An Appraisal of Hydrofoil Supported Craft*

T. M. BUERMANN, Lieutenant Commander P. LEEHEY, U.S.N., and Commander J. J. STILWELL, U.S.N.

The purpose of this paper is to arouse the interest of the naval architectural profession in the potentialities of hydrofoil supported craft and to enlist its aid in solving the problems which stand in the way of fully achieving these potentialities. The paper is essentially expository in nature. The only claims to originality lie in the manner of presentation of known fundamentals and in certain conclusions drawn from them regarding advantages gained and limitations imposed by the use of hydrofoils.

The hydrofoil is described as a hull supported clear of the water surface while underway by the dynamic lift of underwater wings, or hydrofoils. For certain speed-length ratios, it offers a substantial reduction in resistance and a marked improvement in seakeeping capability over a comparable displacement or planing craft. The efforts of various inventors and shipbuilders to produce hydrofoil supported surface craft and seaplanes during the past half century are reviewed. The early sporadic efforts have been followed by government supported programmes in Germany during World War II and subsequently in the United States. The feasibility of hydrofoil craft is considered as well demonstrated, at least for smaller sizes.

To indicate the present state of the art, the most important elements of hydrofoil design are considered in some detail. The methods for determining the principal components of hydrofoil resistance, or drag, are outlined. In a fashion analogous to aerodynamic practice, hydrofoil drag may be separated into profile, parasitic, induced, and wave-making components. A characteristic feature of hydrofoil craft is that the wave drag coefficient decreases rapidly with increasing speed. The "take-off" speed, where the hull first clears the water, is shown to play an important rôle in determining the overall relationships between speed, drag, and power required. Configurations embodying fully submerged foil systems are shown to have "hump" power requirements at take-off speed.

The question of stabilizing hydrofoil craft in a seaway is dealt with in a qualitative fashion. The forces acting on a foil in both ahead and following seas are described and certain tentative conclusions are drawn regarding the ability of various configurations to negotiate these seas.

While it is considered that a detailed consideration of the design and construction of hydrofoil craft lies beyond the scope of this paper, a brief discussion is given of certain points where hydrofoil craft depart from more cenventional ship design practice.

In evaluating the practicality of hydrofoil craft, comparisons are made of specific hydrofoil and conventional designs, both in ranges where the hydrofoil shows a clear advantage and in ranges where the application of foils is obviously absurd. From this, a general study is made to determine where the proper field for hydrofoil applications lies. It is concluded that upper limits on size, together with lower limits on speed, fix the maximum size of hydrofoil craft, consistent with available powering, in the 1,500 to 3,500 ton range, and set the lower limit of Froude number based on over-all length between 0·6 and 0·7. Within these bounds, the prospect is considered favourable for application of hydrofoils to high-speed passenger ferries, small premium cargo carriers, military patrol craft, and pleasure craft.

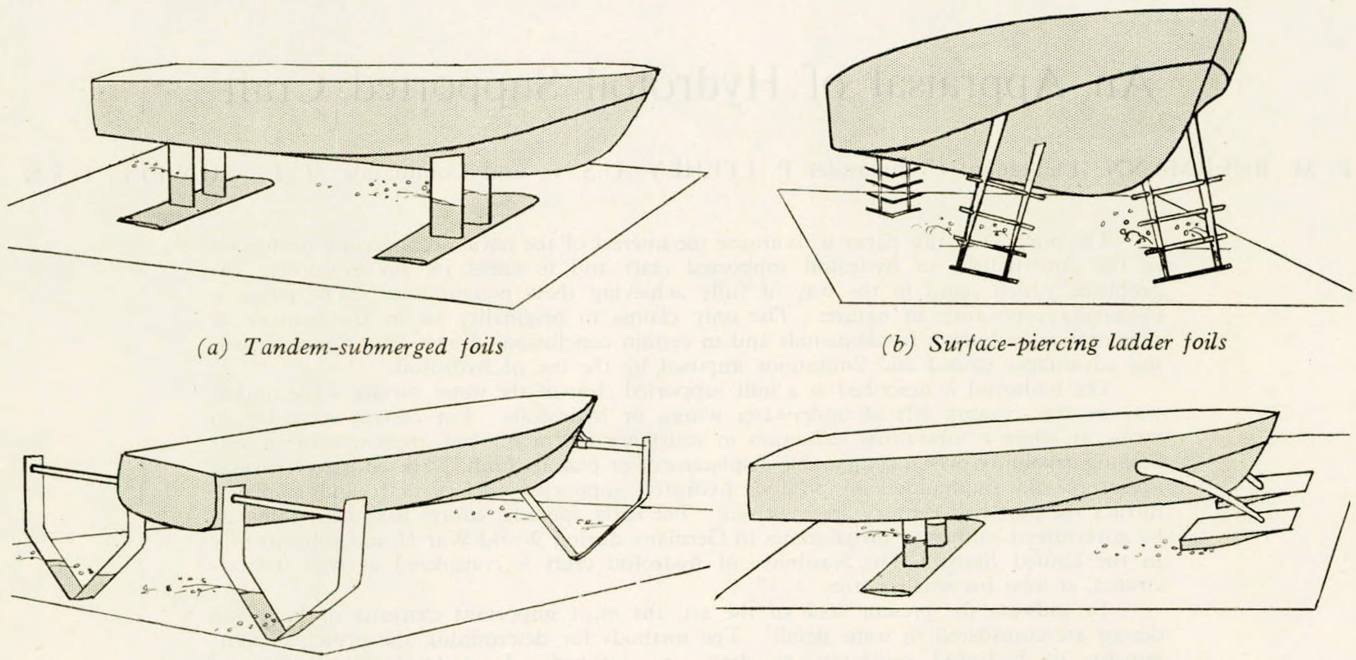
INTRODUCTION

There is a world-wide resurgence of interest in a novel means of reducing the resistance of high-speed boats called the hydrofoil which well merits the attention of naval architects and marine engineers. It is the purpose of the authors in this paper to trace the developments that have taken place in this field, to evaluate the promise of the device, and to outline the problems inherent in development of these vessels with the hope that interest created in this Society may lead to better solutions to some of the problems.

Anyone who has faced the problem of increasing speed of small to moderate sized ships or boats is well aware of the price that must be paid, particularly in craft which depend on the water surface for support. While in most vehicles such as aircraft, automobiles, or trains the power required varies roughly as the cube of the speed or less, in ships and boats at higher speeds the power is proportional to about the fifth power of the speed. This physical fact has brought some criticism to the naval architects from certain lay circles which

* Reprinted by permission of the Society of Naval Architects and Marine Engineers, New York, from their 1953 Transactions, Vol. 61, pp. 242-264.

judge progress in terms of speed. A brief reflection shows that wave making at the water surface is the principal contributor to this disparity between ships and other forms of transportation. When it is seen that in a destroyer type in the 30- to 40-knot range, more than half the required power goes into wave making, and when it is realized that there has been no means developed so far to stop a ship from making waves, the prospects for large increases in speed without large compensating increases in power and size look bleak indeed.



(a) Tandem-submerged foils *(b) Surface-piercing ladder foils*

(c) Surface-piercing V-foils *(d) Grunberg configuration submerged after foils plus surface skids*

FIG. 1—*Typical hydrofoil configurations*

Since about the turn of the century various inventors have attempted to overcome this barrier to higher speeds by lifting the hull of a boat out of the water and supporting its weight by the lift produced by hydrofoils operating in the water. In the course of these experiments, it was discovered that, in addition to substantial reductions of power required, the hydrofoil-equipped boat gave better riding qualities in rough water than a conventional boat of comparable size and speed.

Before proceeding to more detailed discussions of hydrofoils, perhaps some description of hydrofoil-equipped boats might afford better visualization of the problems involved. While many different configurations of hydrofoils have been tried, four general types will suffice to illustrate ways of doing the job. Referring to Fig. 1 we have examples of craft fitted with the following systems:—

(*a*) Tandem-submerged foils
(*b*) Surface-piercing ladder foils
(*c*) Surface-piercing V-foils
(*d*) Submerged after-foil plus surface skids (Grunberg configuration).

