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DESIGN OF TUG PROPELLERS

by

T.P. O'Brien

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A Station of the
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Mr. A. Silverleaf, Superintendent of Ship Division

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Synopsis

This report is a reprint of two articles giving some results of recent work at NPL on tug propulsion and design of tug propellers, and of a third article summarising published data and giving an outline of work in progress at NPL.

The first article (1) describes charts for designing screws and making cavitation estimates, and it includes a procedure for designing tug screws and making estimates of their performance. Formulae for assessing blade stresses and estimating weight and moment of inertia are also given. It summarises single screw tug propulsion data recently obtained at NPL, and it gives worked examples on the design and performance assessment of two tug screws, one designed for free-running conditions, the other for towing conditions.

The second (2) shows that significant improvements in performance can be achieved by using two-speed gear boxes enabling the optimum rate of rotation to be chosen for both free-running and towing conditions. Additional worked examples are given, the results of which show that for a screw designed for free-running conditions and driven via a single speed gear box the loss in towing pull would be 22%, but if a two-speed gear box were fitted the loss in towing pull would only be 3%. Similarly, for a screw designed for towing conditions and driven via a single-speed gear box the loss in free-running speed would be 15%, but if a two-speed gear box were fitted the loss in free-running speed would only be 1½%.

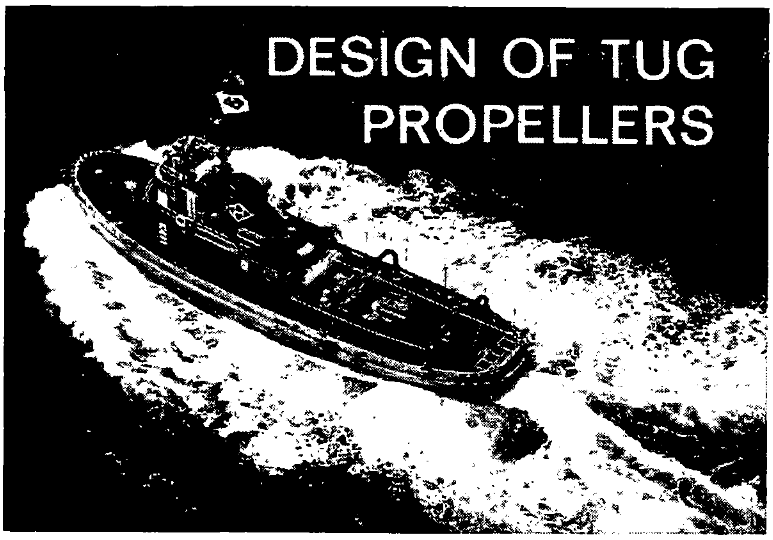
The third article (3) comments on the large amount of basic data that are required and the numerous aspects that need to be considered in designing tug propellers and making estimates of their performance. It gives a summary of available data comprising abstracts of relevant publications. It discusses research in progress at NPL on tug propulsion and propeller design.

(1) O'Brien, T.P. Design of tug propellers, London, Ship and Boat Builder International, April 1965, 18, 22.

(2) O'Brien, T.P. Propeller design and two-speed gear boxes with particular reference to tugs and trawlers, London, Ship and Boat Builder International, Nov. 1964, 17, 41.

(3) O'Brien, T.P. Research on tug propellers, London, Ship and Boat Builder International, 1966, 19, 28 (supplement to February issue).

DESIGN OF TUG PROPELLERS



This series of articles discusses general aspects of propulsion and applications to tug propellers operating at free-running and towing duty conditions. It describes charts for designing propellers and making cavitation estimates, and it includes a procedure for designing tug propellers and making estimates of their performance. Formulae for assessing blade stresses and estimating weight and moment of inertia are also given. It summarises single screw propulsion data recently obtained at NPL, and it gives worked examples on the design and performance assessment of two tug screws, one designed for free-running conditions, the other for towing duty conditions. Sections in the series are: 1 Introduction; 2 Aspects of Propulsion; 3 General Design Considerations (all this month). Future issues will contain sections 4 Screw Design Charts; 5 Cavitation Charts; 6 Blade Stress Calculations; 7 Weight and Moment of Inertia Estimates; 8 Worked Examples and References.

by T. P. O'Brien, C.G.I.A., M.R.I.N.A., Ship Division, National Physical Laboratory

(1) Introduction

THE design of marine screw propellers has been covered in a comprehensive way in a recent book.[1] In a subsequent report[8] the practical aspects of the subject have been discussed and applications to tug screws have been given. In this series of articles, some of the practical aspects are summarised and a procedure is given, in a way more fully covered than in the original work, for designing tug screws and making estimates of their performance.

The design requirements for tug screws needed to operate at low speeds under towing duty conditions differ from those for other vessels operating at moderate speeds under free-running conditions. The conditions for different tugs can also vary. For instance, some tugs are required to produce the maximum two rope pull at zero speed of hull, others are required to operate at a given free-running speed consistent with a stipulated towing performance. Before discussing considerations affecting tug screws, it is desirable to summarise some general aspects of screw propulsion.

(2) Aspects of Propulsion

A marine screw operates by converting the greater part of the power produced by a marine engine, less transmission losses in gearing and shaft bearings, into a thrust horsepower.

The shaft horsepower minus the transmission losses is termed the delivered horsepower DHP which is applied to the screw and absorbed by it in the form of a torque Q at a shaft rate of rotation n. The thrust horsepower THP is the power applied by the screw in propelling the vessel; it is applied in the form of an axial thrust force T when the screw operates at a speed of advance v_A in propelling the vessel at a speed v_S. The delivered horsepower DHP and thrust horsepower THP are defined by:

$$(1)\quad \mathrm{DHP} = \frac{2\pi n Q}{550}$$

$$(2)\quad \mathrm{THP} = \frac{T v_A}{550}$$

The screw efficiency η is the ratio of the power applied by the screw to the power delivered to and absorbed by the screw; it is given by

$$(3)\quad \eta = \frac{\mathrm{THP}}{\mathrm{DHP}} = \frac{T v_A}{2\pi n Q}$$

Since the screw operates in a following current or "wake" behind the hull the speed of advance v_A of the screw is lower than the speed v_S of the hull. These speeds can be linked by the Taylor wake fraction w_T given in the form

$$(4)\quad v_A = (1 - w_T) v_S$$

A given marine vessel requires a certain amount of power to propel it at a specified speed. If the screw were removed and the hull were towed instead of being propelled, the force required to tow the hull at a given speed would differ from the thrust that would have been applied by the screw at the corresponding speed of advance. This is due to the fluid flow around the stern of the hull affecting the performance of the screw.

The power required in towing the hull is termed the effective horsepower EHP which is defined by

$$(5)\quad \mathrm{EHP} = \frac{R v_S}{550}$$

where R is the force required to overcome the resistance of the immersed hull and the air resistance of the superstructure,

v_S is the speed of the hull.

The thrust horsepower and effective horsepower can be linked by the hull factor ξ_H defined by

$$(6)\quad \mathrm{EHP} = \xi_H\, \mathrm{THP}$$

The propulsive efficiency η_p (or quasi propulsive coefficient QPC) is the ratio of effective horsepower to delivered horsepower. It is given by

$$(7)\quad \eta_p = \frac{\mathrm{EHP}}{\mathrm{DHP}} = \frac{R v_S}{2\pi n Q}$$

also

$$(8)\quad \eta_p = \xi_H \frac{\mathrm{THP}}{\mathrm{DHP}} = \xi_H \eta$$

The screw efficiency η_o when operating in uniform flow in open water usually differs from the screw efficiency η_B when operating in non-uniform flow behind the hull. This difference in efficiency can be expressed in the form of a relative flow factor ξ_R defined by

$$(9) \quad \eta_B = \xi_R \eta_o$$

If the principle of thrust identity is applied and it is assumed that the thrust does not vary, then

$$(10) \quad \xi_R Q_B = Q_o$$

and

$$(11) \quad \xi_R DHP_B = DHP_o$$

also, equation 8 can be re-stated in the form

$$(12) \quad \eta_p = \xi_H \eta_B = \xi_p \eta_o$$

where Q_B and DHP_B are the torque and delivered horsepower (in non-uniform flow behind the hull).

Q_o and DHP_o are the torque and delivered horsepower (in uniform flow in open water).

ξ_p is the overall hull factor linking screw efficiency η_o and propulsive efficiency η_p.

(3) General Design Considerations

The main factors influencing the design of a typical screw are outlined below.

(*a*) *Basic Design Data Required*

Hull-type, size and form, speed required, resistance data, stern details.

Engine-type, horsepower, shaft speed (or range of shaft speeds).

Stipulation (if any)—e.g., restrictions on size of screw, specified number of blades, specified material.

(*b*) *Items Considered in Designing a Screw*

Preliminary Propulsion Estimate—propulsion factors, screw efficiency, check on powering data.

Selection of Screw Characteristics—type of screw, number of blades, diameter, blade area, pitch, rake, blade thickness.

Performance Estimates—assessment of performance at design condition and over range of operating conditions.

The preliminary propulsion estimate serves two purposes; it determines the conditions under which the screw is to operate, and it relates the resistance data of the hull to the powering data of the engine.

In selecting the screw characteristics, the diameter and pitch ratio of a screw of a given type and having a given blade area ratio can be determined, and the screw efficiency can be estimated, using a screw design chart. The minimum blade area ratio necessary to avoid cavitation can be estimated using a cavitation chart and the blade thickness is based on strength calculations.

(4) Screw Design Charts

A USEFUL form of design chart in which the variables are expressed in the form of $B-\delta$ coefficients is reproduced in **Fig. 1.** The $B-\delta$ charts originally published by Taylor have subsequently been used by many research workers, notably Troost,[4] for the presentation of standard series screw data. The $B-\delta$ charts are of a form convenient for the determination of the most suitable diameter of screw to operate at specified power, speed of advance and rate of rotation; they also give the pitch ratio and screw efficiency. The $B_p-\delta$ chart, which includes a B_p coefficient related to the delivered horsepower, comprises contours of speed coeffcent δ and screw efficiency in open water η_o on co-ordinates of power coefficient B_p and pitch ratio p as shown in **Fig. 1.** The coeffcents are defined by

$$(13) \quad \delta = \frac{ND}{V_A}$$

$$(14) \quad B_p = \frac{N}{V_A{}^2}\sqrt{\frac{DHP_F}{V_A}}$$

where N is the rate of rotation in revolutions per minute
V_A is the speed of advance in knots
D is the screw diameter in feet
DHP_F is the delivered horsepower corresponding to the freshwater condition in British units.

The $B_p-\delta$ coefficients are dimensional, and B_p as defined above applies to the freshwater condition only. A more convenient form is derived by including the specific gravity. Moreover, differences between the efficiency behind the hull and that in open water can be expressed using the relative flow factor as follows:

$$(15) \quad B_p = \frac{N}{V_A{}^2}\sqrt{\frac{DHP_o}{sV_A}} = \frac{N}{V_A{}^2}\sqrt{\frac{\xi_R DHP_B}{sV_A}}$$

where DHP_o is the delivered horsepower in uniform flow in open water
DHP_B is the delivered horsepower in non-uniform flow behind the hull
s is the specific gravity of the fluid in which the screw operates ($s = 1.026$ for sea water).
ξ_R is the relative flow factor (equation 9).

An examination of the form of the efficiency contours given in the $B_p-\delta$ chart of **Fig. 1** shows that for a given value of efficiency there is a maximum value of B_p located by the intersection of the vertical tangent to the efficiency contour with the scale of B_p. The curve through the points of contact of efficiency contour and tangent represents a line of optimum efficiency corresponding to any given B_p value within the range covered by the chart. This line is termed the optimum efficiency line, and corresponding values of δ which lie on it are termed optimum speed coefficient values δ_o. The point on the optimum efficiency line corresponding to a given value of B_p determines the value of δ_o (and hence the optimum diameter D_o); it also determines the pitch ratio p and efficiency in open water η_o. A screw designed in this way should produce the maximum thrust corresponding to the given combination of delivered horsepower, rate of rotation and speed of advance, which is, of course, represented by the B_p value. If practical considerations limit the screw diameter, the pitch ratio and open efficiency can be determined from a δ value calculated using the limited screw diameter value.

Screw factors

The optimum screw diameter D_o as determined above corresponds to a screw operating in the open water condition. However, there are certain factors which affect a screw in the behind-hull condition which result in an optimum diameter somewhat less than for the open water condition. Results of some work at NPL[5] have shown that increased blade thickness can have an appreciable effect on the optimum diameter of a screw. For example, a blade thickness increase of 50 per cent due to using cast-iron instead of bronze would result in a reduction in screw diameter for optimum performance of about five per cent. Considerations are given below.

For bronze screws the speed coefficient δ_o given by the optimum line of the $B-\delta$ charts should be reduced by about four per cent for single screws and by about two per cent for twin screws.

For cast-iron screws the reduction in the optimum value of the speed coefficient should be about eight per cent for single screws and about six per cent for twin screws.

For screws fitted to tugs or trawlers and designed for free-running conditions—as distinct from towing duty screws discussed below—it may be advantageous to adopt a diameter greater than the optimum value derived from B—δ charts. This results in a small loss in efficiency with associated loss in speed at free-running conditions, but it also results in a greater thrust at towing or trawling conditions. Typical values due to an increase of seven per cent greater than optimum diameter; loss in speed at free-running conditions one per cent; increase in thrust at towing conditions six per cent. For bronze screws this increase in diameter should not be more than eight per cent while for cast-iron screws it should not exceed four per cent.

In designing towing duty screws the screw efficiency η (equation 3) used in designing free-running screws is of lttle practical use since at zero speed of advance its value is zero. Moreover, the low speeds of advance result in large values of B_p and δ, both approaching infinity as the speed of advance approaches zero. Consequently, the screw performance values are outside the range of $B_p-\delta$ charts. However, this difficulty is overcome by using an alternative form of chart (the μ-σ chart) also given by Troost,[4]. These charts are described in a recent article[9], and one of them is reproduced in **Fig 1 of Ref. 9.** In this chart, contours of open efficiency η_o, pitch ratio p and torque coefficient ϕ are given on co-ordinates of torque coefficient μ and thrust-torque ratio σ. The coefficients are given by

$$\phi = v_A\sqrt{\frac{\rho D^3}{Q}} = v_A D\sqrt{\frac{\rho D}{Q}} \tag{16}$$

$$\mu = n\sqrt{\frac{\rho D^5}{Q}} = nD^2\sqrt{\frac{\rho D}{Q}} \tag{17}$$

$$\sigma = \frac{DT}{2\pi Q} \tag{18}$$

where v_A is the speed of advance of the screw in feet per sec.

n is the rate of rotation of the screw in revolutions per sec.

D is the screw diameter in feet

Q is the torque absorbed by the screw in pounds/feet.

T is the thrust applied by the screw in pounds

ρ is the mass density of the fluid in which the screw operates (for fresh water $\rho = 1.938$, for sea water $\rho = 1.988$).

For practical purposes it is convenient to express the thrust in tons. Accordingly, the thrust torque ratio can be re-stated in the alternative form

$$\sigma = 357\,\frac{DT_u}{Q} \tag{19}$$

where T_u is the thrust in tons.

