# The Fundamentals of Ship Propulsion.

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*Reprinted from the Transactions of the Institute of Marine Engineers, Vol. LVIII, March, 1946.*

# The INSTITUTE of MARINE ENGINEERS

Founded 1889. Incorporated by Royal Charter, 1933.

SESSION 1946. *Transactions* Vol. LVIII. No 2.

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# The Fundamentals of Ship Propulsion.

By F. H. TODD, B.Sc., Ph.D., M.I.N.A

*Read on Tuesday, January 8th, 1946, at 5.30 p.m. at 85, The Minories, E.C.3.*

Chairman: A. F. C. TIMPSON, M.B.E. (Past-Chairman of Council).

**Synopsis.**

*This paper is a sequel to that on the Fundamentals of Ship Form given to the Institute last year.*

*After a brief historical review of the development of marine propulsion, the momentum and vortex theories of propeller action are outlined, with some account of the characteristics of blade sections and the application of such knowledge to marine propellers, including the use of model experiments.*

*The next section deals with the interaction between the propeller and the hull of the ship, and the various elements, such as wake, thrust deduction, and relative rotative efficiency, which together make up the propulsive coefficient. After a short account of cavitation phenomena, the paper concludes with a section on propeller design in which some guidance is given as to the probable values of wake and propulsive efficiency for typical single and twin screw ships.*

## 1. Introduction.

At the invitation of your Council I read a paper[1] before your members in 1944 entitled "The Fundamentals of Ship Form", which dealt with the principal features of ship resistance and how they govern the shape of hull for different types of vessels. The discussion on that paper suggested that a similar survey of the field of ship propulsion might be of interest to your members, and this paper is the result.

It does not claim to deal with all the modern theories of screw propeller action, which have been described before this and kindred institutions on many occasions. Rather is it an attempt to show how the propeller and hull interact, and to set out the principal elements which go to make up the overall propulsive efficiency. As in the first paper, the title includes the word "fundamental", and the subject is accordingly discussed *ab initio*, in the hope that it may be of especial interest to young members just beginning their marine engineering experience, and also perhaps to those older members at sea, who have not had the opportunity in the last few years of keeping up their knowledge of the more theoretical side of their profession.

## 2. Types of Propellers.

When a ship is moving, it experiences resisting forces from the water and air which must be overcome by a forward thrust, supplied by some thrust-producing mechanism.

The earliest type of propeller to use mechanical power seems to have been of the jet type, using a prime-mover and a pump, patents for which were granted to Toogood and Hayes[2] in this country in 1661. In such an installation the water is drawn in by the pump and delivered sternwards as a jet at a higher velocity, the reaction providing the forward thrust to propel the ship. At the speeds met with in ships, the jet is materially less efficient than other forms of propellers, and it has never competed with them, its possible use being restricted to very special types of craft.

In 1801 appeared the first steam-driven paddle vessel, the "Charlotte Dundas", built by Symington for service on the Forth-Clyde Canal. Six years later came the famous "Clermont", constructed by Robert Fulton for passenger service on the Hudson River, New York.

From this time until about 1850 followed the heyday of paddle steamers. The Atlantic was first crossed by such a vessel in 1819—the American full-rigged ship "Savannah" with auxiliary steam power—and then followed a line of familiar names—"Royal William", "Great Western", the famous first Cunarder "Britannia", culminating in the last Cunarder to be driven by paddles, the "Scotia", built in 1861.

Paddle wheels were far from ideal for sea-going vessels. The immersion varied with different loadings of the ships, the wheels came out of water when the ship was rolling, causing very erratic course-keeping, and were liable to damage from rough seas. From the marine engineer's point of view, they were too slow-running, involving the use of large heavy engines. Because of the low speed of turning, however, they were reasonably efficient as a propulsive device, but their other operational weaknesses ensured their rapid decline in popularity once the screw propeller arrived. To-day they still have a useful field among pleasure steamers and tugs, plying in river or other smooth waters, in which types the draft does not change materially, and where restrictions of draft due to shallow water prohibit the use of large screw propellers. Side paddles in such vessels have also the added advantage that they give good manœuvring characteristics when berthing.

The first proposal to use a screw propeller appears to have been made in England by Hooke in 1680[2], and its first actual use for such a purpose is usually attributed to an American, Colonel Stevens, in a steam-driven boat at New York in 1804[3]. In 1828 a vessel 60ft. long was successfully propelled by a screw propeller designed by Ressel, of Trieste, obtaining a speed of six knots, but this success was not followed up by the Austrian engineers or shipowners[3]. The first really practical applications came in 1836, by Pettit Smith, a farmer, in this country, and Ericsson in America.

The screw propeller had many advantages over the paddle wheel—it was not materially affected by the changes in draft experienced in the course of normal trading, it was well-protected from damage under the counter, either from seas or accidental collision, it did not increase the overall width of the ship, could be made to run much faster than paddles and still have as good or better efficiency, so that much smaller, lighter, faster-running engines could be used, and the machinery installation did not interfere so much with the internal arrangements of the ship. Consequently it rapidly superseded the paddle for all ocean-going vessels.

The first screw-propelled steamer to make the Atlantic crossing was the British vessel "Great Britain", in 1845, and as already stated the last Transatlantic paddle vessel to be built was the "Scotia" in 1861.

From that time the screw propeller has reigned supreme in the realm of marine propulsion. There remain, of course, many problems, chief among them, perhaps, the incessant quest for propellers to absorb more and more power without an undue onset of cavitation and subsequent erosion, but it is true to-day that the screw propeller has no real rival in the field of ship propulsion.

This brief survey of propeller types and their history should not be closed without a reference to another type of propeller—that in which the screw rotates about a vertical axis, typified by the Voith-Schneider propeller[4]. This consists of a large disc set flush

[1] For references to papers, see Bibliography.

with the lower surface of the hull, and carrying a number of projecting vertical blades rather resembling spade rudders. As the disc revolves around a vertical axis, each of these blades rotates to some extent about its own vertical axis, being so adjusted to the flow that the total thrust from all the blades is concentrated in one direction. This resultant "thrust-direction" can be controlled by different settings of the blades, so as to drive the ship ahead, astern or sideways. The Voith-Schneider screw was first fitted to a 60 H.P. launch in 1929, and by 1939 there were 120 ships so fitted, the maximum power of any installation being 2,200 H.P. It lends itself essentially to craft which need to have great ability to manœuvre. Also, it enables the equivalent of a large diameter orthodox propeller to be fitted to ships which have to operate in shallow water, and in replacing paddles for such purposes it greatly facilitates the adoption of diesel engines, with their advantages over the slow heavy steam engines in such craft.

Although its efficiency does not exceed that of the ordinary propeller, and its maintenance is heavier, the above advantages have naturally resulted in its chief applications being to river steamers, tugs and ferries. In case it may be thought that the Voith-Schneider propeller was a new idea, it is as well to point out that an exactly similar principle was employed in the U.S.S. "Alarm" in 1874[6]. This vessel carried a fixed bow gun and had to be trained to aim the gun. In order to be able to keep the ship steady in a tideway, where a rudder would be useless, what was virtually a feathering paddle wheel was fitted at the stern, completely submerged and rotating about a vertical axis. It was known as the Fowler wheel, and was quite successful as a means of manœuvring the ship, but its propulsive efficiency was low, due to the use of inefficent paddle sections.

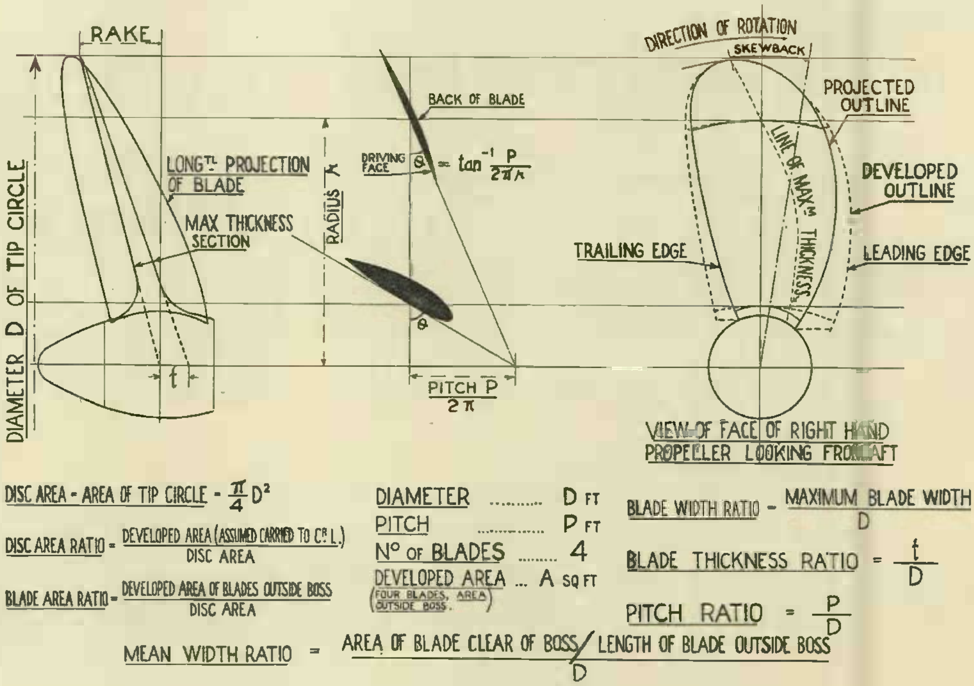


FIG. 2.—*Typical propeller design.*

### 3. Geometry of the Screw Propeller.

In the simplest case, the faces of a marine propeller's blades are portions of a true helical surface, i.e., a surface swept out by a straight line AB (Fig. 1) one end of which (A) advances uniformly

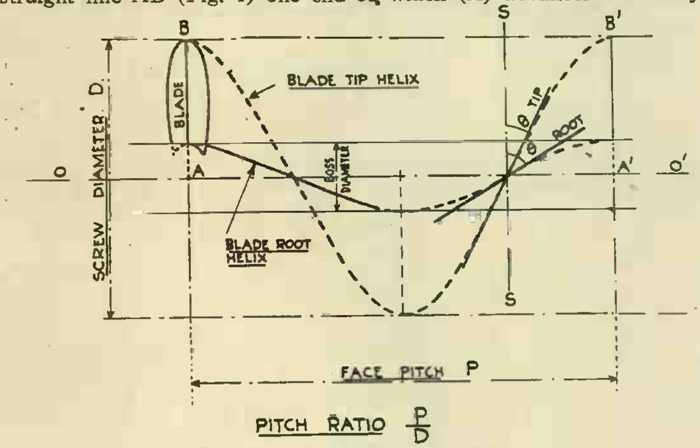


FIG. 1.—*Geometry of propeller.*

along an axis OO', whilst the line itself rotates about the point A with uniform angular speed. When the generating line has made a complete revolution, and is in the position A'B', the distance it has advanced, AA', is called the *face pitch* or *geometrical pitch.*

Any cylinder coaxial with OO' will cut the helical surface in a helix, and the angle between any such helix and a surface normal to the axis is called the pitch angle, $\theta$, as shown in Fig. 1. $\theta$ will be constant for a given helix, i.e., at a given radius, but will increase in value from the tip of the propeller in towards the boss. In practice, the pitch is not always the same at all radii, it being fairly common to have a reduced pitch towards the root of the blade at the boss.

The shapes of blade outline and sections vary greatly according to the individual designers' ideas, and the thickness of the sections is dependent on strength considerations. Fig. 2 shows a typical modern propeller design, and defines most of the terms in common use.

If we consider a section of the propeller blade at a radius $r$ with a pitch angle $\theta$ and pitch P (Fig. 3), and imagine the blade to be working in an unyielding medium, then in one revolution of the propeller, it will advance from A to A', a distance P feet. If we

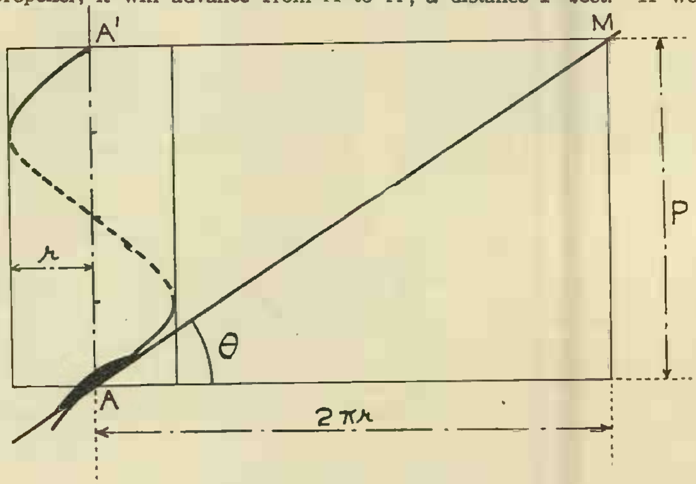


FIG. 3.

develop the helix traced out by A on the cylinder of radius $r$, we shall obtain the straight line AM, and from the geometry of the figure, we see that

$$\tan \theta = P/2\pi r \quad \text{.................................} \quad (1)$$

If the screw is turning at $N$ revolutions per minute, in such a medium, then in one minute it will advance a distance PN feet, and we can obtain a velocity diagram for the section as shown in Fig. 4, which will obviously be of the same shape as Fig. 3.

In actual fact, the medium will yield, and the screw will not advance a distance LM, equal to P×N, in a minute, but some smaller distance LS, the distance MS being called the *slip*, and the ratio MS/ML is called the slip ratio, and the angle MAS is the *slip angle* or *geometrical slip angle*, $\alpha$.

If $V_1$ is the speed of advance of the propeller through the water

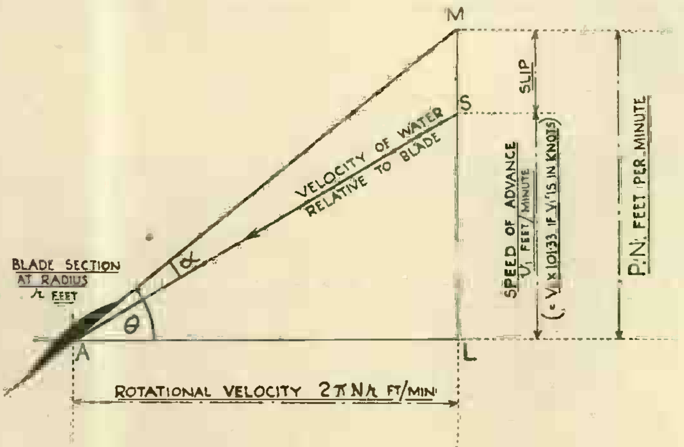


FIG. 4.—*Velocity Diagram for Blade Section.*

in knots, and $v_1$ the same speed in feet per minute, then

$$v_1 = V_1 \times 101{\cdot}33,$$

and by our previous definition

$$\text{slip} = MS = PN - v_1$$

and the *slip ratio* is given by

$$s = MS/ML = (PN - v_1)/PN = 1 - v_1/PN = 1 - 101{\cdot}33\,V_1/PN \quad \ldots\ldots \ (2)$$

Since AL represents the rotational velocity of the section and SL its speed of advance, SA will represent the velocity and direction of the water meeting the blade section, and the angle $\alpha$ is also called the geometrical angle of incidence.

