees

τ = trim angle of mean buttock line, degrees

$(\eta_{CG})_{avg}$ = average vertical acceleration at center of gravity "g"

$(\eta_{CG})_{1/10}$ = 1/10 highest vertical acceleration at center of gravity "g". From the statistical analysis of Reference 3, $(\eta_{CG})_{1/10} = 3.3\ (\eta_{CG})_{avg}$

The previous coefficients are evaluated for a range of initially assumed values of beam :

B	C_Δ	$1/C_\Delta^2$	$H_{1/3}/B$	C_V
13 ft.	1.070	.86	.353	3.76
14	.856	1.36	.328	3.58
15	.695	2.06	.307	3.45
18	.402	6.10	.258	3.16

The following relations between initially assumed beam, assumed length-beam ratio, and speed-length ratio will also prove useful :

	B = 13 ft.		B = 14 ft.		B = 15 ft.		B = 18 ft.	
L/B	L	$V_k/\sqrt{L}$	L	$V_k/\sqrt{L}$	L	$V_k/\sqrt{L}$	L	$V_k/\sqrt{L}$
4	52 ft.	6.2	56 ft.	6.0	60 ft.	5.8	72 ft.	5.3
5	65	5.6	70	5.4	75	5.2	90	4.7
6	78	5.1	84	4.9	90	4.7	108	4.3

Referring to Figures 16, 17, and 18 of Reference 3, which are reproduced as Figures 1, 2, and 3 of the present report, it is seen that the average value of center-of-gravity acceleration is obtained directly from these design charts for arbitrary combinations of L/B, $1/C^2_\Delta$, $V_k/\sqrt{L}$, and $H^{1/3}/B$ such as listed above. It is to be noted that these results are for a trim angle of 4° and a deadrise angle of 20°. Corrections for other combinations of trim and deadrise angle will be described subsequently.

For each assumed value of beam, the average CG accelerations for 45 knots in a state 3 head sea is obtained from the design charts of Figures 1-3 and plotted on the right half of Figure 4 as a function of length-beam ratio. These results are obtained by extrapolations of the design charts as suggested in Reference 3 . The ordinate of the plot in Figure 4 is the quantity :

$$\eta_{CG} \text{ (Average)} \left[\frac{\tau^\circ}{4}\left(\frac{5}{3} - \frac{\beta^\circ}{30}\right)\right]$$

which defines the dependence of acceleration upon trim and deadrise as developed in Reference 3. For $\tau = 4°$ and $\beta = 20°$, the quantity in the square brackets is unity so that η_{CG}(Average) as given in Figure 4, corresponds to the design charts of Reference 3 . Superposed on Figure 4 are curves of constant boat length for various combinations of beam and length-beam ratio.

The quantity $\tau°/4\ (5/3 - \beta°/30)$ is plotted on the left half of Figure 4 for ease of applying these results to arbitrary combinations of τ and β .

Some interesting observations can be made by an examination of the results in Figure 4 :

Effect of Beam on Hydrodynamic Impact :

All other conditions being equal, a reduction of hull beam leads to significant reductions in impact load. For example, a 28% reduction in beam (from 18 ft. to 13 ft.) results in a 69% reduction in impact accelerations (from 1.10g to 0.35g). Even a 1 ft. reduction in beam, from 15 ft. to 14 ft., decreases the impact acceleration by approximately 29%. This powerful effect of beam results from the large dependence of impact upon the inverse of beam loading coefficient $C_\Delta = \Delta/wB$. The effect of C_Δ has long been familiar to the designer of water-based aircraft (References 4 and 5) and has just recently been quantitatively identified by Fridsma[3] for the case of the planing hull.

Effect of Trim Angle on Hydrodynamic Impact :

The hydrodynamic impact load is linearly dependent upon trim angle so that, within the range of data acquisition, a 50% reduction in trim angle (from 4° to 2°) results in a 50% reduction in hydrodynamic impact load. The reference trim angle is the smooth water running trim of the craft for the considered hull dimensions, loadings, and speed.

Effect of Deadrise Angle on Hydrodynamic Impact :

The accelerations decrease linearly with increasing deadrise so that a 50% reduction is achieved by increasing the deadrise from 10° to 30°.

In order of importance, then, impact loads in a seaway can be reduced by providing a hull having a narrow beam, a low running trim angle, and a high deadrise. As will be subsequently developed, this results in as long and narrow a hull as can be accepted without seriously compromising other essential operational conditions.

Relation Between Beam and Trim Angle to Achieve $(\eta_{CG})_{avg} = 0.4g$:

Considering an initial deadrise angle of 20°, the relations between beam and trim angle to achieve $(\eta_{CG})_{avg} = 0.4g$ can be established. The following tabulation follows from Figure 4.

B	$\tau = 4°$, $\beta = 20°$ $(\eta_{CG})_{avg}$ (1)	K (2)	τ (3)
13 ft.	.35 g	1.14	4.5°
14	.50	.80	3.2
15	.72	.55	2.2
18	1.00	.40	1.6

(1) Average value for range of L/B as given in Figure 4.

(2) $K = \dfrac{.40\ g}{(\eta_{CG})_{avg}\ (\tau = 4°,\ \beta = 20°)} = \dfrac{\tau°}{4}\left(\dfrac{5}{3} - \dfrac{\beta°}{30}\right)$

(3) τ = equilibrium trim angle to achieve $(\eta_{CG})_{avg} = 0.40\ g$ for

$\beta = 20°$, $V_K = 45$; $\Delta = 150,000$ lb.

$$\tau = \frac{4K}{\left(\frac{5}{3} - \frac{20}{30}\right)} = 4K$$

Static Displacement Considerations

The block coefficient for planing hulls can vary between 0.40 and 0.50. Having fixed a draft of 3.5 ft. for the present design, the relations between length and beam will take on a more limited set of values than those previously developed from considerations of only hull impact.

$$C_B = \frac{\Delta}{L\ B\ d\ w}$$

where

Δ = 150,000 lbs.

L = length between perpendiculars, ft.

d = maximum draft, 3.5 ft.

w = weight density of water, 64 lbs/ft^3

B	$C_B = 0.40$ L	$C_B = 0.45$ L	$C_B = 0.50$ L
13 ft.	128 ft.	114 ft.	103 ft.
14	119	106	95
15	111	98	89
18	93	82	75

These combinations of length and beam are superposed on the plots of Figure 4. This results in a substantial reduction in length-beam combinations for further study.

Smooth Water Planing (45 knots)

A computational procedure for predicting the smooth water equilibrium conditions of a planing hull is given by Savitsky[1]. This procedure has been programmed for high-speed computers and is generally available to the small boat naval architect. In Reference 2, Hadler extends this work to include the effects of propellers and appendages. Unfortunately, a computer program for this extended computation is not yet generally available to the small boat naval architect.

For the present study, which is intended to define the principal hull dimensions for preliminary design, the simplified computational procedure developed by Savitsky is considered adequate. A 20° deadrise hull is initially assumed and the equilibrium trim and wetted keel length are computed for values of beam between 13 ft. and 18 ft; for longitudinal center-of-gravity positions between 22 ft. and 44 ft. forward of the transom, for a planing speed of 45 knots and a displacement of 150,000 lbs. The results of this computation are plotted in Figure 5. For each value of beam, the trim angle required to achieve $(\eta_{CG})_{avg} = 0.4$ is also indicated.

Using the results of Figure 5, the following relation between beam, LCG, wetted keel length and suggested L (load waterline length) is obtained. These values are also plotted in Figure 6.

B	τ	LCG	L_k	L	C_B	$L/\nabla^{1/3}$
13 ft.	4.5°	32.5 ft.	62 ft.	68 ft.	.76	5.2
14	3.3	38.0	82	90	.53	6.8
15	2.2	42.5	96	106	.42	8.0
18	1.6	44.0	110	121	.31	9.2

The length L is load waterline length and is taken to be equal to 1.10 L_k. It has been found that this relation between L and L_k is most satisfactory. If L is less than 10% greater than L_k, there is substantial bow immersion at high speed resulting in a significant increase in smooth water resistance. When L is much larger than 1.10 L_k, the excessive hull length forward of L_k provides additional impact area when running in a seaway with resultant increasing impact loads.

Also included in the previous tabulation are the corresponding values of block coefficient C_B and slenderness ratio $L/\nabla^{1/3}$. These will be subsequently discussed.