All of these configurations, among others, have been successfully used on small to moderate size boats, and developments are continuing. Regardless of configuration, the lifting force required is generated by the motion of an airfoil section through the water. This hydrofoil is smaller than its sister airplane wing because of the difference in density of the fluids. Basic lift and drag properties for hydrofoils are directly available from existing published airfoil data. However, these data must be corrected for certain effects peculiar to water and the presence of a free surface.

The inherent differences in the four configurations, illustrated in Fig. 1, lie in the methods of obtaining stability and controlling "altitude in flight". The first system requires electric, hydraulic, or mechanical controls which vary the angles of attack of the foils in response to an automatic signal which is a measure of the height of the hull from the water surface. The second scheme (surface-piercing ladder foils) achieves both stability and altitude control by maintaining equilibrium between the lift of the foils that are submerged and the weight of the boat. The third gets its stability and control from the equilibrium between weight and the lift of the portion of the foil remaining submerged. The fourth scheme (a totally submerged after-foil and forward skids) is somewhat more subtle. Here, after speed is reached, the skids plane on the water surface and in effect make the boat pivot about this point. Then, the large foil is designed to respond to the trim of the boat, seeking an equilibrium trim where the lift corresponding to the angle of attack on the foil exactly equals the weight not carried by the skids.

In all these schemes, the boat starts from rest in a displacement condition and is accelerated to a "take-off" speed at which foil lift causes the hull to rise clear of the water, leaving just the propulsion and foil systems in the water. Little imagination is required to recognize the kinship between this craft and an aircraft, but the presence of the free water surface introduces problems not met in conventional aircraft. Thus the foil boat may be said to fall between aircraft and ships.

HISTORICAL DEVELOPMENT

The history of hydrofoil development is a record of many failures italicized by a few notable successes. There is little evidence of a steady improvement in types and much evidence of haphazard approaches to the problems of flight in near proximity to the water surface. In fairness to the many inventors and scientists who turned their efforts to this intriguing problem (including the Wright brothers, Alexander Graham Bell, and Otto Tietjens) it should be stated that there were two fundamental reasons for their meagre success. First, the problems of hydrofoil flight are inherently more complicated than those of subsonic aerodynamics. Second, the aircraft faced no real competitor during its formative years while the hydrofoil faced the prospect of comparison with surface transport from its inception. As a consequence, serious consideration of hydrofoil vessels has had to await the development of materials, power plants, and fundamental understanding of principles, largely derived, fortunately, from the advances in aircraft and ship design over the past fifty years.

At the turn of the century, the men experimenting with the early aircraft and those developing the first successful planing hulls both considered the use of hydrofoils as an integral part of their experiments. As a result it is difficult to determine when the first true hydrofoil boat actually lifted its hull clear of the water by foil, and not planing lift. In France in 1897 the Comte de Lambert drove a catamaran fitted with four transverse "hydroplanes". The floats, or hulls, were raised clear of the water, but it is not clear whether this was accomplished by planing or hydrofoil lift. Perhaps the first true hydrofoil boat was developed in Italy by Forlanini between 1898 and 1905. This craft was supported by a complex system of flat ladder foils. There is little record of its performance other than evidence that it "flew". Also in Italy, and shortly thereafter, Crocco developed a craft supported by monoplane dihedral foils which attained a reported speed of 50 m.p.h.[1]

In 1907 Wilbur and Orville Wright experimented with a foil-supported catamaran on the Miami River at Dayton, Ohio. Testing was abandoned following a river dam failure which resulted in insufficient water depth to permit operating the craft. In 1909 Captain H. C. Richardson, U.S.N.(ret.), fitted tandem biplane foils to a canoe. When towed at 6 knots this craft flew on the lower set of foils. Later, in 1911, Richardson and White outfitted a dinghy with submerged foils employing manual angle of attack control for stabilization and manœuvring (Fig. 2).

About 1911, Richardson and Curtiss in the U.S.A. and Guidoni in Italy began using hydrofoils on seaplane floats to assist in take-off. Guidoni's work, in particular, was quite extensive. Over a period of fifteen years he designed and flew various hydrofoil-fitted seaplanes ranging in weight from 1,400 to 55,000lb. Float sizes were reduced through use of hydrofoils, resulting in a substantial reduction in take-off resistance[1]. However, as seaplane take-off speeds increased, the problems of foil cavitation and stability multiplied. Italian efforts in this direction apparently were at an end by 1925.

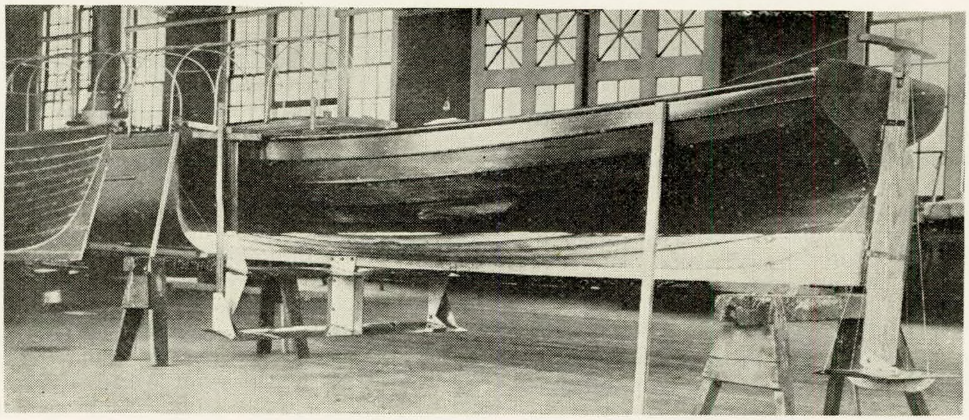
Dr. Alexander Graham Bell's spectacular boat, the HD-4, entered the picture in 1918 (Fig. 3 (Plate 1)). Together with Mr. Casey Baldwin, Dr. Bell produced a craft of outlandish design coupled with a performance which must be considered remarkable even in the light of present knowledge. With a gross weight of 11,000lb., the craft reportedly attained a top speed of 60 knots powered by two Liberty aircraft engines of 350 h.p. each[2]. The foils were of ladder type with dihedral and, despite the complexity of foil and strut intersections, they attained a maximum lift-drag ratio of 8·5 at 30 knots. During the years between the two World Wars, Mr. Baldwin made repeated unsuccessful efforts to interest the Navy Department in the military potentialities of the Bell-Baldwin design. Whether it was the tendency of the HD-4 to porpoise in a seaway or whether it was simply the sheer cumbersomeness of the design which caused these efforts to fail, one may only conjecture. Suffice it to say that there is no evidence of government support of hydrofoil development in any country for some fifteen years after World War I.

During the 1930's there was a renewal of interest in the application of hydrofoils to both seaplanes and surface craft. Dr. Otto Tietjens tested his first hydrofoil speed boat at Philadelphia in 1932, and built and tested a second larger boat near Berlin in 1936. Both craft were supported by a configuration embodying a single large dihedral main foil located somewhat forward of the centre of gravity and stabilized by a smaller elevator foil at the stern. H. F. von Schertel tested his first successful craft in Germany in 1936. The Schertel design different from that of Tietjens in that two V-foils in tandem were used, each carrying approximately one-half the weight of the craft. During this period Guidoni's earlier work on seaplane applications was re-evaluated by the British[3]; and, at the request of the United States Navy Bureau of Aeronautics, the National Advisory Committee for Aeronautics undertook a model test programme in 1936 initially aimed at testing various configurations originally proposed by Guidoni. In Germany, W. Sottorf reported on extensive experiments with various foil sections for high-speed use aimed at possible seaplane applications[4].

The first really practical configuration employing angle of attack stabilization was conceived in France by V. Grunberg in 1935. This system is shown schematically in Fig. 1(d), and its action in a seaway will be described later. The feasibility of this system was demonstrated at that time by model tests in the towing tank of the Institute Aerotechnique de Saint-Cyr[5]. In subsequent years, his idea has been studied by various investigators—attracted by its simplicity and reasonable margin of stabilization.