The slip ratio s is an important factor in a propeller's performance, as we shall see in subsequent sections.

### 4. Theories of Propeller Action.

There have been many theories advanced to account for the way in which a propeller produces thrust. At first relatively crude, these have been elaborated from time to time, stimulated particularly in recent years by the great amount of work, both theoretical and experimental, which has been done in the sister science of aerodynamics.

It is not the purpose of this paper to go into all the latest developments of screw theory (which have, indeed, been admirably and fully dealt with in many recent papers), as these are "refinements" rather than the "fundamentals" of the subject. The theories are dealt with here rather as an introduction to those papers, which to many beginners are not easily intelligible at first reading.

First let it be said that no theory yet advanced takes account of all the various factors entering into propeller action, and the practical design of a ship's propeller to suit a given set of conditions still depends on the results of systematic experiments with model screws. On the other hand, a good theoretical knowledge of how a screw works is essential both to the experimenter and to the practitioner, to guide them as to the best methods to pursue in research, the correct interpretation of the results of their work, and as a basis to explain qualitatively the peculiarities of propeller behaviour.

The screw propeller is essentially a reaction machine, taking in water at a certain speed, and throwing it astern at some greater speed. If the screw acts upon $W$ tons of water per second and gives it an acceleration of $f$ ft. per second per second astern, then the force exerted on the water will be

$$W/g \times f \text{ tons}$$

and the reaction of the water on the propeller blades will appear as a forward thrust. Viewed in this way, it was natural that the earlier theories of propeller action were based upon momentum considerations. They did not lead to any clear idea of the action of individual blades, since the propeller had to be idealised into an "actuator disc" or some similar conception which could cause an instantaneous increase in pressure, and a second line of development was based on a study of the action of each section of a blade, leading to blade element theories and the modern vortex theory.

### 5. Momentum Theories.

These theories are due to Rankine[6], Greenhill[7], R. E. Froude[8], and others. In the ideal conception of the propeller, the latter is regarded as a mechanism capable of imparting a sudden increase of pressure to the fluid passing through it, the method by which it does so being ignored.

The assumptions made are:—

(1) The propeller imparts a uniform acceleration to all fluid passing through it;

(2) The mechanism is frictionless; and

(3) There is an unlimited inflow of water to the propeller.

The first assumption involves a contraction of the race column, and since this contraction cannot take place suddenly at the propeller, the actual accelerations must occur outside the disc and be spread over a finite distance fore and aft.

Consider a propeller disc of area $A$ advancing with uniform velocity $v_1$ into undisturbed fluid. The hydrodynamic forces will be unaltered if we replace this system by a stationary propeller disc in a uniform flow of the same velocity $v_1$, as shown in Fig. 5.

At some distance well ahead of the screw, the velocity of the flow will be $v_1$, and the pressure in the fluid $p$ (Fig. 5). Well

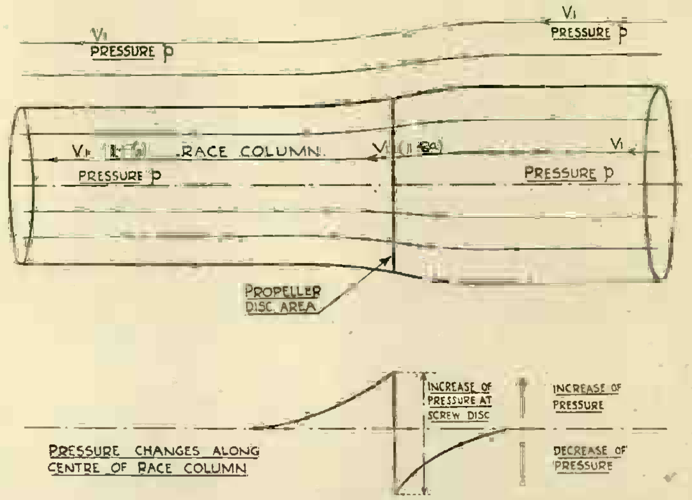


FIG. 5.—*Changes of Pressure and Velocity at Propeller Disc.*

behind the screw, the race column, i.e., the fluid which has passed through the screw disc and been acted upon by the pressure-producing mechanism there, will have some greater sternward velocity, which we may write as $v_1\ (1+b)$. On the other hand, if we neglect any effect of rotation in the race, the pressure will still be $p$, equal to that in the surrounding fluid.

Since the acceleration and contraction of the race column cannot occur instantaneously at the disc, the fluid must acquire some of this increased velocity before it reaches the disc, and the velocity through it will be greater than $v_1$, and we may write it as

$$v_1\ (1+a)$$

where $a$ is an *axial inflow factor*.

The pressure in the race column, which is $p$ well ahead of the screw, will be reduced as the fluid approaches the disc, because of the increase in velocity, as shown in the lower part of Fig. 5. At the disc, the pressure is suddenly increased by the screw mechanism to some value greater than $p$, and then decreases again with the further acceleration in the race until it reaches the value $p$ again well behind the screw when the velocity is $v_1\ (1+b)$.

The theory is developed in Appendix A, and it leads to the important conclusion that

$$a = b/2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (3)$$

i.e., *that one half of the sternward increase in velocity is acquired by the fluid before it reaches the propeller.*

The efficiency of the screw is found to be

$$\eta = 1/(1+a) = (1-s)/(1-s/2) \quad \ldots\ldots\ldots\ldots\ldots\ldots \ (4)$$

where $s$ is the *slip ratio* already defined, and which is thus shown to be a most important factor in the working of the propeller.

The above results neglect any rotational velocity in the race. If the propeller mechanism is a rotating one, and the fluid is not frictionless, some rotation will occur, and the additional rotational kinetic energy must be imparted to the fluid by the propeller. This will involve greater expenditure of power, and a consequent reduction in the efficiency given in equation (4).

Some of this rotational velocity will be acquired by the fluid before it enters the screw disc, just as in the case of the sternward acceleration (Fig. 6), and we can imagine a *rotational inflow factor*, $a'$, analogous to the axial inflow factor $a$. In this case, however, the fluid is set in rotation in the same direction as the propeller disc, so that the rotational inflow factor *reduces* the angular velocity $\omega$ of the disc relative to the water, which accordingly becomes

$$\omega\ (1-a')$$

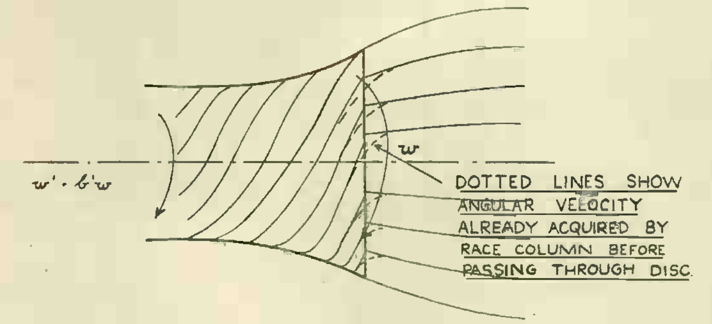


FIG. 6.—*Effect of Race Rotation.*

The above discussion applies to an ideal propeller working in a frictionless fluid, and the results do not help us to design a propeller, nor to visualise the actual mechanism by which the latter develops its thrust. Nevertheless, the conception of inflow factors and the theoretical conclusion that half of the increased velocity is acquired by the fluid before it reaches the propeller, will be found of great assistance and importance in the understanding of modern screw theories.

**6. Blade Section Characteristics.**

In order to follow the reasoning underlying such theories, we must know a little about the behaviour of blade sections when moving through a fluid. Most of our knowledge in this field is derived from aerodynamic research, and the terms used there have been adopted by marine engineers and naval architects.

The fundamental data are derived from tests on long spans, having a constant cross-section shape, placed in uniform flow in a wind tunnel. The two types in which we are most interested are the circular back and aerofoil shapes shown in Fig. 7, which are the

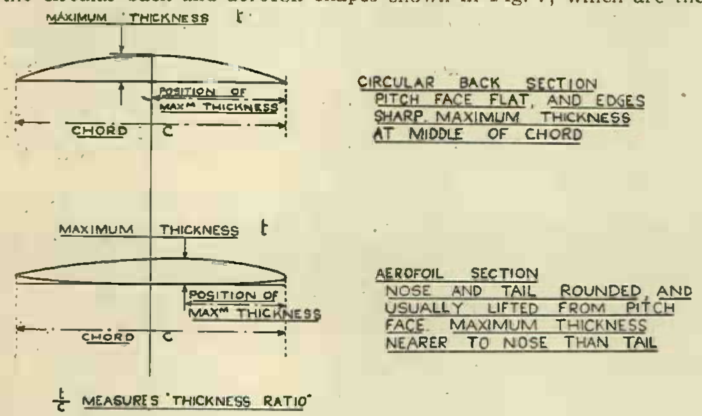


FIG. 7.—*Circular Back and Aerofoil Sections.*

sections mostly used in marine screws. The former has usually a flat pitch face, the back being an arc of a circle, the edges being sharp within the limits of manufacture. The aerofoil section is characterised essentially by having its maximum thickness nearer the nose, and usually has rounded edges and often they are lifted from the pitch face.

When such a span is placed in an air stream with its pitch face set at an angle to the flow (Fig. 8) it experiences a force very

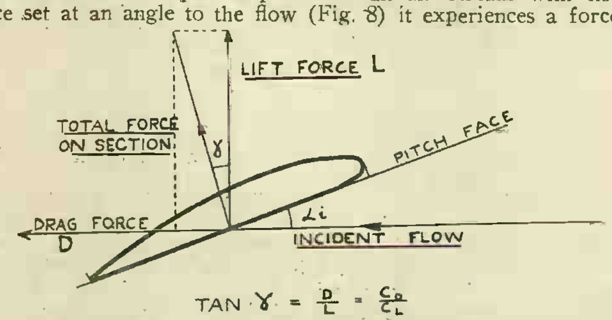


FIG. 8.—*Forces on a Blade Section.*

nearly normal to the face, which is usually resolved into two forces—one normal to the direction of incident flow, called the *"Lift"*, *L*, the other parallel to the flow and called the *"Drag"*, *D*. The angle between the flow and the pitch face is the *angle of incidence*, $\alpha_i$.

These forces are usually expressed in the form of coefficients :—

$$\text{Lift coefficient} = C_L = L/\tfrac{1}{2}\rho A V^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(5)}$$

$$\text{Drag coefficient} = C_D = D/\tfrac{1}{2}\rho A V^2 \quad \ldots\ldots\ldots\ldots\ldots \text{(6)}$$

where $\rho$ = mass density of fluid.
 A = area of plan form of section.
 = chord × span.
 V = velocity of incident flow.

The efficiency of the section as a lifting device can be measured by the ratio

$$\text{Lift/Drag} = L/D$$

since in aircraft wings the provision of maximum lift for minimum drag is of first importance.

When the results of such tests are plotted, a number of interesting points emerge (Fig. 9).

(i) The lift coefficient for small angles of incidence is a linear function of the angle, i.e., $C_L$ plotted against $\alpha_i$ is a straight line.

(ii) Zero lift does not occur at zero incidence, but at a small negative angle. We can thus draw a line from the tail passing above the pitch face which is the *zero lift line* (see Fig. 9), such that when the flow is along this line there will be no lifting force on the section normal to the flow.

(iii) The drag coefficient remains small and more or less constant for small angles of incidence, and then, when the lift coefficient begins to fall off, increases rapidly.

(iv) The lift/drag ratio is a maximum at a small angle of incidence, and for such sections to work efficiently the angle of incidence should be in the neighbourhood of 3 or 4 degrees.

The ratio of span to chord is called the *aspect ratio*. If this ratio were infinite, the flow around a section would be two-dimensional, and the lift distribution along the span would be uniform. With a finite span, a certain amount of "spilling" takes place at the ends, and the lift falls off to zero at those points. The results can be corrected from one aspect ratio to another, and are usually given either for a ratio of 6 or for infinity.

One other feature is of importance in propeller work—the distribution of pressure around a section. A typical example for an aerofoil is shown in Fig. 10. On the face the pressure is increased above that in the free-flowing stream, being greatest quite close to the nose. On the back the pressure is decreased, and with aerofoil sections there is a marked peak in this reduced pressure curve just behind the nose. The lift force generated is the result of the differences in pressure on the two faces of the section, and these reinforce one another, and it is clear from Fig. 10, which is typical, that the reduction of pressure on the back contributes more to the lift than does the increase on the face.

For a circular back section the same general picture holds, but the peak of pressure reduction on the back is not so marked, and for the same area under the pressure curve the maximum reduction in pressure will be less than for an aerofoil. This is important, and will be referred to again. In both types the maxima, both of increase and decrease of pressure, become greater with increase of angle of incidence.

**7. Blade Element Theories.**

In these theories, the screw blades are considered as being made up of successive strips across the blades from leading to trailing edge, and the forces acting on them evaluated from a knowledge of the relative velocity of the strip to the water and the characteristics of the section shape as outlined in the last section (Fig. 11). These elementary forces can then be resolved into the elements of thrust, $\delta T$, in the forward direction, and of torque $\delta Q$, in the plane of rotation. By plotting curves of $\delta T$ and $\delta Q$ along the blade from boss to tip (Fig. 12), curves of thrust and torque loading are obtained which on integrating will give the total thrust T and torque Q for the whole propeller. The efficiency is then

$$\eta = T\, V_1 / 2\pi N Q \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(7)}$$

This theory is developed in Appendix B, and we shall only note here one or two points which it brings out.