<u>Smooth Water Operation (Low Speeds)</u>

Since one of the operational objectives for the craft is to cruise at 12 knots for extended periods, the hydrodynamic resistance should be minimized. For a displacement of 150,000 lbs., the volume Froude number

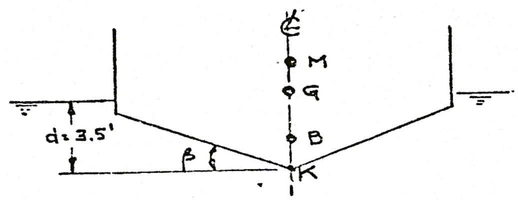
$$F_\nabla = \frac{V}{\sqrt{g\nabla^{1/3}}} = \frac{20.4}{\sqrt{32.2 \times (2350)^{1/3}}} = 1.0$$

From a basic study of the resistance of planing hulls in the prehump speed region (Reference 6), it has been found that low speed resistance is primarily dependent upon slenderness ratio $L/\nabla^{1/3}$. Figure 7 of this report shows the variation of resistance-weight ratio of the Series 62 hull form (Reference 7) as a function of $L/\nabla^{1/3}$ for $F = 1.0$. It is seen that, although the data points represent a wide range of hull loadings and length-beam ratios, they are closely represented by a single curve relating the data to $L/\nabla^{1/3}$. Similar plots for other hull series are given in Reference 6 and also indicate the pronounced dependence of low speed resistance on slenderness ratio. Using these results as a guide, it is concluded that a value of $L/\nabla^{1/3}$ equal to at least 7.0 is essential for low resistance at pre-hump speeds. For a displacement of 150,000 lbs., this results in a minimum hull length of 92 ft.

Comparing this criteria with possible combinations of hull dimensions of page 429, it is seen that a hull beam should be at least equal to or greater than 14 ft. This limitation is also plotted on Figure 6.

Metacentric Stability

The specified requirement for metacentric stability is that the GM be equal to at least 3.0 ft. An engineering estimate of GM for a planing craft with deadrise is made using the following assumptions.



$$KB = \frac{2}{3} d = \frac{2}{3}(3.5) = 2.33 \text{ ft.}$$

$$I = \text{water plane moment of inertia} = .55 \left(\frac{1}{12} L B^3\right)$$

Preliminary weight estimates for the present design showed that the vertical center of gravity of the craft, KG, is 6.2 ft. above the Keel. Considering a minimum boat lenght of 92 ft., as determined from the low speed requirement, the following values of GM are computed for each assumed beam.

B	I	BM	KM	GM
13 ft.	9,110 ft.^{4}	3.9 ft.	6.2 ft.	0.0 ft.
14	11,400	4.9	7.2	1.0
15	14,000	6.0	8.3	2.1
15.7	16,410	7.0	9.3	3.1
18	24,200	10.4	12.7	6.5

where

$$KG = 6.2 \text{ ft.}$$

$$BM = 1/V$$

$$V = 150,000/64 = 2,340 \text{ ft}^3$$

$$KM = KB + BM$$

$$GM = KM - KG$$

A minimum beam of 15.7 ft. is required to attain the specified metacentric height of 3.0 ft.

Selection of Basic Hull Dimensions

The previous elemental hull studies developed the following restrictions on hull dimensions to satisfy the postulated operating conditions.

Low Speed Operation (V_k = 12 knots)

For low resistance in the cruise condition, the hull length should be equal to or greater than 92 ft.

Metacentric Stability (GM = 3.0 ft.)

For a hull length of 92 ft., the required hull beam is 15.7 ft.

Hydrodynamic Impact $(\eta_{CG})_{avg} = 0.4$ g in state 3 head sea)

Two possible options are available for a 20° deadrise hull.

$$(1)\ \left[\begin{array}{lcl} B & = & 15.7 \text{ ft.} \\ L & = & 116 \text{ ft.} \\ LCG & = & 43.5 \text{ ft.} \\ C_B & = & .36 \end{array}\right.$$

$$(2)\ \left[\begin{array}{lcl} B & = & 14 \text{ ft.} \\ L & = & 92 \text{ ft.} \\ LCG & = & 38 \text{ ft.} \\ C_B & = & .53 \end{array}\right.$$

Option (1) satisfies all the hydrodynamic and operational requirements while Option (2) does not satisfy the basic metacentric height requirement. The 116 ft. boat lenght of Option (1) was considered excessive when compared with space requirements for internal arrangements. This excessive length would increase the cost of the boat without materially improving its performance.

As a compromise design, a double chine hull having a length of 92 ft. was selected. A mid-length section through this hull is shown in Figure 8. The inner chine beam is 14 ft. and the outer chine beam is 15.7 ft. In the static or low speed condition, the wetted beam is 15.7 ft. so that the metacentric stability requirement is satisfied by the combination of L = 92 ft. and b = 15.7 ft. At the 45-knot condition, the flow separates from the lower chine so that the 92 ft. long hull has an effective beam of 14 ft. This satisfies the high-speed impact requirements. The metacentric stability of the craft is now satisfied by considering the additional roll stabilizing moments developed by the dynamic loads generated at high speeds.

In summary then, the following basic hull dimensions and loadings are selected for preliminary design

$$L = 92 \text{ ft.}$$

B_{outer} = 15.7 ft.

B_{inner} = 14 ft.

LCG = 38 ft.

β = 20° (Sta. 5)

Δ = 150,000 lbs.

At a smooth water planing speed of 45 knots, this craft is expected to run at a trim angle of 3.3° and develop $(\eta_{CG})_{avg}$ = 0.4 g in a State 3 head sea.

It will be noted that a nominal 20° deadrise hull has been selected. Considerations were also given to deadrise angles of 10° and 30°. The details of these calculations (which follow the previous procedures) will not be presented, but the results will be summarized. Figure 9 presents a plot of resistance-weight ratio (R/Δ) versus trim angle for deadrise angles of 10°, 20°, and 30°. These results follow from Reference 1. Superposed on Figure 9 are the maximum trim angles for combinations of deadrise and beam which will result in $(\eta_{CG})_{avg}$ = 0.4 g when the 150,000 lb. craft runs at 45 knots in a State 3 head sea.

Considering the 30° deadrise case, it is seen that a 14 ft. beam requires a trim angle equal to or less than 5° to satisfy the impact requirements. Further, the minimum drag-lift ratio for a 30° deadrise craft occurs at a trim angle of approximately 5°. This high trim angle could impair visibility during high-speed operation and was thus not considered acceptable. A reduction in trim angle to improve visibility would reduce the impact loads below the maximum acceptable value but, according to Figure 9, would significantly increase the drag-lift ratio to values substantially larger than for the 20° deadrise surface. Thus, the 30° deadrise case was not further considered.

A 10° deadrise surface having a beam of 14 ft. would be required to run at a trim angle of 2.4° so as not to exceed the design impact acceleration. For this case, the drag-lift ratio would be slightly less than that for the 20° deadrise surface. The boat length and LCG position for β = 10°, τ = 2.4°, b = 14 ft., Δ = 150,000 lbs. and V_k = 45 knots are computed from the monograph given in Figure 19 of Reference 1:

$$L_k = 69 \text{ ft.}$$

$$LCG = 35 \text{ ft.}$$

$$L = 1.10 \times L_k = 76 \text{ ft.}$$

The boat length of 76 ft. is 20% less than the 92 ft. length required for low resistance at 12 knots and is thus not acceptable.

In summary then, the 20° deadrise hull was accepted as best meeting the design requirements of the craft.

Description of Final Hull Form

The lines are shown in Figure 10 and show that the principal dimensions of the craft are in substantial agreement with those developed by the design procedures presented herein.

Length, design waterline	92'-0"
Beam, lower chine (nominal)	14'-0"
Beam, upper chine (nominal)	15'-7"
Deadrise, station 5	20°
Deadrise, station 10	10°
Draft (full load)	3.5

It is seen that the craft is a hard, double-chine hull whose high-length beam ratio is favorable for low resistance and good seakeeping. Several detail design features are of interest and will be separately described. Koelbel (Reference 8) provides excellent design guidance in this regard.

Chine Configuration

It has been found that, for planing craft operating at speed-length ratios greater than approximately 2.0 - 2.5, a hard chine is required to assure complete separation of the flow from the bottom. At these speed-length ratios, a round bilge hull will prevent flow separation and result in significant side wetting and thus increase the hydrodynamic drag. The present 45-knot design condition corresponds to a speed-length ratio of 4.6 and clearly requires a hard chine configuration.

In designing a double chine configuration, it is important that the upper chine location be within the cavity found by the boundary of the flow separation from the lower chine. As shown by Korvin-Kroukovsky (Reference 9), the trajectory of the free streamline representing the cavity of the boundary is a function of deadrise angle. Figure 11 of the present report, which is taken from Reference 9, plots the separation trajectory for various deadrise angles. It is seen that the width of the separation cavity increases with decreasing deadrise angle. If the upper chine is located outboard of this cavity boundary, the originally separated flow from the lower chine will reattach to the bottom somewhere between both chines and thus preclude complete flow separation from the lower chine. In the present design, the outer deadrise angle is 45° and the outer chine is approximately 0.80 ft. outboard of the inner chine. This is sufficient to clear the lower chine trajectory at station 5 and result in complete separation from the lower chine. Observations of the wetted bottom areas during model tests confirmed this prediction.