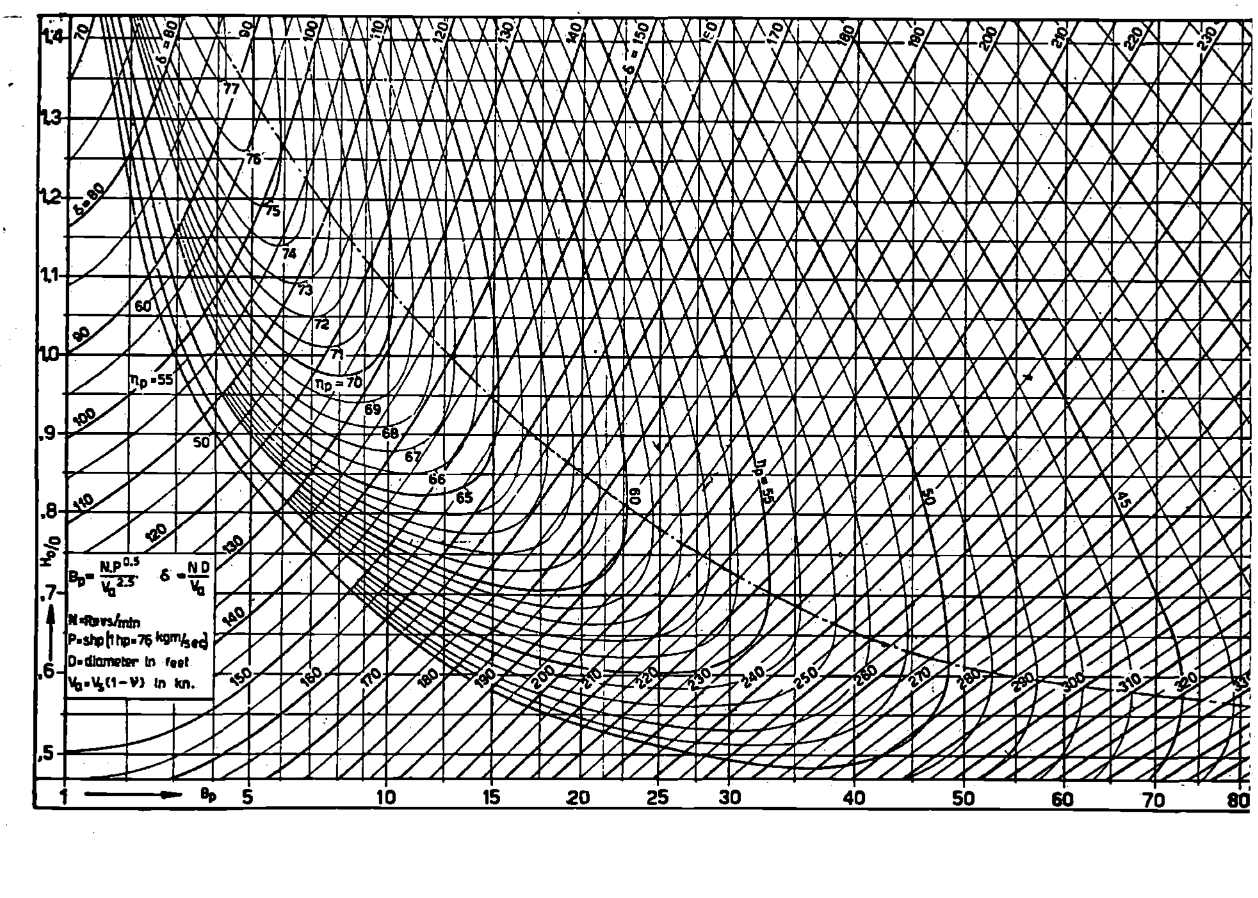
Designing towing duty

In designing towing duty screws using the μ—σ charts, the μ and ϕ coefficients are evaluated for a given screw diameter, torque, rate of rotation and speed of advance. The point on the design chart defined by the values of μ and ϕ (the latter being zero if condition is static bollard pull) enables corresponding values of pitch ratio p and thrust-torque ratio σ to be determined and this enables the thrust to be estimated using equation 19. The corresponding value of tow rope pull P_U is derived from the thrust by applying a pull-thrust ratio τ_p defined by the relation

$$P_U = \tau_p T_u \tag{20}$$

The pull-thrust ratio when based on model experiment data is generally related to a thrust deduction fraction t determined by propulsion experiments which is defined by the relation

Fig. 1



$$(21) \quad t = 1 - \frac{\text{Tow rope pull + hull resistance}}{\text{Screw Thrust}} = 1 - \frac{P_U}{T_u} - \frac{R}{T}$$

also

$$(22) \quad \tau_p = \frac{P_U}{T_u} = 1 - t - \frac{R}{T}$$

Since, for the static bollard hull condition the hull resistance is zero the thrust deduction fraction and pull-thrust ratio are given by

$$(23) \quad t = 1 - \frac{P_U}{T_u} = 1 - \tau_p$$

$$(24) \quad \tau_p = \frac{P_U}{T_u} = 1 - t$$

Similarly, for the free-running condition the two rope pull is zero, hence the thrust deduction fraction is given by

$$(25) \quad t = 1 - \frac{R}{T}$$

Screw diameters

The optimum diameter for a towing duty screw is generally the maximum value that can be selected consistent with adequate tip clearance, but a diameter chosen in this way could be too large for the free-running conditions and this might result in adverse performance. Therefore, free-running performance estimates should be made and the free running speed estimated. If this is lower than required it may be necessary to compromise by re-designing the screw for towing conditions to have a diameter smaller than originally proposed. Alternatively, a higher pitch ratio could be selected, but adoption of this procedure, though improving free-running performance would result in loss in pull since it would not be possible to absorb maximum torque under towing conditions.

In using the $\mu-\sigma$ charts for making towing performance estimates for a screw which had been designed for free running conditions the basic data required are the speed of advance v_A and the maximum torque Q. First, the torque coefficient ϕ (equation 15) is evaluated and plotted on the μ—σ chart at a point the position of which is located by the intersection of two contours, one of ϕ and the other of the pitch ratio p of the screw. This enables corresponding values of torque coefficient μ and thrust torque ratio σ to be read from the chart, and values of rate of rotation n and thrust T to be derived using the above equation for torque coefficient μ and thrust torque ratio σ (equations 16 and 18).

Speed values

In using the $B_p-\delta$ charts for making free-running propulsion estimates for a screw which had been designed for towing duty conditions, the following procedure can be applied. For a series of values of speed of hull V_s covering the expected free-running speed, corresponding values of speed of advance V_A (Equation 4) are derived, and the speed coefficient δ (equation 13) is evaluated for each value of V_A, for the maximum rate of rotation N and for the screw diameter D. For each value of the speed coefficient δ and for the pitch ratio p of the screw, corresponding values of power coefficient B_p and screw efficiency η_o are obtained from the B_p—δ chart. At the same time, the optimum value of the speed coefficient δ_o corresponding to each B_p value is obtained. Since the rate of rotation, the speed of advance and the relative flow factor are known, a value for the delivered horsepower DHP could be derived directly from each value of B_p by applying equation 15. However, for the purpose under discussion it is desirable to adopt a somewhat different procedure. If the speed of advance V_A and the rate of rotation N are both constant the delivered horsepower DHP is directly proportional to the square of the power coefficient B_p; therefore, the following equation is applicable

$$(26) \quad \text{DHP} = \text{DHP}_M \left(\frac{B_p}{B'_p}\right)^2$$

where DHP is the delivered horsepower at which the screw is operating

DHP_M is the maximum value of the delivered horsepower (i.e., the value corresponding to maximum rate of rotation and maximum torque).

B'_p is the value of the power coefficient B_p evaluated for DHP_M.

Values of effective h.p.

Having determined a series of values of screw efficiency η_o and delivered horsepower DHP over a range of speed of hull V_s, corresponding values of propulsive efficiency η_p and effective horsepower available EHP are derived using equations 12 and 7. The values of effective horsepower are plotted on a base of speed of hull together with corresponding values of effective horsepower on trial EHP_T derived from the hull resistance experiments as shown in **Fig. 5.** The intersection of the two EHP curves determines the speed at which the hull will be propelled.

(5) Cavitation Charts

A CONVENIENT form of cavitation chart from which the minimum blade area ratio required to avoid appreciable cavitation under free-running conditions can be estimated is shown in **Fig. 2.**

In applying this chart, corresponding values of operating cavitation number σ_R and thrust loading coefficient K_V are calculated, and this enables the minimum blade area ratio a_D to be estimated.

This provides a fair criterion for the minimum blade area of four-blade screws designed to operate under moderately loaded free-running conditions and of blade thickness within the usual range for bronze screws (τ = 0.050 to 0.055). For screws of greater thickness ratios, correction factors given in a paper on blade thickness variation[5] can be applied.

Correction factors can also be applied to enable the chart to be used for three or five blade screws. These correction factors are given in a recent paper[10] and in Section 6.9 of the book.[1]

Cavitation chart figures

The cavitation number σ_R and thrust loading coefficient k_V are derived from cavitation number σ_A and thrust loading coefficient k_U as follows:

$$(27) \quad \sigma_A = \frac{2(p_o - e)}{\rho\, v_A^2} = \frac{(p_o - e)}{2.76 s V_A^2}$$

$$(28) \quad \sigma_R = \frac{2(p_o - e)}{\rho v_r^2} = \frac{\sigma_A}{\gamma_7}$$

$$(29) \quad k_U = \frac{T}{\rho v_A^2 D^2} = 58.9 \eta_o \left(\frac{B_p}{\delta}\right)^2$$

$$(30) \quad k_V = \frac{T}{\rho v_r^2 D^2} = \frac{k_U}{\gamma_7}$$

where p_0 is the static pressure at the screw axis

e is the saturated vapour pressure of the water in which the screw is operating.

$(p_0 - e) = 2084 + 62.4sI$

s is the specific gravity of the water in which the screw is operating

v_r is the resultant velocity of the blade sectional element at the $x = 0.7$ radius fraction.

I is the depth of immersion of the screw axis

γ_7 is a resultant velocity conversion factor defined by the relation

$$(31)\quad \gamma_7 = 1 + \left(\frac{2.2}{J}\right)^2 = 1 + \left(\frac{\delta}{46.1}\right)^2$$

A chart which enables the cavitation numbers σ_A and σ_R to be evaluated is given in **Fig. 3.**

NPL work on trawlers

There is little published information of the performance of heavily loaded screws operating under cavitating conditions; consequently, there is a dearth of cavitation charts from which blade area estimates could be made for towing duty screws. The results of some work at NPL on trawlers[6] have shown that a screw designed for trawling conditions should have a larger blade area ratio than a corresponding screw designed for the same delivered horsepower and rate of rotation but for free running conditions. For equal margin against thrust breakdown the increase in blade area ratio should be about 20 per cent. Differences between the performance of trawler screws at trawling and free running conditions are analogous to those between tug screws at towing and free running conditions. This suggests that the cavitation chart given in **Fig. 2** can also be used in designing towing duty tug screws providing that an arbitrary increase of about 20 per cent in blade area is applied to make some allowance for the higher thrust loading associated with towing duty conditions.

(6) Blade stress calculations

A chart used in making blade stress calculations is shown in **Fig. 4**. This chart comprises the equations due to Taylor for estimating the compressive and tensile stresses S_C and S_T and additional compressive and tensile stresses due to blade rake S'_C and S'_T, all related to the blade stresses at the $x = 0.2$ radius fraction, which are given by:

$$(32)\quad S_C = \frac{S_2\,\mathrm{DHP}}{\mathrm{BND}^3 \frac{C_2}{D} \tau^2}$$

$$(33)\quad S'_C = S_1\left(\frac{S_3}{\tau} - 1\right)$$

$$(34)\quad S_T = S_C\,(0.666 + S_4 t_2/C)$$

$$(35)\quad S'_T = S_1\left(\frac{2S_3}{3} + \frac{S_5}{C_M/D} + 1\right)$$

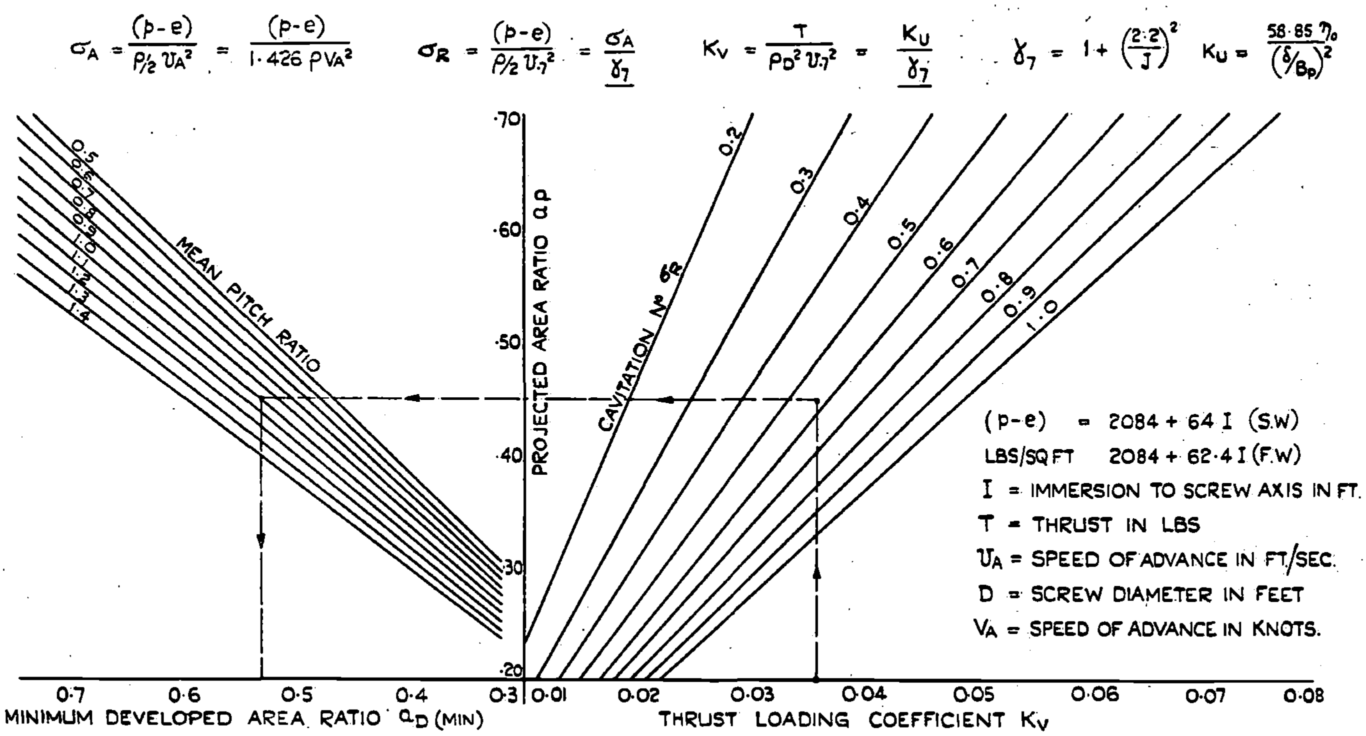
$$(36)\quad S_1 = \frac{1.54\, w\, N^2D^2}{10^7}$$

the values of S_2, S_3, S_4 and S_5 are obtained from **Fig. 4**

where DHP is the delivered horsepower

- B is the number of blades
- N is the rate of rotation in revolutions per minute
- D is the screw diameter
- C_2/D is the chord-diameter ratio at the $x = 0.2$ radius fraction
- τ is the blade thickness—diameter ratio (equivalent value at screw axis)
- t_2/C is the thickness—chord ratio at the $x = 0.2$ radius fraction
- C_M/D is the chord—diameter ratio at maximum chord
- w is the density of the screw material.