In the first place, the working of the blades affects the fluid ahead of the screw, so that the total axial inflow velocity is increased above the speed of advance $V_1$ to some speed $V_1\,(1+a)$, whilst the total rotational inflow velocity is decreased from $2\pi Nr$ to $(1-a')\,2\pi Nr$, in a similar way to that found in the momentum theories. Reference to Figure 11 will show that these two inflow factors $a$ and $a'$ both reduce the angle of incidence at which the water flows on to the blade section to a value considerably below that found on purely geometric considerations neglecting $a$ and $a'$. It must be remembered that this angle of incidence or slip angle is always

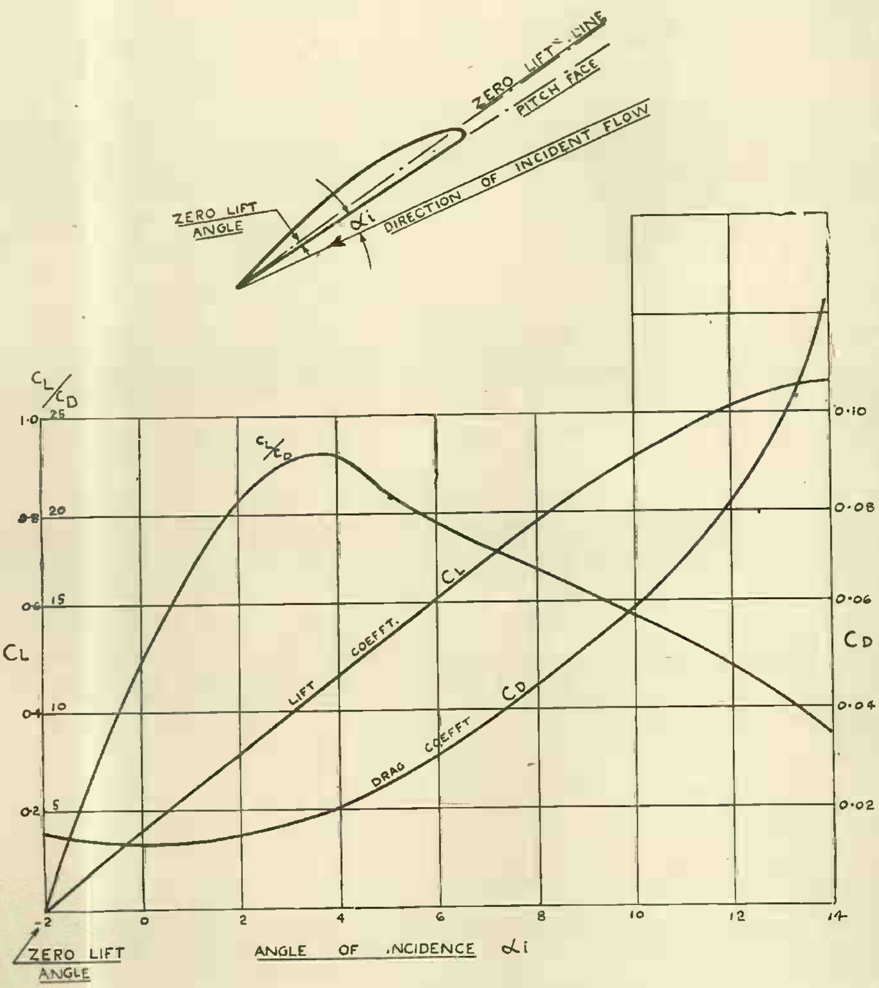


FIG. 9.—*Blade Section Characteristics.*

small in an efficient screw propeller, usually not greater than 3° or 4°. This is as it should be, since we know that both aerofoil and circular back sections give their maximum values of L/D at such angles of incidence.

The elements of thrust and torque on a blade section can be found from data of the kind discussed in para. 6 when the angle of incidence of the fluid to the blade is known. This involves a knowledge of the inflow factors $a$ and $a'$, which can be found by equating the thrust and torque to the axial and rotational momentum put into the race, on the assumption that one half of the acceleration occurs before the fluid reaches the screw disc.

In its original form the blade element theory neglected certain losses in the working of the propeller, with the result that calculated thrusts, torques and efficiencies were in very poor agreement with actual performance. • Later developments in theoretical work on vortex motion have enabled two of these omissions to be rectified to a certain extent —tip losses and blade interference.

We have seen that the pressure on the face of a section is much higher than that on the back—in consequence, at the tips of a blade the fluid tends to "spill" over from face to back, and the continuous operation of this effect causes a vortex spiral to be shed from the tip, which appears as a helical spiral in the propeller race. This tip loss has been the subject of much investigation, and methods of including its effect in the calculation of propeller performance have been given by Goldstein[10], Lock[11] and others.

When one blade is following another, as in a propeller, the pressure field around the second one is affected by the first, and this is important in marine screws, especially near the roots in small pitch ratio propellers. There is a loss in thrust, which can be allowed for if sufficient data are available on the effect of interference between blade sections when run at different distances apart—that is, in "cascade". The most complete information on cascade effect at present available is that due to Gutsche[12] and to Shimoyama[14] but much more experimental work is necessary in this field.

The thrusts and efficiencies of air screws have been predicted with considerable accuracy by the latest vortex theories, the distribution of pressure over an air screw blade has been found to agree well with that over an aerofoil of the same shape of section, and measurements of the pressure and velocity distribution in the neighbourhood of an air screw justify the assumption that half the accelerations occur before the propeller disc.

In the case of marine screws the problem is more difficult. The blades are much wider for their length, and the changes in width are much more rapid than in an air screw blade, causing the flow to be much less two dimensional in character, and hence increasing the tip loss correction. Because of the greater blade

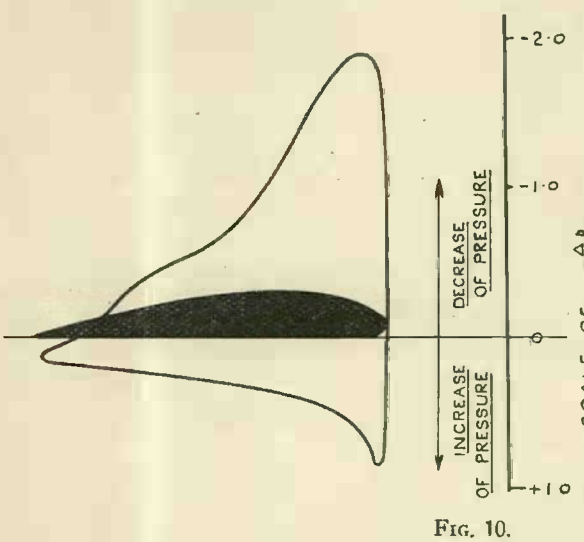


FIG. 10.

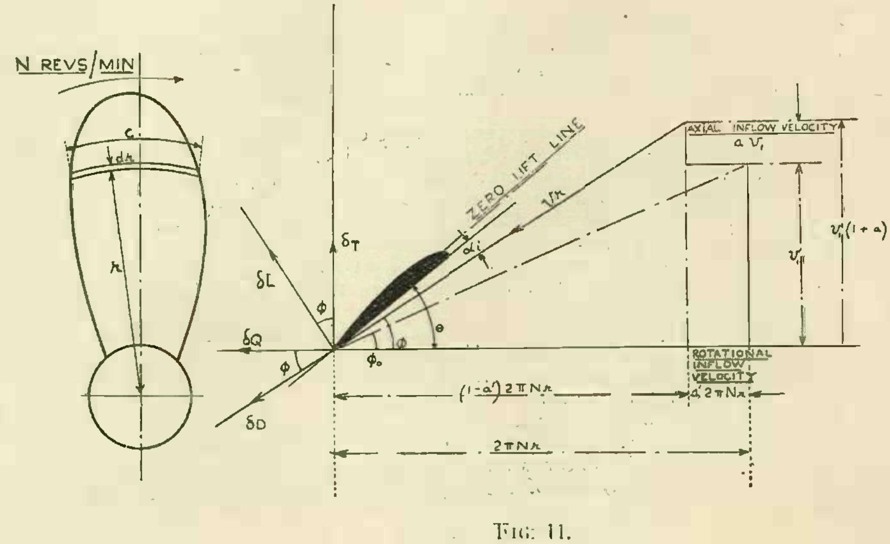


FIG. 11.

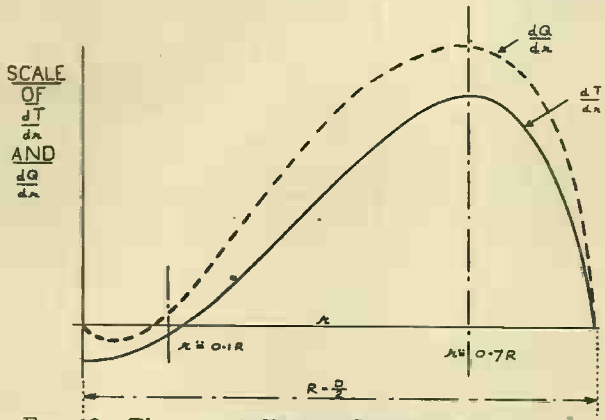


FIG. 12.—*Thrust and Torque Grading Curves along Propeller Blade.*

widths, also, the interference effects are much greater.

But by far the most important difference, and one which greatly restricts the application of such design methods to ship propellers, is the relative positions in which airscrews and marine screws work. In an airscrew working ahead of an aircraft, the inflow is more or less uniform over the disc, and is known fairly accurately. In a marine propeller, working behind a ship, the velocity of the water approaching the propeller disc will vary from point to point over it, the actual variation depending upon the shape of the hull in front of it, the presence of bossings and other appendages, the position of the propeller, the state of the hull surface and the speed of the ship. We shall deal with this question later, but enough has been said to show that the angle of incidence of the water to a given propeller blade section will be varying continuously as it rotates, and no unique ideal design solution will exist.

Another feature of screw propeller performance brought out by Fig. 11 is the fact that the effective pitch is greater than the geometric pitch. When the flow conditions are such that $\alpha_1$ is zero, i.e., the relative velocity $v_r$ is coincident with the pitch face of the section, the thrust provided by the element will not be zero, since the flow is still at a positive angle of incidence relative to the zero lift line. If we calculate slip ratio by using the geometric pitch, therefore, the screw will still deliver thrust when the slip ratio is zero.

The theory of screw propellers outlined above cannot yet replace the use of experiment data—whether model or full scale—but some knowledge of it is of inestimable value in understanding propeller action, in avoiding mistakes in design, and attacking the questions of cavitation, erosion and singing. To those who wish to go more deeply into this subject, a very complete exposition of the application of theory to design has recently been given by Burrill[13] and in three recent papers (15, 16, 17) Baker has dealt with the losses inevitable in a propeller due to its having pitch and experiencing friction, the estimation of effective pitch and the effect of roughness of blades and blade interference on propeller efficiency.

### 8. The Use of Model Experiments.

The use of model experiments to determine ship resistance and the best form of hull to suit given conditions (described in paper 1) was followed by a similar application of model technique to screw propellers.

In the first instance such experiments were confined to measuring the thrust, torque and efficiency of propellers dissociated from any hull—what is referred to as "open water" performance. In the modern experiment tank this is done by running the propeller on a long shaft projecting well ahead of a narrow propeller "boat" containing the recording apparatus. The propeller itself thus advances into undisturbed water, so that its speed of advance $V_1$ is known. Records of thrust, torque and revolutions are taken automatically inside the boat.

It can be shown that if we neglect viscosity, the coefficient

$$T/\rho D^2 V_1^2$$

will be the same for model and full-sized propeller provided that $V_1^2/D$ and $ND/V_1$ are the same in the two cases.

The second condition is equivalent to saying that both screws must be run at the same slip, and the former that if the ratio of the full size to model linear scale is *m*, then the revolutions per minute of the model (*n*) and of the ship (*N*) must be related by the equation

$$n = N\sqrt{m}$$

Thus whilst the speed of advance of the model screw is much less than that of the full size screw, its revolutions per minute are much higher.

The neglect of viscosity in this analysis gives rise to the possibility of some difference between model and full size screws, which is usually referred to as "scale effect", and this must be kept to a minimum by using as large a model screw as possible.

If the model results were plotted as curves of $T/\rho D^2 V_1^2$ and $Q/\rho D^3 V_1^2$ to a base of slip ratio, therefore, the values should be directly applicable to the ship. This is a method often used in model work, but these coefficients have the disadvantage that they become infinite for zero speed of advance (e.g., for the bollard condition in tugs) and it is usual to use the slightly different forms :—

$$\left.\begin{aligned} K_T &= T/\rho N^2 D^4 \\ K_Q &= Q/\rho N^2 D^5 \\ J &= V_1/ND \end{aligned}\right\} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (8)$$

where $\rho$ = wt. of unit volume of fluid/g

In any consistent system of units these coefficients are non-dimensional. An example of such results is given in Fig. 13.

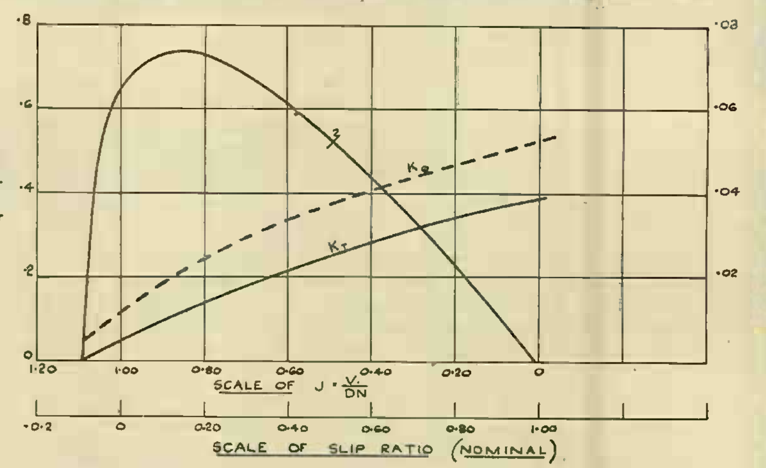


FIG. 13.—*Typical Curves of Thrust and Torque Coefficient and Efficiency for a Screw in "Open".*

$$\left.\begin{aligned} J &= V_1/ND \\ K_T &= T/\rho N^2 D^4 \\ K_Q &= Q/\rho N^2 D^5 \end{aligned}\right\}$$

Where $V_1$ = Speed of advance.
$N$ = Revolutions in unit time.
$D$ = Diameter.
$T$ = Thrust.
$Q$ = Torque.
$\rho$ = Mass density of fluid.
= WT per unit volume/g.

4 bladed screw 0·40 blade area ratio.
1·00 face pitch ratio.

This shows that the screw reaches its maximum efficiency at slips of about 15 to 20 per cent., and also that the thrust does not vanish at $J = 1{\cdot}0$, but at some higher value (nearly 1·10 in this case). This is due to the effect of the zero lift angles of the blade sections, and the value of J at which the thrust does vanish can be used as a measure of the effective pitch ratio.

The first model experiments with a series of geometrically similar screws of different pitch ratios were made by R. E. Froude and published in 1908[18]. Since then more results have become available—Gawn has extended Froude's series to cover much wider blades[19], Taylor in America has run many such families of screws[20], Hughes has run a series of 3 bladed screws[21] and Schaffran in Germany covered a large range of pitch ratios[22]. The most recent and perhaps most useful systematic experiments are those made at Wageningen in Holland by Troost and van Lammeren[23&24], since they include 4 and 3 bladed screws of 2 blade area ratios in each case and cover a much wider range of revolutions than other series, a very valuable quality in these days of fast-running diesel engines.