Section Shapes

The section shapes are slightly convex. This section pounds less than others of equal deadrise because there is less likelihood of instantaneous water contact over large bottom areas.

Planform Shape

At high planing speeds, when dynamic lift predominates, it is usual to narrow the beam towards the stern. This reduces bottom friction without a noticeable loss in lift. The narrow transom also avoids the possibility of reattachment of the separation cavity formed in the region of maximum beam. For the present design, the transom width was determined by considerations of space requirements for the auxiliary machinery in the stern area. This resulted in a slight reduction of beam towards the stern which, in the model tests, was found to be sufficient to avoid flow reattachment.

Bottom Warp

The increase in deadrise with length forward of the transom is referred to as bottom warp and is required to provide a relatively high deadrise in the bow regions. Brown (Reference 10) has shown that there is a slight reduction in planing efficiency for moderate values of warp. The slightly convex bottom sections, from keel to lower chine, used in the planing area aft of the high-speed stagnation line are easily warped to result in increased deadrise and curvature in the bow sections. This combination reduces pounding and impact pressures in a seaway. The transverse section shape above the lower chine is increasingly more concave as the bow is approached. This upper "flare" is desirable to deflect the bow spray outboard of the deck and to provide additional buoyancy to reduce low-speed pitching in a seaway.

Spray Rails

Spray rails are provided along both chines to assure flow separation from the chines. The spray rail for the upper chine must not extend into the separated flow cavity formed by the lower chine. Otherwise flow reattachment will occur at high speeds. Separation from the upper chine occurs at a speed-length ratio between 2.0 to 2.5 while separation from the lower chine is expected to occur at a speed-length ratio of approximately 3.0.

Final Design

An artist's conception of the final design is given in Figures 12 and 13.

MODEL TESTS

Model tests were conducted at the Davidson Laboratory, Stevens Institute of Technology to evaluate the performance of the craft in smooth water and waves. A 1/11-scale model was used to determine EHP and SHP. A 1/16-scale model was used to investigate the seakeeping, maneuvering, and turning ability of the craft. Some of the principal results and test procedures are presented herein.

Resistance and Propulsion

Smooth Water Resistance

A 1/11-scale model was constructed according to the lines of Figure 10. To assure flow separation from the bottom, the upper and lower chines of the model were sharpened by the addition of mylar plastic strips which projected vertically a distance of 1/32 of an inch below each chine. Tests were made for a range of loadings and speeds. The test procedure simulated towing the model through the shaft axis. Measurements were made of the heave, trim, drag and wetted areas. For the purpose of the present paper, Figure 14 presents a comparison between values of trim and drag computed by the procedures of Reference 1 and the results of model tests. The comparison is for a displacement of 150,000 lbs. with an LCG of 38 ft. In the computational procedure, the upper chine beam (15.7 ft.) is used for speeds up to 40 knots and the lower chine beam (14 ft.) is used for higher speeds. This was consistent with test results where complete flow separation from the lower chine was observed at speeds greater than approximately 40 knots (full-scale equivalent). An effective deadrise angle of 20° was used in the computations.

It is seen that the computed and measured results agree well enough to justify use of Reference 1 for engineering estimates of planing boat performance. At speeds below 20 knots, extensive bow immersion precluded application of the methods of Reference 1 which are restricted to prismatic-like planing hulls. Reference 6 will provide procedures for performance estimates at low speeds where bow immersion is significant.

It is interesting to note the complete absence of a "hump" trim in Figure 14. This is attributed to the high-length beam ratio hull which, for normal LCG positions, is constrained to run at low trim angles. The low trim is, of course, most beneficial to improved seakeeping.

Self-Propelled Tests

Self-propelled tests of the 1/11-scale model were carried out to determine propulsion characteristics, e.g. wake fraction, thrust deduction coefficient, relative rotative efficiency and, subsequently, predictions of delivered horsepower.

The test program included resistance tests of the partly appended model, open water tests of the stock propellers used in propulsion tests and self-propelled tests of the 1/11-scale model for overload and underload conditions (so-called "British" method) at a number of speeds and displacement conditions. The open-water tests were carried out with the shaft horizontal and with a shaft inclination of 12°. Self-propelled tests were made with all three propellers driving and instrumented.

The rudders were not fitted for these tests since they are located approximately 4 propeller diameters aft of the propellers, out-of-line with any of the propeller races and, consequently, could have little influence on propeller-hull interaction.

Three propeller dynamometers were installed in the model for measuring thrust, torque and RPM. These were "reaction" type dynamometers having capacities of 10 lb. thrust, 5 in-lb. torque, RPM up to 10,000 and 0.50 HP. The averaging of the force and motion signals, as well as additional data processing, was accomplished using a PDP-8E computer on line. The computer has a built-in analog-to-digital converter and is programmed to carry out operations such as signal averaging, correcting for zero levels, and multiplying by calibration factors to obtain results in engineering units.

Davidson Laboratory uses the overload and underload testing procedure where a group of test runs are carried out at fixed speed with various rates of propeller rotation. This type of test provides information which may be applied for any desired assumptions concerning appendage drag, roughness allowance, scale ratio, air drag or rough water-drag increment.

Some typical propeller-hull interaction factors, derived from the test data for the model self-propulsion point (towing force = 0), are given for a speed corresponding to 45 knots and a displacement of 150,000 lbs.

$$\text{Wake fractions} = \begin{matrix} 1 - W_T = 0.99 \\ 1 - W_Q = 0.96 \end{matrix}$$

$$\text{Relative rotative efficiency} = {}_{RR} = Q_o/Q_B = 0.92$$

$$\text{Thrust deduction} = 1 - t = 0.94$$

These values of wake fraction, relative rotative efficiency and thrust deduction are used to select the particular propeller design which absorbs the installed power at the proper RPM and speed and has good efficiency even while operating under cavitating conditions.

It is interesting to note that the thrust wake fraction is 0.99 indicating an essentially undisturbed flow to the propeller. The thrust deduction is small, 1-t = 0.94, indicating a small effect of propeller-induced flow on the hull resistance.

Rough Water Tests

The rough water performance was measured for several loads and LCG positions in a variety of sea states. Measured quantities included heave and pitch motions, vertical accelerations at the bow and CG, and mean resistance in waves. During each test run, the data were processed by a PDP-8E computer on line. Each channel of data was analyzed at the rate of 200 scans per second and, at the conclusion of each test run, an ordered listing of the peaks and troughs of the pitch and heave motions and the accelerations at the bow and CG were printed out in addition to statistics such as 1/10, 1/3, and average values. This instantaneous output of processed data was extremely useful in interpreting the results.

A comparison between the computed average CG acceleration and the results of model tests is tabulated below for a displacement of 150,000 lbs., a speed of 45 knots and a range of LCG in a head State 3 sea.

	$(\eta_{CG})_{avg}$	
LCG	Computed	Measured
38	0.40 g	0.35 g
34	0.50 g	0.45 g
30	0.60 g	0.55 g

It is seen that the computed values are approximately 0.05g larger than the measured values. The average values are used in this comparison since, in random sea tests, the average statistics include considerably more impact peaks than do the 1/10 highest statistics. Thus, the comparison between measured and computed results are expected to be more reliable. It is interesting to note that a forward movement of LCG from 34 ft. to 38 ft. reduces the impact accelerations by nearly 35 %.

The measured pitch and heave motions and added resistance in waves are not presented in this paper, but are in substantial agreement with results computed by the methods of Reference 3 .

Coursekeeping Stability and Turning Performance

The calm water stability and maneuvering characteristics of the 1/16-scale model with appendages were investigated by means of straight course tests and by rotating arm tests. In both tests, the model was free to heave and pitch, but was restrained in yaw, roll, surge and sway. The restraining forces and moments were measured in a body axis system having its origin at the center-of-gravity.

The straight course tests were made at port yaw angles up to 12° and at port roll angles up to 20°. Also, at zero yaw and roll the effects due to rudder deflection up to 35° were measured.

The rotating arm tests were made at port yaw angles up to 12°, port roll angles up to 20° with the boat making port turns at radii corresponding to 2.5 and 5.0 boat lengths.

The model was tested at a displacement of 120,000 lbs. at an LCG = 34 ft. and at speeds corresponding to 14 and 45 knots. The data obtained during the tests were processed on a digital computer and tables of drag, side force, yaw moment, roll moment, trim and heave were produced as a function of yaw angle and roll angle for each of the radii and speeds investigated. The reduced data were plotted and cross-faired as a function of yaw angle and radius for each of the speeds and roll angles. From these plots, which are not reproduced here, the coefficients needed for stability analysis were determined and are tabulated below. These include the rates of change of side force and yaw moment with yaw angle and radius at zero roll angle.