CAVITATION CHART (4 BLADE SCREWS) FIG. 2



7 Weight and moment of inertia estimates

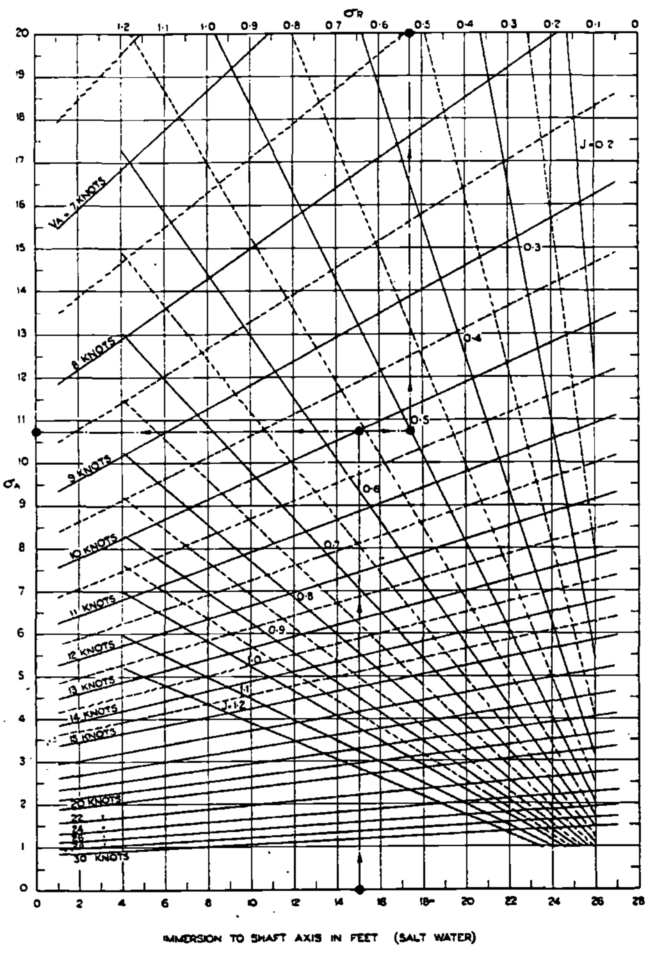
In estimating the weight and polar moment of inertia of a screw it is convenient to consider the blades and boss separately. For screws of NPL standard series type the weight of each blade W_B is given by

$$(37) \quad W_B = \frac{wD^3a_D}{3.69\ B}\ \tau \text{ pounds}$$

where a_D is the developed blade area ratio.

Fig. 3

CHART FOR EVALUATION OF σ_A AND σ_R



A convenient method for estimating the weight W_H of the boss is to divide it into three parts, each of which is assumed to have the same volume as a hollow cylinder of outer and inner diameters equal to the mean outer and inner diameters D_B and D_S of the part boss, and of length equal to the length L_B of the part boss. The weight W_H of the boss is then given by the expression:

$$(38) \quad W_H = \sum W_h = \frac{\pi}{4}\, w \sum L_B (D_B + D_S)(D_B - D_S) \text{ pounds.}$$

The polar moment of inertia I_p of a screw can be estimated for the blades and boss separately by applying a procedure similar to that followed for the weight estimate. For screws of NPL Standard Series type the polar moment of inertia per blade is given by

$$(39) \quad I_{pB} = \frac{D^2W_B}{15.7} \text{ pounds feet}^2$$

The polar moment of inertia I_{pH} of the boss can be estimated by dividing it into three parts as was done in estimating the weight of the boss. It is given by the expression

$$(40) \quad I_{pH} = \frac{1}{8} \sum W_h \left(D_B^2 + D_S^2\right) \text{ pounds feet}^2$$

Entrained water effects

It should be noted that values of the polar moment of inertia calculated using the foregoing formulae do not include the effects of entrained water. If desired, estimates of entrained water effects can be estimated using the data given in the paper[7] by Burrill and Robson.

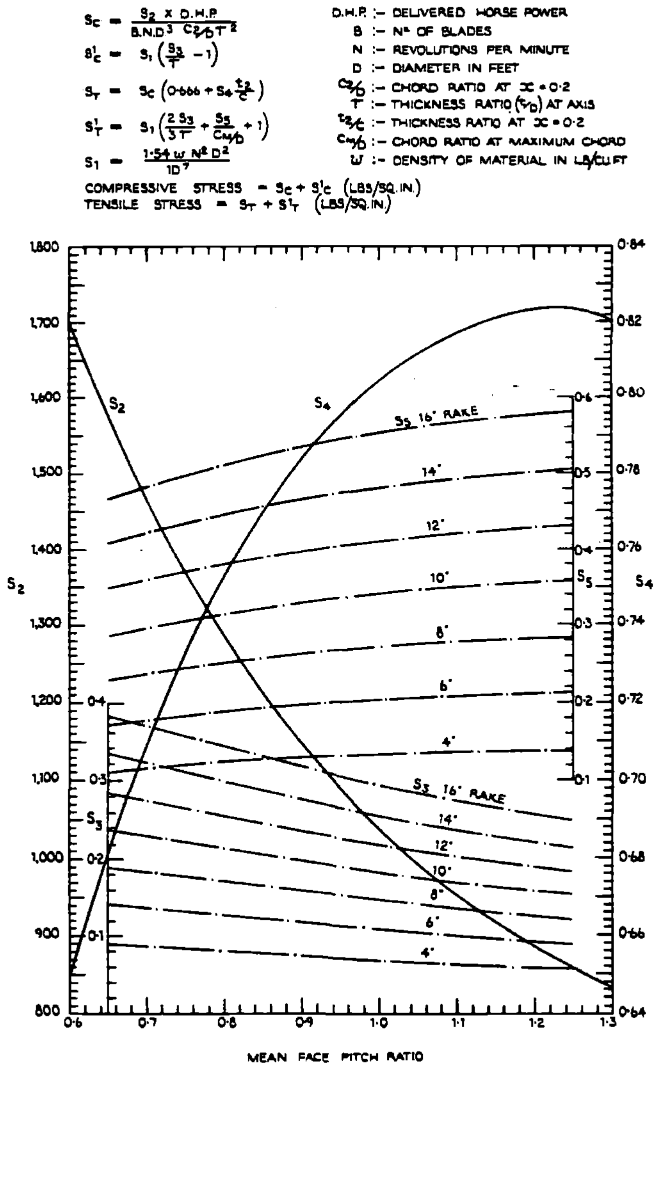
(8) Design Example

It is required to prepare two screw designs for a single screw tug.

The first is to be designed to absorb maximum power for free running conditions, and towing performance estimates are to be made for zero speed of hull.

Fig. 4

STRENGTH OF SCREWS (TAYLOR)



The second is to be designed to absorb maximum power at zero speed of hull, and propulsion estimates are to be made for free-running conditions.

The screws are to be of NPL standard type (Section 3.7 of the book[1]), designed using the Troost B series $B_p - \delta$ charts (Section 3.8) and $\bar{\mu} - \sigma$ charts (Section 3.10).

The blade area ratios are to be determined using the blade area chart for moderately loaded screws (Section 6.4) shown in **Fig. 2**, and the blade thicknesses are to be based on the Taylor strength criterion (Section 8.7) using the chart shown in **Fig. 4**.

Design Data

Hull—Single-screw tug; length 100 ft., breadth 28 ft., draught (aft) 14 ft., rake of keel 4 ft. aft, displacement (mld.) 483 tons, block coefficient 0.502 (other particulars as Model 4033B, ref. 2).

Trial Speed 12.5 knots. Ship-model correlation factor $f_M = 1.10$.

Speed of hull (knots)

V_s	9½	10	10½	11	11½	12	12½	13

Effective h.p. predicted from model experiments

EHP_M	140	176	219	273	349	462	627	873

Effective h.p. on trial

EHP_T =	154	194	241	300	384	508	690	960

$(EHP_T = f_M EHP_M = 1.10\ EHP_M)$.

Engine—Diesel. Delivered h.p. at screw DHP = 1,100, rate of rotation N_F = 200 r.p.m. corresponding maximum torque = 28,900 pounds feet.

For free-running conditions N = 0.98 N_F (2 per cent wake scale effect, see Section 4.9). = 196

Stern Details: Streamlined Rudder, Shaft Immersion I = 7.7 feet.

Stipulation. Max. Dia. 9.0 feet, number of blades 4, material bronze.

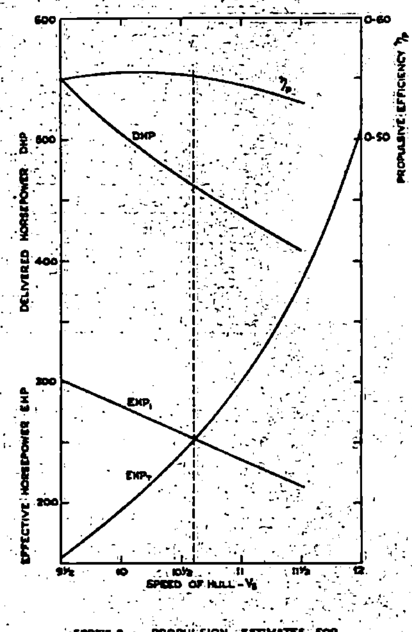
Design Conditions. Screw 1 to be designed to absorb maximum power under trial conditions when running free at a trial speed of 12½ knots.

Screw 2 is to be designed to absorb maximum power at zero speed of hull.

Screw 1—Design Calculations—Free-Running Conditions

In making the screw design calculations given Table 2, the power coefficient B_p and speed ratio δ are evaluated, and corresponding values of pitch ratio p_T and screw efficiency η_o are obtained from the $B_p - \delta$ charts (Section 3.4). Values of propulsive efficiency η_D are derived from screw efficiency η_o using the propulsion factors given in Table 1, and this enables the available effective h.p. EHP_1 to be estimated and compared with the effective h.p. required on trial EHP_T; hence determining whether the specified trial speed would be attained.

Fig. 5.



SCREW 2 — PROPULSION ESTIMATES FOR FREE–RUNNING CONDITIONS

x	P/D	C/D	h_D/D	t/D	h_T/C
1·0	1·00	0	–·053	·0030	0·500
0·95	1·00	0·135	0·020	·0051	0·500
0·90	1·00	0·184	0·051	·0072	0·500
0·80	1·00	0·241	0·092	·0114	0·500
0·70	1·00	0·269	0·118	·0156	0·493
0·60	1·00	0·279	0·132	·0198	0·475
0·50	0·99	0·276	0·137	·0240	0·450
0·40	0·95	0·263	0·136	·0282	0·420
0·30	0·90	0·241	0·129	·0324	0·387
0·20	0·85	0·208	0·116	·0366	0·350

BASIC SCREW N.P.L. STANDARD SCREW SERIES SHEET 1

Nº. OF BLADES 4 BLADE AREA RATIO 0·5 PITCH RATIO 1·0

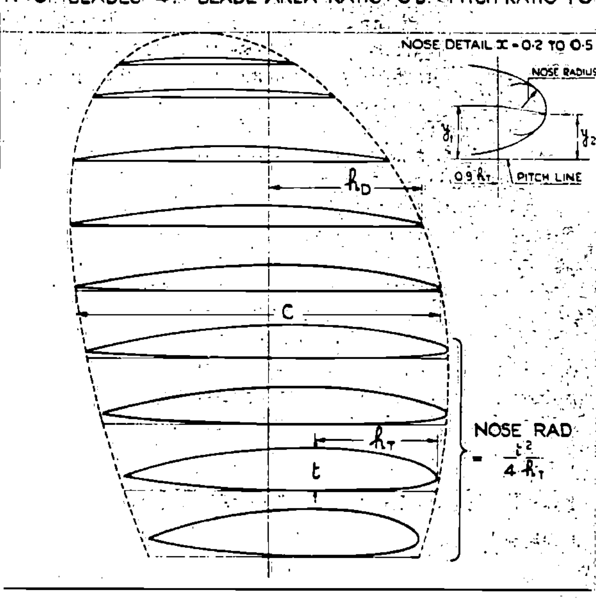


Fig. 6. *Sheet 1 of the geometrical data of the basic 4-blade screw of the NPL Standard Series.* (*See also Fig. 7, p. 32*).

CYLINDRICAL SECTION OFFSETS FROM PITCH LINE

RAD		FROM MAX t TO T.E					MAX	% DISTANCE FROM MAX. THICKNESS TO L.E											RAD
x		T.E.	80	60	40	20	t	20	40	60	70	80	85	90	95	L.E.	y_1	y_2	x
0·95	BACK	0·163	0·464	0·699	0·866	0·967	1·000	0·967	0·866	0·699	0·590	0·464	0·395	0·322	0·245	0·163		0·082	0·95
	FACE	0 -	----	----	----	----	- 0 -	----	----	----	----	----	----	----	----	-0			
0·90	BACK	0·116	0·434	0·682	0·859	0·965	1·000	0·965	0·859	0·682	0·567	0·434	0·361	0·284	0·202	0·116		0·058	0·90
	FACE	0 -	----	----	----	----	- 0 -	----	----	----	----	----	----	----	----	- 0			
0·80	BACK	0·073	0·407	0·669	0·852	0·963	1·000	0·963	0·852	0·669	0·546	0·407	0·330	0·249	0·163	0·073		0·037	0·80
	FACE	0 -	----	----	----	----	0 -	----	----	----	----	----	----	----	----	- 0			
0·70	BACK	0·102	0·436	0·687	0·859	0·965	1·000	0·968	0·866	0·698	0·590	0·465	0·390	0·305	0·210	0·102		0·076	0·70
	FACE	0·048	0·013	0 -	----	----	- 0 -	----	----	----	----	- 0	0·002	0·004	0·012	0·048			
0·60	BACK	0·171	0·477	0·712	0·875	0·970	1·000	0·975	0·881	0·737	0·647	0·530	0·465	0·386	0·298	0·171		0·151	0·60
	FACE	0·129	0·056	0·016	0 -	----	- 0 -	----	----	- 0	0·003	0·012	0·020	0·034	0·058	0·129			
0·50	BACK	0·233	0·521	0·742	0·892	0·975	1·000	0·978	0·900	0·774	0·692	0·591	0·531	0·463	0·377		0·270	0·218	0·50
	FACE	0·199	0·100	0·045	0·013	0·003	0	0	0·004	0·012	0·021	0·037	0·049	0·068	0·101				
0·40	BACK	0·289	0·565	0·769	0·903	0·977	1·000	0·979	0·915	0·804	0·732	0·637	0·582	0·523	0·444		0·307	0·274	0·40
	FACE	0·259	0·139	0·068	0·025	0·005	0	0·003	0·012	0·030	0·045	0·067	0·084	0·108	0·146				
0·30	BACK	0·338	0·598	0·787	0·911	0·979	1·000	0·981	0·924	0·826	0·759	0·676	0·626	0·569	0·497		0·343	0·325	0·30
	FACE	0·312	0·178	0·087	0·033	0·007	0	0·005	0·020	0·048	0·070	0·100	0·121	0·149	0·192				
0·20	BACK	0·386	0·630	0·805	0·919	0·981	1·000	0·984	0·932	0·844	0·783	0·708	0·662	0·608	0·538		0·380	0·375	0·20
	FACE	0·364	0·210	0·105	0·041	0·009	0	0·007	0·029	0·066	0·094	0·132	0·157	0·190	0·237				

In making the cavitation estimates, the cavitation numbers σ_A and σ_R, and thrust leading coefficients k_U and k_V, are evaluated; and this enables a blade area ratio a_D to be selected based on the minimum value obtained from the blade area chart (**Fig. 2**). Next, the blades are determined using the Taylor strength criterion (**Fig. 4**). Since the screw is to be of NPL standard type, the value of the pitch ratio obtained using the Troost B series data is corrected to make allowance for the different blade section shapes as discussed in Section 4.4 of the book.[1]

Screw 2—Design Calculations—Towing Duty Conditions

In making the screw design calculations given in Table 3, the torque coefficient μ is evaluated and corresponding values of pitch ratio p_T and thrust-torque ratio σ are obtained from the $\mu - \sigma$ charts (Section 3.10). The value of the thrust T_U is determined from the thrust-torque ratio σ, and this enables the pull P_U to be derived using the value of the pull-thrust ratio τ_p given in Table 1. The blade area ratio a_D is derived from the corresponding value for the first screw by applying an arbitrary increase of 20 per cent to make some allowance for the higher thrust loading associated with towing duty conditions, as discussed in Section 5, above.