For design purposes, the above coefficients are not very suitable. Various others have been suggested, but those introduced by Taylor and subsequently adopted by Baker and Troost are probably the most useful for everyday use. The results for a family of screws of the same geometrical design but differing in pitch ratio are then presented in a chart such as that shown in Fig. 14, in which are given contours of constant efficiency $\eta$ and advance coefficient $\delta$ with abscissa $B_p$ and ordinate pitch ratio, where

$$\left.\begin{aligned} B_p &= NS\ /V_1^{2\cdot 5} \\ \text{and } \delta &= ND/V_1 \end{aligned}\right\} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

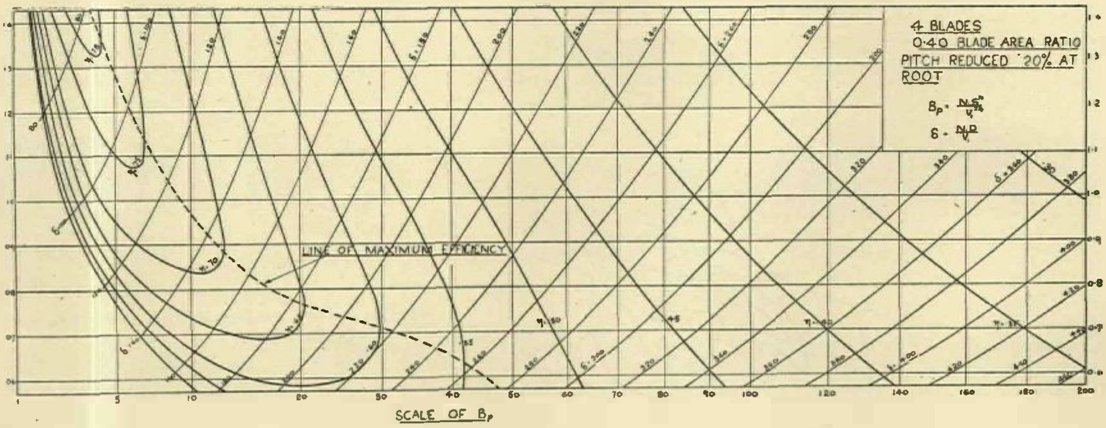


FIG. 14.—*Typical Screw Design Chart—Adapted from Troost.*

In these expressions:—

N = revolutions per minute;
S = horse power absorbed by propeller;
$V_1$ = speed of advance of propeller in knots;
and D = screw diameter in feet.

All these quantities can be estimated when dealing with a propeller design, and from the chart the best combination of diameter, pitch and revolutions selected.

**9. Interaction Between Hull and Propeller.**

When the propeller is working in its correct location behind the model or ship hull, the conditions are modified in several ways.

The water at the stern has, in general, a forward motion due to the hull ahead, and this forward current is called the *wake*. In consequence the true speed of advance of the screw is not the speed of the ship, V knots, but some lower speed, $V_1$, which must be used in the process of designing the screw from charts.

The wake speed is, of course, not the same at all points of the screw disc and the propeller behind the hull will experience a variation of feed velocity and direction from point to point which is absent for the screw in open. Hence, at the same speed of advance and same thrust, the torque and therefore the efficiency will not necessarily be the same—we thus introduce the idea of an *"open efficiency"* $\eta_0$ and a *"behind efficiency"*, $\eta_B$, the ratio of one to the other being called the *"relative rotative efficiency"*, $\eta_B/\eta_0$.

As we have seen when discussing propeller theories, the water is accelerated for some distance ahead of the propeller, and in consequence the pressure there is also decreased. With the hull ahead of the screw, this means that the water pressure over the after part of the ship or model is not so great as when being towed with no propeller, and the resistance is therefore greater by an amount known as the *"augment of resistance"*.

Information on all these factors can only be derived from model experiments, where the performance of the screw can be determined both in open and behind the model. In modern experiment work, the model is fitted up with stern tube and shafting and all external fittings such as sternpost, rudder and bossings. The screw is driven from inside the model and the thrust, torque and revolutions per minute of the propeller are automatically recorded, as are the forward speed of the model and any difference between the thrust of the screw and the resistance of the model, which is constrained by the travelling carriage to run at a constant desired speed. By running a number of such experiments, all at the same speed but at different revolutions, the actual self propulsion point can be accurately determined, and so the correct revolutions, thrust and torque. From this information, together with the open performance of the screw and the hull resistance data, the propulsive coefficient can be determined, and the relative efficiency of different designs of propellers, types of rudder or bossings, or of different revolutions, can be quickly and cheaply obtained. The values of wake and augment of resistance are also easily derived from the results, and this information is of great value in interpreting the experiments.

**10. Wake Fraction.**

Suppose that a propeller driving a hull at V knots develops a thrust T and turns at N revolutions per minute. When this propeller is run in "open water", if the speed and revolutions are maintained at V and N, then it will develop considerably less thrust than behind the hull. In experiments in this country, the method used to obtain the wake is to vary the speed of advance in open water until such a value $V_1$ is obtained that the screw delivers the same thrust T at the same revolutions N as when propelling the hull at speed V.

Then the difference $(V-V_1)$ is the *effective wake speed.*

Froude expressed the value of the wake speed as a fraction of the speed of advance $V_1$, calling this ratio the *wake fraction*, $\omega$, so that

$$\omega = (V-V_1)/V_1$$

and $V_1 = V/(1+\omega)$ ...... (10)

and the expression $(1+\omega)$ is called the *wake factor*. For a forward wake $\omega$ is positive.

Taylor in America introduced another wake fraction, $\omega_T$, by expressing the wake speed as a fraction of the ship speed, so that

$$\omega_T = (V-V_1)/V$$

or $V_1 = V(1-\omega_T)$ ................................ (11)

The second definition has much to recommend it, since a wake of 50 per cent. then means that the wake speed is 50 per cent. of the ship's speed, whereas a 50 per cent. wake in the Froude notation infers that the wake speed is 33 per cent. of the ship's speed. This difference must be carefully remembered in using published data on model work, the British practice being to follow Froude's notation and the American Taylor's. The two wake fractions are related by by the equation

$1/(1-\omega_T) = 1+\omega$ ................................ (12)

It will be seen that this method is based on the identity of thrust behind and in open. A similar wake fraction could be found based on torque identity, and would not, in general, be the same as the thrust wake, except when the relative rotative efficiency has a value of unity. Here again British and American practice differs, the former using thrust wakes whilst the Washington Experiment Basin uses the average of thrust and torque wakes.

The wake is due to three principal causes:—

(1) The frictional drag of the hull surface causes a following current which increases in velocity and volume towards the stern, and produces there a wake having a considerable forward velocity relative to the surrounding still water.

(2) The stream line flow past the hull causes an increased pressure at the bow and stern and a decreased one along the centre part of the length—(see Section 8 of paper 1). This means that at the stern the relative velocity of the water past the hull will be less than the ship's speed, and will appear as a forward or positive wake augmenting the frictional wake.

(3) The ship forms a wave pattern on the surface of the water, and the water particles in the crest of such waves have a forward velocity due to their orbital motion, whilst in the troughs the orbital velocity is sternward. This orbital velocity will give rise to a wake component also, which may be positive or negative according as to whether there is a crest or a trough of the wave system in the vicinity of the propeller.

The total wake is made up of the three components, and is in the vast majority of cases positive. The only exception arises in very high speed craft, such as destroyers and high speed motor boats. In the former, for example, the hull is very fine and the frictional and stream line wakes are small. At a speed of about 34 knots, the wave length of the system created by the ship will be some 640 feet, so that a destroyer 320ft. in length would have a trough in the neighbourhood of the screws, and the "wave" wake would be negative and usually sufficient to about balance the other two components, giving rise to a zero or slightly negative total wake.

The wake factor $\omega$ obtained from behind and open data in this way is, of course, an "average" or "effective" wake. If we wish to find the distribution of wake over the screw disc, other means must be adopted. The wake at actual points in the disc can be obtained by pitot tube measurements. For a single screw ship, such measurements show that the wake will be most intense over the upper part of the disc, rather less so down the vertical centre line, and much smaller over the outer lower quadrants, somewhat as shown in Fig. 15. For a twin screw ship, the average wake will, as a rule, be less than in a single screw ship of the same fullness, but there will be a high concentration across the disc immediately behind the bossing ends, and another in that part of the

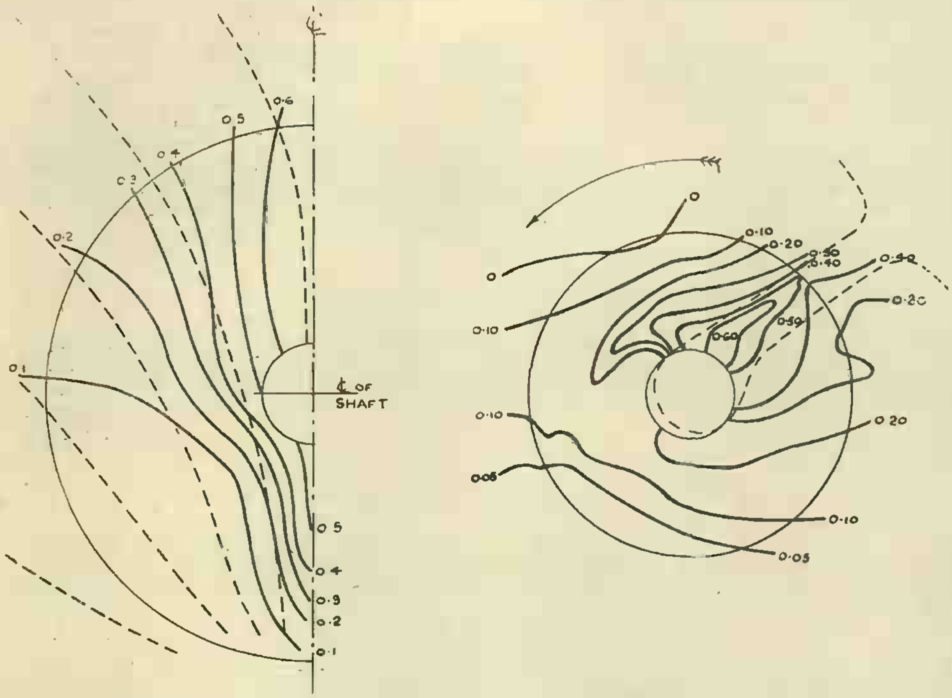


FIG. 15.—*Typical Wake Distribution Over Propeller Disc. Wake Factors are given in Taylor's Notation,* $\omega_T$

disc nearest the hull, especially if the clearance between blade tips and hull is small, so bringing the former into the heavy frictional wake near the surface of the hull.

As a propeller blade revolves, a section at any given radius will thus pass through regions of very different wake concentrations. We can make the propeller with a varying pitch to suit different distributions of *average* wake, and at any given radius the pitch is fixed, and must be fixed, to suit the average wake round the circumference of a circle of this radius. It is thus of interest to measure the wake over such elementary annuli, and this can be done by using blade wheels of different radii or rings of different diameters and measuring their resistance. Both these methods and the use of pitot tubes give the wakes which exist without the propeller in action, and they do not therefore agree with the effective wakes found from the correlation of open and behind results. They are, however, of the greatest value in the interpretation of screw performance.

**11. Real and Apparent Slip Ratios.**

We have defined slip ratio by the equation (2), i.e.,

$$s = 1 - v_1/PN$$

where $v_1$ was the speed of advance in open water. For the screw working behind the hull, it is evident that we can calculate two slip ratios, one using the ship speed v, the other the speed of advance $v_1$ after we have allowed for the effective wake over the propeller disc. These two ratios are referred to as the apparent and real slip ratios respectively, so that

$$\left.\begin{aligned} \text{apparent slip ratio} &= s_a = 1 - v/PN \\ \text{and real slip ratio} &= s = 1 - v_1/PN \end{aligned}\right\} \qquad (13)$$

The former can be calculated from ship figures only, whereas the second, which is the only real guide to the ship's performance, requires a knowledge of the wake fraction $\omega$.

**12. Relative Rotative Efficiency.**

Knowing the wake fraction from the results of open and behind experiments with a propeller, the open and behind efficiencies can be calculated, being given by the expressions

$$\left.\begin{aligned} \eta_B &= TV_1/2\pi NQ \\ \text{and } \eta_o &= TV_1/2\pi NQ_o \end{aligned}\right\} \qquad (14)$$

Hence

$$\text{R.R.E.} = \eta_B/\eta_o = Q_o/Q \qquad (15)$$

If, therefore, it requires the same torque behind as in open to produce the same thrust T at the same revs. per minute N, the relative rotative efficiency will be unity, but not otherwise.

The difference in torque actually found is due to two main reasons—because of the heterogeneous wake behind the model, the flow conditions over a given blade section as it rotates differ from those in open with a homogeneous wake, so that the efficiency of the element will not necessarily be the same, and because of a possible difference in turbulence behind the hull as compared with open water giving rise to scale effect.

The value of relative rotative efficiency does not depart materially from unity, being between 0·95 and 1·0 for most twin screw ships and between 1·0 and 1·10 for single screw.

**13. Augment of Resistance and Thrust Deduction.**

When a hull is towed, there is an area of high pressure over the stern which, by its resultant forward thrust, provides a component reducing the total resistance. With a self-propelled hull, however, some of the water around the stern eventually becomes the race column, and by acquiring half its acceleration before entering the screw disc, materially reduces the pressure on the stern of the hull, thereby increasing its resistance and hence the thrust necessary to propel the model or ship. Thus it is found in model work, where the necessary measurements can be made, that if the resistance of a hull when towed is R, the thrust T necessary to propel the model at the same speed V is greater than R, and the increase is called the *augment of resistance,* and is expressed as the ratio of the increase in thrust required to the resistance, so that

$$a = (T-R)/R = T/R - 1 \text{ or } T = (1+a)R$$

Although this is the more logical way of viewing the problem, the common practice is to look upon this increase in resistance as a deduction from the thrust available at the screw, so that although the screw provides a thrust T tons, say, only R tons are available to overcome resistance, and this "loss of thrust" $(T-R)$, expressed as a fraction of the thrust T is called the "thrust deduction coefficient", $t$, where

$$\left.\begin{aligned} t &= (T-R)/T = 1 - R/T \\ \text{or } R/T &= 1 - t \end{aligned}\right\} \qquad (16)$$

$(1-t)$ being called the "thrust deduction factor".

In modern model work, it is the common practice to fit rudders and other stern appendages for the self-propelled tests, and this has introduced a new feature into the measurement of thrust deduction. R in equation (16) remains the resistance of the naked model (i.e., with no appendages) but T now has to overcome not only the augmented resistance $R(1+a)$ but also the resistance of the rudder and other appendages. This means that the values found for $t$ will not only depend on the shape of hull and propeller characteristics as reflected in the augment $a$ but also on the type of rudder fitted to the model. This problem has been investigated fully by the author and the results published in 1934[28], but they may just be referred to very briefly.