Hydrodynamic and Inertia Coefficients

Speed	N'_v	N'_r	Y'_v	Y'_r	n'_z	m'_y
14 knots	0.110	-.103	-.316	0.0293	1.00	0.651
45 knots	0.096	-.052	-.258	0.0084	1.00	0.651

where

$n'_z + I'_z = 2\,I_z/\rho B^5$ $\qquad$ $m'_y = m' = 2\,W/\rho g B^3$

$N' = N/qB^3$ $\qquad$ $Y' = Y/qB^2$

$q = \rho U^2/2$ $\qquad$ B = beam = 15.7 ft.

N = yaw moment, ft-lbs. $\qquad$ Y = side force, lbs.

W = 120,000 lbs.

Dynamic Course Stability

Dynamic stability relates to the track of a vessel following a small disturbance in, for example, heading angle when no corrective action is taken (i.e., controls fixed). A ship is said to be dynamically stable when, having suffered a disturbance from an initial straight path, it tends to take up a new straight path. The vessel may perform diminishing oscillations about the new track. The degree of stability is measured by the magnitude of a stability index which is negative if the vessel is stable and vice versa.

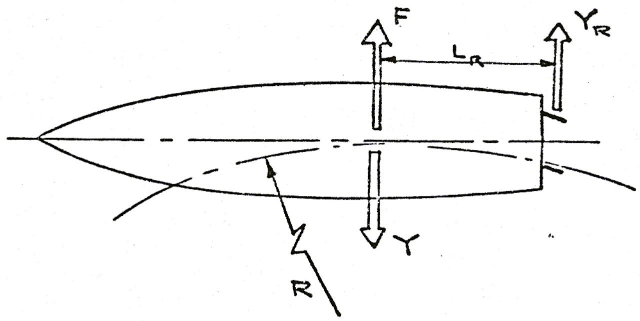
The course stability may be found from linear differential equations governing the craft's motion. The coefficients in these equations may be found from the forces and moments measured during steady state turns, as described above. At both 14 and 45 knots, the craft is statically stable, that is to say that when run at a constant yaw angle the yaw moment tends to reduce the yaw angle. Static stability is measured by the coefficient N'_v and is positive for static stability. However, the degree of static stability is not enough to impair maneuverability. Since it is statically stable, it follows that the craft is also dynamically stable, though oscillatory. These oscillations decay very rapidly, however, being damped to 40 % of the initial disturbance by the time the craft has traveled one boat length. The stability index has been calculated to be

Speed	Stability Index
14 knots	$\sigma = -0.29$
45 knots	$\sigma = -0.22$

Turning Performance

The straight course tests with the rudders deflected showed that the longitudinal position of the center of pressure coincided with the quarter chord point of the mean rudder chord and the vertical location coincided with the depth of the mean rudder chord. Thus, the effect of the rudders can be represented by a force acting at the aerodynamic center of the rudder. The magnitude of the rudder "lift force" was calculated from aerodynamic theory and confirmed by experiment to be represented by a lift curve slope of 0.0373 per degree.

The forces acting on the boat when making a steady turn to port are shown in the following sketch.



The equilibrium equations in side force and yaw moment are

$$Y = F + Y_R \quad \text{and} \quad N = \ell_R Y_R$$

which can be combined into

$$F = Y - N/\ell_R \tag{1}$$

The component of centrifugal force in the x, y plane when the craft has yaw and roll angle of β and ϕ is

$$F = (w/g)\ (V^2/R) \cos\beta \cos\varphi$$

For each speed and radius, the quantity $Y - N/\ell_R$ is plotted as a function of yaw angle for each roll angle and the yaw angles necessary to satisfy Equation (1) are found. At these intersections, the roll moment due to the rudder is found from

$$K_R = (Y - F)\ d_R$$

where d_R is the distance of the rudder mean chord below the craft VCG. This roll moment is superimposed on a plot of roll moment versus roll angle to give the roll angles at equilibrium.

The results of this calculation show that the craft turning diameter is less than 15 boat lengths and that it will roll inboard during turns.

CONCLUSIONS

A quantitative design procedure is described to determine the principal hull dimensions for planing craft intended to satisfy prescribed operational conditions. The method.is applied to establish a hull form required to operate at high speeds in moderate sea states. Principal design features of this craft are described. Extensive model tests were conducted to predict the SHP, EHP, seakeeping, course-keeping stability and turning characteristics of the design. Some of these model test results are presented.

ACKNOWLEDGEMENTS

The authors would like to express their appreciation to Mr. Joseph G. Koelbel, Jr. and to Mr. G. Gordon Sammis for their invaluable assistance in all phases of the development of this new planing craft.

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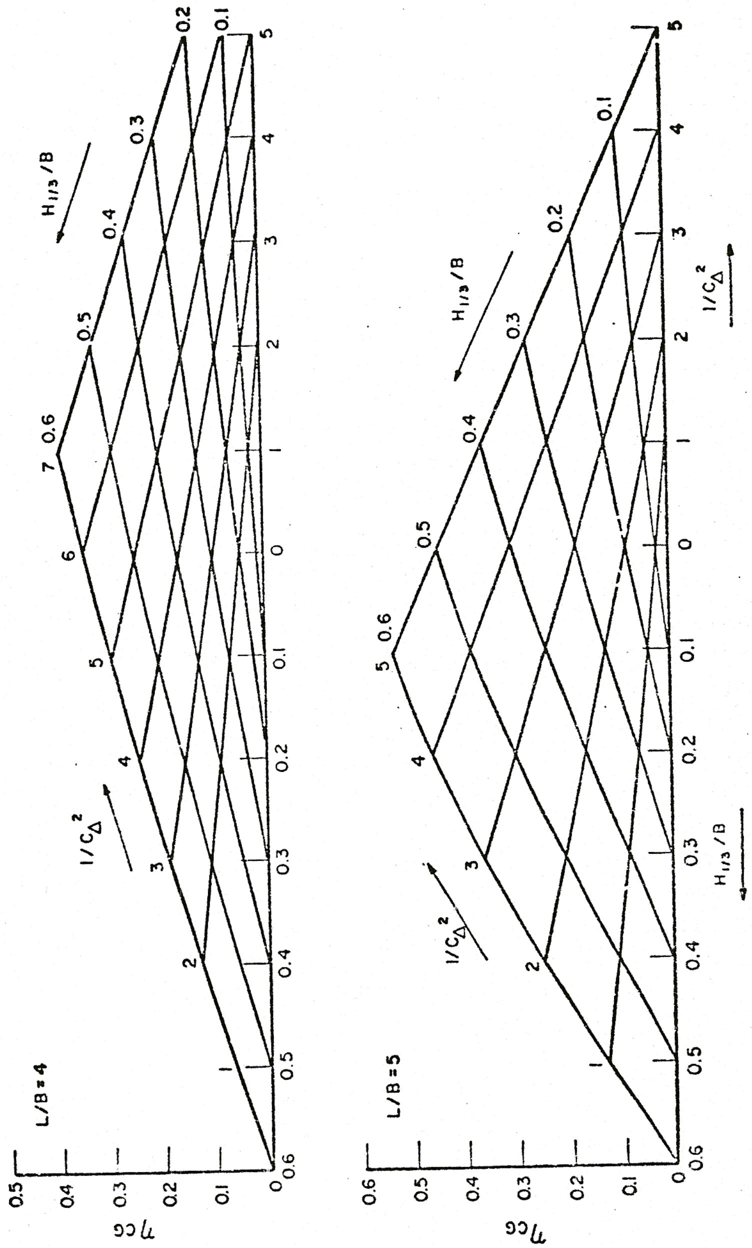

Figure 1. Average CG acceleration at $V/\sqrt{L} = 2$
($\tau = 4°$, $\beta = 20°$)