The first contributions to the understanding of the wave drag of hydrofoils were made by Russian theoreticians Keldysch, Lavrentiev, and Kotchin beginning in 1934[6, 7]. While parallel experimental work in Russia was reported by Vladimirov[8], one finds no serious effort in that country to develop either hydrofoil vessels or hydrofoil-supported seaplanes prior to World War II.

With the advent of World War II, German hydrofoil development, already very active, received substantial support from both the Navy and the Army. At the Sachsenberg Shipyard, Rosslau, a number of craft were designed and constructed along the lines of the basic Schertel concept. The Schertel-Sachsenberg affiliation produced craft up to 80 tons displacement with speeds up to 60 knots. The 17-ton patrol boat VS-6 (Fig. 4 (Plate 1)) and the 80-ton tank transport VS-8 (Fig. 6 (Plate 1)) typify this work. At the same time, a 17-ton craft after the Tietjens design was built at the Vertens Yacht Yard in Schleswig and designated VS-7 (Fig. 5 (Plate 1)).



Official photograph U.S. Navy

FIG. 2—*Dinghy fitted with controllable submerged foils*

None of the German craft was placed in operational use, despite the concentrated development effort. Various reasons have been given for the failure of the programme, chief among which were the insistence of the German high command for quick results and the lack of suitable materials and trained engineers. Allied bombing of the building yards destroyed most of the craft under construction and the deteriorating condition of the German war effort brought an end to the programme in 1945. Tests of VS-6 and VS-7 furnished inconclusive results. VS-7 was the faster boat, but was much poorer than the VS-6 from the point of view of stability and manœuvrability. The ambitious VS-8 was apparently underpowered. During one series of tests she failed to remain foilborne in a following sea and was subsequently beached and abandoned. Some of this work is reported in references 9 and 10.

In the United States, the National Advisory Committee for Aeronautics has continued a modest programme of hydrofoil investigations primarily aimed at seaplane applications[11,12]. Various individuals have developed hydrofoil craft since World War II, both in the U.S.A. and in Europe. A novel extension of the Grunberg concept was devised by Christopher Hook of Cowes, Isle of Wight. The Hook boat (Fig. 7 (Plate 2)) uses two surface skids on long jockey arms ahead of the craft to stabilize independently two forward submerged foils, each bearing one-third the load of the craft. An after foil, integral with the propulsion system at the stern, carries the remaining load and requires only craft trim to achieve adequate stabilization. Two Swedish engineers, Almquist and Elgstrom, built several craft basically of the Grunberg configuration, but with curved main foils to obtain area stabilization as well.

A number of small speed boats employing three retractable V-foils have been developed and constructed by J. G. Baker, of Evansville, Wis. An important feature of his design is the absence of immersed supporting struts. All underwater surfaces, except for the outboard propulsion unit, are lifting surfaces with consequent minimization of parasitic drag. Schertel has resumed activities in the field. His new passenger ferry, PT-30, is similar to VS-6 in appearance. It has been operating recently on Lake Lucerne, Switzerland. Reference 9 indicates that this craft displaces 9·5 tons and achieves 40 knots with 450 h.p.

Since 1947, the Navy Department has supported a programme for hydrofoil research and development involving the co-operative efforts of ship designers and government and university laboratories. An intensive effort has been made to overcome problems of foil design and stabilization which thwarted many early investigators. Studies have been made to determine the limits of practical size and speed, consistent with feasible powering, within which the hydrofoil possesses inherent advantages over the other surface craft. In the course of this work, certain small test craft were developed to permit open water evaluation of performance in a seaway.

Fig. 8 (Plate 2) shows a 12-ft. open-water model employing a pair of biplane V-foils forward with a small V-foil elevator aft. Two basically different test craft of about 20 feet in length are shown in Figs. 9 and 10 (Plate 2). The former has four V-foils arranged in tandem providing inherent stabilization. The latter employs an automatic surface-sensitive incidence control system acting on a tandem submerged foil configuration. Other systems of hydrofoil support are also under active consideration.

In retrospect, it appears that the hardy persistence of the hydrofoil concept for over a half-century testifies both to its basic soundness and to its difficulty in execution.

STATE OF THE ART

The present state of knowledge of hydrofoils may be said to lie somewhere between research and development and practical design, depending on the size of the craft. To best illustrate this status, there follows a discussion of the design problems encountered in these boats and their methods of solution.

The same problem exists in design hydrofoil craft that is involved in conventional ship design, namely the proper balance of the major variables so as to best achieve the required function. For proper consideration of the important factors, a hydrofoil craft should be considered as a carefully designed foil system with an integrated propulsion unit supporting and propelling the necessary mass at a given speed, subject to external conditions.

The estimation of power required depends on reasonably accurate determination of resistance under flying and take-off conditions. Since much of the data required in this estimate is derived from tests of airfoils and aircraft components, aeronautical terminology is used for convenience. Thus resistance is described as drag.

HYDROFOIL DRAG

Since the weight of the craft is supported by dynamic lift of the foils, we equate displacement to lift and write

$$\Delta = L = C_L \cdot \rho/2 \cdot S \cdot V^2$$

where S is the projected foil area*. The selection of the foil section and operating lift coefficient C_L is a careful compromise between conflicting requirements for high foil strength, minimum total drag, and avoidance of foil cavitation at maximum craft speed and flow separation at take-off speed. The method of stabilization and the sea state which the craft must be capable of negotiating on foils play important rôles in this determination. Generally speaking, the selection of a foil section with a flat suction-side pressure distribution, a thickness ratio in the order of 10 per cent, and a design lift coefficient of about 0·25 affords a reasonable starting point for a preliminary drag estimate.

Lift coefficients obtained from aerodynamic data should properly be modified for the effect of foil submergence. However, while a hydrofoil tends to lose lift as it approaches the water surface, the effect on lift coefficient is not particularly significant for submergences of one chord or greater and may reasonably be ignored in preliminary calculations.

The total drag of a hydrofoil craft may be expressed as

$$D = C_D \cdot \rho/2 \cdot S \cdot V^2$$

where the dimensionless total drag coefficient C_D is the sum of profile, parasitic, induced, and wave drag coefficients: —

$$C_D = C_{Do} + C_{Dp} + C_{Di} + C_{Dw}$$

each expressed in terms of the projected foil area. The induced and wave drags are entirely residual in nature, while the profile and parasitic drags have both frictional and form components.

The free surface does not appear to have an important effect upon the profile drags of struts and foils, hence these coefficients may be obtained from known aerodynamic data such as in reference 13. For a lifting surface, the profile drag coefficient is

$$C_{Do} = (C_{Do})_{\min} + KC_L^2$$

expressed as where $(C_{Do})_{\min}$ and K are dependent upon the section used and are functions of Reynolds' number.

The principal sources of parasitic drag are: —

(*a*) Hull windage.
(*b*) Surface interference drag of struts (spray drag).
(*c*) Interference drags of foil-strut intersections.
(*d*) Drag of underwater appendages such as propulsion nacelles, control rods, and hinge joints.

Where intersections of one or more lifting surfaces occur, the corresponding interference drag coefficients are functions of the C_L^2 for the lifting surfaces. No detailed treatment of the various parasitic drags will be attempted here as their determination for hydrofoil configurations follows from the many experimental and theoretical results obtained in aerodynamic and naval architecture work[14].

One parasitic drag, however, is peculiarly important to hydrofoil craft. This is the surface interference drag, or spray drag, occurring at the point where struts or foils pierce the water surface. It results from a complex combination of effects

* All coefficients, such as C_L, appearing in this paper are dimensionless. Where units for physical quantities are not explicitly stated, they may be inferred from context.

