DESIGN DATA FOR HIGH SPEED DISPLACEMENT-TYPE HULLS AND A COMPARISON WITH HYDROFOIL CRAFT

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This paper is intended as a preliminary design guide and as a prelude to a more comprehensive research programme on high speed displacement-type hulls.
It includes a short survey of available NPL data on the resistance, propulsion, and running performance of this type of hull and a brief discussion on the shallow water effect.
The research programme planned at NPL is outlined and a description given of the first of a series of torsionmeters designed at NPL specially for use on trials of small high speed craft.
Finally some estimates are given of the power requirements of a suggested form of mixed craft in which the total weight is supported partly by the buoyancy of the hull and partly by fully submerged hydrofoils.

INTRODUCTION

Many small boats, such as patrol launches, work boats, and pleasure craft, operate at speed/length ratios of 1.2 to 3.5 (or Froude number F_n about 0.4 to 1.2). These boats traditionally have round-bilge displacement hulls, but there is very little published information about them to help their designers. While systematic series of model experiments have been made for most types of larger vessels, the relatively high cost of model tests has made it difficult to persuade owners and builders of small high speed boats to commission model experiments for new designs. Consequently there has long been a tendency to base hull designs for small boats on previous forms presumed to have been successful, without any proper basis for assessment of performance.

This paper is intended as a preliminary design guide and as a prelude to a more comprehensive research programme on this type of hull form which is being carried out at the Ship Hydrodynamics Laboratory of Ship Division, NPL; the large high-speed towing tank there will enable experiments to be made in better conditions than previously, including systematic measurements in head and following seas. The present paper is a short survey of available NPL data on the following aspects of the design of high speed displacement-type hulls:

1. Resistance of Round-Bilge Forms

   (a) Average results for models
   (b) Effect of beam/draft ratio and LCB position

(c) Appendage resistance
(d) Effect of changes in scale.

2. Propulsion

(a) Components of propulsive efficiency
(b) Ship-model comparison.

3. Running Performance

(a) Effect of spray strips
(b) Design of a round-bilge form for good seakeeping.

4. Shallow Water Effects.

This account of the information available at present is followed by an outline of the programme of research now commencing at NPL, and by a description (in the Appendix) of the first torsionmeter specially designed at NPL for power measurements on small boats.

High speed hydrofoil boats are not uncommon, and are being actively developed at present. However, the use of hydrofoils to give partial lift to heavy displacement-type high speed boats operating at speeds less than the planing speed is not well-established. Some preliminary estimates of the value of such devices are given in this paper.

RESISTANCE OF ROUND-BILGE FORMS

Normal round-bilge hull forms designed to operate at speed/length ratios between 1.2 and 3.5 (F_n 0.4 to 1.2 approximately) have the following distinctive characteristics:

1. The afterbody has a flat or convex bottom having a dihedral angle (or rise of bottom) up to 45 degrees which runs smoothly into almost vertical sides with a rounded-bilge shape, not a sharp-angled chine.

2. The buttock lines are usually straight and almost parallel to the centre buttock line which usually runs aft at 10 to 15 degrees to the waterline.

3. The stern has a flat or rounded transom, except for low speed/length ratio forms which may have a cruiser-type stern.

4. The bow above water is designed for minimum interference with the bow wave. Any chine in the forebody is high in profile so as to reduce the breadth at the deck line, and will not extend as far aft as midships.

High speed round-bilge forms incorporate an unusually large number of design parameters. Besides such basic parameters as beam/draft and length/beam ratios, block and prismatic coefficients, features such as depth of transom and angle of rise of afterbuttocks may have considerable effects on resistance and propulsion characteristics.

Average Results for Models

In a first attempt to examine the effects of these design parameters, the measured model resistances for a varied group of about 30 round-bilge forms were plotted as in Figs. 1-11. The resistance coefficient Ⓒ for a form of length $L = 100$ feet is throughout given in terms of the displacement/length ratio $\Delta/(0.01L)^3$ or, what is the same thing for $L = 100$ feet, the displacement Δ in tons. Figures 1-6 give the resistance coefficients for a series of speed/length ratios $V/\sqrt{L}$ from 1.7 to 4.0, with the specific value of the beam/draft ratio B/d shown for each form. These diagrams show no clear evidence of any systematic variation in resistance with B/d at any point in the speed range. In Figs. 8-11 the same results at $V/\sqrt{L} = 1.7$ are replotted, with the specific values of block coefficient, angle of entrance on waterline, position of LCB and length/beam ratio shown in turn. Again there is no clear evidence of any predominant form parameter.

If resistance data were available for a larger number of hull forms, say about 150, it should be possible to determine the influence of each of the principal form parameters by carrying out a multilinear regression analysis, using a high speed computer. With the limited data presently available this is not possible, and as a preliminary step mean lines have been drawn in each of Figs. 1-6. In addition, lines representing 5 and 10 percent deviation from these mean lines have been drawn. These show that most of the results lie within 5 percent of the mean lines. These mean lines are replotted in Fig. 7 to give, for ships of length 100 feet, average resistance coefficients for speed/length ratios from 1.4 to 3.5 for a range of displacements from 50 to 200 tons. It is suggested that for preliminary estimates these average values can be used with discretion to give the resistance generally within 5 percent for good standard hull forms.

Effect of Beam/Draft Ratio and LCB Position

The data in Figs. 1-11 indicate that it is probably unwise to assume the predominance of one parameter, say B/d, as has been done previously [1], and to ignore other form parameters. However, an attempt has been made to assess the effect of beam/draft ratio by comparing resistance coefficients for two models, each run at two displacements, which had different B/d values but were otherwise similar. The comparison is shown in Fig. 12 and Table 1, and suggests that at $V/\sqrt{L} = 2.5$ resistance varies approximately as the cube root of the beam/draft ratio but at $V/\sqrt{L} = 3.0$ the difference is much smaller. A similar comparison for two forms having a chine instead of a round bilge, and a fairly high chine line forward, given in Fig. 13, showed a much smaller variation with beam/draft ratio. Although these data are inadequate, they have been used to provide preliminary guidance on the effect of beam/draft ratio, as shown in Fig. 14.

Table 1
Comparison of Resistance Coefficients for Two 100-Foot Round-Bilge Models with Different B/d Values

Form	B/d	Δ (tons)	$\frac{V}{\sqrt{L}}$	Ⓒ 100′		$(B-A)/A$ (percent)
				Form A	Form B	
A	2.92	100	2.5	1.70	1.88	+11
B	3.94		3.0	1.44	1.48	+3
$B-A$	1.02	80	2.5	1.55	1.72	+11
$(B-A)/A$	0.35		3.0	1.28	1.33	+4

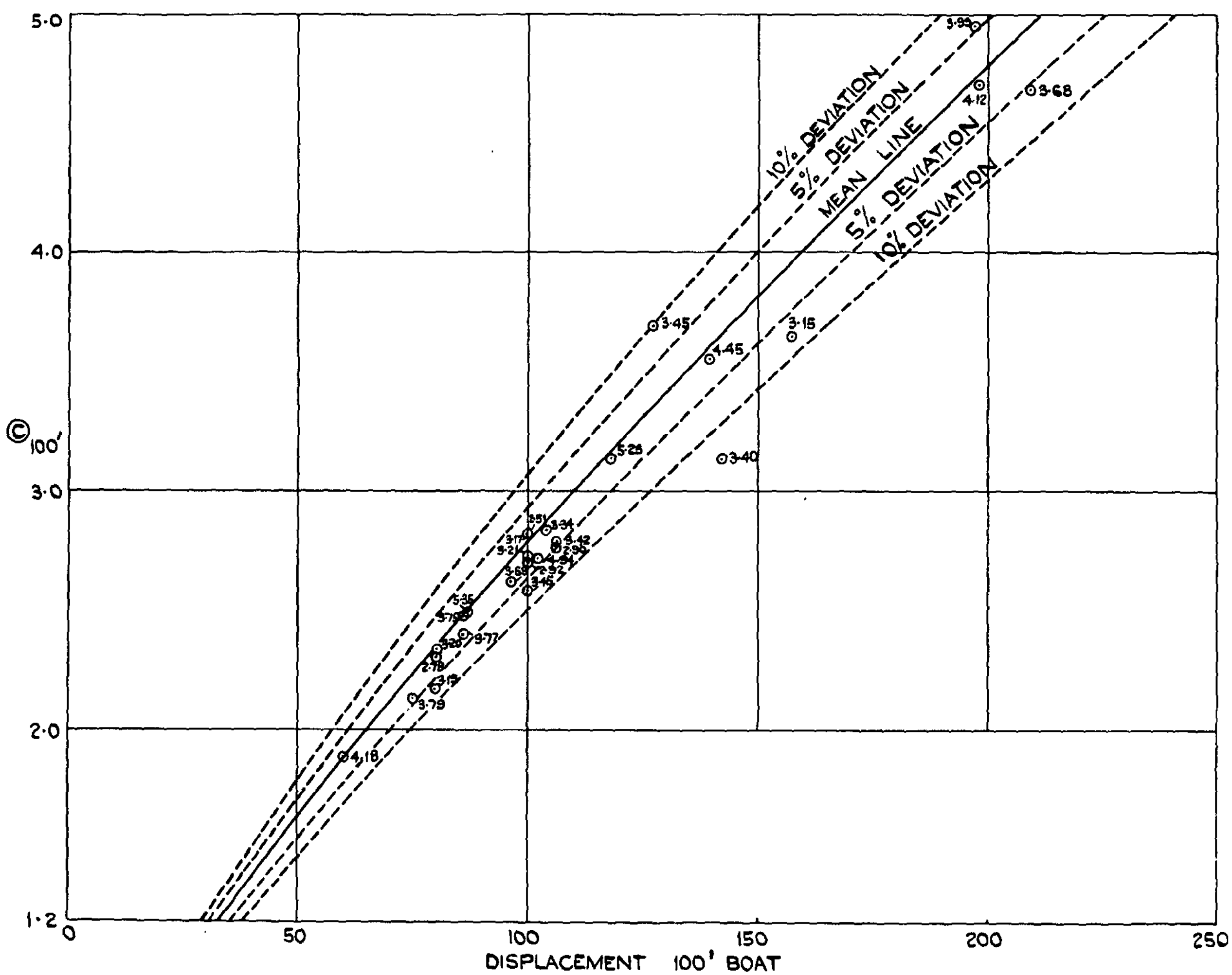

Fig. 1. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 1.70$; B/d values inserted

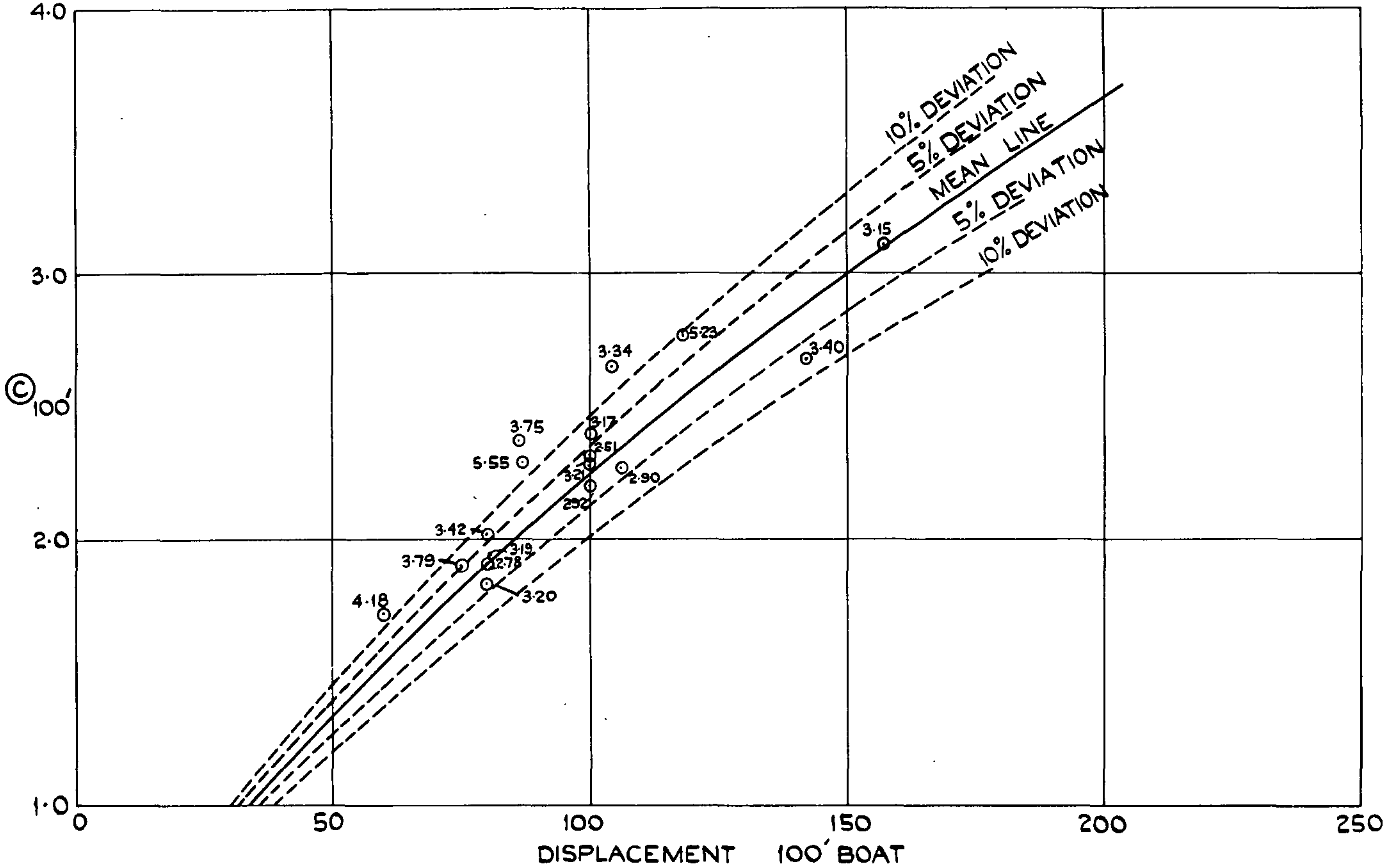

Fig. 2. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 2.10$; B/d values inserted

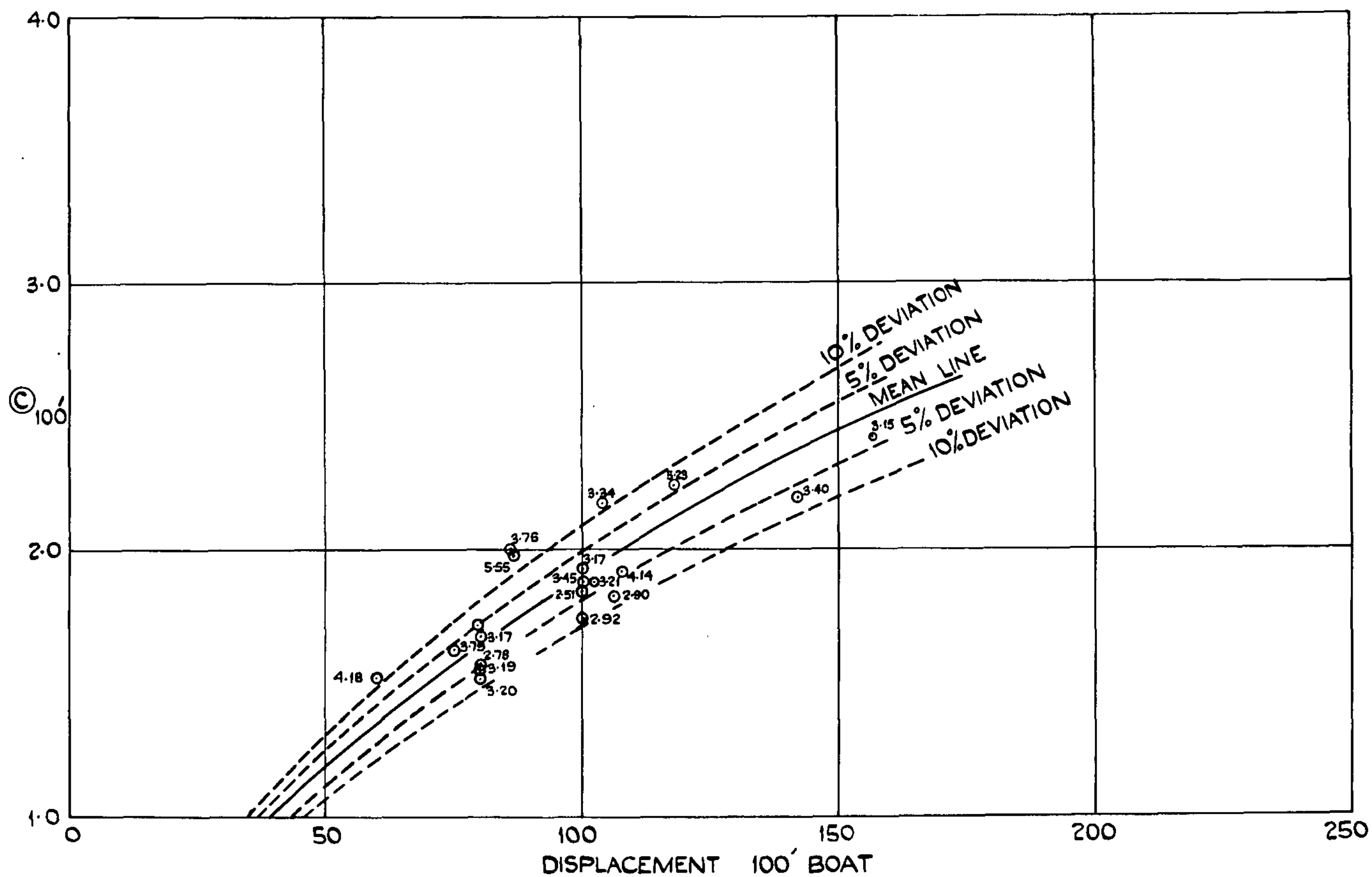

Fig. 3. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 2.50$; B/d values inserted