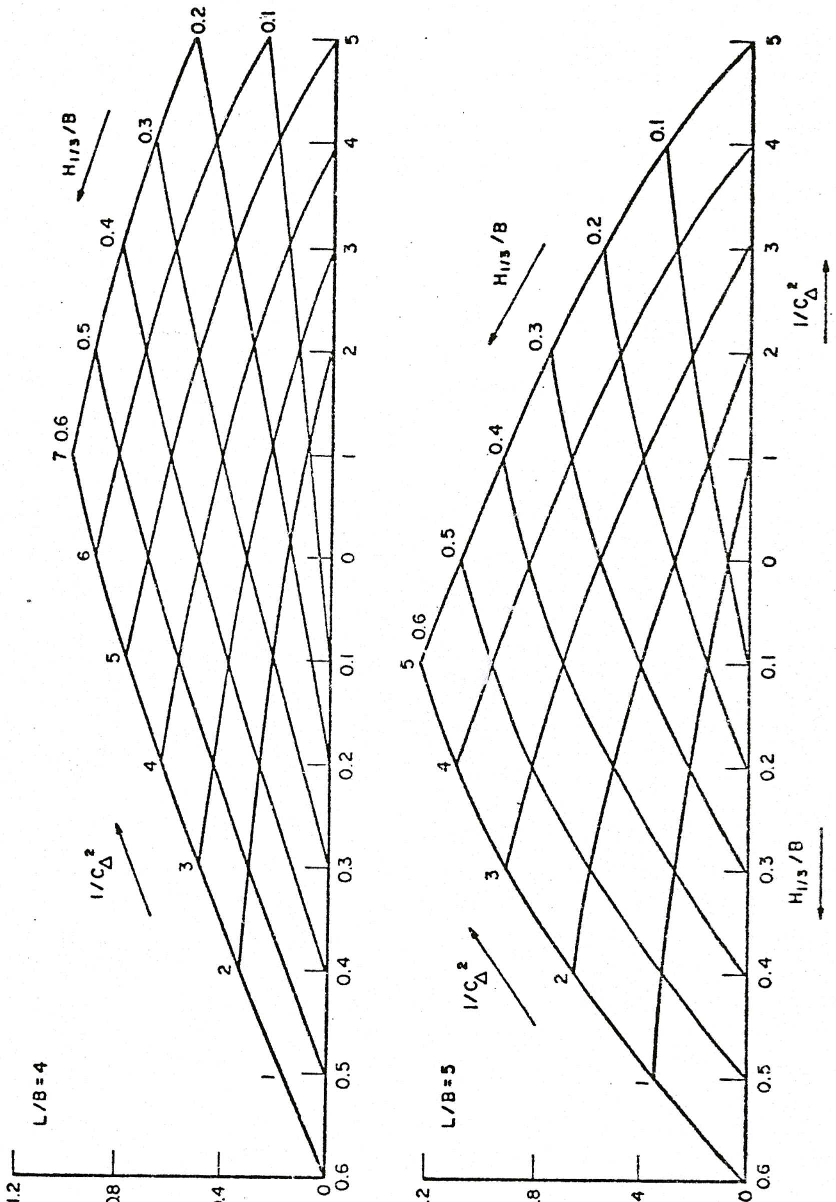

Figure 2. Average CG acceleration at $V/\sqrt{L} = 4$
($\tau = 4°$, $\beta = 20°$)

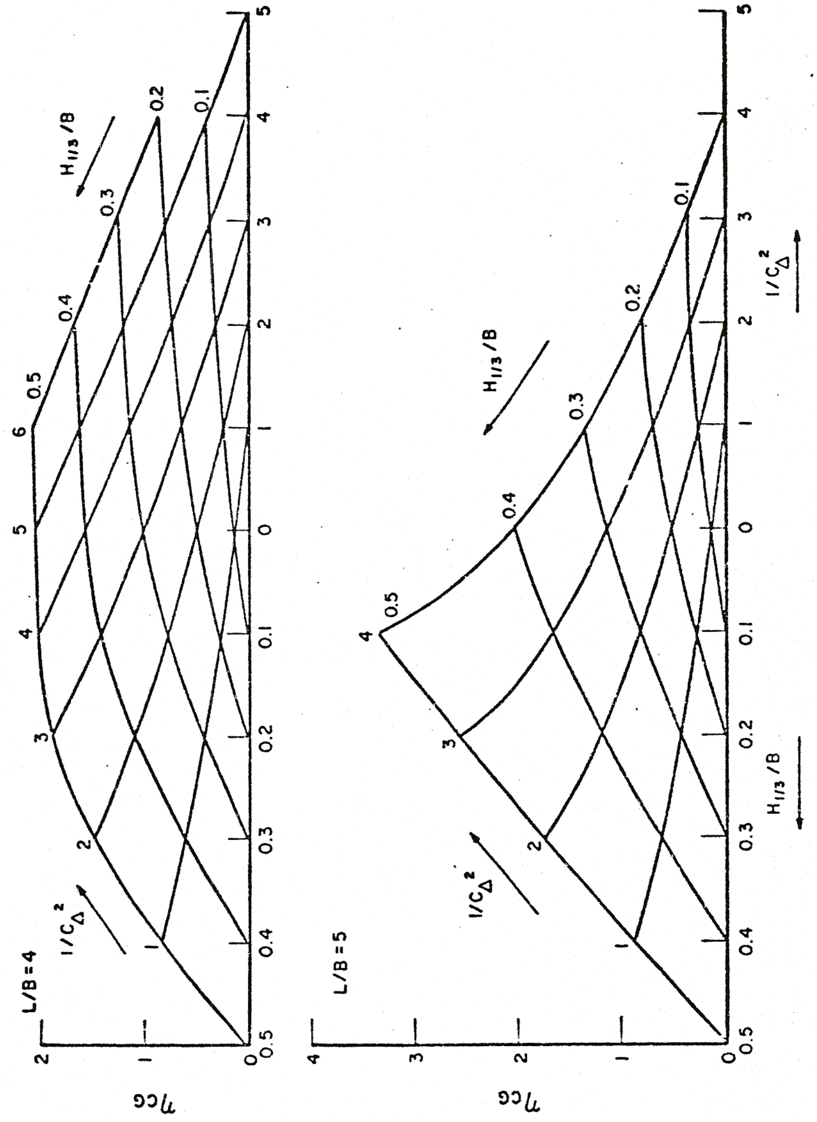

Figure 3. Average CG acceleration at $V/\sqrt{L} = 6$
($\tau = 4°$, $\beta = 20°$)

