SCREW PROPELLERS

AND

MARINE PROPULSION.

BY

I. McKIM CHASE, M.E.

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INTRODUCTION.

The author commenced the preparation of this treatise simply with a view to supply a want that he had found existing, during his extensive experience, among workmen who are engaged in building propelling screws. His purpose was to place before them a comprehensive and practical work that should elucidate the principles of screw propellers, the manner of their generation and their peculiarities, and explain the various methods employed in their construction.

The second part, which is devoted to marine propulsion, was not originally intended to be incorporated in this work; but the author has been induced to embody it, being advised that it will prove to be of interest, not only to the practical workman, but to many of those who are advanced in the profession of marine engineering.

The subject may not be treated herein as ably or as exhaustively as its importance demands; nevertheless, the author feels satisfied with the result of his labor, and he hopes and believes that the work will be a desirable acquisition to the literature of marine engineering.

Most of the works on steam engineering that are in present use contain chapters devoted to marine propulsion and propellers; but these portions are, in many instances, confined to the explanation of engravings of devices or to matters of
construction and arrangement, rather than elucidative of the important principles which underlie the main subject.

There have been a few authors who have striven to diffuse a better understanding of this, the most important and, perhaps, the least understood of any of the branches of marine engineering. Prominent among these, shining as the brightest brilliant of the cluster, is the late Professor Rankine, of Edinburgh. To this gentleman the steam-engineering profession is indebted for much of its literature and the mathematical investigations which he pursued and made public.

In the compilation of this book the author has consulted every accessible work of importance which treats on the subject of propulsion, among which are included the writings of the best modern authorities on the subject of propulsion and propellers. In addition, he has diligently and carefully made reference and research among numerous papers contributed to scientific and engineering periodicals by eminent professional men. He has also collated the results of numerous experiments, many of which have come under his own personal observation, and in cases where he has considered that they would enhance the value of the book he has incorporated the description of the work done in that line.

He is indebted to Bourne's work for much of the information herein contained regarding the history of marine propulsion and propellers.

The propelling screw is, in all probability, a very ancient contrivance. The date of its first application as an instrument of propulsion remains in oblivion, but it is said to have been known in China at a very early period. But it is in its connection with steam as a motive power to propel vessels that the following pages are chiefly devoted.
INTRODUCTION.

Following close upon Watts' successful achievements with the steam engine, numerous attempts were made to supply steam power to the propulsion of vessels. Though the results of nearly all these efforts were such as to convince the experimenters of the feasibility of success and to encourage them to persevere in their endeavors, no practical success seems to have been accomplished until 1808, when Fulton, with a small boat, the "Clermont," successfully demonstrated the practicability of steam propulsion applied to steam vessels.

Naturally, the side or paddle-wheel, from its similarity to the wheels which had been used to drive mills during previous ages, the action of which, in water, was so simple and so universally known, was the first form to which the force of steam was successfully applied for the purpose of propelling a vessel.

Fulton's achievements gave an impetus to the promoters of steam navigation. Soon the steam engine was improved, its development rendering it better adapted to marine purposes, and vessels were so devised as to make a better combination with their engines.

Steam navigation soon became an established fact. Ferries were established. The fast-sailing packets for passenger traffic were supplanted by steam vessels, and but a few years sufficed to witness the permanent existence of steam vessels and steam navigation wherever trade was carried on by water.

Experience accumulated, and the enlargement of steam vessels progressed sufficiently to permit of their being sent on ocean voyages. A very short period of ocean steam navigation demonstrated that the paddle-wheel, which had hitherto marked the progress of steam navigation, did not form an advantageous alliance with sails.

Efforts were then directed to perfect an instrument of pro-
pulsion which would make a better combination with sails. These efforts eventually led to the successful application of a submerged screw to the stern of a vessel. This screw had previously been the subject of experiment by some of the more advanced of the promoters of steam navigation.

One of the earliest authentic applications of steam to the submerged screw for marine propulsion is that of Mr. John Stevens, of Hoboken, New Jersey, who as early as 1802 engaged in experiments with the view of devising some means for driving a vessel through the water by applying the motive power to a propeller in the stern; and with a defective boiler he attained for short distances a speed of seven miles per hour. It is somewhat surprising, therefore, that he should have forsaken the path in which he appears to have made such fair progress. In an experimental boat with two screws built by Mr. Stevens in 1804, and now in the National Museum, Washington, D. C., there appears to be the first conception of the twin-screw system now so generally employed. It was not until 1836 that Captain John Ericsson, a native of Sweden, and Mr. F. P. Smith, a native of England, successfully demonstrated the practicability of propelling a vessel by the application of steam power to a submerged screw.

The result of Ericsson's and Smith's experiments stimulated the improvers of steam navigation and infused new vigor into them. Advancement was made with rapid strides, and it far exceeded the most sanguine expectations of the early promoters. The screw rapidly took the lead as the favorite instrument of ocean-steamer propulsion; gradually and steadily it superseded the paddle-wheel on the seas, until now the latter is confined almost exclusively to inland waters. Nor is this result surprising, when it is considered what
superior advantages the screw and the engine possess over those of the paddle-wheel, especially for ocean navigation. These are, a more advantageous combination with sails, a better protection, more compactness, more simplicity, etc.

The screw encountered much prejudice in its early days. Opposition was not confined to the ordinary ranks, but was manifested by some of the leading intellects of the engineering profession. It arose chiefly from a misconception of the principles involved in marine propulsion, which, indeed, have not been satisfactorily mastered up to the present time. The screw, however, has won its pre-eminent position as a marine propeller upon its merits alone, despite the misgivings of the skeptics who predicted its failure. It is now the reverse of unusual to see vessels entirely devoid of sails depending altogether on the screw and the power applied to it for their propulsion.

Ericsson's original screw was constructed of wrought-iron plate. It consisted of a cylinder, upon whose convex surface was secured a number of helical flanges, and this was attached to a hub by several helical blades, thus making a light and strong instrument.

Smith's original screw was constructed of wood, and consisted of two convolutions of a single thread.

Considerable improvement has been made in the manufacture of screws. They are now cast entire of bronze or iron, and of any desired dimensions. A review of the modern methods of construction will be found in the following pages.

To assist the better understanding of the principles involved in screw construction examples are given of the principal forms now in vogue for marine propellers, as well as a description and elucidation of their generation, etc.
INTRODUCTION.

The author has not formulated any rules of his own, but simply gives the result of his experience, which is extensive.

In the appendix will be found extracts from various works, but chiefly from those of Professor Rankine, whose rules relating to the paddle and the screw are as reliable as any that are to be found.

The quotations include Rankine’s memorable calculation of the speed of H. M. S. “Warrior.”

The author has referred only to so much of the accepted theory of marine propulsion as he believes will be interesting to the general reader. An immense amount of labor has been devoted to the demonstration of the various theories of marine propulsion. But in determining the proportions of screw propellers successful designers do not follow any of the theories. They depend on experimental information, and are guided by the knowledge acquired by practical experience.

For those who desire to pursue the study of the various theories advanced to explain marine propulsion, the volumes of Transactions of the Institute of Naval Architects will afford ample material, commencing with the volume for 1865, which contains Rankine’s celebrated paper on the subject.
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THE SCREW PROPELLER.

SCREWS OF VARIOUS CHARACTER.

The screw is correctly defined as a cylinder or a cylindrical perforation having a continuous rib or thread winding spirally around it at an inclination with the axis of the cylinder. Screws may be either regular or irregular. When the thread winds around the convex surface of the cylinder it is termed a male screw; when around the concave surface a female screw.

When the thread does not wind around a cylinder, but terminates in a line at its axis, it is simply a spiral or right helicoid.

The generatrix and pitch constitute the two essential elements in the generation of screws. The former is the line which generates or determines the form of the helicoidal surface, and the latter is the distance which the generatrix advances axially in making one convolution around the cylinder. When the generatrix advances uniformly, and its circular motion is also uniform, a regular or axially true screw is generated. If the generatrix is a right line and perpendicular to its axis or line of advancement, the screw is a regular one radially. If the generatrix has a variable advancement while
its circular motion is uniform, the screw is an irregular one axially. If the generatrix is any other than a right line perpendicular to the axis of the cylinder, the screw is an irregular one radially.

When the generatrix has a uniformly accelerated advancement while its circular motion remains uniform, the result is what is known in engineering as a screw of uniform expanding pitch. The generatrix can be of almost any form and have almost any position relative to the axis.

Fig. 1, Plate I, gives an illustration of the spiral or right helicoid, where the cylinder necessary to constitute a screw is absent, and where the axial termination of the generatrix is a line. This form, from its want of rigidity, is but little employed in the arts. It is sometimes used, however, in a modified form, and generally in a fixed position. The spiral stairway is one of its modifications, and, indeed, is the principal use that can be practically made of it.

Fig. 2 represents the manner in which such a helicoidal figure may be generated. If a triangle be formed of some sheet material, the base line of which is equal to the circumference of the figure and the perpendicular equal to the pitch of the thread, and the triangle be curved in a circular form, the base line completing the circumference of the circle, the figure thus constructed will be the directrix, or guide for the generatrix, and the hypothenuse of the triangle will coincide with the helix of the periphery of the thread. If a line perpendicular to the base and passing through the centre of the circle be erected, it will represent the axis; if another line be erected perpendicular to the axis, it will be the generatrix. Now, if the generatrix be advanced along the axis while constantly perpendicular to it, and the peripheral end of the gen-
SCREWS OF VARIOUS CHARACTER.

eratrix touch the directrix during the progression, a figure similar to Fig. 1 will be described.

Fig. 3 gives an illustration of the screw in its simplest form. It is a single thread wound around a cylinder, and is a true or regular screw both axially and radially, being generated by a right line perpendicular to the axis of the cylinder, the generatrix advancing uniformly along the axis while having a uniform angular motion around it. Fig. 3 is projected as follows: Describe from the same centre a circle equal to the diameter of the cylinder and another equal to the diameter of the screw. Divide the circles by radii into any number of equal parts, as eight, Plate I. Divide the pitch into the same number of equal parts, and project successively the intersections of the radii with the circles intersecting each successive dividing line of the pitch. A line drawn through the latter intersections will delineate the projected helices of the screw, as represented in the engraving.

Fig. 4 represents the manner in which the screw shown in Fig. 3 may be generated. The procedure and arrangements are similar to those shown in Fig. 2, with the addition of the helix at the juncture of the thread with the cylinder. The remarks on Fig. 2 are also applicable to Fig. 4. If the space inclosed by the directrix and the cylinder be filled with a material capable of being scraped down by the generatrix during its progression, a helicoidal surface will be formed corresponding to the thread of the screw as delineated in Fig. 3. This is essentially the principle of the practical construction of propelling screws by the loam or sweeping-up process.

Fig. 5 gives an illustration of a true screw of the description shown in Fig. 2, with an additional thread, making it a double-threaded screw. The initial points of the threads
are in a plane, cutting the screw perpendicular to its axis and at an angle of 180° with each other. The figure is projected in the same manner as in Fig. 3, the projection of the intersections beginning diametrically opposite to each other. If a fraction be cut from this screw by planes perpendicular to its axis, the fraction between the planes will represent the ordinary propeller screw of two blades, as represented by bold lines in the engraving. This screw, like those in Figs. 1 and 3, is generated by a right line perpendicular to its axis.

Fig. 6 is an illustration of a true screw of four threads. The initial points of two of the threads are the same as those shown in Fig. 5; the other two have the same relative position to each other, but their initial points are advanced axially. If a section be cut from a screw of this description by planes passing through and perpendicular with the axis, the fraction inclosed between the two planes will represent the duplex or "Mangin" propeller, which is one with four blades, two of which are in advance, axially, of the other two. This form, although used to some extent abroad, has been but little employed in this country.

The illustrations show right-hand screws in every case, and when "forward" is mentioned reference is had to the direction in which the screw will progress when properly revolved. When the "after" or pressure side of the screw is mentioned, it refers to the opposite direction.

Fig. 7 gives an illustration of a screw of uniform pitch, the generatrix of which differs from that of the regular screw. Its generatrix is a right line at right angles to the axis, extending to the centre of the thread and thence curving forward to the periphery. If this screw be cut by a plane perpendicular to the screw's axis, as A, B, the line presented, when viewed in
the direction of the axis, is not a radial line, as in the preceding illustrations, but is radial to the centre of the thread, thence curving in the direction in which the screw advances, thereby giving a convex face to the thread from the point where the curve begins. This screw, like all others in which the pitch is uniform radially, may be generated by either of the lines shown. It may be generated by the line exposed by the plane $A, B$ cutting the screw, or by the line exposed by a plane cutting the screw parallel with and passing through the axis, as exhibited by the terminals of the thread in the illustration. It is only necessary that they retain the same relative position with the axis, as observed in the illustration.

This form of screw, when employed for propellers, is known as the "Griffiths."

Fig. 8 represents the manner in which the "Griffiths" screw is generated, and the remarks on Fig. 4 are also applicable in this case.

Fig. 9 gives an illustration of a screw of uniform pitch with a curved generatrix, the curve presenting its concave side to the forward part of the screw and making an acute angle with the axis backward.

This is a form that has been employed for propellers, and belongs to the class known as "overhangs."

Fig. 10 represents the manner in which it is generated.

Fig. 11 gives an illustration of a screw of uniform pitch generated by an arc of a circle, the centre of the circle lying in the axis of the screw in the after direction. If this screw be cut by a plane perpendicular to the axis, as $A, B$, the line exposed, when viewed in the direction of the axis, is a spiral, the curvature of which increases from the axis to the periphery, as shown in the engraving. This also belongs to
the class of "overhangs," the pressure side of the thread being concave. It is one of the earlier forms employed for propellers, being adopted by Hodgson in England in 1844. Combined with the uniform expanding pitch, it is known here as the "Isherwood," having been employed extensively by the eminent gentleman of that name when engineer-in-chief of the United States navy.

Fig. 12 represents the manner in which this screw is generated.

Fig. 13 gives an illustration of a screw of uniform pitch, the generatrix of which is a right line making an acute angle with the axis backward. This screw also belongs to the class of "overhangs," because the thread bends backward and the pressure side is concave.

If this screw be cut by a plane perpendicular to the axis, the line exposed, when viewed in the direction of the axis, is an arithmetical spiral, the curvature of which increases from the periphery to the axis, being the reverse to that of Fig. 11. The degree of curvature will vary with the angle which the generatrix forms with the axis. Like the preceding screws, it may be generated, by either of the lines shown, by a right line making an acute angle with the axis backward, or by the spiral curve on a plane perpendicular to the axis.

Fig. 14 represents the manner in which the screw may be generated. Combined with an expanded pitch, this screw, when used for propellers, is known as the "Hirsch" propeller. But it appears to have been first suggested by Mr. J. W. Nystrom, who obtained letters patent for it, and it is fully described in his treatise on the screw propeller.

Fig. 15 gives an illustration of a screw of uniform pitch axially but of a variable pitch radially, the pitch diminishing
from the periphery to the axis. The generatrix of this, unlike those of the screws described hereinbefore, has no fixed position in regard to the axis, the peripheral end traveling through a greater axial distance than the axial end. The generatrix represented is a right line, which forms a right angle with the axis only in the centre of the screw. If this screw be cut by two planes perpendicular to the axis, one plane cutting forward and the other back of the centre of the screw, as shown by the two bold lines, the lines exposed will be arithmetical spirals, the curvature of which will depend on the angle which the generatrix makes with the axis at those positions. Another peculiarity of this screw is that the thread forward of the centre will have a convex face, and the half back of the centre a concave face.

Fig 16 represents the manner in which this screw may be generated. The length of the cylinder is the axial distance traveled by the axial end of the generatrix in one convolution, and the length of the directrix is the distance traveled by the peripheral end during the same time.

When a screw of uniform pitch is required to be practically constructed, a guide for the peripheral end of the generatrix becomes necessary. This guide can be made of any sheet material and laid out as shown in Fig. 17, Plate II. The base line is equal to the circumference of the circle that the end of the generatrix describes in revolving about the axis of the screw, and the perpendicular of the triangle is equal to the pitch of the screw. A right line drawn from the end of the pitch to the end of the circumference line is the developed helix of the screw. If the triangle be curved so that the base line conforms to the circumference of a circle, as shown at the end of the triangle, it will form what is technically
termed the directrix of a screw. If the screw is to be of uniform pitch from axis to periphery, the axial end of the generatrix is simply required to follow the axial line with the same axial velocity as the peripheral end. If the screw is desired to be of diminished pitch at the axis, then the axial end of the generatrix is required to have a less axial velocity than the peripheral end. This has the effect of altering the angles of the helices of the screw.

Fig. 18 illustrates the difference in the angles between a screw of uniform pitch radially and one of diminished pitch at the axis to the extent of the difference between the perpendicular of the two triangles. The lines drawn parallel with the circumference or base line represent the pitch of the screw as divided into so many fractions by planes cutting the screw through at those points. The lines drawn parallel with the perpendicular or axis line represent the circumference or disk of the screw divided into so many parts by planes cutting through at those points. Now, it will be observed that the same fraction of pitch is equal to the same fraction of circumference. In Plate II, Figs. 17 and 18, both the pitch and the circumference are divided into eight parts, and the angles formed by the hypothenuses of the fractions of pitch and fractions of circumference are all equal, thus proving the screw to be uniform in pitch axially. If the angles were not equal, the screw would be of variable pitch axially. It is well to observe the foregoing facts, as they become of importance in considering fractions of screws and in analyzing and determining their character.
EXPANDING PITCH.

The preceding illustrations and explanations have reference to screws of a uniform pitch in an axial direction; but screws for propelling purposes are not confined to this description, and are frequently made with an increasing or expanding pitch axially.

The line $a$, Fig. 19, Plate II, is the developed helix of a screw similar to Fig. 18. If another line, $b$, is drawn, starting from the end of the circumference line, intersecting the pitch line between its extremity and the circumference line, it will represent the helix of a screw of the same diameter but of less pitch than the previous one. If the greatest of these angles, or pitches, be extended through one-half of the circumference, beginning at the circumference line and terminating at its centre, and the lesser pitch extend through the other half, beginning with the termination of the first pitch and terminating at the pitch line, which it will intersect equidistant between the two pitches, as $c$, the helix thus constructed will be that of a screw of expanded pitch; but the transition from one pitch to the other is abrupt, and would not answer for efficient propellers.

In Fig. 20 the circumference is divided into four parts, and the pitch line is intersected by four angles, corresponding to four pitches—maximum, minimum, and two intermediate. If the maximum pitch be extended through the first fourth of the circumference and the next less pitch extend over the
adjacent fourth, etc., as represented in the illustration, the developed helix will intersect the pitch line at the mean pitch, as before. The transition from maximum to minimum pitch is more gradual than in Fig. 17. This subdivision may be performed again, as in Fig. 21; the greater the division the less abrupt will the helix be. If the division be carried to infinity, a curve will ultimately be obtained which will be the true helix of a screw expanding uniformly from minimum to maximum pitches.

Hence the curve of the helix of the uniform expanding pitch screw may be defined as that described by the angles of the pitches due to dividing the difference between the maximum and minimum into an infinite number of pitches, each successive angle to extend through an equal and successive fraction of the circumference, and the whole to form a continuous line.

Fig. 21 illustrates the manner in which the curve may be attained by construction. The circumference line is divided into any number of equal parts, and the distance between the maximum and minimum pitches is intersected by the same number of lines, all emanating from the extremity of the circumference line; and hence there will be an angle for each space which the circumference line is divided into, the angles representing the helices of screws of so many pitches. Beginning with the maximum angle or pitch at the circumference line, let each successive angle extend through each successive space of the circumference, the least angle, or that of the minimum pitch, terminating at the pitch line, which it will intersect at the mean pitch. The helix will now be composed of a number of angles, but by describing a curve through the intersections of the angles with the ordinates
EXPANDING PITCH.

erected on the circumference line the proper curve will be attained.

A formula for determining the lengths of ordinates by computation will be found on page 33.

All the foregoing illustrations have reference to the whole pitch, but an entire convolution of the helix is never now employed for propellers. If a fraction, as one fourth, be selected from the helix (Fig. 21), the fraction would not contain the maximum and minimum pitches; if selected from the centre, it would only contain the two pitches adjacent to the mean pitch; if selected from either end, it would only contain the maximum or minimum, according to the end selected from and the adjacent pitch. The procedure, however, is the same as for the entire pitch in describing the curve of any fraction. Fig. 21 shows where a fraction of one fourth of the pitch is required to expand from the same minimum to maximum pitches that the entire helix does. The fraction is to be divided and ordinates erected just as in the case of the whole circumference. The angles being the same as for the entire helix, they are extended through the fraction in a similar manner to that previously explained. In other words, the fraction is the entire figure reduced by scale to one fourth size. The ordinates of the fraction are one fourth the length of the corresponding ordinate in the large figure. When pitch is mentioned without being qualified in considering screws of ordinary pitch the mean pitch is to be understood.
FRACTIONS OF PITCH OF SCREWS.

THEIR CONFIGURATION, GENERATRIX, ETC.

In order that the reader may become familiar with the different modifications of the screw employed for propelling vessels and be able to distinguish the character of a screw when but a fraction of it is represented, the whole pitch or an entire convolution of the thread has hitherto been considered.

Although it was with a single-threaded screw of one entire convolution that Smith achieved his first substantial success in the "Archimedes" in 1839, that form of screw is no longer employed for propelling vessels. A fraction of the pitch, which is here understood to be the proportion of length which the propeller bears to one convolution of the thread measured parallel with the axis, independent of the number of blades, is found in practice to be all that is necessary in order to construct an efficient propeller.

A fraction of pitch is also much more conveniently applied to a vessel than the whole pitch.

It sometimes occurs in the experience of workmen employed in the manufacture of propellers that a drawing will be received that has been prepared by a draughtsman unskilled in the making of working drawings of the propelling screw. These drawing may consist of a single view, and that a projection, without figures or explanation. In order to proceed intelligently with the construction of a screw from such
a drawing, it is first required to be analyzed and its generatrix and pitch ascertained. The form of the generatrix and of the helix described by the generatrix in its progress along the axis gives to the screw its peculiar character. Hence, when a fraction of a screw is under investigation and its configuration is such that the character of the screw cannot be distinguished by mere observation, it becomes necessary to analyze it and to determine geometrically its generatrix and pitch.

Four examples are here given of fractions of screws—that is, one of each of the principal forms employed at present for propellers.

Fig. 26, Plate III, represents a fraction of the pitch of a screw of four threads, the generatrix of which, projected on a plane parallel with and passing through the axis, is an arc of a circle. This screw is commonly known as the "Isherwood."

If a screw of one convolution of its thread be viewed in line with its axis, the periphery of the thread will appear as a circle, or, in other words, the thread will bound the entire disk viewed; but if the screw be cut in halves by a plane either perpendicular to or parallel with the axis, and one half be viewed in the same direction as the whole was, the periphery of the thread will only extend through one half of the circumference, or the half of a convolution will cover one half the disk. If the thread be cut in quarters, one fourth of the convolution will cover one fourth the disk, etc. Hence, when the fraction of the circumference through which the blade of a propeller extends is known, the pitch of the screw is readily ascertained.

Let Fig. 26 be the projected drawing of a fraction of a screw of which the pitch and generatrix is desired to be ascer-
tained. First, proceed to describe a circle equal in diameter to the screw; upon the circle project the blade, the periphery of which is presented to view; ascertain what fraction of the circle the blade occupies, and whatever proportion it bears to the whole circle, so will the fraction be to the same proportion of the whole pitch. In the drawing under consideration one blade extends through one sixth of the circle, and it then follows that the fraction is one sixth of the whole pitch. If the screw have an expanding pitch, it would be one sixth of the mean pitch, and the amount of expansion would have to be determined after the manner explained for the measurement of the screw by ordinates. If in the projection of the screw blade on the circle the forward and after edges are found to be radial lines, it then follows that the form of the edges depicted in the elevation is that of the generatrix, which, in the present instance, is an arc of a circle. This being ascertained, the generatrix in the other direction, or that projected on a plane cutting the screw perpendicularly to its axis, is required. Find through what fraction of the pitch the generatrix extends; from this it will be known that the curvature of the line exposed by a plane cutting the screw perpendicularly to its axis will extend through the same fraction of the circle.

In the case under consideration the generatrix is supposed to extend through one twelfth of the pitch. Within this distance draw a series of the generatrix, as \(a\), an equal distance apart. Divide one twelfth of the circle into the same number of equal parts as are contained in the fraction of pitch, as \(b\), and draw radii from each point of division. Draw a line perpendicular to the axis of the screw through the series of generatrix, as \(c\), project the points of their intersection, and intersect the centre line of the thread of the screw, as \(1, 2, 3, 4, 5,\)
etc. Through these intersections describe arcs through the twelfth of the circle, intersecting the various radii. Then a line drawn through the intersections of the arcs and the radii will be the curve required. On the other hand, if the curve projected on a plane, cutting the screw perpendicularly to its axis, be given, the generatrix projected on a plane parallel with and passing through the axis can be ascertained by merely reversing the operation.

Fig. 27 represents a fraction of a screw, the generatrix of which, projected on a plane parallel with and passing through the axis, is a right line, forming an acute angle with the axis backward. This is the "Nystrom," lately known as the "Hirsch" screw. The manner of ascertaining the pitch and generatrix is exactly similar to that of the preceding example. In this screw the section exposed by a plane passing through the screw perpendicularly to its axis is an arithmetical spiral, being the reverse of that of the "Isherwood," in that the curve quickens toward the axis of the screw, whereas in the "Isherwood" the curve quickens towards the periphery.

In the elevation the fraction is configured so as to diminish the fraction of pitch of the blade at the periphery. It will be observed that, though the configuration of a screw may have some influence on its propelling efficiency, the character of the screw is not affected thereby.

Fig. 28 represents a fraction of a screw whose generatrix projected on a plane parallel with and passing through the axis is a right line to the centre of the blade and then curves forward to the periphery of the screw. This is the "Griffiths" screw, which has met with extensive application in England and, to some extent, in this country. The pitch and genera-
trix are ascertained in a similar way to those of the preceding examples.

Fig. 29 represents a fraction of a screw whose pitch expands from hub to periphery, and whose generatrix is a right line having an axial velocity greater at its periphery than at its axis. To ascertain the generatrix of a screw of this description, proceed to project the blade on the circle; and if the two edges are found to be radial lines, it then follows that the form of the edge depicted in the elevation is that of the generatrix. If the periphery extends through one sixth of the circumference of the screw, then the fraction of the pitch at the periphery is one sixth of the entire pitch, and, the edges being radial lines, the blade consequently covers one sixth of the circle at the hub. Hence the fraction of pitch at the hub is also one sixth of the entire pitch at that point. For example, suppose the length of the fraction at the periphery measured on an axial line to be 12 inches and at the hub 11 inches: then the pitch at the former will be 72 inches, and at the latter 66 inches. It will be observed in this example that the generatrix is at a right angle to the axis in the centre of the screw, but converges on either side toward the axis, which results in a convex working surface forward of the centre and a concave surface aft of the centre.

In the foregoing examples the configuration of the fractions, as premised in the elevations, produces radial lines when projected on the circle, in which case the generatrix was at once ascertained to be that of the outline of the fraction in the elevation. It frequently occurs, however, that the configuration of the blade is such as to have a variable fraction of the pitch. For example, in Fig. 28 the blade is represented as having a diminishing fraction of pitch toward the periphery, the
blade being so shaped as to make its generatrix almost impossible to be discerned by mere inspection. In order, then, to ascertain the generatrix, it is necessary to cut the screw by a plane either perpendicular to its axis or parallel with and passing through it. The method of accomplishing this will become obvious after an investigation of the examples given and the acquirement of knowledge as to the properties of their elements.
MEASURING THE SCREW.

Every screw should be accurately measured before being fitted to the vessel. If this suggestion be carried out, many of the seeming anomalies of the performance of screws would disappear. It is only by such a procedure that reliable data may be had upon which to base the performance of a ship. Inaccuracies are liable to occur in the manufacture of screws, even when such are in the hands of the best workmen, and owing to their peculiar form an inaccuracy is not easily discernible by mere observation, whereas it can be readily detected by measurement.

The author was once required to make an accurate measurement of a screw propeller which had been removed from a United States naval vessel in order that another might be substituted. The screw had given good results and an accurate drawing was required of it. It is useless to attempt to attain accuracy in anything by makeshifts; therefore the usual method of measuring screws was discarded and facilities provided to insure correctness. These facilities need not be elaborate, but their accuracy is indispensable.

Fig. 1, Plate IV, shows the method of measurement. First, a substantial straight-edge, somewhat longer than the radius of the screw, was prepared; at one end of this was secured a boss, the lower end of which was turned to fit the shaft hole in the hub of the screw. One edge of the straight-edge was arranged to pass through the axis of the screw. A
MEASURING THE SCREW

block to slide along the straight-edge was next made, which had sides at right angles with each other, and near each end strips, which projected some distance above and below the block, were fastened. These strips were for the purpose of keeping the block flush with the straight-edge, and also to be a guide for the measuring rods in taking the dimensions. A segment of a circle was next made, the inside radius of which was equal to that of the screw. A number of small, straight rods were made upon which to mark the ordinates. The screw was carefully leveled. The end of the hub which had been turned off when the shaft hole was bored was assumed to be perpendicular to the axis of the screw, and the measurement proved such to be the case. The straight-edge, with its boss, was placed in its position and a bolt passed through and was secured at the bottom of the hub. The straight-edge was now leveled and a plumb-line suspended from it touching the periphery of the screw, the screw's radius being thus ascertained. The radius was divided into five equal parts and lines drawn through the points of division across the straight-edge; these lines were lettered A, B, C, D, and E. The measuring rods were marked correspondingly. Small flexible rods were fastened to the edges of the blade by means of clamps similar to clothespins, which rods extended to the points that the blade would have had if the corners had not been cut away to a certain radius. The segment of the circle was fixed in the position represented in the engraving and the straight-edge brought over the extreme points of the blade, as determined by the flexible rods. The fraction of the circumference through which the blade extended was ascertained and marked on the segment. This fraction of circumference was divided into four equal parts, and lines were
drawn across the segment. These divisions represented fractions of circumference. In every position in which the straight-edge was employed it was carefully leveled before the measurements were taken, and hence the base from which all the ordinates were taken was a plane perpendicular to the axis of the screw.

The straight-edge was adjusted over the extreme point of the after edge of the blade and the slide adjusted so that the inside edge of one of its strips coincided with the line $A$ on the straight-edge; the rod $A$ was then passed down alongside of the strip until it touched the blade, and the distance to the straight-edge was marked upon it.

The slide was then moved to the line $B$, and the rod $B$ was proceeded with likewise. In a similar manner the other rods were successively marked, after which the straight-edge was moved to the next division of the circumference, the plumb-line suspended from the straight-edge determining the proper position.

The slide being adjusted, the ordinates were marked successively at the several radii in the same manner as in the preceding instance on another side of the rods. The straight-edge was then adjusted perpendicular to the next division on the segment, the rods again marked, and in a similar manner the remaining divisions were proceeded with. In this way numerous dimensions were obtained both radially and axially.

All of the ordinates were from the same base, and from the conditions of the case their lengths were measured on a line parallel with the axis of the screw. The dimensions of all the blades were taken in a precisely similar manner, and showed the screw to be very irregular. It had evidently been made from a pattern of one blade which had become
MEASURING THE SCREW.

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distorted while being molded. A drawing was made of the screw according to the dimensions, and it was found to be an "Isherwood" of an excessively expanding pitch.

Fig. 2 represents the manner of ascertaining the pitch. First, draw a base line equal in length to the circumference of the screw, at one end of which erect a perpendicular of indefinite length for the pitch line. At the other end lay off the fraction of circumference through which the blade extends, as $b$, $c$, and divide this fraction the same as the segment. From the points of division erect perpendiculars. On the first of these lay off the length of the first ordinate from the after edge of the blade and through the intersection draw the hypotenuse of the triangle from the end of the circumference line, extending to and intersecting the pitch line. Then the distance on the pitch line between the base and hypotenuse will be the pitch of that portion of the blade from the after edge to the first ordinate, which is one fourth. From the end of the first ordinate draw another circumference line, as $d$, $e$, and erect a pitch line, as in the preceding case; lay off the length of the second ordinate, draw the hypotenuse, and ascertain the pitch of the second fourth of the blade. In a similar manner the pitch of the remaining portions of the blade can be determined.

After the pitch of the periphery of the screw has been ascertained, it is necessary to know whether the pitch varies in a radial direction, to determine which the operation just described is repeated, the only difference being the employment of the ordinates and circumferences corresponding to the radius at which they were taken.

Fig. 3 represents the developed helix at each radius.

Fig. 4 represents a side elevation of the blade.
THE SCREW PROPELLER.

The pitch can also be computed by similar triangles, thus: Let the circumference of the screw be 50.265 feet, the fraction of circumference inclosed between two of the ordinates 1.047 feet, and the length of the ordinate or the fraction of pitch .5 feet. The pitch of this portion of the screw is then found by the following proportional statement:

$$1.047 : 50.265 :: .5 : 24 \text{ feet.}$$

As the fraction of circumference is to the whole circumference, so is the fraction of pitch to the whole pitch.

The amount of helicoidal area can be obtained directly from the blades by lining them off, and then computing the sum of the areas of the figures obtained.

A convenient way of obtaining the helicoidal area is to provide a sheet of paper and trace the outline of the blade upon it. The paper can then be stretched upon a board and the figure laid off. The area can then be computed by Simpson's rule, or any of the rules for determining the area of irregular figures.

Plate V illustrates a device for measuring the pitch of screw propellers more elaborate and expensive than the previous one, but in which the same principle is followed. It will be observed by the illustrations that it is a metallic device requiring excellence in construction. It possesses the merit of convenience for measuring the pitch of the blades separately. The details and the application of the contrivance are so plainly illustrated on the plate as to render unnecessary any extended description.

The dotted outline of a blade is shown in position ready to be measured.
DRAWINGS OF SCREW PROPELLERS AND THEIR CONSTRUCTION.

Plate VI gives a working drawing of a true screw propeller made to scale. It has its fraction of pitch reduced near the hub. The shaded parts in the plan and elevation represent sections through the blade. That on the plan is the section exposed by a plane cutting the screw perpendicular to its axis, and that on the elevation is a section exposed by a plane cutting the screw parallel with and through its axis. The shaded sections on the triangles are developments of the different sections of the blade. The base of the triangle is the developed fraction of the circle through which the blade extends on the plan, and the perpendicular ordinates represent fractions of the pitch at the intersections, marked, correspondingly, 1st, 2d, etc. The generatrix of this screw is a right line at right angles to the axis, and, consequently, the same face line is used, whether the mold is swept up or a pattern made for the blade.

Plate VII illustrates a working drawing of an "Isherwood," or, as it is also called, the "Bureau of Steam Engineering" screw, it being the type more generally employed in the United States Navy at one time than any other. It is generally made with a pitch uniformly increasing from the forward to the after edge of the blades about 20 per cent. The shaded section on the elevation is a section exposed by a plane cutting the screw parallel with and through its axis.
THE SCREW PROPELLER.

The face line of this section is the one to be used for the sweep when the mold for the screw is built up in loam. The curved shaded section on the plan is a section exposed by a plane cutting the screw perpendicular to its axis. The face line of this section is the one to be used when a pattern is made for the blade built of parallel thicknesses of wood. The radial shaded section on the plan is merely a scale of thickness through the thickest part of the blade. The sections of the blade shown on the triangles are similar to those explained on Plate IV.

Plate VIII illustrates a working drawing of a "Griffiths" screw cast entire. Explanations which relate to the different views and sections applied to the previous plates are applicable to this.

Plate IX illustrates a working drawing of a "Nystrom" screw propeller.

The "Hirsch" screw differs from the "Nystrom" merely in its configuration.
PLATE X.

ADMIRALTY SCREW.