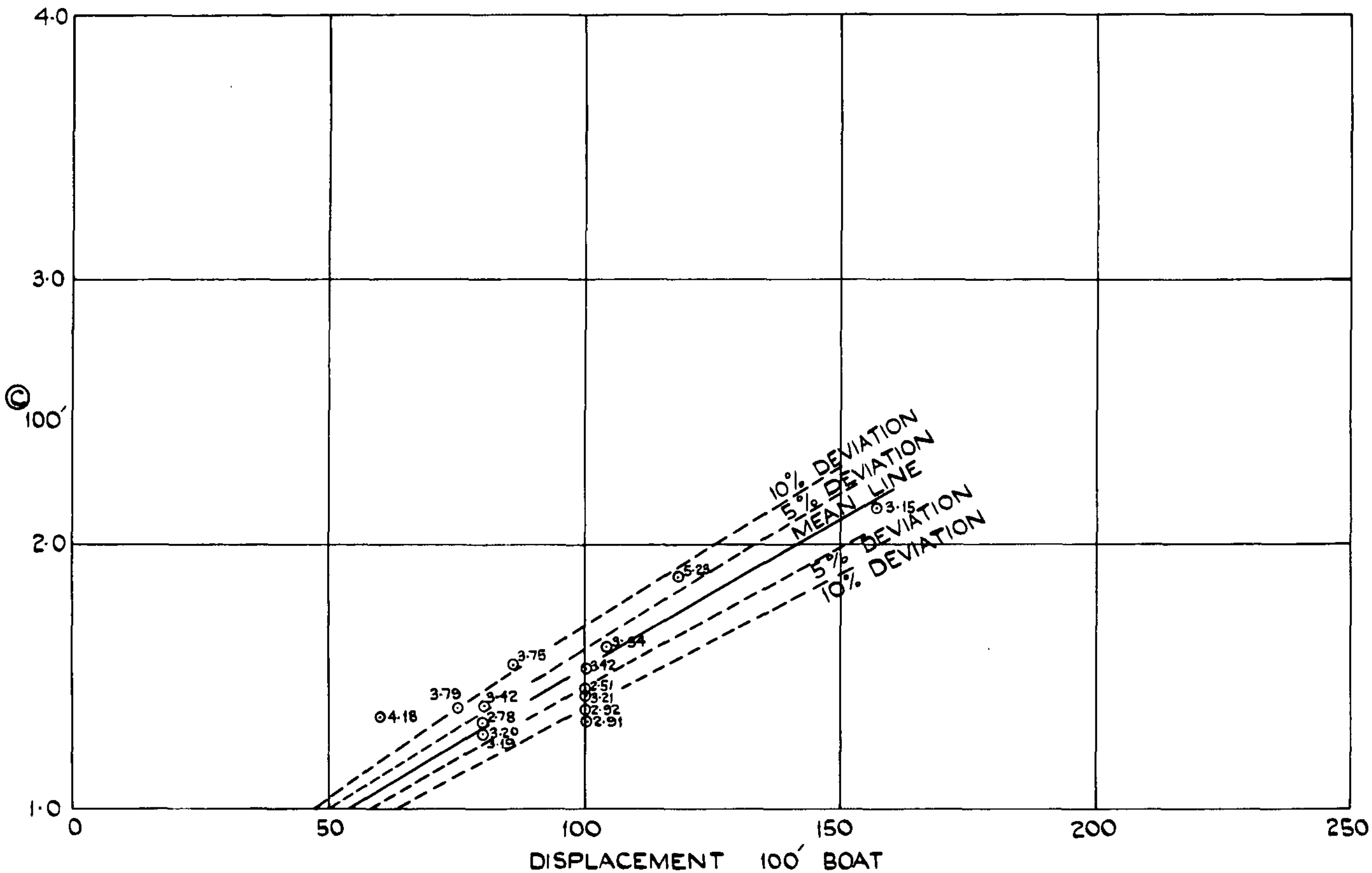

Fig. 4. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 3.0$; B/d values inserted

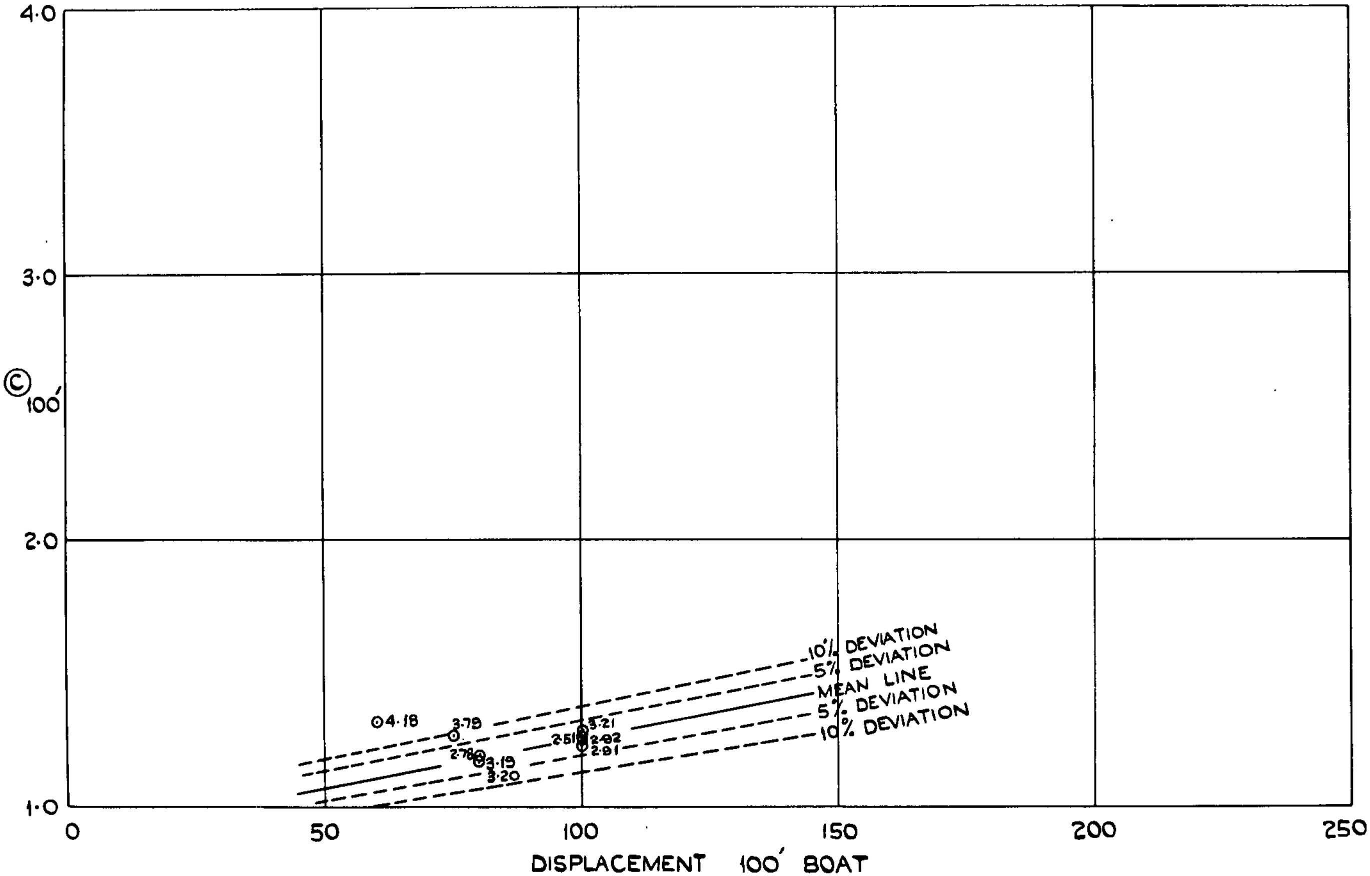

Fig. 5. Resistance coefficient Ⓒ for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 3.50$; B/d values inserted

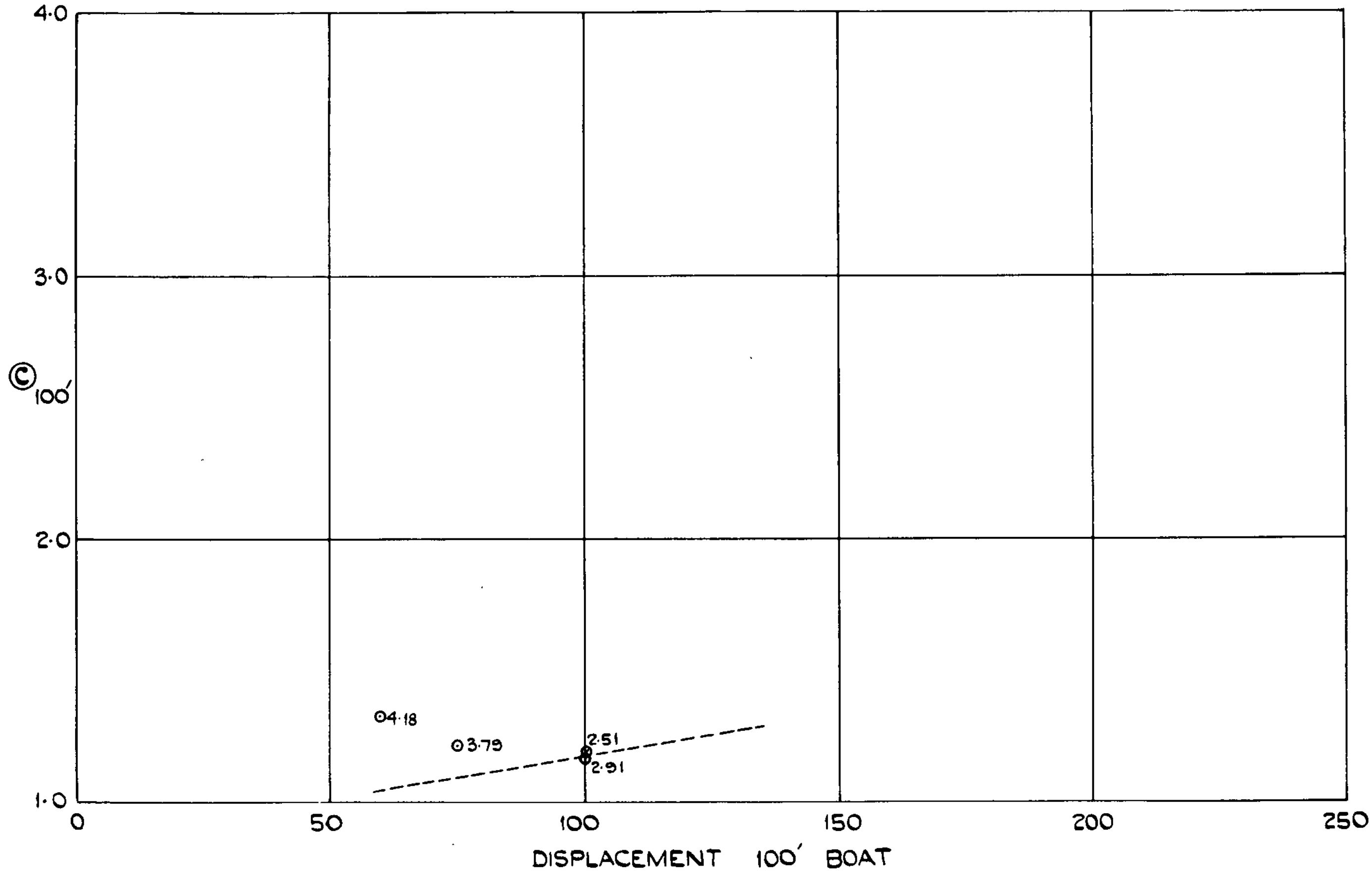

Fig. 6. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 4.0$; B/d values inserted

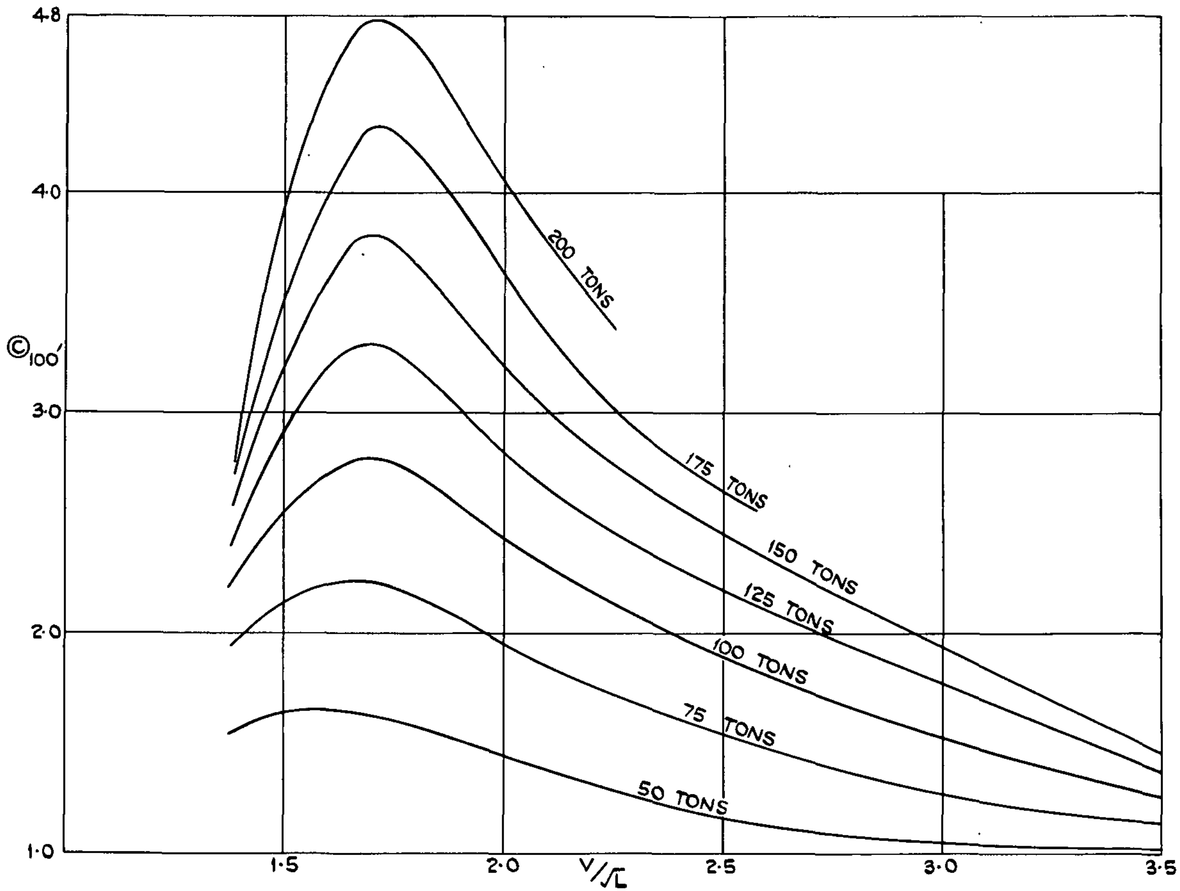

Fig. 7. Averaged values of the resistance coefficient © for a 100-foot round-bilge boat versus $V/\sqrt{L}$; displacement values inserted

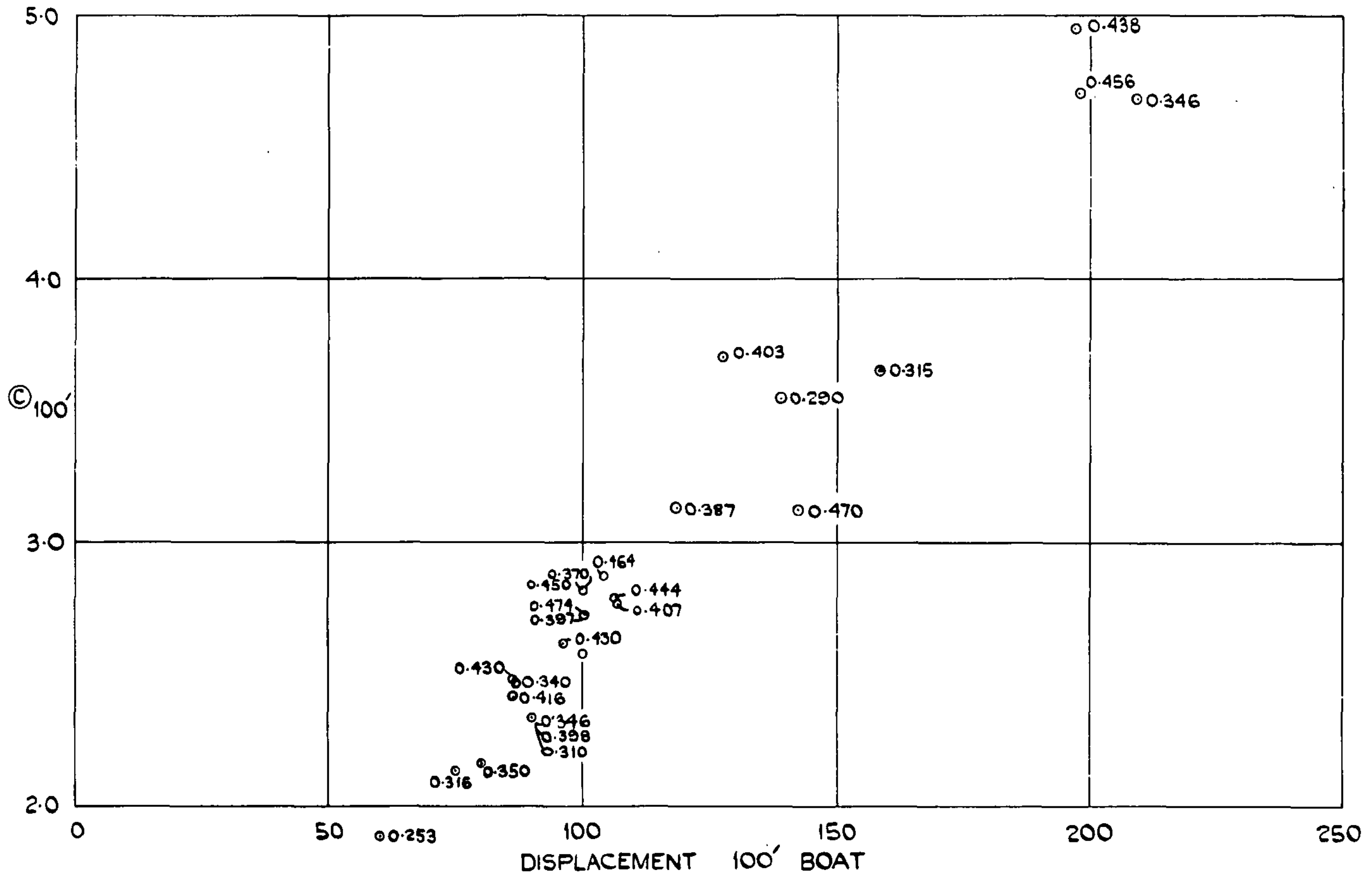

Fig. 8. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 1.7$; block coefficient values inserted