A model representing a 400ft. cargo ship was run self-propelled, but with no rudder or sternpost. The value of $t$ was found to be 0·2, and this was in the main due to the augment of resistance effect. With a plate rudder and square sternpost behind the hull, the value of $t$ went up to 0·29, this representing a considerable loss in propulsive efficiency. When the fore side of the post was faired off into a fin, this value dropped to 0·24, showing that practically all the gain in efficiency from the fin was due to the reduction in head resistance of the post. To carry the matter a little further, the rudder and post were next carried separately from the model, and maintained in their correct position by springs, so that their resistance could be separately measured. Using as the resistance in calculating $t$ the sum of naked hull and this separately measured appendage resistance, the value of $t$ came out the same for all conditions, 0·20, and the changes in propulsive efficiency were then exactly reflected in the changes in total resistance.

When using published figures for thrust deduction coefficient for the purpose of design, therefore, it is most important to know the exact stern conditions on the model for which they were obtained.

**14. Hull Efficiency.**

The work done on the ship in moving her at a speed of V against a resistance R is measured by the product R.V and the power absorbed is called the effective horse power, E.H.P. :—

$$\text{E.H.P.} = (RV \times 101{\cdot}33)/33{,}000 \qquad (17)$$

where R is in lb. and V in knots (see paper 1, section 2).

The work done by the screw is the product of thrust (T) and the speed of advance at which it is delivered ($V_1$), so that the thrust horse power, or T.H.P. is

$$\text{T.H.P.} = TV_1 101{\cdot}33/33{,}000 \qquad (18)$$

The ratio of the work done on the ship to that done by the screw is called the *hull efficiency, h,*

$$\text{so that } h = \text{E.H.P./T.H.P.} = \text{R.V./T}V_1$$

$$\text{i.e., } h = (1+\omega)(1-t) \qquad (19)$$

from equations (10) and (16).

**15. Propulsive Efficiency.**

The overall propulsive efficiency for reciprocating engines is usually given in the form

Propulsive coefficient = E.H.P./I.H.P.

For turbines, the S.H.P. at the turbines is used instead of I.H.P. It is obvious that the coefficient expressed in this way includes the mechanical efficiencies of the main engines, and to avoid differences in this factor a new propulsive efficiency was introduced which measures the relation between the E.H.P. and the power absorbed by the screw, called the delivered horse power, D.H.P., so that

$$\text{Quasi-propulsive coefficient (Q.P.C.)} = \text{E.H.P./D.H.P.} \qquad (20)$$

This can be split up into different elements :—

$$\text{Q.P.C.} = \text{E.H.P./D.H.P.} = \frac{\dfrac{\text{R.V.} \times 101{\cdot}33}{33{,}000}}{\dfrac{\text{T.}V_1 \times 101{\cdot}33}{\eta_B}} = \frac{\text{R.V.}}{\text{T.}V_1} \times \eta_B = (1-t)(1+\omega)\frac{\eta_B}{\eta_0} \cdot \eta_0$$

$$\left.\begin{array}{l}\text{i.e., Q.P.C.} = (1-t)(1+\omega)(\text{R.R.E.}) \times \eta_0 \\ \text{or Q.P.C.} = \text{hull efficiency} \times \text{R.R.E.} \times \text{screw efficiency in open}\end{array}\right\} \qquad (21)$$

The quasi-propulsive coefficient is thus found to depend on the different elements we have already studied, and this brings out their importance in all attempts at estimating powers for new designs.

**16. Cavitation.**

When discussing blade section characteristics in para. 6, attention was called to the distribution of pressure around a section, a typical example of which is shown in Fig. 10. It is evident from such pressure diagrams that the reduction of pressure on the back contributes more to the thrust of the screw than does the increase of pressure on the face. As the revolutions of the propeller are increased and the velocity of the blade section through the water increases, the maximum reduction of pressure on the back grows larger and larger. There is a limit to this reduction, and when the peak pressure has fallen to that of the pressure of water vapour, no further reduction is possible, and for any higher revolutions the water will no longer remain in contact with the blade at that point, bubbles or cavities appear which are filled with water vapour (and some air which has been in solution in the water), and for this reason the phenomenon is known as cavitation. As the revolutions are further increased, the area of the back of the blades affected by cavitation spreads, and the screw develops less thrust than would otherwise be the case. This loss of thrust is the major effect of cavitation, and may be responsible for a vessel failing to reach the desired speed. There are other effects—erosion of the blades, vibration, and possibly cavitation is involved in some way with singing—and altogether it is a feature of propeller operation which is to be avoided at almost any cost.

The maximum pressure reduction on the back of the blade which may occur before cavitation begins will obviously depend to a large extent upon the total pressure head in the water at the point where the particular section is working at any instant. This pressure head is made up of the actual water head plus the equivalent water head of the atmospheric pressure. In model experiments the former is reduced to scale but the latter remains at its full value. At any particular section of a model propeller blade, therefore, the pressure is much greater than it should be for true similarity and as a result model propellers can be run to much higher slips than the ship propellers before cavitation occurs. This does not invalidate the model results so long as cavitation does not occur on the ship, because the excessive pressure acts equally on both faces of the blades, but in high speed craft such as destroyers and cross channel ships where cavitation is to be feared, no indication of its onset will be given by the model screws. To obtain information on cavitation, some other model technique is necessary, and this has led to the use of cavitation or water tunnels, in which the model propeller is placed in a closed water circuit, the water is circulated past it at the desired speed, and the air above the water is removed to the degree necessary to give the correct scale pressure at the propeller. Curves of $K_T$, $K_Q$ and $\eta$ can then be obtained and visual observation made of the propeller's behaviour. Such experiments show that as slip is increased, the first visual effect is that the tip vortices become cored, and are visible as helices trailing down the race column, one from each blade. There is no apparent loss of thrust at this stage. At somewhat higher revolutions, a "blister" appears on the back of the blade at the tip, and the tip vortices become wider and give to the trailing vortices a ribbon-like appearance. From this point onwards there is a definite loss of thrust as compared with the propeller at the same revolutions under non-cavitating conditions, and this loss increases whilst the blister gradually spreads over the whole of the back of the blade with increasing revolutions. This is what is usually termed the "cavitation region". With still further increase in revolutions, no further cavitation can occur on the back, as it is already completely denuded of water, and the increased speed of the section gives a greater pressure on the face, and the slope of the thrust curve increases again, the propeller now working in what is generally referred to as the "super-cavitation" region. Curves for a screw working in the tunnel at Teddington throughout this whole range were given by the author in the discussion on M. Posdunine's paper before the Institution of Naval Architects in 1944, and anyone particularly interested in this matter should study that paper and the discussion.

The erosion of the backs of propeller blades is usually attributed to the collapse of the cavitation bubbles as they move into regions of higher pressure towards the trailing edge, and for screws working in the cavitation zone this erosion is usually serious and quite rapid—it has been noticed on destroyer propeller blades after only 2 hours full power trial.

For screws in the super-cavitation zone, on the other hand, little or no erosion occurs, presumably because all the bubbles are carried away into the race, and none collapse on the blades. Thus in high speed motor boats where super-cavitation conditions have to be accepted because of the need for small, fast running screws driven by light, high speed engines, the propellers do not suffer from serious erosion or pitting.

The efficiency of the cavitating screw is considerably less, and with fully developed cavitation may fall as low as 30 per cent., but in certain cases this has to be accepted.

In addition to cavitation associated with general pressure distribution, it may also result from local causes due to the shape of the blades. Particular care is necessary in finishing the backs of the blades especially in the region of the maximum pressure reduction to ensure as even a curvature as possible. At very low slips, face cavitation may occur near the roots at the leading edge, but this is not normally of any great consequence.

Many criteria have been proposed as a means of predicting the probable presence or absence of cavitation in a new design. The earliest was based solely on the average thrust intensity per unit area of blade surface, but from what has been said above it should be obvious that this alone is not a sufficient test. For example, for the same lift coefficient the maximum decrease of pressure on the back of an aerofoil section may be 15 per cent. greater than on a circular back section of the same thickness ratio, and the reduction of pressure and consequently the onset of cavitation would therefore occur at lower thrust and lower loading per unit area on the aerofoil screw than on the circular back one. For this reason the blades of destroyer propellers are always made with the latter type of section, even though the efficiency at normal cruising speeds thereby suffers a little. Any suggested criteria must take account of these factors, and it is extremely difficult to find one which is at once simple and yet reliable. Charts showing the limiting values of pressure per unit area and formulæ for calculating whether a given screw will cavitate have been given by many investigators—Eggert[26], Lerbs[27], Burrill[28], van Lammeren and Troost[29], for example —to which reference may be made. A detailed investigation of the pressure distribution around the blade sections can also be carried out on the basis of the blade element theory outlined earlier in this paper. Knowing the real incidence angle at which any particular section is working, the pressure distribution can then be found from tests made in the wind tunnel on aerofoils of the same or similar shape, and the maximum reductions on the back determined and compared with the total head due to water and atmosphere.

The true angle of incidence will depend very largely on the wake value, and if the calculation is made using the average wake around the whole circumference at any particular radius, then cavitation will actually occur at somewhat lower thrusts because of the local concentration of wake at certain points, and some margin is necessary to allow for this feature. For the same reason, cavitation will be delayed by any means which reduces the variation of wake over the disc, and in screws where such trouble is probable, especial care should be given to clearances between propeller and hull. The ends of bossings in twin screw ships should be made as fine as possible consistent with strength requirements, the tip clearance from the hull should be sufficient to avoid the greatest wake near the hull, and the blades should be raked to this end. The whole bossing should also be designed to interfere as little as possible with the normal flow. In single screw ships the aperture should be designed to give good clearances between the stern frame and rudder post and propeller blades, and the stern frame should be shaped and not square in section.

The effect of local wake concentrations on the ship also introduce complications into model testing in the cavitation tunnel, and the ship screw will show cavitation earlier than the model screw in the tunnel, where there is no such variation in the inflow velocity. A great deal has yet to be done in this field, and in particular much data is required on the behaviour of aerofoil and circular back sections under cavitating conditions in order that calculations can be made, similar to those outlined earlier in this paper, but under reduced pressure. Although cavitation is normally looked upon as a failing of high speed ships only, it is probable that many propellers of merchant ships suffer from it to a small extent when being over-loaded in bad weather or driving a ship in need of docking and cleaning.

**17. Factors Affecting Propeller Efficiency.**

The many systematic series of model propellers which have been run allow us to draw certain conclusions regarding the effect of several factors on the open efficiency of the propeller.

(a) Diameter. In general the larger the diameter the better the efficiency—few ships suffer from having too large a screw. If we consider the loading as being measured by the thrust coefficient $K_T$ (equation (8)) this latter will decrease with increase in diameter (for the same required thrust), and model tests show that in general the lower $K_T$ results in a higher efficiency. There are, of course, exceptions to this, especially in the case of small, high revolution diesel engines, and the diameter has often to be limited for other reasons than efficiency—the tips must be given adequate cover to ensure absence of racing and air-drawing in moderate seas, and sometimes the height of shaft centre is a modifying feature. In twin-screw vessels increase in diameter means bigger bossings and consequently greater resistance, which may more than offset the gain in propeller efficiency, and in such ships the factor of vulnerability of the screws to accidental damage is also of importance.

In any given case the optimum diameter can only be fixed by careful investigation of all the factors involved.

(b) Pitch ratio. The pitch ratio, in association with the diameter chosen, decides the slip at which the screw will work. From what has been said above, it should be clear that the real slip ratio should be in the region of 20 to 25 per cent. to ensure maximum efficiency and the screw dimensions should be chosen accordingly. Slips below 15 per cent. should be avoided—if our assumptions are only a little in error in such a case, and the real slip is only slightly less than this, a serious loss in efficiency will result, as is evident from Fig. 13. At low slips—from about 20 to 40 per cent.—high pitch ratios give the best efficiency, whilst at higher slips, low pitch ratios are to be preferred.

(c) Number of blades. If additional blades of exactly the same size are successively added to a propeller, the efficiency falls off in passing from two blades upwards, due to interference effects between the blades becoming more marked. On the other hand, two bladed propellers are not used in ships, largely because the restriction in diameter would mean very wide blades to obtain reasonable thrust loading, and because of the uneven torque which would result. There is not much to choose between 3 and 4 bladed designs—the former are a little more efficient at low slips, whilst at high slips the four bladed are as good or better.

(d) Blade area. Excessive blade area causes a loss of efficiency, especially with low pitch-ratio screws, where blade interference effects are thereby increased. On the other hand, too narrow blades lead to thick sections which also cause a loss in efficiency. The normal propeller has a blade area ratio of some 0·40, and this has to be increased only where the thrust loading per unit area is too high. For average merchant ship propellers an increase of 10 per cent. in area causes a loss of about 2 per cent. in efficiency, but there is a strong school of thought which believes that whilst this loss occurs in smooth water, in rough weather the added area is a valuable aid to speed-keeping qualities.

(e) Blade outline. Most blades are elliptical in outline, and the chief difference between designs is the amount of skewback given to the blades. It is doubtful if this feature has much effect upon efficiency in open water, but behind the ship it may reduce the thrust and torque variation as the blade passes into the regions of concentrated wake. For highly loaded screws wide-tip blades have sometimes to be used in order to obtain the necessary area, and whilst these give more thrust and torque at the same slip than narrower tip designs, they are less efficient.

(f) Thickness of blades. Thin blades are to be preferred on two counts—they are more efficient as lifting sections, i.e., their lift/drag ratio is higher, and they delay the onset of cavitation. The actual thickness in any given propeller will depend on strength considerations, and if this is found to differ materially from that of the parent propeller of the series being used for the purpose of design, some adjustment of the pitch will be necessary—if the actual blades are to be made thicker, for example, the zero lift line will be altered so as to give a virtually greater effective pitch, and the face pitch will need to be reduced accordingly.

(g) Rake. Rake of the blades increases the clearance of the tips from the hull and bossing webs in twin screw ships, and from the stern frame in single screw, thus avoiding some of the eddies shed by these features, and reduces the tendency to vibration. On the other hand, due to the additional centrifugal forces introduced it necessitates thicker blades. Rakes up to 15° have been found beneficial in model tests.

(h) Shape of blade section. With the thick sections near the root of the blade, aerofoil shapes are found to be more efficient, but for the outer sections the superiority of aerofoil over circular back shape is much less, and since the latter will go to higher loading before cavitation begins, most average or highly loaded screws have circular back sections over the outer third of the blade. It is claimed by at least one experimenter that these circular back sections near the tip also avoid or at least mitigate another trouble experienced in certain marine screws—"singing"—and they have been rather widely adopted of recent years.

**18. Propeller Design.**

The design of a propeller is almost invariably based in the first instance on a standard chart giving the results of open water tests on a series of geometrically similar model screws, such as that shown in outline in Fig. 14.