The blade thicknesses are determined and the blade section shape corrections are made using the same procedure as for the first screw. Since the blade thickness ratio ($\tau = 0.052$) is appreciably greater than that of the basic Troost B screw series ($\tau = 0.045$), a pitch correction factor is applied to make allowance for increased blade thickness, as discussed in Section 4.3 of the Book[1].

Screw 1—Towing Performance Estimates

In making the towing performance estimates given in Table 4, corresponding values of torque coefficient μ and thrust-torque ratio σ are obtained from the μ–σ charts (Section 3.10). The rate of rotation n at which the screw is operating and the resulting delivered horsepower DHP are determined from the torque coefficient μ. The thrust T_U is determined from the thrust-torque ratio σ, and this enables the pull P_U to be derived using the value of the pull-thrust ratio τ_p given in Table 1.

Screw 2—Propulsion Estimates—Free-Running Conditions

In making the free-running propulsion estimates given in Table 5, the procedure outlined in Section 4 above, is followed using the B-4-55 $B_p - \delta$ chart. In deriving the screw efficiency η_o, a reduction of 1 per cent is applied: this makes allowance for departure from basic blade area ratio ($a_D = 0.6$, basic value 0.55) and increase in blade thickness ratio ($\tau = 0.052$, basic value 0.045). The former correction is obtained by interpolating between efficiency values obtained for screws of different blade area ratio ($a_D = 0.55$, $a_D = 0.70$), while the latter is determined using the data given in the paper[5], as described in Section 4.3 of the book[1].

The results of the calculations are plotted in the form of values of propulsive efficiency η_p, delivered h.p. DHP and effective h.p. available EHP_1, together with corresponding values of effective h.p. on trial EHP_T, all on a base of speed of hull V_s, as shown in **Fig. 5**. The speed at which the hull will be propelled is given by the value of the speed of hull located by the intersection of the two curves of effective horsepower.

Screw 2—Screw Manufacturing Data

The screw manufacturing data for Screw 2 are given in Table 6 in the form of a series of correction factors to be applied to the geometrical data of the basic screw of the NPL Standard Series shown in **Figs. 6 and 7.**

Table 1 Propulsion Data for a Single Screw Tug

Hull: 100 ft. × 28 ft. × 12 ft. draught mean (14 ft. draught aft)
Block coefficient 0.502. Other particulars as Model 4033 (reference 2)

FREE-RUNNING CONDITIONS

Hull	Screw	Diameter (ft.)	Design condition	Design: Delivered h.p.	Design: Rate of rotation (r.p.m.)	Operating (at design R.P.M.): Delivered h.p.	Operating (at design R.P.M.): Speed of hull (knots)	Wake fraction	Thrust deduction fraction	Hull factor	Relative flow factor	Overall hull factor	Propulsive efficiency
		D		DHP	N	DHP	V_s	w	t	ξ_H	ξ_R	ξ_P	η_p
4033A	W157	9.5	Bollard	1,100	130	487	11.24	0.210	0.231	0.97	1.02	0.99	0.686
	W158	9.0		1,100	200	452	10.66	0.203	0.220	0.98	1.01	0.99	0.570
	W159	7.8		1,100	270	506	10.80	0.210	0.205	1.01	1.00	1.01	0.544
4033B	W160	10.25		1,100	130	462	11.07	0.213	0.239	0.97	1.02	0.99	0.676
	W174	10.25	Free-running	1,100	130	1,073	12.59	0.223	0.223	1.00	1.02	1.02	0.689
	W175	9.0		1,100	200	1,061	12.51	0.225	0.206	1.02	1.00	1.02	0.654
	W156	7.8		1,100	270	1,081	12.45	0.234	0.200	1.04	1.00	1.04	0.615

TOWING DUTY CONDITIONS
BOLLARD PULLS AT ZERO SPEED OF HULL

Hull	Screw	Diameter (ft.)	Design condition	Design: Delivered h.p.	Design: Rate of rotation (r.p.m.)	Operating (at design torque): Delivered h.p.	Operating (at design torque): Rate of rotation (r.p.m.)	Thrust (tons)	Pull (tons)	Thrust deduction fraction	Pull-thrust ratio
		D		DHP	N	DHP	N	T_U	P_u	t	P_u/T_U
4033A	W157	9.5	Bollard	1,100	130	1,100	130	15.35	15.05	0.020	0.980
	W158	9.0		1,100	200	1,100	203	16.00	15.60	0.025	0.975
	W159	7.8		1,100	270	1,100	275	14.30	14.00	0.021	0.979
4033B	W160	10.25		1,100	130	1,100	129	16.95	16.60	0.021	0.979
	W174	10.25	Free-running	1,100	130	791	94	12.00	11.65	0.029	0.971
	W175	9.0		1,100	200	762	139	11.85	11.60	0.021	0.971
	W156	7.8		1,100	270	807	198	11.60	11.35	0.022	0.978

Table 2 Screw 1—Design Calculations: Free-running Conditions

Design Conditions: DHP = 1,100, N = 196 R.P.M., V_s = 12.5 knots,
D = 9 feet, I = 7 7 feet

Propulsion Factors (Table 3) $w = 0.225$, $t = 0.206$, $\xi_p = 1.0$, $\xi_R = 1.02$

Screw	B_p (1)	δ (2)	δ_o (3)	p_T (3)	η_o (3)	η_P (4)	EHP_1 (5)	EHP_T (6)	σ_A (7)	k_u (8)	γ (9)	σ_R (10)	k_V (11)	a_D (12)
B-4-40 B-4-55	22 22	182 182	189 182	0.840 0.830	0.635 0.610	0.647 0.620	712 685	690 690	9.75	0.532	16.6	0.587	0.0321	0.50

for $a_D = 0.5$ $p_T = 0.833$ (interpolation between $a_D = 0.4$ and 0.55)

(1) (equation 14) $B_P = \dfrac{N}{VA^2}\sqrt{\dfrac{\xi_R DHP}{sV_A}}$

(2) (equation 13) $\delta = \dfrac{ND}{V_A}$

(3) Values from B_P–δ charts (Section 3.4)

(4) (equation 12) $\eta_P = \xi_p \eta_o$

(5) (equation 7) $EHP_1 = \eta_p DHP$

(6) Effective h.p. on trial (given hull-resistance data)

(7) (equation 27) $\sigma_A = \dfrac{(P_o - e)}{2.76sV_A^2}$

(8) (equation 29) $k_u = 58.9\eta_o\left(\dfrac{B_P}{\delta}\right)^2$

(9) (equation 31) $\gamma = 1 + \left(\dfrac{\delta}{46,1}\right)^2$

(10) (equation 28) $\sigma_R = \sigma_A/\gamma$

(11) (equation 30) $k_V = k_u/\gamma$

(12) Value of blade area ratio from Blade Area Chart (Fig. 2)

Strength Calculations

C.2/D = 0.208 (Fig. 6), S_2 = 1,240,
S_3 = 0.21 (Fig. 4) assumed thickness ratio $\tau = 0.047$

(equation 32) $$Sc = \frac{S_2 DHP}{\tau^2 BND^3 C.2/D} = 5,100$$

(equation 33) $$S^1_C = S_1\left(\frac{S_3}{\tau} - 1\right) = 910$$

$$S_C + S^1_C = 6,010$$

Correction factors for departure from standard—for blade sections of M.S. type increase pitch ratio by 2½ per cent (Section 4.4).

Particulars of Screw: Diameter D = 9.0 ft.
Number of Blades B = 4
Blade Area Ratio $a_D = 0.50$
Pitch Ratio $p = 0.853$
Blade Thickness Ratio (at axis) $\tau = 0.047$
Rake Ψ = 10 deg.

Table 3 Screw 2—Design Calculations: Towing Duty Conditions

Design Conditions: DHP = 1,100, N = 200 R.P.M., Q = 28,900 lb. ft.
$V_s = 0$ n = 3.333 RPS, D = 9.0 ft.

Blade Area Ratio $a_D = 0.6$ (i.e. 0.5 + 20%, see Section 5)

Screw	ϕ (1)	μ (2)	p_T (3)	σ (3)	$\frac{T_U}{\text{Tons}}$ (4)	$\frac{Pu}{T_U}$ (5)	$\frac{Pu}{\text{Tons}}$ (6)
B-4-55 B-4-70	0 0	6.71 6.71	0.565 0.570	1.67 1.64			
for a_D =	0.6		0.565	1.66	14.95	0.975	14.60

(1) (equation 16) $\phi = v_A D\sqrt{\dfrac{\rho D}{Q}}$ $v_A = 0$, hence $\phi = 0$

(2) (equation 17) $\mu = nD^2\sqrt{\dfrac{\rho D}{Q}}$

(3) Values from μ–σ charts (Section 3.10)

(4) (equation 19) $T_U = \dfrac{Q\sigma}{357D}$

(5) Pull-thrust ratio τ_p (Table 1)

(6) (equation 20) $Pu = \tau_p T_U$

From results of strength calculations similar to those given in Table 2, $\tau = 0.052$

Correction factors for departure from standard—
for blade sections of M.S. type; increase pitch ratio by 3 per cent (Section 4.4)
for increased blade thickness; reduce pitch ratio by ½ per cent (Section 4.3)

Particulars of Screw: Diameter D = 9.0 ft.
Number of blades B = 4
Blade Area Ratio $a_D = 0.60$
Pitch Ratio $p_T = 0.58$
Blade Thickness Ratio (at axis) $\tau = 0.052$
Rake Ψ = 10 deg.

Table 4 Screw 1—Towing Performance Estimates

Operating Conditions: Maximum torque Q = 28,900 lb. ft at zero speed of advance, $\phi = 0$

Screw	p_t (1)	μ (2)	σ (2)	n RPS (3)	N RPM (4)	DHP (5)	T_U Tons (6)	$\frac{Pu}{T_U}$ (7)	$\frac{Pu}{\text{Tons}}$ (8)
B–4–40 B–4–55	0.84 0.83	5.10 4.70	1.37 1.28						
for $a_D = 0.50$		4.83	1.31	2.40	144	792	11.82	0.971	11.47

(1) Pitch ratio values from Table 2
(2) Values from μ–σ charts (Section 3.10)
(3) (equation 17) $n = \frac{\mu}{D^2}\sqrt{\frac{Q}{\rho D}}$
(4) $N = 60n$
(5) $DHP = \frac{2\pi nQ}{550}$
(6) (equation 19) $T_U = \frac{Q\sigma}{357D}$
(7) Pull-thrust ratio τ_P (Table 1)
(8) (equation 20) $Pu = \tau_p T_U$

Table 5 Screw 2—Propulsion Estimates: Free-running Conditions

Operating Condition: Maximum rate of rotation N_F = 200 R.P.M.
(N = 196 R.P.M.)

Propulsion Factors (Table 1) $w = 0.225$, $t = 0.206$, $\xi_R = 1.0$, $\xi_P = 1.02$
Delivered h.p. (Maximum) $DHP_M = 1{,}100$

V Knots	V_A (1)	$V_A^{2.5}$	δ (2)	B_P (3)	$\eta_o{}^1$ (3)	η_o (4)	η_p (5)	$B_P{}^1$ (6)	$\frac{DHP}{DHP_M}$ (7)	DHP (7)	EHP_1 (8)
9½	7.36	1,470	240	31.0	0.545	0.540	0.550	43.7	0.502	552	303
10	7.75	1,672	228	26.0	0.550	0.545	0.555	38.4	0.459	503	279
10½	8.14	1,890	217	22.3	0.547	0.542	0.552	34.0	0.430	474	263
11	8.52	2,120	207	19.0	0.540	0.535	0.545	30.3	0.394	434	236
11½	8.91	2,370	198	16.5	0.525	0.520	0.530	27.1	0.371	408	216

(1) (equation 4) $V_A = (1-w)V_s$
(2) (equation 13) $= \frac{ND}{V_A} = \frac{1.764}{V_A}$
(3) Values from B—4—55 B_p—δ chart (Fig. 1) at pitch ratio $p_T = 0.565$ and δ from column (2)
(4) Chart values reduced by 1 per cent (departure from basic blade area ratio and thickness ratio)
(5) (equation 12) $\eta_p = \xi_p\eta_o = 1.02\,\eta_o$
(6) (equation 15) $B_p{}^1 = \frac{N}{V_A{}^2}\sqrt{\frac{\xi_R DHP_M}{s\,VA}} = \frac{6{,}415}{V_A{}^{2.5}}$
(7) (equation 26) $DHP = DHP_M\left(\frac{Bp}{B_p{}^1}\right)^2$
(8) (equation 7) $EHP_1 = \eta_p\,DHP$
From Fig. 5 V = 10.6 knots, DHP = 460, $\eta_p = 0.55$

Table 6 Screw 2—Screw Manufacturing Data—Particulars of Ship Screw

Model Hull No. 4033B

The screw recommended for the ship should have the following main features:—

Number of blades		4
Diameter		9.0 feet
Pitch ratio at tip	p_T	0.58
Blade area ratio		0.60
Thickness ratio at axis	t/D	0.052
Rake angle	Ψ_o	10
Boss diameter ratio at rake line	d_B	0.167
Material		Bronze

The detailed features of the screw should resemble those of the Basic Screw of the N.P.L. Standard Series, as shown on Sheets 1 and 2.

For the ship screw the following modifications to the Basic Screw are required:—

(1) Correction to basic P/D values to allow for differing Pitch Ratio i.e. (P/D) ship screw = (P/D) basic screw × 0.58.