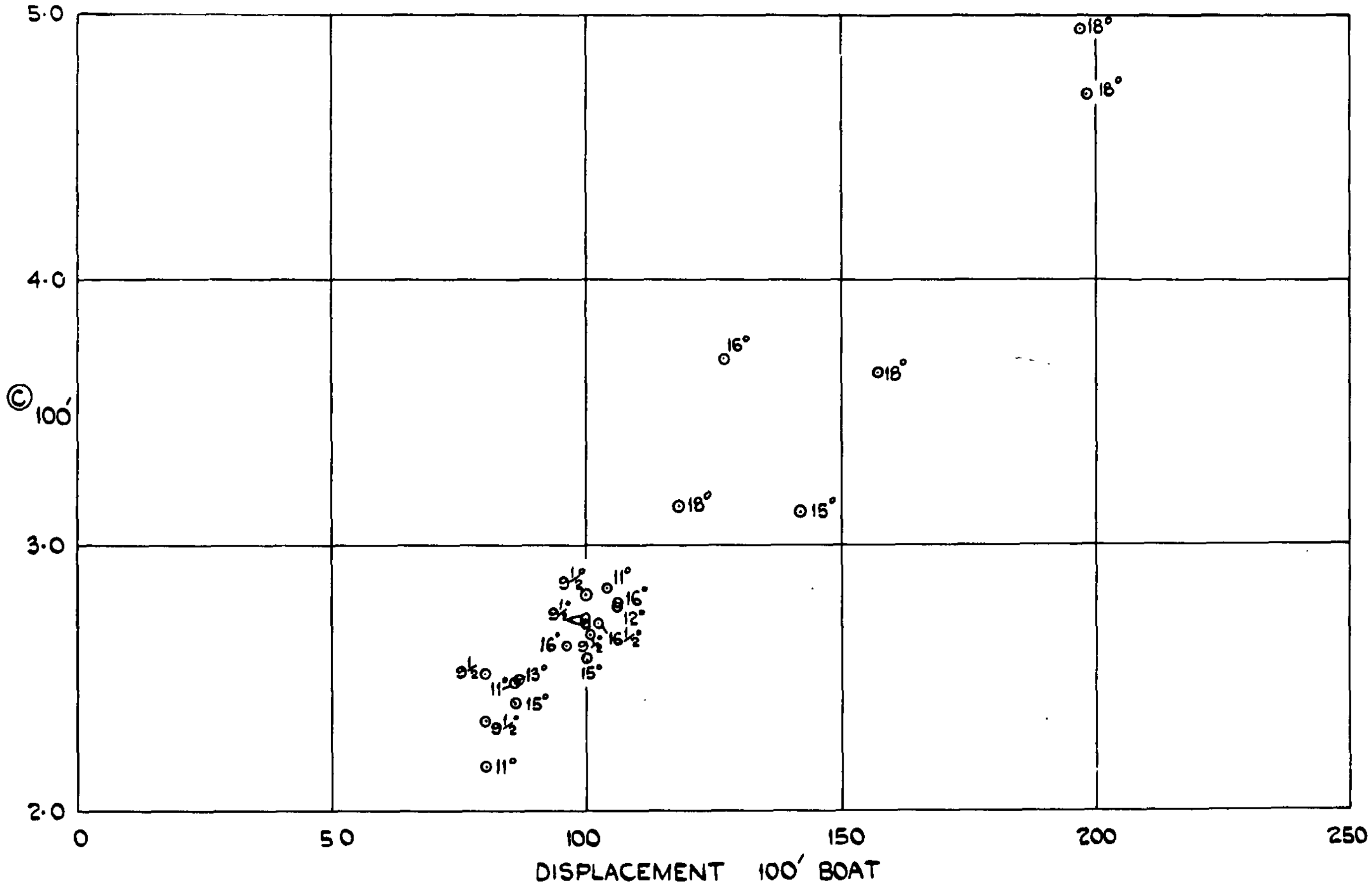

Fig. 9. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 1.7$; half angle of entrance values inserted

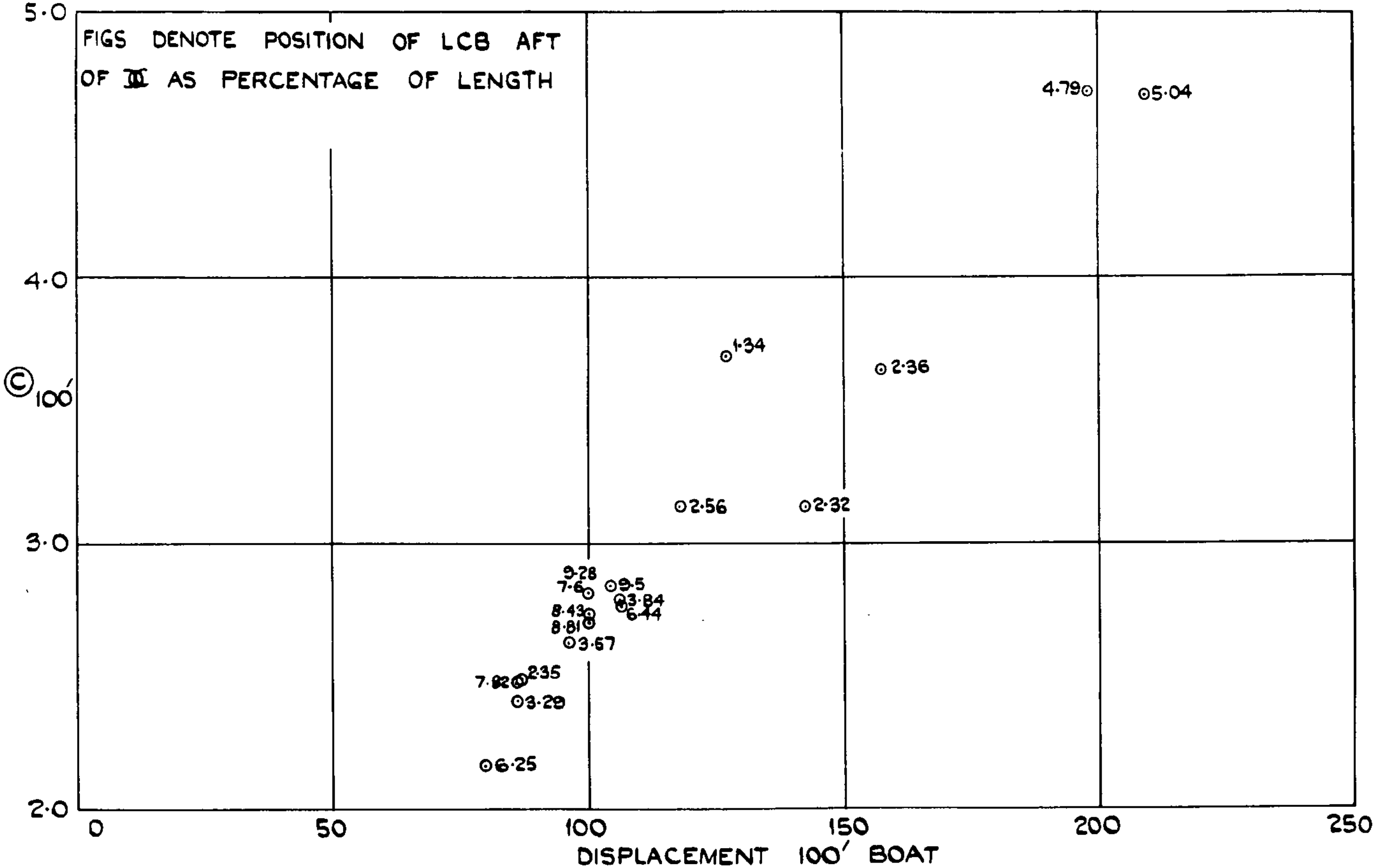

Fig. 10. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 1.7$; longitudinal centre of buoyancy values inserted

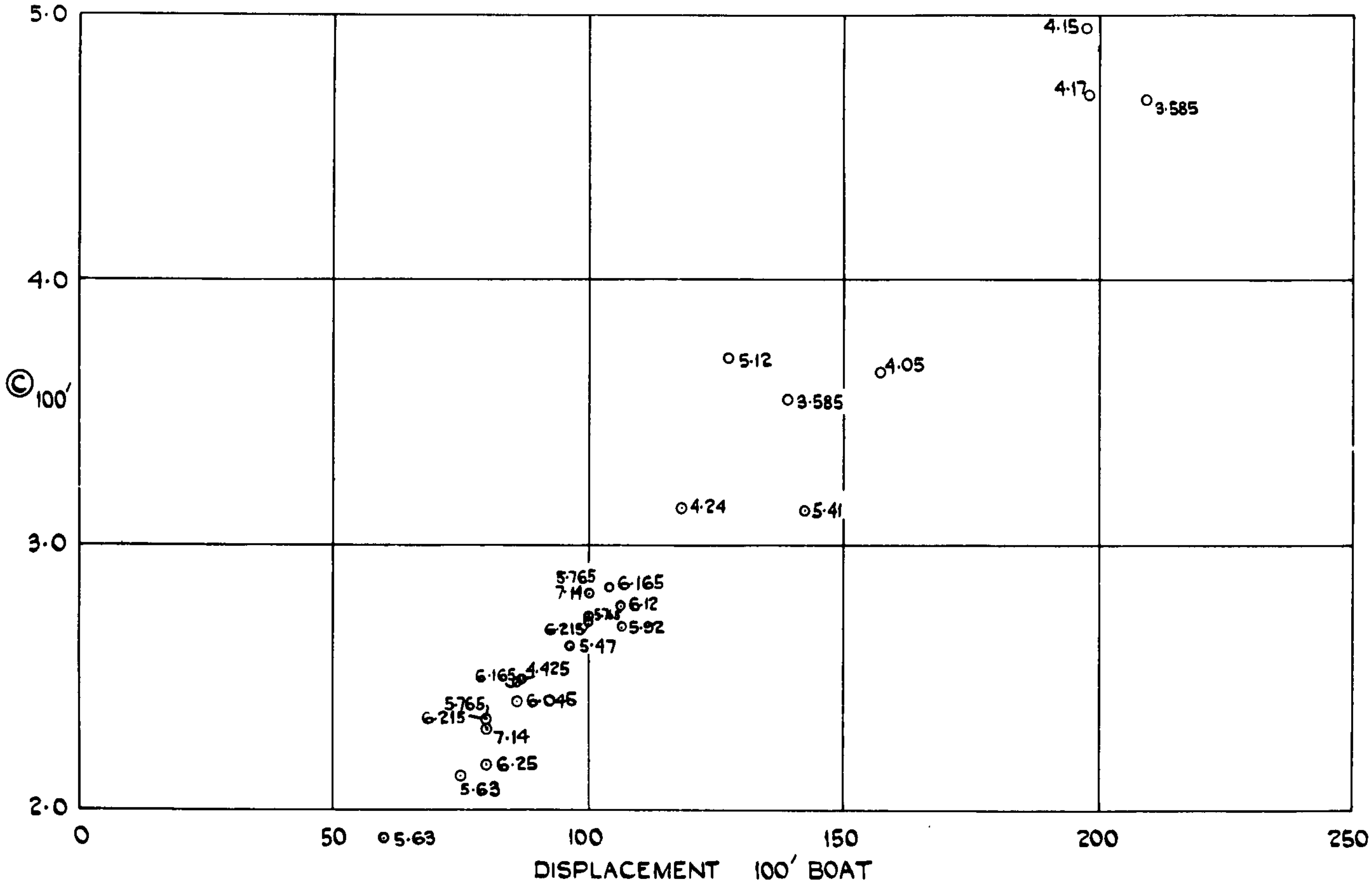

Fig. 11. Resistance coefficient © for a 100-foot round-bilge boat versus displacement in tons; $V/\sqrt{L} = 1.7$; L/B values inserted

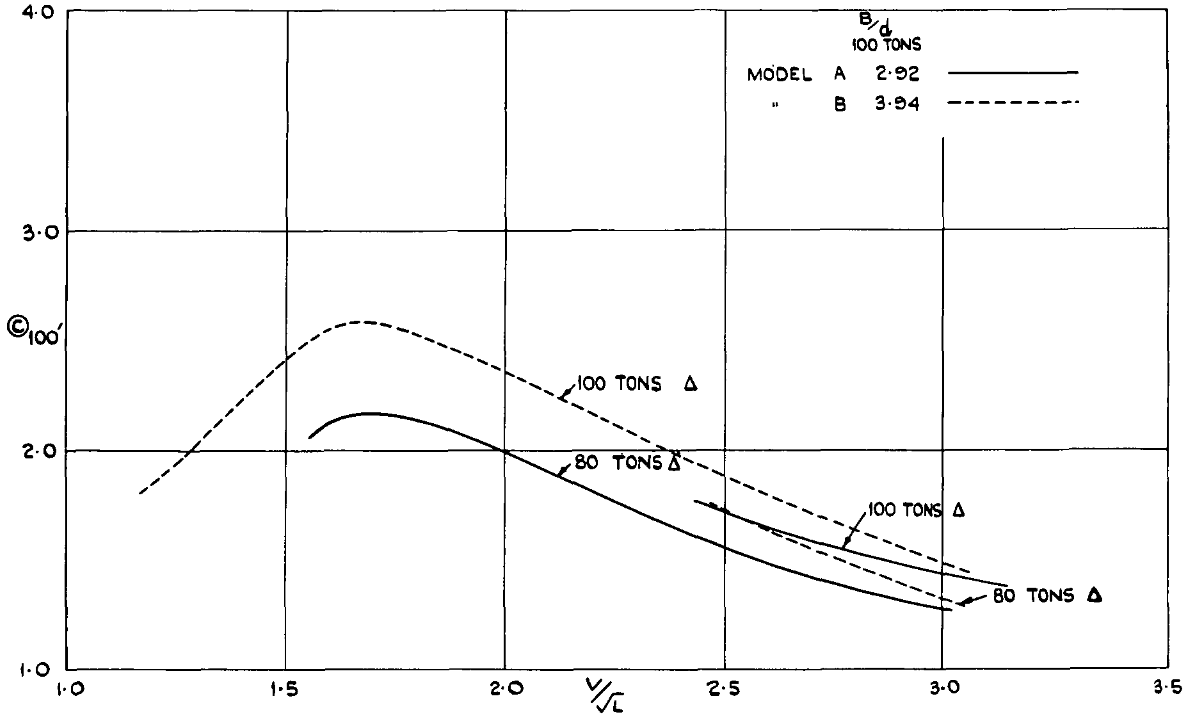

Fig. 12. Effect of B/d on the resistance coefficient for a 100-foot round-bilge boat

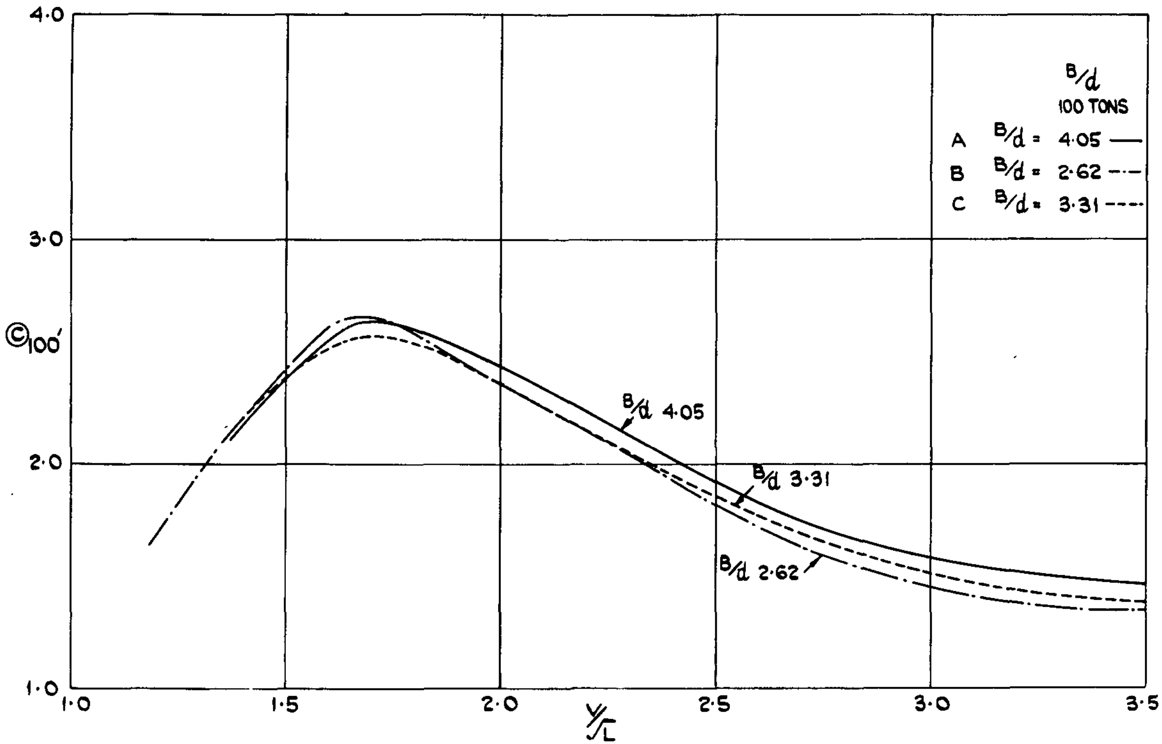

Fig. 13. Effect of B/d on the resistance coefficient for a 100-foot hard chine boat

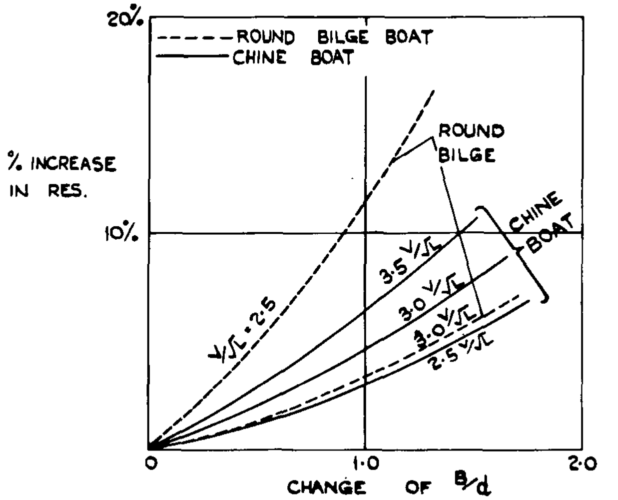

Fig. 14. Comparison of the effect of B/d on round-bilge and hard chine boats

The available information does not permit any reliable estimate to be made of the effect of changing the position of the longitudinal centre of buoyancy (*LCB*), although this is generally considered to be an important parameter. As shown in Fig. 15 two forms, otherwise similar, with *LCB* positions 6.25 percent and 8.81 percent of the length aft of midships had almost identical resistance coefficients, possibly because the maximum advantage to be gained by moving the *LCB* aft had been achieved in the first of them. In both forms the forward waterlines were straight, and by moving the *LCB* further aft the half angle of entrance at the waterline, initially 11 degrees, was reduced to 9-1/2 degrees, probably the practical minimum. It is suggested that so long as the *LCB* is far enough aft to permit straight waterlines, there is little to be gained by further movement aft.

Appendage Resistance

Appendages for high speed displacement-type hulls generally comprise: shaft brackets, either *A* or *I* type, propeller shafts, usually angled to the flow, stub bossings at the hull, skegs, rudders, either single or twin, bilge keels, and bar keels.