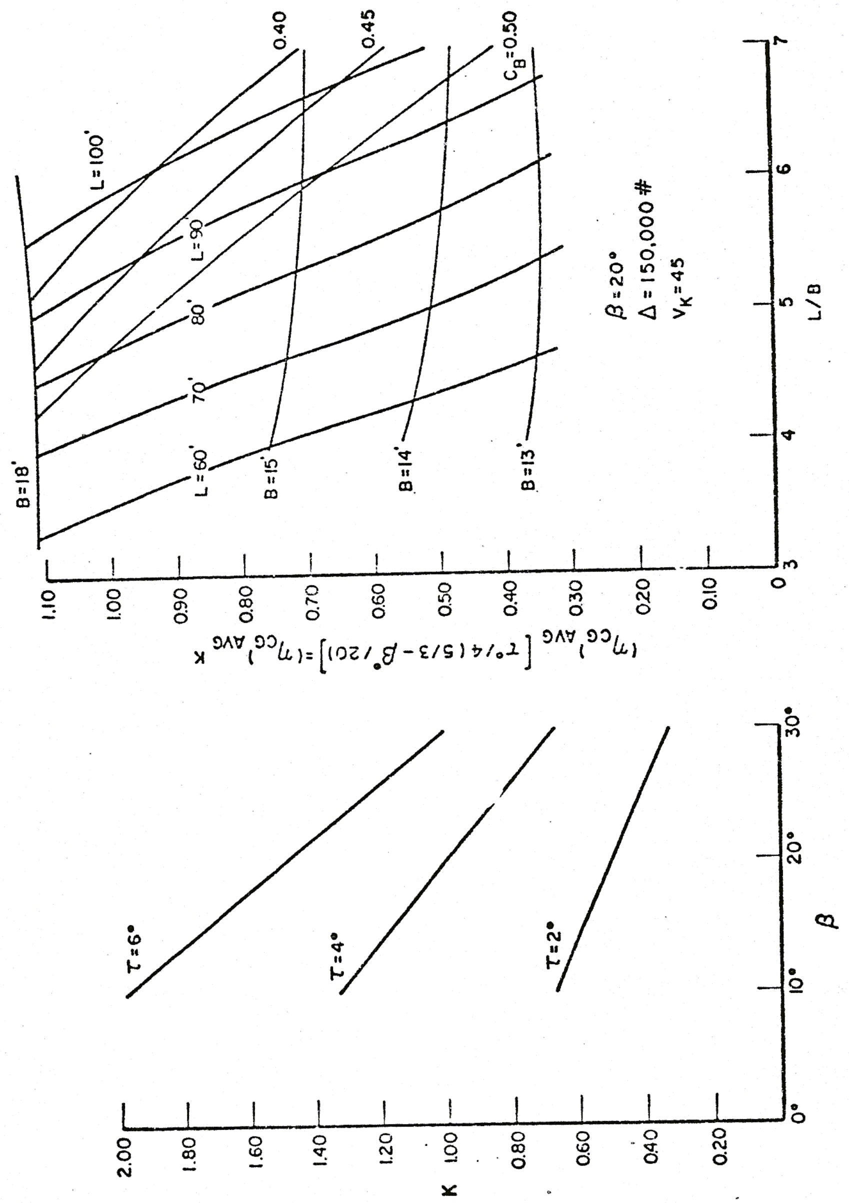

Figure 4. Average CG acceleration for various values of beam

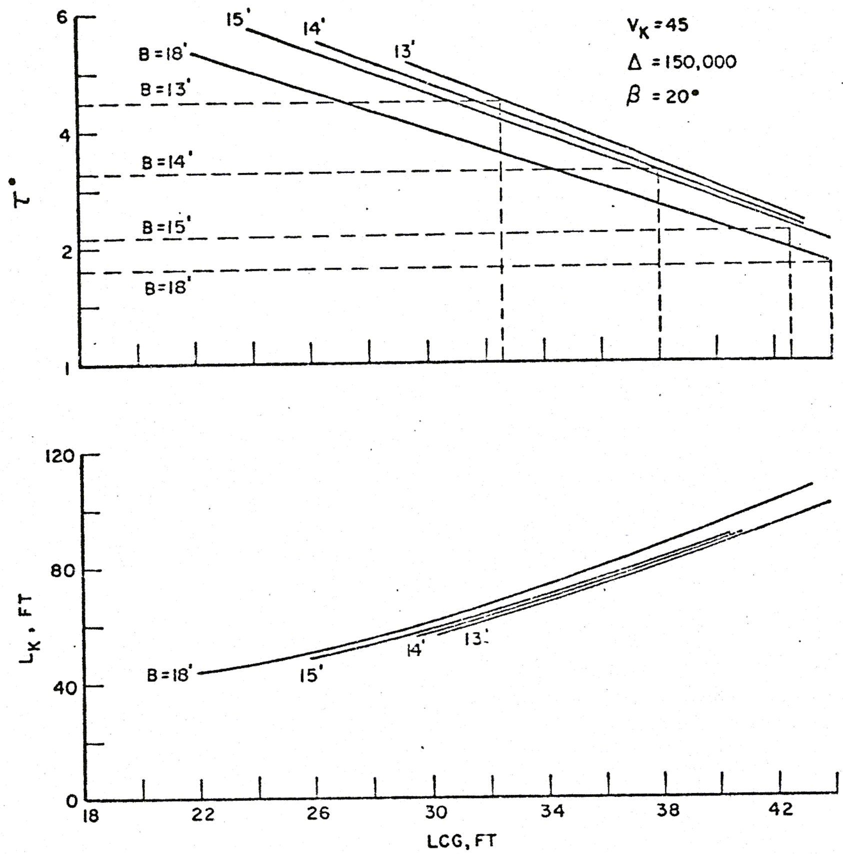

Figure 5. Equilibrium conditions in smooth water for various values of beam and LCG

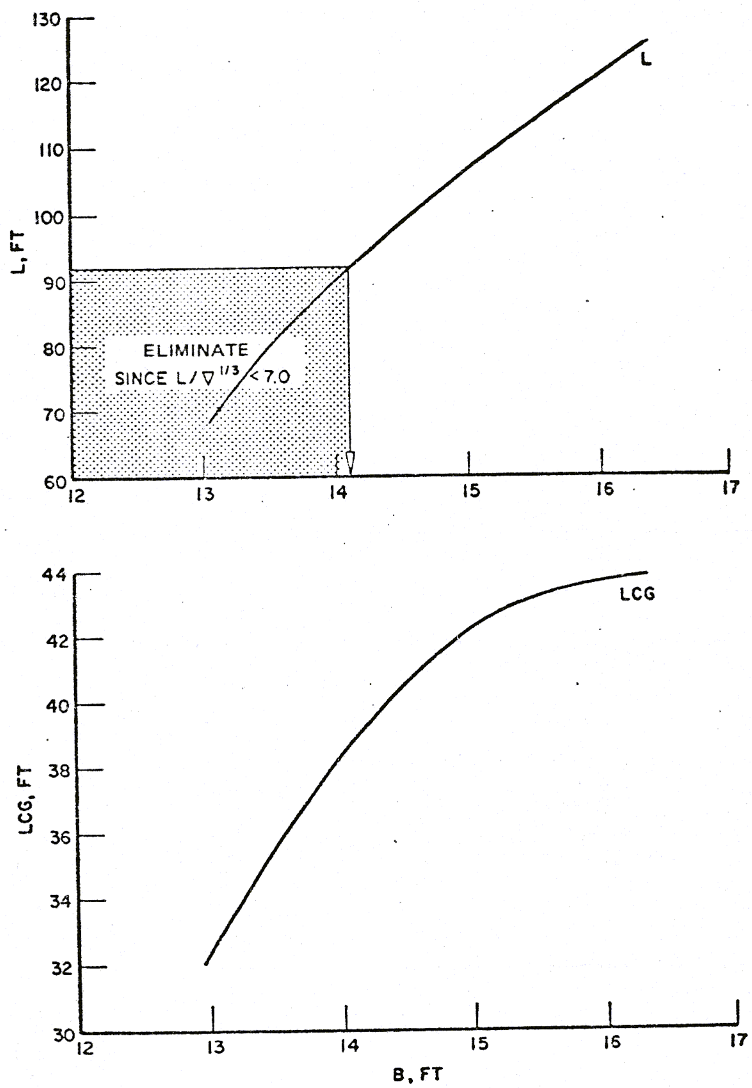

Figure 6. Relation between beam and length to achieve $(\eta_{CG})_{avg} = 0.4$ g $V_K = 45$ knot $\Delta = 150,000$ lb.