(2) Correction to basic C/D values to allow for differing Blade Area Ratio,

$$\text{i.e. (C/D) ship screw} = \text{(C/D) basic screw} \times \frac{0.6}{0.5}$$

$$\text{and } \left(\frac{h_D}{D}\right) \text{ ship screw} = \left(\frac{h_D}{D}\right) \text{ basic screw} \times \frac{0.6}{0.5}$$

(3) Correction to basic t/D values to allow for differing thickness,

$$\text{i.e. } (t/\text{D}) \text{ ship screw} = (t/\text{D}) \text{ basic screw} \times \frac{0.052}{0.045}$$

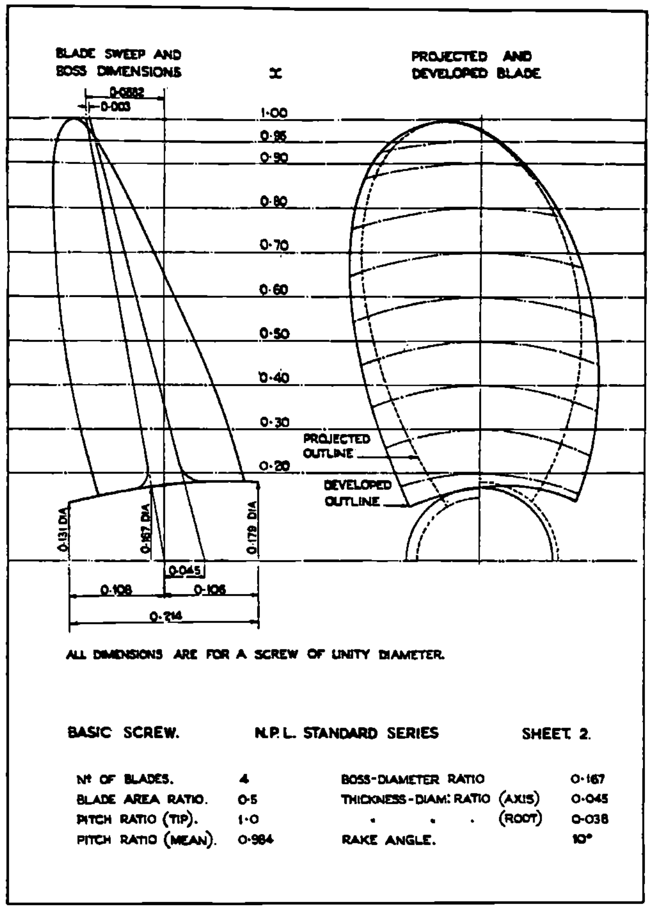


Fig. 7

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Propeller Design and Two-Speed Gearboxes

. . . with particular reference to tugs and trawlers

by T. P. O'Brien, C.G.T.A., A.M.R.I.N.A., Ship Division, National Physical Laboratory.

This two-part article discusses the differences in performance of screws designed for free-running conditions and towing duty conditions, the former when towing and the latter when running free. It shows that significant improvements in performance for both types of screw can be achieved by using two-speed gearboxes enabling the optimum rate of rotation to be chosen for both free-running and towing conditions. Equations are derived and coefficients are given in part one to enable design and operating conditions to be chosen to give optimum performance.

1. Introduction

Unlike those for other vessels, screws for tugs and trawlers are dual purpose propulsion devices since, in addition to operating at free-running conditions, they are also required to run at low speed towing duty conditions. Some screws are designed to give best performance at free-running conditions and do not operate so efficiently when towing. Others designed to give best performance at towing duty conditions, suffer adverse performance when free-running.

For the former, the loss in towing performance could be a 20 per cent reduction in towing pull, while for the latter the loss in free-running performance could be a 15 per cent reduction in ship speed. Some screws are, of course, designed to operate at conditions that are a compromise between free-running and towing.

A marine screw can be designed to absorb a stipulated horsepower when running at a given rate of rotation and speed of advance in propelling the hull. If the screw is designed for free-running conditions, it has a moderate pitch ratio and operates at a moderate speed of advance when running at its design condition. However, when the screw operates at low speed towing duty conditions, the maximum torque applied by the engine will be reached at a low rate of rotation, consequently there will be a reduction in delivered horsepower resulting in low thrust and pull, *i.e.*, for towing conditions the screw is overpitched.

Conversely, if the screw is designed for towing conditions, it has a low pitch ratio and operates at a low speed of advance when running at its design condition. However, when this screw operates at moderate speed free-running conditions, the maximum rate of rotation will be reached at a torque value substantially lower than the maximum. Therefore, the engine will not be able to apply maximum torque, and there will be a reduction in delivered horsepower resulting in low free-running speed, *i.e.*, for free running conditions the screw is underpitched.

The foregoing restrictions apply if the screw is driven either directly from the engine or via a single-speed gear box; however, significant improvements in performance can be achieved by introducing a two speed gear box. For screws designed for free-running conditions, a second gear can be chosen to enable the screw to run at a higher rate of rotation, and so operate at maximum power when towing. Similarly, for screws designed for towing duty conditions, a second gear ratio is chosen to enable the screw to run at a higher rate of rotation and so operate at maximum power when free-running.

2. Basic equations and μ–σ coefficients

A form of chart convenient for designing and making performance estimates for both towing duty and free-running screws is the μ–σ chart given in a paper by Troost.[1] Some of the charts given by Troost are reproduced in a book[2] and in a report[3] by the present author, where worked examples are included illustrating practical applications. In this chart, contours of open water efficiency η_0, pitch ratio p and torque coefficient ϕ are given on co-ordinates of torque coefficient μ, and thrust-torque ratio σ, as shown in **Fig. 1.** In their basic form, the coefficients are given by:—

$$(1)\quad \phi = v_A\sqrt{\frac{\rho D^3}{Q_0}} = v_A D\sqrt{\frac{\rho D}{Q_0}}$$

$$(2)\quad \mu = n\sqrt{\frac{\rho D^5}{Q_0}} = nD^2\sqrt{\frac{\rho D}{Q_0}}.$$

$$(3)\quad \sigma = \frac{DT}{2\pi Q_0}$$

$$(4)\quad \eta_0 = \frac{Tv_A}{2\pi n Q_0}$$

If desired the thrust-torque ratio and screw efficiency can be linked by the advance coefficient J defined by

$$(5)\quad J = \frac{v_A}{nD} = \frac{\eta_0}{\sigma}$$

where v_A is the speed of advance of the screw in feet per second

n is the rate of rotation of the screw in revolutions per second

D is the screw diameter in feet

Q_0 is the torque absorbed by the screw when running in open water in pounds feet

T is the thrust applied by the screw in pounds

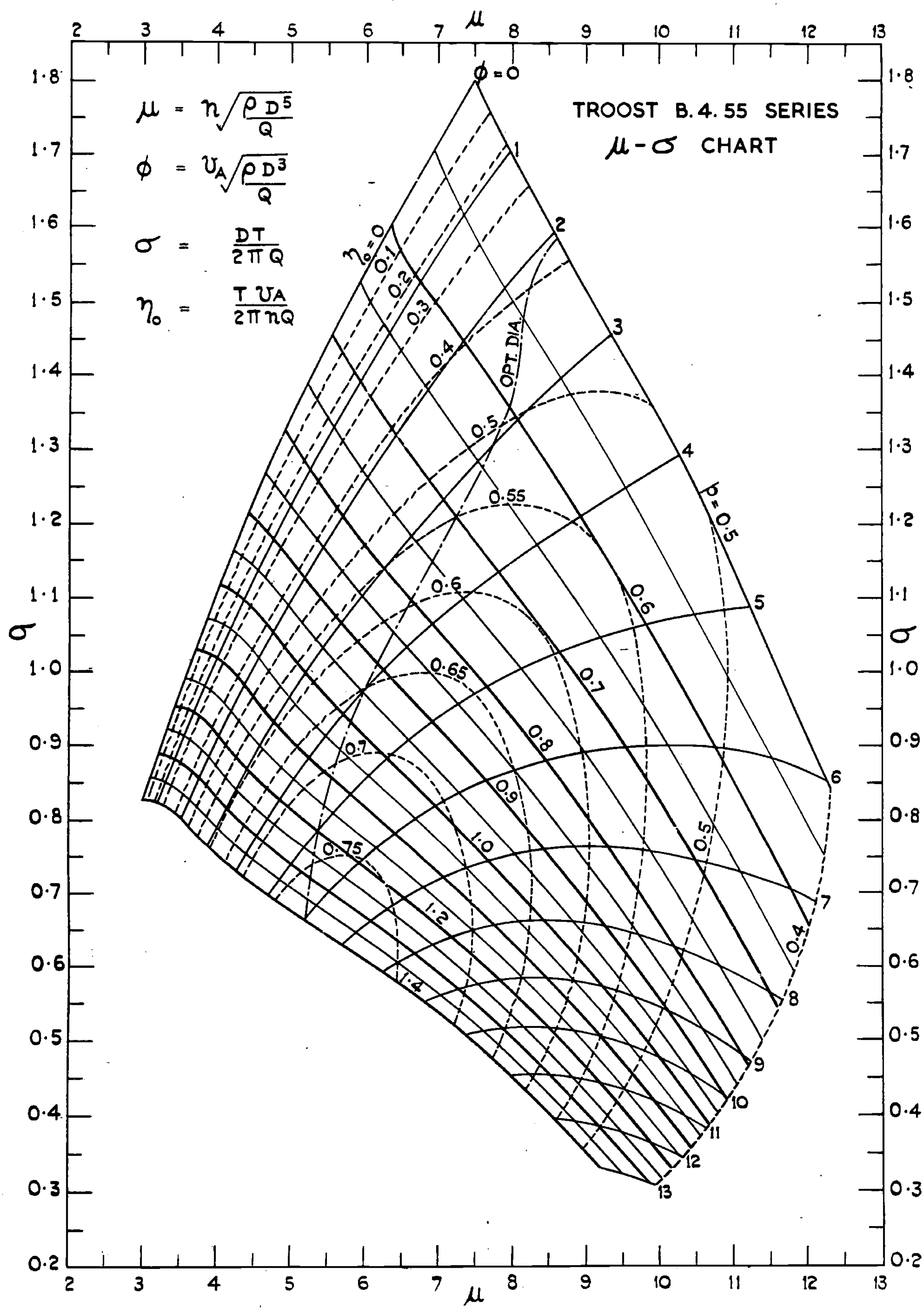
ρ is the mass density of the fluid in which the screw operates

(for fresh water $\rho = 1.938$, for sea water $\rho = 1.988$)

For practical purposes it is convenient to express speed of advance in knots; rate of rotation in revolutions per minute and thrust in tons; moreover, it is desirable to apply the principle of thrust identity as discussed in Section 2.6 of the book.[2]

Accordingly, the coefficients are re-stated in the form

$$(6)\quad \phi = 1.689\, V_A D\sqrt{\frac{\rho D}{\Sigma_R Q}}.$$



TROOST B. 4. 55 SERIES
μ - σ CHART
$\mu = n\sqrt{\frac{\rho D^5}{Q}}$
$\phi = v_A\sqrt{\frac{\rho D^3}{Q}}$
$\sigma = \frac{DT}{2\pi Q}$
$\eta_o = \frac{T v_A}{2\pi n Q}$
$\phi = 0$
$\eta_o = 0$
OPT. DIA
$p = 0.5$
μ
σ

Fig. 1

$$(7) \quad \mu = \frac{ND^2}{60}\sqrt{\frac{\rho D}{\Sigma_R Q}}$$

$$(8) \quad \sigma = \frac{357DT_U}{\Sigma_R Q}$$

$$(9) \quad J = \frac{101.3V_A}{ND}$$

$$(10) \quad \eta_0 = \frac{602T_U V_A}{N\Sigma_R Q} = J\sigma$$

where V_A is the speed of advance of the screw in knots

N is the rate of rotation of the screw in revolutions per minute

T_U is the thrust applied by the screw in tons

Σ_R is the relative flow factor linking the screw efficiency η_B when operating in non-uniform flow behind the hull, and the screw efficiency η_0 when operating in uniform flow in open water.

Applying the principle of thrust identity

$$(11) \quad \Sigma_R Q_B = Q$$

In applying the μ–σ coefficients in designing a towing duty screw of given diameter to absorb a stipulated delivered horsepower DHP when running at given rate of rotation N and speed of advance V_A, first, the torque Q is computed using the formula

$$(12) \quad Q = \frac{33{,}000\ DHP}{2\pi N}$$

This enables torque coefficient ϕ and μ to be evaluated, and the point on the μ–σ design chart defined by these values enables corresponding values of pitch ratio ρ and thrust-torque ratio σ to be determined. The thrust T_U is calculated using equation 8, and the related tow rope pull P_U is derived from the thrust by applying a pull-thrust ratio τ_P defined by the relation

$$(13) \quad P_U = \tau_P T_U$$

A worked example following the above procedure is given in Table 5 of the report.[3]

The foregoing considerations apply if the rate of rotation is fixed. However, if it is possible to select a set of gear ratios to give a range of values of rate of rotation N_1 by applying a factor k to the basic value of rate of rotation N_1 and applying the condition that the power remains constant, the following relations can be derived:—

$$(14) \quad N = kN_1$$

$$(15) \quad Q = \frac{Q_1}{k}$$

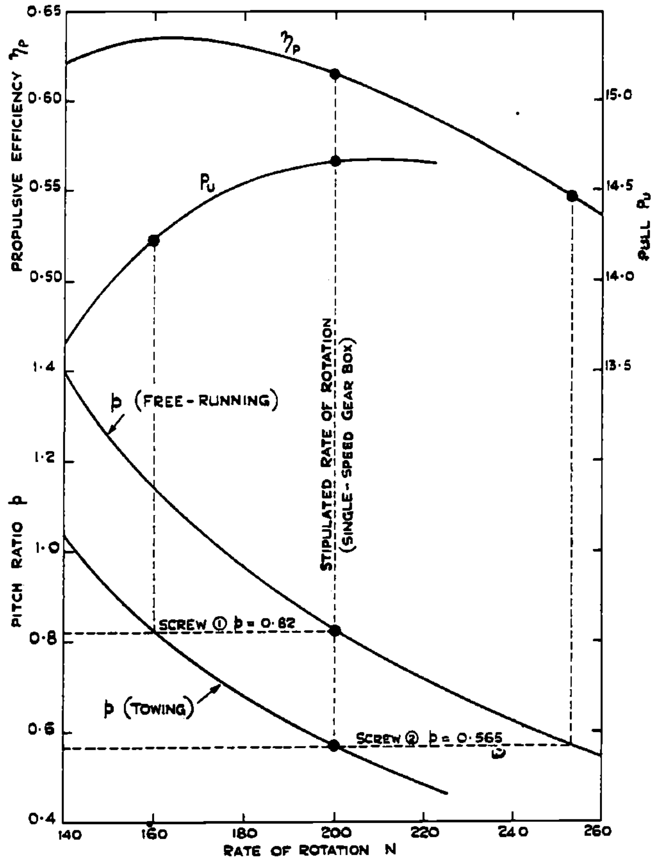
where Q_1 is the basic value of the torque corresponding to the basic rate of rotation N_1 which, when substituted in the equations for torque coefficients enable these equations to be re-stated in the form

$$(16) \quad \phi = 1.689V_A D\sqrt{\frac{\rho D_k}{\Sigma_R Q_1}} = k^{\frac{1}{2}}\,\phi_1$$

$$(17) \quad \mu = \frac{kND^2}{60}\sqrt{\frac{\rho D_k}{\Sigma_R Q_1}} = k^{3/2}\mu_1$$