PLATE X is a drawing that represents the type of screw adopted by the English admiralty. The same type is also much employed in this country at the present time, especially in modern United States naval vessels, and is of the modified "Griffiths" form, with separable blades. While this construction is more expensive and also heavier than that in which the screw consists of a single casting, it possesses several advantages over the latter. The breaking of a blade, which is a not infrequent occurrence, can be readily made good by the substitution of another. Indeed, it is customary for vessels fitted with this kind of screw to carry spare blades, and thus be prepared to meet such cases of emergency. The mean pitch (within a limited extent) may be varied by proper adjustment of the blades, the bolt holes in the flange of the blade being elongated to admit of this being done. They are constructed as true screws; that is, are generated by a right line at right angles to the axis and have a uniform pitch. The guide iron, as designated on the drawing, is by some made of plate metal shaped to the proper angle, then bent to the required curvature, and screwed to a base of wood which supports it in position. It is intended to be used when the blade is swept up. But in a screw of this construction it is preferable to make a pattern of the blade by building up pieces of parallel thickness of pine to the necessary height and then shaping
the blade according to the drawing, as described in connection
with Plate XIV. It has been the author’s invariable prac-
tice to make the guides of wood, according to the manner de-
scribed in connection with Plate XIV, and when employed to
build the pattern on the radius of the guide is made to equal
the radius of the propeller.

In commencing the design of a propeller for a given ship
the diameter and pitch are first to be decided upon.

The Admiralty screw is constructed as a true one; that is,
it is generated by a right line at right angles to its axis and
has a uniform pitch. The form of blade developed on a plane
of the standard Admiralty screw is an ellipse, the major axis
of which equals the radius of the propeller, the minor axis
being equal to four tenths of the major axis.

When, owing to the diameter being limited, sufficient
blade area is not obtainable by these proportions, the ellipse
may have its minor axis increased to 5 or 5½ tenths of the
major.

With regard to the number of blades, if the necessary disk
area can be obtained with a three-bladed propeller, this form
is preferable; but if the draft is limited, a four-bladed pro-
peller may be necessary. The boss or hub will equal the di-
ameter of the propeller divided by 3½ to 3¾, which will cut
away about one fifth of the area of standard ellipse.

In laying down the propeller, the plan (Fig. 1), with the
developed blade, is first drawn.

The pitch angles for the different sections are next deter-
mined. Let the base of the angle at the periphery, as indi-
cated by A, B (Fig. 2) equal the radius of the propeller. Then
the perpendicular B, C, or distance between the intersection of
the angles and base with the axis, is equal to the pitch divided
by 6.2832. It is sufficient to compute for the periphery only
and then divide the base or radius of the screw according to
the sections of the blade desired, as indicated by 1, 2, 3, and 4.
From these intersections of the base draw lines intersecting the
axis at C, and these hypothenuses will be developed pitch
angles at the different sections of the blade. The width of
the blade taken from the development on the plan is laid off
on the angle corresponding to the section selected. The
intersection of the different widths on the pitch angles deter-
mines the outline of the blade as viewed from its periphery
toward the axis of the screw or end view of the blade of the
screw. The end view of the blade having been determined,
the projected area of the blade on the plan, as well as the
'thwartship view on the elevation, is determined by projecting
the intersections thus obtained on the plan and elevation, as
is plainly shown in the drawing.

The thickness of the blade is next to be determined. Mr.
S. W. Barnaby says that the thickness of the tip should be as
small as possible consistent with good casting or forging, as
the case may be. When gun-metal is employed, it is generally
made about \( \frac{1}{4} \) of an inch for each foot in diameter of the
screw. For the thickness radially, it is customary to assume
that the total pressure on the blade tending to break it about
the root is proportional to the indicated thrust, or to \( \frac{P}{\rho R} \)

where \( P \) = I.H.P. per blade,
\( \rho \) = pitch of propeller,
\( R \) = the revolutions.

The "leverage" is proportional to \( D - d \),
where \( D \) = diameter of propeller,
\( d \) = diameter of boss.
THE SCREW PROPELLER.

The moment of inertia of the root section of breadth $b$ and the depth $h$ is proportional to $bh^2$; then $h$, the required thickness of the middle of the root, is obtained from the formula

$$h' = \frac{cP \cdot (D - d)}{\rho \cdot R \cdot b}.$$

the value of the coefficient $c$ being about 230 for gun-metal and 90 to 100 for forged steel blades.

In the above formula $\rho$, $D$, and $d$ are to be taken in feet and $b$ and $h$ in inches.

The thickness of the blade at the tip and root being decided, the thickness at the different sections may be determined as follows: Draw a straight line, as $C, D$, on the elevation for the face of the blade; project the intersections 1, 2, 3, and 4 onto this line, which will locate the different sections of the blade; lay off the thickness at the root and tip as determined, and join these by a straight line; then the distance between the face and back lines at the different sections is the greatest thickness of the blades at those points.

In a working drawing the developed thickness through the different sections should always be shown. A convenient way is to arrange them as represented by Fig. 3, where the developed pitch angles, as previously determined for the elevation, are laid off. On each of these angles the length of the elliptic arc of the corresponding section, as shown on the developed blade in the plan, is laid off. These lengths are bisected, and at these points the greatest thickness of blade at the respective sections, as previously determined, is laid off. Circular arcs described through these middle points and the
extremities of the sections determine the areas of the different sections.

While the foregoing is correct for steel or bronze, the edges of which are worked down after casting or when the blade is forged, it will make the edges too thin, especially at the outer sections, to insure a good casting in gun-metal or iron. Hence it is advisable to describe an arc of small radius at the extremities of the sections or edges of the blade, and make the circular arc of the back of the section intersect the middle point and the arcs at the extremities.

The radius of the edges can vary, according to the diameter of the screw, from $\frac{1}{8}$ to $\frac{1}{4}$ inch for gun-metal or bronze, and from $\frac{4}{8}$ to $\frac{6}{8}$ inch for cast-iron.

As shown by the drawing, the thickness of the blade at the root is not disposed symmetrically about the centre of the flange. This allows an advantageous arrangement of the bolts, and enables one more to be put on the face side, where it is more needed than on the back. To determine the size of the bolts:

Let $P$ be the I.H.P. per blade, and $k$ the distance of the centre of pressure of the blade from the under side of the flange, in feet;

$K$, the coefficient, ranging from 18 to 21 for gun-metal;

$C$, the diameter of the bolt circle in inches;

$R$, the number of revolutions per minute;

$B$, the combined area, in square inches, of the bolts at the bottom of their threads, on the face or pressure side of the blade:

Then

$$B = \frac{PhK}{CR}.$$
The number of bolts having been decided, their diameter is readily found by dividing the aggregate area, as previously found, by the number of bolts; the resultant quotient is the area of a single bolt. All the bolts on the sides of the blade are made the same size for convenience.

These illustrations include the types of screws most efficient in practice which are the most generally employed for propellers. Innumerable forms have been tried for propelling instruments, but all the merits of a good propeller are included in the foregoing types.

To enumerate all the different modifications of submerged propellers would entail a vast amount of labor and would possess no other than a mere historical value.

A working drawing of a screw propeller should always show the different sections, as illustrated in the plates.

The cheapest screw propeller is one cast entire. When the blades are made separately and secured to the boss, the cost is increased from 25 to 50 per cent.

Mr. Thornycroft employs a very good construction in his torpedo-boat propellers (Fig. 1). The root of the blade, as

![Fig. 1.]

well as the key, is made wedge-shaped, the latter being tapered also in the direction of its length. When this key is driven hard in the blade is held absolutely fast without
any projecting bolts or flanges. By making the blades of forged steel they can be made extremely thin and sharp, which is a matter of great importance.

Fig. 2 shows another neat construction, more suitable for larger propellers, especially when made of bronze. A pin formed on the end of the blade fits into a socket in the boss and is keyed fast thereto. By making the blades separate from the boss they can be more conveniently finished. A propeller cast entire, especially if very large, is a difficult object to handle.
PLATE XI.

THE EXPANDING OR INCREASING PITCH SCREW.

This type is very old, and would seem to have originated with one of the earliest investigators of screw propulsion. H. Emerson, in 1754, distinctly and definitely advocated its principles. It was subsequently revived by Mons. Bourdon in 1824, and by Tredgold in 1827, and later by Woodcroft. Of late years it has had extensive application.

The alleged advantages of a propeller of expanding pitch are, that as the screw advances through the water the forward portion encounters a resistance due to a solid body moving through water at rest; but water being an exceedingly mobile substance, that with which the screw first comes in contact has motion imparted to it by the pressure exerted by the screw, and this motion of the water is accelerated by the portions of the blade following, and, consequently, the after portions of the screw will press upon the water with a motion opposite to that in which the screw is advancing. In order, then, to equalize the pressure over the whole blade, the pitch must be increased proportionate to the motion acquired by the water. The increase in pitch usually employed is from 15 to 25 per cent.

The absence of method and accuracy among draftsmen in laying down screws of expanding pitch attracted the attention of Mr. D. M. Greene, formerly of the engineer corps of the United States Navy, who, in a communication published in the Journal of the Franklin Institute in 1866, gave an elaborate
THE EXPANDING OR INCREASING PITCH SCREW.

demonstration of the principles involved, and accompanied it with the following formula for computing the ordinates for the curve of the helix and a method of laying down a working drawing of an expanding pitch screw. Mr. Greene says: "We are of those who believe if there is any merit in the screw of uniformly expanding pitch, that advantage can be taken of it only by making it so. With this view we have constructed a formula for calculating the ordinates of the curve of the guide plate used in sweeping-up the mold, and have adopted a new method of constructing the sections of the blades."

Example: Let the maximum pitch equal 24 feet, the minimum pitch 20 feet. Let one blade cover .16 of the circumference of the screw, and the number of ordinates be 10.

Then will:

\[ y_1 = \frac{16}{100} \cdot [24 \left( 10 - \frac{1}{2} \right) + \frac{20}{3}] \]

\[ y_2 = \frac{16}{100} \cdot [24 \left( 2 \times 10 - \frac{1}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_3 = \frac{16}{100} \cdot [24 \left( 3 \times 10 - \frac{1}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_4 = \frac{16}{100} \cdot [24 \left( 4 \times 10 - \frac{3}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_5 = \frac{16}{100} \cdot [24 \left( 5 \times 10 - \frac{4}{3} \right) + \frac{20}{3} \times 20] \]

\[ y_6 = \frac{16}{100} \cdot [24 \left( 6 \times 10 - \frac{5}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_7 = \frac{16}{100} \cdot [24 \left( 7 \times 10 - \frac{5}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_8 = \frac{16}{100} \cdot [24 \left( 8 \times 10 - \frac{5}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_9 = \frac{16}{100} \cdot [24 \left( 9 \times 10 - \frac{5}{2} \right) + \frac{20}{3} \times 20] \]

\[ y_{10} = \frac{16}{100} \cdot [24 \left( 10 \times 10 - \frac{5}{2} \right) + \frac{20}{3} \times 20] \]

In the value of \( y_{10} \), if we put unity for .16, the value of \( y_{10} \) becomes 22 feet = \( \frac{24 - 20}{2} \) = mean pitch. It will be observed, also, that the second differences are constant, both of which facts prove the accuracy of our results.
We now proceed to illustrate our method of constructing the working drawing of the screw given in the preceding example, so far as it is essentially different from the methods we have seen employed.

Let $R = 8$ feet; draw $KK'$, Fig. 3 (Plate XI), as the centre line of the blade and its sections; take any point $E$ as a centre, and with a radius equal to the radius of the screw describe an indefinite arc. Calculate .16 of $360^\circ = 57^\circ 48'$, and lay off half of this angle on each side of $KK'$; the sector $EFH$, thus determined, will represent the projection of one blade on a plane perpendicular to the axis. Next divide the radius into any number of equal parts, say four, and through the points of division draw the arcs $ab$, $cd$, and $ef$ concentric with $FH$; also draw an arc $AC$ 6 inches outside the periphery, to represent the position of the guide plate.

Now, divide the arc $AC$ into any convenient number of equal parts, generally ten; but in this case we use only five. Through the points of division draw radii dividing the projection of the blade into five equal sectors. Next calculate the circumference of the radius $AE = 8' 6''$, and find .16 of that circumference, which will be 8.5452. Draw $A'C'$ tangent to $AC$ at its middle point; make it equal to 8.5452 feet; divide it into five equal parts, and draw lines from $A'C'$ and the points of division to the centre $E$; also draw tangents to the middle points of each of the arcs $FH, ab, cd,$ and $ef$.

Then will $A'C', FH', a'b', c'd', e'f'$ be the developments of the corresponding arcs upon the tangents drawn to their middle points, and, consequently, perpendicular to $KK'$. The intersections of these tangents by the lines drawn from $A'C'$ to the centre will evidently be equidistant, and will represent the
positions of the points of division of the arcs in their respective developments.

Next, upon the opposite side of $E$, at equal intervals, each slightly greater than the length of the screw (or the greatest computed ordinate), draw as many perpendiculrars to $KK'$ as there are arcs upon the blades. Then project the extremities and points of division of $FH'$, in lines parallel to $KK'$, upon the most remote of the perpendiculrars last drawn; and in like manner project $a'b', c'd'$, and $e'f'$ upon corresponding perpendiculrars. $F'H'$ will then be found in $F''H''$, $a'b'$ in $a''b''$, $c'd'$ in $c''d''$, and $e'f'$, in $e''f''$.

Finally, lay off at the points of division of $F''H''$ the ordnates $y_1, y_2, y_3, y_4$, already computed, and draw a curve through their extremities; in like manner lay off these same ordinates at the points of division of $a'', b'', c'', d''$, and $e''f''$ and draw curves through their extremities.

The curves thus constructed will represent the exact lengths (not projections, as is usually the case) of the parallel helices of the blade, of which the arcs $FH, ab, ed$, and $ef$ are the projections. Laying off upon these curves the proper thicknesses of the blade, and drawing another set of curves through their extremities, we have the sections (A), (B), (C), and (D) of the blade represented in their actual dimensions, from which the measurements for the "thickness strips" can be directly taken.

Fig. 4 (Plate XI) represents a "thickness strip" made in a single straight piece, then cut away, by sawing, at small intervals, so that it may be easily bent to fit the circular arc of the blade.

To construct the drawing for the guide plate, Fig. 2 (Plate XI), we have only to lay off $AC = 8.5452$ feet, divide
it into five equal parts, erect perpendiculars, lay off the computed ordinates at the points of division, and draw a curve through their extremities.

We have in this example used only five ordinates, that number being sufficient for our purpose of illustration. In a working drawing, however, it is of course better to use ten.

Figs. 2 and 3 (Plate XI) comprise all that is necessary in a working drawing of a screw propeller of uniformly expanding pitch excepting only the length of the hub, which we have not represented, and the sweep, which, in the case of a blade curving backward, would require a representation either of the sweep or of its curvature.

In the case of a screw having a uniformly varying pitch from hub to periphery, if such a screw or propeller were desirable, our formula is equally applicable, it being only necessary to construct guide curves at equal intervals from hub to periphery, or simply two curves, one at the hub and the other at the periphery. We should have remarked before that the ordinates calculated for the periphery of a uniformly varying pitch "fore and aft" answer equally as well for all sections between the periphery and hub as for the curves of the guide plate.

The length of the ordinates may also be computed by the following rule: Divide the fraction of maximum pitch by the number of ordinates, and the quotient is the first constant; multiply the quotient thus obtained by the number of the ordinate for the first product. From the quotient of the fraction of the maximum pitch divided by the number of ordinates subtract the quotient of the fraction of the mean pitch divided by the number of ordinates; divide this difference by the number of ordinates, multiply this last quotient by the square of the number of the ordinate, and subtract the
last product from the first. The difference is the length of the ordinate required.

Example: Let the circumference of a screw equal 53.4 feet and the blade extend through .16 of the circumference. Let the maximum pitch be 24 feet and the mean 22 feet; then base or circumference equals $53.4 \times .16 = 8.5452$ ft.

\[ 24 \times .16 = 3.84 \text{ ft.} = \text{fraction of maximum pitch} \]
\[ 22 \times .16 = 3.52 \text{ ft.} = \text{fraction of mean pitch}. \]

Let the number of ordinates be ten; then

\[ 3.84 \div 10 = .384 = \text{one tenth of the fraction of maximum pitch}, \]

and

\[ 3.52 \div 10 = .352 = \text{one tenth of the fraction of mean pitch}. \]

\[ \frac{.384 - .352}{10} = .0032 = \text{one tenth of the difference} \]

between the one tenth fractions of maximum and mean pitches; and this last quotient is the second constant to be used in the formula for obtaining the lengths of ordinates, which are the following:

\[ y_1 = .384 \times 1 - (.0032 \times 1) = .3808 \]
\[ y_2 = .384 \times 2 - (.0032 \times 4) = .7552 \]
\[ y_3 = .384 \times 3 - (.0032 \times 9) = 1.1232 \]
\[ y_4 = .384 \times 4 - (.0032 \times 16) = 1.4848 \]
\[ y_5 = .384 \times 5 - (.0032 \times 25) = 1.8400 \]
\[ y_6 = .384 \times 6 - (.0032 \times 36) = 2.1888 \]
\[ y_7 = .384 \times 7 - (.0032 \times 49) = 2.5312 \]
\[ y_8 = .384 \times 8 - (.0032 \times 64) = 2.8672 \]
\[ y_9 = .384 \times 9 - (.0032 \times 81) = 3.1968 \]
\[ y_{10} = .384 \times 10 - (.0032 \times 100) = 3.5200 \]
PLATE XII.

PATTERN-MAKER'S PREPARATIONS.

PLATE XII illustrates the preparations necessary to be made by the pattern-maker when a screw is to be swept up, in order to enable the molder to construct the mold.

Fig. 1 represents the guide. This is usually set six inches beyond the periphery of the blade, to allow for the joint of the mold. Its base is curved to a radius, due to the distance of its position from the axis. An inclined rail of the same cylindrical curvature as the base, supported by uprights, is attached to the base. The ordinates a, b, and c, which, for like letters in the figure, are of equal length, show how the curvature of the rail may be obtained. The fraction of circumference through which the blade extends is laid off on the base, and perpendicul ars are erected at these intersections. A centre line is laid off on the base, and from it a perpendicular is erected. The base is divided according to the number of ordinates given on the drawing for the guide, and perpendiculars are erected from the intersections. A line parallel with the base is laid off on the inside. The distance of this line from the base will depend on the distance from the guide seat to the lowest point to which the blade extends, which distance is governed by the amount allowed for the joint below the blade, usually about four inches, and the character of the generatrix. The length of the ordinates given on the drawing
are laid off on the lines provided for them, and a thin batten is tacked on the rail, the edge of which passes through the upper extremities of the ordinates. A line drawn along the edge of the batten gives the directrix or path which the sweep is to travel at the guide. In a true screw the development of the guide line is a straight line, but in an expanding pitch screw it is a curve.

Fig. 2 shows the guide finished.

It is preferable to have a guide at the hub also, as greater accuracy is obtained by this arrangement. It may be determined in the same manner as in the case of the outside guide.

The blade, face sweep or generatrix is made from plain board, according to the character of the screw, as delineated by the drawing. The working edge is the line exposed by a plane cutting the screw parallel with and through its axis, when the thickness of the blade is all on one side or back of the generatrix, as determined above. It sometimes happens that a drawing will show the thickness divided by such a line as in the case of a uniform V-thread, in which case the generatrix will have to be determined by cutting the screw obliquely to its axis, the sweep being set accordingly. An easy way of solving this problem is to work it out on a block of wood to scale and develop the generatrix from the line produced on the block.

The face sweep is divided according to the number of thickness pieces intended to be used. The distance between these pieces is usually from 12 to 18 inches. A small notch is cut into the edge of the sweep at these points, which leaves a line marked on the face of the mold; by this the thickness pieces are set and nailed to the mold.

The thickness pieces are made of board three quarters of
an inch thick, their dimensions being obtained from the drawing. They are kerfed with a saw in lines parallel with the axis of the screw, to facilitate their being bent to the lines made by the sweep on the mold.

Fig. 3 represents a thickness piece kerfed and bent.

Fig 4 represents an arrangement by which any kind of screw may be swept. If the screw is to be a true one radially, the sweep is secured to the end of the rod in a fixed position; but if the screw is to vary in pitch radially, the sweep is pivoted to the rod, in order that its ends may travel at different axial velocities. The rod is free to slide vertically through the arms secured to the spindle, and the latter is free to revolve about its axis. The illustration shows the guides and sweep set ready for the piers to be built on when the blades are formed.
PLATE XIII.

SWEEPING UP PROPELLER.

PLATE XIII illustrates different stages in the molding of a propeller by the sweep method.

Fig. 1 shows the first stage, in which a foundation plate is provided. Upon this a course of soft brick is laid in loam. A coating of loam is spread over the brick, and this coating is swept by the seat sweep, which is secured to the spindle and revolves about its axis. The seat sweep provides the positions for the hub and guide.

Fig. 2 shows the second stage, in which the hub is formed by a sweep secured to and revolving with the spindle. The guide is shown on its seat. In the method here shown the hub is the pattern which is destroyed when the mold is stripped. It is generally built of brick and covered with loam.

Fig. 3 shows the third stage, in which the blade sweep slides on the spindle. By this way only screws having the same pitch at the hub and periphery can be swept. A guide is necessary at the hub to secure accuracy in consequence of the sweep being cut away to clear the hub. Without a guide at the hub the sweep will spring, and a consequent distortion of the surface will result. It was the practice of the author to make a cylinder of wood for the hub somewhat less than the hub's least diameter. Upon this cylinder kerfed
strips were secured, one edge of which coincided with the pitch angle at the hub and formed a guide for the hub end of the sweep. Nails driven into the cylinder and allowed to project held the loam which covered it while the hub was being shaped by its sweep. The device is shown in Fig. 4, Plate XII. One pier is shown in course of construction and two ready to be lined off for the blades. The piers are built of soft brick and loam, with sufficient straw or other material to render them porous; the face is covered with loam, which is reduced to the intended form by the sweep following the guide as it is moved up and down.

Fig. 4 shows the fourth stage. One blade is shown lined off, with its thickness-pieces set. Another shows the pattern of the blade formed by filling in between the thickness pieces with green sand and then dressing it down to conform to the shape as determined by the thickness pieces. In this condition it is ready for the cope, or the upper part of the mold, which is built upon it. The cope is shown completed on the other pier. The copes in the method illustrated are built of soft brick and loam strengthened by plates and rods, and are made porous, to admit of the free escape of the gases. Sometimes copes are built of cast-iron staves or plates, provided with prickers for holding the loam in contact with the pattern. Adequate means are provided for handling the different pieces by a crane. After drying, the copes are lifted off, the pattern taken out, and the mold dressed and dried. When sufficiently dried it is ready to be assembled and rammed up. It is generally assembled in a pit and securely clamped together, after which sand is filled around it and well rammed, in order that the mold may be able to withstand the pressure resulting from filling it with fluid metal.
The foregoing is an outline of the method adopted in the molding of a propeller by sweeping up. With different molders the procedure will vary somewhat in detail. After casting the mold should not be opened too soon, and when it is opened the whole upper part should be uncovered at the same time, in order that the distortion from shrinkage may be as uniform as possible in all the blades.

By the behavior of a mold in a certain case while it was being filled with metal it was thought that the propeller casting had failed to be a sound one. The following morning one of the blades was uncovered in order to examine the casting, which proved to be sound. The other blades remained covered for two or three days longer before they were uncovered. When the screw was measured, the author’s prediction was confirmed by the blade which was uncovered first showing a greater distortion than the others.

When screws are made by the method just explained, with the face down, as they should be made, the invariable tendency of the distortion is to reduce the pitch. By keeping a record of these distortions the author was able to make allowances to counteract them and to produce in the casting a pitch nearly equal to that required by the drawing.
PLATE XIV.

MAKING PATTERN OF BLADE.

PLATE XIV illustrates the method of constructing a pattern for a propeller. As is plainly represented, the blade is built up of pieces of parallel thicknesses. The shaded part in Fig. 1 shows a section of the blade as exposed by a plane cutting it perpendicular to the axis of the screw. The line of the face of this section, that is, the working face of the screw, is a straight line radiating from the centre. For a right screw, as represented, this line will be the same for all the pieces, while the line on the back will vary as the thickness varies. In a screw of less pitch at the axis than at the periphery the face line will be a radial line in the centre of the screw only or midway between the extremities of the pitches. From this centre forward the face line will change to a convex one, and from the centre toward aft the face line will be a concave one. Therefore the face line in such an instance will change with each parallel piece, making it a matter of considerable labor to determine the face line for the different sections. The lines can be determined by the analytical method explained in connection with Plate II, or the pieces, instead of being parallel in thickness, may be tapered—that is, may be made proportionately thicker at their peripheral ends than at their inner ends, in proportion as the pitch at the periphery is greater than that at the axis.
This type of screw, however, is now seldom employed for propellers.

In cases where the pitch is the same at both axis and periphery the face line remains the same for all the parallel pieces. With the face line determined, it becomes an easy matter to lay off the different pieces. The shaded sections show the developments on a plane of the curved thickness of blade where they are correspondingly marked hub 1st, 2d, 3d, etc., on the plan, Fig. 2.

A templet of the face line, made of thin stuff, and which has the radii of the different sections marked on it, facilitates the laying off of the pieces. A templet for each section and periphery, one edge of which coincides with the curvature of the arc described by the radius and the other end coincides with the face line, should also be provided.

Beginning with the first or bottom piece, proceed to prepare it as follows: Lay off the face line by the templet and mark the positions of the sections and periphery. Work out the face-line square with the sides; lay off by the templet the arcs due to the different radii of the sections; step off with dividers the length of the bottom rectangle in the section marked hub, and transfer this length to the arc corresponding to the radius of the hub on the piece. Step off the length of the rectangle of the same piece at the next section marked 1st, and transfer it to its arc. Proceed similarly for the other sections and periphery, then tack a batten on the piece whose edge intersects the length of the rectangles on the arcs, and draw a line along the edge, which will be the back line. Work the piece to this line and square with the sides, when the piece will be ready to be set in place to begin the building up of the blade. It is advisable to leave a surplus of length at
each end of the piece to be cut off when it is fitted in building up the blade. All the parallel pieces are laid off in a similar manner.

It is preferable to build the blade pattern with the face down. It is then necessary to provide two guides, one corresponding to the pitch at the periphery, the other to the pitch at the hub. The guides are made in accordance with the manner described on Plate XII.

A surface board is prepared, and an outline of the blade is drawn upon it with arcs extending through where the different sections are located. A central radial line is drawn and the guides are screwed to the surface board according to the positions determined by the lines drawn. If the screw is to be cast entire, a hub is prepared and secured to the board the axis of which coincides with the centre from which the arcs on the board have been struck. If the blades are to be bolted on the boss, the flange is set up in place of the hub. The necessary preparations are now complete to allow the building of the blade to proceed. When the propeller is cast entire one blade pattern only need be made, as it can be shifted in molding to make the number of blades required.

In building up the blade pattern the parallel pieces are fitted against the guides and hub or flange, one upon another, and are glued and also nailed together wherever the nails will not interfere with the tools in working the pattern off. After the building of the blade is completed, the back is roughed off; it is then turned over and the face finished. It is then laid off to the required configuration and the surplus material removed, after which the back is finished, and the pattern, when shellacked, is ready for the foundry.

If the drawing, as sometimes occurs, does not show the
thickness as illustrated in the plate, the pattern maker can determine the deficiency in the following manner: Lay off the outline of the blade on a plane perpendicular to the axis of the screw (Fig. 2), and draw radial lines at a tangent to the widest parts of the blade. Describe arcs where the thickness sections are desired; develop the arc at the periphery of the screw, as shown in the view where the triangles are. From the extremities of this developed arc draw lines of equal length to the centre; bisect the angle so formed, and draw another line to the centre for a centre line. On this centre line mark off the different radii of the sections and draw lines through these intersections at right angles with the centre line, and intersecting the two outside lines. Then the lengths of the lines so intersected will be the developed arcs of circles the radius of which is the distance between their intersection of the centre line and the centre of the screw. They are also the bases of the triangles by which the angle for the thickness section is determined. If the screw be a right one, the perpendicular of the triangle will bear the same proportion to the whole pitch that the arc does to the whole circumference; that is, if the arc extends through one sixth of the circumference, the perpendicular will be one sixth of the pitch, and the inclined line connecting the extremities of the perpendicular or pitch line and the base or circumference line is a right line. If the screw is of expanding pitch, the perpendicular will be one sixth of the mean pitch, and the inclined line will be a curve. The casting of a screw is never exactly like the pattern, owing to its distortion by shrinkage. The author has kept a record of the dimensions of patterns of a number of screws and the measurements of the castings made from them, and has found the
THE SCREW PROPELLER.

distortion somewhat irregular, but having an invariable tendency to reduce the pitch. The amount of this reduction will average about 1.7 per cent. This amount, therefore, should be added to the pitch in making the pattern, in order to compensate for the distortion of the casting.

Extended elucidation of the descriptive geometry of screws of various kinds, with their analyses, is contained in McCord's "Lessons in Mechanical Drawing."
PLATES XV AND XVI.

ILLUSTRATIONS OF LIFTING SCREWS AND THE BEVIS FEATHERING SCREW.

Plate XV illustrates the arrangement for a lifting screw. This is constructed in such a manner that the propeller can be lifted entirely out of the water when the ship is under sail. It is now employed only in special cases, as it is found that the gain in resistance this arrangement offers over that experienced when the screw is disconnected from the engine and allowed to revolve freely does not compensate for the additional complications that it entails. The screw is provided with journals and suspended in a banjo frame. The forward journal is fitted with a T-head, which fits into a cheese coupling on the end of the shaft, by which the rotation of the shaft is transmitted to the propeller. In connecting and disconnecting the propeller from the shaft it is necessary to bring the latter into a fixed position.

Feathering screws possess many of the advantages of lift-ings crews, and are much less difficult and objectionable in their application. They are preferable wherever it is desirable to combine steam-power as auxiliary with sails, as they permit the latter to be advantageously employed when the propeller is not in operation. One of the most practical feathering screws is that of R. R. Bevis, of England. H. M. S. "Calliope," which successfully escaped from the harbor of
Apia, Samoa, during the disastrous hurricane in 1890, was fitted with one of these screws.

Plate XVI gives an illustration of the Bevis screw.

This arrangement is as follows: A short shaft or spindle at the root of each blade passes inside the hollow boss and has a lever or arm attached firmly to it. The stern shaft is made hollow, and rods are carried through the centre of the shaft from the ends of the arms on the blades pindles to a collar near the forward end of the shaft, which can be moved forward or backward by means of a nut working over a thread cut on the outside of the shaft. The details are clearly shown in the diagrams. The dotted lines in the figure on the right hand show the position of the arms when the blades are set at the working angle, and the full lines their positions when the blades are feathered and placed in a fore-and-aft direction. This arrangement has the additional advantage of enabling the propeller blades to be readily set to any desired pitch without docking the ship, the regulating screw on the stern shaft being suitably graduated for this purpose. These propellers are fitted to a large number of steam yachts.
PLATE XVII.

THORNYCROFT'S SCREW TURBINE—PLATE XXX, MAL-LORY STEERING PROPELLER—PLATE XXXI, Mc-GLASSON REVERSIBLE PROPELLER.

PLATE XVII illustrates the latest form of the Thornycroft screw turbine propeller.

Figs. 1 and 2, in which the propeller is represented as cut in half, will be seen to comprise a cylinder containing within it a body or boss of such a shape that the channel is gradually contracted from the forward to the after end. Within the forward part of the propeller there are revolving blades attached to the forward part of the boss, which is keyed onto the shaft and is separate from the after part. The pitch of the forward edge of the blades, multiplied by the number of revolutions, is approximately equal to the velocity of the supply water. The pitch increases uniformly along the length of the blade, imparting a uniform acceleration to the water.

Aft of the revolving blades are numerous guide blades of contrary curvature, which are fixed to the rearward portion of the boss and to the cylinder. The cylinder is attached to the stern of the vessel.

The area of the channel through the propeller is so proportioned as to suit the acceleration of the water which has been caused by the blades. Thus, at the forward end is a large opening which will admit a certain quantity of water at
the velocity of the supply; at the after end the area is restricted to that necessary to allow the exit of the water at the speed of the discharge. The long, tapering body forming a prolongation of the boss outside of the cylinder allows the annular stream of water to close gradually without the formation of eddies.

As the long pitch of the screw causes considerable rotation of the water, the guides are so formed as to direct the water into a straight line aft, and the rotary motion is utilized without loss except by friction.

It is claimed that the thrust delivered by the curved guides amounts to about one third of the whole thrust. Forward of the cylinder and keyed on the shaft are screw blades of the same radius as the cylinder, the function of which is to propel when going astern. They are of the same pitch as the leading edge of the turbine propeller blades, and advance through the water without propelling when the vessel is going ahead. When the vessel goes astern these blades receive water not only through the turbine cylinder, but also from outside of it.

Fig. 3 shows the screw turbine as ordinarily applied to a single screw vessel.

Fig. 4 shows an arrangement of the screw turbine suitable for ferry steamers. The sternway screw being placed in the bow, it is in the most favorable position for propelling and steering astern.

It is claimed for the screw turbine that it is a more efficient propeller than the paddle-wheel for shallow waters not encumbered by grass and weeds.

Steering propellers have been used to some extent. The most common of these have a universal joint between the stern bearing and the screw. The after end of the screw is sup-
ported in a frame hinged in a vertical position. By moving this frame to one side or the other of the ship the axis of the screw is made to assume an angular position in regard to the axis of the tail shaft, and thus, while the engine is running, a turning of the vessel is effected.

The Mallory system of propulsion, illustrated on Plate XXX, was applied to the United States torpedo boat "Alarm." By it the vessel could be not only propelled and steered both backward and forward, but could be made to perform with great celerity and in small space the most difficult evolutions, utterly beyond the capacity of rudders and fixed screws, while the engines were uninterruptedly running one way. The system, however, was complicated and liable to derangement, and has had, therefore, but limited practical use.

Another reversible screw, by means of which the speed can be altered and a stop or reversal of the ship made while the engines continue to run in one direction, is the invention of Mr. Robert McGlasson, of England. The device is handy and capable of being made substantial and durable.

Plate XXXI shows the details of the McGlasson feathering or reversible screw. Figs. 1, 2, and 3 show the details of the gear for small craft, and Figs. 4, 5, and 6 show the details of the gear for larger vessels.

The blades are made flat, and their section presents an easy form for passing through the water. The ends of the blades entering the boss are made conical, with a short arm projecting from a side of each; to these arms are pivoted short connecting rods, the other ends of these rods are pivoted to a cross-head fitted to the end of a rod which runs through the tail shaft, the latter being made hollow for the
purpose. The inboard end of the internal rod is fitted with a flat key, which key passes through slots in the sides of the hollow shaft. The slots extend along the shaft, to enable the key to be moved along in them. The ends of the key are secured to a grooved sleeve, which can be moved along while it revolves with the shaft. When the sleeve with the key is moved along the shaft, the rod and cross-head, with the connecting rods, etc., attached to the end inside of the boss, are caused to move also; and, as a result, the stems of the blades are made to turn in their sockets. The boss is made in parts, to enable the mechanism to be inserted. One joint of the boss passes through the axis of the blade at right angles to the axis of the shaft.

Various devices are employed to move the sleeve. For small craft it is moved by means of a hand lever, as illustrated; but for larger vessels gearing driven by the main shaft is made to do the work when a clutch is thrown in gear. A hydraulic engine, which can be operated from the bridge if required, is also proposed for moving the sleeve.

The inventor states: “My outboard gears are operated while the engine is running; they are in frequent use, so it is impossible for them to stick through inattention or corrosion, especially as they are effectively lubricated, the boss being kept full of oil. Experiments that I have made prove the fact that they have to perform but light work. Developments of the propellers and gears that I advocate have been running for over twelve months and have given great satisfaction. They are easily operated, and one opened out after several months’ service was found in as good condition as when floated.”
The molecules of a fluid have no relative cohesion. The force that holds the fluid together is that of gravity. The colder or heavier molecules will find their way to the bottom and the lighter ones to the top or surface of the fluid. Any force superior to the gravitating force of the molecules that is brought to bear on them will cause their displacement.