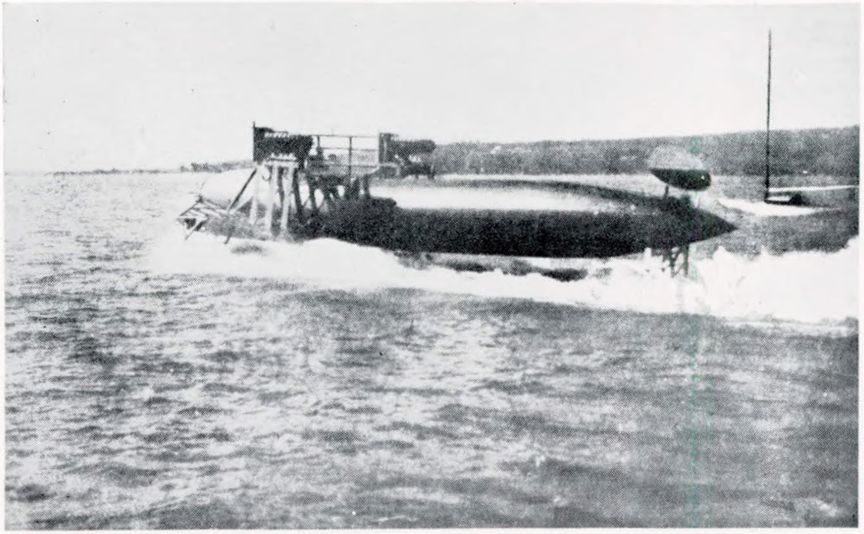


FIG. 3—*Alexander Graham Bell's HD-4*

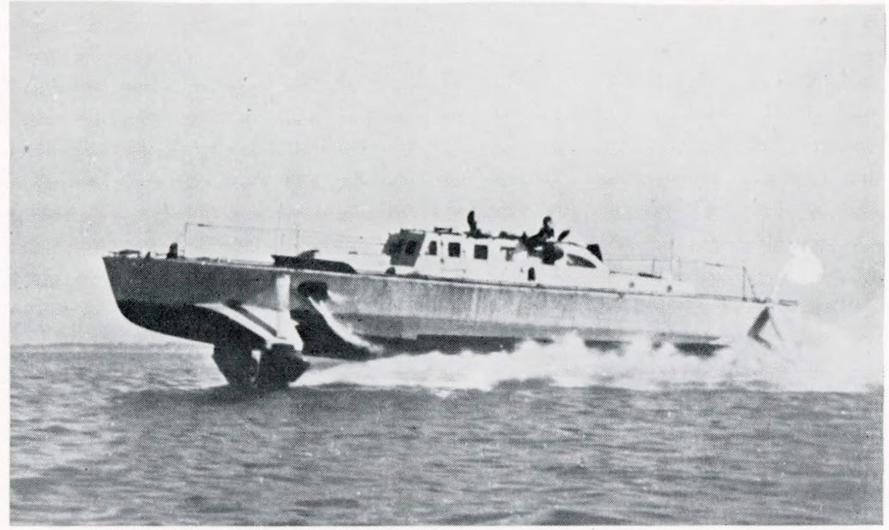


FIG. 4—*Schertel's VS-6 on trials*

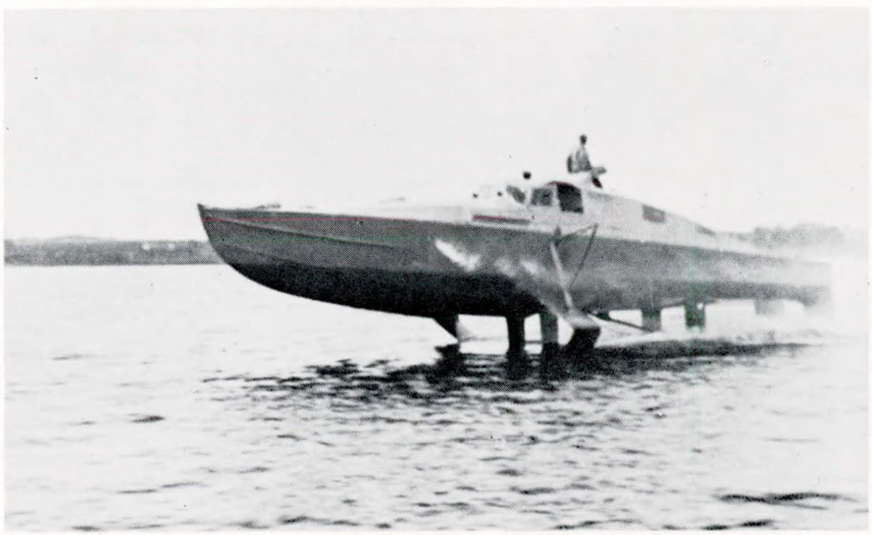


FIG. 5—*The VS-7, Tietjen's counterpart to the VS-6*

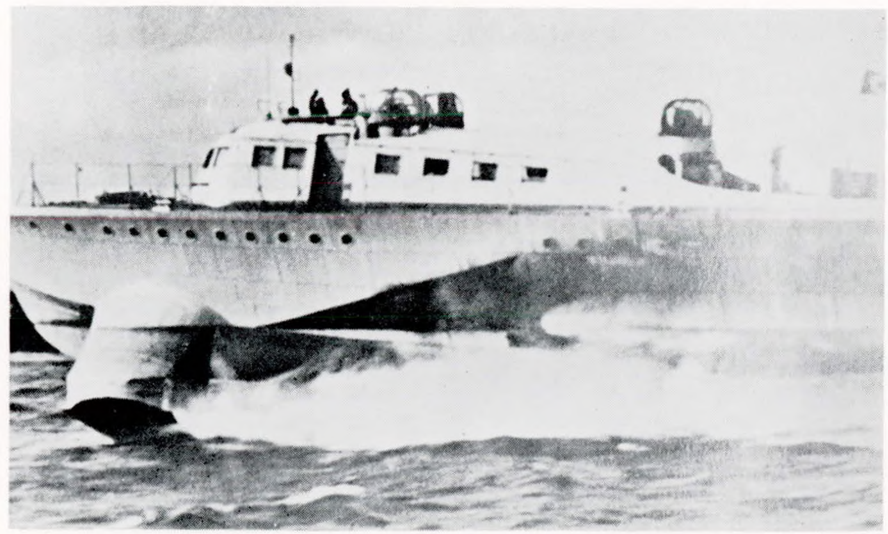


FIG. 6—*The Tank transport VS-8, by Schertel-Sachsenberg*

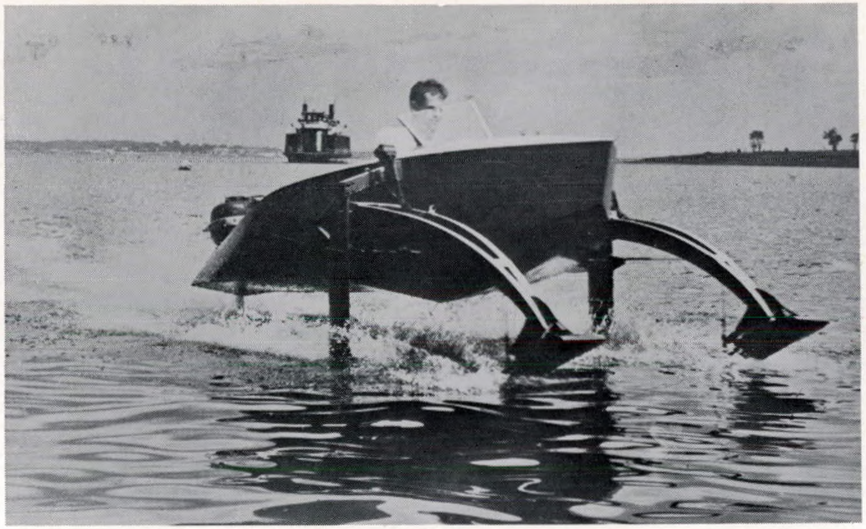
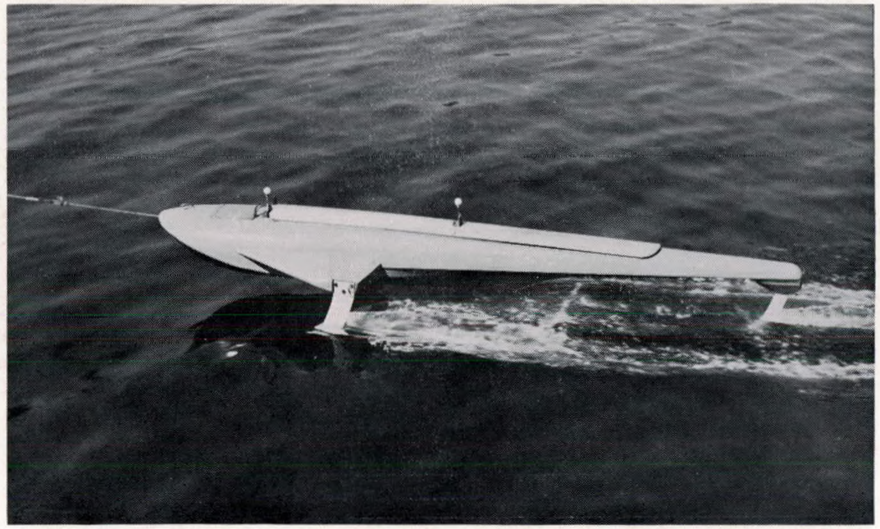
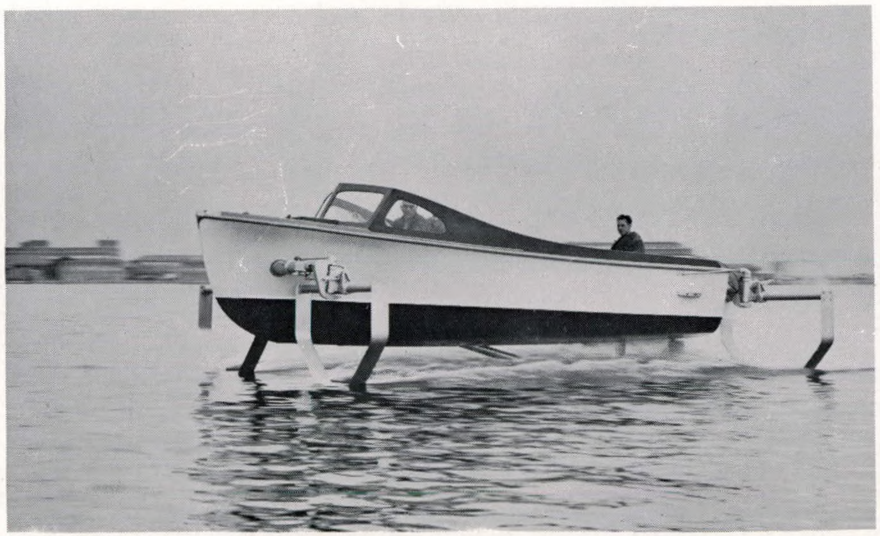


FIG. 7—*Christopher Hook's hydrofin*



Official photograph U.S. Navy

FIG. 8—*12-ft. open water model fitted with biplane V-foils*



Official photograph U.S. Navy

FIG. 9—*United States Navy test craft employing tandem V-foils*

Official photograph U.S. Navy

FIG. 10—*Automatic incidence control stabilization on a 20-foot United States Navy Test Craft*

involving air entrainment and spray formation. The wave-making component of this drag is of little consequence because of the very high Froude numbers, based upon strut chord, c, at which most hydrofoil craft operate. A reasonable measure of this drag has been derived from the tank tests as

$$R = C_{Dt} \cdot \rho/2 \cdot V^2 t^2$$

where t is the strut thickness and where an approximate value of the coefficient $C_{Dt} = 0{\cdot}2$ may be assumed for $F_c = V/\sqrt{gc}$ greater than 10. For lower Froude numbers, C_{Dt} assumes much higher values which are roughly proportional to the thickness ratio t/c. The dependence of spray drag upon strut profile and rake angle make further generalization on its magnitude difficult.

An important assumption is made in the calculation of the remaining portions of the drag of hydrofoil craft; namely, that the induced and wave drag components may be treated separately. Considering a uniformly loaded submerged hydrofoil represented by a horseshoe vortex, this assumption amounts to an arbitrary separation of the bound and trailing vortex drags. The induced, or trailing vortex, drag may be calculated by use of classical biplane theory considering the image vortex system to have circulations of the same direction and magnitude as those representing the submerged hydrofoil. The separation between the hydrofoil and its image is, of course, twice the foil submergence. The induced drag of the hydrofoil is then the sum of the drag derived from the effect of trailing vortices of the foil itself plus that derived from the effect of the image trailing vortices on the foil. This may be expressed in coefficient form as

$$C_{Di} = \frac{C_L^2}{\pi \text{A.R.}} (1 + \sigma)(1 + \delta)$$

where σ is the Munk interference factor and δ is the planform correction[15]. This representation is consistent with the free surface conditions only for infinite craft velocity (or infinite Froude number $F_h = (V/\sqrt{gh})$). However, since the Froude numbers, F_h, of most hydrofoil craft at their designed speeds and foil submergences are usually quite large, and the corresponding wave drags are usually small, this formulation agrees reasonably well with experimental results.