The separate resistances of these appendages were measured in one case at NPL by removing each one in turn from a fully fitted model. The measured model resistances, expressed as percentages of the naked hull resistance, were: twin "*A*" brackets, 7.5%; shafts and stub bossings, not measurable; twin rudders (large), 10.5%; bilge keels (large) 4.0%; bar keels, 5.0%; for a total of 27.0%.

In accordance with current NPL practice for such appendages, this total was halved in estimating the additional full-scale resistance. At the design speed/length ratio 1.35, this gave an additional resistance coefficient δ © of 0.27. For two other models the full-scale addition δ © was 0.21; one of these had a single rudder and the other twin rudders, though neither had bilge keels, which are not usually fitted to launches and similar vessels. In all three cases the appendage resistance coefficient showed little variation with change

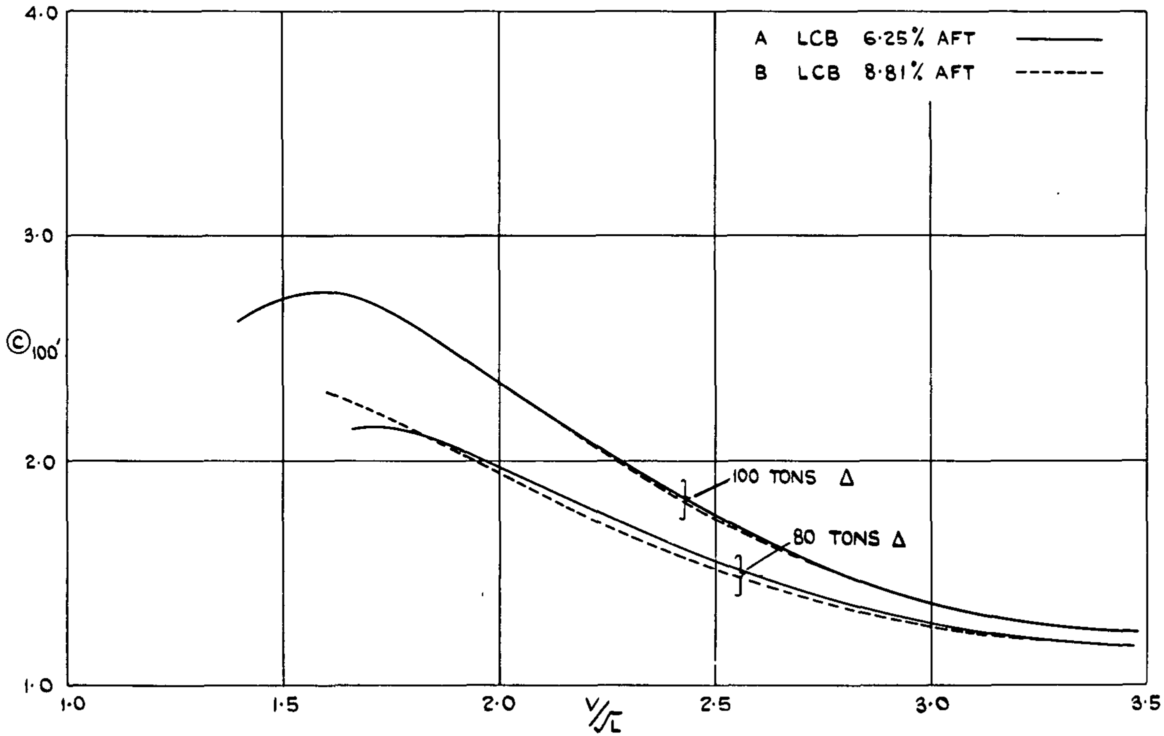

Fig. 15. Effect of the longitudinal centre of buoyancy on the resistance coefficient for a 100-foot round-bilge boat

of speed. These measurements suggest that the total resistance for typical appendage arrangements is more closely estimated by a constant δ $\textcircled{C}$ rather than by a constant proportion of the naked hull resistance. A tentative value for the resistance coefficient of normal stern arrangements, excluding bilge keels, is thus $\delta\ \textcircled{C} = 0.20$.

Effect of Changes in Scale

The average resistance coefficients $\textcircled{C}$ in Fig. 7 apply directly only to vessels of length $L = 100$ feet; for vessels of other lengths a skin friction correction must be applied. The conventional relation for this is

$$\textcircled{C}_{100'} - \textcircled{C}_L = (O_{100'} - O_L)\ \textcircled{S}\ \textcircled{L}^{-0.175}$$

where the subscripts 100′ and L refer to vessels of length 100 feet and L feet respectively, "O" is Froude's friction coefficient, and $\textcircled{S}$ and $\textcircled{L}$ are Froude's circular wetted-surface and speed/length constants respectively. This does not hold for high speed displacement-type forms, but a modified relation does give friction corrections with practical accuracy. This modified relation is

$$\textcircled{C}_{100'} - \textcircled{C}_L = k(O_{100} - O_L)\ \textcircled{S}\ \textcircled{L}^{-0.175}$$

where the factor k may be taken as a function of displacement/length ratios $\Delta/(0.01L)^3$ alone with sufficient accuracy for preliminary design purposes. Values of k and of Froude's "O" are shown in Fig. 16. It should be noted that the wetted surface area of a round-bilge displacement-type form does not vary sufficiently with speed to cause appreciable errors in using a fixed value of wetted area. For preliminary design purposes it is generally adequate to estimate the wetted area S from a simple formula such as the Denny-Munford relation

$$S = L(1.7d + BC_B)$$

where C_B is the block coefficient and the other symbols are as previously defined.

PROPULSION

Components of Propulsive Efficiency

Components of propulsive efficiency have been derived from the results of propulsion experiments made with eight different models. In these experiments the thrust was measured in the direction of the propeller shafting, and corrected to eliminate the effect of the weight of the propeller. The resistance was measured as the horizontal force in the direction of motion, but no attempt was made to compute the fore and aft component of the thrust. Open water experiments with propellers alone were made with the shaft horizontal.

Values of the thrust deduction fraction t, the wake fraction w based on thrust identity, the relative rotative efficiency η_R, and the hull efficiency η_h are given in Figs. 17-20 in terms of the speed/length ratio $V/\sqrt{L}$. These show that the thrust deduction fraction t tends to vary much as the resistance coefficient $\textcircled{C}$, with a maximum near $V/\sqrt{L} = 1.5$ and

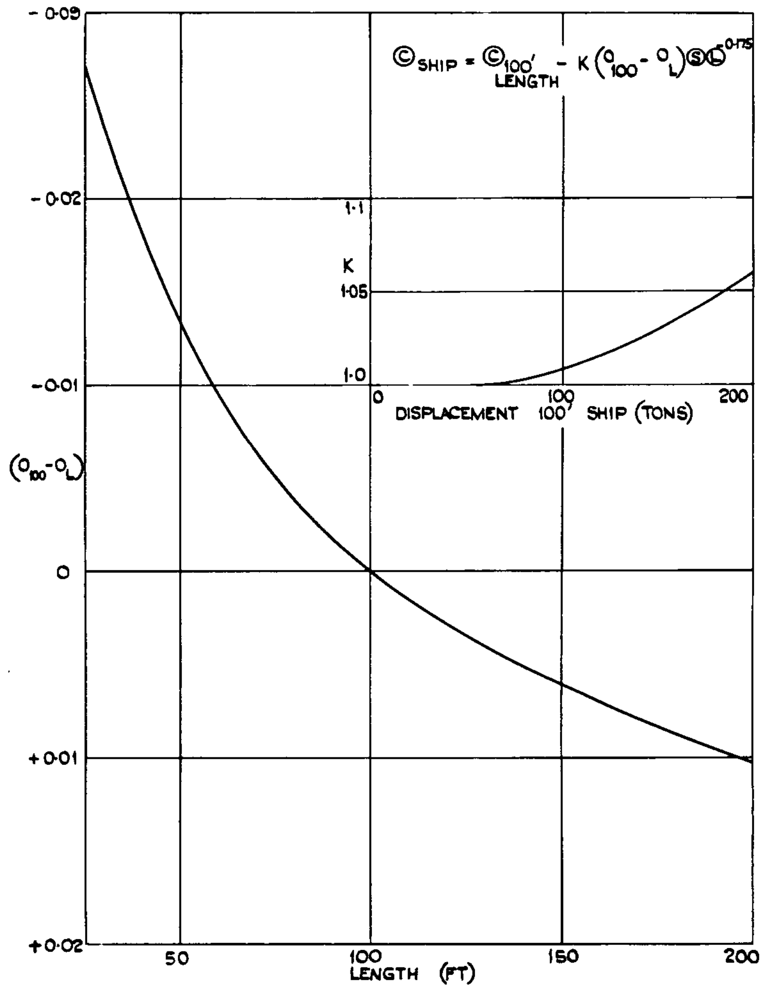

Fig. 16. Values of k and of Froude's "0" for a round-bilge boat

decreasing at higher speeds, finally approaching open water conditions. The wake fraction w is small, as expected, and slightly negative at high speeds.

From these factors and the propeller open water efficiency η_o, which depends principally on the permissible propeller diameter or the engine-propeller gear ratio, the quasi-propulsive coefficient η can be estimated as

$$\eta = \eta_o \eta_R \eta_h .$$

Ship-Model Comparison

Good data from full-scale trials under ideal conditions are required to determine the correlation factors linking predicted performance from model experiments with that measured on shipboard. Little reliable data for high speed displacement-type forms are available, and

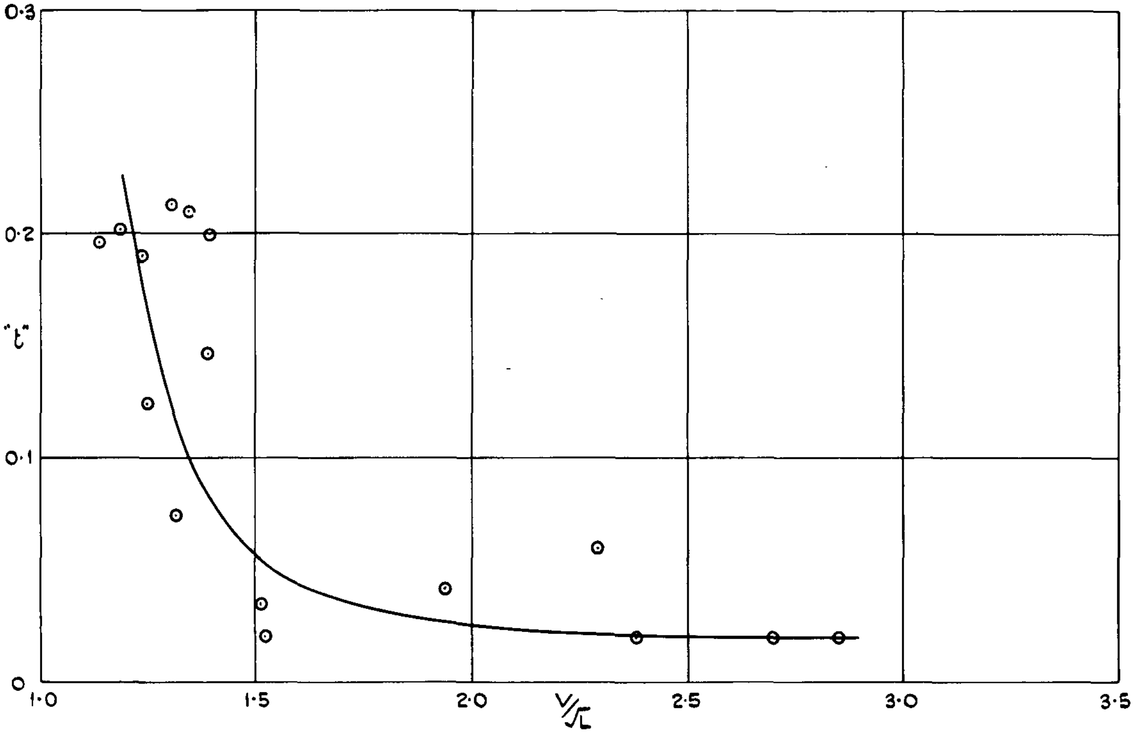

Fig. 17. Variation of the thrust deduction fraction t with $V/\sqrt{L}$ for a round-bilge boat

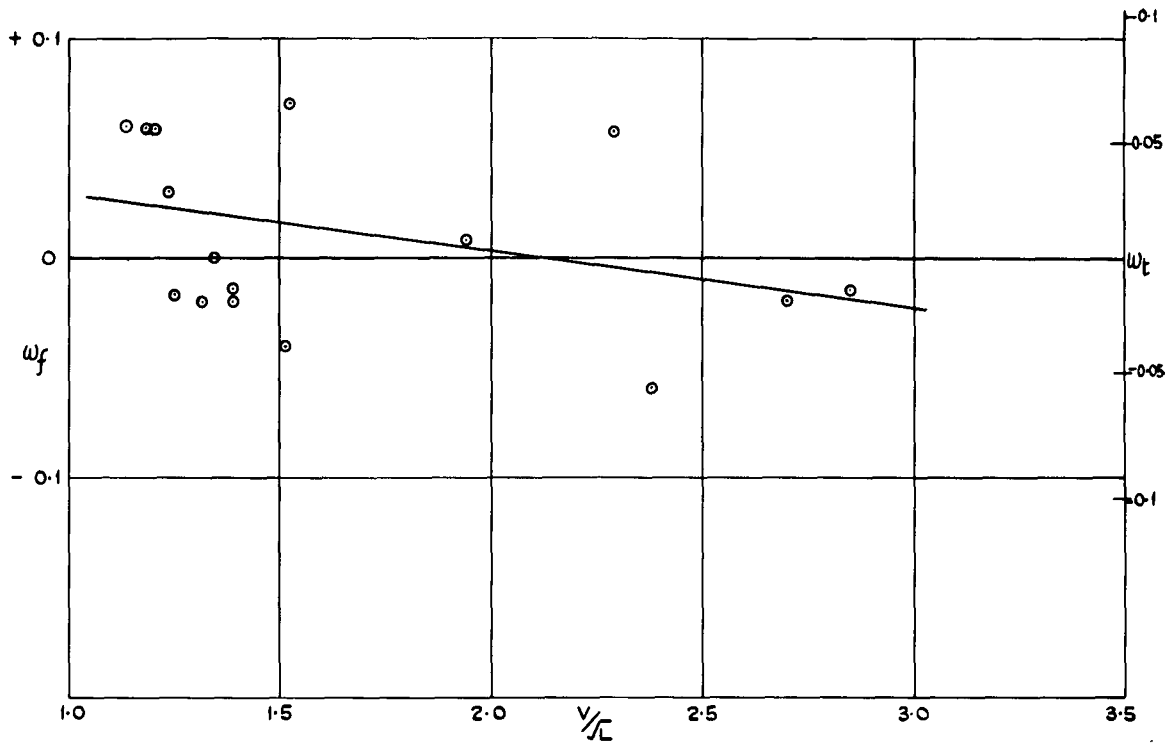

Fig. 18. Variation of the wake factor w_f with $V/\sqrt{L}$ for a round-bilge boat

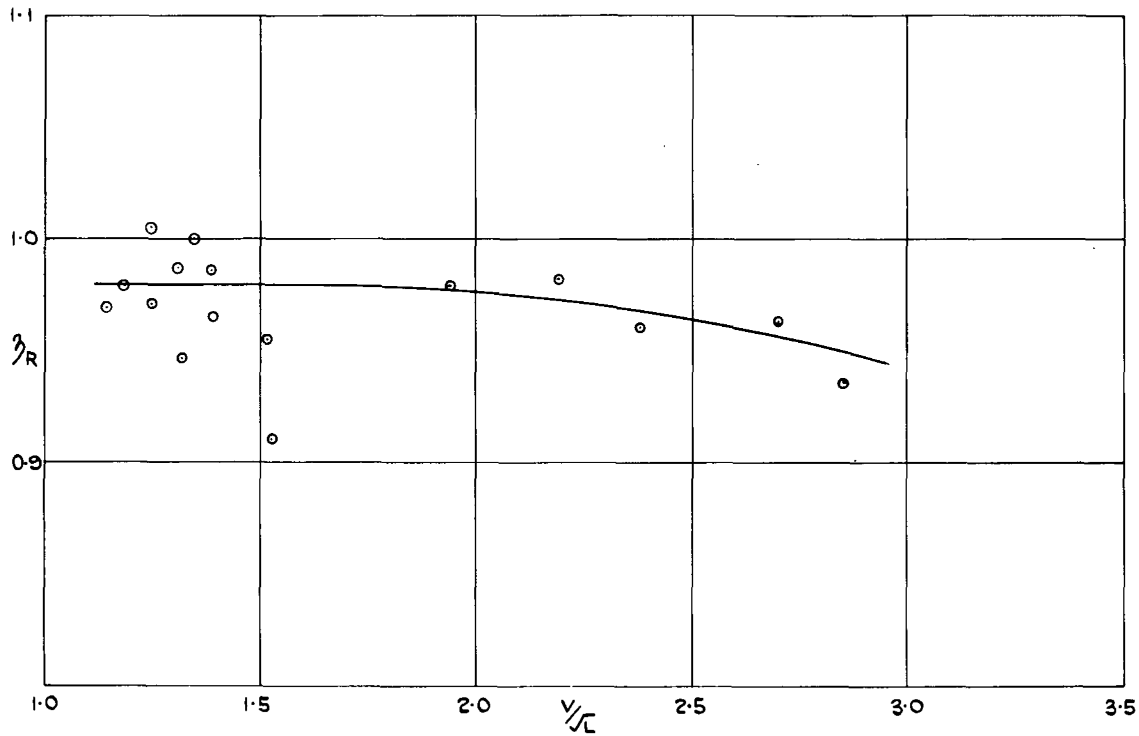

Fig. 19. Variation of the relative rotative efficiency η_R with $V/\sqrt{L}$ for a round-bilge boat

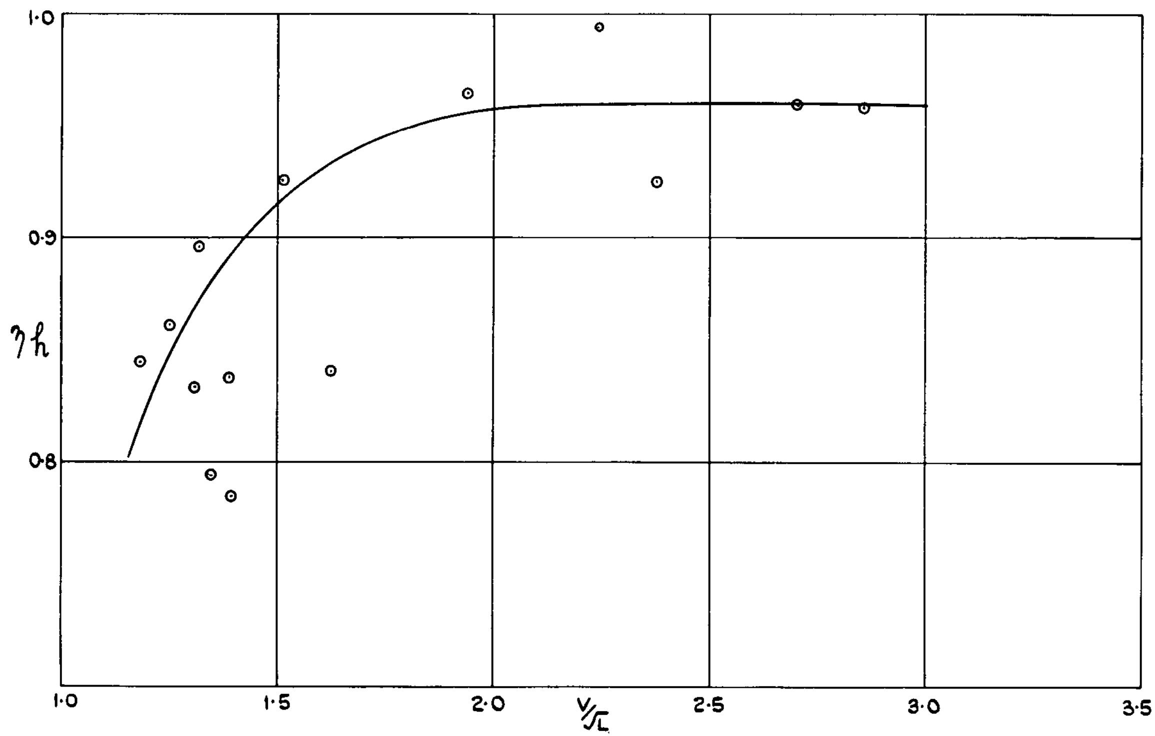

Fig. 20. Variation of the hull efficiency η_R with $V/\sqrt{L}$ for a round-bilge boat

an attempt to obtain further results is being made by the Ship Division, NPL. For this purpose two portable torque meters, of strain gauge type, have been designed and constructed at NPL to fit propeller shafts having diameters from 1-1/4 inches to 6 inches. A description of the first of these is given in Appendix A.