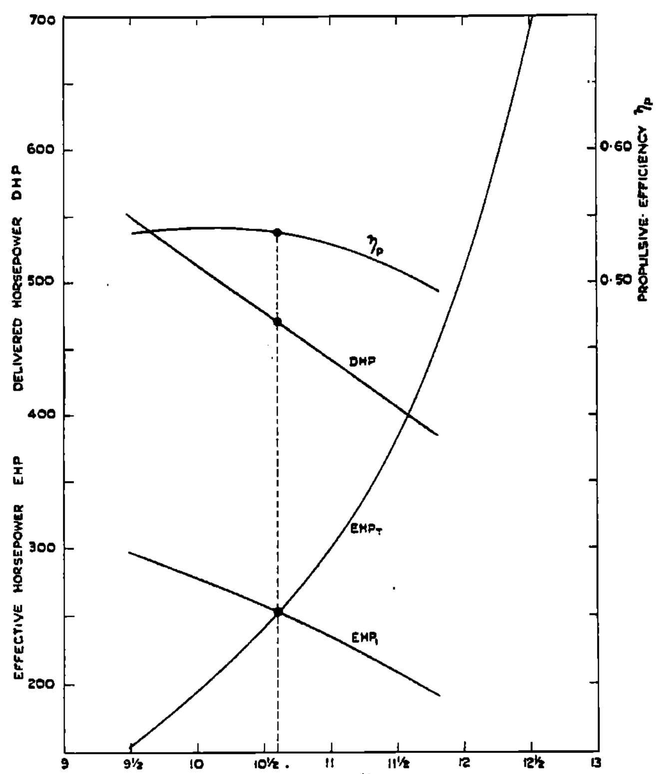
where ϕ_1 and μ_1 are the basic values of torque coefficient evaluated using basic value of rate of rotation N_1 and for which the coefficient k is equal to unity. Pairs of corresponding values of pitch ratio p and thrust torque ratio σ are determined using the μ–σ chart as before (for fixed rate of rotation) and thrust values are calculated using a modified form of equation for the thrust torque ratio given by

$$(18) \quad T_U = \frac{Q_1\sigma}{357kD}$$



ABOVE: **Fig. 2.** *Propulsion estimates (free-running) and performance estimates (towing) for varying rate of rotation*

BELOW: **Fig. 3**



SCREW 2 - PROPULSION ESTIMATES (FREE-RUNNING) FOR FIXED RATE OF ROTATION

Propeller Design and Two-Speed Gearboxes . . .

The results of the calculations are plotted in the form of value of pitch ratio p and thrust T_U (on pull P_U) on a base of rate of rotation N, as shown in **Fig. 2**.

In applying the μ–σ coefficients in designing a free-running screw, a procedure similar to that used for towing duty screws as given above is followed. However, for free-running conditions the performance criterion is the screw efficiency per cent instead of the thrust T_U used for towing conditions. Accordingly, values of the advance coefficient J are calculated and the screw efficiency η_0 is determined using equation 5. The results of the calculations are presented in a similar way as for towing conditions, but here the parameters are screw efficiency η_0 (or propulsive efficiency η_P) on a base of rate of rotation N, as shown in **Fig. 2**.

The towing performance curves of **Fig. 2** show that optimum towing performance is associated with high rate of rotation and low pitch ratio, as might be expected. From the free-running performance curves of **Fig. 2**, the value of the rate of rotation to given maximum screw efficiency can be selected and the corresponding value of the pitch ratio for a screw designed for free-running conditions can be determined. The optimum rate of rotation for this screw when operating at towing duty conditions, and the corresponding value of pull, can be determined from the towing performance curve.

Towing duty

The performance data given in **Fig. 2** can also be applied in designing towing duty screws and estimating their performance at free-running conditions; since, having chosen the design rate of rotation and pitch ratio, the optimum rate of rotation and related value of screw efficiency for free-running conditions can be determined from the free-running performance data.

The performance data of **Fig. 2** do not give performance values for screws designed for free-running conditions, nor do they give free-running performance values for screws designed for towing duty conditions if it is not intended to fit two speed gear boxes. For screws designed for free-running conditions, but operating at towing duty conditions, maximum torque is reached at a rate of rotation lower than the design value, and this needs to be estimated.

This can be done by first computing the torque coefficient ϕ (equation 6) which is then plotted on the μ–σ chart at a point the position of which is located by the intersection of two contours, one of ϕ the other of the pitch ratio p of the screw. This enables corresponding values of torque coefficient μ and thrust torque ratio σ to be read from the chart, and values of screw rate of rotation N and thrust T_U to be derived using the equation given above for torque coefficient μ and thrust torque ratio (equation 8). At the same time, the delivered horsepower DHP can also be evaluated using equation 12.

As stated above, a towing duty screw operating at free-running conditions at the design rate of rotation does so at reduced torque and delivered horsepower. In making free-running performance estimates using the μ–σ coefficients, it is convenient to introduce a torque reduction factor K defined by the relation

$$(19)\quad Q = KQ_M \text{ and } DHP = KDHP_M$$

where the suffix M denotes the maximum values of torque Q and delivered horsepower DHP.

Incorporating the torque reduction factor K in the equations for torque coefficient ϕ and μ and thrust torque ratio σ (equations 6, 7 and 8) they can be re-stated in the form

$$(20)\quad \phi = 1.689\, V_A D \sqrt{\frac{\rho D}{\Sigma_R K Q_M}} = \frac{\phi_M}{K^{\frac{1}{2}}}$$

$$(21)\quad \mu = \frac{ND^2}{60}\sqrt{\frac{\rho D}{\Sigma_R K Q_M}} = \frac{\mu_M}{K^{\frac{1}{2}}}$$

$$(22)\quad \sigma = \frac{357 T_U D}{KQ_M} = \frac{\sigma_M}{K}$$

where ϕ_M, μ_M and σ_M are values of the torque coefficients ϕ and μ and the thrust-torque ratio σ computed using the maximum value of the torque Q_M.

In making free-running performance estimates, a range of values of torque reduction factor K is selected and a corresponding set of values of torque coefficient μ is evaluated. Values of torque coefficient ϕ and thrust-torque ratio σ are read from the μ–σ chart at points the positions of which are located by the intersection of two contours, one of torque coefficient μ, the other of the pitch ratio p of the screw.

This enables the speed of advance V_A and the advance coefficient J to be determined using equations 6 and 9, respectively. The screw efficiency η_0 is determined using equation 10, and the speed of hull V_s, the propulsive efficiency η_p and the effective horsepower available EHP_1 are derived using the equations given in the report,[3] which are reproduced below.

$$(23)\quad V_S = V_A(1 - w_T)$$

$$(24)\quad \eta_p = \Sigma_p \eta_0$$

$$(25)\quad EHP_1 = \eta p DHP$$

where V_S is the speed of the hull in knots
w_T is the wake fraction
Σp is the overall hull factor.

The values of effective horsepower available EHP_1 are plotted on a base of speed of hull V_S, together with corresponding values of effective horsepower on trial EHP_T derived from the hull resistance experiment results, as shown in **Fig. 3**. The value of the speed at which the hull will be propelled and the corresponding value of the effective horsepower are determined by the co-ordinates of the point of intersection of the curves of effective horsepower.

If desired, the value of propulsive efficiency η_p can also be plotted in **Fig. 3**, and this enables the value of propulsive efficiency at the trial speed to be determined, from which the delivered horsepower on trial DHP can be derived using equation 25.

References

1. Troost, L. *Open-Water Tests with Modern Propeller Forms.* Trans. N.E. Coast Instn. Engrs. Shipb., Vol. 67, 1951.
2. O'Brien, T. P. *The Design of Marine Screw Propellers.* Hutchinson's Scientific and Technical Press, London, 1962.
3. O'Brien, T. P. *Some Practical Aspects of Marine Propeller Design with Particular Reference to Single-Screw Tugs* N.P.L. Ship Division Report, November, 1964.

Propeller Design and Two-Speed Gearboxes

. . . with particular reference to tugs and trawlers

Part Two

In this concluding article, the author gives worked examples, the results of which show that for a screw designed for free-running conditions and driven *via* a single speed gearbox, the loss in towing pull would be 22 per cent, but if a two speed gearbox were fitted the loss in towing pull would be only three per cent. Similarly, for a screw designed for towing conditions and driven *via* a single-speed gearbox, the loss in free-running speed would be 15 per cent, but if a two-speed gearbox were fitted the loss in free-running speed would be only 1½ per cent.

by T. P. O'Brien, C.G.I.A., A.M.R.I.N.A., Ship Division, National Physical Laboratory.

It is required to prepare the preliminary design calculations and make performance estimates for the propellers for a single screw tug.

The first is to be designed to absorb maximum power at a stipulated gear ratio and rate of rotation for free-running conditions. Towing performance estimates are to be made for zero speed of hull for two conditions: (*a*) at same gear ratio as for free-running conditions; (*b*) at a gear ratio selected to enable the screw to run at a rate of rotation to absorb maximum power at towing conditions.

The second is to be designed to absorb maximum power at a stipulated gear ratio and rate of rotation for towing conditions. Propulsion estimates are to be made for two free-running conditions (*a*) at same gear ratio and rate of rotation as for towing conditions; (*b*) at a gear ratio selected to enable the screw to run at a rate of rotation to absorb maximum power at free-running conditions.

The computations are to be made using the μ–σ coefficients derived in Section 2 above and the Troost B-4-55 series μ–σ chart given in Figure 1.

Design Data

Hull—Single-screw tug; length 100 ft., breadth 28 ft., draught (aft) 14 ft., rake of keel 4 ft. aft, displacement (mld) 483 tons, block coefficient 0.502. Other particulars as Model 4033B, reference 4.

Hull speed (knots)	V_S	10	10½	11	11½	12	12½	13
Effective h.p. (model experiments)	EHP_M	176	219	273	349	462	627	973
Effective h.p. (trial conditions)	EHP_T	194	241	300	384	508	690	960

$EHP_T = f_M EHP_M$
$\quad = 1 \cdot 10\ EHP_M$

Table 1 Screw 1—Design Calculations: Free-running Conditions

Design Conditions DHP = 1,100, Q_1 = 28,900 pounds feet N_F = 200 RPM
$D = 9$ feet, $V_S = 12.5$ knots $\quad N_1 = 196$ RPM
Propulsion Factors $\quad w = 0.225,\ t = 0.206,\ \Sigma_R = 1.0,\ \Sigma_p = 1.02$
(Ref. 3, Table 3)
Basic Torque Coefficients (equations 16 and 17)

$$\phi_1 = 1.689\, V_A D \sqrt{\frac{\rho D}{\Sigma_R Q_1}} = 3.66 \qquad \mu_1 = \frac{N_1 D^2}{60}\sqrt{\frac{\rho D}{\Sigma_R Q_1}} = 6.57$$

k	N_F	N (1)	$k^{\frac{1}{2}}$	$k^{3/2}$	ϕ (2)	μ (3)	p (4)	σ (4)	J (5)	η_o (6)	η_p (7)
0.7	140	137	0.837	0.586	3.06	3.86	1.40	0.765	0.795	0.608	0.620
0.8	160	157	0.894	0.715	3.27	4.71	1.14	0.895	0.696	0.623	0.635
0.9	180	176½	0.949	0.855	3.47	5.62	0.97	0.995	0.618	0.615	0.628
1.0	200	196	1.000	1.000	3.66	6.58	0.82	1.090	0.556	0.606	0.617
1.1	220	215½	1.049	1.154	3.84	7.59	0.72	1.145	0.506	0.580	0.592
1.2	240	235	1.095	1.314	4.01	8.65	0.62	1.195	0.464	0.555	0.566
1.3	260	255	1.140	1.483	4.17	9.76	0.54	1.23	0.428	0.526	0.536

(1) $N_1 = 0.98\ N_F$ (Wake scale effect, Ref. 2, Section 4.9); (2) $\phi = k^{\frac{1}{2}}\phi_1$ (equation 16); (3) $\mu = k^{3/2}\mu_1$ (equation 17); (4) Values from μ–σ chart (Fig. 1); (5) $J = \dfrac{101.3\ V_A}{ND}$ (equation 9); (6) $\eta_o = J\sigma$ (equation 10); (7) $\eta_p = \Sigma_p \eta_o$ (equation 24). For $N_F = 200$, pitch ratio $p = 0.82$, propulsive efficiency $\eta_p = 0.617$.

Table 2 Screw 2—Design Calculations: Towing Conditions

Design Conditions DHP = 1,100, Q_1 = 28,900 pounds feet, N_1 = 200 RPM
D = 9 feet, $V_S = 0$

Propulsion Data $\Sigma_R = 1.0$, Pull-Thrust Ratio $\tau_p = 0.975$
(Ref. 3, Table 3)

Basic Torque Coefficient $\mu_1 = \frac{N_1 D^2}{60}\sqrt{\frac{\rho D}{\Sigma_R Q_1}} = 6.71$
(equation 17)

k	N	$k^{1/2}$	$k^{3/2}$	ϕ	μ	p	σ	T_U Tons	P_U Tons
					(1)	(2)	(2)	(3)	(4)
0.7	140	0.837	0.586	0	3.93	1.030	1.09	14.0	13.65
0.8	160	0.894	0.715	0	4.80	0.820	1.30	14.6	14.25
0.9	180	0.949	0.855	0	5.73	0.675	1.49	14.9	14.55
1.0	200	1.000	1.000	0	6.71	0.565	1.67	15.0	14.65
1.1	220	1.049	1.154	0	7.74	0.480	1.84	15.0	14.65

(1) $\mu = k^{3/2}\mu_1$ (equation 17); (2) Values from μ–σ chart (Fig. 1); (3) $T_U = \frac{Q_1\sigma}{357kD}$ (equation 18); (4) $P_U = \tau_P T_U$ (equation 13).

For N = 200, pitch ratio $p = 0.565$, pull $P_U = 14.65$ tons.

Engine—Diesel. Delivered horsepower at screw d.h.p. = 1,100, engine speed 600 r.p.m., stipulated gear ratio 3:1 giving 200 r.p.m. for screw, corresponding maximum torque at screw = 28,900 pounds feet.

For free-running conditions N = 0.98 N_F (2 per cent wake scale effect, see Section 4.9 of Ref. 2).
= 196 RPM

For towing conditions N = N_F = 200 rpm

Stern Details—Streamlined rudder, shaft immersion I = 7.7 ft.

Stipulations. Screw diameter 9.0 ft., number of blades 4.

Design Conditions. Screw 1 to be designed to absorb maximum power under trial conditions when running free at a trial speed of 12½ knots.

Screw 2 to be designed to absorb maximum power under towing conditions at zero speed of hull.