A two-dimensional treatment of wave drag effect based upon bound vortex circulation appears adequate for estimating the wave drag coefficient. This result, obtained in 1934 by Keldysch and Lavrentiev[6], is here modified by the assumption that the effect of submergence upon lift may be neglected. The resulting expression in coefficient form is

$$C_{Dw} = 0{\cdot}5 \frac{C_L^2}{F_c^2} \cdot e^{-2/F h^2}$$

where $F_c = V\sqrt{gc}$ is the Froude number with respect to foil chord c and $F_h = V/\sqrt{gh}$ is the Froude number with respect to foil submergence h.

It is thus seen that the wave drag coefficient is a function, not only of C_L^2, but also of two distinct Froude numbers F_h and F_c. However, since practical considerations of strut drag and strength limit foil submergences to the order of one to three chords, a single Froude number F_c may be considered to govern the magnitude of the wave drag component for feasible hydrofoil designs. Here the "effective craft length", or characteristic dimension, is simply the chord of the lifting hydrofoil, consequently all hydrofoil craft operate in a *régime* where wave drag decreases with increasing speed. As may be seen in Fig. 11, very small, high-speed craft have almost negligible wave drag.

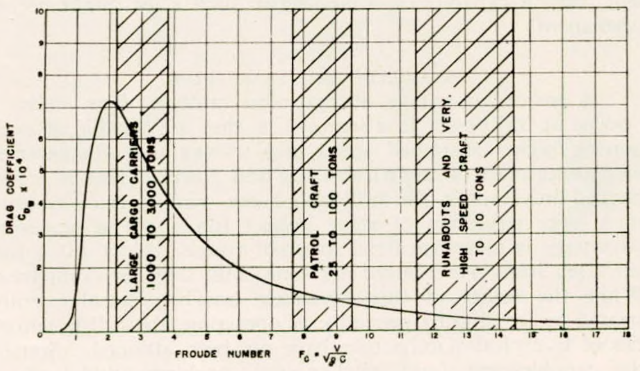


FIG. 11—*Relative importance of wave drag for various hydrofoil types*

With the increase in size of hydrofoil craft, and corresponding decrease in their designed speeds to limits imposed by available powering, the wave drag becomes a very appreciable component of total drag. A possibility then arises for the elimination of a substantial portion of the wave effect by use of a tandem foil configuration with foil spacing so adjusted to the design speed that the wave created by the forward foil is partially annulled by the wave from the after foil. The two-dimensional treatment of reference 6 gives the disturbance of the free surface at a distance x in feet behind a bound vortex of strength Γ at a submergence h as

$$y = \frac{-2\Gamma}{V} e^{-1/Fh^2} \sin \frac{gx}{V^2}$$

where V is the free stream velocity in f.p.s. Strictly speaking, this formula applies only for very large x. Practically, it is a good approximation to the surface for distances of one-quarter wave length or more behind the foil. The wave length of this disturbance is $2\pi V^2/g$, hence locating a second bound vortex of the same strength a distance $\pi V^2/g$ feet behind the vortex would result in a complete cancellation of the disturbance. For a hydrofoil ship with a designed speed of 30 knots, the required foil spacing would be about 250 feet. This must be regarded as approximate, for the formula is only asymptotically correct and does not consider the bound vortex and source distribution necessary to form a mathematical model of a foil whose chord/submergence ratio is of the order of unity. More importantly, the three-dimensional effect of the trailing vortices results in transverse waves characterized by the appearance of a roach or "rooster-tail" in the wake of a single submerged foil. This non-uniformity in flow may well affect the foil spacing required for optimum recovery of wave drag and almost certainly precludes the possibility of complete wave drag recovery as predicted by the elementary theory.