Using this torque meter, good trial results have recently been obtained for two twin-screw launches, one 71 feet in length and the other 51 feet. Resistance experiments were made at NPL with both forms, and propulsion experiments with the model of the smaller vessel.

The results of the trials with the 71-foot launch are given in Fig. 21; this shows the measured power for one shaft (only one torque meter was available) and also calculated

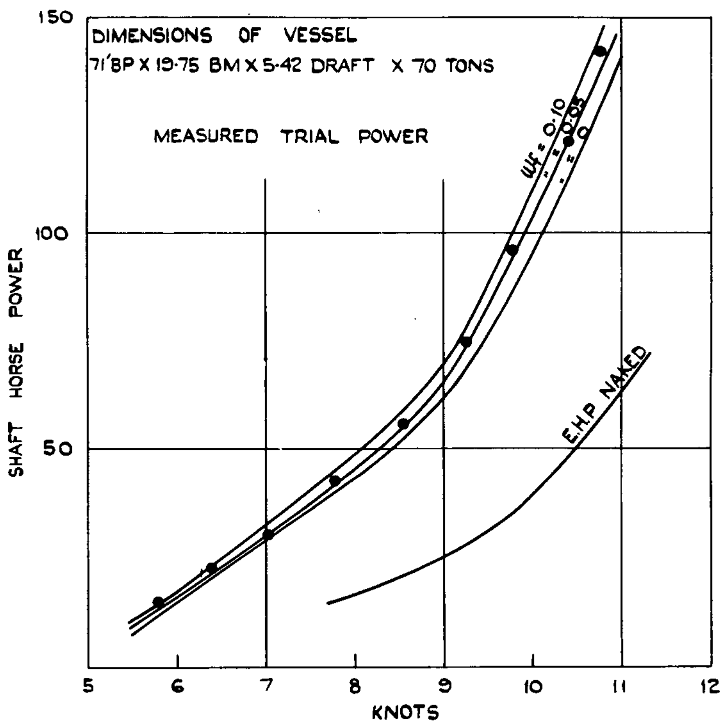

Fig. 21. Results of trials with a 71-foot round-bilge launch in a water depth of 50 feet

values of the power absorbed by the propeller for assumed wake fractions w_f of 0, 0.05, and 0.10. Closest agreement between measured and calculated power occurs for w_f = 0.05; the estimated value was 0.03. The measured power at the highest speed is greater than the absorbed power calculated for w_f = 0.05; this may have been caused by a 2-1/2 percent difference in the shaft revolutions. Had both shafts run at the higher speed of that with the torque meter, the total power, and consequently the speed, would have been slightly higher. In this case w_f = 0.05 would have given close agreement at the highest speed.

The full-scale power was estimated directly from the model results, taking η_R = 0.99, η_o = 0.65, η_h = 0.83, and appendage resistance of 0.20 Ⓒ corresponding to a factor of 0.10. This gave a delivered horsepower (*dhp*) equal to 118 for a speed of 10.77 knots; wind resistance estimated from measured velocities increased this by 3 percent to *dhp* = 121.5. The measured power was 138; the correlation factor (ratio of measured to estimated power) is thus 1.135, slightly higher than the current NPL value 1.10 for this type of vessel. Part of the discrepancy may be due to a small difference in trim between model and full scale. These trials were run in water of depth about 50 feet, considered sufficient not to introduce any depth effect.

The trials with the 51-foot launch were run in water of depth 20 feet or less while the model experiments were made in water of 45-foot equivalent depth. A possible shallow water effect thus confuses the ship-model comparison; Fig. 22 shows that the greatest discrepancy between measured and predicted powers occurs at speeds close to the critical speed for water of depth 17 feet. In this trial also only one torque meter was fitted, but it was here assumed that both propellers absorbed equal powers. A power estimate based on

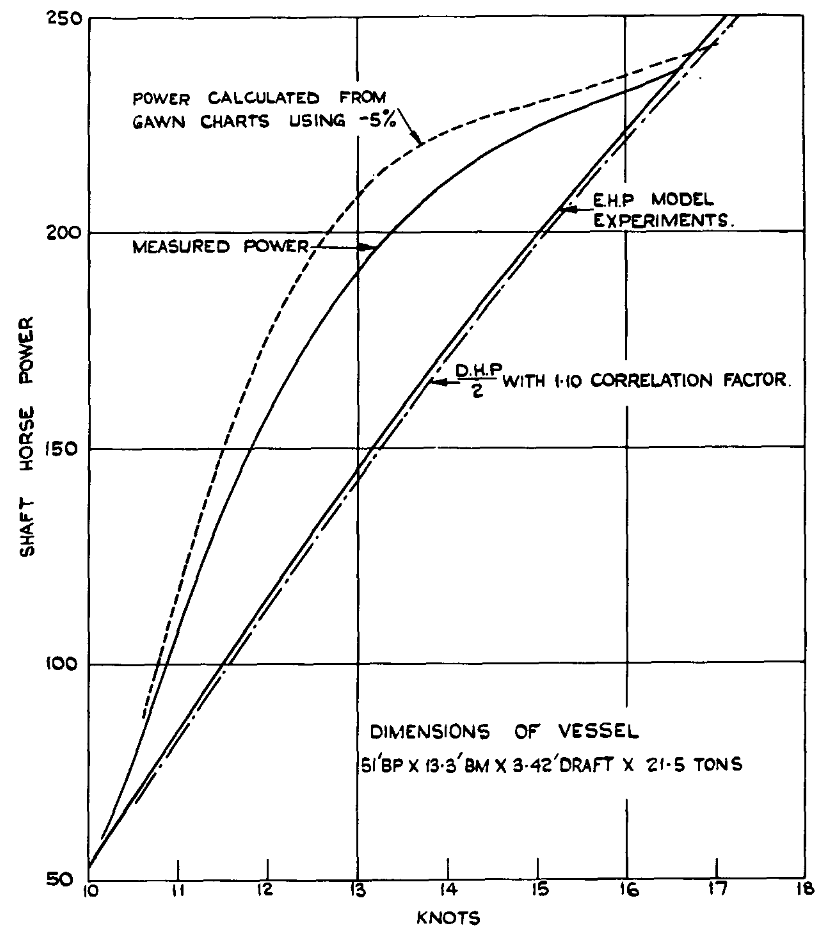

Fig. 22. Results of trials with a 51-foot round-bilge launch in a water depth of 20 feet or less

a correlation factor of 1.10 and a measured model wake fraction of 0.05 gave good agreement with the measured power, particularly at higher speeds.

The results of these two trials suggest that a ship-model power correlation factor of 1.10 is reasonable; further trial data are needed to confirm this estimate.

RUNNING PERFORMANCE

Effect of Spray Strips

At speed/length ratios of 3.5 and above ($F_n \geqq 1.2$), a round-bilge displacement-type hull develops a troublesome bow wave. This occurs at about 25 to 40 percent of the length aft of the bow, and for a 100-foot vessel can be 4 to 5 feet above the still water level. Unfortunately, this wave often is covered by a fine spray film which starts at the bow profile and in still water clings to the hull above the main wave profile. In bad or windy weather this film breaks away from the hull as spray which can cause considerable wetness of the afterstructure.

Model experiments were made with spray strips in an effort to combat this effect. On the form shown in Fig. 23 a strip 1 inch × 1 inch (for a 100-foot vessel) was fitted at about 3 feet above the still water line from the bow to midships. This prevented the film rising to the top side of the model, and enabled the speed to be raised from 30 to 42 knots without serious trouble.

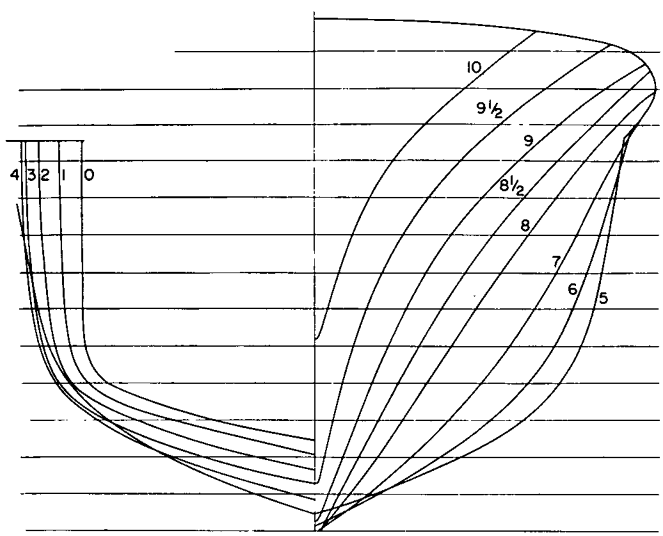

Fig. 23. Model 2084; 100 feet × 16 feet × 5.48 maximum draft × 100 tons

It has been found that spray strips either reduce the resistance of a model or have no measurable effect. Model trim is increased by about 0.5 degrees at the highest speeds.

Design of a Round-Bilge Form for Good Seakeeping

For a small vessel, less than 100 feet in length, designed to operate at sea, the performance in waves can be more important than in smooth water. Near-synchronous pitching conditions may well be met, and waves no more than 3 feet in height can cause slamming decelerations up to 9 g, injurious to both structure and personnel, as recorded at sea [3] and observed in model experiments.

The requirements for a small vessel with good seakeeping qualities include: (a) a dry deck, especially forward of the wheelhouse so as not to interfere with navigation, and (b) avoidance of slamming, and (c) minimum resistance so that available power is efficiently used in maintaining speed in a seaway. Other qualities are also required, but these are not discussed here.

An attempt to design a hull form with these qualities was made some time ago at NPL. Initial experiments were carried out with models of hard chine forms similar to those used on wartime air-sea rescue launches. However, for a design maximum speed/length ratio of 3.4, it was soon found that a round-bilge form with a fine bow and a transom stern had a superior smooth water resistance. Such a form, designated Model 2084, was designed as shown in Fig. 23, and for comparison a hard chine form, designated Model 2117, shown in Fig. 24, was also evolved; boats with this hard chine form were known to have had

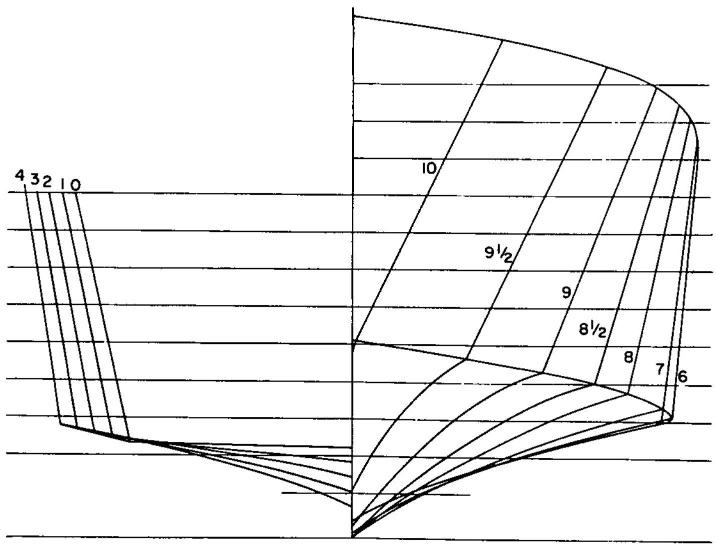

Fig. 24. Model 2117, 100 feet × 18.8 feet × 4.80 maximum draft × 100 tons

to reduce speed considerably in moderate seas. These two forms had the following main dimensions:

Type	Model	LWL (ft)	B on WL (ft)	Max. d (ft)	Δ (tons)
Round bilge	2084	100	16.0	5.48	100
Hard chine	2117	100	18.8	4.8	100

Smooth water resistance data for the two forms are given in Fig. 25; the round-bilge form behaved well up to 30 knots ($V/\sqrt{L}$ = 3.0), and a spray strip fitted as described previously helped to keep the hull clear at higher speeds.

Experiments with 1/15 scale models were also made in head seas. In these each model was towed by a bridle held by hand on the towing tank carriage. Hand towing was preferred to a rigid attachment to allow slight retardation of the model when its resistance increased in passing through a wave.

Figures 26-28 show extracts from continuous film records taken of both hulls in the wave conditions indicated in the figures. These film extracts have been chosen to show the vessels in their worst position, low in the wave with spray being thrown upwards or to the side.

The chine form is "stiff" in waves and tends to slam violently. The low chine forward throws water forward and up, obscuring wheelhouse vision and producing a wet ship. The round bilge form pitches more but this reduces slamming. The flare forward, which was designed with particular care, is very effective in throwing water away from the hull. The film records clearly show the superior wave-performance of the chineless, round-bilge form. Although the behaviour of the hard chine form could be improved by raising the chine line forward, it was not possible to reduce its resistance to that of the round-bilge form. It is hoped that a vessel having this round-bilge form will shortly be built.

SHALLOW WATER EFFECTS

Many builders of small boats do not appear to be aware of the marked effect of depth of water on wavemaking resistance. Trials are frequently run in river estuaries over accurately measured distances in water between 10 feet and 15 feet deep, which is considered to be ample for vessels with drafts of 2 feet to 3 feet. Not surprisingly, often the designed maximum speed is not achieved.

The 51-foot twin screw launch mentioned earlier demonstrates this shallow water effect. Its draft was 3.42 feet, and trials were run in about 20 feet of water, for which the critical speed ($v = \sqrt{gD}$) is about 15 knots. Figure 22 shows that the measured and predicted power curves differ most at about 13-1/2 knots. Saunders [2] quotes a method for assessing the depth of water effect both below and above the critical speed; this is based on data from trials of a German destroyer, the only available information for speeds above the critical value. Using this method, and taking the depth of water as 20 feet for the 51-foot launch, the shallow water effect given in Table 2 has been calculated.

The considerable differences between the power ratios directly measured for the 51-foot launch and those deduced from Saunders' data are not surprising, since Saunders states that other model and full-scale data do not agree with the German results. It is clear that further information is necessary, and a series of NPL trials in different depths of water with a 27-foot launch are being completed.