Suppose Fig. 2, Plate XVIII, to be a plane submerged in a fluid. Let the plane revolve in the direction of the arrow and come in contact with a particle, as $a$. The particle will be deflected in the direction in which the plane or force is moving, as $a'$. Should the plane again come in contact with it in this last position, it will be again deflected in the direction of $a''$, and, finally, beyond the circle described by the periphery of the plane, as $a'''$.

The action of flying off of a body when acted upon by a revolving force is known as centrifugal action. If the plane is advanced while it is revolving, instead of the particle flying off at right angles to the axis of rotation it will traverse the plane and leave it at some point, as at $b$, Fig. 3; and the path so described by the particle on the plane will be a parabolic curve.

If the plane is fixed obliquely to the axis of rotation instead of parallel to it, the particle will then be acted upon by
two forces. The resultant of one is the centrifugal action, and of the other a direction of motion of the particle at right angles to the plane, as shown in Fig. 4.

The resultant of the two directions of motion given to the particle is a spiral path, according to the laws of the resolution of forces.

If the axis to which the plane is fixed is free to move longitudinally, the reaction of the impact or resistance the particle offers to being moved will cause the plane to move in an axial direction opposite to that of the moving particle. During this axial movement of the plane there will be a succession of impacts with the particle, each one accelerating its motion, and the particle will follow a spiral path of increasing curvature from its first to its last impact with the plane. If the force applied to move the particle is greater than the gravitation of the latter, it is impossible for the particle to take any other course under the foregoing conditions.

Revolving in a fluid, the advancing edge of the plane would meet with a succession of particles, when the displacement would take the form of streams, as illustrated in Fig. 5. With each successive impact of the plane with the streams the velocity of the latter is accelerated, in consequence of which the streams will gradually diminish in width toward their point of discharge from the plane.

When a propeller which is a fraction of a right screw revolves the water between the blades is acted upon at the same moment by two forces, one being the propulsive force resulting from the oblique action of the revolving blades, the other being the centrifugal force generated by their rotation.

Attempts have been made to counteract the centrifugal action by making the generatrix of the screw a curve, as
shown in Fig. 2, Plate XVIII, instead of a right line at right angles to the axis. Such a screw was patented in this country by Nystrom in 1850, and was described by him as a centripetal propeller. Some years later it appeared in a modified form as the Hirsch propeller. In practice, no advantage seems to have been derived from this modification.

A screw drives the water astern in various directions, more or less oblique, and the reaction due to the transverse component of the motion of each particle of water has no effect in propelling the vessel.

The mere outline shape or configuration of a screw has no effect on the direction in which the water is displaced. The direction of displacement of water by a screw depends solely on the form of its generatrix and the obliquity of its generated surface to the axis of rotation. In other words, the direction of displacement depends altogether on the form of the helicoidal surface, and not on its outline shape.

It can be understood from the foregoing why it is not advantageous to employ very long screws. Although the quantity of water acted upon by a screw will increase with its length, more energy is wasted in the divergent direction given to the displaced water, and more energy is also absorbed in overcoming the friction of the increased surface of the longer screws. The gain from the increased quantity of water acted upon does not compensate for the loss incurred in accomplishing it. Machines for pumping water have been devised in which the water is moved by screws instead of by a piston. It was found disadvantageous in those machines to use long screws, owing to the rotary motion which the water acquired from its continued contact with the screw. The most successful of those machines were constructed with a number of
short screws fixed to a shaft, with spaces between them. Thus, upon the water leaving one screw, an interval occurred before it was acted upon by the next, and, consequently, there was less rotary motion given to the water than would have been given had the screw been continuous.

In Smith's first successful experiments with the Archimedes in 1839 he used a single-thread screw of one convolution. During a trial of the vessel on the Thames the screw struck an anchor chain, the force of the blow breaking off the after part of the blade. By this fortunate accident the speed of the vessel was accelerated, and the lesson was duly profited by. A double-threaded screw of one half a convolution was then tried with better results, but Smith obtained his best result with a two-thread screw of one sixth of a convolution. Thus he discovered by trial, as was afterward demonstrated in other ways, that it is not economical to employ the entire pitch of a propeller.

It can be observed, from what has already been said, that if the displaced water is given a divergent action, there must, necessarily, exist in the immediate wake of a propeller a condition of less density than obtains elsewhere in the water, a series of whirls—eddies, as they are called.

When the screw revolves in a fixed position the discharged water assumes the form of a truncated cone, with less density in the interior; but when the screw is advancing, the cone is drawn out, approximating a cylindrical form.

As to the number of blades and the extent of acting surface a screw should have much confusion exists. A propeller, to be steady in action and not produce injurious vibration, should have as few blades as possible. The author has seen screws of one blade experimented with, and, so far as
propulsion is concerned, they prove very efficient; but their
twisting action and the vibration which they caused precluded
their practical use. The result of these experiments, and the
well-established fact that in numerous cases blades have been
removed from a screw or broken off by accident—and if no in-
crease of efficiency resulted from the operation of the screw
with the remaining blade or blades, at least no noticeably
detrimental result followed—have convinced the author that
the performance of a screw is more affected by its length than
by the number of blades. Of course, in the application of a
screw in discharging a mass of water the velocity of its rotation
is an important factor. When revolving slowly, a screw hav-
ing but a single blade would probably not displace so great a
quantity of water as one with several, because with slow rotation
some of the supply water doubtless enters at the periphery,
especially at the bottom; but when the velocity of rotation is
sufficiently great for a single blade to displace the water as
fast as the water can gravitate to it, then the addition of more
blades can only exclude water.

The following illustration of the action of a single-thread
screw in displacing water will, in some measure, elucidate the
foregoing argument:

Suppose a single-thread screw of one convolution and 6.68
feet in diameter, or a disk area of 35 feet, having a pitch of
10 feet, be caused to revolve in a vertical tube; suppose the
water to flow in at the bottom of the tube as fast as it is dis-
charged at the top, the end of the tube and the top of the
screw being in the same plane; then the greatest quantity
of water that the screw is capable of discharging at one revo-
lution is the volume of the tube minus the volume of the
screw. For instance, 35 feet of sea water weighs one ton.
The tube, therefore, will hold $35 \times 10 = 350$ cubic feet, or 10 tons of sea water. Let the volume of the screw equal 10 cubic feet. Then the greatest possible amount of water that could be discharged in one revolution is $350 - 10 = 340$ cubic feet, or 9.714 tons. Now, it is evident that if the screw is cut in halves one half the length of the whole screw can only discharge one half the quantity of the whole screw. If the screw is reduced to one fourth the pitch, only one fourth, or 85 cubic feet, equal to 2.428 tons, can be discharged per revolution. Let the length of the screw and tube be reduced to one fourth of the pitch of the screw, or to a length of 2.5 feet; the tube will then contain 85 cubic feet of water and 2.5 cubic feet of screw. Now, suppose another blade of 1 cubic foot in volume is added, then the screw's volume will be increased to 3.5 cubic feet. Now, add two more blades of 1 cubic foot each, and the screw's volume will then be increased to 5.5 cubic feet and the water reduced to 82 cubic feet. This operation of increasing the number of blades can progress until the tube is filled entirely with the screw and the water excluded therefrom.

As to the most advantageous length for constructing a screw propeller there is no rule which can in every case be applied in order to obtain it. The best that can be done is to observe the practical results of screws in actual operation and to follow the dimensions of those which give the best results under the nearest conditions to that to which the one under consideration is to be subjected.

It is very doubtful whether there will ever be devised a formula that can universally be applied to determine the best form and the most advantageous dimensions for a screw propeller under any given set of conditions.
The following regarding the theory of propulsion is extracted from Mr. R. Sennett's work, "The Marine Steam Engine:"

The principle of momentum and its application to the problems of propulsion were clearly explained in papers contributed to Nos. 2 and 3 of the Annual of the Royal School of Naval Architecture and Marine Engineering. The two principles (momentum and work) are analogous and convertible, as may be seen by the following illustration, and we will point out the features in which they differ and the reason why the principle of momentum is more suitable for solving the general problems connected with propulsion.

If a body move in a straight line under the action of a constant force, \( P \), then at the end of time \( t \),

\[ Pt = mv, \]

where \( m \) = the mass of the body and \( v \) = velocity in feet per second. The mass of a body is a constant quantity depending on its size and density, and is equal to the weight of the body divided by the force of gravity. If \( D \) equals the weight of the body, and \( g \) represents the accelerating force of gravity,

\[ \text{Mass} = \frac{D}{g}. \]

For a cubic foot of sea water, which weighs 64 lbs,

\[ \frac{D}{g} = 2, \text{ nearly}. \]

The product \( mv \) is called the momentum of the body, so that the force multiplied by the time through which it acts is equal to the momentum of the body; if the body had, initially, a
given velocity, change of momentum should be substituted for momentum.

If in the time $t$ the body has moved through a space $x$, we have

$$Px = \frac{1}{2}mv^2,$$

where $\frac{1}{2}mv^2$ is called the half vis viva of the body; so that the force multiplied by the distance through which it acts (which is work performed) is equal to half the vis viva generated; if the body had initially a given velocity, the work done would be equal to one half the change in the vis viva.

Therefore, in the simple case of a constant force acting on a body in a given direction, we see that if it be considered by the time during which it acts its measure is the momentum or change of momentum produced; while if it be considered with respect to the space through which it acts, it should be estimated by the change in the half vis viva produced.

By comparing the two systems of measuring the effect of a force the principle of momentum is more readily applicable to propulsion than the principle of work, etc.

The following is another way of considering the same matter, as given by Seaton, in his "Manual of Marine Engineering":

Let $V$ be the velocity of the stream of water with respect to the ship and $v$ the velocity of the ship; then the velocity of the stream with respect to the water $= V - v$. This is called the slip of the stream, or the velocity with which it moves over the water on which it is projected.

The percentage of slip, therefore, is $\left(\frac{V - v}{V}\right) \times 100$. The propelling effect of this stream of water will depend on the mass projected in a given time.
If the water be taken from the surrounding water, whose velocity with respect to the ship is \( v \), the velocity imparted by the machinery is then \( (V - v) \), which is the slip. Then, if \( M \) be the mass of water projected in a second, \( (V - v) \) its velocity in feet per second, \( W \) its weight in pounds, and \( g \) its gravity (= 32), the momentum = \( M (V - v) = \frac{W}{g} (V - v) \). If \( A \) is the area in square feet of the issuing stream and \( V \) the velocity of issue, the weight of a cubic foot of sea water being 64 lbs., the weight of water projected in a second = \( A \times V \times 64 \). Therefore, the momentum of stream =

\[
\frac{A \times V \times 64}{32} \times (V - v) = (A \times V) \times (V - v).
\]

And this is the measure of the propelling force, and is equal to the resistance of the ship in pounds, and, consequently, it is also equal to the thrust of the shaft on its bearings.
PLATE XIX.

MARINE PROPULSION.

So much has been written on the subject of marine propulsion, no small portion of it having been by men eminent in the engineering profession, that it may seem superfluous to some that the author should make any addition thereto. But from a perusal of the matter herein contained it may be seen that, in the literature referred to, many varied statements have been made, some of them complicated and mystified by a display of mathematics, others exhibiting greater ability in the construction of sentences than in the elucidation of the subjects upon which they treat, while many are erroneous and misleading.

In the following pages will be found an exposition of marine propulsion which, while intended to contain matter of general interest, is especially addressed to practical men.

Marine propulsion has remained in an unsatisfactory condition ever since the first successful attempt was made to propel vessels by the agency of the steam engine. In many cases designers have been content to fashion propelling screws more in accordance with their peculiar fancy than with regard to the principles involved. There are a few eminent scientists who have striven to disseminate the correct principles of marine propulsion, and conspicuous among them were Professors Rankine and Froude. Numerous papers on the subject, containing
considerable argumentative force, have appeared at different times, but they have lacked one most important essential, namely, that of illustration, and in that have failed to convince even the ordinary reader.

The following pages will, therefore, be accompanied by numerous plates and figures, which will aid materially in the acquirement of a comprehension of the correct theory of marine propulsion.

Before entering upon the subject of the action of marine propelling instruments it will be advantageous to first consider, in the light of modern science, the principles which govern the behavior of fluids on submerged bodies.

The three elements constituting the total resistance to the ship’s motion which the propeller has to overcome are:

1. Frictional resistance, due to the gliding of the particles of water over the skin of the ship.
2. Eddy-making resistance, due to the wake at the stern.
3. Surface disturbance or wave-making resistance.

The first of these is by far the most important.

Dr. Froude says: "It is a common but erroneous belief that a body completely submerged, moving in a straight line at a uniform speed through an unlimited ocean of fluid, experiences resistance to its onward motion by an increase of pressure on its head and a diminution of pressure on its tail end. There would, in a perfect fluid, be absolutely no other resistance than that resulting from the friction of the particles gliding over the surface of the body. A perfect fluid is one free from viscosity or quasi-solidity, such as is possessed by tar or syrup. Water undoubtedly possesses this property, but to such a small degree as to be practically inappreciable."
This may be understood by taking an object the specific gravity of which is but very little greater than that of water. Place it in that fluid, and it will descend through it by its own gravity. If the water possessed viscosity or cohesion among its molecules, even to a small extent, the resistance resulting therefrom would considerably retard or prevent the passage of the body through it.

While there is, practically, no resistance offered by water to a submerged body passing through it other than that of friction, there is a certain limit of velocity due to every form, beyond which there is a resistance arising from the displacement, in the form of eddies, of the water in front faster than it can fall in behind, to equalize the hydrostatic pressure on the body.

This can be demonstrated by taking a plane, as a piece of glass, for instance, and allowing it to descend through the water by its own gravity, first edgewise and then sidewise. Although in each case it would present the same surface and be urged by the same force, the velocity with which it passes through the water would be much greater when descending edgewise than when sidewise.

It follows, then, from the above demonstration that a circular spindle will have a higher normal fluid velocity than a sphere, and the latter a greater velocity than a cube, though they may be of equal surface and urged by equal forces.

The resistances offered by water to a ship are: surface friction, eddy resistance, and wave resistance. Of these the first named, as affecting a large ship with easy lines, is much the greatest item. Dr. Froude relates an instance in the case of a bluff ship of 1100 tons, only 170 feet long, and having a thick stem and stern posts, thereby making considerable eddy
resistance, and at ten knots visibly making large waves, where the surface friction was 58 per cent. of the whole resistance offered at that speed.

Of eddy resistance very little is at present known by which its amount can be even approximately arrived at in all cases; but continual investigation will, no doubt, throw much light on this, the most perplexing of resistances that a body moving through water experiences.

Fig. 1, Plate XVIII, illustrates the theory of stream lines, in explanation of which Dr. Froude, who has done so much to develop the subject, has said:

"Every particle of fluid composing an ocean that passes a body submerged in it must follow some path or other, though it may not be possible to discover what path; and every particle so passing is preceded and followed by a continuous stream of particles all following the same path. The ocean may be imagined to be divided into a series of streams of any size and cross-section, provided they fit into one another and occupy the whole space, and provided that the boundaries which separate the streams exactly follow the natural course of the particles. The streams forming this system must not only be curved, to get out of the way of the body, but might even be required to have, to some extent, a different sectional area, and, therefore, a different velocity of flow at different points of their course. If the streams be traced to a sufficient distance ahead of the body, the ocean will be found flowing on completely undisturbed by the submerged body; there all the streams will have the same direction, the same velocity of flow, and the same pressure.

"Again, if their course is pursued backward to a sufficient distance behind the body, they will all be found again flowing
in their original direction and to have assumed their original velocity.

"The combination of curved streams surrounding the body, which together constitute the ocean flowing past it, return finally to their original direction and velocity, and cannot administer to the body any endwise force.

"The difference between the behavior of water and a perfect fluid is twofold, as follows: first, the particles of water, unlike those of a perfect fluid, exert a drag or frictional resistance upon the surface of the body as they glide over it. This action is commonly known as surface friction, and constitutes almost the whole resistance experienced by bodies of easy shape traveling under water at any reasonable speed.

"Secondly, the mutual resistance experienced by the particles of water in moving past one another, combined with the almost imperceptible degree of viscosity which water possesses, somewhat hinders the necessary stream-line motions, also their nice adjustment of pressure and velocities, and thus defeats the balance of the stream-line forces and induces resistance.

"This action is imperceptible in forms of fairly easy shape. On the other hand, angular or very blunt features entail considerable resistance from this cause, because the stream-line distortions are in such cases abrupt, and degenerate into eddies, thus causing great differences of velocity between adjacent particles of water and consequent friction between them."

The greatest velocity at which a body can move through a fluid without the formation of eddies may be called its normal fluid velocity.

It is an inherent property of a flowing fluid that its pressure under a given head will vary with the velocity of the
fluid. If its velocity is reduced at any point in its path, the pressure of the fluid on the conduit through which it flows is increased at that point. If the velocity is increased, the pressure will become less. If a fluid flows through a conduit whose area of section varies, the pressure exerted by the fluid on the wall of the conduit will be greatest where the area is greatest and least where the area is least, and the velocity of flow will be least where the area of section is greatest and greatest where the area is least.

It will be observed that the streams broaden when they approach the submerged body, which broadening indicates a reduction of their velocity and an increase of their pressure on the body. In flowing by the body they converge until their last breadth is attained, which is at the point of the greatest cross-section of the body, at which point the streams attain their greatest velocity and also produce the least pressure on the body.

A body to pass through water with little resistance should have an easy form and smooth, uniform surface. This applies to the blades of a propeller as well as to the hull of a vessel.

A marine propeller is a device to which the force derived through the engines is transmitted. This force is expended partly in driving the ship in one direction and partly in moving a mass of water in another direction. The fundamental principles of the action of every marine propeller are the same, whether it be a screw, paddle, turbine, or other instrument. The propeller drives backward a certain quantity of water at a certain speed. In so doing it presses backward against the propeller with an equal force, which is called the reaction of the water. This force is transmitted to the framework of the machinery and thence to the vessel, and is the force that
drives her forward. When the ship is starting from a state of rest or is increasing her speed the driving force must be greater than the resistance, but so long as the speed is uniform the driving force and the resistance are equal.

That the foregoing are the correct principles of marine propulsion may be more readily comprehended by reference to Fig. 1, Plate XIX. Let A and B be two similar planes, with a cylinder fitted to B and a piston fitted to A, the planes being immersed under exactly similar conditions. In the case of a vessel being propelled by its machinery there is simply a transmission of the force developed by the engines to the propelling instrument and thence to the water. In the transmission of this force to the water there is a reaction, just as there is in the discharge of a gun, the force from the powder reacting on the gun with a power equal to that with which it projects the shot in the opposite direction.

Let the plane B represent a vessel and A the propeller. Now, if a force be evolved in the cylinder, its effect will be to drive the cylinder in one direction and the piston in another. As the planes are similar, are immersed under similar conditions, resisted by equal amounts, and acted upon by the same force, it follows that the movement of each must be equal and in opposite directions.

The plane B can now be modified so as to give it the form of the forebody of a vessel. As before explained, it will now be capable of moving with a greater velocity, although urged by the same force as before, in consequence of the water being deflected aside, instead of being moved ahead.

Let the force be again generated in the cylinder as before. B, modified, will move faster than the plane A; yet the latter will have moved in the opposite direction to that of B, and
in doing so it will have moved a mass of water; and the extent of the movement of B will depend altogether on the velocity of the water moved by A.

The form of B may be further modified by making the after part resemble the afterbody of a vessel, as indicated by the dotted lines. This form will facilitate the water falling in behind and exerting a forward pressure, which will enable the body to move with a higher velocity, though urged by the same thrust as in the previous instance.

This illustrates marine propulsion in its simplest form.

The following is another way in which the action of marine propellers can be illustrated and the thrust shown to be dependent on the mass of water moved by the propeller:

Let Fig. 3, Plate XIX, represent a wheel constructed in the manner of ordinary paddle-wheels, and Fig. 4 a wheel of similar dimensions, but the construction of which is the reverse of that shown in Fig. 3, in that the spaces between the paddle boards are filled in so as to make them solid and the paddle boards removed. It is obvious that the former, in Fig. 3, would have all the propelling power of an ordinary paddle-wheel, while the latter would have scarcely any, simply from the fact that, while the first is capable of moving a quantity of water equal to the depth of immersion and the length and width of the spaces between the paddle boards, the quantity of water which the latter (Fig. 4) can move is limited to the volume of the open spaces, or that of the paddle boards, in Fig. 3.

The quantity of water that any propelling instrument is capable of moving in a given time is a very important factor in the consideration of its efficiency.

Suppose the paddle-wheel (Fig. 3, Plate XIX) be made to
revolve freely in a close-fitting race or flume, so that no water can flow through the flume unless the wheel be revolved. Let the wheel be one foot wide and of such a diameter that the annular ring of one foot wide at the outer radius will contain an area of 12 square feet, then the volume of the annular ring one foot deep on the face will equal 12 cubic feet. Let the sum of the paddle boards equal one cubic foot. Now, if the wheel be caused to make one revolution, it will move eleven cubic feet of water along the flume, and the reaction resulting from the movement of this water will produce a thrust of the bearings of the wheel, tending to move them in an opposite direction to the movement of the water. Now, suppose the wheel shown in Fig. 4 be made to revolve in a like manner in the same flume. The sum of the volumes of the filling, which in this case may be considered the paddle boards, equals eleven cubic feet where the wheel enters the flume, which makes it possible for the wheel to move only one cubic foot of water for each revolution and to produce a thrust on its bearings of one eleventh of that produced in the case of Fig. 3. It is an axiom that no two bodies can occupy the same space at the same time. Consequently, if the paddle boards enter the stream in the flume they exclude an amount of water equal to their volume.

The same elucidation is applicable to the action of a screw in moving water.

Let Figs. 6 and 7 represent screws of equal diameter, the former an ordinary single-thread screw and the latter the screw reversed; that is, the former consists of a cylinder having a helicoid wound about its periphery, and the latter a cylinder having within it a cylindrical and helical perforation. Assume that the screws revolve freely in close-fitting tubes,
and, when revolved, will displace the water as fast as it flows to them. Further assume, as Rankine did, that the particles of water on being displaced by the screws move in lines parallel with their axes, and that all particles move with the same velocity. Let the screws be of six feet pitch and six feet long, and the tubes of the same length and of such diameter as to make the volume of the tubes twelve cubic feet each. Let the volume of screw, Fig. 6, equal one cubic foot, or one twelfth that of the tube, and of screw, Fig. 7, equal eleven cubic feet, or eleven twelfths the volume of the tube.

Now, if the screws are caused to revolve under the foregoing conditions, the results will, obviously, be the same as those in the preceding example of the paddle-wheels, Fig. 6, and will discharge eleven cubic feet of water with each revolution, while those in Fig. 7 can discharge only one cubic foot with each revolution.

The foregoing is analogous to cases where screws have been constructed to work within an encircling casing or tunnel, in which case it is obvious that the area of the cross-section or disk area of the tunnel limits the quantity of water that can pass through it, and if a portion of this area is occupied by a screw, the area for the passage of water is reduced to that extent.

Instruments for marine propulsion should contain, therefore, the least possible amount of material consistent with their duty. It may be observed as a rule, that, everything else being equal, that propeller is the best which displaces the least water by its own form.

The foregoing plainly illustrates the basis of marine propulsion. In the case of the paddle-wheel filled (Fig. 4) it is
possible for it to move only one cubic foot of water per revolution. The weight of this, multiplied by the velocity with which it is moved and divided by the force of gravity, gives the thrust or reaction. On the other hand, the wheel (Fig. 3) with the ordinary paddle is capable of moving eleven cubic feet per revolution, which, if moved at the same velocity, will produce eleven times the thrust.

The same illustration is applicable to the screws shown in Figs. 6 and 7. The one displacing the least water by its own form is capable of producing eleven times the thrust that the other can accomplish in one revolution, provided the revolutions of the respective wheels are made in equal times.
MARINE PROPULSION. — Continued.

INERTIA AS A FACTOR.

Referring again to Fig. 1, Plate XIX, it is evident that the same pressure or thrust that operates to move a plane in one direction also operates to move the other plane in the opposite direction, and, in order that one may move at a greater velocity than the other, the resistance to the one that it is desired to move the faster must in some manner be reduced, as giving it an easy form for passing through the water—a form that will deflect the streams and allow them to accommodate themselves to the moving body, instead of producing eddies, which the plane itself would do.

If the water displaced by a propeller is moved in the line of the keel, then the thrust or pressure operating to move the keel in the opposite direction will depend on the weight and velocity of the water so moved. The following formula will denote the thrust:

\[ T = \frac{Wv}{g} \]

Where \( W \) is the weight of the water moved by the propeller, \( v \) its velocity, and \( g \) the force of gravity, the energy then absorbed by a moving body is found by the formula—

\[ \frac{Wv^2}{2g} \]
THE SCREW PROPELLER.

The thrust of a propeller on the water is exactly equal to the thrust of the water on the propeller. It is a case of the equality of action and reaction.

The energy developed in a marine engine is divided (eliminating that absorbed by the engine and shafting and that also by the propeller, which is principally surface friction and form resistance in the latter) between the useful work of driving the vessel and the wasteful work of displacing water sternward. If the weight and condition of the water that a propeller takes in and the direction of the displacement be known, the thrust can be closely calculated from the data.

It is just here that the whole theory of marine propulsion has failed to agree with the results of practice.

There is no such thing as a satisfactory and universally applicable formula for solving the problems of propulsion. One reason is that little is known as to the actual weight of water, its condition, and the direction of its displacement which any given propeller can deal with in a given time. It appears certain, from numerous experiments, that propellers can deal with much greater quantities of water than they have a chance of obtaining in the ordinary way. Screws situated as twin screws are prove more efficient than they would when located in the stern aperture of a vessel, because the screws occupy a more favorable position for obtaining a full supply of water.

Mr. Seaton says that although very much has been written on the subject of marine propulsion, and men of undoubted ability have spent much time and money in making researches therein, the best informed among them are still liable to make the most egregious mistakes in designing
screws. This was evidenced on the first trials of H.M.S. "Iris." The performance of a screw when the vessel is under way cannot be determined with any great accuracy from its performance when the vessel is made fast, because the condition of the supply water is different in the two cases, as will here after be explained.

An ideal propeller is that in which every particle of water displaced by it will have a direction in line with the keel of the vessel and in an opposite direction to that in which the keel moves. If these conditions are assumed, the thrust and the energy absorbed in producing it, as well as the energy absorbed in driving the ship, may be calculated by the aid of well-known formulæ.

Referring to Fig. 2, Plate XIX, let the large plane of 35 square feet area be arranged with a curb of 4 feet in depth. It will then be capable of holding 4 tons of sea water. Suppose the small plane is 17.5 square feet in area and similarly arranged with a curb of 4 feet in depth, it will then be capable of containing 2 tons of sea water. Let the velocity with which the 4 tons are moved astern be 12 feet per second; then the thrust will equal

\[
T = \frac{Wv}{g} = \frac{4 \times 12}{32.2} = 1.49 \text{ tons.}
\]

Evidently the thrust on the small plane must be the same, while the velocity of the small plane will equal

\[
\frac{32.2 \times 1.49}{2} = 24 \text{ feet per second.}
\]

The kinetic energy of a moving body varies as the square
of its velocity. Hence the energy absorbed by the large mass which is assumed to be moved by the propeller is equal to

\[
\frac{W \times v^2}{2g} = \frac{4 \times 144}{64.4} = 8.95 \text{ foot-tons per second;}
\]

and the energy absorbed in moving the smaller mass, which is assumed to represent the vessel, is

\[
\frac{2 \times 576}{64.4} = 17.9 \text{ foot-tons per second,}
\]

making an aggregate of \(8.95 + 17.9 = 26.85\) foot-tons of energy per second necessarily expended in driving the vessel at the above speed. The equivalent in horse power of this amount is

\[
\frac{26.85 \times 2240 \times 60}{33000} = 109.38 \text{ H. P.}
\]

Assuming this to be all the power that can be devoted to propulsion in a given case, the only way in which a propeller could be made more efficient is to have it move a greater mass of water at a slower velocity, and produce at least an equal thrust with less expenditure of energy, as the following example will reveal:

Let the propeller move 6 tons of water at a velocity of 8 feet per second; then the thrust will equal

\[
\frac{6 \times 8}{32.2} = 1.49 \text{ tons.}
\]

just as in the first example; but the energy expended in producing this thrust is only

\[
\frac{6 \times 64}{64.4} = 5.96 \text{ foot-tons.}
\]
This leaves \(26.85 - 5.96 = 20.89\) foot-tons available for driving the vessel. Assuming the weight moved by the small plane, or the resistance of the vessel, to be the same as in the first example, the velocity that it will attain by the expenditure of 20.89 foot-tons is

\[
\frac{64.4 \times 20.85}{2} = 672.658,
\]

and \(\sqrt[4]{672.658} = 25.93\) feet per second, against 24 feet in the former example.

It can be understood by the foregoing examples that it is advantageous to produce as great a thrust as possible with the least expenditure of energy, because the energy thus expended is necessarily wasted, and is not available for driving the vessel.

The foregoing elucidates the theory of marine propulsion as propounded by Professor Rankine and accepted by the engineering fraternity as correct, and which led him to enunciate as an axiom that that propeller is the best, other things being equal, which drives the largest quantity of water astern in line with the keel at the least velocity.

But there is another important factor in the problem of marine propulsion that has only recently been recognized, and that is the effect of the inertia of the water or the resistance it offers to being put in motion from a state of rest by the propelling instrument.

In some experiments made with aero planes by Dr. Langley and Mr. Maxim, where the plane was set obliquely to the line of force which drove it through the air, it was found that its lifting force increased faster than that of the force required to drive the plane.
THE SCREW PROPELLER.

Let Fig. 3 represent a plane submerged in a fluid and driven by a force in the direction indicated by the large arrow. It is evident that, in order for the plane to advance its own thickness, a volume of fluid equal to that of the plane must be displaced. Owing to the obliquity of the plane, and to its driving force being greater than that of the gravitation of the fluid, the displaced fluid will have to take a downward direction, the reaction of which will have a tendency to lift the plane.

Now it will require time for this displacement of the fluid. It cannot be started from a state of rest instantaneously with the first application of the force, because each particle of the mass moved must feel the effect of the moving force before it can move. The tendency of the plane to lift is opposed by the force driving it and the pressure of the fluid above. Consequently the plane will tend to move in the path of least resistance, which is the direction that the plane occupies, as indicated by the small arrow. Should the plane be fixed to a rotating axis, the resultant of this tendency will be in the direction of the axis.

As evidence that it requires time for a force to become diffused throughout a mass before it will move, Dr. Anderson observed by the mark made on a parapet by the discharge of
a disappearing gun that the shot must have left the muzzle before the gun commenced to recoil.

A familiar illustration of the effect of the inertia of a fluid is shown in the instance of a shot striking the water and then ricocheting: the energy in the shot is insufficient to displace the water with the velocity at which it is traveling, and which, consequently, it is prevented from penetrating.

Another striking instance of the effect of the inertia of fluids is observed in the explosion of nitroglycerine. The disintegration of the substance and the evolution of its gases take place so rapidly, that before the air can be forced back to give place to the expansion of the gases it has offered resistance sufficient to enable the gases of the substance to destroy almost any solid material upon which it may have exploded.

Professor Rankine laid emphasis on the great loss that arises from the propeller giving motion to the water suddenly. Dr. Froude very properly took exception to the phrase "suddenly," and contended that a body always acquires its motion by degrees from a state of rest. The period in which a body will acquire motion from a state of rest depends on the weight of the body and the intensity of the force applied in giving it motion. When force is first brought to bear on a body of solid substance in order to give it motion, owing to the resistance which the body offers to being started from a state of rest, the first effect of the application of the force is a distortion of the body to a greater or less extent. Now the energy absorbed in this distortion can have no direct tendency to move the body. In the case of a soft, non-compressible substance like water this distortion does not occur in a mass of it; consequently, it does not absorb energy by impact, as a body does which is composed of hard substance. It only
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THE SCREW PROPELLER.

absorbs energy when it is given motion, or when its temperature is increased.

Let a blade, as an oar, be immersed in the water vertically and a fulcrum provided between the outer end and the water, about which the oar may turn. Now if a force is applied to the outer end of the oar to move the water, a great difference in resistance will be experienced whether the force is applied quickly or slowly. If the blade be inclined to the line of motion and the force applied quickly, there can be felt a sensible effort of the blade to move in the direction of the plane which the blade occupies.

The foregoing are some of the reasons that have influenced the author in forming the opinion that the inertia of the water plays a part in propulsion as well as the displacement.

In many instances screw propellers have been replaced by others of smaller diameter, greater pitch ratio, and less helicoidal area, and the latter have proved the more efficient, because more revolutions were rendered possible, and, consequently, the water was acted upon by the blades quicker, and an increased benefit derived from its inertia.
The reactionary effect of a discharged fluid is very different from that of a solid. If a rigid bar is pushed out by a piston and it meets with an obstruction, a reactionary effect from the collision is immediately communicated to the piston through the bar, but the energy impressed into the bar is expended on the obstruction.

When a fluid, as water, is impelled by a piston from an orifice, no additional reactionary effect is felt by the piston in consequence of the water meeting with an obstruction, unless the impediment is so near the orifice as to reduce the area necessary to allow the water to be discharged at the velocity with which it issues from the orifice. If the impediment is so near the orifice as to reduce this area—that is, less than the area of the orifice divided by its circumference—in order that the quantity of fluid discharged may not be reduced also its velocity of discharge must be increased, which implies that more kinetic energy is impressed in it, and, consequently, more work required of the piston. This applies to all devices that discharge fluids.

In hydraulic systems of mining water issues from nozzles with an enormous amount of energy, and the stream is, apparently, as dense and rigid as a bar; yet no difference in its reactionary effect is experienced, whether the water impacts on a bank at 6 or at 24 feet from the nozzle. The reactionary
effect on the nozzle is not altered until the point of impact is so near that the eddies produced by the breaking up of the column of water have a tendency to reduce the area of discharge.

The same phenomenon is observed in impact water wheels of the Pelton type.

The following will demonstrate the reactionary effect of a stream of water issuing from an orifice: Let Fig. 4 represent a vessel immersed in a fluid and fitted with a pipe projecting vertically, the lower end of which terminates at the after part of the vessel. Suppose the pipe to be so arranged that a constant level of water can be maintained in it at a height of 39 feet above the level of the fluid in which the lower end of the pipe terminates. Then there will exist a constant "head" of 39 feet of water.

According to well-known laws of hydraulics the velocity
of water issuing from an orifice will be the same as that attained by a body falling freely from the surface of the water in the pipe to the surface of the fluid into which the falling water flows.

The velocity that a body acquires in gravitating through a height of 39 feet will be

$$8 \sqrt{39} = 50 \text{ feet (nearly) per second.}$$

Suppose the quantity of water discharged to be ten imperial gallons, or 100 pounds, per second. Then we have a column 600 inches long, weighing 100 pounds, and having a volume of 2772 cubic inches discharged per second.

The area of orifice to allow this quantity to be discharged, eliminating retarding influences, is

$$\frac{2772}{600} = 4.62 \text{ square inches.}$$

The reactionary effect, or thrust tending to move the orifice in the opposite direction to that in which the jet is issuing, will equal

$$\frac{100 \times 50}{32.2} = 155.27 \text{ pounds.}$$

The amount of energy necessary to produce this thrust, which is the kinetic energy contained in the moving column of water, is

$$\frac{100 \times 50^3}{64.4} = 3882 \text{ foot-pounds.}$$

Now 3882 foot-pounds per second is all that can be developed under the foregoing conditions, and may be compared to the maximum power developed by the engine. If the vessel is prevented from moving in the opposite direction, all this energy goes out in the moving mass of water. If the column should impact on any object that would destroy its motion,
the energy would be expended on the obstacle, but would have no reactionary effect on the orifice as a result of the impact.