Screw 1—Design Calculations—Free-running conditions

In making the design calculations given in Table 1, first, the basic values of the torque coefficients ϕ_1 and μ_1 are calculated using the given screw diameter D and speed of advance V_A and the basic values of delivered horsepower DHP, torque Q_1 and rate of rotation N_1. Next a series of values of torque coefficients ϕ and μ are derived covering a range of screw rate of rotation and applying the constant power condition. This enables a series of corresponding values of pitch ratio p and thrust torque ratio σ to be obtained from the μ–σ chart shown in **Fig. 1**. Finally, a set of values of screw efficiency η_O are derived from the chart values of σ. This enables a set of values of propulsive efficiency η_p to be derived from η_O, and these are plotted on a base of rate of rotation N, together with the values of pitch ratio p, as shown in **Fig. 2.**

Screw 2—Design Calculations—Towing Duty Conditions

In making the design calculations for this screw given in Table 2, the procedure is similar to that followed for Screw 1. However, since the speed of advance is zero the torque coefficients ϕ become zero; consequently, the values of pitch ratio p and thrust torque ratio σ are read from the μ–σ chart at points located by the contour $\phi = 0$ and the co-ordinate μ. A set of values of screw thrust T_U are derived from the chart values of σ. A set of values of pull P_U are derived from T_U and plotted on a base of rate of rotation N, together with the values of pitch ratio p, as also shown in **Fig. 2.**

Screw 1—Towing Performance Estimates

(*a*) *Single Speed Gear Box.* In making the towing performance estimates given in Table 3, corresponding values of torque coefficient μ and thrust-torque ratio are read from the μ–σ chart at the point determined by the intersection of the contours torque coefficient $\phi = 0$ and value of pitch ratio p for the screw. The rate of rotation N at which the screw is operating and the resulting delivered horsepower DHP are determined from the torque coefficient μ. The thrust T_U is determined from the thrust-torque ratio σ and this enables the pull P_U to be derived.

(*b*) *Two-Speed Gear Box.* In addition to giving the performance of screws designed for towing duty conditions, the towing performance data shown in **Fig. 2** can also be

Table 3 Screw 1—Towing Performance: Estimates

Operating Conditions: Maximum torque Q_1 = 28,900 pounds feet at zero speed of advance $V_A = 0$, $\phi = 0$. Screw diameter D = 9.0 feet.

p	μ	σ	N RPM	DHP	T_U TONS	τ_P	P_U TONS
(1)	(2)	(2)	(3)	(4)	(5)	(6)	(7)
0.82	4.79	1.305	143	785	11.75	0.971	11.4

(1) Pitch ratio as determined for free-running condition (Table 1); (2) Values from μ–σ chart (Fig. 1); (3) $N = \frac{60\mu}{D^2}\sqrt{\frac{\Sigma_R Q}{\rho D}}$ (equation 7); (4) DHP = $\frac{2\pi NQ}{33{,}000}$ (equation 12); (5) $T_U = \frac{\Sigma_R Q\sigma}{357D}$ (equation 8); (6) Pull-Thrust Ratio (Ref. 3, Table 3); (7) $P_U = \tau_P T_U$ (equation 13); For two speed gear box N = 160, P_U = **14.25** **(Fig. 2).**

Table 4 Screw 2—Propulsion Estimates: Free Running Conditions

Operating Conditions: Rate of rotation for stipulated gear ratio

$N_F = 200, N = 196$

Propulsion Factors. $w = 0.225$, $\Sigma_R = 1.0$, $\Sigma_p = 1.02$
(Ref. 3, Table 3)
Screw diameter $D = 9.0$ feet pitch ratio $p = 0.565$ (Table 2).

Basic Torque Coefficient $\mu_M = \dfrac{ND^2}{60}\sqrt{\dfrac{\rho D}{\Sigma_R Q_M}} = 6.58$ (equation 20).

K	$K^{\frac{1}{2}}$	μ (1)	ϕ (2)	σ (2)	V_A knots (3)	V_S knots (4)	J (5)	η_o (6)	η_p (7)	DHP (8)	EHP_1 (9)
0.50	0.707	9.31	3.94	1.245	7.35	9,49	0.423	0.527	0.537	550	296
0.45	0.671	9.81	4.50	1.155	7.97	10.29	0.458	0.529	0.540	495	267
0.40	0.632	10.41	5.13	1.050	8.57	11.06	0.493	0.517	0.527	440	232
0.35	0.592	11.12	5.86	0.920	9.17	11.84	0.527	0.485	0.495	385	191

(1) $\mu = \mu_M/K^{\frac{1}{2}}$ (equation 21); (2) Values from μ–σ chart (Fig. 1); (3) $V_A = \dfrac{\phi}{1.689D}\sqrt{\dfrac{\Sigma_R K Q_M}{\rho D}}$ (equation 20); (4) $V_S = \dfrac{V_A}{(1\text{-}w)}$ (equation 23); (5) $J = \dfrac{101.3V_A}{ND}$ (equation 9); (6) $\eta_o = J\sigma$ (equation 10); (7) $\eta_p = \Sigma_p \eta_o$ (equation 24); (8) DHP = K DHP_M (equation 19); (9) $EHP_1 = \eta_p$ DHP (equation 21). From Fig. 3, $V_S = 10.6$ knots, $\eta_p = 0.54$. For two-speed gear box, $N = 253$, $\eta_p = 0.55$ (Fig. 2). EHP $= \eta_p$ DHP $= 605$ $V_S = 12.3$ (Fig. 3).

used to make towing performance estimates for screws designed for free-running conditions. In applying the procedure, corresponding values of rate of rotation N and tow rope pull P_U are obtained from **Fig. 2** for the value of the pitch ratio p as previously determined for free running conditions. From the rate of rotation N the values of the second gear ratio for the gear box is derived.

Screw 2—Propulsion Estimates—Free-running Conditions

(*a*) *Single-Speed Gear Box.* In making the propulsion estimates given in Table 4, first, the basic value of the torque coefficient μ_M is calculated using the given screw diameter D, the maximum value of torque Q_M derived from the maximum delivered horsepower DHP_M and the stipulated value of the rate of rotation N. Next, a series of values of torque coefficient μ are derived from μ_M for reduced torque and power covering a range of values of torque reduction factor K. This enables a series of corresponding values of torque coefficient ϕ and thrust torque ratio σ to be obtained from the μ–σ chart shown in **Fig. 1,** at points the positions of which are located by the co-ordinate of μ and the contour of the pitch ratio p of the screw. This enables a series of values of speed of advance V_A to be calculated from ϕ, and a series of values of screw efficiency η_o to be calculated from σ. Finally, a series of values of speed of hull V are

Table 5 Screws 1 and 2—Comparisons of Free-Running and Towing Performance

Screw	Design Condition	Free-Running DHP	Free-Running N RPM	Free-Running η_p	V_S Knots	Towing (at $V_S = 0$) DHP	Towing N RPM	Towing P_U Tons	Gear Box	Gear Ratios
(1)	Free running	1,100	200	0.617	12.5	786	143	11.40	One speed	3:1
		1,100	200	0.617	12.5	1,100	160	14.25	Two speed	3:1 3.75:1
(2)	Towing	470	200	0.54	10.6	1,100	200	14.65	One speed	3:1
		1,100	253	0.55	12.3	1,100	200	14.65	Two speed	3:1 2.37:1
(1)	% loss in towing pull							22	One speed	3:1
								3	Two speed	3.75:1
(2)	% loss in free-running speed				15				One speed	3:1
					1½				Two speed	2.37:1

derived from V_A, and a series of values of propulsive efficiency η_P are derived from η_O: the product η_P DHP gives the effective horsepower available EHP_1, the values of which are plotted on a base of speed of hull V , together with the values of the effective horsepower on trial EHP_T obtained from the hull resistance data, as shown in **Fig. 3.** The speed co-ordinate of the point of intersection of the two effective horsepower curves determines the speed at which the hull will be propelled. Similarly, the effective horsepower co-ordinate gives the corresponding value of the effective horsepower, and the propulsive efficiency curve gives the value of the propulsive efficiency η_P from which the delivered horsepower DHP can be derived.

(*b*) *Two-Speed Gear Box.* In making these estimates, a procedure similar to that used in making the towing performance estimates is followed, and corresponding values of rate of rotation N and propulsive efficiency η_P are obtained for the value of the pitch ratio *p*, as previously determined for towing duty conditions. From the rate of rotation N the value of the second gear ratio for the gear box is derived. Since the propulsive efficiency is lower than for Screw 1, which had been designed for free-running conditions, the speed of hull will also be lower than for Screw 1. The value of the speed of hull is determined from the hull resistance data given in **Fig. 3**, at the co-ordinate of V_S corresponding to the co-ordinate of EHP determined by the product η_P DHP.

Comparison of Results

The results of the foregoing calculations are compared in Table 5. These show that there are significant advantages in fitting a two-speed gear box, as summarised below. For Screw 1, designed for free-running conditions and driven via a single speed gear box, the towing pull would be 22 per cent lower than for Screw 2, designed for towing conditions; however, if a two speed gear box were fitted the loss in towing pull would be 3 per cent.

For Screw 2, designed for towing conditions and driven via a single speed gear box, the free-running speed would be 15 per cent lower than for Screw 1, designed for free-running conditions; however, if a two speed gear box were fitted the loss in free-running speed would be 1½ per cent.

Reference

4. Parker, M. N. and Dawson, J. *Tug Propulsion Investigation. The Effect of a Buttock Flow Stern on Bollard Pull, Towing and Free-Running Performance.* Trans. Roy. Instn. Nav. Archit., Vol. 104, 1962.

(The figures referred to in this article appeared in Part One, November issue)

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RESEARCH ON TUG PROPELLERS

by T. P. O'Brien, C.G.I.A., M.R.I.N.A., Ship Division, National Physical Laboratory

IN designing tug propellers and making estimates of their performances a large amount of basic data are required and numerous aspects need to be considered. These include: Propulsion data for free-running and towing conditions and performance data for alternative types of propulsion machinery to enable preliminary propulsion estimates to be made; design charts, cavitation charts and strength criteria to enable the overall geometric features of basic screws to be chosen; and correction factors to make allowance for variation in particular geometric features (blade-section shape, blade thickness, number of blades, blade-area ratio, blade outline and boss-diameter ratio) to enable the detailed geometric characteristics of the screws to be selected.

These comments apply generally to all marine screws, but there are two factors affecting tug screws that are significant. For some classes of vessel (for example, tankers and cargo vessels) an extensive amount of systematic propulsion data are available, but for others (including tugs) there is a dearth of published data. Moreover, owing to the differing conditions under which tugs operate some tug screws are designed for free-running conditions and performance estimates are required for towing conditions, while others are designed for towing conditions and propulsion estimates are required for free-running conditions.

A summary of published data

What follows is a bibliography of data comprising information which has proved useful in designing tug screws and making estimates of their performances. Design topics and the particular publications in which they are discussed are listed in **Table 1.** The subject matter of these publications relevant to tugs is summarised below.

1. ARGYRIADIS, D. A. *Modern tug design with particular emphasis on propeller design, manoeuvrability and endurance,* Trans. Soc. Naval Arch. & Mar. Engrs., 1957, *65.*

Several types of main propulsion machinery power plants are discussed and the merits of each one are presented. Propeller design is discussed at some length.

Table 1. Design Topics and Relevant Publications

Item	Topic	Publications
1	Aspects of screw design	1, 2, 3, 4, 5, 6, 12, 13
2	Propulsion data	1, 2, 3, 6, 7, 8, 9
3	Propulsion machinery	1
4	Design charts	2, 3, 4, 5, 12, 15
5	Cavitation	2, 3, 5, 12, 13, 14
6	Strength	2, 3, 12, 13
7	Variation in geometric features	2, 3, 5, 6, 7, 8, 10, 11, 12, 13
8	Correction factors and formulae	1, 2, 3, 5, 6, 7, 8, 9, 10, 11 12, 13, 15
9	Worked examples	1, 2, 3, 5, 12, 13, 15

Preliminary design formulae are given for both bollard pull and towing thrust. Comparisons between the different types of propellers are made and a method for calculating the performance of the propeller at any speed of the vessel is presented.

2. O'BRIEN, T. P. *The design of marine screw propellers,* Hutchinson Scientific and Technical Press, London, July, 1962.

Chart methods for the design of screws and prediction of screw performance are described in detail, and correction factors given to enable varying screws to be compared. Modern developments in applied circulation theory, cavitation and its associated problems and the adaptation of aerodynamic data to the design of blade sections are all fully and practically presented. There are chapters on model experiments, strength, design factors and screw geometry and a large number of useful tables and charts throughout. A comprehensive set of worked examples is given, ranging from uses of simple design charts to practical applications of theoretical design methods.

NOTE: Subsequent to publication it was found that the design charts (pages 79 and 83) had been interchanged during printing. Consequently, the captions should read as follows:

Page 79, Fig. 3.16 Troost B.4-55 Bp — δ Chart.

Page 83, Fig. 3.13 Troost B.3-50 Bp — δ Chart.

Some small errors which have been found in the text are listed in an errata, copies of which can be obtained from the author.

3. O'BRIEN, T. P. *Design of tug propellers,* SHIP AND BOAT BUILDER INTERNATIONAL, London, April, 1965, *18,* 22.

This article discusses general aspects of propulsion and applications to tug propellers operating at free-running and towing-duty conditions. It describes charts for designing propellers and making cavitation estimates, and it includes a procedure for designing tug screws and making estimates of their performance. Formulae for assessing blade stresses and estimating weight and moment of inertia are also given.

It summarises single-screw tug propulsion data recently obtained at the N.P.L., and it gives worked examples on the design and performance assessment of two tug screws, one designed for free-running conditions, the other for towing-duty conditions.

NOTE: An improved form of cavitation chart used in making the design calculations discussed above is given in an article mentioned below (14). It is reproduced in **Fig. 1.**

4. TROOST, L. *Open-water test series with modern propeller forms,* Trans. N.E. Coast Inst. Engrs. Shipb. 1951, *67.*

This paper records the results of experiments with a systematic series of two-bladed and with two systematic series of five-bladed aerofoil propeller models, designed in accordance with practice in 1950. Previously published results with three- and four-bladed test series are completed with results of tests with propellers of larger disc-area ratio and smaller pitch ratio. Charts of all series are extended by so called μ—σ design and analysis diagrams.

5. WRIGHT, B. D. W. *The N.S.W.B. standard series propeller data and their application,* British Ship Research Assocn. T.M. 213, June, 1965.

The Netherlands Ship Model Basin have extended the range of open-water tests on the Troost B-series propellers to include six- and seven-bladed propellers and have increased the range of blade-area ratios of the two-, three-, four- and five-bladed series. In this memorandum the results of the whole series of tests are presented in the form of B_p—δ and μ—σ design diagrams, together with a selection of worked examples illustrating the use of these diagrams.

A brief discussion is included on (*a*) factors influencing propeller design, (*b*) cavitation and (*c*) the use of correction factors when departing from standard series propeller dimensions.

6. HARVALD, S. A. *Tug propulsion—wake, thrust deduction and r.p.m.,* European Shipbuilding, Oslo, 1963, No. 3.

The variation of wake fraction and thrust deduction coefficient with speed, advance coefficient and thrust load coefficient has been determined on the basis of model experiments published by different authors. The result is applicable for preliminary design of tugs. A diagram linking wake fraction thrust deduction coefficient and block coefficient is given. Finally, the questions of the most suitable number of revolutions and the propeller diameter are discussed.