Even a brief discussion of hydrofoil drag such as this one should not omit some consideration of the troublesome question of the effect of surface roughness and fouling upon the profile drags of foils and struts. It seems that the acceptance of drag coefficients, based upon standard roughness as defined by the N.A.C.A.[13], leads to pessimistic conclusions regarding the performance attainable by hydrofoil craft. Such conclusions are probably unwarranted, especially in the case of smaller craft employing retractable foils. Here it appears quite feasible to construct and maintain foil surfaces with a degree of smoothness practically unattainable in aircraft wings (with due regard to proper scaling in this comparison). On the other hand, acceptance of very low profile drag coefficients obtained for "laminar flow" sections is equally unjustified in view of the turbulent nature of the seaway and the variable operating conditions required of the foil.

For large hydrofoil craft—where foil retraction becomes clearly impractical—fouling and corrosion will certainly take a heavy toll in increased drag. This situation will probably be only partially mitigated by the possibility of improving the relative smoothness of foils during construction because of the increased craft sizes. Little or no data exists which would permit a quantitative evaluation of these effects. There is, perhaps, some small comfort in the realization that the knowledge of the effects of surface roughness on the full scale resistance of conventional ships and aircraft is far from adequate, despite years of investigation.

TAKE-OFF

In the design, and particularly in the powering of hydrofoil craft, careful attention must be paid to the conditions of take-off. Take-off may be defined as the instant that the hull leaves the water, or the instant at which the hull ceases to contribute to either the lift or the water drag of the boat.

This discussion is limited to the "aircraft type" take-off which may be described as follows: At rest, the entire weight of the boat is supported by the hull buoyancy; as the craft accelerates, the foils increase their lift, unloading the hull, until at take-off speed the entire weight is on the foils and the buoyancy (or lift) of the hull is reduced to zero*.

The phenomenon that causes the most trouble in foil craft take-off is the drag hump. Characteristically, the thrust required to propel the hull and foils increases to a maximum as take-off is approached. Then, as the hull clears the water, thrust requirements drop to a minimum value at a speed slightly over take-off and then climb again, until with proper design, the full power is reached at maximum speed. Although the power available for full speed is usually considerably more than the maximum hump requirement at take-off, the normal characteristics of propellers, attempting to deliver high power at relatively low speed of advance, limit thrust available. Therefore, the maximum resistance at take-off rather than the power required is the controlling factor (Fig. 12). In addition, the

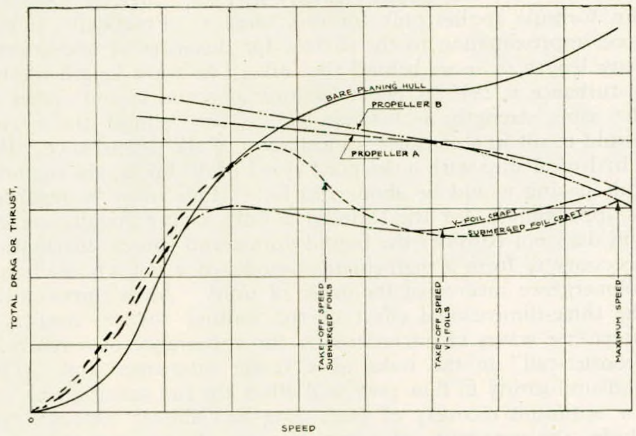


FIG. 12—*Typical thrust-drag curves for two hydrofoil craft*

margin of thrust available over resistance is the accelerating force, the amount of which determines the time and distance required for take-off. A good thrust margin is required because take-off may be necessary in rough seas and under other adverse conditions that increase resistance, particularly of the hull. Even in ideal conditions, a craft that can just get to a condition of equilibrium between thrust and resistance at take-off speed will not take off, as there must be a slight excess of lift over weight to provide for the vertical acceleration needed to lift the boat to its flying attitude.

The speed selected for take-off coupled with the foil lift coefficient that can be tolerated in this condition will determine the minimum foil area required for the craft and thus will affect the top speed, unless a surface piercing or ladder-type foil is used where some of the area required for take-off comes out of the water at top speed. In choosing take-off speeds and lift coefficients, it is convenient to use a relatively high value of lift coefficient and a relatively low take-off speed to avoid prolonged runs at full power and with objectionable wave encounter. It can be shown that take-off distance is a function of speed squared so that an increase of take-off speed causes a large increase in distance. However, since the induced drag of the foils increases as the square of the lift coefficient, this practice may induce unacceptable power humps requiring a compromise in top speed or the inclusion of variable pitch propellers in an already complicated propulsion problem.

* An alternative "elevator type" take-off has, on occasion, been used with submerged foil craft. Here the craft accelerates to flying speed without lift on the foils, then, suddenly, lift is applied and the craft rapidly ascends to its flying attitude. The authors find no advantage in the elevator type take-off.

The problem of designing a hull and estimating its performance under these conditions of unloading with increased speed is peculiar to the hydrofoil and seaplane fields. Since the amount of foil lift and drag imposed on the hull depends on the speed and the trim, and since the trim during take-off is a resultant between the natural trim of the hull at the given speed and the impressed trim from the action of the forward and after foils, the estimation of the total resistance at any pre-take-off speed is a complex job that is best handled by model experiments.

For the initial selection of the hull form for this service, certain guides can be set down. To reduce impact loading during inadvertent high-speed touch down or during momentary wave slap when flying, V- or U-shaped sections appear to be preferable to flat sections. As for the best form for take-off, the speed at take-off and the size of the craft must be considered. If the take-off speed is in the range of speed-length ratios for the craft that are favourable for displacement operation, then a displacement type hull should be used. If, however, the speed is such that in a normal craft of the size planing would be an advantage, then a planing type is indicated. In addition to these considerations of impact and minimum hull resistance, if the foil configuration is a surface-piercing, Grunberg, or ladder type, the trim of the hull under the speed and load conditions to be met must be investigated, since, in these cases, the ability of the foils to achieve a proper lift coefficient to lift the craft depends to a certain extent on the hull trim.

The surface-piercing and ladder type configurations generally have a smooth take-off with little drag hump as compared to the fully submerged foil types. On the other hand, the submerged types usually can take off at lower speed but with a sharper transition evidenced by a considerable drag hump (Fig. 12). This condition is brought on by the fact that the submerged foil uses the same amount of area at take-off as it does at top speed. Thus there must be a large lift coefficient at the low take-off speed to support the weight. This can only be achieved by increasing the angle of attack. Assuming a take-off at one-half of top speed, the submerged foil will require a lift coefficient equal to four times that at top speed, and since the induced drag varies as C_L^2, the induced drag at take-off will be about four times its value at top speed. With the surface-piercing type foil, on the other hand, it is common practice to use about one-half of the total area available for top speed operation, and almost all the available area at take-off. Therefore, the take-off lift coefficient can be about twice that for top speed and the induced drag, taking into account the effects of aspect ratio change, will be about the same for both conditions. The above effect is so marked that in actual operation it is often difficult to ascertain the exact take-off point with surface-piercing foils.

All of these factors affecting take-off assume added importance when it is realized that a hydrofoil is usually capable of flying through sea conditions which it would have difficulty negotiating as a displacement craft, and that unless it has excellent take-off capabilities, a landing in such a sea might become permanent.