So long as the vessel is prevented from moving in the opposite direction to that of the issuing jet the whole energy goes out in the moving mass of water; but if the vessel is relieved of resistance and allowed to move, then part of the energy is absorbed in moving the vessel; and herein lies the difference between the object sought to be accomplished in marine propulsion by the jet principle and hydraulic mining. In the former, it is advantageous to have as little energy as possible impressed into the moving water, in order that as much as possible may be utilized in moving the vessel in the opposite direction. It is the economical movement of the vessel that is desired, and, as will be shown, the total energy expended is required to be divided between the movement of the ejected water and that of the vessel. In hydraulic mining, the object sought to be accomplished is the reverse of that in marine propulsion. It is the destructive effect of the impact of the issuing column that is desired to be utilized, and it is advantageous to have all the energy possible impressed into the discharged water.

As the head of water and the area of the discharge orifice are fixed quantities, the thrust is the maximum that can be obtained from the reaction of the jet under the foregoing conditions.

The thrust cannot be increased by the jet impacting on an object, as the following will show. The velocity of discharge is determined by the head of water, and the quantity by the velocity and the area of the discharge orifice. Now if the area is reduced, there will be less water discharged, and,
consequently, less energy developed. The distance from the
discharge orifice to any object opposite that will deflect the
moving column of water necessary to allow the free discharge
of the volume issuing from the orifice cannot be less than the
quotient of the area divided by the circumference, or one
fourth of the diameter of the discharge orifice.

The diameter of a circle containing 4.62 square inches area
is 2.42 inches, and one fourth of this is 0.6 inch approxi-
mately. Now let the plate (Fig. 1) be placed 0.3 inch from
the orifice; then the area of discharge will be reduced to one
half, or 2.31 square inches, and consequently, the velocity
remaining the same, only one half the quantity of water can
be discharged than under the former conditions; that is, 50
pounds at 50 feet velocity per second. The reaction or thrust,
therefore, will equal

$$\frac{50 \times 50}{32.2} = 77.63 \text{ pounds,}$$
or one half that in the former case, and the kinetic energy
in the discharged water will equal

$$\frac{50 \times 2500}{64.4} = 1941 \text{ foot-pounds per second,}$$
also one half the first result.

As the energy contained in the discharge-water after it
leaves the orifice is lost so far as it can produce thrust, it be-
comes important in marine propulsion, where a given amount
of energy is developed, to waste as little as possible in that
direction, in order that more can be utilized in driving the
vessel. The energy in a moving mass varies as the square of
its velocity; less will be expended in moving a greater mass
at a slower velocity.
Suppose the "head" (Fig. 4) is reduced to 16 feet, which will produce a velocity of 32.2 feet per second. Let the thrust required be the same as in the former case, that is:

\[
\frac{155.27 \times 32.2}{32.2} = 155.27 \text{ pounds.}
\]

The weight of water necessary to be discharged per second in order to accomplish this thrust will require the area to be enlarged to 7.18 inches.

The kinetic energy in this weight falling with a velocity of 32.2 feet per second will be

\[
\frac{155.27 \times 32.2^2}{64.4} - 2 \text{500 foot-pounds (nearly) per second.}
\]

Thus, the same thrust is produced as in the first case, with considerably less expenditure of energy, simply by allowing a greater weight of water to fall with a lower velocity.

Those who are engaged in hydraulic mining observe the difference in the effect of impact between a large amount of water issuing at a low velocity and a smaller quantity issuing at a higher velocity, and, consequently, sources of supply are sought in high elevations, rather than larger quantities in lower elevations.

From the foregoing demonstration there is little prospect that the jet principle of marine propulsion will ever supplant the screw except in special cases, where economy is not of leading importance.

When the reaction resulting from projecting a body is to be utilized, as in the case of propelling a vessel, it makes a vast difference in the economy of energy or the efficiency of the propeller whether the force applied to project the water is one acting constantly or one acting impulsively. A propeller acting impulsively may be described as follows:
A method of propulsion has been tried where the vessel was fitted with a cylinder and piston, which constituted a pump. One end of this cylinder terminated at the stern, the axis being parallel with the keel. In operation, the piston being at the inboard end of the cylinder, the latter was allowed to fill with water, which the piston was made to expel by causing it to move to the outboard end of the cylinder. The piston was then returned to its inboard position, and, of course, during this return stroke no thrust was produced, as the movement of the vessel resulting from the thrust of the outboard stroke extended through the period of the inboard stroke.

It has been demonstrated in a previous example how 100 pounds of water gravitating through 39 feet acquire a velocity of 50 feet per second and produce a reactionary pressure of 155.27 pounds, and that there will have been absorbed in producing this effect 3882 foot-pounds of energy. We will now consider the case of a force acting impulsively, and demonstrate its greater waste of energy over that where the force acts constantly, eliminating the effect of inertia.

Suppose the level of the water in the stand-pipe (Fig. 4) be maintained at a height of 156.25 feet above the level of the water into which it discharges. Then the water in falling will acquire a velocity of \(8 \sqrt{156.25} = 100\) feet per second, and assuming that the area of orifice remains the same as in the previous example, there will be 200 pounds of water discharged in a second, making a column 100 feet long of 2 pounds to a foot.

Now suppose the water is allowed to flow for half a second and then is stopped for half a second. There will be a column 50 feet long, weighing 100 pounds, discharged in half a second. This is the same quantity as in the previous example, but its velocity is double, or 100 feet per second.
The thrust or reaction resulting from 100 pounds having a velocity of 100 feet per second will equal
\[ \frac{100 \times 100}{32.2} = 310.55 \text{ pounds}, \]
or double that of the first example.

In driving a vessel by the reaction of a jet issuing intermittently every half-second the thrust acts while the jet is issuing, and the motion acquired therefrom continues for the remaining half-second, or during the period the jet ceases. Hence the mean thrust for the entire second is equivalent to one half the thrust produced, which in the present instance will equal
\[ \frac{310.55}{2} = 155.27 \text{ pounds}, \]
the same as in the first example. As before stated, the energy impressed into a moving mass or body varies as the square of the velocity which it attains. Hence the energy contained in 100 pounds moving with a velocity of 100 feet per second will be
\[ \frac{100 \times 100^3}{64.4} = 15,528 \text{ foot-pounds}, \]
or four times that of the first example, while the reactionary force is the same. This results from the necessary increase in velocity required to be given to the discharged water in order to produce the thrust required.

With a screw propeller it is just as necessary to have free discharge as it is to have free access for the supply water. If the rear of the screw is obstructed by any object that will impede the free discharge of the water, the latter will be given greater rotary motion and will partly be discharged tangentially. At each revolution the screw takes in a certain quan-
tity of water, which must be discharged, and the nearer its discharge approximates the velocity and direction of its entry the less will be the energy absorbed by the water from the screw. Hence the velocity with which water is ejected from a propeller is an indication of the efficiency of the instrument in regard to economical propulsion. The greater the velocity of the water discharged the greater will be the waste of energy.

The pump in the jet system takes in a smaller amount of water than the screw and discharges it at a higher velocity; consequently, it is more wasteful of energy.

Riggs proposed to utilize some of the energy contained in the water discharged from a propeller by fixing guide blades behind the propeller upon which the discharged water impinged. These blades were so shaped as to take the rotary motion out of the water and leave it moving directly astern. Later, Mr. Thornycroft combined Riggs’ ideas with his own screw turbine, and claims for the device great efficiency, especially in shallow water.

Eliminating friction and form losses in the instrument itself, the efficiency of any propeller is measured by the amount of thrust which it produces and the amount of energy expended in producing the thrust.

That propeller is the most efficient which, in producing a given amount of thrust, discharges the water with the least amount of energy contained therein; because the thrust is measured by the velocity which the weight of water attains in being moved by the propeller.

It is an error to assert that one propeller may discharge a given weight of water at double the velocity of another and both have the same efficiency.
THRUST, ADVANCE, SLIP, FRICTION, EFFICIENCY.

THRUST.

The thrust which a propeller produces on the bearings of the shaft is equal to the resistance of the vessel.

Professor Rankine gives the following rule for computing the thrust of a propeller:

Rule V.—To calculate the thrust of a propelling instrument (jet, paddle, or screw) in pounds, multiply the transverse sectional area in square feet of the stream driven astern by the propeller, the speed of the stream relatively to the ship in knots, the real slip; or part of that speed which is impressed on that stream by the propeller, also in knots, and the constant (5.66) for sea water or (5.5) for fresh water. That is, if $S$ is the speed of the screw in knots, $s$ the speed of the ship in knots, $A$ the area of stream in square feet (sea water), the thrust in pounds equals

$$A \times S (S - s) \times 5.66.$$  

In 1876 the late Dr. Froude read a paper at the Institution of Naval Architects, in which he proposed an extension and improvement of Mr. Denny's system by the substitution of the indicated thrust of the propeller for the indicated horse power as the ordinates of the diagram. From the curves of indicated thrust most valuable results have already followed, the most
important probably being that the indicated thrust due to the
initial or constant friction of the machinery may be determined
by a simple geometrical construction.

The indicated horse power of itself is not altogether reli-
able as a measure of propulsive efficiency, as it combines in
one item the performances of the ship, machinery, and pro-
peller. Mr. Froude, therefore, deemed it desirable to reduce
the indicated horse power to a force factor by dividing it by
the speed of the propeller, the result being what he termed
indicated thrust.

The indicated thrust may be estimated in either of two
ways:

1. By multiplying the mean effective pressure on the
piston in pounds per square inch by twice the length of the
stroke in feet, and dividing the product by the pitch of the
propeller in feet.

2. By multiplying the actual indicated horse power by
33000, to bring it to foot-pounds, and dividing the product by
the pitch of the propeller in feet multiplied by the number of
revolutions per minute, which is the speed of the propeller in
feet per minute.

If \( P_m \) = mean effective pressure on the piston in pounds
per square inch,

\( S \) = length of stroke in feet,

\( P \) = pitch of propeller in feet,

\( N \) = number of revolutions per minute,

Indicated thrust is \( \frac{P_m \times 2S}{P} \), or

\[
= \frac{I. H. P. \times 33000}{P \times N};
\]
and either of the above formulae, as may be most convenient in any given case, may be used in the calculations.

Having calculated the indicated thrusts for the trial speeds of the ship, a curve is constructed, with the speeds as abscissae, as in the indicated horse-power curves, but with the indicated thrusts as ordinates instead of the indicated horse power.

Fig. 5 is an example showing the construction of the indicated thrust curve. Vertical ordinates equal to the indicated thrusts at the respective speeds are set up, and a series of points, $a$, $b$, $c$, $d$, and $e$, are thus obtained, through which a fair curve is drawn.

The indicated thrust is composed of several elements, which may be thus enumerated.

1. Useful thrust, or ship's true resistance.
2. Augment of resistance, due to the action of the propeller in diminishing pressure under the stern of ship.

3. Equivalent of friction and resistance of the screw blades in their edgewise motion through the water.

4. Equivalent of friction, due to the dead weight of the working parts, piston packings, etc., which constitutes the initial or slow-speed friction of the engines.

5. Equivalent of friction of the engines, due to the working load.

6. Equivalent of the air-pump and feed-pump duty, etc.

It is probable that items 2, 3, and 5 are all very nearly proportional to the useful thrust. Item 6 is probably nearly proportional to the square of the number of revolutions, and thus, at the lower speeds at least, to the useful thrust. Item 4 probably remains constant at all speeds, and for convenience may be regarded as constant.

It will be observed, on drawing the thrust curves, that they do not descend to the thrust zero when the speed disappears, but tend to cross the vertical axis at some distance above it, representing a considerable amount of thrust at the zero of speed. This apparent thrust, when the speed is reduced to zero, and when, consequently, there can be no real thrust, will represent the real thrust equivalent to the initial or constant friction of the engines, and if a horizontal line be drawn through this point of intersection, the height thus cut off from the thrust ordinates would represent the deduction to be made from them in respect of the constant friction, and the remainder of the ordinates between the new base line would represent approximately the variations of thrust due to the ship's true resistance.
ADVANCE.

The advance of a ship is the distance which it progresses; its velocity is the distance advanced in one hour of time. The advance of a screw is the distance progressed in an axial direction; its slip is the difference between its actual advance and that due to its pitch multiplied by the number of revolutions which it makes. Thus, a screw of ten feet pitch, making ten revolutions, if working in a solid nut, would advance 100 feet; but working in water it may advance only 90 feet in the same number of revolutions. It is then said to have a slip of ten per cent.

SLIP.

Slip is the movement of the water acted upon by the propeller, which movement is opposite to that of the ship. The percentage of slip is usually computed according to the following expression:

\[
\text{Speed of screw} = \frac{85 \times 21 \times 60}{6080} = 17.60 \text{ knots.}
\]

\[
\text{Speed of ship} = 12.27 \quad "
\]

\[
\text{Slip of screw} = 5.33 \quad "
\]

\[
\text{Slip} = \frac{5.33 \times 100}{17.60} = 30 \text{ per cent.}
\]

It is important to distinguish between the apparent slip, which is the difference between the speed of the screw and that of the ship, and the true slip, which is the difference between the speed of the screw and that of the water relatively to the ship in which the screw is working.
In connection with his demonstration of the action of an oblique plane in passing through the water Dr. Froude said that later experiments show conclusively that the decrease in efficiency consequent on increased slip with screws of ordinary proportion is scarcely perceptible even when the slip ratio is as large as 30 per cent with screws working in undisturbed water. The results so shaped themselves as to point to the conclusion that for some reason or other the coefficient of surface friction began to diminish when the slip ratio became as much as 15 per cent, and near about halved when it was 30 per cent. It appears probable that with increasing slip a more or less pronounced eddy might become established at the back of the blades, so as to more or less completely neutralize the friction on that surface. An experiment was made by moving a plane obliquely through the water with various angles of slip, in a position where the effect could be observed; and in fact, when the angle between the plane and its line of motion was about 10°, the water at the back assumed the form of an eddy having nearly the speed of the plane, and overran the plane when the angle was increased to 15°.

Slip, as heretofore explained, is unavoidable, and is necessary to enable a screw to produce thrust with economy. Yet its abolition has been the aim of a large majority of persons, who have sought to make the screw a more efficient instrument of marine propulsion than it is.

It does not necessarily follow because a screw shows a large percentage of slip that it is inefficient. Some of the most wasteful screws exhibit very small slip. In fact, screws having a negative slip, that is, showing a greater advance than that due to the pitch multiplied by the number of revolutions,
are known to be very wasteful of power. A paper by Professor Rankine, in which he explains how the phenomenon of negative slip may occur, is given in the appendix.

But the author is of the opinion that in many instances where this phenomenon is supposed to have occurred it is the result of the application of erroneous data. A drawing of a screw propeller is made to have a certain pitch; a screw is made from it, and if the configuration is according to the drawing, that is, if the screw looks like the drawing, it is assumed to be correct, and the pitch called for is taken as a factor of the data in estimating its performance.

Owing to the peculiar shape of screw propellers, nothing can be known of the nature of their generation by their appearance. The only way by which to ascertain exactly what a screw propeller is with regard to its generatrix and pitch is to measure it after it is finished, and so find out. As the author has experienced in his practice, mistakes will occur through ignorance and accident. The shrinkage of the casting in cooling will cause it to warp and be distorted from the form of the pattern, and the extent of its departure from the drawing can only be ascertained by the actual measurement of the screw. By the system of measurement employed by the Bureau of Steam Engineering at the Washington navy yard many irregularities were discovered which would not otherwise have come to light.

Blades are frequently made for one pitch and set at another. In such a case the instrument, although a propeller, is not a screw. In instances of this kind the pitch is measured at different distances from the axis, and the mean of those pitches is taken as the pitch of the screw.

It is necessary to correlate the losses resulting from skin
friction and other elements of edgewise resistance inherent in propeller blades with the loss of slip.

The coefficient of frictional force per square foot as a unit of speed varies greatly with the length of plane in line of motion and with the quality of surface.

Regarding surface friction, Dr. Froude says that his "experiments for the admiralty show that it varies about as the power 1.85 or 1.9, but for convenience we may adhere to the usual expression that it varies as the square of the speed.

"A screw of 20 feet diameter, making 80 revolutions, will have the tips of the blades traveling at a speed of about 50 knots per hour.

"The resistance of a surface so short in the line of motion as a screw blade, even when its surface is quite smooth, is as much as 1 1/4 pounds per square foot at 10 knots, and is nearly as the square of the speed; and as each square foot or blade area involves 2 square feet of skin, the resistance of each is over 60 pounds. Thus, making some allowance for thickness and bluntness, there is involved in driving it at 50 knots at least 10 indicated horse power, and collectively the outermost foot of four such blades, each 3 feet wide, would absorb fully 120 indicated horse power in surface friction.

"The frictional coefficient adopted in calculating the resistance that the screws of the 'Iris' experienced was:

For screws Nos. 1 and 2...........0.00564
" " " 3..................0.00345
" " " 4.................0.00332

(See Trans. Inst. Nav. Arch. for 1879, and also extracts from the reports of the steam trials of the "Iris," in the appendix.)

The frictional resistance of screw propellers is always a
fruitful source of inefficiency. With a given screw the loss due to friction may be taken approximately as the square of the speed. This is not to say that the frictional resistance is greater in proportion to the thrust at high than at low speeds. The blades of screws should be as smooth and clean as possible, but for high-speed screws the absolute saving of friction may be considerable with an improvement of the surface. There is no permanent advantage in polishing the blades. No doubt there is some advantage for a little time, but the blades soon become rough, and shellfish and weeds appear to accumulate as rapidly on recently polished blades as on those with ordinary surfaces.

Dr. Froude made a theoretical investigation of the action of a plane passing through the water, and assumed its action as analogous to that of a screw propeller. From this investigation he made the following deductions:

"That the area that will drive the ship with a given slip ratio is directly as the ship's resistance and is inversely as the square of her speed; and since at moderate speeds a ship's resistance may be taken as proportional to the square of her speed, the same area of propeller will at all moderate speeds drive a given ship with the same slip ratio.

"To produce maximum efficiency, the propelling plane ought to stand at an angle of 45° with the line of the ship's motion, and this equally happens whatever be the coefficient of surface friction or normal pressure."

He further says that the salient conclusions to which we are led by this review of elementary principles, on which the efficiency of the screw propeller depends, are:

1. That a very much longer pitch than has been commonly adopted is favorable to efficiency.
THRUST, ADVANCE, SLIP, FRICTION, EFFICIENCY. 101

2. Instead of its being correct to regard a large amount of slip as proof of a waste of power, the opposite assertion is the true one. (Trans. Inst. Nav. Arch., 1878.)

The efficiency of a propeller is the proportion which the useful work performed by it in driving a vessel bears to the whole energy expended in moving it. In the absence of all friction and all action on the water, except that of driving the particles directly astern, the efficiency would be simply the ratio of the speed of the vessel to that of the propeller; that is to say, the speed of the vessel divided by the speed of the vessel plus the slip. But in every case there are resistances to overcome and transverse motions impressed on the water, which reduce the efficiency to a smaller fraction.

The efficiency of a propeller may be very much impaired if it be so placed and the vessel so shaped that the action of the propeller increases the resistance to the motion of the vessel. Such is the case when the lines of the after body of a screw vessel are too full for the action of the screw. This causes a diminution of the pressure of the water against the stern of the vessel, and has the same effect as an increase of pressure against the bow.

When a propeller lays hold of water that is already in motion through the action of the vessel, the change in pressure produced in the water by the action of the propeller on it is transmitted to some part of the vessel's bottom, and thus the resistance to the ship is altered.

This is what Dr. Froude called resistance augmentation, which, in some instances with single screws, he estimated to amount to from forty to fifty per cent of the natural resistance of the ship, or the resistance of the ship experienced when towed at the same speed. In a well formed single-screw
ship, except for the effect of the stern and rudder posts, the resistance augmentation due to the action of the screw is estimated by Dr. Froude to be generally from fifteen to eighteen per cent of the ship's natural resistance. In the "Iris" and other twin-screw ships of similar form the increase is only from ten to twelve per cent, and in some cases it has been found to be as low as eight per cent.

The alteration of resistance so produced constitutes a difference between the total thrust and the effective thrust. The effect is always to produce a waste of power when the propeller works in water that has been previously set in motion by the vessel.

The ratio borne by the useful work done in driving the vessel to the whole work done in moving the propeller gives the following results when friction is eliminated, and is applicable to all kinds of propelling instruments:

When the propeller works in previously still water, there is a loss of work simply proportional to the slip of the propeller, so that the efficiency is represented by the slip of the propeller divided by its speed.

In the case of screws or other oblique surfaces acting on water the loss stated above comprehends the effects of rotary or transverse as well as the backward slip of the water.

When a propeller works in water previously set in motion by the vessel, there is in the first case a loss of work proportional to the real slip of the propeller relatively to that of the moving water, and then a further loss of work proportional to the square of the previous velocity of the water.

When a screw works in water having a motion in the direction in which the screw is progressing, its effect is first to bring the water upon which it acts to a state of rest and then start it from a state of rest and move it in the opposite direction.
THE SUPPLY OF WATER TO A SCREW

When a screw is revolving while a vessel is made fast it operates simply as a pump. The water flows to it at its forward end and is discharged from its rear. The motion of the water flowing to the screw is the effect of its gravitation, and this effect sometimes extends for a considerable distance in advance of the screw. The water flows to the screw from all directions around its forward end—that is, from all around and in front of the circle described by the forward tips of the blades—but by far the greater part of this supply comes upward from below the axis of the screw, for the reason that the hydrostatic pressure is greatest below, and also because the eddies are less below than at or near the surface. This gravitation of the supply water toward the screw occurs also when the screw is advancing and when not counteracted by the frictional wake; but it diminishes as the velocity of advance increases, in consequence of the less time the force of gravity has in which to act.

When a screw revolves slowly, some of the supply water, doubtless, enters the screw at its periphery, especially at the bottom, because the water will be displaced at a less velocity than that due to its gravitation at that point. This falling upward, as it may be called, is a property possessed by fluids. But when the velocity of rotation is sufficient to enable the
screw to discharge a column at a velocity due to the gravitation of an equal mass, it is impossible for the supply to enter at any other place than the forward end.

When a screw revolves slowly, the motion of the water upon which it acts is, no doubt, influenced by gravitation as well as by the pressure of the screw blades, and the resultant direction while passing through the screw will be, to some extent, that of the least resistance, or toward the surface.

When revolving rapidly, the influence of gravitation is overcome, and its force is not exerted until the discharged water has left the screw.

During the dock trials of a double-hull vessel built to ply across the English Channel, which had its paddle wheel between the two hulls, it was observed with astonishment that pieces of wood thrown into the water between the hulls and in front of the revolving wheel did not readily enter the wheel, but would sometimes float about in front of it for a time, while pieces thrown into the water on the outside of the hulls disappeared quickly, and would reappear in the rear of the revolving wheel between the hulls.

The operation of the wheel caused currents to flow down on the outside of the hulls under the bottoms and upward between the hulls, to replace the water displaced by the paddle wheel (see Fig. 6).

No doubt part of this phenomenon is due to the water near the surface being more easily disturbed or thrown into eddies by the periphery of the paddle boards.

Isherwood observed, during his screw experiments at Mare Island, that with the vessel made fast "the surface water was absolutely quiescent; that it had no movement in any direction. The water supplying the screw came up from beneath
THE SUPPLY OF WATER TO A SCREW.  

in a vertical column. The depth of water at the wharf was very considerable, and it had a free movement between the bottom and the vessel's keel. An unbroken wave or elevation of water covered the screw during its action, the height of the

![Diagram]

Fig. 6.

wave varying, of course, with the rapidity of the rotation of the screw."

Fig. 7 serves to illustrate an instance in which a screw revolves without advancing, as in the case of a vessel made fast. Here the screw simply acts as a pump. If its revolutions are sufficiently rapid to preclude water entering at the periphery, the supply, water will enter at the forward end, and will have given to it by gravity a motion toward the screw, or, as some writers describe it, by the suction of the screw. The supply, water, as shown, is supposed to be divided into streams. These streams are represented as flowing toward the screw in an irregular converging channel. A preponderance of the streams composing the channel come from below the axis of the screw, indicating a greater velocity of the supply water from below the axis of the screw than from above. This is caused by the greater hydrostatic pressure that exists there, and by the water having freer access below than above.
In passing through the screw the streams are given a divergent direction and also a rotary motion, in consequence of the centrifugal action of the screw. The water is discharged with a divergent spiral motion, the mass resembling a twisted truncated cone, the density of which internally is less than it is externally. As the water is discharged divergently and occupies a greater bulk than when it enters the screw, and as the surrounding water into which it is discharged cannot be compressed, it takes up the necessary space by pushing back the surrounding water, just as an increase in bulk of an object in the atmosphere will push the air back. The water in being pushed back naturally seeks the direction of least resistance (the surface of the surrounding water), and thus produces the elevation of the surface, or wave, noticeable above a revolving screw.
THE SUPPLY OF WATER TO A SCREW.

In an account of the speed trial of the triple-screw U. S. S. "Columbia" the following is related: "The most noticeable feature of the trial of the ship itself was the remarkable absence of all wave. The triangular foaming cataract at the stern formed with its apex about ten feet from the ship and then subsided in height as it spread in width, until it disappeared fifty feet further aft into a series of gentle waves, similar to those which are seen in the wake of a stern-wheel steamboat."

In some instances where the screws revolve very rapidly, and the form of the after part of the vessel and position of the screws are favorable for it, the elevation resulting from the discharge of screws is excessive. In the case of the torpedo boat "Cushing" it attains a height of three or four feet above the normal surface.

When the immersion of the screw is sufficient, there may be little or no noticeable elevation of surface over it, because the discharged water will be widely dispersed and equilibrium will be restored before the surface can be affected to a sensible degree.

In Trans. Inst. Nav. Arch., 1879, Mr. Griffiths recites the results of experiments he was allowed to make on H. M. steam pinnace No. 22 in 1875 by measuring, with apparatus specially constructed for the purpose, the rate at which the water flowed through the screw disk while the boat was being towed. These experiments showed that over the bottom half the water was little interfered with, but at the top half of the disk the water was dragged with the boat to a certain extent, and only flowed through the screw at about half the speed of the boat's progress.

When a ship is being driven by a screw propeller, the con-
ditions are somewhat different from those which prevail when the screw revolves while the ship is made fast; yet there must be a greater quantity of water passing through the lower than through the upper half of the screw's disk, if the varying velocity of the frictional wake, which always follows an advancing vessel to a greater or less extent, is what it is generally considered to be; that is, if the velocity of the wake is greatest at the surface of the water and diminishes with its depth.

In cases like the foregoing there must, of necessity, be less water passing through the upper half of the disk than through the lower half, because while the water moves in the same direction as the screw, the action of the latter is to first absorb this forward motion while the water is passing through the screw and then to give it motion in the opposite direction. When this forward motion is barely absorbed, and the water is discharged from the propeller in the condition of water undisturbed by the advance of the vessel (still water), there will be no slip in that portion of the screw. If all the forward motion of the wake is not absorbed by the screw, then there will exist a negative slip in that portion of the screw through which such water passes.

Suppose an advancing vessel to have a wake following her in which the screw works, of a certain velocity at the surface or at the top of the screw, and this velocity to diminish uniformly to nothing at the bottom of the screw, where the water is supposed to remain undisturbed by the vessel; suppose the velocity of this wake to be such that in passing through the upper half of the screw's disk it has the forward motion absorbed and a motion in the opposite direction given to it
that will have a mean velocity when discharged by the screw of four feet per second. Now the mean velocity of the wake which passes through the lower half of the screw's disk being less than that which passes through the upper half, there will be less forward motion of the wake to be absorbed while passing through the screw, and consequently the mean velocity of the water discharged from the lower half of the screw will be greater than that discharged from the upper half.

If the mean velocity of the water discharged by the lower half of the screw is eight feet per second, then the quantity discharged by the lower half must necessarily be double that discharged by the upper half. Hence, notwithstanding the wake offers a greater initial resistance in the upper half of the screw's disk, the thrust or pressure on the screw may be much greater in the lower than in the upper half, in consequence of the greater quantity of water there discharged in a given time.

The author believes the foregoing to be the actual condition that obtains in practice when a screw is driving a vessel and working in a following wake. One of the reasons for his expressed opinion is that a preponderance of the supply water flowing to a screw comes from below its axis, since, if a greater quantity of water is discharged there, a greater quantity must there be supplied.

When a screw is advancing and a frictional wake is following the vessel, the supply water is no doubt influenced by the screw before the latter meets with it, but to what extent this influence goes it is impossible to determine, especially in the case of a fast vessel. Although the forward velocity of that part of the frictional wake which supplies the screw may
be reduced by its "suction," it is very doubtful in the case of a fast vessel whether there is any running of such water toward the screw.

Screw propellers have been placed at the bows of vessels, in which position there is freer access of the supply water to them, and where their performance is not affected by the ship's wake, as when placed at the stern, but owing to their disturbance of the stream-line action of the water at the bow and the production of eddies, or from other causes not understood, they have not proved advantageous in practice, except in special or exceptional cases, as, for instance, in double-end ferryboats. Screws projecting below the keel, with no shoe or other obstacle to impede the flow of supply water from that direction, and twin screws, which are generally obstructed but little below, prove the most efficient.

Professor Cotterill, in a paper on the changes of level in the surface of water surrounding a vessel produced by the action of a propeller and skin friction (Trans. Inst., Nav. Arch., 1887), says: "It appears that the augmentation of the resistance of a vessel by the action of a propeller is of two distinct kinds. The first is due to the disturbance of the wave system of the vessel by throwing additional water into a state of wave motion or by the direct action of the race. This is probably small at small speeds, when the wave resistance is small, and only becomes important at speeds at which the wave resistance is the ruling element of the resistance of the vessel. It occurs in all propellers alike. The second is due to a propeller at the stern of a ship accelerating the water which supplies it before entering the propeller. This occurs principally in screw propellers, and is then due to suction at the back of
the blades and to the rotation of the race. In a Whitehead torpedo it is the sole cause of augmentation, because in a deeply submerged body there is no wave resistance. It consists in a lowering of the pressure in the whole mass of water lying between the screw and the vessel, thus augmenting the resistance of the vessel and the thrust of the screw. The two effects are, perhaps, nearly equal. It is by far the most important part of the total augmentation of resistance at low speeds, or when the supply of water to the screw is impeded. Any augmentation of thrust in the screw not represented by sternward momentum, and the corresponding augmentation of the turning couple on the screw, is employed in accelerating the rotation of the race."

An expedient often adopted with stern-wheel steamboats when difficulty is experienced in passing over a bar in the usual way is to turn the boat around and proceed stern foremost. When advancing bow foremost, the supply water to the wheel comes chiefly from below and under the stern of the boat; the pressure of the water on that part of the boat is thereby reduced, and, consequently, the stern settles deeper in the water. When the boat advances stern foremost, the wheel being reversed, the supply water is received from the opposite direction before and below the wheel, and the water, when discharged by the wheel, is forced against the stern, where it produces greater pressure. The result is a tendency to elevate the stern and decrease the draught of the boat at that point.

One of the most singular circumstances connected with marine propulsion, and for which up to the present time no generally accepted explanation has been given, is the retard-
ing influence that shallow water has on the speed of a vessel.

This phenomenon is no fancy, but a fact established by indubitable evidence in numerous instances. It is now generally recognized that higher speeds can best be attained in deep water, and interested parties insist that speed trials shall take place in such water. If the phenomenon was confined to depths of water but a few feet below the keel of the vessel, a plausible explanation might easily be given, but such is not the case. There is evidence that this influence extends to depths three and four times the draught of the vessel, and the higher the speed the more pronounced becomes the retarding influence of shallow water.

Mr. Edwin Cramp, in an account of the official trial of the U. S. S. "Columbia," says: "When at maximum speed there was little or no vibration of the hull, except when passing over shoal places, when the engines would be slowed down and a panting, leaping motion would become apparent, as if the ship was being held back and was striving to break its bonds."

The slowing down of the engines would clearly indicate that the ship was meeting with greater resistance, rather than that the screws were not performing with regularity and were not receiving their usual supply of water, because if the supply of water to the screws was diminished while the resistance to the ship remained the same the engines would increase instead of diminish their revolutions.

If the resistance to a moving ship is increased as the water in which it is moving decreases in depth, which condition is now conceded to be a fact, it is fair to presume that the bottom on which the water rests is the cause of this increase of
resistance, and that the water intervening between the bottom and the ship is the medium through which this resistance is transmitted. According to this view, the frictional wake must extend for a considerable distance below the bottom of the ship, as this retarding effect is believed to extend to depths several times greater than the draught of the ship.

The water below a ship may be supposed to be divided into a series of layers. The ship's movement produces a corresponding motion in the layer of water immediately adjacent to the ship's skin; this layer produces motion in the layer immediately below it, and so on, as far as the wake-motion extends, each succeeding layer having a uniformly reduced motion, until it becomes nil (when the water is of sufficient depth).

If the bottom is reached before this motion ceases, the layer immediately in contact with the bottom is dragged to some extent over it and thereby has its motion retarded. This retardation is communicated to the layer immediately above it, and this again retards the layer immediately above it, and so on, until the retarding influence of the bottom reaches the ship. Thus the ship's resistance is increased, because the movement of the ship is the primary cause of the motion of the water.

This retardation caused by the bottom is analogous to that of a flowing stream. It is well known that the velocity of the current of a stream is least below and along its banks, where it is retarded by the bed and banks. From the bottom upward the velocity of flow gradually increases, the greatest velocity being at the surface in mid-stream.

In the case of a flowing stream, gravitation is the force
which produces the motion of the water; but in the case of the frictional wake the cause of its motion is the movement of the ship, against the surface of which the particles of water rub and have given to them, by reason of this friction, a whirling and advancing motion in the same direction in which the ship is advancing. The wake, in acquiring this motion, also absorbs a considerable amount of energy from the ship; in fact, it is said to be the principal source to which the energy produced in the ship is imparted.

When a ship is moving through water of sufficient depth it meets with a certain amount of resistance, resulting principally from the motion which it produces in the mass of water through which that motion extends, and this requires the expenditure of a certain amount of the energy produced in the ship. When a shoal is encountered, the lower part of the wake is dragged over the bottom, the resistance resulting therefrom being transmitted through the water to the ship, and hence the ship is either retarded, or else more energy must be expended in order to maintain a uniform speed.

On the subject of the increase in resistance to a ship in shallow water Mr. D. W. Taylor says:

"According to the trochoidal theory of waves, the orbits of the particles cease to be circular in shoal water, and become elliptical, the eccentricity of the ellipse increasing with the shoalness of the water; and other changes—in period, etc.—take place.

"At depths where a trochoidal wave of such length as to travel at a given speed is practically unchanged from the deep-water wave at the same speed, the resistance of a ship at that speed will not be affected."
"The following depths of water are appropriate to various speeds:

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<th>Speed of ship in knots</th>
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"The increase of resistance due to shoal water appears liable to exaggeration. Mr. R. E. Froude has stated (Trans. Inst. Nav. Arch., 1892) that for a 5000-ton ship at speeds no greater than 17 knots, in water of 7 fathoms depth, the increase in resistance above that in deep water is but some 3 or 4 per cent."
THE POSITION OF THE SCREW.

No great revolution in marine propulsion can be accomplished by any modification of the instruments now in use for the purpose.

There are many screw propellers in use at this time which are as efficient as any means of marine propulsion can possibly be made. One great desideratum necessary for the greatest efficiency is a favorable position of the screw for allowing the freest access of water to it. Another is that the screw should be well immersed; that is, its upper edge should always be well covered with water. It is better to use a smaller diameter and have the screw well immersed than to use a larger diameter and have the blades reach the surface of the water, because with the larger diameter working near the surface the water is more easily broken up into eddies, and the loose water and air in passing through does not offer as much resistance to the screw as the water below; and, moreover, screws well immersed are much steadier in operation.