The parameters used in this connection are $B_p$ and $\delta$, defined in equation (9).

The information usually available may be listed as follows :—

(a) Principal dimensions and proportions of the ship.
(b) The E.H.P. derived either from tests on a model of the ship or estimated from published data.
(c) The engine power and rated revolutions.
(d) The speed of the ship.

The first step is to make an estimate of the power required at the desired speed. This can be done from the E.H.P. by estimating the probable value of the quasi-propulsive coefficient, after which the power at the propeller is given by

$$S = (\text{E.H.P.} + \text{allowance})/\text{Q.P.C.}$$

The allowance is necessary to take account of appendage resistance such as bilge keels, air resistance of the above water form, and the effect of weather. The standard allowance for the first two items in a single screw ship is taken as 8 per cent. for ideal measured mile conditions—smooth sea, no wind and a clean ship. For a twin screw vessel, a further addition must be made for the bossing resistance, which if the bossings are carefully designed, preferably after stream-flow tests, should vary between 3 per cent. for full ships and 5 per cent. for fine ships, where the bossing is necessarily larger and longer.

If the design is to be for service conditions, it is the practice at Teddington to add a further 15 per cent. for average fine weather at sea. For certain ships and services this is probably insufficient—for example, winter service in the North Atlantic and small coasters in the steep, short seas of the North Sea—and the allowance must be varied in accordance with experience on each route.

The quasi-propulsive coefficient depends on many factors, and it is impossible to give more than an extremely approximate guide to its value for any given case. The primary feature affecting the propulsive efficiency is the revolutions per minute, and in Fig. 16 is shown a plot of quasi-propulsive efficiency against revolutions per minute for a 400ft. ship. There are two curves, one for single

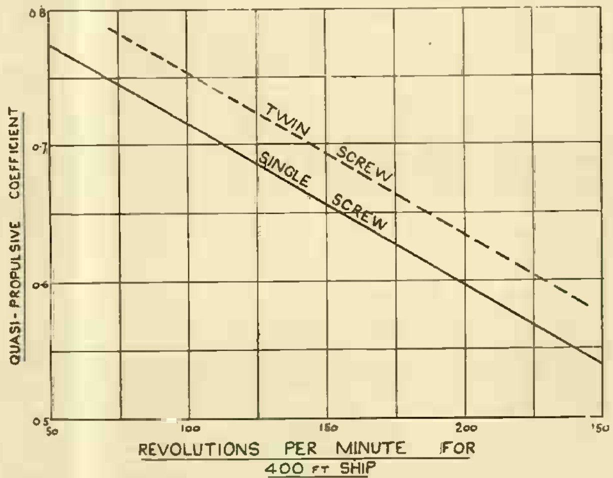


FIG. 16.—*Average Values of Quasi-Propulsive Coefficient for Single and Twin Screw Ships.*

screw and the other for twin screw vessels. In both cases the value falls with increasing revolutions, due primarily to a reduction in the open screw efficiency. These curves are the result of a plot of the results for some 100 models, and will apply only to twin screw vessels with well designed bossings and outward turning screws and smooth rudders, whilst for single screw ships they will only be reasonably correct for vessels with a double plate rudder and shaped sternpost and with good clearances in the aperture and a modern screw. For the effect of other stern arrangements on efficiency reference should be made to the large amount of published data in the transactions of the different institutions.

Knowing the power required at the propeller, the brake horse power at the engine can be found by using a reasonable figure for the losses in the stern tube and shaft bearings—some interesting figures for engine mechanical efficiencies and shaft friction losses have recently been given by Sir Amos Ayre[30]. The revolutions per minute N for the desired power can then be found from the engineer's rating of the engine.

To find the speed of advance $V_1$ from the ship speed V knots, it is necessary to estimate the wake fraction $\omega$. Many formulæ and charts have been published giving such information, and an excellent summary of these is given in Chapter III of reference (2).

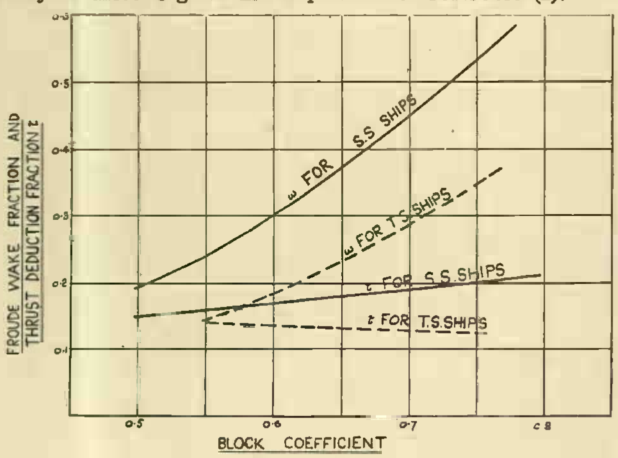


FIG. 17.—*Average Values of Wake Fraction $\omega$ and Thrust Deduction Fraction t.*

In Fig. 17 are shown the wake fractions for the same models as used in the plot of quasi-propulsive coefficients, and they are therefore subject to the same conditions as those coefficients regarding stern arrangements. The scatter of the spots is quite large and is to be attributed to the many other variables which occur besides the primary one of block coefficient which has been used. A better plot would probably be obtained by using some coefficient defining the fullness of the after body only, but block has been adopted as being simple and known even in the earliest design stages.

Of the other features affecting wake, one is the position of the screw in the aperture in single screw ships, as exemplified by the ratio of the height of shaft above base to the draft. The greater this is, the larger the wake, because the tips of the screw at the top of the disc are nearer the water surface where there is a heavy concentration of slow moving wake water. On the other hand, reduction of diameter beyond a certain point also increases wake, because a greater proportion of the disc is being concentrated in the region of high wake on each side of the centre line. The wake fraction in single screw ships is also very sensitive to the type of rudder and sternpost. The values in Fig. 17 are for vessels with fairly slim double-plate rudders and fins on the fore-side of the sternpost, and a clearance between nose of fin and trailing edge of propeller blades of about 12in. on a 400ft. ship. For less clearance or for fatter rudders $\omega$ will be higher, and the same will apply if the sternpost is rectangular in section without a fin.

In twin screw vessels the wake is less than on singles for the same block coefficient, and will also be less the greater the tip clearance from the hull. In the majority of cases plotted in Fig. 17 the clearance is between 24in. and 30in. for a 400ft. ship. The angle of the bossings to the horizontal also affects the wake factor, and this is usually determined by stream flow tests to be between 25° and 30° for outward turning screws—for inward turning the bossing should be nearer the vertical.

Having in this way estimated the values of S, N and $V_1$, the value of $B_p$ can be worked out. The use of a chart such as Fig. 14 will then enable a value of $\delta$ and of pitch ratio to be chosen to give the maximum open efficiency. Since

$$\delta = ND/V_1$$

the optimum value of D is thus found, and the corresponding open efficiency. Of course it is not always possible to use this diameter because of other restrictions, chief among which are draft and the necessity of giving the screw tips adequate cover when the ship is not fully loaded and in a seaway. If the optimum value of D cannot be used, a new $\delta$ can be found from the known values of N and $V_1$ and the maximum permissible value of the diameter D. This in association with the value of $B_p$ already found will give a new screw of different pitch ratio and open efficiency, and one can see from these figures the loss in efficiency caused by the necessary use of a smaller diameter.

The probable quasi-propulsive coefficient can now be estimated from equation (21), i.e.,

$$\text{Q.P.C.} = (1+\omega)(1-t) \times \eta_0 \times \text{R.R.E.}$$

$\eta_0$ has just been found from the design chart, $\omega$ has already been estimated, and the values of $t$ for the same models are also shown on Fig. 17. The relative rotative efficiency, if no other information is available, may be taken as 0·98 for twin screw ships and 1·05 for singles.

If this deduced value of the Q.P.C. agrees fairly closely with the one assumed to obtain the estimate of the power S, well and good—if it does not, the calculation must be done again using the new value of Q.P.C. until such agreement is obtained.

The area to be given to the propeller blades depends upon the thrust which must be developed, and if there is any danger of cavitation this must be carefully looked into either by one of the semi-empirical methods mentioned above, or by carrying out a complete strip calculation and determining the maximum reductions of pressure on the backs of the sections. Such a calculation will also give information as to the efficiency with which the blades are working at different radii and so may lead to the adoption of some other pitch variation over the propeller blades. In this connection, it may be pointed out that a given section of the blade in making a complete rotation meets with a very variable wake distribution and therefore will work over a range of angles of incidence, and as the pitch must be a fixed one it can only be chosen to suit the mean wake averaged circumferentially at that radius. The section will therefore not always be working at its maximum efficiency but if we have reliable information as to the distribution of mean wakes at different radii, we can ensure that the difference will never be large. There is not a great deal of published data on such wake distribution, but charts for single and twin screw ships will be found in reference 29. If, as a result of such work, there is found to be some danger of cavitation, steps must be taken to obviate it or at least reduce it as far as possible, by increasing area, thereby reducing the loading and making the sections relatively thinner, by special attention to pitch distribution and to the shape of the blade sections.

It will be seen from equation 21 that for a given open efficiency, a high propulsive coefficient will be associated with a high hull efficiency. Any feature of design which causes a high value of $\omega$ and a low value of $t$ will thus be beneficial in obtaining a good propulsive efficiency, but there are other considerations which considerably modify this view.

A full after body will increase $\omega$, but $t$ will also be larger, and therefore the gain is not as great as at first it might appear, and of course the effect on hull resistance has also to be considered. A fine after body will reduce $\omega$ and $t$, the former effect usually being predominant. In a slow speed ship such a change will also generally reduce resistance, but due to the concurrent reduction in hull efficiency this is not all reflected in the final power required. In fact in one case in the author's experience a ship was built with a finer and longer run than an otherwise sister ship, and although the E.H.P. was reduced the reduction in wake resulted in a quasi-propulsive coefficient so much lower that the horse power required at the propeller was actually higher in the second case. The choice of hull form thus affects both factors, and a nice discrimination is sometimes necessary to decide how far such changes can be carried.

To obtain high wake values also means coming nearer to the surface, with a consequent danger of loss of efficiency in rough weather or in ballast, or nearer to the hull, with a danger of vibration. Also, in obtaining a high average wake by such methods, the variation in wake over the disc is made larger, with a danger of lowering the relative rotative efficiency and, maybe, an added risk of cavitation or of singing. The search for high values of wake factor must therefore not be pushed too far, and experience of many model and ship results is necessary in order to make a suitable choice of all the factors involved.

**19. Conclusion.**

It is impossible within the limits of a single paper to do justice to such a wide and intricate subject as ship propulsion.

My object has been rather to review the field from as broad a viewpoint as possible in the hope of interesting the young members and those others who have had too little contact with recent developments.

The problem of ship propulsion is a fascinating one, and I am sure those who once begin to study it will find great profit and pleasure therefrom, and I hope the references I have given will help to introduce them to the vast literature which is in existence in the transactions and libraries of the different institutions.

There are still many new ideas coming along in the realm of propeller design and theory. In the past, practice and experiment have always kept ahead of theory, and this is still to a large extent true to-day, but modern theories have made great advances and are now invaluable both for an understanding of experimental results and as a guide to future research. In both fields there are still many unexplored regions and no marine engineer who takes up the serious study of propellers and their ways need fear a lack of work or of interest.

**20. Acknowledgment.**

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**Appendix A.**

**Momentum Theory of Propeller Action.**

Consider a propeller disc of area $A$ sq. ft. in a uniform flow of velocity $v_1$ ft./min. Well ahead of the screw all the fluid will have this velocity $v_1$ and the pressure will be $p$ lb. per sq. foot.

Well behind the screw, the race column will have some larger sternward velocity, which we may write as

$$v_1(1+b)$$

but if we neglect any effect of rotation in the race, the pressure will still be $p$, equal to that in the surrounding fluid. (See Fig. 5).

Since the acceleration and contraction of the race cannot occur instantaneously at the disc, let the velocity through it be

$$v_1(1+a)$$

where $a$ is the *axial inflow factor*.

Then the quantity of water passing through the disc in unit time, $Q$ cubic feet say, will be

$$Q=v_1(1+a)A$$

Neglecting the effect of rotation, the change of momentum in unit time is

$$\rho Q[v_1(1+b)-v_1]$$

where $\rho$ = mass density = wt. per unit volume/g

$$\text{or } \rho Q v_1 b$$

and this must be equal to the thrust $T$ of the disc (since force = rate of change of momentum).

Hence

$$T=\rho Q v_1 b \quad \text{(A.1)}$$

The total work done per unit time (or the power expended) is equal to the increase in kinetic energy of the fluid (since we are neglecting friction) and if there is no rotation of the race, the increase in kinetic energy in unit time is given by

$$\begin{aligned} &\tfrac{1}{2}\rho Q[v_1^2(1+b)^2-v_1^2] \\ &=\tfrac{1}{2}\rho Q[v_1^2b^2+2bv_1^2] \\ &=\rho Q v_1^2 b[1+b/2] \\ &=Tv_1(1+b/2) \quad \text{(A.2)} \end{aligned}$$

This increase in kinetic energy has been provided by the work done *on the water* by the thrust, which is

$$Tv_1(1+a) \text{ in unit time.}$$

Hence we shall have

$$Tv_1(1+a)=Tv_1(1+b/2)$$

$$\text{or } a=b/2 \quad \text{(A.3)}$$

That is, *one half of the sternward increase in velocity is acquired by the fluid before it reaches the propeller.*

The useful work obtained from the screw, i.e., *done upon the ship*, is $Tv_1$ and therefore the power lost in the screw is

$$Tv_1(1+a)-Tv_1$$

or $Tv_1a$, which by equation A.3 is also equal to

$$Tv_1b/2$$

The screw efficiency will be

$$\begin{aligned} \eta &= \text{useful work obtained/work expended} \\ &= Tv_1/Tv_1(1+a)=1/(1+a) \quad \text{(A.4)} \end{aligned}$$

If there were no "slip" in such an ideal propeller disc, the speed of advance would be $v_1(1+b)$. Actually the speed of advance is $v_1$.

Hence, by our earlier definition of slip, we can write

$$\text{Slip ratio}=s=\frac{v_1(1+b)-v_1}{v_1(1+b)}$$

$$=b/(1+b)$$

$$\text{Hence } b=s/(1-s)$$

$$\text{and } a=b/2=s/2(1-s)$$

Therefore, from equation A.4,

$$\eta=\frac{1}{1+a}=\frac{1}{1+\dfrac{s}{2(1-s)}}=\frac{2(1-s)}{2-s}$$

$$\text{or } \eta=(1-s)/(1-s/2) \quad \text{(A.5)}$$

If in addition to being given an acceleration in the line of the race axis, the fluid is also given rotation in passing through the disc, the kinetic energy in the race will be increased by the energy of rotation in the fluid. To develop the same thrust, therefore, more energy will have to be imparted to the water, and the effect of rotation in the race will thus be to reduce the ideal efficiency below that given by A.5.