STABILITY AND SEAKEEPING

A major advantage of hydrofoil craft is their ability to operate at full power in a seaway, i.e. they are capable of maintaining higher sustained speed in a seaway than displacement or planing craft of comparable size and power. This is accomplished by keeping the hull above the wave surface resulting in a large reduction of wave impact forces. The seakeeping advantage is most beneficial to small vessels (below 1,000 tons) since sea states experienced are frequently large by comparison. While the reality of this advantage has been amply demonstrated by small craft presently in operation, the full potentialities of hydrofoil stabilization have not been attained. Some of the troublesome (and challenging) problems involved are indicated.

Most hydrofoil craft can be stabilized in calm water and the requirements are clear. However, with few exceptions hydrofoil craft are required to operate in waves, and the combined requirements of stability and wave response are not fully understood. Speaking in general terms the flying characteristics of a hydrofoil should be a compromise between two requirements. From the point of view of comfort and minimum acceleration, the craft should fly in a straight path with the irregular water surface positioned within the gap between the hull and foils. This implies no wave response and is indeed the most satisfactory means of flying in small waves. However, from the point of view of maintaining the hull above the water surface the craft should closely follow the wave pattern as the waves get larger. Since these are limiting conditions, a compromise between the two is strived for in most hydrofoil designs.

Surface-piercing V-foils have been used extensively since they serve as stabilizing members as well as lifting surfaces. Lateral stability is inherent if the centre of gravity is not too high, since heel provides increased foil area on the low side, and the increased lift produces a righting moment. Longitudinal trim is maintained in a similar manner since when a foil sinks below the equilibrium position it adds more foil area producing a restoring force. It should be noted that as speed increases a craft with a constant chord V-foil configuration tends to rise in the water. If this proves undesirable, the rates of change of lift with submergence for the forward and after foils can be physically altered so that the trim of the craft and the resultant angle of attack of the foils is reduced as the speed is increased. This prevents the craft from rising too high.

Where permissible foil dimensions unduly limit the rate of change of lift with submergence of V-foils, or where an increased range of flying speeds is desired, ladder foils can be used. Dihedral is usually incorporated in the ladders making the stabilizing action much the same as that of V-foils. However, the operation of a ladder configuration in a seaway produces complex spray and interaction effects characterized by erratic variations in hydrodynamics forces on the system.

The Grunberg configuration has one main submerged foil aft with surface stabilizers forward and lateral and longitudinal stability are achieved by the forward planing surfaces. These surfaces, or "skids", have strong depth stability and poor lift/angle of attack curve slopes. Since these are no better than conventional planing hulls, they are not used for main lifting surfaces, and normally carry only 10 to 20 per cent of the weight of the craft. The remaining weight is carried on the main foil placed aft of the centre of gravity. However, the "skids" are quite satisfactory as stabilizing surfaces, providing a positive force to locate the craft in the proper relation to the water surface. The craft trims about the planing surfaces. Consequently, if the main foil tends to rise, it automatically decreases its angle of attack and lift as the hull trims, thus the craft returns to the original position. The dynamics of such a craft can be investigated by making the assumption that the skids follow the wave contour. In practice it has been shown that is true for low frequencies of wave encounter but is not realistic for high frequencies. Therefore, such calculations must be properly weighed, and since they are tedious, model tests are probably more desirable. The tendency of the "skids" to skip and the light damping of disturbance motions are disadvantages.

With fully submerged foils the situation is somewhat different. Since the foils must remain submerged at all times, they are not affected by their depth of submersion. Essentially, no inherent stability or control is present and some type of stabilization must be provided*. The possible configurations are a large foil under the centre of gravity supporting most of the weight of the craft with a small tail foil aft for balance purposes and the tandem foil arrangement where the two foil areas are approximately equal. There has also been some use of the "Canard" configuration with the small foil forward. All of these must be artificially stabilized.

Submerged foil configurations can be provided with incidence or flap control for the foil sections. The angle on the foils or flaps is governed either by a very lightly loaded planing surface and mechanical linkage or by a water surface detector providing a signal to an autopilot which in turn motivates an actuator to position the angle of attack of the foil. With a configuration of this type it can be assumed that the foils are completely submerged at all times and the stability may be investigated satisfactorily by theoretical means, namely the traditional aircraft approach to dynamic stability and servo mechanisms, modified by the hydrodynamic surface effects.

The ability of the hydrofoil craft to fly in waves of various heights depends upon the relation of strut to craft length, the frequency of encounter with succeeding wave crests, and the effects of orbital velocities. The relation of strut to craft length determines what vertical displacement the craft must make to keep the hull clear of the wave profile. The choice of strut length governs the maximum craft motion normal to the wave profile as well as the admissible normal accelerations equivalent to limiting column loading of the struts.

The frequency of encounter with wave crests is a function of craft velocity relative to the sea and of the wave length. It governs the time in which corrective actions at the foils must act, and conversely, the time in which orbital velocities may affect the foil lift. The higher the frequency of encounter with waves, the less time there is for variations in lift to act, and the smaller the vertical displacement. For the same boat speed the frequency of encounter is lower while running with the waves than when going into them. Hence the need for changing the attitude of the foil or foil area is not as great in the case of head seas as in following seas.

The orbital velocity of the water, added vectorially to the craft velocity and to the instantaneous velocities due to pitch and heave of the craft, determines effective change in the angles of attack of the foils from their steady state values. These effects on foil lift must be coupled with the stabilizing effects of the configuration (area change for V-foils or angle change for controlled submerged foils) in relation to the instantaneous position of the craft in a wave train.

In the head sea condition as a foil system approaches the face of a wave, either more foil area is added or in the case of a controlled submerged foil a signal developed indicating a depth error in a direction that causes the foil to increase its angle of attack, thus causing the foil to rise. Also as the foil enters this portion of the wave it enters an area of upward water particle motion which further increases the effective angle of attack of the foil relative to the water flow, hence causing an additional upward force on the foil.

There are two basic differences between head and following seas which affect the behaviour of hydrofoil craft. The frequency of encounter in following seas is greatly reduced, giving the craft more time to respond, and the orbital motions are in a detrimental rather than favourable direction. The orbital motion of the water particles in the back slope of a wave is downward which, when vectorially added to the horizontal velocity of the water past the foil due to the forward velocity of the craft, has the effect of decreasing the angle of incidence of the foil relative to the water flow. If the geometric foil angle is not changed or if the slope of the lift/angle of attack curve is not altered by other means, the foil experiences a reduced lift force at the very time that lift should be increased in order for the foil to rise and track the wave it is approaching and overtaking. The reverse and equally detrimental condition occurs under the forward face of the wave. There, the orbital velocities tend to raise the foil as it approaches a trough. The above is premised on the practical assumption that the craft velocity is higher than that of the waves. In the unusual case, where the following sea overtakes the craft from astern (very large fast waves), the orbital velocities aid the craft response as in a head sea.

* A small stabilizing force exists due to a decrease in lift as a foil approaches the surface. It is possible in model scale to produce a configuration with fixed submerged foils which is stable for small disturbances. This is of little practical importance in full scale operation in a seaway.

The actions of submerged and surface-piercing foils in response to these dynamic conditions are quite different. For V-foils the principal restoring forces are provided by changes in foil area. At design speed these foils operate about half submerged, hence the maximum restoring force is approximately equal to the steady state lift. Submerged foils, on the other hand, operate with a constant lifting area and meet changing flow conditions by changes in angle of attack as signalled by some surface sensing device. At low flying speeds, restoring forces in the order of three to four times steady state lift are obtainable without stalling. However, at high speeds in a seaway, the permissible variations in lift coefficient, and the restoring forces available, must be severely limited if cavitation is to be avoided*.

One cannot make generalizations about the relative seakeeping capabilities of various configurations without considering the foils in their relation to the motion of the entire craft. Certainly trim, for example, plays an important rôle.

Despite the many variables involved, a few qualities appear inherent in each system operating in specific sea states. From the foregoing discussion one may conclude that an automatically controlled submerged foil craft will be superior to a V-foil craft in a following sea where relatively large, slow changes of flow occur. Conversely, a V-foil craft should perform better in head seas where high frequency of wave encounter is present. Ladders offer one possibility of improving the following sea operation of area stabilized craft, if the added drag penalties can be accepted, since the higher foils of the ladder system can be placed at greater angles of attack. The relative insensitivity of planing skids to orbital velocities indicates why a Grunberg configuration shows better ability to fly at low frequencies of encounter than a V-foil craft.