Mr. Barnaby, in his "Marine Propellers," recites:

"The speed with which water can follow up the blades of a screw depends on the head of water over it; but if the immersion is sufficient to exclude air, a head equivalent to 30 feet
is supplied by the atmosphere. This is a fact which was clearly pointed out by Professor Reynolds.

"As a result of a change from a diameter of 5 feet 10 inches to 4 feet 6 inches, the speed of a first class torpedo boat was increased from eighteen to twenty knots, other conditions remaining the same. This is a most important feature in a screw. It is a common thing to see screw steamers in their light condition with the tips of the propeller blades two or three feet above the surface. This implies an immense loss of efficiency."

The configuration of a propeller does not affect the direction of the water it discharges; but it may affect the quantity of water the propeller is able to deal with, especially when situated in an aperture in the stern of a vessel.

The position of a screw propeller with reference to its surroundings has a great deal to do with its economical performance. For this reason twin screws are more efficient.

In small high-speed steam launches and torpedo boats the screw has been placed to project below the keel, and also placed outside the rudder, with advantage; but this is impracticable in very large vessels.

The veteran experimentalist on screw propulsion, Mr. Robert Griffiths, in a paper read before the Institute of Naval Architects in 1879, gives an account of experiments conducted by him in order to determine the best position in which to place a screw when situated in a stern aperture, and he deduced therefrom that the screw should be placed at a distance of not less than two thirds of its diameter behind the stern-post. He further says: "As the ship moves forward the water closes in from each side of the stern to fill the space she occupied. When the screw is in operation it sucks away the
eddy water which accumulates at the top part of its disk, chiefly owing to the fulness of the run from the level of the screw shaft to the top of the screw, where water would follow the ship if the screw did not draw it away; by this the thrust given to the screw shaft is increased, which increases the resistance of the ship to nearly twice what it would have been had the screw not been in operation. The only remedy that I have been able to discover is to place the screw as far back from the tapered or wedge part of the stern as will allow the eddy water to remain there, for which purpose the screw must be placed about two thirds its diameter behind the stern-post, so that the currents which flow in on each side of the deadwood will, by being drawn by the action of the screw, meet in front of it and supply it with water to act upon."

It seems to be well established that a moving ship carries along with it (owing to the friction of the water on the skin) the water for some distance from the point of contact, and that the velocity of this water gradually diminishes as its distance from the ship increases, until its condition is the same as that outside the ship's influence. While the water is thus moving it exerts a certain pressure on the hull. Now, if the water thus following the ship is drawn away at any part, there is a disturbance of this pressure and a diminution at that part.

The effect of the action of a screw propeller when placed close to the ship at the stern is to draw away the water from that part of the ship, or, more correctly speaking, to cause it to flow to the screw to supply it, and the forward pressure is thereby diminished and the resistance to the ship augmented. Mr. Froude estimated the augmentation of resistance in single-screw ships from the action of the screw to
amount, in some cases, to 40 or 50 per cent of the ship's resistance, but in well-formed twin-screw ships it is much less, in some cases as low as 10 per cent.

It appears from the foregoing elucidation that it is advantageous to place the screw at a sufficient distance from the ship to annihilate the augmentation of resistance by its action. It is very evident, however, that there are important practical considerations involved in the location of screws, especially twin screws, that to a great extent determine their location.

As evidence that there is a mass of water flowing along with the ship, Mr. E. J. Harland in 1879 recited his experience in regard to the application of a novel speed indicator. A part of the apparatus consisted of a hollow sword, with a hole near its lower end. This sword passed through a stuffing box in the bottom of the ship, with the hole toward the bow. A pressure gauge communicated with the hollow in the sword. When the ship was under way the pressure resulting from the sword passing through the water operated through the hole and was communicated to the pressure gauge, the speed of the ship, by an ingenious calculation, being thereby determined. It was intended to be a continuous speed indicator, by which the speed of the ship could be read off in any part of the vessel where the indicator might happen to be placed, but an unexpected circumstance befell the device. The sword was first made to project two feet below the bottom of the ship, where the speed indicated was only about half that of the speed of the vessel. The sword was then lengthened to more than 3 feet and was tried with the hole at different distances from the skin of the ship; but even at 3 feet 6 inches the speed indicated was far below what the
ship was actually traveling, thus showing at that part of the
ship, which was 100 feet from the bow in a ship 450 feet long,
that there was surface water following fully 3 feet in thickness.
The conclusion ultimately reached was that the water imme-
diately next to the skin of the ship may be traveling almost at
the speed of the ship, and that the speed of the water dimin-
ishes as the distance from the skin increases; but even at 3
feet 6 inches from the ship the water could not be said to be
undisturbed by the ship's motion.

In very fine ships the extent of outside shafting, with the
necessary supports, becomes a serious consideration, in conse-
quence of the increase in resistance which the supports cause.

In order to reduce this objection to a minimum, an ar-
range ment of twin screws with overlapping disks was intro-
duced by Messrs. Rankine and Blackmore in 1876. One screw
was set in advance of the other, the blade tips passing through
an aperture in the deadwood. Figs. 8 and 9 show twin
screws as thus fitted in the "Buzzard," a small coasting
steamer. The arrangement has proved successful, and was
adopted by Messrs. Harland & Wolff in the steamships "Teu-
tonic" and "Majestic." The screws of the "Teutonic" are
19 feet 6 inches in diameter, and the distance between the
shafts is 16 feet; they are right and left handed and turn out-
ward. A twin-screw torpedo boat built by Mr. Normand,
with overlapping screws, has both screws arranged to turn the
same way. The water thrown up by the ascending blade of
one screw is met by the descending blade of the other, and the
slip is thus reduced. When thus arranged, it is found that the
aftermost screw turns slower than the foremost one, the con-
trary being the case when they turn in opposite directions.

Multiple screws are being introduced. The U. S. S. "Co-
"Columbia" and "Minneapolis" are fitted with three screws each, the centre one having a greater immersion than the side screws. This is believed to be a better arrangement for triple screws than that of having them all of the same immersion.

The "Columbia" and "Minneapolis" have proved very successful on trial, the results of which are given elsewhere.

Several advantages accrue from the adoption of triple screws in war ships of high speed. It is possible by this
means to affect a reduction in the weight of the machinery, since a high rotative velocity is admissible. There will also be a saving of fuel when cruising at low speed by using the centre screw alone.

At the thirty-fourth session of the Inst. Nav. Arch., 1893, Mr. George Calvert read a paper on the investigation of the relative speeds of the water in different parts of the ship’s wake. He described experiments that he had made during several years with specially prepared apparatus, with a view to ascertain the actual condition of the water around the stern of a moving ship and the nature of the currents set up in consequence of the ship’s motion.

He first experimented with a, full-sized ship 260 feet in length, and afterward with a model one ninth of the lineal dimensions of the ship. The deductions that he drew from the results were confirmatory of the general opinion which had previously existed, viz., that the frictional wake is greatest at the surface of the water and immediately in contact with the ship, and that it diminishes thence with the depth and distance from the skin.

Mr. Calvert prepared a flat plank, 28 feet long, in the direction of its motion, and fitted it with measuring appliances, in order to ascertain (1) the maximum forward speed induced in the water by the passage of the plank through it at various velocities up to 460 feet per minute; (2) the distance from the surface to which the disturbance of the water extends, and (3) the speeds at intermediate distances.

The first series of experiments was made in order to ascertain the velocity of the frictional wake in contact with the surface of the plank, which had been coated with black-varnish. The speeds recorded at 1 foot, 7 feet, 14 feet, 21
THE POSITION OF THE SCREW.

feet, and 28 feet from the leading end were, respectively, 16, 37, 45, 48, and 50 per cent of the velocity of the plank. Other experiments were made with this plank to show the manner in which the motion of the water in contact with the surface of the plank was gradually imparted to layers of water lying underneath. The results of these experiments at 200, 300, and 400 feet per minute indicated that the velocity decreases in geometrical progression as the distance from the surface increases in arithmetical progression. Thus, at a distance of 28 feet from the bow end, with a quality of surface similar to that of the plank, the forward velocity of the water in contact with that surface being 50 per cent of the velocity of the surface at a distance of $\frac{1}{6}$ inch therefrom, it is reduced to one half, or 25 per cent; at $1\frac{1}{4}$ inches distance, to one fourth, or $12\frac{1}{2}$ per cent, and so on; so that at a distance of about 5 or 6 inches from the surface the water remains practically undisturbed. The retardation of velocity is in the somewhat analogous condition of orbital wave motion, and the flow of rivers.

An interesting mathematical investigation of the thickness of the frictional wake will be found in the report of the British Association for 1874.

As regards the total value of the frictional wake, there is, perhaps, little chance of mistake, as our knowledge of frictional resistance (due principally to Dr. Froude's researches) is fairly complete; and the connection between that form of resistance and wake motion is based on "the universal law connecting force and momentum."

Any inclination of the axis of the propeller with the line of the advance of the vessel is a disturbing element as to the proper performance of the screw, for its effect is equivalent
to a distortion of the pitch, and is also a cause of vibration. If the axis of the screw be placed perpendicular to the line of advance, the pitch of the screw, so far as it relates to advancing the vessel, would be nil.

Twin screws turning outward, as they are usually made to do, in order to avoid the risk of floating objects becoming jammed between the upper blade and the ship’s counter, should cause least vibration when the shafts are slightly inclined upward and outward, starting from the stern; those, on the contrary, which turn inward would work most favorably with the shafts inclined downward and inward. Probably Mr. Thornycroft was the first to draw attention to the injurious effect of an excessive vertical inclination of shaft, and also to the possibility of neutralizing the action of the wake by inclining the shaft horizontally in single-screw vessels.

Mr. D. W. Taylor, U. S. N., has compiled a table, from which the necessary inclination may be obtained, depending upon the pitch and slip of the screw. (See Journal of the American Society of Naval Engineers, Vol. III (1891), page 12.)

With twin screws the clearance allowed between the tips and the hull should be not less than one inch for each foot in diameter of the screw.
CAUSES OF VIBRATION.

The vibration resulting from the operation of screw propellers is the effect of several causes. When the blades of a two-bladed screw are in line with the stern post, the latter interferes with the water that supplies the screw, and it experiences less resistance at this moment with a resulting shock. Isherwood found in towing a vessel with a two-bladed screw revolving freely that there was a decided retardation in the velocity of the screw's revolution whenever it became masked by the stern post. This is also true to some extent with twin screws when placed close behind a bracket. Vibrations and shocks occur when screws work with varying immersion, as when the vessel is pitching or rolling. Screws are seldom tested to ascertain whether they are balanced. There is no doubt that in many instances they are considerably out of balance, in which case, when revolved, such condition will cause vibration. Especially will this effect be made manifest when the screw is revolved at a high velocity. The breaking of a blade throws the screw out of balance, and the effect when the ship is under way is immediately felt. As previously stated, any inclination of the axis of a screw propeller with the line of advance of a ship will have a tendency to cause vibration.

The "Griffiths" configuration, or its modified form, is the
one most generally employed in propellers. By having its width of blades diminished toward their tips the centres of pressure on them are brought nearer to the centre of the shaft than if the edges of the blades were radial lines, and for the same reason when in a stern aperture it interferes less with the action of the rudder.

This can be understood by reference to the cut (Fig. 10). Let the radial lines represent the outlines of a two-bladed screw, in which the area of each blade is 1000 square inches and the depths of immersion of the centres of pressure of the blades are for the upper one 4.5 feet, and for the lower 13.5 feet. Assuming a column of water of 2.25 feet to equal one pound pressure, then the upper blade will be opposed by a pressure of $1000 \times 2 = 2000$ pounds, and the lower one by $1000 \times 6 = 6000$ pounds. It is evident that there will exist from this condition a tendency to twist the stern (athwart-ship thrust) in an opposite direction to that in which the lower blade is moving.
CAUSES OF VIBRATION.

Let the dotted lines in Fig. 10 represent blades of the "Griffiths" form of the same area as those of radial outline, but whose centres of pressure are for the upper blade 6.75 feet, and for the lower one 11.25 feet. Then the pressure will be, for the upper blade $1000 \times 3 = 3000$ pounds, and for the lower one $1000 \times 5 = 5000$ pounds, the same aggregate (8000 pounds) as in the former case; but there will be less twisting action, because the centres of pressure are nearer to the centre of the shaft, and there will also be less difference in those pressures. The nearer the centres of pressure are to the axis of the screw the less will become the twisting moment.

When a screw revolves without advancing, there is no doubt it experiences its greatest resistance below its axis, because the hydrostatic pressure is greatest there. When a screw is driving a vessel the conditions are different, and whether the screw will then experience greater resistance above or below the axis will depend on the condition of the water which supplies it when the screw comes in contact with it.

If a vessel with a right screw is arranged to swing freely while the bow is made fast, the stern will move to starboard when the screw revolves for going ahead, and to port when the motion of the screw is reversed.

It appears from experiments on the steering qualities of some ships that the action of a right-handed screw is to throw her head to starboard. This would indicate that the tail shaft was bearing hardest on the port side of its bearing, caused by the screw experiencing greater resistance above than below its axis. This effect is attributed to the screw working in a frictional wake which has its greatest forward velocity at the surface of the water. The fact is, that but little is known of
the condition of the frictional wake. This lack of knowledge is one of the reasons which have made the problems of propulsion so difficult of solution. The only experiments with which the author is acquainted having for their object the investigation of the relative speeds of the water in different parts of the ship’s wake, the actual condition of the water around a ship’s stern, and the nature of the currents set up in consequence of a ship’s motion, are those of Mr. George Calvert, mentioned elsewhere.

Professor Reynolds observes that when a vessel is going ahead the onward motion of the wake is very different at the surface and the keel. He agrees with Professor Rankine that the mean speed in a fine-lined vessel may be ten per cent of the vessel's speed, but believes it varies from twenty per cent at the surface to nothing at the keel, and the upper blade experiences greater resistance than the lower from the effect of the wake.

The speed of the wake depends on the nature and extent of the surface rather than on the form of the vessel. When the screw works in the wake, it is able to recover some of the energy which has been expended by the ship in giving it motion.

The effect on the screw of increased hydrostatic pressure consequent upon increased depth of immersion was demonstrated by George Rennie in 1856 (Trans. Inst. Nav. Arch., 1878).

A screw of 1 foot 9 inches diameter and 346.3 square inches disk area was arranged to be immersed and rotated at various depths. A constant number of revolutions was given the screw during the various trials. The thrust was accurately measured for each immersion, and was as follows:
7 lbs. per square inch with screw just immersed;
28 lbs. per square inch with 6 inches of water over the screw;
56 lbs. per square inch with 54 inches of water over the screw.

The following is an instance in which the result is similar to that of the foregoing. During a shop trial of a launch engine, by way of experiment it was loaded as follows: A wooden tank was prepared with a stuffing box fixed on one side. The engine shaft passed through the stuffing-box, and a screw propeller was fitted on the end of the shaft in the tank. When the engine was running it was found that the load could be varied by increasing or decreasing the depth of water in the tank. As the depth of immersion was made greater the hydrostatic pressure on the screw was increased, and therefore more power was required to displace the water.

Increase of immersion of a screw tends to make its action more steady and also to increase its efficiency. The deeper its immersion the greater the fluid pressure on the screw. When water under greater pressure is displaced it is equivalent to an increase in the weight of the water displaced. This is one of the advantages that immersed screw propellers possess over the paddle-wheel.
FORMS AND DIMENSIONS OF SCREW PROPELLERS.

There is no problem in marine engineering in regard to which the opinions and practice of leading designers differ to a greater degree than in that of the screw propeller. But it can scarcely be disputed by any practical man that the true secret of success lies in the correct proportion of the three cardinal elements—diameter, pitch, and blade surface—to each other in such a manner as to give the best possible balance to the engine power, and that the shape of the blades (within certain limits), their thickness, and the condition of their surfaces are of minor importance.

The above is, briefly, what an English designer of large experience wrote some years ago, and the words may be truthfully repeated to-day with regard to the practice in this country.

Two theories of the action of a screw propeller have been advanced, and by the aid of the deductions therefrom it has been thought possible to design the most suitable screw for a given case.

The earlier of the two is the blade theory, which is based on the assumption that the action of each blade is independent of the other and that the thrust is proportional to its area. Dr. Froude followed this theory in his demonstration of the action of a plane moving obliquely through the water.
The blade theory, as expounded by Professor Cotterill (Trans. Inst. Nav. Arch., 1879), is founded on the development of the action of a small flat plate moving through the water in a spiral path.

On the other hand, the disk theory was propounded by Professor Rankine (Trans. Inst. Nav. Arch., 1865). According to this theory, the water is supposed to be sent back in a column consisting of cylindrical rings sliding and rotating within one another, all having the same velocity.

Both the disk and the blade theories require considerable modification in their practical application.

Experiments with models have also been made, with a view to constructing formulae by which screws of large diameter might be designed. But it seems that little practical benefit can be derived from the data obtained from the experiments with small models of screws. In this connection the elder Froude asserted that the results of experiments on small models of screws are likely to be of less value, or, at any rate, less directly trustworthy, than experiments on models showing the resistance of ships.

It is proper here to state that two high authorities on the subject of screw propulsion (Messrs. R. E. Froude and S. W. Barnaby) place a high estimate as to the value of experiments on model screws.

While it is true that no formula exists by which the proportions of a screw may be determined which will give the best result in any given case, much has been learned by experience during the last decade or thereabout. Practice has furnished examples so numerous, varied, and accessible that there is little difficulty in selecting a vessel whose performance has been satisfactory, and which can be used as a model;
and by the method of comparison the proportions of a screw may be arrived at which, if not absolutely the best for the intended vessel, will closely approximate thereto.

The diameter of a screw propeller is the diameter of the circle swept by the tips of the blades.

The pitch is the axial distance between the same thread at one convolution.

The disk area is the area of the circle described by the tips of the blades less the area of the hub. (The area of the hub is included by some authorities.)

The developed or helicoidal area is the actual area of the working face of the blades when laid out on a plane.

The projected area is the area of the projections of the blades on a plane generally taken at right angles to the axis of the screw.

The length of a screw is the length of a blade measured parallel with the axis.

The speed of a screw is the pitch multiplied by the number of revolutions.

The pitch ratio is the quotient resulting from dividing the pitch by the diameter.

The slip ratio is the quotient resulting from dividing the speed of the screw by the speed of the ship.

A screw propeller is called a true screw when its pitch is uniform, and an expanding-pitch screw when the pitch increases from the forward to the after edge.

Sometimes the pitch is made to vary radially, and again the pitch may be made to vary both radially and axially.

When the pitch is uniform and the generatrix is a right line at right angles to the axis, the instrument is a right screw.

A modified Griffiths screw propeller is one having blades
of any curved shape diminishing in width toward the tips and having their broadest part nearer the boss than the tips.

The mass of water or section of column acted upon by a screw propeller in one revolution is the volume generated by the propeller when revolving in a fixed position, less the volume of the instrument itself; i. e., the boss and blades.

Designers of large experience have, generally, their own particular ideas as to the proper proportions of screws. These ideas are, of course, acquired by experience, and from them an empirical rule is constructed, the application of which will, in most cases, give fairly good results, although they may not be in all respects the best possible.

Some designers consider diameter the first important element to be determined; next, pitch, and lastly blade area. Others reverse this order.

It has often been proved in practice that a screw can be made too large in diameter as well as too small.

The reason given for considering the diameter of leading importance is that any change in it will affect the result in a greater degree than an equal change in the pitch or the blade surface.

Some designers proportion the diameter of the screw according to the immersed midship section.

One successful steamship builder in this country made the disk area of the screw for single screws 33 to 36 per cent of the immersed midship section of the vessel, and the aggregate area for twin screws from 42 to 46 per cent. The pitch was obtained by dividing the required speed by the number of revolutions, allowing 15 per cent for slip. The blade or helicoidal surface was made 40 to 42 per cent of the disk area of
the screw. The configuration was the modified Griffiths type.

The most suitable diameter of screw for a given case is the most difficult as well as the most important factor to determine. Mr. Thornycroft has remarked, regarding the diameter, "How little we know about it."

A high speed of revolution will permit of the diameter of screws being reduced and placed lower, thereby securing the important object of greater "head," and will also admit of the driving machinery being made lighter.

A fast vessel should have rapidly revolving screws. In designing the machinery for such a vessel it is therefore desirable to fix upon the highest practical speed of revolution.

By some the pitch is determined, first, by dividing the desired speed by the number of revolutions, allowing for slip from the quotient thus obtained; the diameter of the screw is then decided by making the ratio of pitch to diameter from 1.25 to 1.65. The blade surface is made about 35 per cent of the disk area of the screw.

The percentage of slip now allowed is from 16 to 20 in large screws for good-sized ships, in which the engines make from 115 to 150 revolutions per minute.

The type of vessel and the character of the work that she will have to perform govern, in great measure, the number of blades that will be most usefully employed and the amount of blade surface that a screw propeller should have.

Isherwood deduced from his Mare Island experiments, "in the case of screws having the same kind and quantity of surface," that "their propelling efficiency in smooth water is not affected by either the number or the position of the blades." It appears certain that the fewer the blades that will insure
steadiness of action the more satisfactory will be the operation of the screw, because the propelling efficiency is not impaired by a reduced number of blades, while it can be made lighter and less expensive to construct.

In practice, the limit of the number of blades is reached when four are used. Six and even more blades have frequently been tried, and invariably an inferior performance has been the result.

Steadiness in operation is affected by the number of blades, and should, generally speaking, determine the number.

Situated in a stern aperture, four blades give the greatest satisfaction, especially where the vessel traverses rough water. Formerly screws with two blades were more extensively used than now, but owing to the vibration that they cause and the tendency of modern practice to higher rotative velocities and increase of thrust their use has become limited, although they are efficient instruments so far as propulsion only is concerned, as they absorb less power in surface friction and form resistance than screws with a greater number of blades.

Situated as twin screws are where the supply water is not so much interfered with as at the stern, and where the immersion can generally be made more ample, in consequence of reduced diameter, three-bladed screws are preferable, and are now more extensively employed than any other type.

It is sometimes impossible, owing to restricted draught or other cause, to obtain sufficient blade area with three blades and to have the screw well immersed without recourse to an undesirable length of screw. In such cases four blades are used.

An idea prevails, even with some reputable authorities on screw propulsion, that in respect to the displacement of
water the blades act independently of each other; that is, if one blade of a given area discharges a given quantity of water, another blade added thereto will discharge double the quantity in the same time, three blades treble the quantity, and so on. If such be true, the quantity of water displaced should increase with the number of blades. It would be difficult to explain why four should be the limit in practice, but the author has previously explained that as more blades are added the disk area is reduced for the passage of water.

Regarding the amount of blade surface that should be given to a propeller, Mr. A. E. Seaton contends for what seems to be the preponderance of opinion, namely, that a propeller giving the best results in smooth water will not, necessarily, give the best results in rough water, and that propellers for ships traversing rough water and subject to head winds should have more blade surface than would be necessary if the ships ply only in comparatively smooth water.

Experiments have repeatedly been made with the view of ascertaining the effect of removing two of the four blades of a screw. In Trans. Inst. Nav. Arch., 1888, Mr. Andreae gives the results of such experiments, where a four-bladed screw, the disk area of which was 33 per cent of the immersed midship section and whose blade area was 47 per cent of the disk area, was first tried with four blades. The two opposite blades were then removed, which reduced the blade area to 23.5 per cent of the disk area. There was found to be a gain of 10 per cent in the number of revolutions in the case of the screw with two blades, and at some of the speeds tried it proved to be the more efficient.

About the same results were obtained in the memorable trials of the "Iris," where the two blades of No. 1 screws
proved more efficient than the same screws with four blades. Much speculation and computation were indulged in by those who maintained that the blade area should be divided between the number of blades, regardless of the length of the screw, to account for this result with the two-bladed screws of the "Iris." With the screws having four blades the ship was driven 15.123 knots with 5251 I. H. P.; but when the two alternate blades of each screw were removed, the ship was driven 15.726 knots with 4368 I. H. P. An attempt was made to prove that the increase of power was absorbed in the surface friction of the two extra blades, but, after much ingenious figuring the advocates of the theory above referred to were compelled to admit that the friction of the two extra blades could not account for all the difference in results which the experiments had elicited.

A screw having two blades or threads when in operation cuts slices of water, so to speak, of a certain thickness. If two more blades are added, making it a four-thread screw, these slices of water are reduced to one half their former thickness less the thickness of the blades. Consequently, less water is acted upon, and there is a greater interference of the blades with each other in obtaining dense water. This interference or disturbance of water by the blades is aggravated as the pitch of the screw becomes less, because the slices then become thinner.

Practice varies as to the pitch ratio of screw propellers. Pitch, like other features of screw propellers, should be governed to a great extent by the type of vessel and the character of the service it will have to perform. For torpedo boats, yachts, and other high-speed light-draught boats, which usually run in comparatively smooth water, and where high
rotative velocities prevail, diameters being restricted in consequence of small draught, the pitch of screws varies from about 1.4 to double the diameter. For the larger high-speed ocean vessels, which traverse water more or less rough, and with which ample diameters and immersions of the screws can be obtained, the pitch will vary from about 1.2 to 1.6 the diameter. For the slower-speed vessels, such as cargo steamers, towboats, etc., smaller pitch ratios prevail, ranging from about 1.1 to 1.3 the diameter.

As to the distribution of surface, which is determined by the configuration or shape of the blades, there is not so great a diversity in present practice as formerly existed.

The configuration of a screw propeller is of less consequence than either of the other important elements—diameter and pitch. It has little influence on the direction in which the water is displaced, because the direction in which the water is discharged depends on the character of the generatrix or form of the acting surface; but it may influence the quantity of supply water to the screw, according to the position it occupies in reference to the hull. With single screws in a stern aperture it has been found advantageous to shape the blades so as to give a good clearance between the stern post and the screw for the access of supply water. With this view the Dundonald and Isherwood blades, when viewed athwartship, incline aft, the former by a straight line, and the latter by an arc of a circle, their forward edges merging into their tips with much larger radii than the after edges.

Captain John Ericsson, in his later practice employed a screw of this character. This propeller was a true screw, having the blades inclining backward; that is, the section or fraction of the screw used for his propeller was a conical sec-
tion of a true screw, the forward and after edges making acute angles with the axis backward. This form was patented by the Earl of Dundonald in 1843. Its only advantage consists in allowing more space between the propeller and the stern post than when the blades project from the boss at right angles to the axis and when the boss is close to the stern post. The increased space gives freer access for the supply water.

The modified "Griffiths" shape of blade, which in some cases is oval and in others resembles the section exposed by cutting a pear longitudinally through the middle, is more extensively employed, especially for twin screws, than other shapes. When made pear-shape, it is for the purpose of getting the greatest possible surface near where the pitch angle is $45^\circ$, as this is considered the most efficient angle; and by the blade becoming diminished in width toward the tips the surface friction is thereby reduced and room is obtained for the access of the supply water.

When a screw propeller has ample and free access for the supply water and like provision for the egress of the discharged water, and has its length suitably proportioned, and where the forward and after edges in revolving describe planes perpendicular to the axis and parallel to each other, as good a performance may be obtained as with any other form, provided all other conditions are equal.

In Trans. Inst. Nav. Arch., 1882, Mr. Charles Hall relates that he was provided with $\frac{1}{4}$-scale models of several steamers, to which he fitted powerful clockwork, and that he made a series of eighty trials to ascertain the results of model screws representing 17 feet 4 inches diameter and 28 feet uniform pitch, but with a varying amount of blade surface. The screw was of the Griffiths type, representing 147 feet of blade area,
and was progressively cut down laterally until the area became 87 square feet and the edges of the blades parallel. The power delivered was 765 foot-pounds per minute. The time in which the same power was delivered progressively diminished as the area was reduced from 60 seconds, when the screw represented 147 square feet of blade area, to 49.6 seconds, when the screw represented 87 feet area.

After the contractors' trial of the U. S. S. Detroit, when the ship was docked, some surprise was occasioned by the discovery that one blade of each of the three-bladed screws was similarly broken off short, while the remaining blades were distorted and cracked. Only one blade out of the six was not cracked, and that was bent. There is little doubt that the blades gave out during the official speed trial. Not long after getting under way it was noticed that the revolutions of the starboard engine suddenly increased, and the vibrations of the stern, which had previously been moderate, became severe. There were indications that the other screw was broken when a very short turn was made on the run home over the course. The speed made with these broken and distorted screws was so high that many will not believe that they were broken during the speed trial. The blades were made of manganese bronze.

As it is the length of a screw propeller rather than its number of blades that affects its power to move a given columnal mass of water, the most advantageous length becomes an important element to determine.

Examples of practice furnish the best criterion by which to proportion the length. A screw can be made too long, just as it can be made too large in diameter for a given case. When made too long, the loss occurring from the divergent
and rotary direction given to the discharged water, and also from the surface friction of the blades, more than counteract the advantage of the increased mass of water acted upon.

The length of screw propellers should vary with their rotative speeds. The most efficient length appears to lie somewhere between 18 and 20 per cent of the diameter—the former for high speeds, and the latter for lower speeds, as for cargo ships. The average of a number of high-speed screws of good performance, of modified "Griffiths" shape give a maximum length of 19 per cent of the diameter, and for a number of lower-speed screws of good performance of modified "Griffiths" shape the average maximum length was 19.6 per cent of the diameter.

Where the diameter of a screw is restricted in consequence of a small draught of water, the length may be advantageously increased to 20 or 23 per cent of the diameter.

The efficiency of a screw is affected by the conditions under which it operates. For propelling a fast vessel in smooth water with a high rotative velocity a short screw will prove more efficient than a long one, for the reason that it lays hold of successive masses of water more rapidly, and thereby receives from the inertia of the water an advantage that cannot be obtained with slower velocities, and will also offer less surface friction. Under such a change of conditions as would be produced by strong headwinds and rough water, when fewer revolutions prevail, a longer screw would prove more efficient, because it cannot lay hold of new masses of water so rapidly, and therefore needs to act on a longer column in order to produce the necessary thrust.

A screw whose generatrix is a right line at right angles to
the axis will, in operation, produce a divergent and a rotary motion to the discharged water, and the extent of these motions will depend on the length of the screw, as explained in connection with Figs. 2, 3, 4, and 5, Plate XVIII. If the generatrix is a line curved in the direction of rotation, as with the Isherwood and the Nystrom screws, then the divergency is diminished and the rotary motion increased. On the other hand, if the generatrix is a line curved opposite to the direction in which the screw revolves, then the divergency is increased and the rotary motion diminished.

The author once devised an experimental apparatus for an inventor which produced ocular demonstration of the foregoing phenomena. When the working face of the blades was made sufficiently concave, the air discharged was given a convergent direction. This was the object which the inventor sought to accomplish. The result was plainly visible when a beam of sunlight was allowed to fall on the revolving propellers.

A launch propeller was made to produce this convergency of discharge, but when tried it showed a performance inferior to that of a true screw of like diameter.

Some authorities treat with levity the centrifugal or divergent action of screw propellers on the water which they discharge or ignore it altogether as unworthy of notice, but they lay great stress on the rotary motion given to the discharged water. Unless screws are supposed to act quite differently when submerged to drive a vessel to what their known action is in other analogous positions, there must result both a rotary and a divergent motion given to the water discharged by a right screw, as explained in connection with Plate XVIII.

Mr. Thornycroft found it necessary with his screw turbine or guide-blade propeller to have the screw fit the casing as
close as possible, to prevent the water acted upon by the screw escaping around the tips of the blades, which tendency is caused by the centrifugal action of the screw, which, in this case, is made unusually long.

Mr. Griffiths discovered in the course of his experiments that the boss or hub might be made one fourth or more the diameter of the screw without impairing its efficiency; but his experiments were generally made with single-screw wooden vessels having large stern posts and bearings, which obstructed the supply of water to the screw and shut out a considerable volume.

Mr. Charles Hall experimented with a view of learning the effect of increasing the diameter of the boss. With a model screw, to which was applied a given number of foot-pounds of energy, he obtained a speed of 188 feet per minute. When the diameter of the boss was increased to practically shorten the blades one fourth, the speed fell off to 180 feet. When the boss covered the blades so as to practically shorten them one third of their length, the speed was brought down to 158 feet per minute.

It is true that the area for the passage of water diminishes rapidly as the axis is approached, owing to the blades increasing in thickness where the disk area diminishes rapidly. It is also true that the obliquity of the acting face decreases as it approaches the axis, and, therefore, has a greater tendency to carry the water around with the revolving screw; yet a large boss does not remedy these defects, but simply obliterates them by introducing a factor which increases the weight and expense of the screw. The only good reason that the author can assign for the use of a large boss is its necessity when the blades are bolted to it.
Although the author has formulated no rules of his own for determining the dimensions of screw propellers, but simply quotes those in successful use for the purpose, he entertains opinions regarding the form and dimensions of screw propellers which are derived from very large experience in their manufacture, the prosecution and observation of many experiments, and the collation and preservation of their results, as well as careful attention, extending over many years, to the actual performance of numerous modifications of the screw.

He would always have the screw well immersed; that is, would have a certain proportion of the draught covering the top edge of the screw when the vessel is down to the load-line. It is better to use a smaller screw with greater proportionate length, and have it well immersed, than to use a screw of larger diameter that would break up the water at the surface and allow the admission of air to the screw. It can be safely accepted as a rule that as the immersion of a screw propeller increases its diameter may diminish and equal efficiency be the result. He would fix the pitch altogether by the diameter and the intended number of revolutions of the screw.

For the slower speeds, as of cargo steamers and towboats, he would employ as large diameters as the vessel’s draught would permit, would have the screw well immersed, and of rather fine pitch. For the higher-speed vessels he would employ screws of smaller diameters and of greater pitches relatively to the diameters.

On the official trial of the U. S. S. “Detroit,” to which reference has been made, an average speed of 18.71 knots was attained during a four-hours’ continuous run. Some months later the U. S. S. “Marblehead,” practically a dupli-
cate of the former vessel, was tried under equally favorable, if not superior, conditions, and an average speed of only 18.44 knots was attained during a run of the same length of time. In seeking a cause for this difference of results, about the only thing discovered that could be likely to cause it was the difference in the pitch of the screws. Although the screws of the "Marblehead" attained a greater number of revolutions, they had less pitch than those of the "Detroit." When the fact of the damaged condition of the screws of the latter are taken into consideration, the result is even more favorable to the coarser pitch.

By increasing the pitch of the screws of the U. S. S. "Baltimore" a better result was obtained. A similar result followed when the pitch of the screws of H. M. S. "Iris" was increased.

Dr. Froude noted the diminution of friction with increase of slip, and stated that at a slip of 30 per cent the friction at the back of the blade appeared to be diminished by half.

As to the type of screw, the author is inclined to favor the Isherwood, with expanding pitch, when placed in the stern, as its configuration affords better access of the water to it. This type is especially suitable for long-voyage vessels with single screws, in which case there is but little backing to do. The greater convexity of its back operates against its backing power.

Situated as twin screws are, the author is inclined to favor the modified "Griffiths" configuration. For towboats, ferryboats, and the like, all of which are subject to frequent and sudden stoppages and backing, the true screw is preferable with a greater proportionate length than is required for other purposes, having a greater length toward the tips than at the
hub, and the forward corners of the blades of a much greater radius than the after corners, being similar in this respect to the configuration of the “Isherwood.” The depth of immersion of the upper tips of the blades should increase with the draught of the vessel or the diameter of the propeller. It should not be less than the following proportion when the vessel is down to her load-line:

For 4 feet and less diameter ........ 1/10 of diameter.
From 4 to 8 feet diameter ........... 1/8 " "
" 8 to 12 " " ............... 1/4 " "
" 12 to 16 " " .............. 1/6 " "
" 16 to 20 " " .............. 1/8 " "
20 feet and above ............... 1/8 " "

Situated in a stern aperture, the distance between the forward edges of the screw and the stern post should be the greatest that the design will allow.

The performance of screw propellers is often most perplexing and unaccountable.