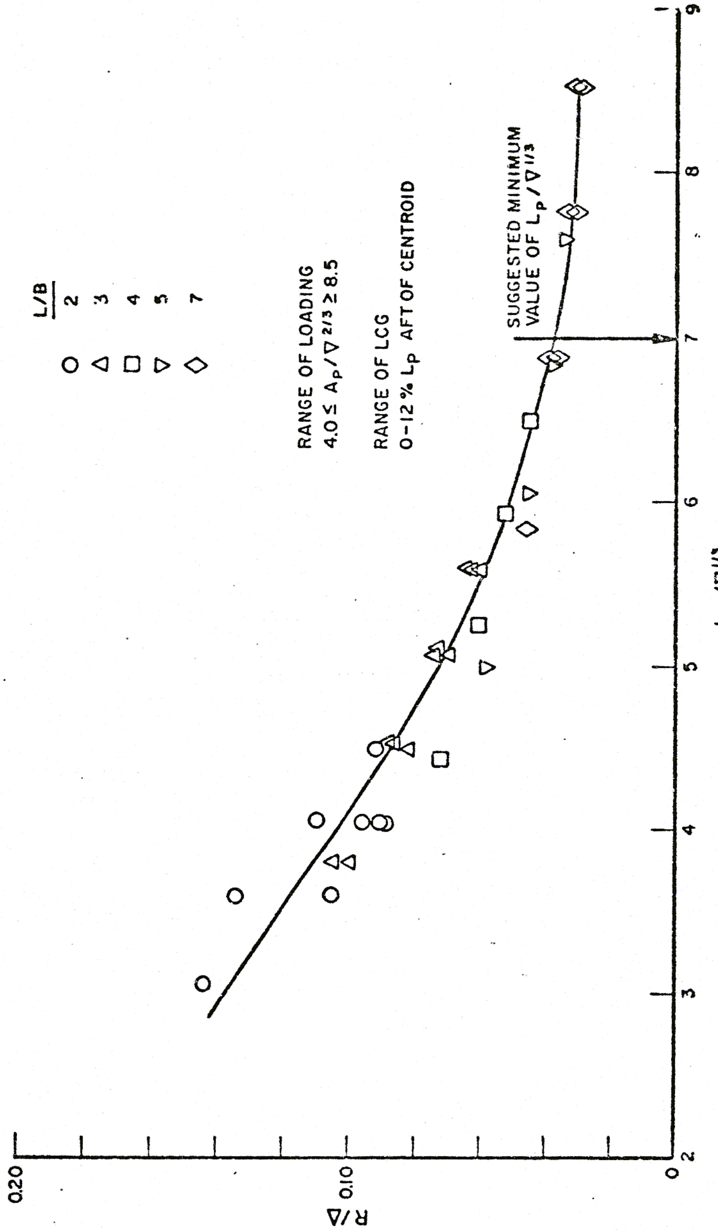

Figure 7. R/Δ for F_∇ = 1.0 series 62 hulls

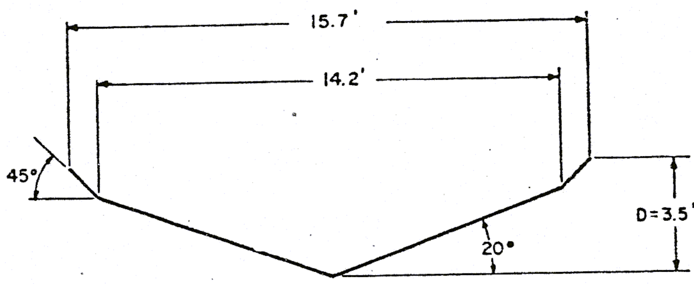

Figure 8. Double chine hull form selected for design

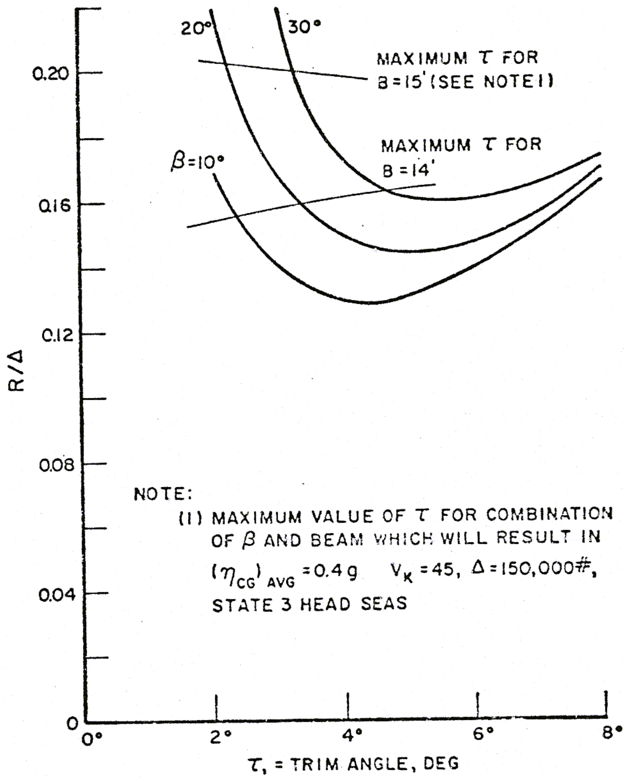

Figure 9. Variation of R/Δ for various τ and β

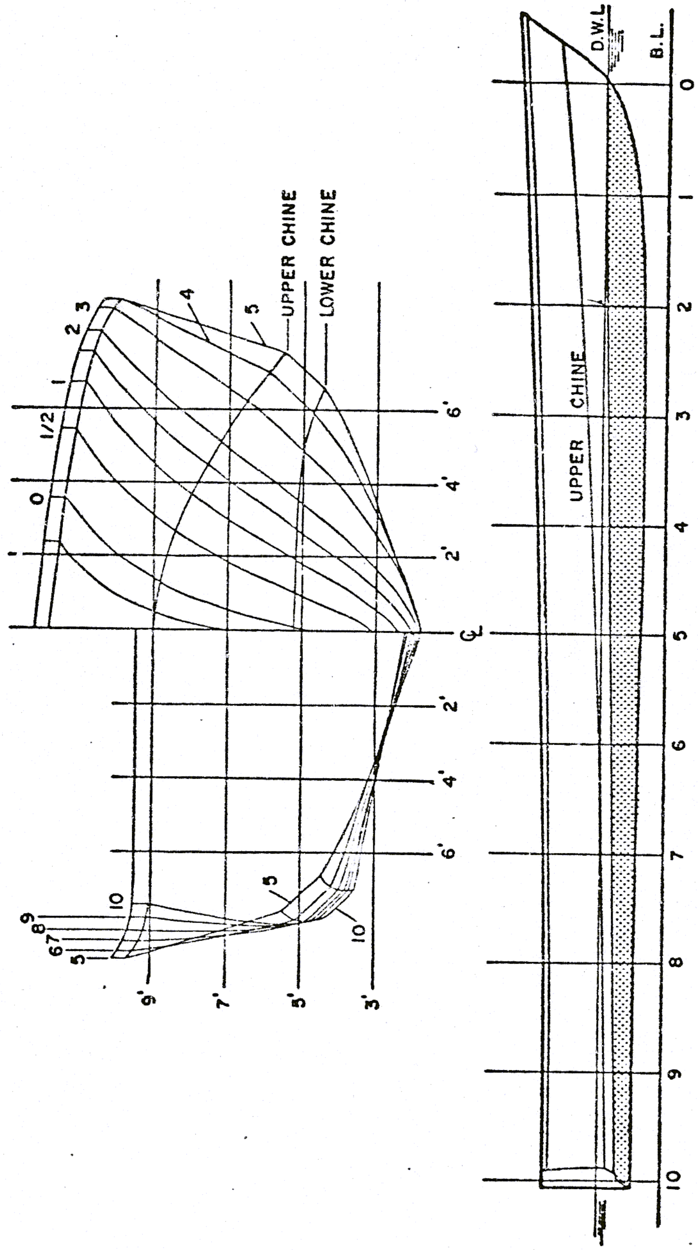

Figure 10. Line plan of final configuration

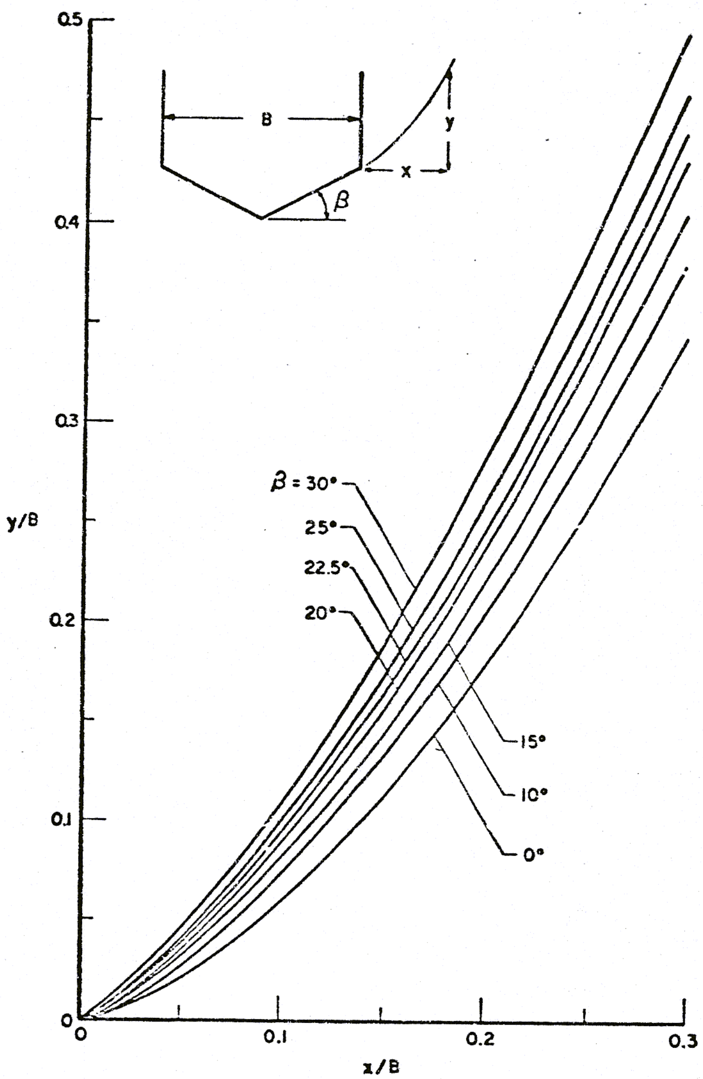

Figure 11. Shape of free streamline for immersed V-bottom

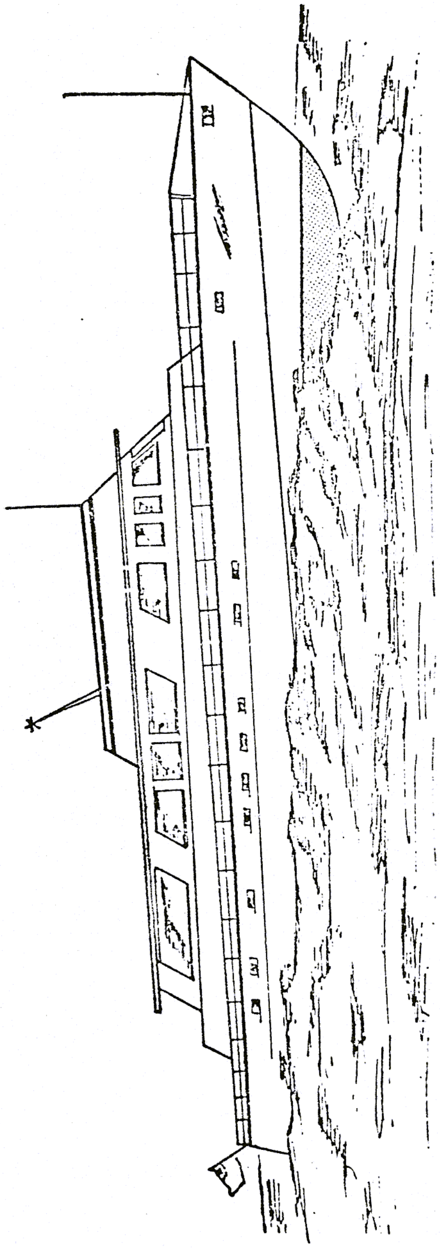

Figure 12. Final configuration

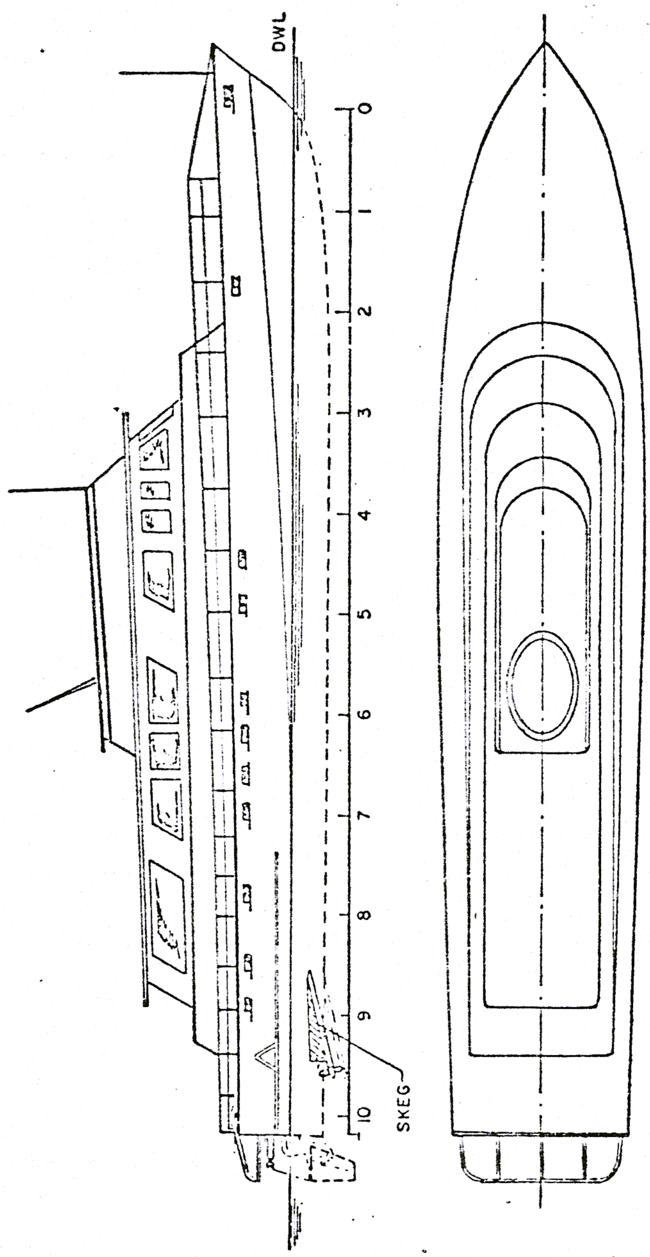

Figure 13. Outboard profile and plan view

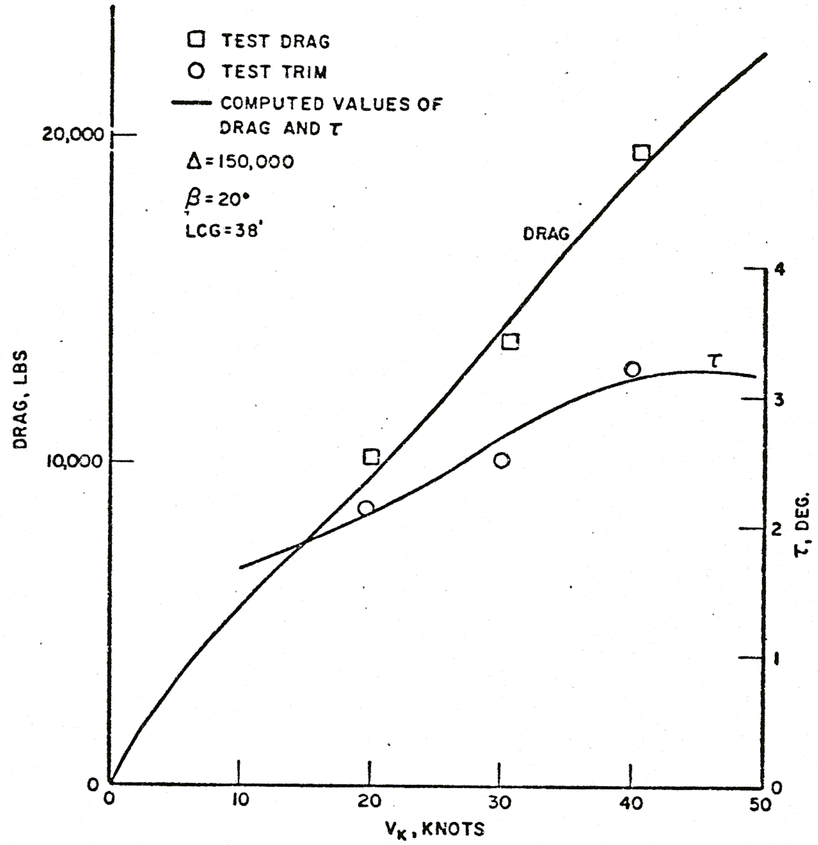

Figure 14. Computed and measured values of resistance and trim in smooth water

DISCUSSION

Manley Saint-Denis
University of Hawai
Honolulu, U.S.A.

I am very happy that seakeeping has been treated in this paper as the pre-eminent factor in the design of planing craft, for to the present, seakeeping has been introduced in the design of small craft hardly at all. Indeed it has been ignored except for raising the chine at the bow and narrowing the beam. And then the designers have simply put their trust in God, hoping that he would be kind to them and to the sea-beaten crews that would man in the open sea the craft they had designed. Therefore, I am grateful that something is being done because, waves being independent of the size of the vehicle that ventures over the surface of the sea, I suppose it is not a profound revelation to state that the smaller the craft the more she suffers. For a large vessel, even a heavy sea can be only an inconvenience, but for a small craft even a modest sea can lead to a very miserable experience. Therefore, starting the design of planing craft by considering the sea behaviour as the very first step in the process is the correct way to proceed, and I am glad to see the authors have done just this.

My second point relates to the authors' conclusion that if you narrow the beam, reduce the trim angle and up the dead-rise, things will be better rather than worse ; and while the exposition of the paper itself gives quite some insight into the sensivity of how impact, sea behaviour and other effects are related to the design features, the designers do not unfortunately go further into the matter.