Suppose the disc to be rotating with angular velocity $\omega$, and that in the race, well behind the screw, the water is rotating as a whole with angular velocity

$$\omega^1=b^1\omega$$

In an actual fluid, the rotation $\omega^1$ cannot be acquired instantaneously, and in fact some rotation of the fluid is already present at the disc, in the same way that some of the fore and aft acceleration has also been acquired at that point (Fig. 6).

In a similar way, we can call the rotational speed of the water at the disc $a^1\omega$, and it should be noted that in this case it is in the same direction as the movement of the disc. Hence the rotational velocity of the disc relative to the water will be

$$\omega-a^1\omega$$

$$\text{or } \omega(1-a^1) \quad \text{(A.6)}$$

and $a^1$ is the *rotational inflow factor*.

**Appendix B.**

**Blade Element Theory of Propeller Action.**

In the momentum theory the propeller is considered as a mechanism for increasing the momentum of the race, and no attempt is made to explain how this is done. In the blade element theory the problem is approached by considering the forces acting on one small strip of blade as it rotates and advances, the elementary contributions of thrust and torque evaluated, and then these are integrated over the whole blade from root to tip.

Consider a strip of blade $dr$ wide at radius $r$, advancing with speed $v_1$ and with rotational speed N revolutions per minute (Fig. 11).

Let $a$ and $a^1$ be the axial and rotational inflow velocity factors as already described in Appendix A. The velocity of the water relative to the blade element will then be given by a vector $v_r$ in Fig. 11, and we see that

$$\tan\phi=\frac{v_1(1+a)}{2\pi Nr(1-a^1)}=\frac{1+a}{1-a^1}\cdot\tan\phi_0$$

$$\text{and } v_r=V_1(1+a)/\sin\phi \quad \text{B.2}$$

The angle of incidence, $\alpha_1$, is given by $(\theta-\phi)$, where $\theta$ is the pitch angle. We thus see that the induced velocities $av_1$ and $-a^12\pi Nr$ *both decrease the slip angle* or angle of incidence and it must be remembered that this angle is always small in a screw propeller, usually not greater than 3° or 4°.

Suppose the propeller has $B$ blades, the chord at radius $r$ is $C$, and that the section of the blade there has lift and drag coefficients $C_L$ and $C_D$ at an angle of incidence $\alpha_1$.

Let the lift and drag of an element of the propeller blade of length $\delta r$ along the blade be $\delta L$ and $\delta D$ respectively.

Then for a propeller with $B$ blades we shall have

$$\delta L = \tfrac{1}{2}\rho \times \text{area} \times (\text{velocity})^2 \times C_L$$
$$= \tfrac{1}{2}\rho C dr B \times v_1^2(1+a)^2 C_L/\sin^2\phi$$

and

$$\delta D = \tfrac{1}{2}\rho C dr B \times v_1^2(1+a)^2 C_D/\sin^2\phi$$

Now $\delta L$ and $\delta D$ are respectively normal to and along the direction of the relative velocity $v_r$.

Hence the thrust and torque contributed by these elements will be (see Fig. 11)

$$\delta T = \delta L \cos\phi - \delta D \sin\phi$$
$$\text{and } \delta Q = (\delta L \sin\phi + \delta D \cos\phi) r$$
$$\text{Now } \delta T = \delta L\left(\cos\phi - \frac{\delta D}{\delta L}.\sin\phi\right)$$
$$= \delta L\left(\cos\phi - \frac{C_D}{C_L}.\sin\phi\right)$$
$$= \delta L(\cos\phi - \tan\gamma\sin\phi) \text{ where } \tan\gamma = C_D/C_L \text{ (See Fig. 8).}$$
$$= \delta L(\cos\phi\cos\gamma - \sin\phi\sin\gamma)/\cos\gamma$$
$$= \delta L[\cos(\phi+\gamma)/\cos\gamma]$$

Then

$$\delta T = \tfrac{1}{2}\rho C dr B v_1^2(1+a)^2 C_L \times \cos(\phi+\gamma)/\sin^2\phi\cos\gamma \quad \ldots \text{ B.3}$$

and

$$\frac{dT}{dr} = \tfrac{1}{2}.\rho.C.B v_1^2(1+a)^2 C_L . \frac{\cos(\phi+\gamma)}{\sin^2\phi.\cos\gamma} \quad \ldots\ldots\ldots\ldots \text{ B.4}$$

Similarly,

$$dQ/dr = \tfrac{1}{2}\rho C B r v_1^2(1+a)^2 C_L \times \sin(\phi+\gamma)/\sin^2\phi\cos\gamma \ \ldots\ldots \text{ B.5}$$

$dT/dr$ and $dQ/dr$ may now be plotted on a base of radius $r$ and the total thrust and torque obtained by integration. Such curves show that most of the thrust and torque are developed over the outer part of the blade, the maxima occurring at about 0·7 R (Fig. 12).

The efficiency of a blade element is given by

$$\eta = v_1\delta T/2\pi N\delta Q$$
$$= \frac{v_1 \times \delta L \times \dfrac{\cos(\phi+\gamma)}{\cos\gamma}}{2\pi N r \delta L \ \dfrac{\sin(\phi+\gamma)}{\cos\gamma}}$$
$$= \frac{v_1}{2\pi N r} . \frac{1}{\tan(\phi+\gamma)}$$
$$= \tan\phi_0/\tan(\phi+\gamma)$$
$$\text{i.e. } \eta = \frac{1-a^1}{1+a} . \frac{\tan\phi}{\tan(\phi+\gamma)}$$

Having obtained curves of $dT/dr$ and $dQ/dr$ and integrated them to find the total thrust and torque T and Q, the efficiency of the whole propeller will be

$$Tv_1/2\pi NQ$$

The performance of each blade element can only be determined when values of a, $a^1$, $C_L$ and $\gamma$ are known.

$C_L$ and $\gamma$ can be found from tests on sections of the same shapes as the different blade elements, at the angle of incidence $\alpha_1$ at which the section is working.

To find $a$ and $a^1$ it is necessary to equate the thrust to the momentum put into the race, and the torque to the change in rotational momentum, in much the same way as for the momentum theory.

In practice, $a$ and $a^1$ can be found on the assumption that one half of the final increase in velocity, both axial and rotational is gained before the screw disc is reached, their actual values being found by graphical plotting.

Of recent years corrections have been introduced to take account of two other features of propeller working—the losses at the tips due to the tip vortices, and the loss due to interference between one blade and the next. This is most serious near the boss, and is usually referred to as "cascade effect". These are refinements of the theory, and it is not proposed to discuss them here. Those who wish to pursue the matter will find the necessary references in the Bibliography, the tip correction being given in the work of Goldstein[10] and Lock[11], and the cascade effect in that of Gutsche[12]. Recently Burrill[13] has given a very complete example of the application of these theories to marine propellers.

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    W. P. A. Van Lammeren, L. Troost and J. G. Koning. (published by H. Stam, Amsterdam, 1942).
30. An Approximate and Simple Formula concerning 4 bladed Propellers of Single Screw Cargo Ships.
    Sir Amos L. Ayre, K.B.E. Trans. N.E. Coast Inst., 1945.


## Discussion.

**Mr. W. A. R. Atkinson** (Member), opening the discussion, said he would like to know something about the closing-in apertures of older ships, and whether it was worth while.

At one time it was thought that the closing-in of the aperture (by streamlining) at the forward side of the propeller would improve the performance of some of the older vessels. Could Dr. Todd tell whether this practice was worth while?

**Major-General A. E. Davidson, C.B., D.S.O.** (Member), said he had asked two or three authorities what was the actual percentage of propeller slip in a ship, but he had never had a clear-cut answer. Perhaps the reason was that there was not one. He gathered from one diagram that it was never less than 20 per cent., but other people talked about 7 per cent. and 12 per cent. The slip, of course, might be variable under different conditions, but it would be interesting to have expert information on this question.

**Dr. E. V. Telfer** said that Dr. Todd had delivered a masterly exposition of the subject in record time, aided no doubt by the fact that most of the matters dealt with were of a non-controversial character.

He would have liked to have seen a more detailed exposition of different designs of propellers. Such an addition would have been greatly appreciated by seagoing members who would then have been able at sight to identify such screws as the Wawn (with or without slip-correctors), Wilcox, Freidanthol, Unislip, Scimitar, Heliston, to mention only a few. The advantages of bronze blades over cast-iron, the price to be paid in extra power for rough and broken blades over smooth blades would also have been useful information. Built blades versus solid propellers involving the issue of big bosses versus small (a physical and not political issue!) were also of great interest.

There was one further point with which he might deal. On Fig. 2 the author defined skew-back in a way that appeared new to him. The method might of course be that standardized at Teddington, but he did not think that it was so well defined. Skew-back, in a propeller with an even number of blades, measured the tip separation of opposite blades, say, passing through the aperture. This separation was quite independent of root symmetry or position on boss. It thus would be better to define skew-back without reference to the root. The definition given in Dr. Baker's book did not appear to be the same as given by the author.

**Mr. R. K. Craig** (Member) wished to know whether he was correct in assuming from Fig. 16 that the propulsive efficiency of twin screws was better than that of a single screw.

**Mr. A. W. Jones, B.Sc.** (Associate Member), said an interesting point from one's watchkeeping experience was that often the engineer who worked out the apparent slip for the day, from the ship miles and propeller miles, found very great variations, one day giving no apparent slip and other days of similar weather conditions and r.p.m. giving slips up to 15 per cent. He would like to know whether such variation was just due to a straightforward error in the figures used, or whether it could possibly occur under such conditions (the effect of currents being excepted).

**Mr. F. D. Clark** (Associate Member) said that when at sea with the ship in a favourable current it was not unusual when calculating the average slip at the end of the watch to obtain a negative value. It was appreciated that true slip could not be negative, but could the author suggest how such a value should be expressed in the engine-room log book?

Would it be true to say that the present method of calculating the slip at sea was so approximate that it could be misleading when one was endeavouring to judge the efficiency of the propulsion unit?

**Mr. N. H. Denholm** (Visitor) said that, although he was an electrical engineer and therefore not qualified directly to assess the technical content of Dr. Todd's paper, he had attended many meetings of kindred societies where the subject of ship propulsion had been discussed. He had been able to enjoy the clearest picture of the phenomena under discussion from the author's adherence to "The Fundamentals of Ship Propulsion".

The many figures and curves given by the author were analogous to the characteristics of electrical machinery, and the concept of "slip" was commonly met with in the alternating current equipment used to provide the power to drive such screw propellers.

He hoped the author would include in the published paper the illustration which they had seen on the screen showing the distribution of pressure over the longitudinal section of the ship. It was almost precisely the same in contour as that displayed when the character of the vertical component of a ship's magnetic field was measured at the sea bed.

He thought that the paper would be of benefit not only to himself as a fruitful source of reference, but also to the young students who were primarily instructed in the fundamentals on which Dr. Todd's whole argument was based.

**Mr. T. W. G. Knowles** (Member) said he would like to mention a phenomenon which had come to his notice some years ago, and which he still did not understand. It was in connection with a propeller which was tested behind a model hull in the Teddington Tank. In the original test it was found that the propeller tended to run away periodically and then pick up again. He said he believed that some improvement was effected by moving the model propeller further back, and in the actual vessel the stern frame was so altered that the propeller was about six inches further back than originally intended. Although this movement no doubt largely overcame the difficulty, there were variations in the power output of the Diesel engine, which was fitted with a speed governor which maintained constant revolutions per minute. He said he could not understand why the modification to propeller position should have got over the trouble, and he would be interested to have the author's views on the matter.

He continued by saying that he had been concerned with the design of propellers for vessels in Malaya, where the ratios of beam to draught were very large, and up to the time the Japanese occupied the country, he had been unable to obtain data on the wake which one might expect with such very large beam draught ratios. He wished to know whether there had been experiments carried out with models resulting in such data being available, which would be invaluable for his purpose.

On the proposal of **Mr. A. M. Riddell**, seconded by **Dr. E. V. Telfer**, a vote of thanks to the author was carried with acclamation.

## BY CORRESPONDENCE.

**Mr. A. Emerson, M.Sc.:** Attention is drawn to the different meanings of the single- and twin-screw propulsive coefficient lines in Fig. 16. The line for single-screw ships is obtained from an empirical formula given by the writer in 1942 (Transactions Institution of Naval Architects, Vol. 85, page 182) for single-screw cargo ships. The limitations of this formula ($Q.P.C.=0{\cdot}83-N\sqrt{L}/18{,}000$ where $N$=r.p.m. under trial conditions and $L$=ship length in feet) are discussed in a paper which it is hoped will be published later this year, but in general it may be taken to give the propulsive coefficient which should be obtained with a correctly-designed single-screw ship.

It is not possible to give so simple a law for a twin-screw ship, but in this case the primary variable is open screw efficiency; if it is considered desirable to plot on a base of revolutions then it is necessary to give a series of lines corresponding to different $D/V_1$ ratios.

Presumably the line given in Fig. 16 represents approximately one such line, *i.e.* the single-screw line is a general result, the twin-screw line a particular result.

**Mr. L. Troost** (Ned. Scheepsbouwkundig Proefstation, Wageningen): This excellent paper calls for some comment with regard to the fundamental pressure diagram in Fig. 5.

Although the calculation of the pressure decrease ahead of the screw and the increase behind, has been left out of consideration, it might be of interest to state that your diagram as well as most diagrams of the sort in handbooks is somewhat misleading, in so far that it suggests an equality of the decrease and increase referred to, which is not the case as soon as a change of momentum or thrust is assumed. As a matter of fact, the decrease of pressure ahead of the screw disc is smaller than the increase behind, the ratio of these differences decreasing with increase of thrust and ultimately approaching 1/3 for the screw with zero speed of advance. Only in the case of vanishing thrust are the pressure differences equal. This variation of pressure difference is interesting from the point of view of variation of the thrust-deduction factor with thrust loading.

If we apply Bernoulli's Law both in front and aft of the screw disc, assuming $p_1$ to be the pressure just ahead of, and $p_2$ just behind the disc, $(p_2-p_1)$ being the increase of pressure at the screw disc, we get:

$$p_2-p=\frac{\rho}{2}\left[\{V_1(1+2a)\}^2-\{V_1(1+a)\}^2\right] \text{ and}$$

$$p-p_1=\frac{\rho}{2}\left[\{V_1(1+a)\}^2-V_1^2\right]$$

$$\frac{p-p_1}{p_2-p}=\frac{2+a}{2+3a} \qquad \text{(1)}$$

If we adopt a thrust constant $C_T=\frac{T}{\frac{1}{2}\rho V_1^2 A}$. we find:—

$$C_T=\frac{\rho.Q.V_1.2a}{\frac{1}{2}\rho V_1^2 A}=\frac{\rho V_1^2(1+a)2a.A}{\frac{1}{2}\rho V_1^2.A}=4a(1+a) \qquad \text{(2)}$$

From (1) and (2) we derive:—

$$\frac{p-p_1}{p_2-p}=\frac{3+\sqrt{C_T+1}}{1+3\sqrt{C_T+1}} \qquad \text{(3)}$$

As $\gamma=1/(1+a)$, we can also write:—

$$\frac{p-p_1}{p_2-p}=\frac{1+\gamma}{3-\gamma} \qquad \text{(4)}$$

From (3) we see that the ratio$=1$ when $C_T=O$, and from (4) that it is 1/3 when $\gamma=O$, which is the case with zero speed of advance.