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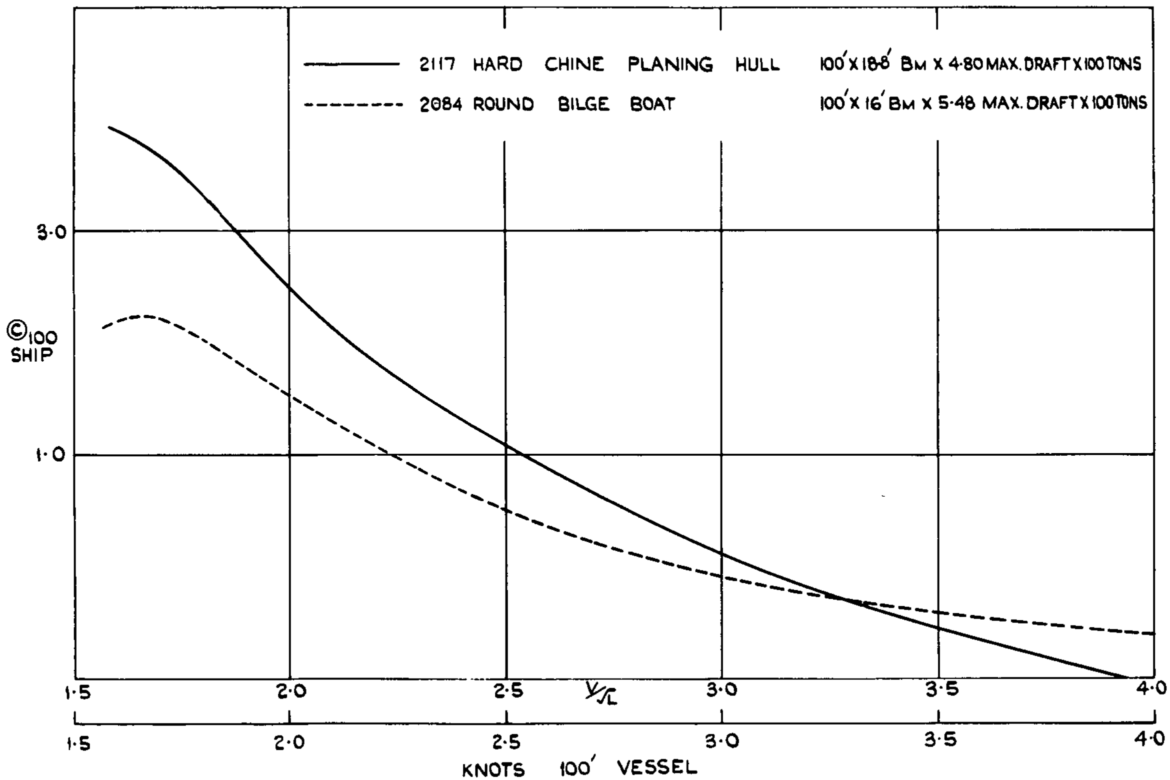

Fig. 25. Comparison of the resistance coefficient © for a round-bilge boat and a hard chine boat

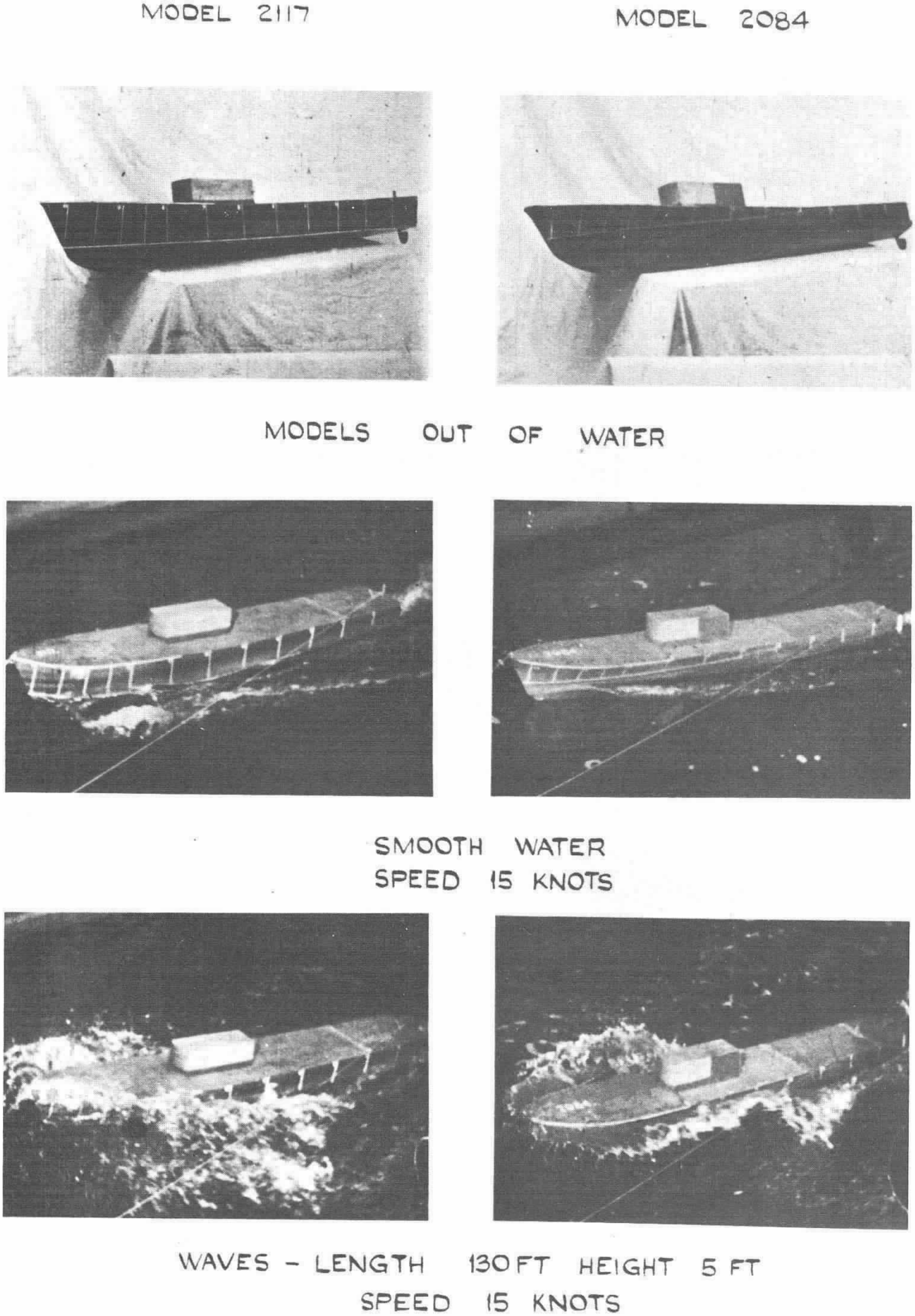

Fig. 26. Behaviour of the two models of Figs. 23 and 24 in waves

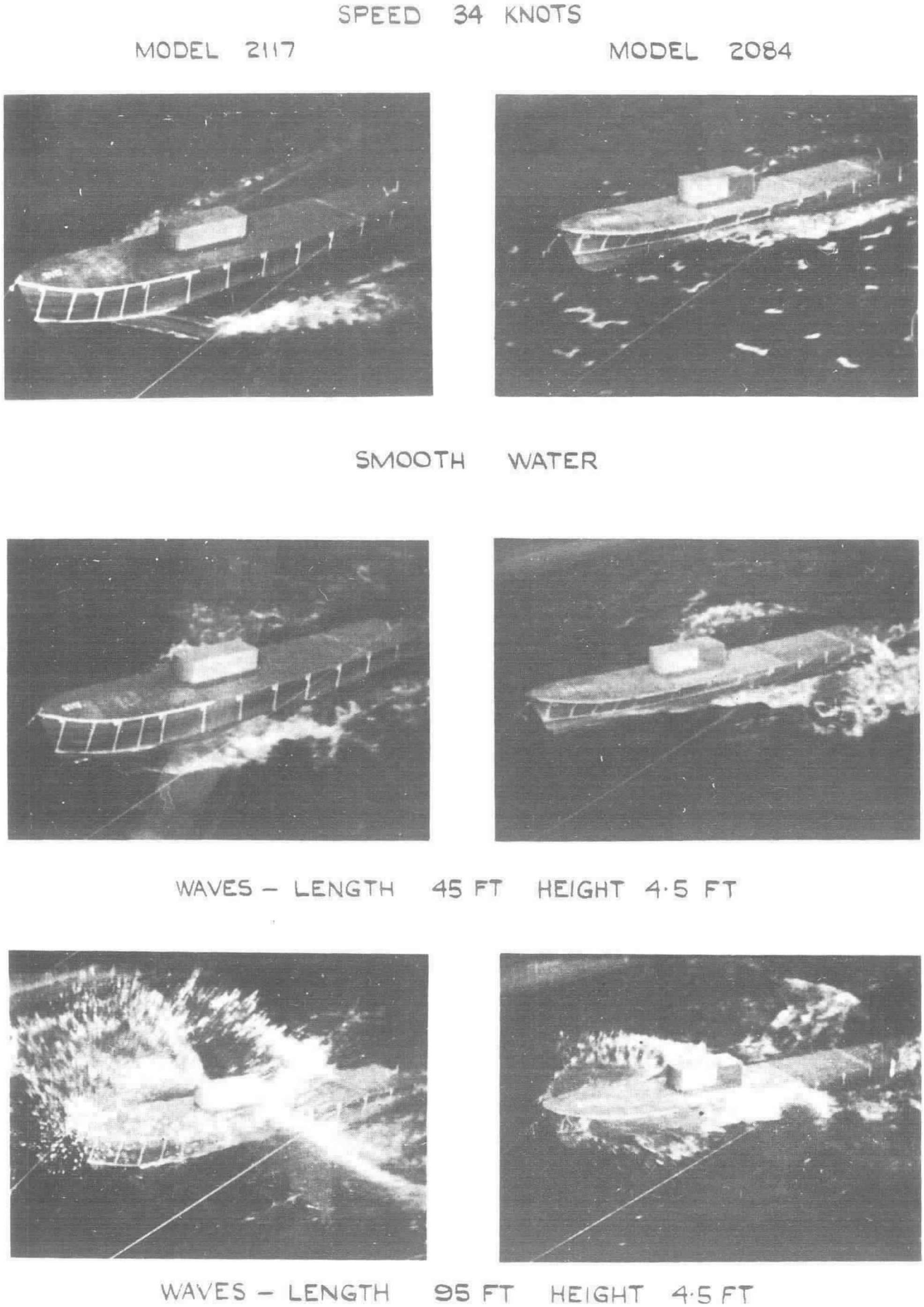

Fig. 27. Behaviour of the two models of Figs. 23 and 24 in waves

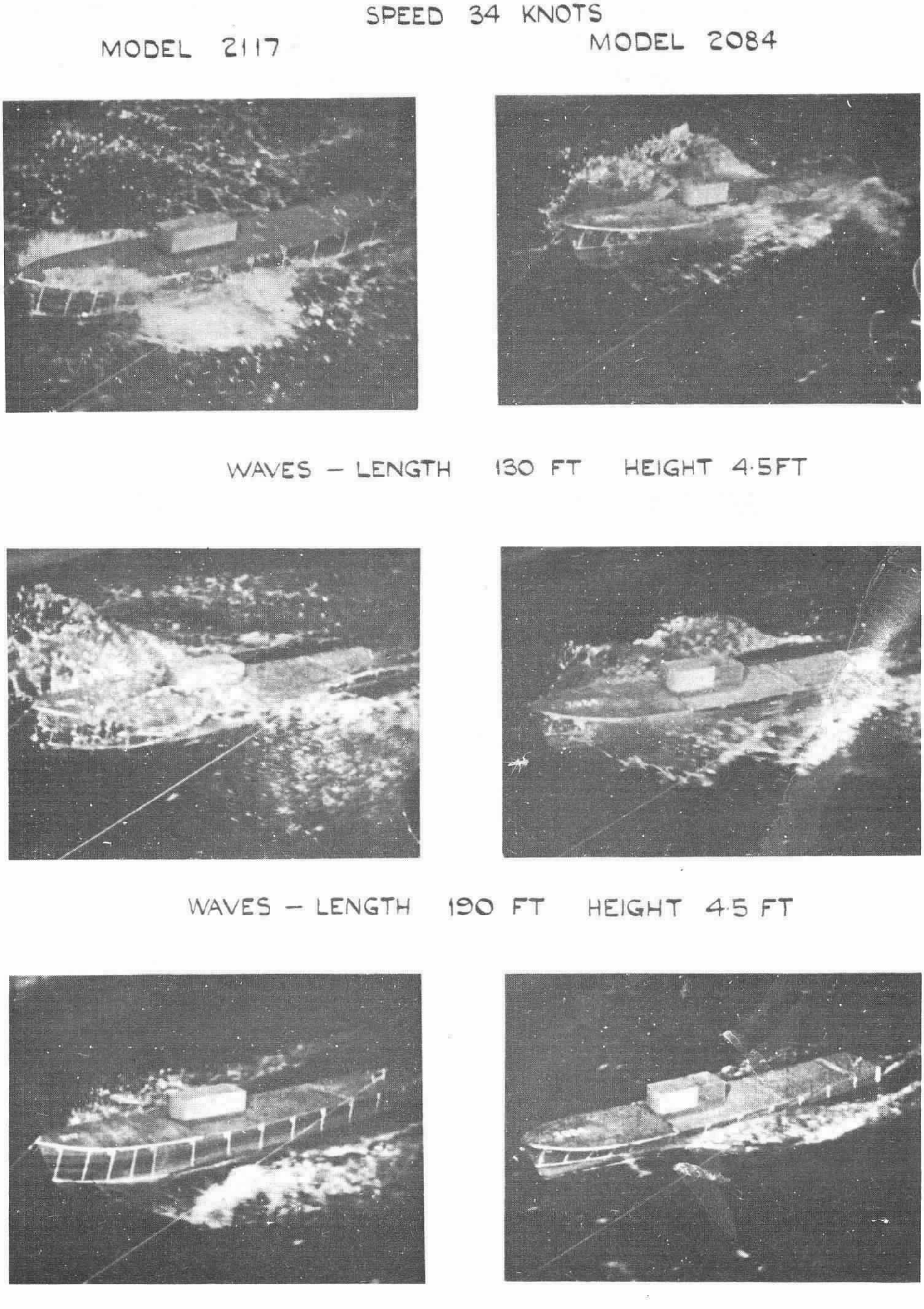

Fig. 28. Behaviour of the two models of Figs. 23 and 24 in waves

Table 2
Shallow Water Effect for a 51-Foot Launch in 20 Feet of Water

Speed/Depth Ratio $v/\sqrt{gD}$	Power in Shallow Water / Power in Deep Water	
	Predicted	Measured
0.720	1.0	1.285
0.850	1.25	1.364
0.867	1.5	1.330
0.885	2.00	1.313
0.915	2.50	1.270
1.010	2.50	1.120
1.080	2.00	1.035
1.170	1.50	Probably
1.230	1.25	becoming
1.370	1.00	< 1
1.400	0.95	
1.500	0.90	

FUTURE WORK ON ROUND-BILGE FORMS

While it is hoped that the preceding summary of data will be of use to the designer, it has also emphasized the scanty nature of the information presently available. Consequently, systematic investigations into high speed round-bilge forms are to be carried out as part of the research programme of the Ship Division, NPL.

The round-bilge form Model 2084 has been taken as the parent form. Systematic changes in form will cover variations in displacement, beam/draft ratio, and position of *LCB*, and features such as the depth and width of transom and the slope of afterbuttocks will also be varied. The displacement range will cover that shown in Fig. 7, and speed/length ratios from 1.2 to 3.5, and to 4.0 in certain cases, will be investigated.

Most of the model experiments will be made in the new large No. 3 tank at the Ship Hydrodynamics Laboratory of NPL. Its greater depth will allow higher speed/length ratios to be reached without encountering serious interference at the critical speed $v/\sqrt{gD}$, its greater size will enable larger models to be run so that reasonable size propellers can be fitted for propulsion experiments, and the wavemaker control system will facilitate experiments in irregular head and following sea conditions. The propulsion experiments will be made with at least three propellers of different pitch and diameter for each form so that data appropriate to different engine-propeller gear ratios will be obtained. Some detailed measurements of hull pressures are also planned.

DISPLACEMENT HULLS WITH PARTIAL HYDROFOIL SUPPORT

Recent studies of hydrofoil supported craft have included some interesting comparisons with high speed displacement-type hulls (e.g., Ref. 4). These naturally show that the effective lift/drag ratio, or displacement/resistance ratio, is higher for displacement hulls at low speed/length ratios, and then significantly higher for hydrofoil craft at high speed/length ratios (say above $V/\sqrt{L} = 4.0$). The hydrofoil craft considered in these comparisons have been those in which the complete hull is above the water at the design operating speed, only a minimum of foil and support structure remaining immersed.

It seems worthwhile to consider whether a mixed-type craft has any advantages. In this the total weight of the craft and any foil support structure would be balanced partly by buoyancy, as in a conventional displacement vessel, and partly by dynamic lift generated by hydrofoils below the surface. Such a mixed type craft would thus move through the water rather as though it were a displacement vessel operating at a lighter draft than in its deep load condition. It seems possible that such a craft might have operating advantages compared with a fully supported hydrofoil craft, and consequently some preliminary estimates of the resistance of such a craft are given here.

Among the available NPL data for high speed displacement hulls are some results for one form at different displacements. These indicate that, at a fixed speed/length ratio the displacement/resistance ratio is approximately constant between full and half displacement. This enables an estimate to be made of the variation of resistance with displacement over this range. For any assumed reasonable value of the lift/drag ratio for a hydrofoil support system (which could closely resemble the simple arrangement familiar in ship roll stabilisers), it is then possible to estimate the total resistance of the partly supported mixed-type craft. Making a reasonable allowance for the additional weight of the foil support structure, it is then possible to compare the total resistances of a series of partly supported craft based on a single parent normal displacement-type hull. This simple analysis may be expressed thus:

$$r + D = R$$

$$\Delta + L = W + S$$

$$L/D = \epsilon$$

$$S = kL$$

where r – Resistance of hull remaining in water
Δ – Displacement of hull remaining in water
L, D – Lift, drag respectively of hydrofoil system
ϵ – Lift/drag ratio of hydrofoil system
R – Total resistance of "mixed" craft
W – Weight of vessel excluding foil and support structure
S – Weight of hydrofoil and support structure
k – Weight ratio of hydrofoil and support structure to lift of foil system.

Figure 29 shows a group of resistance curves derived on this basis from the data obtained from a model of a displacement-type craft having a full displacement of 80 tons and length 100 feet. At each speed/length ratio selected, estimates have been made for $L/D = 10$, 15, and 20 for the hydrofoil unit, and for $k = 0.15$. Variations in L/D significantly affect the result, but reasonable variations in k have little effect.

The shaded areas in Fig. 29 represent regions in which resistance, and thus power reductions are achieved. It is obvious that as L/D increases, so the possible reductions also increase. It is also clear that if displacement-type hulls can be designed so that the displacement/resistance ratio decreases with the displacement, then significant gains may also be achieved; this is now being investigated.

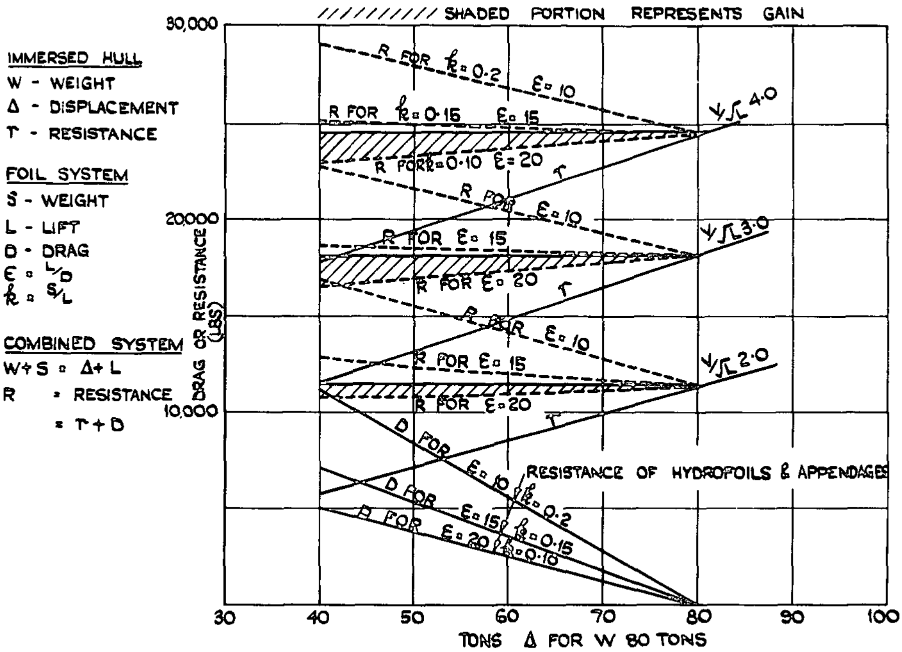

Fig. 29. Possible performance of a partially immersed 100-foot boat and fully immersed hydrofoils

Preliminary calculations suggest that hydrofoil support systems to carry half the weight of a conventional displacement type vessel should present no serious hydrodynamic or structural design problems, even if the foils are restricted to noncavitating types.

ACKNOWLEDGMENTS

The work described in this paper forms part of the research programme of the National Physical Laboratory, and the paper is published by permission of the Director, NPL. The assistance of Messrs. James and Stone, Ltd., Brightlingsea, Essex, in providing facilities for trial measurements on two launches, is gratefully acknowledged.