I should like to point out that if all the authors wanted to do was to show how to develop a design to fulfil some very rigid specifications, such as the inflexible ones they have stated, the design process could be shortened considerably. In fact, one could develop a simple computer program that would yield an almost instantaneous answer, for the line of logic is simple and unambiguous in such a case. However, the point I should like to raise is that the specifications are not always quite as rigid as the authors have listed them, that indeed one has to play with them somewhat, giving up somewhat little here to gain somewhat more elsewhere : for example, take the problem of the transverse metacentric height, the metacentric height is reduce by decreasing the beam, but this step also lessens the impact force, and so one might be better off. Is it worthwhile ? How much is it worthwhile ? The answer is not easy, of course, but such type of problems - so called trade-off problems - are not treated. The computer technique is in hand for coping very nicely with such problems and it is feasible to set up a programme that would yield a design, by satisfying in an optimum manner an imposed set of trade-off criteria. Therefore I humbly suggest that the authors, having been successful so far, should continue their quest for further success by applying themselves to this step.

DISCUSSION

Reuven Leopold
U.S. Navy. Naval Ship Engineering Center
Hyattsville, Maryland, U.S.A.

The importance of high endurance, and hence low resistance, at low -about 12 knots- cruising speeds is emphasised. Certainly a broad transom will have an adverse effect on resistance at these low speeds. This point is not discussed in the paper. Was the possibility of incorporating a method of trim control in a design to reduce or eliminate transom immersion at low cruising speeds considered in this design methodology ?

REPLY TO DISCUSSION

Daniel Savitsky
Stevens Institute of Technology
Hoboken, New Jersey, U.S.A.

Yes. Deliberate trim control was considered, but it is not presented here.

DISCUSSION

Reuven Leopold
U.S. Navy. Naval Ship Engineering Center
Hyattsville, Maryland, U.S.A.

The requirement to achieve 45 knots in state 3 seas is emphasized. However the effects of air drag and sea state on the power required to propel the craft at 45 knots is not discussed. Assuming that a state 3 sea is generated by a 15 knot wind, how much is the calm water 0 kn. relative wind resistance of the craft at 45 knots increased by the presence of a 60 knot relative wind and a state 3 sea ?

REPLY TO DISCUSSION

Daniel Savitsky
Stevens Institute of Technology
Hoboken, New Jersey, U.S.A.

It is important to emphasize that the present paper presents a methodology for rational design of planing hulls. The method has been applied to a particular set of design parameters to demonstrate its validity. It now remains to use this technique to develop optimum designs as suggested by Dr. Saint-Denis and Dr. Leopold.

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