7. PARKER, M. N., and DAWSON, J. *Tug propulsion investigation—the effect of a buttock-flow stern on bollard pull, towing and free-running performance,* Trans. Roy. Inst. Naval Arch., 1962, *237.*

A series of model experiments was carried out to determine the effect of introducing buttock-flow stern lines on the performance of a tug under static (*i.e.,* bollard trial), towing and free-running conditions. The models were tested with a series of propellers, designed to cover a range of revolutions for the same power absorption, corresponding to 1,100 h.p. for a 100-ft. full-scale tug.

Four model hulls were used: (1) Conventional form A—this represented a typical modern single-screw Diesel tug; (2) conventional form B—this was similar to conventional form A, but the aperture was enlarged to enable a propeller of greater diameter to be fitted; (3) buttock-flow form A—this had a buttock-flow afterbody, but was otherwise the same as conventional form A; and (4) buttock-flow form B—this had the same afterbody as buttock-flow form A, but the forebody was redesigned on buttock-flow principles.

The results of the experiments indicated that the differences in the bollard and towing performances of the hull are small when they are fitted with the same propeller, but, in all conditions, from static to free-running, the conventional stern tends to give a better performance than the buttock-flow stern. When running free at a speed of 11 knots the buttock-flow forms require 11 to 16 per cent more power than the conventional forms, depending upon the propeller fitted.

The results also show the advantage of a large-diameter propeller with low revolutions, particularly in the static condition, but it is pointed out that if the diameter is made too great there is the possibility that the propeller may "sing" when the tug is running free.

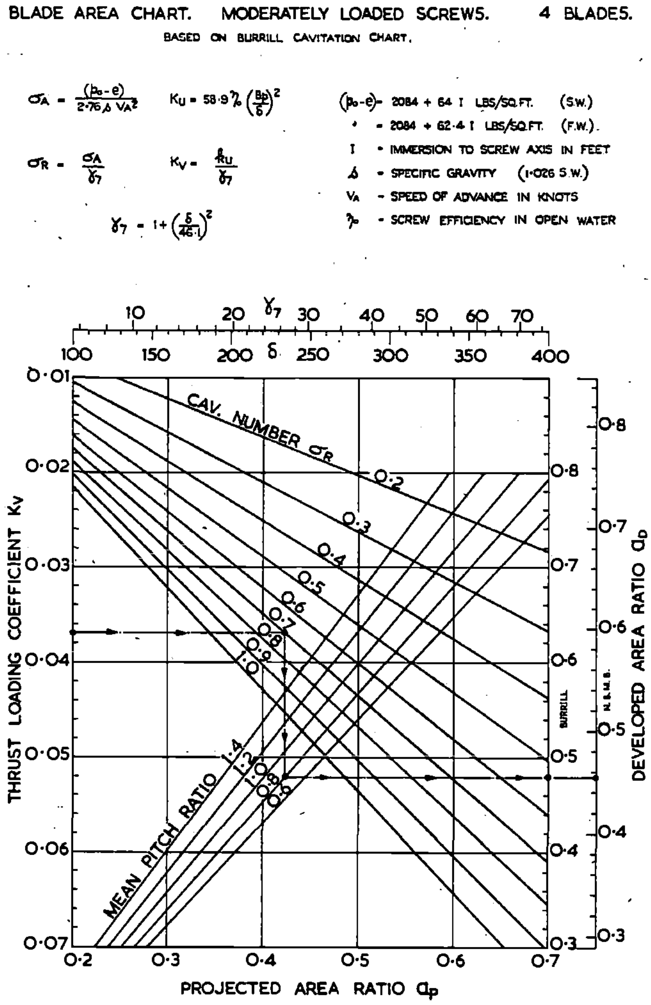

Fig. 1

8. DAWSON, J. *Tug propulsion investigation—the effect on performance of designing propellers for the free-running condition,* Trans. Roy, Inst. Naval Arch., 1964, *106.*

This paper presents the results of experiments carried out for the B.S.R.A. on a single-screw tug model, which had a conventional stern, to investigate the effect on performance of designing propellers for the free-running condition. The results are a continuation of those given for bollard design propellers in an earlier paper.

For many tugs speed is important, and it was considered that results for free-running design propellers, when compared with those previously obtained, would throw some light on the performance of controllable-pitch propellers. Three propellers designed to cover a range of revolutions for the same power absorption were used and the propulsion experiments covered the static condition, towing at low speeds, and free running.

The marked effect of whether a propeller is designed for the bollard or the free-running condition is clearly shown, and this indicates the possible advantages of a controllable-pitch propeller. In the range 130 to 270 r.p.m. covered by the investigation, free-running speeds have been improved by approximately 14 to 17 per cent, compared with the performance of corresponding bollard design propellers.

The results also show that the original bollard pulls, with screws designed for the static condition, are from 43 to 19 per cent better than those obtained with the corresponding free-running design propellers. The performance at low speeds could also be improved if the propellers were designed specifically for the towing condition or, in the case of controllable-pitch propellers, if the pitches were adjusted to intermediate positions.

9. MOOR, D. I. *An investigation of tug propulsion,* Trans. Roy. Inst. Naval Arch., 1963, *105*.

This paper presents the results of a series of tests carried out on a model of a single-screw tug in an investigation of its propulsive characteristics.

Resistance experiments were conducted over a range of speed-length ratios of 0.40 to 1.30. Propulsion experiments, both free-running and with various towing pulls, were made over the same speed range and in the bollard condition, with three different propeller designs. The results are presented for a 100-ft. ship.

On the average, the maximum towing h.p. is available at 0.64 of the running-free speed attainable, and is 0.45 of the d.h.p. The corresponding r.p.m. are 0.93 of those when the vessel is running free at the same d.h.p.

The bollard pull with the propeller revolving ahead is 1.66 of the towrope pull corresponding to the maximum towing h.p., and the corresponding r.p.m. are 0.81 of the running-free r.p.m. For a given towing pull, with the propeller revolving astern, d.h.p. and r.p.m. are, on the average, respectively 1.92 and 1.25 of those obtained with the propeller revolving ahead.

10. O'BRIEN, T. P. *Some effects of blade-thickness variation on model screw performance,* Trans. N.E. Coast Inst. Engrs. Shipb., 1957, *73,* 405.

Calculations and experiments have been made for two groups of screws, one having N.P.L. sections and the other having segmental sections, each of varying blade thickness. Performance values have been calculated for non-cavitating conditions by alternative methods and the results compared with those obtained by testing model screws in open water. The calculations have been extended in a simplified form to cover the range of a four-bladed model screw series so as to provide thickness correction factors for pitch, power and efficiency. Some of the model screws have been tested in the Lithgow water tunnel.

In an appendix some calculated results are given to enable the increase in blade area for a given increase in blade thickness to be estimated.

11. O'BRIEN, T. P. *Some effects of variation in number of blades on model screw performance,* Trans. N.E. Coast Inst. Engrs. Shipb., 1965, *81,* 233.

This paper gives the results of calculations and experiments for two groups of model screws, one of standard type and the other of non-standard type, both comprising screws having three, four and five blades. It includes comparisons of performance under non-cavitating and cavitating conditions based on calculations, open-water experiments and water-tunnel experiments.

For standard type screws, correction factors are derived and design data are given which enable three- and five-bladed screws to be designed using four-bladed standard series data as the bases. For non-standard type screws, correction factors additional to those applied in making the basic design calculations are given, enabling closer agreement to be obtained in the performance of three-, four- and five-bladed screws designed using detailed calculations.

12. O'BRIEN, T. P. *Design of tug propellers—performance of three-, four- and five-bladed screws,* SHIP AND BOAT BUILDER INTERNATIONAL, London, November and December 1965, *18,* and January 1966, *19*.

This group of articles discusses differences between the performance of three-, four- and five-bladed screws both under non-cavitating and cavitating conditions. It summarises N.P.L. model experiment data, and it comprises correction factors and design data which enable three- and five-bladed screws to be designed, and comparative performance estimates to be made, using four-bladed standard series data as the bases. It gives examples on designing additional three- and five-bladed screws for a single-screw tug for which data for four-bladed screws are available.

The results of the calculations show that reducing the number of blades from four to three results in improved performance, but that increasing the number of blades from four to five generally results in adverse performance. For a three-bladed screw designed for free-running conditions the increase in efficiency would be 2½ per cent, and at towing conditions the increase in pull would be 1½ per cent. For a three-bladed screw designed for towing conditions the increase in pull would be 1 per cent, but at free-running conditions there would be no change in performance. For a five-bladed screw designed for free-running conditions the reduction in efficiency would be 4 per cent, and at towing conditions the reduction in pull would be 5 per cent. For a five-bladed screw designed for towing conditions the reduction in pull would be 5 per cent, but at free-running conditions there would be an increase in speed of ½ per cent.

13. O'BRIEN, T. P. *Design of tug propellers—optimum screw diameter and rate of rotation* (not yet published).

This discusses the results of varying the screw diameter and the design rate of rotation and the effects on performance under both non-cavitating and cavitating conditions. It describes a procedure for making allowance for variation for departure from basic blade-area ratio and blade-thickness ratio. It comprises worked examples for tug screws designed for both free-running and towing conditions.

The results show that for the free-running screws small improvements in performance can be achieved if the design rate of rotation can be selected, and that significant improvements in performance can be achieved if also the screw diameter can be increased within practical limits of aperture size (for single screws) or tip clearance (for twin screws). For these screws operating at towing conditions the change in rate of rotation results in adverse performance, but the change of both rate of rotation and diameter results in improved performance.

For the screws designed for towing conditions the change in rate of rotation results in adverse performance, but the change in both rate of rotation and diameter results in improved performance. For these screws operating at free-running conditions both the change in rate of rotation only and the change in rate of rotation and diameter result in improved performance.

Selecting a typical 9 ft.-diameter screw designed to operate at 200 r.p.m. as the basis, it was found that the optimum rate of rotation would be 160 r.p.m. and the resulting increase in efficiency would be 2 per cent. Moreover, if the maximum diameter consistent with aperture size (D=10.25 ft.) were selected, the optimum rate of rotation would be 140 r.p.m. and the resulting increase in efficiency would be 8 per cent. At towing conditions the change in rate of rotation would result in a reduction in pull of 4 per cent, but the change in both rate of rotation and diameter would result in an increase in pull of 6½ per cent.

For a 9 ft.-diameter screw designed to operate at 160 r.p.m. at towing conditions there would be a reduction in pull of 2½ per cent, but at free-running conditions there would be an increase in speed of 4 per cent. For a 10.25 ft.-diameter screw designed to operate at 140 r.p.m. at towing conditions there would be an increase in pull of 7 per cent, and at free-running conditions there would be an increase in speed of 3 per cent.

14. O'BRIEN, T. P. *Graphs and contour charts and applications to propeller design*, European Shipbuilding, Oslo, March 1965, *14*, 2.

This note describes simple graphs relating two variables and contour charts comprising three variables. It describes methods for constructing contour charts, and it gives two examples; one where the relation between the variables is of a simple mathematical form, and the other where it is of an empirical form.

15. O'BRIEN, T. P. *Propeller design and two-speed gearboxes with particular reference to tugs and trawlers*, SHIP AND BOAT BUILDER INTERNATIONAL, London, November 1964, *17*, 41 (reprinted in Ship Division Tech. Memo. 79, February 1965).

This article discusses the differences in performance of screws designed for free-running conditions and towing-duty conditions, the former when towing and the latter when running free. It shows that significant improvements in performance for both types of screw can be achieved by using two-speed gearboxes enabling the optimum rate of rotation to be chosen for both free-running and towing conditions. Equations are derived and coefficients are given to enable design and operating conditions to be chosen to give optimum performance.

Worked examples are given, the results of which show that for a screw designed for free-running conditions and driven via a single-speed gearbox the loss in towing pull would be 22 per cent, but if a two-speed gearbox were fitted the loss in towing pull would only be 3 per cent. Similarly, for a screw designed for towing conditions and driven via a single-speed gearbox the loss in free-running speed would be 15 per cent, but if a two-speed gearbox were fitted the loss in free-running speed would only be 1½ per cent.

Research in Progress at the N.P.L.

In addition to the list of data already mentioned, two research projects at present being undertaken at the N.P.L. are listed below. The first (16) is a continuation of an investigation into the effects of varying the geometric features of a series of model screws (10 and 11 above). The second (17) will provide both propulsion data and screw design data for tug propellers. It forms two parts—(I) on tug propulsion and (II) on propeller design. The preliminary work for Part I is in hand: the items comprising Part II, some of which have been published (3, 12 and 15), are listed in **Table 2.** A synopsis of the first project and an outline of the second project are summarised below.

16. O'BRIEN, T. P. *Some effects of variation in blade area, blade outline and boss diameter on model screw performance.* (Completed—awaiting publication.)

This paper gives the results of experiments for three groups of model screws, covering variations in blade area, blade outline and boss diameter. It includes comparisons of performance under non-cavitating and cavitating conditions based on open-water experiments and water-tunnel experiments. Correction factors are derived and design data are given which enable screws of different blade area, blade outline and boss diameter to be designed, and comparative estimates of their performance to be made using data for a standard screw as the bases.

Table 2. Design of Tug Propellers—Proposed Outline of Work, January 1966

Item	Topics	Remarks
1	Aspects of propulsion Design considerations Design charts. Cavitation charts. Stress calculations. Weight and moment of inertia Worked examples	Published April 1965 (Reference 3)
2	Comparison of three-, four- and five-bladed screws	Published November 1965 (Reference 12)
3	Effects of variation in diameter and rate of rotation	Not yet published
4	Effects of variation in blade shape and boss diameter	Preliminary work in hand
5	Comparisons of ahead and astern performance. Design of astern-duty screws	
6	Controllable-pitch screws	
7	Nozzle screws	
8	Alternative propulsion units (steam, Diesel, Diesel-electric)	
9	Two-speed gearboxes	Published November 1964 (Reference 15)
10	Overall comparisons, optimum combination of screw and propulsion unit	

17. O'BRIEN, T. P. *A thesis on tug propulsion and propeller design.* (In preparation.)

Part I—Propulsion: Analysis of N.P.L. resistance and propulsion data for tugs, derivation of basic screw design data. Proposed outline is as follows: Analysis of propulsion experiment results. Assessment of basic data, including geometric features of screws, values of wake fraction and hull factor at free-running conditions, and values of pull-thrust ratio and pull at towing conditions. Derivation of design data comprising charts for estimating propulsion factors for free-running screws and pull criteria for towing-duty screws.

Part II—Propeller Design: Dissertation on general aspects of propulsion, application to tug screws operating at free-running and towing conditions, design charts, cavitation charts, strength criteria. Worked examples on designing screws and making performance estimates at free-running and towing conditions. Investigation into effects of variation in number of blades, blade shape, boss diameter, screw diameter and rate of rotation. Comparisons of performance of fixed-pitch, controllable-pitch and nozzle screws. Propulsion estimates for astern conditions. Alternative propulsion units and two-speed gearboxes. For proposed outline see **Table 2.**

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