NOMENCLATURE

General

Δ = Displacement

L = Length (waterline), feet

V = Speed, knots

v = Speed, ft/sec

d = Draft, feet

B = Beam (waterline), feet

g = Gravitational constant, ft/sec^2

D = Depth of water, feet

C_B = Block coefficient

Resistance

$V/\sqrt{L}$ = Speed/length ratio

F_n = Froude number $V/\sqrt{gL}$

0_{100} = Froude skin friction coefficient for 100-foot ship

0_L = Froude skin friction coefficient for L-foot ship

Ⓢ = Wetted surface × 0.09346 / $\Delta^{2/3}$

Ⓛ = Froude speed/length ratio = 1.055 $V/\sqrt{L}$

r/Δ $= \dfrac{©}{1.319}\left(\dfrac{V}{\sqrt{L}}\right)^2 \dfrac{L}{\Delta^{1/3}}$

r = Resistance, pounds

© $= \dfrac{ehp \times 427.1}{V^3 \Delta^{2/3}}$

Propulsion

t = Thrust deduction

η_h = Hull efficiency

η_r = Relative rotative efficiency

η_o = Open water propeller efficiency

η = Quasi-propulsive coefficient

w_f = Froude wake fraction

w_t = Tayler wake fraction

dhp = Delivered horsepower at propeller

ehp = Effective horsepower = © $V^3\Delta^{2/3}$ / 427.1

Note: The constants used above are for ships in salt water.

REFERENCES

[1] Kafali, K., "The Powering of Round Bottom Motor Boats," International Shipbuilding Progress, Vol. 6, No. 54, Feb. 1959

[2] Saunders, H.E., "Hydrodynamics in Ship Design," Vol. 2, p. 408

[3] DuCane, P., "The Planing Performance, Pressures and Stresses in a High-Speed Launch," Trans. I.N.A. 98:469 (1956)

[4] Crewe, P.R., "The Hydrofoil Boat; Its History and Future Prospects," Trans. I.N.A. 100:329, Oct. 1958

APPENDIX A

NPL Small Portable Torsionmeter No. 1

This torque meter (Fig. A1) was designed to be readily portable, and to fit shafts having diameters from 1 to 6 inches without any need for removing any part of the propeller shaft. To meet these requirements a strain-guage-type meter was designed, with the gauges mounted directly on the shaft and their output leads taken through a readily dismountable slip ring assembly.

Four small resistance strain gauges are bonded to the shaft by an appropriate strain gauge cement at a position close to, and inboard of the stern gland. The gauges lie on 45-degree helices on the shaft surface, so placed that there is symmetry relative to the shaft axis and also to a plane normal to the shaft. Hence gauges respond only to torsion in the shaft and the gauge output is not affected by bending or thrust in the shaft. The gauge

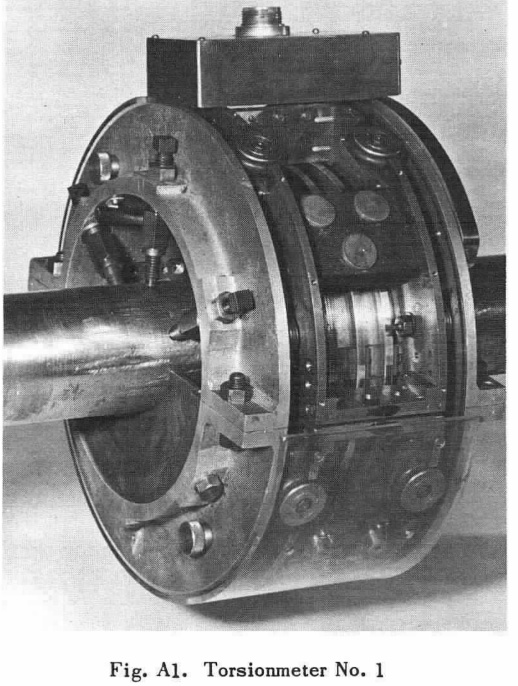

Fig. A1. Torsionmeter No. 1

leads are taken to a terminal block fixed to the shaft which is, in turn, connected to the inner unit of the slip ring assembly.

This slip ring assembly has two main units: an inner unit fixed to the shaft which carries the slip rings themselves, and which rotates with the shaft, and a fixed outer ring which carries the pickup brushes. Both units are made in two halves, so that they may each be placed around the shaft and bolted together. The inner unit of the slip ring assembly is large enough to fit around a 6-inch-diameter shaft; it is fitted to shafts of smaller diameter by inserting a distance piece in one half, and clamping the other half in place by adjustable, screwed feet. The outer unit carrying the pickup brushes is kept in position by rubber-tired rollers both radially and axially.

The five slip rings are of silver-plated brass, accurately scarphed at the joints of the two half units. Four rings are used for gauge leads, and the fifth for counting revolutions of the shaft. The brushes are of silver graphite. The gauge outputs are measured either on a Baldwin and Southwark SR 4 or a Siemens strain indicator. The complete meter is calibrated statically by applying known torques to the shaft close to the propeller when the vessel is out of the water.

The system has proved very reliable. The slip rings have behaved well, the strain readings are steady, and there is no appreciable zero drift.

DISCUSSION

Peter du Cane (Vosper Limited, Portsmouth)

We at Vosper have been interested in this paper as it discusses the range of high speed craft in which we have specialised over the last 25 years.

Within reason we agree that for $V/\sqrt{L} = 3.4$ a round-form hull as suggested by the authors would be our selection though only because if 3.4 is assumed as a maximum the probability is that for continuous cruising $V/\sqrt{L} = 2.5$ might be expected. At $V/\sqrt{L} = 3.4$ we could certainly produce a hard chine planing form with better resistance qualities than the round form and in our opinion with at least comparable seakeeping qualities.

It may be conceded that the subject of the relative merit of hard chine versus round form in waves is very relevant to this Symposium so that it is proposed to discuss it at some length:

Reference 4, which is a paper I submitted to the R.I.N.A. in 1956 and which was mainly concerned in recording extreme values, is quoted in justification of the following judgement on the qualities of the hard chine planing form: "waves no more than 3 feet in height can cause slamming decelerations up to 9 g, injurious to both structure and personnel, as recorded at sea."

It is felt there must be some misunderstanding of the data recorded in Ref. 4. Certainly there was no question of 9 g being recorded in 3-foot waves. If this is inferred anywhere it is incorrect, though it is conceivable that such an acceleration could be recorded at say the stem in exceptional conditions of speed and irregular wave formation.

In the particular case of this reported research in Ref. 4, however, the objective was to find extremely adverse conditions which did not prove easy and eventually resulted in a slightly hazardous attempt to run across the bow wave train of the "Nieuw Amsterdam" at 35 knots.

As this bow wave train was approaching us at a speed estimated to be 14 knots the conditions of encounter were equivalent to the launch meeting the waves at (35 + 14) = 49 knots.

Under these circumstances it is not surprising that 6 g was experienced in the wheelhouse. The maximum acceleration at the stem was recorded as 8.28 g but this was in exceptionally adverse conditions.

The relative merits of the round form and the planing form (so called) are discussed at some length in Ref. 4 and in the discussion thereto.

In this connection it is interesting to mention N.P.L. Report S.H.MV. 5 of Feb. 28, 1955, where a hard chine and round form were tested in waves. It was reported that there was little to choose as regards vertical accelerations between the hard chine and the round form. If anything the accelerations appeared less in the case of the planing form and certainly the spray on decks was reported as less in the case of the planing form.

This, of course, does not accord with the comment in the paper we are now discussing which reads as follows:

"The chine form is 'stiff' in waves and tends to slam violently. The low chine forward throws water forward and up, obscuring wheelhouse vision and producing a wet ship. The round bilge form pitches more but this reduces slamming. The flare forward, which was designed with particular care, is very effective in throwing water away from the hull. The film records clearly show the superior wave performance of the chineless, round-bilge form. Although the behaviour of the hard chine form could be improved by raising the chine line forward, it was not possible to reduce its resistance to that of the round-bilge form. It is hoped that a vessel having this round-bilge form will shortly be built."

Among other comparisons between planing and hard chine craft which are available in Ref. 4 is a closely reasoned and cautious judgement made in the course of the discussion by Dr. Gawn based on his great experience at the Haslar Tank and his collated data from sea reports. His contribution is offered verbatim:

"The author makes no secret of his preference for a hard chine hull and it does appear this type can be drier and generally no less satisfactory at sea than the round bilge. There is a need for firm fact to replace some of the contentious opinions often expressed on sea behaviour, and the author's tests go some way in this direction. However, the issue is not clear cut in a general sense but depends on the size and speed of the craft. If, for example, the speed-length coefficient is less than about 3.7 there is much to be said for the round-bilge form because of its resistance advantage in calm water. If, on the other hand, as in the launch dealt with in the paper the speed is much greater, then the hard chine hull can have a clear advantage as regards resistance in calm water and in my judgement is to be recommended, provided there is no overriding emphasis on economic cruising at low speed."

Since the date of that paper, quite a lot more evidence on this matter has reached me, both from actual seagoing experience and model tests in waves. In particular and resulting from model tests carried out in irregular waves at the Davidson Laboratory of the Stevens Institute in 1959, Figs. D1 and D2 were reproduced in a report of the "Development and Running of the 'Brave' Class Fast Patrol Boat" read at Göteborg in February 1960.

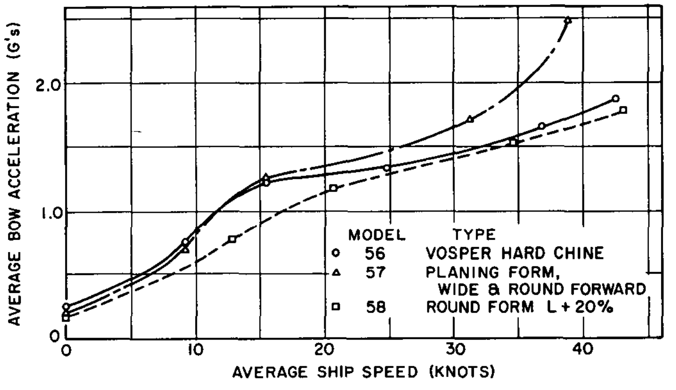

Fig. D1. Average bow acceleration versus average ship speed in an irregular head sea; Beaufort scale sea state 5; average height, 5.16 feet; signal height, 8.10 feet

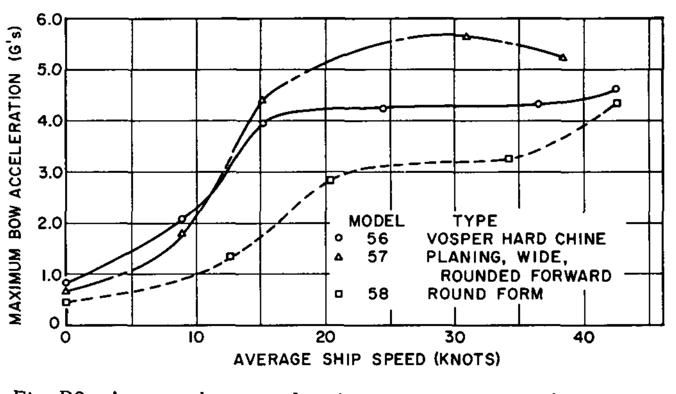

Fig. D2. Average bow acceleration versus average ship speed in an irregular head sea; Beaufort scale sea state 5; average height, 5.16 feet; signal height, 8.10 feet

This shows remarkably little difference between the round form 58 and the Vosper hard chine 56, especially when it is realised the round form was 20 percent longer for the same displacement.

To summarise, there is little to choose between the forms for speed/length ratios between say $V/\sqrt{L}$ = 2.5 to 3.8. After this the hard chine planing form is a "must" for reasons of resistance.

The planing form probably *can* be made to slam more unless trim is adjustable by transom flap or similar device. Equally the round form can undoubtedly be made to bury her forecastle in the waves and wash down her bridge. Directional stability is much better in the planing form and there is less liability to yaw and broach in a following sea. In head seas the hard chine craft will lose more speed due to the energy lost in generating spray which undergoes an acceleration relatively forward and outward from rest. This feature tends to keep the decks and bridge dry.

So far we have only discussed the relative merits of the round form and hard chine planing craft and it may be appropriate to point out here that in the light of development over the years the difference between the two classes of craft which used to be clearly defined are becoming very much less.

For instance, both types are in fact planing craft in the generally accepted sense. In both types it is fully appreciated that the waves must be met from ahead by relatively soft V-shaped or rounded sections. Equally, to obtain good resistance qualities the aft sections in both types are relatively flat and wide and run to a "chine" at the sides. To control the amount of spray on deck both types work a substantial flare in forward sections leading to a "knuckle" or chine.

In fact it is probably fair to say that in theory the optimum would be represented by a rather wide form incorporating the forward sections of a round-form craft with the aft sections of a planing form. The draw-back here is that the round form leads to wetness on deck and width leads to rather excessive slamming.

Such a compromise form is represented in Figs. D1 and D2 by model 57.

As the subject of the Symposium covers high speed craft it may not be out of place here to state the admittedly personal opinion that no hydrofoil craft of the surface piercing type has given any real evidence of being able to deal with substantially rough conditions in the open sea more successfully than the round form or hard chine.

It seems possible the submerged foil arrangement as demonstrated by Dr. Hoerner during this Symposium may be superior in this respect but at the moment this awaits practical proof at sea, at least in the larger sizes.

The ground effect machine may seem to be limited at the present state of the art by the very severe effect upon lifting efficiency likely to be experienced if cushion heights suitable for passage over the ocean waves are to be generated.

Finally as a constructive suggestion we should like to suggest to the authors that we could submit for trial a hard chine planing form which would prove a much more serious competitor to the round form proposed than is the form exemplified in model 2117.

It would also be most valuable to run a hydrofoil configuration of comparable dimensions and load-carrying capacity in comparison with the above forms.

D. Savitsky (Davidson Laboratory, Stevens Institute of Technology)

I would like to comment on the interpretation given by the authors to the relative seakeeping abilities of the so-called hard chine and round-bilge boats. It is the authors' contention that the round-bilge boat exhibits superior performance (less pitching motions and considerably less bow spray) than the hard chine boat. The point that I would like to clearly emphasize is that the existence of a hard chine or round bilge per se has no relevance to its behavior in waves. Heavy bow spray and large pitch motions are primarily dependent upon the fullness of the bow. For very sharp bow forms (high deadrise forward) the rough water behavior will be relatively mild and the development of bow spray very small. The reverse is true for a full bow (low deadrise forward). It is to be noted from the authors' paper that their round-bilge boat, which exhibited relatively mild rough water motions was indeed designed with large deadrise forward while their poorly performing hard chine boat was designed with low deadrise forward. With such design features it is expected in advance that the hard chine boat will perform poorly in waves. If the full bow is, for reasons of construction, necessarily inherent in a hard chine boat then the authors' general comments on the relative merits of each boat are correct. However, if this is not the case, that is, if high deadrise forward can be designed into each boat, then general conclusions on rough water behavior of round-bilge versus hard chine boats cannot yet be made.

In reply to the authors' plea for basic force data on planing boats, I would like to suggest the many reports prepared by the Davidson Laboratory of the Stevens Institute of Technology on this very subject. These basic experimental and analytical studies were sponsored at our laboratory by the Office of Naval Research.

E. V. Telfer (Technical University of Norway)

All I want to say is once more in connexion with presentation. In this paper the authors make the very positive statement that "again there is no clear evidence of any predominant form parameter." Against this I would like to suggest that what the authors really mean to say, but clearly do not, is that because they have adopted a basic presentation in terms of the displacement of the 100-foot ship, which is of course the Froude Ⓜ $= L/\nabla^{1/3}$ value in another form, this is clearly in itself the dominant form parameter and any other usual form parameter is likely to be of minor importance. Had the authors' diagrams such as Fig. 8 been presented in terms of Ⓜ the importance of relative length on constant displacement would have been more evident and the distribution of the individual spots would have more uniformly covered the Ⓜ base.

The issue is, however, not quite as simple as this. It should be noted that while at any constant displacement the Ⓒ values correctly grade the corresponding powers, a comparison of the Ⓒ value at one displacement does not give a direct comparison of the corresponding power ratio at any other displacement. The authors have made their Ⓒ comparisons under the condition of constant $V/\sqrt{L}$, or since $L = 100$ feet, under the condition of constant speed. It follows therefore that to get the relative power ratios P_1 and P_2 for two different displacements Δ_1 and Δ_2, the respective Ⓒ values must first be multiplied by the two-thirds power of their displacements. Alternatively to express the true relative power per ton displacement over the displacement range covered the Ⓒ values first require multiplication by Ⓜ. When values of Ⓒ Ⓜ are plotted to the displacement base, then the comparison is both correct and compatible. The authors' presentation is neither. For example, in Fig. 8 for 200 tons displacement a Ⓒ value of 4.9 may be indicated, while at 50 tons the Ⓒ value is probably only 2.45 at most for the same B/d ratio. The corresponding Ⓒ Ⓜ comparison is thus $4.9 \times 17.1 = 84$ against $2.45 \times 27.2 = 66.6$. The 50-ton boat thus requires 79.4 percent of the power per ton displacement needed by the 200-ton boat.

The authors' presentation might have tempted the unwary to believe that the 50-ton boat was twice as efficient as the 200-ton boat. It follows therefore, since the authors' presentation has very seriously exaggerated the apparent importance of Ⓜ value, that the influence of other factors will not necessarily have quite the insubordinate influence which the authors are led to believe. I am not, of course, suggesting that the authors themselves in their specialised tank practice would be at all unwary; but I do feel very strongly that all data presentation for general professional consumption should be visually correct, so that he who runs may read, and read quickly and accurately.

R. N. Newton (Admiralty Experiment Works)

Any paper which provides data which can be usefully employed by a designer is very welcome at any time or place. This paper by Messrs. Marwood and Silverleaf is just such a paper and any opportunity to add to the information they present should not be missed, for the same reason.

Experiments on high speed planing forms having been carried out at A.E.W. for many years, with forms of different lengths and coefficients, in calm water and in waves, andwith various devices for improving performance, I will attempt to take this opportunity to qualify some of the statements in the paper and add to them, in a general way, in terms of the ratio of length of wave to length of craft and in terms of Froude number.

Here let me hasten to add that while I incline very much to Prof. Telfer's opinion that we should use more rational performance coefficients and parameters to show the trends arising from changes in form and displacement for the present purpose I must adhere to the commonly used ones, such as Froude number and length of wave to length of ship ratio.

First, then, speaking generally, on the effectiveness of steps in the bottom, a large number of resistance experiments conducted at A.E.W. between 1935 and 1944 led to the general conclusion that although stepped hulls had advantages over unstepped hulls in some ways, they also had inherent disadvantages and particularly increase of resistance at low speeds.