During the years 1870 and 1871 Chief Engineer Charles H. Loring, U. S. N., conducted an extensive series of experiments at the Washington Navy Yard with screw propellers applied to a steam launch. Many different forms were experimented with, and there was surprisingly slight difference in the performance of each. The screws were of like diameter and varied little in pitch.

Among the number of propellers was one constructed of four flat blades, or paddles, attached to a boss by round bars, the blades being set obliquely to the axis of rotation. The performance of this propeller, as compared with that of others of like diameter, indicated that there was some plausi-
bility in the assertion of Mr. Griffiths that "four strips of plate iron, set at an angle on the shaft which would hold the engine to the speed you required, would give you within half a knot of the best screw ever made."

The patent records of the United States show that more than two hundred patents have been taken out for various forms of screw propellers, all of which embody the general principle of the rotation of the blades, having their surface placed obliquely to the axis of the shaft that turns them. These forms are varied in almost every conceivable manner, many being whimsical and even grotesque, and yet any one of them will propel a vessel.

A screw propeller should be as light and contain as little material as possible commensurate with sufficient strength to withstand any stress that it may encounter by contact with the water while the engine is developing its maximum power. Its entire surface should be as uniform and smooth as possible without polishing. The section of the blades should present an easy form for passing through the water, and the edges,

![Entering Edges](image)

**Fig. II.**

especially the forward ones, should be sharp and of the shape shown in Fig. II, or similar to the sections of blades illustrated in Plate XXIX. The lightest and cheapest screw propeller is one cast entire, but such a screw does not admit
of easy repair in case of an accident that might damage a blade. The practice of making the blades separately and then bolting them to the boss has of late years become more general than heretofore, on account of the facility which it offers for the replacement of a broken blade, and also for the alteration of the pitch within a limited number of degrees.

It has been stated by Mr. Hall-Brown that for vessels of very full form—a class with which he has had large experience—a large diameter and a small pitch ratio are essential to success; and he attributes this necessity to the influence of dead water at the stern, as distinguished from frictional wake. In such cases the blades must reach well into the water clear of the stern, so that the proportion of dead water to the total area of stream acted upon may be as small as possible.

Mr. Hall-Brown gives the particulars of what is found to be a good screw for a cargo-vessel of the following dimen-

![Fig. 12.](image)

sions: Length B. P., 277 ft.; beam (molded), 37.5 ft.; draught, 19 ft. 11 in.; displacement, 4670 tons; block coefficient, .792; I. H. P., 825; speed, 9 knots.

- Diameter of screw: 16 ft.
- Pitch: 16 “
- Revolutions: 64
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Fig. 12 illustrates the form of screw employed by Mr. Thornycroft for torpedo boats and other fast vessels.

The following are the dimensions of a Griffiths screw, as given by Burgh:

Depth of top of blades from line of flotation = Diameter of screw ÷ 10 to 12.

Pitch of blades at edge = Diameter × 2 to 1.25.

Length of screw = Diameter ÷ 5 to 6.

Diameter of boss = Diameter ÷ 3 to 4.

Length of boss = \{ Diameter of flange + half the diameter of shank in some cases = diameter of boss.

Diameter of flange of blade = \{ Diameter of boss × .5 thickness of edge of flange.

Lap of blade on boss beyond flange = \{ \(\frac{3}{4}\) inch per foot of diameter of screw = diameter of screw ÷ 20.

Width of screw blade at widest part = Diameter of screw ÷ 3.

Width of blade at point = Diameter of screw ÷ 7.

Thickmess of blade at root = \{ \(\frac{1}{4}\) inch to each foot of screw's diameter.

Thickness of point = \(\frac{1}{8}\) that at root.

Diameter of shank = Diameter of screw × .25.

Metal around shank = Diameter of boss ÷ 23 to 24.

Metal beyond flange and cotter = \(\frac{7}{12}\) of the depth of cotter.

Width of main cotter = Diameter of shank × .5

Thickness of main cotter = Diameter of shank ÷ 6.

Thickness of small cotter = Its width ÷ 2.

Width of small cotter = Diameter of boss ÷ 20.

Angle inside of wedge box = 71\(\frac{1}{2}\) degrees.

Metal in cheeks where crotters enter = Diameter of boss ÷ 40.
Thickness of plate over \( \text{wedges} \) = Diameter of boss \( \div 48 \).

"Blades of screw curve forward a half inch to each foot of diameter of screw from face to root, the curve to commence at the centre of blades."

The following are the principal dimensions of a two-bladed Griffiths screw as given by Rankine:

Length of blades at tips, about \( 0.07 \) of pitch.
At the longest part, being about \( \frac{4}{10} \) of the radius from the axis \( 0.167 \) " "
At the root, about \( 0.11 \) " "
Mean length of blade \( 0.12 \) " "
Mean aggregate length of two blades \( 0.24 \) " "

The outlines of the blades are rounded. The tops of the blades are slightly bent toward the vessel, so as to present a somewhat convex surface to the water they act upon.

Prof. Rankine gives the following rule for the required thickness of cast-iron blades of ordinary section of which the working strength may be taken as from 4000 to 4500 pounds per square inch. The working strength of the shaft is taken as from 8000 to 9000 pounds per square inch.

**Rule**: Multiply the cube of the diameter of the shaft by \( 4 \) and divide by the number of blades multiplied by the axial length of the blade at its root in inches. The square root of the quotient will be the greatest thickness of the blade at the axis of the shaft.

Let \( D \) be the diameter of the shaft; then

\[
\sqrt{\frac{D^3 \times 4}{\text{No. of blades} \times \text{length at root}}} = \text{thickness at axis.}
\]
If the blades are made of material having equal strength with the shaft, the multiplier will be 2 instead of 4.

In Fig. 13 let $A$ represent the axis of the shaft, the circle $G, D, H$ the outline of the boss of the propeller, and $A, B$ a radius drawn from the centre of shaft to tip of blade.

Make $A, C$ equal to the thickness, as computed by the foregoing rule, and join $C, B$. Then, the triangle $B, A, C$ will represent the section through the centre of the blades that gradually diminishes in strength from centre to tip. In practice, the tip of the blade cannot be made quite sharp; so from $B$ set off a distance, as $B, E$, equal to the minimum practicable thickness, and from $E$ draw $E, F$ parallel to the face or radius; join $E$ and $D$ at $F$ with the curve. In this method of construction, if a blade should happen to break it will probably do so near the tip and leave a portion still available for propulsion.

Example.—Given a screw propeller of three cast-iron blades of 15 feet diameter, with a shaft of 12 inches diameter, required the thickness of blades at the centre of the shaft, the blades having an axial length of 34 inches:

$$\frac{12^3 \times 4}{3 \times 34} = 67.76.$$  

Thickess $= \sqrt{67.76} = 8.2$ inches.

The following extracts regarding the diameters and revolutions of propellers are from Mr. S. W. Barnaby's work on
"Marine Propellers." Whatever he says on the subject is of value, because of his known ability and extensive practical experience with the subject in connection with the business of Mr. Thornycroft.

Mr. Barnaby says: "The whole problem of screw propulsion is to fix upon the best diameter of a propeller, and the best ratio of pitch to diameter, under any given conditions."

It is sufficient, in most cases, to take an actual propeller which is known to give a good performance and to use that propeller as a model.

Then the following rules will enable you to find the diameter and revolutions suitable.

To find the diameter of a propeller for a given I. H. P. and a given speed from the diameter of another similar propeller at a different I. H. P. and a different speed:

If \( d \) = diameter of model which may be larger or smaller than \( D \);

\( D \) = diameter of required propeller;

\( \dot{p} \) = I. H. P. of model;

\( P \) = I. H. P. of required propeller;

\( v \) = speed of vessel with model propeller;

\( V \) = speed of vessel required;

\( r \) = revolutions of model propeller;

\( R \) = revolutions required; then

\[
D = \sqrt{d^4 \times \frac{v^4}{V^4} \times \frac{P}{\dot{p}}}.
\]

*Example.*—If \( d = 5.0 \) feet, \( \dot{p} = 670 \) I.H.P., \( v = 18 \) knots,

\( P = 1800 \) I.H.P., \( V = 20 \) knots, then

\[
D = \sqrt{5^4 \times \frac{1800}{670} \times \frac{18^4}{20^4}} = 7 \text{ feet},
\]

if model is smaller.
If the model is larger, the ratios are reversed, thus:

\[ D = \sqrt{7^2 \times \frac{670}{1800} \times \frac{20}{18}} = 5 \text{ feet.} \]

Having, therefore, a given speed of vessel and a given horse-power to start with, fix upon the diameter of the propeller, then upon the revolutions suitable for the propeller.

To find the suitable number of revolutions for a given propeller at a given speed from the revolutions of a similar propeller at a different speed, "the revolutions per minute are proportional to the speed and inversely proportional to the diameter."

If \( D \) = diameter of given propeller, \( d \) = diameter of smaller model, \( V \) = given speed, \( v \) = speed of model, \( R \) = revolutions of given propeller, \( r \) = revolutions of model; then

\[ R = r \times \frac{V}{v} \times \frac{d}{D}. \]

Example.—If \( D = 7 \) feet, \( d = 5 \) feet, \( V = 20 \) knots, \( v = 18 \) knots, \( r = 400 \) revolutions; then

\[ R = 400 \times \frac{20}{18} \times \frac{5}{7} = 318 \text{ revolutions.} \]

If the model used is larger than the given propeller, the ratios are reversed.

The pitch of the propeller should then be made the same ratio to the diameter as in the model.

The surface of the blades should be about 35 per cent of the disk area.
In the application of the system of comparison for determining the dimensions of a screw propeller—that is, by taking a propeller which has given a good practical performance and using it as a model by which to determine the proportions of another—it is important that both the model screw and the vessel to which it is applied should conform in general proportions, as nearly as possible, to those of the proposed screw and vessel. The nearer these conditions prevail the closer the result will approximate the object sought.

A METHOD OF DESIGNING SCREW PROPELLERS WHICH HAS BEEN EMPLOYED AT TIMES BY THE BUREAU OF STEAM ENGINEERING U. S. N., BY PASSED ASSISTANT-ENGINEER W. M. McFARLAND, U. S. N.

This method is based on the idea that to a certain extent the size of the propellers should depend on the thrust of the screw in propelling the vessel. As it is impossible to make anything more than an approximation to an actual thrust, it is just as well, and much simpler, to use the indicated thrust in calculation, the indicated thrust being equal to the horse-power in foot-pounds divided by the product of pitch and revolutions.

An examination of the performances of a number of vessels whose propellers have undoubtedly been designed so as to produce the best results shows that for engines running at moderate speed it is safe to allow about a thousand pounds of indicated thrust per square foot of developed or helicoidal area of the screw, while for fast-running engines, where a larger amount of slip is allowed, the amount of indicated thrust per square foot of developed area may run from 1100 to 1300 pounds.

The process to be followed in designing a screw would
then be about this: The horse-power of the engines having been decided from the requirements of the ship and the number of revolutions per minute fixed, consideration being given both to the propelling instrument and the circumstances of the case for the engines themselves, the next thing would be to settle upon the pitch of the screw. This can be readily done by allowing such a percentage of slip as from experience would be likely to occur under the circumstances of the case. Now, having the indicated horse-power and the pitch of the screw, as well as the number of revolutions, it is very easy to get the indicated thrust; and, as already stated, the developed area would then be equal to the indicated thrust divided by the number of pounds which we have decided would be best.

Now, having the developed area of the blades, we find the area of each blade from the number of blades desired to be used, so that we get the area of each one. We can then decide upon the diameter either by comparison of other screws of somewhere near the same size, or, if preferred, by assuming that the blade will be an ellipse. This, of course, is a detail which can be readily worked out in any drawing-room where propellers are usually designed.

We now have the diameter, pitch, and developed area of the blades, so that, of course, the remainder of the problem is merely a matter for descriptive geometry.

The following example will illustrate the several steps in the case:

*Example.*—Given for a ship with twin screws of

5400 I.H.P.,

175 revolutions,

18.5 knots,
required the pitch and helicoidal area of a screw, allowing 15 per cent slip and 1100 pounds per square foot of helicoidal area—

The aggregate I.H.P is divided by the number of screws employed.

\[
\text{Pitch} = \frac{18.5 \times 6080}{175 \times 60 \times .85} = 12.6 \text{ feet;}
\]

\[
\text{Helicoidal area} = \frac{2700 \times 33000}{175 \times 12.6 \times 1100} = 36.75 \text{ sq. feet.}
\]

Required the same, allowing 20 per cent slip and 1300 pounds per square foot of helicoidal area—

\[
\text{Pitch} = \frac{18.5 \times 6080}{175 \times 60 \times .80} = 13.4 \text{ feet;}
\]

\[
\text{Helicoidal area} = \frac{2700 \times 33000}{175 \times 13.4 \times 1300} = 29.2 \text{ sq. feet.}
\]

The method of design followed by the bureau is practically that given in Seaton's "Marine Engineering" on pages 287 to 296, with constants based on successful ships already tried and such changes as study of tabulated data suggests.

This method has been found to give very good results.

Mr. Seaton gives the following regarding the proportions of propellers:

Pitch of Propeller.—For all ordinary cases, take the intended speed of the ship in feet per minute (1 knot is equal to 101.33 feet per minute), increase it by one tenth, and divide by the intended number of revolutions per minute; the quotient will be the pitch in feet. For the large propellers of very bluff cargo boats, increase the intended speed by one twentieth only.
The general expression for the pitch for any percentage (x) of apparent slip is, then,

\[
Pitch \text{ of propeller} = \frac{S \times 101.33}{R} \times \frac{100}{100 - x} = \frac{S}{R} \times \frac{10133}{100 - x'}
\]

where \(S\) is the speed of the vessel in knots and \(R\) the revolutions per minute.

**Apparent Slip of Propeller.**—This is usually reckoned as a percentage of the nominal axial advance of the propeller \((P \times R)\), and its amount in any given case is

\[
\text{Apparent slip (per cent)} = \left(\frac{P \times R}{P \times R} - \frac{S \times 101.33}{P \times R}\right) \times 100,
\]

where \(S\) is the speed of the ship in knots, \(P\) the pitch of the propeller in feet, and \(R\) the revolutions per minute.

The apparent slip should generally be from 8 to 10 per cent at full speed. Should it in any case be less than 5 per cent, it may be taken for granted that the propeller is not suited to the ship. In bluff cargo boats, however, the slip rarely exceeds 5 per cent; so that the preceding remark does not apply to them. A larger slip (say 12 to 18 per cent) does not necessarily imply a waste of power, and may be due to a small diameter of propeller; but a larger percentage, again, would probably mean that the blade area or "surface" was too small. If this latter is the case, it at once becomes apparent, when the indicated thrusts are plotted down and a curve drawn, for the curve will show a want of augmentation at the higher speeds.

\[
\text{Indicated thrust} = \frac{\text{I.H.P.} \times 33000}{P \times R}.
\]
For method of constructing curve, see page 94.

_Diameter of Propeller._—For all ordinary cases, the diameter of the propeller should be that given by the following formula:

\[
\text{Diameter of propeller} = K \sqrt{\frac{\text{I.H.P.}}{\left(\frac{P \times R}{100}\right)^4}}
\]

where \(P\) is the pitch in feet, \(R\) the revolutions per minute, and \(K\) a coefficient which has the values shown in the following table:

<table>
<thead>
<tr>
<th>PROPELLER COEFFICIENTS.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Description of Vessel.</td>
</tr>
<tr>
<td>--------------------------</td>
</tr>
<tr>
<td>Bluff cargo boats.</td>
</tr>
<tr>
<td>Cargo boats, moderate lines.</td>
</tr>
<tr>
<td>Pass. and mail boats, fine lines.</td>
</tr>
<tr>
<td>Pass. and mail boats, fine lines.</td>
</tr>
<tr>
<td>Pass. and mail boats, very fine.</td>
</tr>
<tr>
<td>Naval vessels, very fine.</td>
</tr>
<tr>
<td>Naval vessels, very fine.</td>
</tr>
<tr>
<td>Torpedo boats, very fine.</td>
</tr>
<tr>
<td>Torpedo boats, very fine.</td>
</tr>
</tbody>
</table>

Or, if \(\left(\frac{P \times R}{100}\right)^4\) be used instead of \(\left(\frac{P \times R}{100}\right)^4\), the coefficient \(K\) becomes practically constant, and the formula for four-bladed propellers becomes

\[
\text{Diameter} = 7.7 \sqrt{\frac{\text{I.H.P.}}{\left(\frac{P \times R}{100}\right)^{1.14}}}
\]
For three-bladed screws, multiply the diameter thus found by 1.05.

Blade Area or Surface of Propeller.—For all ordinary cases satisfactory results will be obtained if the blade area be that given by the following rule:

\[ \text{Total developed area of blades} = C \times \sqrt{\frac{\text{I.H.P.}}{R}} \]

where \( R \) is the number of revolutions per minute, and \( C \) is a coefficient having the values given in column 6 in preceding table. In naval vessels the centre of the propeller is usually more deeply immersed in proportion to diameter, and the revolutions are higher for a given power than is the case in merchant vessels.

The thickness of blade can be determined easily and with a close approach to accuracy by means of the following formula:

\[ T = \sqrt{\frac{d^*}{n \times b}} \times K \]

where \( T \) is the thickness of blade at the centre of shaft, \( d \) the diameter of tail shaft in inches, \( n \) the number of blades, \( b \) the breadth of blade in inches where it joins the boss, or commonly the length of the boss, and \( K \) a coefficient having values as follows:

- For cast iron, \( K = 4 \).
- " cast steel, \( K = 1.5 \).
- " gun metal, \( K = 2 \).
- " high-class bronze, \( K = 1.5 \).
The thicknesses given by the above rule for gun metal and bronze blades are such as are suited to propellers revolving at speeds usual in naval work. The thickness of the blade at tip should be as follows:

Thickness at tip, cast iron \[ .04D + .4 \text{ inch} \]
" " " cast steel \[ .03D + .4 \text{ "} \]
" " " gun metal \[ .03D + .2 \text{ "} \]
" " " high-class bronze \[ .02D + .3 \text{ "} \]

where \( D \) is the diameter of propeller in feet.

Blade flanges should be of the following proportions:

Diameter of flange = \( 2.25 \times \) diameter of tail shaft;
Thickness of flange = \( .85 \times \) diameter of stud for gun metal, bronze, or steel;
" " = \( 1.4 \times \) diameter of stud for cast iron.

Proportions of propeller boss:

Length of boss, cast iron = \( 2.7 \times \) diameter of tail shaft;
Diameter " " = \( 2.7 \times \) " " " " " "

The fore and aft section of boss should be oval, the principal radius being \( .8 \times \) diameter of boss.

The length of taper bore should be divided into three approximately equal parts, of which the two end ones will bear on the shaft, while the centre one will be cored back.

The thickness of shell or central cored-out portion of the boss should be:

Thickness of boss = \( .65 \times \) thickness of blade or shaft axis.

Where the blade flanges are all within the sphere of the boss,
the extreme diameter of the boss is about $3.3 \times$ diameter of tail shaft, the length of the boss being $2.7 \times$ diameter of tail shaft. When the propeller is solid the breadth of blade at root $= 2.7 \times$ diameter of tail shaft. When blades are separable the breadth of blade at flange $= .8 \times$ diameter of flange $= 1.8 \times$ diameter of tail shaft.

*Tail shafts*, the taper of the part which fits the boss, should be about one inch on the diameter for each foot of length, but never less than three fourths of an inch. The thread of the large nut that holds the propeller on the shaft is $2\frac{1}{2}$ threads per inch, regardless of diameter. It should be left-handed when the propeller is right-handed, and *vice versa*. The nut should be securely locked, preferably by a plate fixed to the after end of the boss by tap screws.

The propeller should be secured by one feather or key extending the whole length of the boss, the proportions of which may be,

\[
\begin{align*}
\text{Breadth of key} &= 0.22 \times \text{largest diameter of shaft} + 0.25; \\
\text{Thickness of key} &= 0.55 \times \text{breadth}.
\end{align*}
\]

The diameter of the screwed end of the shaft should be sufficiently reduced to allow the key to be fitted in from the after end clear of the thread.

*Studs or Screws for Attaching Blades to Boss.*—These are usually of "naval brass" (Muntz-metal with the addition of one per cent tin) or one of the stronger bronzes, for gun metal or bronze propellers, and of mild steel for cast-iron or cast-steel propellers. They should be of the size given by the following formula:

\[
a \times N \times r = \frac{T \times L}{K}
\]
where \( a = \) area of one stud or screw at bottom of thread in square inches;

\( N = \) number of screws or studs for one blade (usually 7 to 11);

\( r = \) radius of studs or screws in inches;

\( T = \) indicated thrust, i.e., \( \frac{I.H.P. \times 33000}{\text{pitch} \times \text{revolutions}} \);

\( L = 0.6 \times \) total length of blade in feet (flange joint to tip);

\( K = 17 \) for steel studs, 14 for naval brass or bronze.

The following method of determining the thickness of the blade at the centre of the shaft is employed by some, and gives a close approximation when there is not a great variation in the shape of the blade:

Cast iron . . . . allow .5 in. for each foot in diameter of screw;

TABLES OF DATA RELATING TO THE PERFORMANCE, TYPE, AND DIMENSIONS OF SCREWS.

Experienced designers of screw propellers are guided almost entirely by experimental information, and to such all trustworthy data that relate to the character, dimensions, and performance of screw propellers become valuable.

I am indebted to Chief Engineer Harrie Webster, U.S.N., their compiler, for the use of the following tabulated data.

Chief Engineer Webster says:
FORMS AND DIMENSIONS OF SCREW PROPELLERS. 163

"The accompanying table and illustrations of the screws to be found elsewhere have been compiled in the belief that they will prove of interest to all marine engineers, and may assist in investigations tending toward a solution of the problem of the scientific design of screw propellers.

"While certain general principles in connection with propeller designs are now pretty well settled, the fact remains that some of the most important features are still determined empirically or at times by the whim of the designer.

"This table, which contains the results of trials of fifteen ships, taken in connection with the reduced working drawings of the propellers, will at least give the careful student something on which to base empirical formulæ.

"The data of the propellers themselves are absolutely correct, as are those of the hull dimensions. Those of the machinery are also thoroughly trustworthy, but prior to the trial of the 'Yorktown' the horse-powers were not corrected for indicator error, as was subsequently done. This, however, is of less importance, as the screws of the older vessels were much larger than would now be designed.

"The speeds of some of the vessels, which were tried for horse-power only, are not entirely reliable, owing to the fact that on the trial they were determined by patent log and bearings with an estimate of the tide. The values given in the table for these cases have been adopted after comparison with similar ships where the speed was accurately determined, taking into consideration the circumstances of wind and weather, engine revolutions, etc. As the table is not constructed to support any theory, these computed values are believed to be entitled to nearly as much weight as those obtained by direct observation."
"The illustrations are reductions of accurate copies of the working drawings, permission having been kindly granted by Engineer-in-Chief Melville.

"Most of the records are the results of trials for four or more hours at full power, but, where the data for shorter periods were more carefully determined, they have been used. This is the case with the 'Boston,' 'Chicago,' and 'Yorktown,' where the results of measured-mile performances are given, and the 'Vesuvius,' whose horse-power was only determined for a short run. The speeds are in knots of 6080 feet. The indicated thrust is computed from the ordinary formula.

\[ T = \frac{\text{I.H.P.} \times 33000}{P \times R} \]

where I.H.P. = indicated horse-power of main engines only;
\[ R = \text{revolutions of propeller per minute}; \]
\[ P = \text{mean pitch of propeller in feet}. \]

"In the case of twin screws, the indicated thrust is the mean for the two screws.

"Two performances of the 'Baltimore' are given, because there was a decided change in the pitch and horse-power, and, in connection with the tables of engine dimensions and trial data in Vol. III of the Journal of the American Society of Naval Engineers, they enable the effect of change of pitch to be studied. As this change was made by simply shifting the blades slightly on the bosses, all other propeller data are the same for both cases."

The official speed trials of the U.S.S. "New York," "Columbia," "Detroit," "Montgomery," "Olympia," and "Minneapolis" occurred subsequent to the publication of Chief Engineer Webster's paper in the Journal of the Ameri-
can Society of Naval Engineers. The data of those trials, kindly furnished by him, are given in an appended table, which affords additional practical and useful information.

The screws of the "New York," "Columbia," and "Minneapolis" were designed by the Messrs. Cramp; those of the "Detroit" and "Montgomery" by the Bureau of Steam Engineering U.S.N.

The screws of the "Olympia" were designed at the Union Iron Works, San Francisco. They are of expanding pitch. The pitch of their forward edges is 18 feet, and that of their after edges 20 feet.

Referring to Plate XXII, and particularly to the sections of the blades therein shown, it will be observed that the screws of the torpedo boat "Cushing" present an easy form for passing through the water, the front and back of the blades being alike. This form renders the screw efficient for backing and manœuvring, an important consideration with a torpedo boat. But this advantage is attained to the detriment of the driving-ahead qualities of the screws, in consequence of the convexity of face produced on the screw blades. As the screws are employed in driving the boat ahead the chief part of the time during which they are in operation, it is doubtful whether this sacrifice of efficiency in driving ahead is justified in view of the advantage above named.

The screws of the "New York," Plate XXV, are of uniform pitch, and are of the same character as those illustrated in Fig. 27, Plate III, and also in Plate IX. The generatrix is a right line inclining backward about 15°, or making an angle of about 75° with the axis. If such a screw be cut by a plane perpendicular to its axis, the face line exposed will be a spiral curve similar to that employed by Hirsch. As previously
stated, the effect of this curvature is to cause the screw to dis-
charge the water in a more or less converging direction, but in
doing so greater rotary motion is imparted to the water dis-
charged. If there is any merit in departing from the genera-
trix of a right screw, an arc of a circle, as employed by Isher-
wood, is preferable.

It will also be observed that the sections of the blades are
given an easy form for passing through the water, but to a
less detrimental degree for going ahead than in the screws of
the torpedo boat "Cushing."

It will be observed in the illustrations of the screws of the
"Columbia," Plates XXVII and XXVIII, that they are of the
same general character as those of the "New York," but have
a higher pitch ratio than the screws of the latter, which feature,
no doubt, contributed to the obtaining of the superior
result.

In the results of the trial of the "Columbia" an opportu-
nity is presented for the comparison of the value of increased
immersion. The engines of the different screws being similar,
it is fair to assume that their initial resistance was the same.
If the centre screw had the same immersion as the port screw,
it would, undoubtedly, have made more revolutions, being one
foot smaller in diameter, with less blade area, and its engine
having a greater mean pressure on its L.P. cylinder; but, as
the table reveals, it made fewer revolutions, which is evidence
that the water which it displaced offered greater resistance to
it than was offered to the port screw by the water which it
displaced.

This increased resistance was owing to the greater hydro-
static pressure, due to the greater immersion of the centre
screw, which had, likewise, less slip than either of the others.
Similar phenomena are observed in the trial record of the "Minneapolis."

The centre screw engines developed ..... 7218.82 I.H.P.;
The starboard engines " ..... 6586.72 I.H.P.;
The port engines " ..... 6560.69 I.H.P.

The centre screw being one foot less in diameter and of less pitch and helicoidal area than the side screws, if immersed under the same conditions, should have made more revolutions than the mean of the side screws, but, as the table reveals, it made less.

The mean indicated thrust on the starboard and port screws was 1385.6, while that of the centre screw was 1573.7. The slip also of the centre screw was less than either of the others, all of which goes to corroborate the opinion expressed in regard to the advantage of deep insertion.

Mr. E. A. Linnington says, in Trans. Inst. Nav. Arch. for 1887, that one of the most interesting and valuable features in the development of naval construction in recent years is the great advance that has been made in the speeds of our [English] warships.

From the first-class armored fighting ship of about 10,000 tons displacement down to the comparatively diminutive cruiser of 1500 tons the very desirable quality of high speed has been provided. These are all twin-screw ships. The speeds attained indicate a high efficiency with twin screws. In all ships, but more especially in high-speed ships, success depends largely upon the provision of propellers suited for the work they have to perform; and where a high propulsive efficiency has been secured, there is no doubt the screws are working with a high efficiency.
The table gives the chief points of several classes of ships, the details of the screws, and the results obtained on the measured-mile trial of a ship of each class. The vessels whose trials are inserted in the table have not been selected as showing the highest speeds for the several classes. They are ships which have been run on the measured mile at or near the designed load-water line. On light-draught trials speeds have been attained of from half a knot to a knot higher than those recorded. All these measured-mile trials were made under the usual admiralty conditions—that is to say, the ships' bottoms and the screws were clean, and the force of the wind and state of the sea were not such as to make the trials useless for the purpose of comparison.

There are a few points of detail about these propellers which deserve passing notice. On Fig. 1, Plate XXVI, is shown a fore-and-aft section through the boss. The flanges of the blades are sunk into the boss, and the bolts are sunk into the flanges. The recesses for the bolt heads are covered with a thin plate having the curve of the flange in such direction that the flanges and boss form a section of a sphere. The conical tail is to prevent loss from eddies behind the flat end of the boss, and is particularly valuable with the screws of high-speed ships. The light hood shown on the stern bracket is for the purpose of preventing eddies behind the boss of the stern bracket and to save the resistance of the flat face of the boss. The edges of the blades were cast sharp, and not rounded, with a small radius, as in the usual practice. The driving key extends throughout the whole length of the boss, and the taper shaft fits also throughout its length.

It appears that within certain limits, where the shape of the blades does not affect the efficiency of the screw, but with
a given number of blades and a given disk area, the possible variations in the form or distribution of a given area are such that different results may be realized. The shape of the blades of these propellers are shown in Figs. 2, 3, and 4, Plate XXVI. Fig. 2 shows the blades for the A screw. C and D have the same form. Fig 3 shows in full lines the blades of the B screw, and, though very narrow at the tips, they, like A, resemble the Griffiths pattern. The blades of E and F are of similar shape, as shown in Fig. 4, and approach an oval form rather than that of the Griffiths pattern.

All the screws worked outwardly, that is, the upper tips turned away from the hull. The averages of the clearances between the tips of the blades and the respective hulls is about one eighth of the diameter of the screw. These screws are of gun metal, and they are fitted to the ships with the blade surfaces in the same condition as that in which they left the foundry.

When deciding upon the position of twin screws there is room for variation longitudinally and transversely. The immersions given in the table are the vertical positions. The immersion of A is 9 feet, and shows what may be done in a deep-draught ship with a small screw. Whatever the value of deep immersion in smooth water may be, there can be no question that it is materially enhanced in a seaway. The longitudinal positions are such that the centre of the screw is about one fifth of the diameter forward of the aft side of the rudder post. The forward edges of the blades are from 2 to 3 feet clear of the legs of the bracket which carries the after bearing.

An interesting and noteworthy fact in connection with these propellers is that wide differences exist in pitch and revolu-
tions, though the products of the two do not greatly vary. As a general rule, with (revolution \times pitch) a constant, an increased number of revolutions, even with a consequent decrease of pitch, allows a diminution of disk area and blade areas, other modifying conditions, such as thrust, slip, number, and pattern of blades, being the same.

At the present time the engineer who designs a screw propeller depends mainly on information such as that which is given in the table; and as published data of this kind are somewhat rare, it is hoped that these will be useful to designers of screws for high-speed ships.

**PARTICULARS OF SOME RECENT HIGH-SPEED TWIN SCREWS.**

<table>
<thead>
<tr>
<th>Ship</th>
<th>A</th>
<th>B</th>
<th>C*</th>
<th>D</th>
<th>E</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length, feet</td>
<td>325</td>
<td>315</td>
<td>300</td>
<td>300</td>
<td>220</td>
<td>250</td>
</tr>
<tr>
<td>Breadth, feet</td>
<td>68</td>
<td>61</td>
<td>56</td>
<td>46</td>
<td>34</td>
<td>32.5</td>
</tr>
</tbody>
</table>
| Draught \( \{ \) forward | 26' 2" | 24' 6" | 21' 00" | 15' 6" | 12' 10" | 13' 1"
| on trial \( \} \) aft. | 27' 3" | 25' 6" | Mean | 19' 6" | 15' 2" | 14' 7"
| Displacement, tons | 9690 | 7645 | 5000 | 3584 | 1560 | 1544 |
| I.M.S., square feet | 1560 | 1287 | 1000 | 744 | 438 | 392 |
| Speed of ships, knots | 16.92 | 17.21 | 18.75 | 18.18 | 16.91 | 17 |
| I.H.P. | 11,610 | 10,180 | 8500 | 6160 | 3115 | 3045 |
| Revolutions per minute | 107.2 | 88 | 120 | 122.6 | 150.4 | 132.1 |
| Pitch of screw | 19' 5" | 22' 0" | 18' 6" | 17' 6" | 12' 7.5" | 14' 9"
| Slip per cent | 17.6 | 10 | 14.2 | 9.7 | 11.4 |
| Diameter of screw | 15' 6" | 18' 0" | 14' 6" | 13' 0" | 10' 6" | 11' 0"
| Diameter of boss | 4' 4" | 4' 11" | 3' 9" | 3' 5" | 2' 9" | 2' 10"
| Number of blades | 4 | 4 | 3 | 3 | 3 |
| Blade area of one screw | 72 | 87 | 60 | 47 | 24 | 24 |
| Shape of blade | Fig. 2 | Fig. 3 | Fig. 2 | Fig. 2 | Fig. 4 | Fig. 4 |
| Pitch | 1.25 | 1.22 | 1.3 | 1.34 | 1.2 | 1.34 |
| Diameter | 2.62 | 2.92 | 2.75 | 2.82 | 3.6 | 3.96 |
| Disk | 2.92 | 2.75 | 2.82 | 3.6 | 3.96 |
| Blade area | 9' 0" | 5' 3" | 4' 4" | 2' 9" | 1' 10"

*The following results were obtained on a trial in April, 1887, at the designed draught of water: I.H.P., 8620; speed, 19 knots; revolutions per minute, 119.
USEFUL MEMORANDA FOR STEAMSHIP PERFORMANCE.

Cubic feet of displacement $\times 0.0279 = \text{tons of fresh water.}$

$\text{Miles } \times 0.87 = \text{knots.}$

$\text{Knots } \times 1.15 = \text{miles.}$

$\text{Feet per minute } \times 0.01 = \text{knots per hour.}$

$\text{Pounds of coal per hour } \times 0.010714 = \text{tons per 24 hours.}$

$I.H.P. \times 0.01607 = \text{tons of coal per 24 hours at the rate of 1}\frac{3}{4}$

$\text{pounds per H.P. per hour.}$

$I.H.P. \times 0.02143 = \text{ditto at 2 pounds.}$

$I.H.P. \times 0.02679 = \text{ditto at 2}\frac{1}{4} \text{ pounds.}$

Coefficient of fineness used in comparing the performance
of vessels, dimensions in feet:

$D = \text{displacement in cubic feet;}$

$L = \text{length of ship between perpendiculars;}$

$B = \text{extreme width of beam;}$

$W = \text{mean draught of water, exclusive of keel;}$

$A = \text{midship area in square feet;}$

$K = \text{coefficient of fineness } = \frac{D}{LBW};$

$k = \text{coefficient of water lines } = \frac{D}{LA}.$

The displacement is estimated at 35 cubic feet = one ton.
MATERIAL BEST SUITED FOR PROPELLER BLADES.

At the twenty-ninth session of the Inst. Nav. Arch., March, 1888, Mr. W. C. Wallace read an interesting paper on the above subject, from which the following is extracted:

"The chief problem connected with the choice of the most suitable material for screw propellers is that of economy.

"Until recent years cast iron has been the material most generally used for propeller blades, and some engineers are still of the opinion that there is much to be said in its favor. Its most serious defect consists in its liability to fracture upon a comparatively slight blow. Instances have been known to occur where blades of cast iron have been entirely carried away in heavy seas from the racing of the engines.