## The Author's Reply to the Discussion.

**Mr. Atkinson:** The results of the first experiments made at Teddington on the effect of closing in the aperture in single-screw ships were published in 1928*. These showed that a fin plate fitted in the aperture in front of the screw was disadvantageous to propulsive efficiency, the loss being some 3 to 5 per cent. At the same time, the body post in front of the propeller should be shaped to give a taper finish to the waterlines, preferably not exceeding 15 degrees to 18 degrees to the centre-line plane, and not be of the old square section once so common. This had the double effect of avoiding a certain amount of eddy-making behind the post and of reducing the risk of vibration when the propeller blades passed behind the stern post and through the concentrated wake belt which occurred there. Streamline fins on the fore side of the rudder post, and a smooth surface to the rudder, with all openings shielded, were wholly beneficial to propulsion, and might result in an increase in propulsive coefficient of from 5 to 8 per cent. (see reference 25) due essentially to the elimination of the head resistance of the square post. One point which must be remembered in connection with such fins added to an existing post was the necessity for strong rigid construction to avoid vibration.

**Mr. F. D. Clark:** The replies to other contributors to the discussion would answer most of Mr. Clark's queries. There would appear to be no alternative but to enter negative slip values in the log book so long as apparent slip was tabulated there. It was impracticable to replace this by true slip ratios, since this necessitated an accurate knowledge of the ship wake at all speeds, in different sea conditions and for different states of the hull surface, which information was not at present available.

**Mr. R. K. Craig:** It did not follow from Fig. 16 that the overall propulsive efficiency of twin screws was better than that of a single screw for any particular design. The two lines shown there did not represent optimum results from calculations, but were average lines drawn through some 100 spots taken from the results of experiments made at Teddington over the last three or four years, in the course of the ordinary work going through the Tank for different firms.

They thus represented an average of present-day trends in marine engineering design, and in general it would be agreed that for a given total power, twin engines would normally be designed to run faster than a single. This would reduce the apparent advantage of twins shown in Fig. 16. Again, in large cargo-liners it was not always possible to choose the optimum screw diameter for a single-screw installation, because of the limitation in diameter due to draft, whereas for a twin design the best diameter could probably be used, and a good result obtained. This factor would also be reflected in Fig. 16, since in most of the cases on which this diagram was based the twin-screw arrangement would only have been adopted because of some limiting factors unfavourable to the single-screw arrangement.

The quasi-propulsive coefficient was also not the sole criterion of the final power—in the twin-screw ship the appendage resistance would be larger than in the single due to the presence of the bossings, as pointed out in Section 18 of the paper. This would amount to some 5 per cent. in the case of well-designed bossings, preferably drawn out from the results of streamflow tests in the Tank, and might be considerably greater if an ill-chosen bossing shape or position was used. This factor alone would balance most of the difference in quasi-propulsive coefficient.

The question of the choice of single or twin screws was thus seen to be extremely complicated, and the decision could only be made on the merits of individual cases, and would involve questions of engine design, first cost of the two alternative installations, engine-room space, and so on, in addition to the hydrodynamic aspects. Mr. Craig would find some discussion of this subject in a paper by Dr. G. S. Baker on "Some Considerations in the Design of High Speed Cargo Vessels" read in 1942 before the North-East Coast Institution of Engineers and Shipbuilders.

**Major-General A. E. Davidson, C.B., D.S.O.:** The confusion which always seemed to arise over the term "slip" was due to the fact that people did not distinguish clearly between "apparent" slip and "real" slip. The propeller, as explained in the paper, did not advance through the water with the same speed $v$ as the ship, but at some reduced speed $v_1$. This was because of the wake behind the hull, in which water was being dragged forward relatively to the sea-bed, and thus virtually reduced the speed of feed of water to the screw. The speed of advance of a propeller of pitch $P$ feet turning at $N$ revolutions per minute would, if the medium were solid, be $P.N.$ feet per minute, and the real slip of the screw if it in fact advanced through the water at only $v_1$ feet per minute, was

$$P.N.-v_1$$

or the *real* slip ratio $= \dfrac{P.N.-v_1}{P.N.}$

The maximum propeller efficiency would occur at a *real* slip ratio of 12 to 20 per cent., depending upon the pitch ratio, and in general the screw should work at somewhat higher average slips than those giving maximum efficiency in order that it might always work on that part of the efficiency curve where the change in efficiency with slip was smaller—*i.e.* to the right of the peak value in Fig. 13. *Real* slip was the only true criterion of the propeller performance, and for all ships would be found to lie in the region between 15 and 40 per cent.

When small slips of the order of 5 per cent., or negative slips, were quoted, it might be safely assumed that they were *apparent* slips, based upon the ship speed and not upon the speed of advance of the propeller through the wake water. The *apparent slip ratio* was given by

$$\frac{P.N.-v}{P.N.}$$

where $v$ was the ship speed in feet per minute, and as in all mercantile vessels $v$ was greater than $v_1$, it followed that the apparent slip ratio was always less than the real slip ratio. For example, in a vessel travelling at 15 knots, propelled by a screw of 14·5 feet pitch at 110 revolutions per minute, the *apparent slip ratio* was

$$S_a = \frac{14{\cdot}5\times110-15\times101{\cdot}33}{14{\cdot}5\times110} = 0{\cdot}046 \text{ or } 4{\cdot}6\%.$$

But the wake fraction in this case was 0·44, so that the actual speed of advance of the propeller through the *water* was only

$$\frac{15\times101{\cdot}33}{1{\cdot}44} \text{ or } 1{,}056 \text{ feet per minute.}$$

Hence the *real slip ratio* was

$$S = \frac{14{\cdot}5\times110-1{,}056}{14{\cdot}5\times110} = 0{\cdot}34 \text{ or } 34\%.$$

**Mr. N. M. Denholm:** The illustration showing the distribution of pressure over the longitudinal section of the ship to which Mr. Denholm referred was given in the first part of this paper to the Institute in 1944 (Ref. 1, Fig. 7) and further examples would be found in most text books on the subject and in the papers published by Eggert in the U.S.A.

**Mr. A. Emerson:** The fact that propulsive efficiency depended to a large extent upon propeller revolutions was not new. The author had frequently drawn attention to this in his published papers on Coasters previous to 1942—"In the propeller experiments the propulsive efficiency was investigated for values of $^N/_v$ from 9·5 to 30, ................ The efficiency was found to decrease with increase in revolutions ..............."*; "The effect of increased propeller revolutions is to reduce the propulsive efficiency" ................ "and that linear interpolation of the propulsive coefficient for intermediate values of $^N/_v$ can safely be employed"†; "................ the propulsive efficiency varied from 0·66 ................ for $^N/_v$ values of 18·6 ................ while ................ it fell to 0·57 for the high $^N/_v$ value of 32·2. This represents a loss of propulsive efficiency from the highest value of some 16 per cent., which is due primarily to the higher propeller revolutions"‡.

The line for single-screw ships given in Fig. 16 of the present paper was, as stated therein, derived from a plot covering the results of some 70 models (the remainder of the 100 being twin-screw designs) mostly run at Teddington in the last three years, and chosen so as to give as wide a range of revolutions as possible, and with due regard to the type of rudder and stern frame, as clearly stated in Section 18. Mr. Emerson's statement that it was obtained from his formula was not correct, which was indeed obvious, since the latter did not give the same line.

It might be as well to repeat here the very clear warning given

*"Experiments on the Propulsion of a Single Screw Ship Model", by G. S. Baker and J. L. Kent. Trans. I.N.A. 1928.

*Further Resistance and Propeller Experiments with Models of Coasters. I.N.A. 1938.

†Experiments with Models of Cargo-carrying Type Coasters. Institute of Marine Engineers. 1940.

‡Further Experiments with Models of Cargo-carrying Coasters. N.-E. Coast Inst. 1942.

in Section 18 that such a diagram could only be used as an "extremely approximate guide" to the value of the propulsive efficiency, and it was only intended for use in the early design stages as representing the probable value in a modern, well-designed ship. Later, many other features would have to be taken into account and a more definite value derived from accumulated experience.

**Mr. A. W. Jones, B.Sc.:** The apparent slip, as had been seen above, depended on the three factors, propeller pitch, revolutions per minute and ship speed. Under identical conditions of weather and sea, with the same condition of hull surface, loading of the ship, and trim, the same revolutions should give the same ship speed and therefore the same apparent slip.

If the basic data were correct, then any change in slip must be due to a change in weather, to different loading or to fouling of the hull. It had been found that the performance was susceptible to quite small changes in weather, which might not be noticed as between one watch and another in the engine room. But it was far more probable that the differences were due to errors in one item of the basic data—the ship speed. This was affected by ocean currents and other errors of observation which it was very difficult to eliminate, whereas propeller revolutions were easily measured to any order of accuracy desirable.

**Mr. T. W. C. Knowles:** It was very difficult to answer Mr. Knowles' questions without knowing more details of the ship concerned.

The symptoms he described would suggest that there was instability of flow behind the model, and that this gave rise to periodic breakdown of thrust in the propeller. Such instability might be due to the periodic shedding of eddies from the stern, and this would be more likely the fuller the form aft—it would be interesting to know whether the form was, in fact, a very full one. This suggested explanation was rather borne out by Mr. Knowles' statement that moving the propeller aft improved matters, since this would increase the clearance between the body post and the blades, and remove them from the region of most intense eddies just behind the post, and so reduce the variation in thrust and torque on the screw as the blades passed through the aperture.

The most complete set of data relating to wake values for single-screw ships was that published by Professor E. M. Bragg before the American Society of Naval Architects and Marine Engineers in 1922. This covered a range of beam to draft from 2·0 to 3·0, and indicated that, other things being equal, the increase in beam to draft ratio caused an increase in wake fraction. This was generally known to be the case. For example, in a certain model of about 0·6 block coefficient, in which a wake value of 0·30 might reasonably be expected for a beam to draft ratio of 2·5, the wake measured for the actual ratio of 4·5 was 0·65.

**Dr. E. V. Telfer:** Bronze blades owed their superiority over cast-iron ones to two features—they were thinner for the same design conditions, and the surfaces were smoother and retained this smoothness much longer. Some experiments on the effect of blade thickness had been done by Baker and Riddle*, in which the thickness at the root was halved, all other features of the propeller remaining unaltered. At heavy loadings the efficiency was unaltered by the change in thickness, but at lighter loadings the efficiency was higher with the thin screws, particularly those having circular back sections. Thus at a $B_p$ value of 10, the increase in efficiency was from 0·61 to 0·67 for the circular back type, and from 0·67 to 0·69 for the aerofoil. At a $B_p$ of 25, on the other hand, all screws gave approximately the same efficiency of 0·57. For all screws the reduction in thickness meant an increase in revolutions of some 3 to 4 per cent. when absorbing the same power. Taylor stated in his book "Speed and Power of Ships" that practicable variations in blade thickness would have comparatively little effect upon efficiency.

McEntee had tested model screws with various surface finishes*, including cast steel, iron and bronze, and finished smooth bronze. At working slips all the cast surfaces gave much the same efficiency, which was about 10 per cent. below that for the smoothly-finished screw.

Baker had recently given some estimates of the effect of roughness of blades on propeller efficiency based upon experiments on model propellers in water and on the effect of roughness on sections in the wind tunnel†. He found that the frictional drag of the blade surface was much more important with low pitch than with high pitch ratio screws, and the former especially required a polished surface. Tests were made with a model screw 12in. in diameter in the smooth, finished condition, and with four different types of roughened surface. With an open-spaced roughness consisting of grooves 1/1000in. deep spaced 0·25in. apart, the loss in efficiency was about 3 per cent. at the highest efficiency, disappearing at higher slips. Closer spacing or an increase in depth led to rapid loss in efficiency. With grooves 0·0013in. deep, 0·12 to 0·26in. apart, along two diagonals at right angles on both face and back, the efficiency was reduced from 0·652 to 0·575.

Prandtl had said that the permissible roughness for a surface to be considered "smooth" would be a little less than half a thousandth of an inch on a ship screw, and Baker had estimated that a roughness of two thousandths would reduce the efficiency 4½ per cent. Measurements made by Dr. Berndt gave the roughness of propeller bronze as normally worked (in Germany) as one thousandth of an inch, but this could be reduced to half a thousandth by regrinding and polishing. Unworked cast-iron had a roughness of seven thousandths of an inch.

It would be realized from these figures that serious wastage of power might occur due to the roughness of surface of blades, particularly after some time in service when erosion might have become evident and fouling occurred, and it behoved all superintendents to pay attention to this feature of their ships.

The definition of skewback given in the paper was that normally used in the Tank screw reports.

If a built-up and a solid propeller were tried behind the same hull, the propulsive efficiency was somewhat higher with the solid type. Three twin-screw models showed gains varying between 2 and 3½ per cent., and three single-screw designs gains of from 1 to 4½ per cent. On the other hand, if the hull was designed to suit the larger boss, and the latter could be provided with a long cone to give a reasonable taper to the after end, little or no loss in propulsive coefficient need be incurred, but the hull resistance would probably be a little higher because of the larger bossing required on the hull itself.

**Mr. L. Troost:** The pressure changes in the screw race as shown in Fig. 5 were not meant to be in any way quantitative, but only to show in a general way the changes which occurred in velocity and pressure. The ratio of decrease of pressure in front of the disc to the increase behind varied, of course, with the thrust loading, and they were indebted to Mr. Troost for his very clear illustration of this fact, which formed a valuable addition to the paper.

The author was particularly glad to have this contribution from Mr. Troost because, he believed, it was the first time since 1940 that he had been able to take part in the discussion of a paper before a British Institution. As Superintendent of the Wageningen Experiment Tank in Holland he had always been a most welcome visitor to this country, and they all looked forward to many further contributions by him to the problems of ship resistance and propulsion.

*Screw Propellers of Varying Blade Section in Open Water—Part II. Trans. I.N.A. 1934.

*Notes from Model Basin—American Soc. Naval Architects and Marine Engineers. 1916.

†The Efficiency of Marine Propellers and the Drag Coefficient. N.-E. Coast Inst. 1945.