Experiments conducted with two, three, and five steps in the bottom, at different spacings and positions along the bottom indicated that these gave generally inferior performance to a single step on the same form, particularly as regards an increased tendency to porpoise and, judged by tests of one form in waves, inferior behaviour in waves.

There is an optimum position for the step which effects a comparison between reduction in resistance above planing speed, increase in resistance below planing speed, and the liability to porpoise. The risk of porpoising increases as the step is moved forward. As the step is moved aft the speed at which planing starts increases until when the step is well aft the craft can hardly be said to be planing at all in the usual sense of the word.

Secondly as regards resistance in calm water, the position of the center of gravity is critical and a small movement has a marked effect. Generally speaking a moderate stern trim in still water is an advantage at planing speeds but adds to resistance at lower speeds.

Too large a dihedral angle amidships adds to resistance due to increased running incidence and reduced rise. A low dihedral angle is good for calm water performance but detracts from behaviour in waves.

A narrow transom gives increased running trim but the effect on resistance is complicated. There is an optimum transom width depending on speed, trim, loading, and other form parameters. For high speed a wide transom is desirable, but if the width is excessive stern trim is impeded. If the width exceeds 0.8 of the maximum beam the sides of the hull may not be "clean" at top speed.

Running attitude at a given speed is mainly governed by loading and static trim, although chine dihedral, curvature, and width of transom can have appreciable effects as noted previously. Excessive trim causes high hump resistance which can lead to "locking up" of the engines. The effect of propellers is to increase the running trim by as much as 1 degree.

Next as regards resistance and motion in waves, as the authors have shown, a form which has a low calm water resistance by virtue of high running trim can be a poor boat in a seaway, particularly as regards pitch and slamming. On the other hand too small a trim leads to wetness and pounding even though slamming and motion may be reduced.

In $L/30$ waves maximum motion in ahead seas usually occurs at λ/L between 2.5 and 3.0 when the pitch is about 1.1 to 1.2 × maximum wave slope, out to out, and the heave about 1.1 × wave height.

In $L/20$ waves the maximum motion occurs at λ/L about 2.0 or less, pitch being about 1.35 × maximum wave slope and heave 0.9 × wave height.

The motion is characterised, at planing speeds, by notably small movement at the stern.

Mention is made in the paper of accelerations as high as 9 g and this has been referred to by previous speakers. Such accelerations can be produced of course but conditions on board would hardly be tenable and there would be grave risk of structural damage. Local accelerations of this order occur of course when the craft slams.

For a given maximum acceptable acceleration, which may be decided for strength or other reason, there is a limiting speed and wave slope; e.g., in a 70-foot boat if the maximum acceptable acceleration is 2 g the boat should not be driven into seas steeper than $L/20$ at speeds greater than $F_n = 0.7$. Steeper waves can only be encountered, to keep below 2 g, if they are shorter than ship length.

Now as regards the advantages of round bilge and hard chine forms there is much that can be added to the information given in the paper and among some of the more important facts are these.

In calm water the round bilge form is less resistful than the hard chine form up to at least $V/\sqrt{L} = 3.2$ ($F_n = 0.96$). Thereafter the hard chine form shows to advantage. This agrees with Fig. 25 of the paper being discussed.

In waves the resistance of the round bilge form is still less than the hard chine form up to $V/\sqrt{L} = 3.7$ ($F_n = 1.1$), but only in waves up to $3.0L$ when the difference is of small magnitude only.

As regards motion in waves it is necessary, I think, to qualify the statement in the subsection of the paper titled "Design of a Round-Bilge Form for Good Seakeeping" in terms of F_n and L_w/L_s.

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Up to a certain speed, usually about $F_n = 0.7$, both pitch and heave in ahead seas are greater for the round-bilge boat, and less than the hard chine boat beyond this speed. This applies in waves up to 1.5 or $2.0L$. In longer waves the round-bilge form continues to pitch more but there is little difference in heave between the two forms.

In the following seas the pitch of the round-bilge form is slightly greater in waves up to $2.0L$ and slightly less in longer waves. The reverse is true of heave.

In shallow seas the accelerations at the fore end of the round-bilge form are slightly higher than for the hard chine design, and this corresponds with the slightly greater pitch and heave. Accelerations at the after end are much the same for both designs. In steeper seas the accelerations of the hard chine form are greater both at the fore end and at the after end.

Slamming is generally more pronounced in the hard chine form for the same still-water loading condition as regards displacement and running trim.

With regard to wetness, as would be expected the water is thrown low and clear by the hard chine forms and higher and nearer the hull in the round bilge form, although both are generally dry in seas dead ahead. The effect of wind on the bow is to increase the degree of wetness, and the round-bilge form is wetter than the hard chine in this condition.

Finally there is one feature of high speed craft, of which the paper makes no mention, which I should like to remark on briefly, viz., the use of a transom flap in the form of a plate hinged at the lower edge of the transom so as to be adjustable and generally about 1/20th the length of the craft in width.

Model experiments show that in calm water running, stern trim and rise are reduced by the use of a transom flap at all speeds, from which it might be expected that the resistance would also be reduced at all speeds. In fact the measured model resistance is reduced by the use of a flap only up to a moderate speed, beyond which it increases, compared with no flap.

Nevertheless sea trials have shown that the flap improves the speed up to full power due to the change in interaction between hull and propellers having a favourable effect upon the hull efficiency elements. For a model of a 70-foot-long hull the augment of resistance was reduced and the wake increased to give an increase in hull efficiency of 8% at top speed, for a flap angle of 5 degrees.

Both model experiments and sea trials have demonstrated, however, that as the maximum speed increases, or as displacement decreases, the flap becomes less effective and smaller angles of incidence of the flap are required to obtain maximum effect. Ultimately a condition will be reached when the flap is of no propulsive advantage. One explanation, or part explanation, for this may be that in designs with high top speed, not fitted with a flap, the running trim ceases to increase above a certain speed and then begins to reduce, although rise may still continue to increase. Full consideration must be given to this fact when contemplating the incorporation of a transom flap in a new design.

As regards behaviour in a seaway in waves less than ship length and height, in the region of $L/30$ the flap has an appreciable effect upon pitch or heave at any speed. In waves of critical length about $2.5L$ when motion is usually severe the flap reduces the pitch and heave by about 10 percent.

In longer and steeper waves around 2 ship lengths and height $L/20$ and at F_n greater than 1.1 there is a more significant decrease in both pitch and heave of the order of 30 percent and 45 percent respectively.

At these high speeds, however, there is a grave danger of the craft plunging. In fact during one experiment with a model at $F_n = 1.4$ the model buried so deeply into the slope of the oncoming wave as to cause it to sink. Obviously in such conditions the ship would suffer grave damage or loss.

In following seas the use of a flap or wedge introduces a distinct tendency of the model to bury into the slope of the wave—much more so than in ahead seas. For instance, in the case mentioned previously the same model, in waves $1.5L$ using 5-degree flap incidence, plunged and sank at $F_n = 1.4$. The same occurred also at the same speed in waves $2.5L$. In longer wavelengths the model could only be controlled with difficulty and then only up to $F_n = 1.06$, when there was still a tendency to broach-to.

As a result of close study of films of models in waves and experimental data a general conclusion can be drawn. This is that whereas a transom flap can prove of advantage propulsively in calm weather, the degree of this advantage being determined by consideration of the top speed and displacement, before the flap can be used to improve the motion in waves, i.e., to make conditions on board more comfortable, it is essential for the operators to be able to recognise the sea conditions at the time, or to be provided with some scientific indication that the craft might be in danger. There is no doubt that with long experience of such craft, judgment can be placed upon the "average" length and height of sea which may be running, but this is very different from being able to predict the length and height of oncoming waves. For the time being, therefore, until much more research has been done into the statistics of wave spectra in a given ocean area, and until such time as the spectrum can be indicated with some degree of reliability, the use of transom flaps in rough weather cannot be recommended—at least not for craft designed for really high speed, say beyond 45 knots.

In conclusion it is perhaps pertinent to remark, in the light of these general observations that any change of form, or device, introduced into a new design, although it may improve performance from one aspect, may well detract from it in another aspect, and a compromise is very often necessary, and can only be found by careful model experiments.

K. C. Barnaby (John I. Thornycroft Company, Southampton)

The authors of this interesting paper referred to the difficulty of finding a suitable parameter for expressing their data. It is, of course, obvious that a conventional constant such as © is quite unsuitable. Once dynamic lift has taken charge, the wetted surface is no longer constant, but must diminish rapidly since the product $C_1 \frac{1}{2} \rho S V^2$ must always be less than the original displacement *weight,* by at least the amount of submerged hydrostatic buoyancy. Also, the resistance to motion increases with approximately V and very definitely not with V^2. Thus both the terms $\Delta^{2/3}$ and V^3 become incorrect at high speeds.

The one factor that cannot change as the speed increases is the actual weight and this forms the basis of a suitable parameter. If one plots the ratio $R/\Delta V$, that is, the total resistance expressed in pounds per ton divided by the speed in knots, for a common parent form at various displacement ratios and speeds, a family of curves is obtained that, at sufficiently high speed/length ratios will merge into one line that is nearly independent of displacement ratio and varies only slightly with speed. In general, there will be a slight increase with speed, except in the case of round-form hulls, which usually show a slight falling off.

This parameter $R/\Delta V$ is conveniently expressed in the modified form of $K = V/\sqrt{bhp/\Delta}$, where K is a constant depending on the type of craft and the static waterline length (or perhaps more correctly the beam as being an unaltered dimension). I have examined several hundred cases. Those giving $R/\Delta V$ directly have mainly come from tests at Mr. Thornycroft's Fort Steyne Model Basin. The more numerous K values have come from trial results and from published data. In no case have I found any wide discrepancy from Table D1. These latter are of course only approximate, but in naval architecture practice it is often more useful to have an approximation that can be made in a few minutes than a more elaborate method which takes a considerable time. The approximation can at least show whether a proposal is feasible or quite impossible.

Table D1
Values of the Constant K in the Formula* $K = V/\sqrt{bhp/\Delta}$
for Planing and Semiplaning Types

Length (ft)	Value of K for Various Types of Hull and Limits of Speed/Length Ratio			
	Round bottom, transom stern, very flat aft	V-Chine, stepless	Stepped	Hydrofoil
	$V/\sqrt{L}$ = 2.5-3.5	$V/\sqrt{L}$ = 2.75-4.5	$V/\sqrt{L}$ = 3.5-6.5	Up to 60 knots
20	2.25	2.75	3.6	5.3
30	2.60	3.10	3.96	5.5
40	3.05	3.65	4.3	5.7
50	3.34	4.00	4.6	5.9
60	3.45	4.20	4.8	6.1

*Δ in tons and V in knots.

Surprise may be expressed at the inclusion of round-bottom craft in the table. It is however necessary for such a type to have a transom stern and a very flat stern in order to reach speed length ratios of over $V/\sqrt{L}$ = 2.5. Under these circumstances, the stern lines are not very different from those of a normal V-chine type and a nearly comparable amount of dynamic lift can be obtained at high speeds. The increased drag due to the round bilge is however reflected in lower K values. It is assumed in all cases that the form is suitable for the speed and runs cleanly with a suppressed bow wave that does not lap high up the side but is deflected outwards. It is also assumed that the propulsive efficiency is normal, say, about 0.6.

Powering figures stated by Mr Rader, during the meeting, give a K of 4.76 at 40 knots, increasing to 4.90 at 55 knots. These were for a 100-ton V-chine hull of about 50 percent increased length over the maximum table length. They are thus in line with the table values.

J. B. Hadler and E. P. Clement (David Taylor Model Basin)

The authors have suggested that a high speed displacement-type hull combined with partial hydrofoil support may show performance which is superior to that of a conventional boat. A number of years ago the Model Basin gave consideration to the same suggestion and established a program of investigation under our Bureau of Ships Fundamental Hydromechanics Research Program. A partial support hydrofoil system, composed of two submerged foils, each on a strut attached to the side of the hull were installed on a model of the U.S. Navy 52-foot aircraft rescue boat. These foils were arranged such that their fore and aft location could be varied as well as their angle of attack. The optimum location for these foils was 32.7 percent of the length forward of the center of gravity, and at a minus 3-1/2 degree angle to the keel of the boat. At this condition the resistance was reduced throughout most of the higher speed range. For speeds below 15 knots, the resistance of the hybrid craft was somewhat greater. Between 15 knots and 30 knots the resistance decreased to about 75 percent of that of the conventional craft. Between 30 knots and 40 knots, the average reduction in resistance was about 25 percent. Although these results indicated favorable performance for the hybrid craft, no further work was undertaken at the Model Basin because of the urgency of the hydrofoil program itself. Further research should be continued in this area, particularly in investigating the performance of the hybrid craft in seaways and with foils of the surface-piercing type. The full details of this work are contained in a Model Basin report entitled, "Tests of a Planing Boat Model with Partial Hydrofoil Support," TMB Report 1254 dated August 1958.

The resistance data for high speed displacement-type hulls which are made available in the paper are very welcome. A similar presentation was made by H. F. Nordstrom in his 1951 publication, "Some Tests with Models of Small Vessels," but in that paper the maximum speeds are considerably below the maximum speeds for which data are given in the present paper. The data on the components of propulsive efficiency are particularly interesting and valuable, since little information of this kind has been previously made available. It is noted that there is considerable scatter in the plots of these data, however, presumably because of significant difference in the hull forms and appendages of the boats represented. The usefulness of the data would be enhanced considerably if the authors would provide some additional information on those designs which were self-propelled. A profile drawing of the stern, including the appendages, would be particularly helpful. This would make it possible for designers to select values of wake and thrust deduction for the design most like the one with which they were concerned at the moment, and thereby obtain assistance in making accurate predictions of performance.

W. J. Marwood and A. Silverleaf

The authors first wish to thank all those who contributed to the discussion of this paper; their comments have greatly enhanced its value.

Several contributors have discussed the relative merits of round-bilge and hard chine hull forms; however, the aim of the paper was solely to provide data on high speed displacement-type hulls, and there was no desire to make comparisons of a general nature between different basic forms. Commander du Cane's views on this important practical point naturally deserve close attention, but it is necessary to use carefully defined terms when comparing hull forms. The round-bilge form as defined and described in the paper is without any chine or knuckle line which could affect the flow, and is in extreme contrast to the hard chine form used in some of the experiments in waves. Forms of these two types can be designed to differ much less, and the differences between them in performance will then also be much less. Indeed, the round-bilge form as quoted by Cdr. du Cane retains a chine line forward and aft intended to influence the flow pattern.

There seems to be some doubt as to the correct interpretation of some of the results given by Cdr. du Cane in Ref. 4. Table IV of that paper gives for Run 35 the maximum acceleration in the fore peak (position A1) as 8.36 g in waves 2 to 3 feet high. On the other hand, Table III gives a value of 2.86 g for the same run; this may be an average value of the measured acceleration, and agrees well with model values measured at N.P.L.

The experiments quoted by Dr. Graff, carried out in Duisburg with high power round-bilge forms in deep and shallow water, are of considerable interest. A full account of these experiments has been given,* and the results show the same characteristics as those given here for similar experiments. However, detailed comparison shows some differences in the absolute values of resistance coefficients, and these discrepancies emphasise the need for further investigations of shallow water effects.

Mr. Newton's contribution is a most valuable addition to the paper, for which we are very grateful. In drawing on the extensive store of information available at Haslar he has raised many points which we are not qualified to discuss, and we shall restrict our comments to four of the topics he mentions. First, we agree that a narrow transom does give increased running trim, but only for speed/length ratios $V/\sqrt{L}$ greater than 1.8, and it then also increases the resistance. However, at lower speeds, particularly below the resistance hump around $V/\sqrt{L} = 1.5$, when there is little or no hydrodynamic lift, a narrow transom is found to decrease resistance. Indeed, at lower speeds still it becomes advantageous to fit a cruiser stern. Second, we are slightly surprised at Mr. Newton's statement that propellers increase running trim significantly. Provided the model is towed along the shaft axis when the resistance and trim are measured, we should not expect the trim to alter appreciably when it is self-propelled, unless the propellers have a marked influence on the average surface pressures over the after body. Third, we agree that at planing speeds the pitching centre is well aft; indeed, it can be aft of the transom. Fourth, experiments made at N.P.L. support Mr. Newton's views about transom flaps. Attempts to turn or "hook" the extreme afterbuttock lines have also succeeded in reducing calm water resistance. However, we believe that, in a vessel designed to operate mostly at one speed, transom flaps are only needed when the basic hull design is faulty. For a vessel intended to operate at several different speeds, the fitting of flaps is more easily justified. Some interesting effects of flaps on heave and pitch have been observed in experiments at N.P.L., and these agree well with the comments made by Mr. Newton.

Mr. Savitsky points out that the terms round bilge and hard chine are not satisfactory descriptions for comparing ship performance in waves. We agree with this criticism, and agree also that the deadrise angle forward is a more realistic criterion to adopt. The principal justification for the descriptive terms used in this paper is that they are commonly employed by naval architects, and it was hoped that they would be sufficiently precise for the purposes of this preliminary report. We must also apologise for not referring explicitly to the Davidson Laboratory reports on planing craft.

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* W. Sturtzel and W. Graff, "Systematische Untersuchungen von Kleinschiffsformen auf Flachem Wasser im Unter- und Uberkritischen Geschwindigkeitsbereich," Report 617, Versuchsanstalt für Binnenschiffbau, Duisburg, 1958.

Professor Telfer rightly draws attention to our statement that "again there is no clear evidence of any predominant form parameter." We agree that, for clarity, this would have been better if qualified by the important reservation "apart from displacement/length ratio or its equivalent Ⓜ ." We also agree that a presentation based on Ⓜ would be better than one with displacement alone as basis. Professor Telfer's advocacy of © Ⓜ as the best criterion for power comparisons at different displacements also has much to commend it. However, we were primarily concerned with presenting the basic information as simply as possible, and presumed that those who used it would know how to do so correctly.

Mr. Barnaby also advocates an alternative basis of comparison. However, his factor $K = V\sqrt{\Delta}/P$ also has disadvantages. It has the dimensions of a velocity, and it seems that a nondimensional factor $K/\sqrt{V}$ would serve Mr. Barnaby's purpose better. This is closely related to what is sometimes called transport efficiency, and when the values quoted by Mr. Barnaby are converted into this form the very valid practical advantages which he claims for this type of comparison factor are demonstrated more clearly. It is also worth pointing out that much of the data given in the paper are for speeds at which hydrodynamic lift is relatively unimportant, and to that extent © is still a reasonable drag criterion.

The comments of Mr. Hadler and Mr. Clement also form a valuable addition to the paper. We hope that further work will be undertaken at DTMB and other establishments to examine the potentialities of hybrid craft with partial hydrofoil support. The additional information requested on details of propulsion arrangements will be included in a subsequent paper intended to give results of systematic propulsion experiments with models of high speed displacement-type hulls.

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