"The resistance of cast iron to corrosion is superior to that of steel. The average time that a cast-iron blade will run without requiring renewal is from five to six years; and even after that it is not always quite useless, for in many cases the tip of the blade may be burned on and its life thus prolonged.

"As compared with steel its surface is initially smoother, and in casting the true form is better preserved. The want of ductility in cast iron goes far to unfit it for work where trustworthiness is a desideratum. In point of cost it is far below that of all other materials used for the purpose.

"The ultimate tensile strength of cast iron suitable for propeller-blades is from 8 to 12 tons per square inch."
"The greatest thickness of a cast-iron propeller blade at the root is usually .5 inch for every foot in diameter for the propeller.

STEEL.

"The introduction of mild steel in the manufacture of propellers has wrought therein a revolution, even as it has in that of many other structures. By its use propeller blades are produced of which no fear of fracture need be entertained from any cause except such as would entail serious damage to the stern of the ship as well as to the propeller. Satisfactory as steel is in point of strength, it is most unsatisfactory in the matter of corrosion. Blades become so pitted and corroded in the course of a few years as to require renewal. The length of time that steel blades will last without renewal is very variable. Swift Atlantic steamers have, in some cases, used the same blades for more than six years, while, on the other hand, other and slower steamers have retained theirs as long as ten years. Instances are on record of steamers trading in all parts of the world having propellers the lives of which lasted but three years. Many expedients have been proposed and resorted to in order to protect steel blades from corrosion, such as covering the back with tin, brass plate, etc., but it is extremely difficult to prevent water getting behind this casing and causing galvanic action. The ultimate tensile strength of steel suitable for propeller blades is from 28 to 34 tons per square inch. Elongation, 6 to 12 per cent.

GUN METAL.

"The use of gun metal for propeller blades dates nearly as far back as that of cast iron, but its use is generally confined
to launches, warships, yachts, and similar vessels. For such purposes it has proved most satisfactory. But a comparison between gun metal and such materials as have been used in the mercantile marine is difficult to make, as, under ordinary circumstances, naval ships are not subjected to the rough usage that frequently falls to the lot of ocean-going mail steamers. Its resistance to corrosion in iron and steel vessels is all that can be desired, the life of the blades being, in most cases, greater than that of the ship to which they are fitted. To prevent serious pitting and corrosion of the stern frames and plating, resulting from the galvanic action produced, strips of zinc are placed around the inside of the stern aperture. Galvanic action then goes on between the propeller and the zinc; the latter is eaten away and the hull is preserved. The quantity of zinc required for a 20-foot propeller is about 5 hundred-weight, which has to be renewed every twelve or fifteen months. In the manufacture of heavy gun-metal castings, as with many of the bronze alloys, great care must be taken to prevent segregation before the metal has time to set. This is liable to occur in the heavier parts, as at the root of the blade; and when it does take place it reduces the strength of the metal where it is most needed. The ultimate tensile strength of good gun metal, having in its composition not less than 8 per cent of tin nor more than 5 per cent of zinc, is from 12 to 16 tons per square inch. Elongation, 3 to 8 per cent

**MANGANESE BRONZE.**

“This alloy, which is extensively used in the manufacture of propeller blades, is very variable in quality. The good
results obtained with some test pieces in the early days of its use led to a too great reduction in the thickness of blades. This reduction, combined with difficulties in casting, led to the unexpected breakage of a number of blades.

"Like gun metal, manganese bronze is all that can be desired with regard to freedom from corrosion. The same means are employed with this as with gun metal to prevent injurious action on the hull from galvanic influences. Accompanying the freedom from pitting and the gradual wasting away of the blades is the freedom from the loss of power which pitting involves. This saving of power compared with corroded steel and cast-iron blades is computed to be about 4 per cent. The ultimate tensile strength of manganese bronze is from 12 to 17 tons per square inch.

PHOSPHOR-BRONZE.

"This is another important alloy of copper that is capable of giving good results when used for propeller blades. Propellers have been cast of this bronze for torpedo boats and launches, but up to the present time, so far as is known, its use for propeller blades has not extended to sea-going steamers. Although it can be safely asserted that a phosphor-bronze propeller would possess the same advantage of freedom from corrosion as gun metal and manganese bronze, experience of the alloy in large castings is wanting, and it cannot, in this respect, be brought in comparison with the two alloys just mentioned. The ultimate tensile strength of phosphor-bronze, according to Kirkaldy, is 15.8 tons per square inch. Elongation, 17.5 per cent."
DELTA METAL.

"This is an alloy of copper, zinc, and iron having remarkable properties with regard to strength. Propellers have been cast of delta-metal for sea-going steamers, but information as to its trustworthiness and lasting qualities in actual use is wanting. There is reason to believe that it possesses the same advantage of freedom from corrosion and pitting as gun metal and other bronzes. The ultimate tensile strength is from 15 to 23 tons. Elongation, 10 to 20 per cent.

ALUMINIUM BRONZE AND BRASS.

"These materials have been but little used for propellers of large or moderate size. With regard to corrosion, blades of aluminium alloys stand, without doubt, on the same footing as gun metal and other bronzes. Certain precautions have to be taken in casting these alloys, owing to liability to the oxidation of aluminium when in a liquid state. Tests made by the United States Government at the Watertown Arsenal give the following results:

<table>
<thead>
<tr>
<th>Bronze composition.</th>
<th>Elongation, per cent.</th>
<th>Tensile strength, tons per square inch.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper and 8 per cent aluminium and silicon</td>
<td>23.7</td>
<td>26.1</td>
</tr>
<tr>
<td>&quot; 8½ &quot;</td>
<td>26.0</td>
<td>27.2</td>
</tr>
<tr>
<td>&quot; 10 &quot;</td>
<td>3.2</td>
<td>30.4</td>
</tr>
<tr>
<td>Brass composition.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Copper, 3½ aluminium and 33½ zinc</td>
<td>1.6</td>
<td>31.3</td>
</tr>
<tr>
<td>&quot; &quot;</td>
<td>2.5</td>
<td>36.8</td>
</tr>
</tbody>
</table>

These test bars were 2 inches in diameter, the elongation being taken on 15 inches.
MATERIAL BEST SUITED FOR PROPELLER BLADES. 177

COMPARISON OF FOREGOING.

"The following table shows the comparative strength of the materials that have now been referred to:

<table>
<thead>
<tr>
<th>Metal</th>
<th>Ultimate tensile strength, tons per sq. in.</th>
<th>Elongation, per cent.</th>
<th>Weight to break bar 1 in. x 1 in. 12&quot; between supports, cwt.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast iron</td>
<td></td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Steel</td>
<td></td>
<td>31</td>
<td>10</td>
</tr>
<tr>
<td>Gun metal</td>
<td></td>
<td>14</td>
<td>9</td>
</tr>
<tr>
<td>Manganese bronze</td>
<td></td>
<td>15</td>
<td>9</td>
</tr>
<tr>
<td>&quot; &quot; by Bronze and Brass Co.</td>
<td></td>
<td>16</td>
<td>17</td>
</tr>
<tr>
<td>Phosphor-bronze by Phosphor Bronze Co</td>
<td></td>
<td>19</td>
<td>15</td>
</tr>
<tr>
<td>Delta-metal</td>
<td></td>
<td>34</td>
<td>2</td>
</tr>
</tbody>
</table>

"Practical experience has led to steel blades being made 25 per cent thinner at the root than cast iron. About manganese bronze there is a difference of opinion, but 30 per cent thinner than cast iron for the bronze supplied by the Manganese Bronze Company may be said to be the outside that ought to be allowed. For aluminium brass an allowance of 30 per cent may also be assumed, keeping the reduction low on account of the want of ductility of this alloy. Delta-metal may be allowed 25 per cent reduction, and gun metal and phosphor-bronze each a reduction of 10 per cent. (See note x.) Making these allowances, taking the cost per ton as previously noted, the relative cost of blades compared with cast iron may be approximately stated as under.

"The following table (p. 178) gives results very much in favor of cast iron and steel for first cost.

"Now, assume the life of cast-iron blades to be (apart from accident) six years, and that of steel to be four, for a steamer
THE SCREW PROPELLER.

of, say, 5000 tons, having four propeller blades, the combined weight of which is twelve tons. Reducing the above to a common term of four years, the expenditure for cast iron would amount to £192, and for steel £396; the saving in cast iron over the four years would therefore allow of three out of the four blades being renewed in that time after failure due to accidents of such nature as would leave steel intact.

"If so high an allowance for the failure of cast-iron blades is unnecessary, then the opinion mentioned earlier in this paper favorable to the use of this metal is shown to be well founded.

"Turning from the comparison of cast iron with steel to the comparison of steel with the bronzes, there are three matters to which consideration has to be given:

"(1) The larger coal bill entailed by steel blades.

"(2) The necessity of the renewal of steel blades on account of pitting and corrosion.

"(3) The possibility of having to renew the blades of gun metal or other alloys on account of failure from immediate ostensible cause.

<table>
<thead>
<tr>
<th></th>
<th>Cost per ton, based on copper at £80 per ton and tin at £15 per ton.</th>
<th>Equivalent cost of blades for every ton of cast-iron blades.</th>
<th>Cost per ton, based on copper at £15 per ton and tin at £10 per ton.</th>
<th>Equivalent cost of blades for every ton of cast-iron blades.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cast iron</td>
<td>£24</td>
<td>£24</td>
<td>£24</td>
<td>£24</td>
</tr>
<tr>
<td>Steel</td>
<td>38</td>
<td>33</td>
<td>38</td>
<td>33</td>
</tr>
<tr>
<td>Gun metal</td>
<td>130</td>
<td>144</td>
<td>90</td>
<td>100</td>
</tr>
<tr>
<td>Manganese bronze</td>
<td>135</td>
<td>123</td>
<td>127</td>
<td>116</td>
</tr>
<tr>
<td>Phosphor-bronze</td>
<td>170</td>
<td>189</td>
<td>128</td>
<td>142</td>
</tr>
<tr>
<td>Delta-metal</td>
<td>115</td>
<td>110</td>
<td>96</td>
<td>92</td>
</tr>
<tr>
<td>Aluminium brass</td>
<td>145</td>
<td>130</td>
<td>120</td>
<td>108</td>
</tr>
</tbody>
</table>

Note.—The admiralty engineers make gun-metal blades thinner at the root than is stated above, and it would seem that experience has justified them in so doing.
"Under the head of manganese bronze it has been said that for the same speed, with bronze and steel blades, respectively, some 4 per cent difference in power is to be expected. Assuming the same difference to apply to the comparison of steel with the other copper alloys, and taking again the special case of a 5000-ton ship with a coal consumption of 50 tons per day, this 4 per cent difference of power means 2 tons of coal per day, of a mean value of £2. This is a charge against steel blades amounting to £300 per year, taking 150 as the number of days at sea. This amount for 4 years, added to the first cost of steel blades, amounts to £1596 as the total disbursement on their account every four years.

"The price of gun metal blades for the same steamer would amount to £1728 each time they are renewed. If renewals occur at intervals of 52 months, the disbursement for gun-metal blades over this period would be identical with the disbursement for steel over the same period, neglecting the charges for docking in order to renew the gun-metal blades and the cost of zinc for protecting the hull.

"Making a similar calculation for the other alloys, and tabulating the results, it appears that economy in the use of alloys would be the same as in that of steel, if, on an average, one blade had to be renewed every

13 months for gun metal;
11 " " manganese bronze;
17 " " phosphor-bronze;
10 " " delta-metal;
12 " " aluminium brass.

"This is the furthest point to which the comparison has been carried."
THE SCREW PROPELLER.

"Special pains have been taken in obtaining these figures."

It will be observed in the foregoing comparisons that the two elements which tell against the use of steel are the necessity of renewal of steel blades and the extra coal bill as compared with the other alloys. In strength, mild steel is decidedly superior to all other materials mentioned.

Mr. G. W. Manuel (member): "My Lords and Gentlemen—We have to thank Mr. Wallace for his valuable paper, and the able manner he has brought comparisons of the various metals suitable for screw propeller blades before us. Having considerable experience with propellers and their blades, also with the material used for their construction in the mercantile marine, and specially in the Peninsular and Oriental Company's service, I shall endeavor to give you some information from actual practical results in 1880, owing to the number of breakages of cast-iron blades in this company's service, caused in some instances from sheer weakness of design, defects in the material, defects due to operations during casting; also, on the voyage, breakages caused by the propellers striking on the banks and other obstacles in the Suez canal, entering harbors, and at sea—the blades breaking off in various parts, some at the root, others at the middle, and others at a short distance from the points, causing loss of speed, detention, necessity of docking, or in some cases tipping the steamer to ship spare blades, thus entailing unnecessary expense and delay (most undesirable in a mail service); and besides this, considerable loss of efficiency was entailed by the corrosion of the blades by sea water at the forward side and entering edges of the blades, increasing the resistance to cleave the water. In order to reduce these losses, chiefly the breakage of blades,
MATERIAL BEST SUITED FOR PROPELLER BLADES. 181

I advised the use of mild steel as made by Messrs. Vickers & Sons, of Sheffield, and I quite agree with Mr. Wallace that no better material have we got at the present time to resist breakage of propeller blades than mild steel. The result has been that for eight years no blades of this material have broken, though often most severely tried, and steamers have been enabled to continue on the voyage that would have been disabled if the blades had been made of cast iron. The use of this steel entirely settled the question of breakages of blades; but as regards the efficiency of steel blades to withstand the corrosive action of sea water I found that steel corroded more rapidly than cast iron, the blades becoming unfit for use in two or three years. I endeavored to prevent this corrosion at the forward surface of the blades, where corrosion usually takes place, by protecting it with a sheathing of soft brass \( \frac{1}{8} \) in. to \( \frac{1}{2} \) in. thick, carefully fitted, the blade at this part being recessed for the sheathing, so that when finished it was flush with the uncovered part of the blade, about 2 in. being left unprotected at the edges to take the shock when the blade struck any object instead of the sheathing, which would otherwise be damaged and turned over, retarding the vessel's speed. This sheathing was secured to the steel blade by \( \frac{1}{4} \)-in. tap rivets, and when carefully done the duration of the blade's efficiency was increased from two to three years to seven years, and even longer, with slight repair to the sheathing; some, where injured by striking, have had the sheathing removed. There was still considerable corrosion at the forward cleaving edge of the blade left uncovered, as mentioned. This part soon became blunted, and we had to contend with the considerable loss, due to this blunted edge, about \( \frac{1}{4} \) in. flat, cleaving the water at the points at the rate
of 37 knots per hour. To prevent this I designed a soft brass shoe having a fine edge to cover the blade entirely at the cleaving edge, joining it closely to the brass sheathing on the forward side of blade and extending about 5 in. on the after side. Several propeller blades are running with this, giving good results. At the same time, being, as it were, a patch, and striking any hard object, it is liable to get turned over, and thus increase the resistance of the blade itself through the water. The whole may, therefore, be considered a patch to preserve steel blades, and I concluded that while steel blades are almost unbreakable, owing to their rapid corrosion in sea water, it was not the best material for screw propellers. Gun metal has been used in war vessels, also in the French mercantile marine, but was found to be wanting in strength to withstand breakage; and becoming, as it were, doubled up at the points when striking any object, a better material was sought for. Manganese-bronze had been tried in several mail steamers—one of the metals mentioned in Mr. Wallace's paper as possessing an ultimate tensile strength of from 12 to 17 tons per square inch and breaking at a weight of 28 hundred-weight on a bar 1 in. square and 12 in. between supports. These estimates are rather under the real value, as found by practical tests, the tensile strength being 22 to 26 tons per square inch, and weight supported by 1 in. square bar 44 hundred-weight. The directors of the Peninsular & Oriental Company had this metal brought before their notice for trial, but owing to the number of blades made of this material giving way, more particularly in the Atlantic mail steamers and New Zealand steamers, they declined then to adopt it. At the same time this metal appeared to possess such suitable qualities for propeller blades that the question of strength was looked into, and the man-
ager of the Manganese Bronze Company’s works, Mr. Parsons, investigated it, when it was found that the blades of the Atlantic and other steamers which had broken at sea had not been made by the Manganese Bronze Company, although the metal in ingots had been supplied by them, and that none of the blades cast by this company had given way. It appears that great care is required in recasting from the original ingots, the different alloys requiring correction to insure uniform strength. It was therefore decided by the directors of this company, the metal being the most suitable, to fit one of the company’s steamers with a set of manganese bronze blades; and in order to arrive at the correct value of the advantage of this metal over that of steel, it was decided to take off the steel blades that had been in use for four years, the performance of which had been carefully noted, and replace them with the bronze blades cast by the Manganese Bronze Company at Deptford of the same diameter, pitch, and area, and in every way duplicate, except thickness of material and fine cleaving edge. These blades were successfully cast, found to be correct duplicates, and fitted to the S.S. “Ballarat;” and the results from an Australian voyage, compared with similar voyage, circumstances, and displacement, were:

<table>
<thead>
<tr>
<th></th>
<th>Speed, knots</th>
<th>Coal per day, tons</th>
<th>I.H.P.</th>
<th>Slip, per cent</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel blades</td>
<td>12.11</td>
<td>63.8</td>
<td>2828</td>
<td>13.1</td>
</tr>
<tr>
<td>Bronze blades</td>
<td>12.35</td>
<td>55.0</td>
<td>2577</td>
<td>9.7</td>
</tr>
</tbody>
</table>

giving a saving of coal of 700 tons on the round voyage. The directors of this company then decided to fit out four new steamers then building—two with manganese bronze and two
with steel blades, all of similar dimensions and pitch—and I hope further on to give you some particulars of the results. Meanwhile I give you the trial results at measured mile for sea work of the first two steamers, viz.:

<table>
<thead>
<tr>
<th>Vessel</th>
<th>Material</th>
<th>Steam, lbs</th>
<th>Revolutions</th>
<th>I.H.P.</th>
<th>Speed, knots</th>
<th>Slip, per cent.</th>
<th>Displacement, tons</th>
</tr>
</thead>
<tbody>
<tr>
<td>Victoria</td>
<td>Bronze</td>
<td>146</td>
<td>63</td>
<td>6084</td>
<td>16.52</td>
<td>8</td>
<td>8124</td>
</tr>
<tr>
<td>Britannia</td>
<td>Steel</td>
<td>146</td>
<td>64</td>
<td>6203</td>
<td>16.47</td>
<td>10</td>
<td>8040</td>
</tr>
</tbody>
</table>

"It is difficult on long voyages at different periods to get fair comparisons. Those given are practically correct. I may state that in my opinion the superior efficiency of bronze over steel is due to the fine cleaving edge which this bronze gives and maintains from its anti-corrosive qualities, rather than to any reduction of general thickness, for I have found, where blades of cast iron that gave way were replaced by new ones of same material, but about two inches thicker at the root and proportionally up to points, that no loss of speed occurred with same indicated horse-power.

"I will not detain you any longer, but again thank Mr. Wallace for bringing before us such an important paper."

Later experience than the foregoing, with the use of manganese bronze for propeller blades, has given very satisfactory results. The propeller blades of nearly all of the modern United States naval vessels are of this material. Its required manipulation is now so well understood that there is little difficulty with new metal in obtaining reliable castings.
APPENDIX.

EXTRACTS FROM THE SPEED TRIALS OF H. M. S. "IRIS."

By J. Wright, Esq., Vice President Institute of Naval Architects.

The original screws of the "Iris" were four-bladed. The particulars of the speed trials with those screws are in the table annexed. It will be observed that the highest speed obtained with 7503 indicated horse-power was only 16.577 knots, or about a knot below what was expected. Those screws will be referred to as No. 1 screws.

The second trial was with the same screws, but with two alternate blades removed, no alteration being made in the pitch of the remaining two blades on each screw. With these screws, which were not run with maximum power, a speed of 15.726 knots was attained with nearly 2000 indicated horse-power less than when four blades were on each screw.

The third trial was made with screws designated herein as No. 2 screws (two-bladed Griffiths). With 7556 indicated horse-power a speed of 18.587 knots was reached, an increase of over two knots over No. 1 screws. The vibration to the vessels with these screws was greater at all times, and at a speed of over 12 knots it became so excessive that it was decided not to leave them on the vessel.
### Summary of trials.

<table>
<thead>
<tr>
<th>Description</th>
<th>No. 1 Screws</th>
<th>No. 2 and 3 Screws</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power required to overcome resistance of vessel when towed at speed of 15.746 knots</td>
<td>2160 E.H.P.</td>
<td>2160 E.H.P.</td>
</tr>
<tr>
<td>Power to overcome friction of machinery</td>
<td>300 I.H.P.</td>
<td>300 I.H.P.</td>
</tr>
<tr>
<td>&quot; blades (?)</td>
<td>829 &quot;</td>
<td>829 &quot;</td>
</tr>
<tr>
<td>Power not used in propulsion but absorbed by detrimental actions of screws Nos. 1, 2 and 3</td>
<td>2728 &quot;</td>
<td>1298 &quot;</td>
</tr>
<tr>
<td>Total power that was found necessary to drive vessel at 15.746 knots by means of her screws</td>
<td>6800 &quot;</td>
<td>3958 &quot;</td>
</tr>
</tbody>
</table>
APPENDIX.

The "Iris" was subsequently fitted with two four-bladed screws (No. 3) of 2 feet 3 inches less diameter than the No. 1 screws. The tips of the blades were made narrow and curved aft a little. The mean pitch was set at 19 feet 11½ inches. The water had better access to this screw, and the result was that a speed of 18.57 knots was attained, an increase over the No. 1 screw of two knots, and almost equal to the speed attained by the two-bladed No. 2 screw, which had to be taken off on account of excessive vibration. These four-bladed screws showed a slip of 2.97 per cent, and the vibration was very little. The augment of resistance to the vessel occasioned by the detrimental action of these screws amounted to over 32 per cent; or, in other words, the power required to overcome the detrimental action was 32 per cent of the active power employed to drive the vessel at the speed mentioned.

Authorities on the subject of screw propulsion teach us that the effective thrust or efficiency of a screw propeller depends upon the quantity of water acted on in a given time and the sternward velocity impressed upon it, and that it is always preferable to make the sternward velocity of the current small, by adopting a form of screw which will act on the largest possible quantities of water. Indeed, it has been said that if it were not for the objectionable element "surface friction" there would theoretically be no objection to an indefinite extension of blade area. In short, it has been always considered essential to the efficient performance of screws generally that their blade area should be large and that the water must be permitted to flow freely to them.

Now, No. 1 screws with four blades appear to fulfill the above conditions very completely, for the blade area is undoubtedly very large; and as the run of the "Iris" is very fine,
there is no obstruction to an ample supply of water for the screws. Notwithstanding this, however, the results of the trials appear to show that on account of the large diameter of the screws, the shape, and the large area of the blades, combined with a fine pitch and consequent high rotary velocity, an extraordinary large amount of power was absorbed in the surface friction of the blades. But seeing what a large amount of power appears to have been wasted at the highest speeds with No. 1 screws, it is difficult to believe that it can be all credited to this cause, and it is interesting to form an estimate of the amount absorbed in the surface friction of the blades. In our present state of knowledge it is difficult to determine this with even approximate accuracy, but the elaborate and valuable experiments made by Mr. Froude in 1872 on the friction of planes with surfaces of various kinds, traveling edgeways through water, furnish data which at all events enable us to form an estimate of the friction of screw blades under certain conditions and with certain reasonable assumptions. The following are the results of the calculations which have been made:

For No. 1 screws, which were coated with Sims' composition, the coefficient of friction has been taken as a mean of those given by Mr. Froude for a surface varnish and one of fine sand. For a mean length of surface of $\frac{3}{4}$ feet, the width of the blades, this amounts to $0.00564$, and adding ten per cent for edge resistance the coefficient becomes $0.0062$ for these screws.

The polished surfaces of No. 3 screws were probably less smooth than Mr. Froude's tinfoil surface, and about intermediate between it and the varnished surface. For a mean
width of blade of 3 feet the coefficient for this screw, with 10 per cent added as before, would be .0038.

No. 2 screws had a similarly smooth surface, but the mean width of the blade being 4 feet, the coefficient becomes .00365, inclusive of 10 per cent of edge resistance.

Time will not admit of the details of the calculations being given, but the results showing the horse-power expended on blade friction are as follows:

<table>
<thead>
<tr>
<th>No. 1 SCREW (4 BLADES).</th>
</tr>
</thead>
<tbody>
<tr>
<td>At 91.04 revolutions per minute = 1112 horse-power.</td>
</tr>
<tr>
<td>&quot; 84.14 &quot; &quot; &quot; &quot; = 820 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 65.10 &quot; &quot; &quot; &quot; = 420 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 43.35 &quot; &quot; &quot; &quot; = 120 &quot; &quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No. 2 SCREWS.</th>
</tr>
</thead>
<tbody>
<tr>
<td>At 93.25 revolutions per minute = 330 horse-power.</td>
</tr>
<tr>
<td>&quot; 76.93 &quot; &quot; &quot; &quot; = 200 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 59.38 &quot; &quot; &quot; &quot; = 90 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 39.15 &quot; &quot; &quot; &quot; = 30 &quot; &quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No. 1 SCREWS (2 BLADES).</th>
</tr>
</thead>
<tbody>
<tr>
<td>At 88.89 revolutions per minute = 530 horse-power.</td>
</tr>
<tr>
<td>&quot; 81.02 &quot; &quot; &quot; &quot; = 400 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 65.06 &quot; &quot; &quot; &quot; = 210 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 45.07 &quot; &quot; &quot; &quot; = 70 &quot; &quot;</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>No. 3 SCREWS (4-BLADED, SMALLER DIAMETER).</th>
</tr>
</thead>
<tbody>
<tr>
<td>At 97.189 revolutions per minute = 420 horse-power.</td>
</tr>
<tr>
<td>&quot; 85.388 &quot; &quot; &quot; &quot; = 280 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 61.343 &quot; &quot; &quot; &quot; = 110 &quot; &quot;</td>
</tr>
<tr>
<td>&quot; 40.963 &quot; &quot; &quot; &quot; = 30 &quot; &quot;</td>
</tr>
</tbody>
</table>
According to Mr. Froude's experiments with the model of the "Iris," the I.H.P. necessary to overcome the resistance of the hull alone, i.e., the E.H.P., would be 2160, at 15.726 knots speed. We see, therefore, that blade friction alone cannot account for the bad performance of these screws, and that other sources of loss must be looked for.

It is probable that the four-bladed No. 1 screws were more fruitful in eddy making than the two-bladed No. 2 screws, and encountered more loss from this cause; then, as the blades followed so closely on each other, taking, as it were, slices of water about 4\(\frac{1}{2}\) feet thick, they injuriously disturbed the water for each other. That is, each blade probably came in contact with eddies set up by the blade preceding it. The eddies would no doubt be worse where the blades approached the sides of the ship and the heads of the brackets which carry the bearing.

As the blades of the two-bladed No. 2 screws followed each other at intervals about twice as long any eddies set up by one blade would probably be left behind before the other could sweep around them.

The No. 1 screws also interfered more with the stream line motion of the water, closing in at the stern to balance the head resistance.

Mr. Froude estimated that these disturbances by screws in well-formed single-screw ships is equal to from 40 to 50 per cent of increase upon the resistance of the ship when towed at the same speed. With twin screws there is good reason to believe that the augment of resistance has a less percentage than with single screws; but, on the other hand, there is the drag of the brackets or struts and the shaft tubes to be overcome. All this may help us to understand why the original
APPENDIX.

screws of the "Iris" gave results so much worse than the others.

The great amount of attention which has of late years been given to the improvement of the designs of ships and engines has, it is feared, for a time prevented such full consideration from being given to the improvement and proper adaptation of the screw propeller as such an important subject deserves.

Professor Rankine states that through the discoveries of Mr. Scott Russell a vessel can be designed in which the resistance due to the production of waves shall be insensible and all resistance rendered insensible except that which is unavoidable.

THE RESISTANCE DUE TO FRICTIONAL EDDIES.

The resistance is a combination of the direct and indirect effects of the adhesion between the skin of the ship and the particles of water which glide over it, which adhesion, together with the stiffness of the water, occasions the production of a vast number of small rapids or eddies in the layer of water immediately adjoining the ship's surface. The velocity with which the particles of water whirl in those eddies bears some fixed proportion to that with which those particles glide over the ship's surface; hence the actual energy of the whirling motion impressed on a given mass of water at the expense of the propelling power of the ship, being proportional to the square of the whirling motion, is proportional to the square of the velocity of the glidings; in other words, it is proportional to the height due to the velocity of the gliding.

Hence the resistance to the motion of the ship, due to the
production of frictional eddies by a given portion of her skin, is the product of the following factors:

1. The arc of the portion of the ship's skin in question.
2. The cube of the ratio which the velocity of the gliding of the particles of water over that area bears to the speed of the ship, being a quantity depending on the figure of the ship and the position of the part of her skin under consideration.
3. The height due to the ship's speed; that is,

\[
\frac{(\text{speed in feet per second})^3}{64.4}
\]

or

\[
\frac{(\text{speed in knots})^3}{22.6}
\]

4. The heaviness (or weight of a unit of volume) of the water (64.4 lbs. per cubic foot for sea water).
5. A factor called the coefficient of friction, depending on the material with which the ship's skin is coated and its condition as to roughness and smoothness.

The sum of the products of the factors 1 and 2 for the whole skin of the ship has of late been called her augmented surface; and the eddy resistance of the whole ship may therefore be expressed as the product of the augmented surface by the factors 3, 4, and 5, above.

The coefficient of friction employed is that deduced by Prof. Weisbach from the experiments on the flow of water in iron pipes, viz.,

\[ f = .0036 \]

and that value has given results corroborated by practice for surfaces of clean painted iron.

The preceding value of the coefficient of friction leads to the following very simple rule for clean painted iron ships.
APPENDIX.

At ten knots the eddy resistance is one pound avoirdupois per square foot of augmented surface, and varies for other speeds as the square of the speed.

COMPUTATION OF AUGMENTED SURFACE.

To compute the exact augmented surface of a vessel of any ordinary shape would be a problem of impracticable labor and complexity.

The method employed, therefore, as an approximation for practical purposes is to choose in the first instance a figure approximating to the actual figure, but of such a kind that its augmented surface can be calculated by a simple and easy process, and to use that augmented surface instead of the augmented surface of the ship, care being taken to ascertain by comparison, with experiments on ships of various sizes and forms, whether the approximation so obtained is sufficiently accurate.

The figure chosen for that purpose is the trochoid, or rolling-wave curve, extending between a pair of crests, such as $A$ and $B$, Fig. 14. It is found that the augmented surface of a

![Fig. 14]

trochoidal riband of a given length in a straight line and of a given breadth is equal to the product of that length and breadth multiplied by the following coefficient of augmentation:

$$1 + 4 \left(\text{sine of greatest obliquity}\right)^3 + \left(\text{sine of greatest obliquity}\right)^4,$$
the *greatest obliquity* meaning the greatest angle $B, E, D$ made by a tangent $D, E$ to the riband at its point of contrary flexure $D$ with its straight chord $A, B$.

In approximating to the augmented surface of a given ship by the aid of that of a trochoidal riband the following values are employed:

1. For the length $A, B$ of the riband, the length of the ship on the plane of flotation.

2. For the total breadth of the riband, the *mean immersed* girth found by measuring on the body plan the immersed girths of a series of cross-sections and taking their mean by "Simpson's rule," or by measuring mechanically with an instrument the sum of a number of girths and dividing by their number.

3. For the coefficient of augmentation, the mean of the values of that coefficient, as deduced from the greatest angles of obliquity of the series of water lines of the face body, as shown on the half-breadth plan. It is not necessary to measure the angles themselves, but only their sines. The augmented surface is then computed by multiplying together those three factors.

The probable resistance in pounds is computed by multiplying the augmented surface by the square of the speed in knots and dividing by 100 for clean iron ships.

In computing the *probable engine power at a given speed*, allowance must be made for the power wasted through slip, through wasteful resistance of the propeller, and through friction of the engine. The proportion borne by the wasteful power to the effective or *net* power employed in driving the vessel of course varies considerably in different ships, propellers, and engines, but in several good examples it has been
found to differ little from 0.63; so that as a probable value of the indicated power required in a well-designed vessel we may take, net power $\times$ 1.63.

Now an indicated horse-power is 550 foot-pounds per second, and a knot is 1.688 feet per second; and therefore an indicated horse-power is $\frac{550}{1.688} = 326$ knot-pounds nearly, or 326 pounds of gross resistance overcome through one nautical mile in an hour. If we estimate, then, the net or useful work done in propelling the vessel as equal to the total work of the steam divided by 1.63, we shall have $\frac{326}{1.63} = 200$ knot-pounds of net work done in propulsion for each indicated horse-power.

Hence the following rule: Multiply the augmented surface in square feet by the cube of the speed in knots and divide by 20,000; the quotient will be the probable indicated horse-power.

The divisor 20,000 in this rule expresses the number of square feet of augmented surface which can be driven at one knot by one indicated horse-power. It may be called the coefficient of propulsion. It is, of course, understood that the exact coefficient of propulsion differs in different vessels, according to the smoothness of the skin, the nature of its material, and the efficiency of the engines and propellers, being the greatest in the most favorable examples.

In clean iron ships, with no evident fault in shape or dimensions or in the propeller and engine, it has been found on an average to be somewhat above 20,000, and this value 20,000 may be taken as a probable and safe estimate of the coefficient of propulsion in any proposed vessel designed on good principles.
THE SCREW PROPELLER.

In vessels sheathed with copper or coated with smooth pitch the coefficient of propulsion is unquestionably greater; but in what precise proportion it is at present difficult to say, owing to the scarcity of experimental data.

COMPUTATION OF PROBABLE SPEED.

When the augmented surface of a ship has been determined, her probable speed with a given power is computed as follows:

Multiply the indicated horse-power by the coefficient of propulsion (say for clean iron ships 2000), divide by the augmented surface, and extract the cube root of the quotient. This gives the probable speed in knots.

Example 1.—Calculation of probable speed of H. M. S. "Warrior:"

Displacement on trial .......... 8997 tons.
Draught of water

<table>
<thead>
<tr>
<th>Water lines</th>
<th>Sine of obliquity</th>
<th>Square of sine</th>
<th>4th power of sine</th>
</tr>
</thead>
<tbody>
<tr>
<td>L. W. L.</td>
<td>.370</td>
<td>.1369</td>
<td>.01874</td>
</tr>
<tr>
<td>2. W. L.</td>
<td>.315</td>
<td>.0992</td>
<td>.00984</td>
</tr>
<tr>
<td>3. W. L.</td>
<td>.290</td>
<td>.0841</td>
<td>.00707</td>
</tr>
<tr>
<td>4. W. L.</td>
<td>.265</td>
<td>.0702</td>
<td>.00492</td>
</tr>
<tr>
<td>5. W. L.</td>
<td>.235</td>
<td>.0552</td>
<td>.00304</td>
</tr>
<tr>
<td>6. W. L.</td>
<td>.165</td>
<td>.0272</td>
<td>.00074</td>
</tr>
<tr>
<td>Keel</td>
<td>.000</td>
<td>.0000</td>
<td>.00000</td>
</tr>
</tbody>
</table>

Means............. .0674 .00583

\[1 + (4 \times .0674) + .00583 = 1.275\], coefficient of augmentation.
## APPENDIX.

<table>
<thead>
<tr>
<th>Half girths from body-plan, feet.</th>
<th>Simpson's multipliers</th>
<th>Products.</th>
</tr>
</thead>
<tbody>
<tr>
<td>21.0</td>
<td>1</td>
<td>21.0</td>
</tr>
<tr>
<td>27.2</td>
<td>4</td>
<td>108.8</td>
</tr>
<tr>
<td>30.8</td>
<td>2</td>
<td>61.6</td>
</tr>
<tr>
<td>34.6</td>
<td>4</td>
<td>138.4</td>
</tr>
<tr>
<td>38.8</td>
<td>2</td>
<td>77.6</td>
</tr>
<tr>
<td>41.5</td>
<td>4</td>
<td>166.0</td>
</tr>
<tr>
<td>42.6</td>
<td>2</td>
<td>85.2</td>
</tr>
<tr>
<td>44.0</td>
<td>4</td>
<td>176.0</td>
</tr>
<tr>
<td>44.0</td>
<td>2</td>
<td>88.0</td>
</tr>
<tr>
<td>44.0</td>
<td>4</td>
<td>176.0</td>
</tr>
<tr>
<td>43.3</td>
<td>2</td>
<td>86.6</td>
</tr>
<tr>
<td>42.1</td>
<td>4</td>
<td>168.4</td>
</tr>
<tr>
<td>40.3</td>
<td>2</td>
<td>80.6</td>
</tr>
<tr>
<td>38.1</td>
<td>4</td>
<td>152.4</td>
</tr>
<tr>
<td>36.0</td>
<td>2</td>
<td>72.0</td>
</tr>
<tr>
<td>35.0</td>
<td>4</td>
<td>140.0</td>
</tr>
<tr>
<td>32.0</td>
<td>1</td>
<td>32.0</td>
</tr>
</tbody>
</table>

Divide by ........................................ 3)1830.6 sum
Divide by $\frac{1}{3}$ number of intervals .......... 8)610.2
Mean immersed girth ................................ 76.3
$\times$ length .................................... 380
Product ........................................... 28994
$\times$ coefficient of augmentation ............... 1.275
Augmented surface ................................ 3697 sq. ft.
Indicated horse-power on trial ..................... 5471
$\times$ coefficient of propulsion .................. 20000
Divide by augmented surface ....................... 36979)109,420,000 product
Cube of probable speed .......................... 2959
Probable speed computed .......................... 14.356
Actual speed on trial ............................ 14.354
Difference ....................................... 0.002
Example 2.—H.M.S. "Fairy" will next be taken as an example, on account of the great contrast in size between her and the "Warrior."

Displacement .................. 168 tons.
Draught of water .................. 4.83 feet.

<table>
<thead>
<tr>
<th>Water lines</th>
<th>Sine of obliquity</th>
<th>Square of sine</th>
<th>4th power of sine</th>
</tr>
</thead>
<tbody>
<tr>
<td>L. W. L.</td>
<td>.23</td>
<td>.0529</td>
<td>.0028</td>
</tr>
<tr>
<td>2. W. L.</td>
<td>.22</td>
<td>.0484</td>
<td>.0023</td>
</tr>
<tr>
<td>3. W. L.</td>
<td>.21</td>
<td>.0441</td>
<td>.0019</td>
</tr>
<tr>
<td>4. W. L.</td>
<td>.17</td>
<td>.0289</td>
<td>.0008</td>
</tr>
<tr>
<td>Keel</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Means ................. .1304 .0015

\[ 1 + (4 \times .0304) + .0015 = 1.123, \text{ coefficient of augmentation.} \]

Length of water line .................. 144 feet
\[ \times \text{ mean immersed girth (measured mechanically with an instrument)} \] ........................ 19
\[ \times \text{ coefficient of augmentation} \] ........................ 1.123
Augmented surface .................. 3072 sq. ft.
Indicated horse-power on trial ........ 364
\[ \times \text{ coefficient of propulsion} \] ........................ 20000
\[ \div \text{ augmented surface} \] ........................ \( \frac{3072 \times 7,280,000}{3072} \)
Cube of probable speed ............ 2370
Probable speed, computed ............ 13.333 knots
Actual speed on trial ............... 13.324
Difference ........................ .009
RULES APPLICABLE TO FEATHERING PADDLES WORKING IN UNDISTURBED WATER.

RULE 1.—To determine the proper area of a pair of feathering floats for paddle wheels of a given vessel to be driven at a given speed with a given slip, the paddles working in water that is not sensibly disturbed by the ship, calculate the probable resistance of the ship at the intended speed; divide that resistance by the intended speed of the centres of the paddle floats relatively to the water (or slip) by their intended speed relatively to the vessel (or sum of the speed of the vessel and slip), and by the mass of a cubic foot of water, viz., for resistance in pounds and velocities in feet per second, 2, and for resistance in pounds and velocities in knots, 5$. The quotient will be the required area in square feet.

RULE 2.—To solve the same question when the vessel is so proportioned that her resistance depends on her "augmented surface" only, divide the augmented surface by the ratio which the intended slip bears to the intended speed of the vessel, by the ratio which the intended speed of the centres of the floats bears to the intended speed of the vessel, and by the constant, 566. The quotient will give the area required.

In different states of ship's bottoms the constant divisor may have different values, which in the present state of an experimental knowledge may be taken as ranging between 500 and 600.
Example from the steamer "Admiral."

Augmented surface .......................... 8560 sq. ft.
Intended ratio of slip to speed of vessel .......... 0.28
Intended ratio of speed of centres of floats to speed of vessel ......................... 1.28

\[
\frac{8560}{0.28 \times 1.28 \times 566} = 40 \text{ sq. ft.,}
\]

the required area of a pair of floats; giving 20 sq. ft. for a single float. The actual floats measure 7 ft. broad and 3 ft. deep.

Rules for the Screw.

The pitch of the screw is the length, measured along the axes, of a complete turn; and the speed of the propeller, in the case of a screw, means the pitch multiplied by the number of turns in a unit of time. The case of a screw having different pitches at different parts of its surface will be considered further on. The effective area or disk of a screw propeller is measured on a thwart-ship plane, and has for its outer boundary the circle swept by the tips of the blades and for its inner boundary the outline of the boss. The calculation of the proper disk area for a screw by following theoretical principles precisely is of great length and complexity, and for its details reference must be made to the Translations of the Institute of Naval Architects for 1865. For ordinary practical purposes, when the circumference of the screw is not less than \( \frac{1}{4} \) times nor more than \( 3\frac{1}{2} \) times its pitch, the following approximate rule is sufficient:

Rule 1.—Divide \( \frac{9}{10} \) of the pitch by the circumference and subtract the quotient from 1. The remainder expresses the ratio of the area of an equivalent feathering paddle to that of the screw. Then find, by rules 1 and 2, the area of a
feathering paddle required to drive the ship at the certain speed with the given real and apparent slip. Divide that area by the ratio already found; the quotient will be the required effective area of the screw, and the efficiency of the screw, neglecting friction, will be very nearly equal to that of the paddle. To allow for the friction of the screw in the water about 2\(\frac{1}{4}\) or 3 per cent may be subtracted from the efficiency in ordinary cases.

**Rule 2.** To find the diameter, circumference, and pitch of a screw whose effective area is given: From 1 subtract the square of the ratio in which the diameter of the boss is to be less than that of the screw, and divide the effective area by the remainder; the quotient will be the area, including the boss. Multiply that area by 1.273 (or multiply by 14 and divide by 11); the square root of the result will be the required diameter. That diameter, multiplied by 3.1416 (or by \(\pi\)), will give the circumference, which, divided by the predetermined ratio of the circumference to the pitch, will give pitch.

**Example** (from H. M. S. "Warrior").—Area of equivalent feathering paddle (as already computed in the preceding article), \(\frac{1}{16}\) of the augmented surface of the ship.

\[
\frac{\text{Circumference of screw}}{\text{Pitch}} = 2\frac{1}{4}, \text{ nearly.}
\]

\[
1 - \frac{0.8}{2\frac{1}{4}} = 1 - 0.32 = 0.68
\]

\[
\text{ratio of area of equivalent feathering paddle to area of screw. Then}
\]

\[
\frac{1}{120 \times 0.68} = \frac{1}{81.6}
\]

\[
\text{ratio of required effective area of screw disk to augmented surface.}
\]

Augmented surface, \(\frac{36,979}{81.6} = 453\) square feet, effective area.
The diameter of boss to be one fourth of that of screw; then, \(453 \times \frac{1}{4} = 483\) square feet, total area.

\[\sqrt{483} \times \frac{1}{4} = 24.8\] feet, calculated diameter.

The actual diameter is \(24\frac{3}{8}\) feet;  
Calculated circumference \(77.91\) feet;  
Actual circumference \(76.97\) \(\frac{\text{"}}{30.00}\).

**Rule 3.** To find the mean moment of torsion on the propeller shaft (excluding friction of bearings): Multiply the thrust of the screw by the pitch and divide by 6.2832. (The thrust may be taken, for the purposes of this calculation, at about 3 per cent greater than the resistance of the vessel.)

*Example from H. M. S. "Warrior."

\[
\text{Thrust, } 86,000 \text{ lbs.} \times \frac{\text{pitch, } 30 \text{ feet}}{6.2832} = 410,000 \text{ foot-lbs., nearly.}
\]

**Rule 4.** To design a pair of twin screws, equivalent to a given single screw: Make the pitch, diameter, and all the other dimensions of the twin screws \(
\frac{1}{\sqrt{2}} = 0.7071
\) of the corresponding dimensions of the single screw. The twin-screws must then make \(\sqrt{2} = 1.4142\) revolutions for each revolution of the single screw, and the mean moment of torsion on each of the shafts will be \(
\frac{1}{2\sqrt{2}} = 0.3535
\) of that on the shaft of the single screw.

The preceding rules are based on the supposition that the screw is free from any defects in construction or position tending to impair its efficiency. What such defects are, and how they are to be avoided, will be considered further on.
Experiments have been made by Mr. Rigg upon a screw fitted with an appendage astern of it, which consists of a set of radiating fixed blades, so shaped as to receive the obliquely moving streams of water which come from the screw and turn those streams into a direction right aft. If the action of this apparatus were theoretically perfect, its effect would be to make the screw equivalent in thrust and efficiency to a feathering paddle of the same area.

RULES FOR RADIAL PADDLES.

The sectional area of the stream driven aft by a common or radial paddle is the product of the breadth of the paddle into the greatest depth of immersion of its lower edge.

The slip differs at different parts of the stream. For purposes of calculation, it is convenient to take the slip at the greatest depth below the surface. The apparent slip at that point is the excess of the speed of the outer edges of the paddles above the speed of the vessel, and it may have to be corrected to find the real slip.

The action of radial paddles on the water is very complex. Each particle of the stream that is driven aft quits the paddle at its outer edge and is driven obliquely upward with the exception of the lowest layer of particles only, which are driven horizontally. The following rules are founded on a mathematical investigation, which is confirmed in a general way by such comparisons as exist between the performances of the same vessels with radial and with feathering paddles; but so few of these comparisons have been conducted as to give definite results that the rules must be regarded as provisional approximations only.

RULE 1. To find the sectional area of the stream to be driven back: Proceed as for feathering paddles, supposing the
slip of the feathering paddles to be equal to that of the lower edges of the radial paddles, and having due regard to the effect which the varying immersion of the vessel may have upon the depth of the stream.

Rule 2. To find the efficiency: Multiply the efficiency of the corresponding feathering paddle by the square root of the fraction of the outer radius of the radial paddle-wheel which stands above water. The reciprocal of that square root is, of course, the ratio in which the power required to drive the paddle-wheels is greater than that required to drive the corresponding feathering paddles. For example, if a radial paddle-wheel has 0.36 of its outer radius immersed, so that 0.64 of that radius is above water, its efficiency will be nearly \( \sqrt{0.64} = 0.8 \) of that of the corresponding feathering paddle, and \( \frac{1}{0.8} = 1.25 \) times the power will be required to drive it at the same speed, the speed being measured at the centres of the feathering paddles and at the outer edges of the radial paddles.

Rule 3. To compute the mean moment of torsion on the paddle shaft (exclusive of friction): Multiply half the estimated resistance of the wheel by the outside radius of the paddles and by the square root of the ratio in which that outside radius is greater than the height of the axis above the surface of the water.

Dimensions of Shafts.

It has been shown how to compute the mean effective moment of torsion on the shaft, either of a screw or of a paddle-wheel. The strength of the shaft, however, must be suited to bear, not merely the mean effective moment, but
the greatest total moment of torsion, and in determining the greatest moment from the mean moment the following rules are to be observed:

**Rule 1.** For a screw-propeller shaft: Add three tenths to the mean effective moment for friction; this will give the mean total moment. Then, to find the greatest total moment, multiply by one or other of the following factors:

- If the screw is driven by a single engine. ............. 1.57
- " " " " " pair of engines. ........ 1.11
- " " " " " three engines. ...... 1.05

**Rule 2.** For paddle shafts, either in one piece for both paddle-wheels or coupled amidships by an intermediate shaft or otherwise: In this case regard must be had to the fact that, while the ship rolls in a seaway, one paddle-wheel may occasionally be lifted wholly out of the water, so as to throw the whole power of the engines on the shaft of the other wheel. Therefore the mean twisting moment on the shaft of one paddle-wheel is, in the first place, to be doubled. Then add one-fourth for friction and multiply by one or other of the following factors:

- If the paddle-wheels are driven by a single engine. ...... 1.57
- " " " " " pair of engines. ..... 1.11
- " " " " " three engines. ..... 1.05

**Rule 3.** The greatest twisting moment on an intermediate shaft is that due to one engine only; nevertheless the changes of stress on the intermediate shaft are so irregular and so sudden that it is found necessary in ocean steamers to swell its diameter in the middle to about one and a sixth that of the paddle shafts.

**Rule 4.** For each shaft of a pair of independent paddle-
wheels, take half the moment given by Rule 2, as the case may be, observing that the factor expressing the ratio in which the greatest exceeds the mean total moment depends on the number of engines that drive each independent shaft; so that, for example, if a pair of independent paddles are each driven by a single engine, the multiplier is 1.57.

**Rule 5.** To deduce the mean total moment from the total indicated power of the engines: Multiply the I.H.P. by 33,000 for foot-lbs. per minute and divide by 6.2832 multiplied by the number of revolutions per minute; the quotient will be the mean total moment in foot-lbs.; or, otherwise, multiply the I.H.P. by 8250 and divide by the number of revolutions per minute.

**Rule 6.** To find the diameter of a shaft suited to bear a given greatest twisting moment: Reduce the moment to inch-lbs.; divide it by 0.196 multiplied by the working modulus of stress in pounds on the square inch; the cube root of the quotient will be the required diameter in inches.

For various comparisons of the diameters of shafts used in practice, with the estimated greatest twisting moments to which they are exposed, it appears that the working modulus of stress ranges from 8000 lbs. to 10,000 lbs. on the square inch; so that the divisor in the rule ranges from \(0.196 \times 8000 = 1568\) to \(0.196 \times 10,000 = 1960\).

The higher values of the modulus are, on the whole, from examples of paddle shafts; the lower, from examples of screw shafts.

Considering the comparative weakness of iron in large forgings, the preceding values of the modulus of working stress correspond to the factors of safety of from \(5\frac{1}{2}\) to \(4\frac{1}{2}\); the smaller factors being for paddle shafts. The durability of paddle shafts
under those circumstances is probably due to the fact that they are only occasionally exposed to the severe stress which takes place when one wheel is lifted wholly out of the water.

**Example of Rules 1 and 6.**

Mean effective twisting moment on a screw shaft ........................................ 410,000 ft.-lbs.
Add three tenths for friction .................................................. 123,000 "
Mean total moment ................................................................. 533,000 "
Shaft driven by a pair of engines multiply by .................................... 1.11

Greatest total moment .............................................................. 591,630 ft.-lbs.
To reduce to inch-lbs. multiply by ............................................. 12
Taking as the working modulus 8000 lbs.\{ 1568\}7,099,560 in.-lbs.
on the sq. in., divide by 8000 × 0.196 = \{ \]
Cube of diameter ................................................................. 4,528
Required diameter ............................................................... 16.58 inches

**Example of Rules 2 and 6.**

Given mean effective twisting moment exerted
by one paddle-wheel ......................................................... 23,000 ft.-lbs.
Add one fourth for friction .................................................. 5,750 "
Mean total moment for one wheel ............................................. 28,750
Multiply by .......................................................... 2
Mean total moment for both wheels ......................................... 57,500
Paddle-wheels driven by a pair of engines multiply by .............. 1.11

Greatest twisting moment ..................................................... 63,825 ft.-lbs.
To reduce to inch-lbs. multiply by ............................................. 12
Taking 9000 lbs. on the sq. in. as the working modulus, divide by 9000 × 0.196 = \{ \]
Cube of diameter ................................................................. 434
Required diameter ............................................................... 7.57 inches
THE SCREW PROPELLER.

Example of Rules 4 and 6.

Independent paddle shaft for same vessel:
Mean total moment for one wheel, as before... 28,750 ft.-lbs.
Multiply by .................................................. 1.57
Greatest twisting moment .......................... 45,137.5 "
To reduce to inch-lbs. multiply by ............ 12
Divide, as before, by ................................. \( \frac{1764 \times 541,650}{1764} \) in.-lbs.
Cube of diameter ...................................... 307
Required diameter ...................................... 6.75 inches

Example of Rules 5 and 6.

Indicated horse-power of engines........... 5,471
Multiply by .................................................. 33,000
Divide by .................................................. \( \frac{6.2832 \times 180,543,000}{100} \) per min.
Divide by number of revolutions per min-
ute .................................................. 54\( \frac{28,734,000}{54} \) nearly
Mean twisting moment ............................... 529,600 ft.-lbs.
Multiply (as in Rule 1) by ......................... 1.11
Greatest total moment .............................. 587,856
Multiply by .................................................. 12
Divide (as in the first ex.) by 1568) .......... \( \frac{7,054,272}{1568} \)
Cube of diameter ...................................... 4.499
Required diameter ...................................... 16.52 inches.
APPENDIX.

PARTS, FIGURE, AND DIMENSIONS OF SCREW PROPELLERS.

As to the pitch, circumference, and disk area, the following statements have to be made in addition:

The form of screw propeller in general use consists of a boss fitted on the end of the propeller shaft, with two or more blades arranged symmetrically around it, so as to balance each other. Single-bladed screws have been proposed, but are bad, because the weight and inertia of the single blade, being unbalanced, tend to produce vibrations and to overstrain the propeller shaft and the framing of the ship. The shaft sometimes passes through the boss, and has an after bearing in the after stern post or rudder post. When there is no such bearing the screw is said to overhang. Sometimes an overhung screw projects abaft the rudder, the shaft passing through an oval eye in the rudder-stock. The boss is of small diameter (say about twice the diameter of the shaft), and is usually cylindrical; if of large diameter (say from one fourth to one half of the diameter of the screw), it is spherical or spheroidal. Should a large boss be made cylindrical, it is advisable to give it spherical ends.

It has been proved that a large spherical boss (as in Griffiths screw, Plate VIII of from one fourth to one third of the diameter of the screw, and sometimes even more, is favorable to efficiency, for it meets with very little resistance from the water, and occupies the place of a part of the screw blades which meets with much resistance and has little effect in propulsion. It is also practically convenient, because its blades can be made separate from it, and are then easily fixed and unfixed and replaced when lost, and may also have their position adjusted, so as to alter the pitch slightly, if required. The centre of the boss has usually a round hole through it turned to fit the shaft, upon which it is fastened with keys.
wedged endwise into longitudinal grooves, and sometimes with a transverse bolt in addition to the keys. The most common number of blades in the screws of ships of war is two; of merchant ships, three or four. Every time a screw blade passes the stern post there is a slight shock given to the after part of the ship, tending to make it vibrate horizontally. This is caused by the impediment offered by the stern post to the water which the screw carries along with it. When the screw has an even number of blades, as two or four, each pair of opposite blades produces shocks in opposite directions at the same time, which partially counteract each other's effects; when the number of blades is odd, as three, shocks take place in opposite directions alternately, and the vibration produced is greater than with an even number of blades.

A screw of two blades can be disconnected from the shaft and drawn up in the screw well in the stern of the vessel when she is to go under canvas alone. When the screw is simply disconnected from the engine and allowed to revolve freely in the water, Mr. Scott Russell, by experiment, found that the loss of speed in a vessel under sail was a half-knot out 10 knots, showing that between 14 and 15 per cent of the whole resistance was produced by the dragging of the screw and the friction of its bearings. Another mode of lifting the screw out of the water (adopted by Mr. Russell) is to have a universal joint in the shaft, allowing the screw to be hauled up into the stern. The moveable part of the shaft works in a narrow longitudinal well amidships.

The length of a single blade of the screw is measured parallel to the axis of the screw; and its proportion to the pitch is also the proportion borne by the arc of the disk occupied by that blade to a whole circumference.
APPENDIX.

The aggregate length of all the blades of a screw is the sum of their several lengths measured parallel to the axis; and its proportion to the pitch is also the proportion borne by the whole angular space on the disk occupied by the blades to a whole circumference.

It is known by practical experience (and also by the experiments of MM. Bourgeois and Moll, for an account of which in English see Mr. Bourne's treatise on the Screw Propeller) that when the number of blades does not exceed four the most advantageous aggregate length lies between 0.45 and 0.27 of the pitch, and that in ordinary cases its value is about one third of the pitch; also that such aggregate length may be divided equally among the blades, whether two, three, or four in number.

When the number of blades is increased beyond four, the same aggregate length is no longer sufficient.

In the common screw the length at every part of each blade is the same; so that the blades, when projected on a fore-and-aft plane, appear of a rectangular figure, except that the corners are in general rounded, that having been found to save power.

In the form of blade arrived at by Mr. Griffiths by means of numerous and varied trials the length is not uniform at different points of the same blade, but is diminished toward the tip and toward the root. For example, in a two-bladed screw the lengths are as follows:

At the tip, about ......................... 0.07 of the pitch.
At the longest part, being about \( \frac{1}{4} \) of the

radius from the axis ..................... 0.167  "
At the root, about ......................... 0.11  "
Mean length of one blade .................. 0.12  "
Mean aggregate length of the two blades... 0.24  "
the outlines being rounded. The effect of this on the present screw, whose circumference is $2\frac{1}{3}$ times its pitch, is to make the breadth of the blade, as projected on a thwartship plane, nearly the same at the top and at $\frac{4}{9}$ of the radius from the axis, and equal to one sixth of the pitch nearly.

The tips of the blades are slightly bent toward the vessel, so as to present a somewhat convex surface to the water they act upon. This is better seen in screw blades which are shown in Plate VIII.

In order to work with the greatest possible efficiency the screw should have ample clearance in its aperture both afore and abaft. The length of the screw aperture should be double the greatest length of the screw. The chief advantage of increasing the number and diminishing the length of the blades of a screw is to obtain more clearance between the stern post and rudder post, or, if a balanced rudder is used, its forward edge should be bearded, to let the water move freely toward and from the screw. The top of a screw should, as far as possible, be well immersed.

Mr. Woodcraft's gaining pitch screw, in which the pitch gradually increases from the leading to the following edge, has greater efficiency than the screw of uniform pitch, but in what proportion it is difficult to state with precision.

The following is theoretically the best way of designing a gaining-pitch screw.

In Fig. 15 suppose the paper to present the development upon a flat surface of a cylinder of the circumference $A, B$, described about the axis of the screw. Let $B, C$ ($B$ being forward and $C$ aft) be the distance run by the ship while the screw makes one revolution (being the pitch of an imaginary screw turning with the real screw, but working in a solid). Join $C, A$ ;
then, if $A$ is the leading edge of a blade, $A, C$ should be a tangent to that edge, which ought to touch the imaginary screw of the pitch $B, C$ in order that it may gently cleave the water

![Diagram](image)

**Fig. 15.**

without striking it. Produce $B, C$, and in it take any distance $B, D$ that may be determined upon as the mean pitch of the screw blades.

Let $B, C$ be the length of the blade parallel to the axis; draw $E, F$ parallel to $B, A$, cutting $A, D$ in $F$; then $F$ is the following edge of the blade—that is to say, the leading edge and following edge are to lie in one screw surface of the uniform pitch $B, D$. Draw $A, G$ perpendicular to $A, C$; bisect $A, F$, and through the point of bisection draw $H, G$ perpendicular to $A, D$, cutting $A, G$ in $G$. About the centre $G$ draw the circular arc $A, H, F$, which will touch $A, C$; that arc will be the proper section for the screw blade upon the cylindrical surface whose circumference is $A, B$; and in the same manner a series of sections may be found at a series of cylindrical surfaces.

Supposing that a screw thus designed were executed with
absolute precision, and that all the particles of water moved by it were acted on similarly to those in contact with the blades; its efficiency would be equal to and its thrust double that of a screw of the same area and of the uniform pitch $B, D$ turning at the same speed, and with the slip $C, D$ at each turn. But it is very doubtful whether so good a result as this can be practically realized.

Rounding off the corners of screw blades and making them so that their edges present continuous convex curves, whether projected on a fore-and-aft plane or on a thwartship plane, diminishes the risk of their getting fouled by floating bodies.

In order to diminish the resistance to the motion of a screw blade through the water its edges should be thin and its surfaces smooth. As the figure of the face of the blade or surface which passes on the water in driving ahead is determined by the pitch and other dimensions, the thin edges are to be obtained by giving a suitable convex form to the back of the blade, so that the thickness may gradually diminish from the middle to the edges. The thickness in the middle depends on conditions of strength, which will be considered in the next article. To maintain smoothness of surface bronze and yellow-metal screws should be polished and iron screws should be coated with zinc and burnished.

In vessels with a pair of twin screws one screw should be right-handed and the other left-handed, so that, being turned in contrary directions in driving ahead, they may counteract each other's tendencies to produce lateral vibration.

For the description of endless varieties of forms of screw propellers and experiments as to their action reference must be made to Mr. John Bourne's work, already cited.
STRENGTH OF SCREW PROPELLERS.

The conditions to be fulfilled by a screw blade as to strength are the following: First, it should be strong enough, at least, to bear its share of the twisting moment on the shaft; secondly, it should be weaker than the shaft against the straining action of a blow of a hard body upon the tip of the blade; and the moment of resistance of the blade in going from the boss toward the top should diminish faster than the moment of such a blow; that is, faster than the distance from the tip to which that moment is proportional. This latter condition (the importance of which has been pointed out and acted upon by Mr. Scott Russell and Mr. J. R. Napier) is laid down in order that, if the tip of the blade strikes against a hard substance, the fracture, if any, may take place neither in the shaft nor in the blade near boss, but as near as possible to the tip, so that the broken blade may still be useful for propulsion and capable of being repaired.

Those conditions may be fulfilled in the following manner: Suppose the developed cross-section of the blade as made by a cylindrical surface to be of the general character of that represented in Fig. 16, $A, B$ being the face and $A, C, B$ the back, the curve of the back being nearly a semi-ellipse, and transverse strength, or moment of resistance, nearly the same as that of an elliptical cross-section of the same thickness; then considering, first, the moment of resistance of a circular or elliptical section to bending is half of its moment of re-
istance to twisting; and, secondly, that the working modulus of stress for the material of the blades is probably about half of that for the material of the shaft (say from 4000 to 4500 lbs. on the square inch for the blades, and from 8000 to 9000 lbs. on the square inch for the shaft), the following rule will give the proper relation between the scantlings of the shaft and blades:

**Rule 1.** Multiply the cube of the diameter of the shaft by 4 and divide by the number of blades and by the length of a blade at its root measured parallel to the axis; the square root of the quotient will be the greatest thickness of a blade at the axis of the shaft, supposing the blade to be continued inward so far.

The aggregate strength of the blades against the reaction of the water will then be equal, or nearly equal, to that of the shaft, and the strength of any one blade will be a fraction of that of the shaft.

When the blades are made of the same material as the shaft, the multiplier is two, instead of four.

If the cross-section of the back of the blade is to be parabolic, as in Fig. 16a, instead of semi-elliptic, as in Fig. 16, add one eighth to the thickness as given in the rule for the semi-elliptic section.

Then to find the greatest thickness of the blade at other points proceed as follows:

**Rule 2.** In Fig. 17 let the point $A$ be the axis of the shaft, the circle $D, G, H$ the outline of the boss on a thwartship plane, and $A, B$ a radius drawn from the axis to the top ($B$) of a blade. Perpendicular to $A, B$ draw $A, C$ equal to the thickness found by Rule 1 and draw the straight line $C, D, B$. Then the perpendicular distance of $D, B$ from $G, B$ at any
point will represent the proper thickness for the middle of the blade at the corresponding point, measured in a direction normal to its face; and if this rule be observed, the moment of resistance of the blade will diminish more rapidly in going from $G$ toward $B$ than the moment of a force applied at $B$.

Should it be considered not advisable to reduce the thickness of the blade at the tip below a certain least thickness, set off the least thickness $(B, E)$ at the tip, and through $E$ draw a straight line parallel to $B, G$; then connect the two straight lines from $E$ to $D$ with each other at $F$ by means of a short curve. The point $F$ will be that at which the blade will probably break on receiving a blow from a hard body at or near the tip.

**Example.**

Suppose diameter of shaft $= 16$ inches;
Length of root of blade parallel to axis $= 40$ "
Number of blades $= 2$ "

Then by Rule 1, $\frac{16^2 \times 4}{2 \times 40} = 204.8$, and $\sqrt{204.8} = 14.3$ in.

$= A, C$, in Fig. 4.

Suppose, further, that radius of boss $A$, $G = \frac{1}{4} A, B$; then by Rule 2 the thickness of the middle of the blades at $G$ is found to be 10.7 inches, diminishing uniformly towards the tip.
Fittings and Supports of Screw Propeller Shafts.

The shaft of the screw propeller passes inboard through the shaft pipe or shaft tube, which in iron vessels is of cast iron, and in wooden vessels of galvanized cast iron or of yellow metal. The shaft pipe is accurately turned and bored inside at and near the ends, so that the shaft may rotate in it easily. In wooden vessels it extends through the stern post and dead wood, in which a hole is bored to receive it. In iron vessels it extends through an eye in the stern post and through the part corresponding to the dead wood. The frames have circular arcs formed in them to curve round the pipe. The forward end of the pipe passes through a water-tight bulkhead, and contains a stuffing box.

From that bulkhead to the water-tight after bulkhead of the engine-room extends a compartment containing the shaft, called the shaft alley or tunnel, which is wide enough and high enough to allow workmen easy access to the shaft. Where the shaft and ship are both large, the shaft-alley is from 6 to 8½ feet high and 9 feet broad, but smaller dimensions are sufficient in smaller vessels. The sides of the shaft alley are sometimes water-tight bulkheads, and its roof is a water-tight platform. Frequently the bottom of the tunnel also is a water-tight platform.

In some vessels the thrust of the propeller shaft in driving ahead is communicated to the ship by means of a flat steel pivot at the forward end of the shaft pressing against a step composed of four or five bronze disks, shaped alternately like convex and concave lenses, the box containing the disks be-
ing pressed aft against the pivot by means of an adjusting wedge. In driving astern, the thrust of the screw is exerted through a pivot at the after end of the shaft against an elm or lignumvitæ step in the rudder post.

According to the construction which is now most common, the thrust of the shaft, whether exerted ahead or astern, is communicated to the ship by means of a journal of a length equal to about twice the diameter of the shaft, and having a number of fillets or collars projected from it and fitting into corresponding circular grooves in a bronze bush that embraces the shaft journal. That bush is carried by a strong plumber block securely framed to the bottom of the ship. The plumber block is so placed as to be easily accessible for the purpose of lubricating the bearing. The collars are usually about an inch broad and an inch deep, and have an inch clear space between them.

Another way of receiving the thrust of the screw is to let its boss bear against a collar which surrounds the eye of the stern post, and which has a bearing surface of wood with the fibres endwise to the thrust, formed by driving a circle of elm or lignumvitæ wedges into a ring-shaped groove. The water lubricates this bearing sufficiently.

When the propeller shaft, as is often the case, is in several lengths, they are usually coupled together by means of flanges and bolts. To find the proper diameter for the bolts, divide the cube of the diameter of the shaft by the radius of the circle in which the centres of the bolts lie, by the number of bolts, and by 3; the square root of the quotient will be the required diameter of each bolt spindle measured inside the thread. For example, let the diameter of the shaft be 16
inches, the number of bolts 8, and the radius of the circle in which their centres lie 12 inches;

then \( \frac{16}{12 \times 8 \times 3} = 14.22 \), and \( \sqrt{14.22} = 3.77 \) inches

are the required diameters of bolt spindles. The weight of such shafts is supported by additional bearings at the rate of one to each division of the shaft.

The shaft pipes for a pair of twin screws are sometimes made to pass through the ship's skin at suitable points in the after part of her bottom, and are supported at their after ends by iron brackets. Sometimes they are contained in a pair of twin stern posts and twin runs, the ship being so designed as to have a single forebody and a double afterbody.

When the after end of a shaft-pipe is supported by a bracket, the bracket may consist of an upright arm, hanging downwards from the ship's quarter, and a thwartship arm. The arms should be in the plane either of the stern post or of a thwartship bulkhead; they should be broad in a fore-and-aft direction and thin transversely. If sharp-edged fore and aft they will cause the least resistance. It is difficult to lay down any precise rule for their scantlings; but a sectional area equal to about half that of a screw blade at its root ought to be sufficient for each of them. (On the subject of twin screws, see a paper by Captain Symonds, R.N., in the Transactions of the Institution of Naval Architects for 1864.)

A paper on "Some Remarks on Apparent Negative Slip," by Professor Rankine, was read at the Institution of Naval Architects in April, 1867, and is here reproduced:

1. When the attempt has been made to account for the apparent negative slip of a screw propeller by the fact of its
laying hold of a current of water that is following the ship, the objection has been raised that the forward momentum impressed on the current in a second is equivalent to the resistance of the ship; that the backward momentum impressed by the screw on the propeller race in a second is equivalent to the thrust of the screw, which is equal and opposite to the resistance of the ship; and that, consequently, even if the screw were to take hold of every particle of the following current, that fact would account for a diminution of positive slip only, but not for negative slip.

2. If the velocity of the following current in which the screw worked were simply the mean forward velocity of the ship's wake, the objection in question would be unanswerable; for it is the momentum per second due to that mean velocity which is equivalent to the resistance of the ship, and to which the reasoning just mentioned applies.

3. The water affected by the passage of the ship through it has various reciprocating or wave-like motions combined with the mean velocity of the wake, and, in particular, there is forward motion under every crest and backward motion under every hollow, of the waves that accompany the ship. The velocity of those reciprocating motions is not connected directly with the resistance of the vessel—in fact, their resultant momentum is equal to nothing—and it is only the momentum of the uniform current, which remains after the wave motions have died out, that is equivalent to the ship's resistance.

4. Hence, if there happens to be, as there generally is, the crest of a wave following or filling under the ship's counter, the water which the screw lays hold of has a temporary forward velocity over and above the permanent velocity of the wake. That temporary velocity, indeed, may be many times greater
than the permanent velocity of the current whose momentum is equivalent to the resistance of the ship, and thus any extent of apparent negative slip may be accounted for.

5. The existence of the following wave explains also the fact that any considerable apparent negative slip is always accompanied by waste of motive power, the resistance to the motion of the engine increasing in a greater proportion than its speed is diminished. For among the laws of wave motion are the following: that all forward motion of the particles in a wave is accompanied by an elevation in the level, and the pressure against a body in front of the wave, due to that elevation of the level, is exactly equal to the pressure required to impress the forward motion upon the particles of water.

Such is the pressure exerted upon the stern of a ship by the wave which follows under her counter when that wave is undisturbed by the action of the screw. But the screw, by checking or reversing the motion of the particles of water, lowers the level of the crest of the following wave and diminishes the forward pressure which that wave exerts on the vessel. That diminution of pressure is virtually equivalent to an increase of the ship's resistance; so that the thrust of the screw must be equal not merely to the resistance properly due to the dimensions and figure of the ship, but to that resistance increased by a force equal to the diminution which the action of the screw produces in the pressure exerted on the ship by the following wave. Thus the total thrust of the screw is increased above its effective thrust—that is, above the proper resistance of the ship—in a proportion greater than the proportion in which the speed of the screw is diminished through apparent negative slip, so that the result is an increased ex-
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penditure of the motive power above what would be required if the screw acted in water not affected by wave motion.

6. The principles of the preceding paragraph do not apply to uniform forward motion of the particles of water produced by friction, because such motion is not accompanied by the production of a swell; and hence the permanent following current in the ship's wake due to frictional resistance does not give rise to a loss of thrust as the wave motion of the particles of water does.
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