It is perhaps too much to expect that the future will see the development of a single method of stabilization which is optimum for all operating conditions. Yet the state of hydrofoil art, as demonstrated by the various radically different configurations presently in use, is curiously akin to that of aircraft just prior to the first World War. The underlying cause is the problem of stabilization, now as then. It is here, more than in any other particular, that emphasis on research and development must be placed if the hydrofoil is to realize its full potentialities. Modern techniques of analysis, model studies, and full scale evaluations developed in allied fields are certainly applicable—although the problem is severely complicated by the seaway. Even a cursory examination of the reported effects of sea state on sustained speed for conventional ships, however, shows that the speed losses incurred by them are great. It is felt that the prospect of alleviating this situation, for certain size-speed ranges, by use of the hydrofoil is great enough to warrant further serious consideration.

MISCELLANEOUS DESIGN CONSIDERATIONS

Sufficient information on hydrofoils is available to enable the designer to select the foil-strut combination to satisfy the design conditions. In most cases, however, it is difficult to rule out all but one possibility and two or more approaches may necessarily be developed in preliminary design form before the choice becomes apparent.

Generally, for small high-speed craft, the surface-piercing systems such as V or ladder configurations are preferred for simplicity and reliability.

As the size of the craft increases, geometry usually dictates that the main lifting surfaces be placed under the hull indicating submerged foil configurations. The size at which the complication of automatic control of submerged foils is required has not yet been definitely set but it appears to be in the 25-50 ton range. Even below this size, submerged foils with surface sensing such as the "Hook" configuration may be worth the extra complications if extreme seaways are anticipated and good riding qualities are required. The ladder foil system will also give excellent seakeeping but inherently gives a rougher ride. The V-foil system, while not as good in a seaway as some of the others, has proved to be one of the lowest drag configurations and should be considered wherever speed and simplicity are important.

Once the configuration has been indicated, it becomes necessary to decide on a distribution of the foil area. Here the location of the centre of gravity is of prime concern. The single foil with the main lifting surface under the centre of gravity is practical in some instances, but may involve an added drag penalty due to the necessity of a non-lifting stabilizing surface. Also, placing all the area in one foil increases the span and this becomes a disadvantage in the larger sizes. Foil areas can be distributed fore and aft, in a "tandem" arrangement with two nearly equal foils. This has advantages but in quartering seas produces asymmetrical loading and subsequent racking. The third possibility of main area aft is not as common as the others, but works satisfactorily in some configurations. Generally, if the idea of a main foil supporting most of the weight is used, the closer the centre of gravity of the boat is to this foil, the better the seakeeping qualities of the boat.

Foil areas must be carefully selected after due consideration of the effect of various foil loadings. For area contributes greatly to the parasite drag and should be kept to reasonable proportions. In order to respond to sea conditions and the variation in speed from take-off to full speed, a lift of three to four times the weight of the craft should be possible without loss of lift. The stall lift coefficient equals 0·9 or 1·0; so, in practice, the design lift coefficient at operating speeds cannot vary greatly from about 0·2 to 0·3 except for very high speeds where lower values may increase cavitation free speed.

A great variety of foil sections have been utilized for hydrofoil craft. Although most of them have been standard aircraft sections there has been no indication of a universal preference. The thickness ratio ranges from 4 to 18 per cent with the thinner sections being necessary for higher speed craft. To delay the inception of cavitation, and to reduce drag, sections should be as thin as possible consistent with strength.

Various standard aeronautical devices such as sweep back, taper, dihedral, and careful intersection and tip design can be employed with equal success in hydrofoils to improve the drag and "flying" qualities of the configuration selected. It is, of course, important to keep all the immersed portions of the configuration hydrodynamically clean by the elimination of unnecessary intersections, sharp corners, or unfaired protuberances.

The primary requirement for strut design is adequate strength with minimum drag but consideration must also be given to providing sufficient lateral area for turning. The maximum lift coefficient obtainable for a strut piercing the surface at 90 degrees is about 0·3. The optimum strut shape varies from the submerged portion to the point of surface penetration. However, most struts are made uniform in section and simple ogival (arc form) sections have proved satisfactory in service.

Hull design is a straightforward naval architecture problem influenced strongly by such factors as take-off performance, wave impact, and the concentration of loads over the strut attachments. In smaller sizes, experience has shown that hulls designed for displacement or planing operation usually have sufficient strength for foil operation. In fact, in one case, a hull that was satisfactory with foils was found to fail when subjected to low-speed planing operation.

POWER PLANT AND PROPULSION

The overall design of hydrofoil boats is materially affected by the characteristics of available power plants. As in any

* One may fairly ask if it is really necessary to avoid cavitation. Reference II, for example, presents test data indicating that the inception of cavitation is not accompanied by catastropic effects on lift and drag. From the point of view of stability, however, there is little doubt that it introduces a significant, and relatively unknown, factor into an already complex problem. Here such danger as control reversal must be considered.

boat or ship in the higher speed ranges, lower machinery weight and lower fuel consumption give a better design. In larger sizes of hydrofoils, the great growth of foil size with increased overall weight places an added penalty on heavy machinery. Since, in general, machinery weight per horsepower increases with the capacity per unit, this process is magnified to the point where very large hydrofoil craft may well depend for their existence on the development of very light high-powered machinery. In small sizes, however, the hydrofoil can compete with other craft on an equal machinery basis. Fuel rate, of course, cannot be unduly sacrificed in the search for light machinery if reasonable range is to be kept.

Transmission and shafting design is unique in the hydrofoil case. Here, the power must be transmitted from the prime mover in the hull down to a propeller at about the deepest level of foil submergence, a distance in even a small boat of several feet. In addition, the propulsion supports, gearing, bearings, and associated equipment cannot present a bulky mass under the water or large drag penalties are incurred. Several solutions have been used, such as inclined shafting from a point well forward in the boat, double right-angled gearing permitting a vertical shaft, and outboard motors in small sizes. The maximum torque that can be transmitted through any of these schemes is, to date, lower than that which can be carried on conventional shafting, thus establishing the number of shafts required in larger hydrofoil plants greater than in a conventional plant. For large sized hydrofoils, transmission development beyond presently available components is believed necessary.

In estimating propeller performance and propulsive coefficient, it is found that, unless the propeller is placed quite near foil structure or other sources of flow disturbance, the hydrofoil provides excellent flow conditions resulting in considerably higher values of propulsive coefficient than is normal for conventional craft of similar size and speed.

When considering very high speeds, the simplicity of air screw propulsion is most attractive and should be evaluated on an efficiency and weight basis against marine propulsion. To achieve reasonable ideal efficiency of air propulsion, either very large diameters or speeds of advance that are exceptionally high in terms of marine practice are required. Thus, unless speeds well over 50 knots are desired, air propulsion is not desirable. In addition, the effects of wind on advance speed could be troublesome with air propulsion.

ARRANGEMENT AND FITTINGS

The tasks of locating machinery, shafting, and other components in the hull, and the determination of arrangements, are generally controlled by the foil configuration and propulsion means selected, and by weight and moment considerations. However, good protection against spray for the pilot and excellent visibility should receive prime consideration in these high-speed vessels. Also, if passengers are to be carried, suitable means should be provided to prevent injury from high accelerations in the event of inadvertent high-speed landing. Likewise cargo-securing provisions should receive particular attention.

The ultimate use of the craft and the means for handling it in harbours or at piers should dictate the amount of complication and weight that can be afforded for retraction of foils and propulsion. Where size permits, some means of retracting foils is desirable on any hydrofoil craft for maintenance of foils without drydocking. In craft which may be required to operate alongside piers or other ships, a means of retracting the foils to positions within the over-all dimensions of the hull is nearly essential.

EVALUATION OF THE HYDROFOIL CONCEPT

Having examined the design problems of hydrofoils, it is apparent that the initial cost of such craft will undoubtedly be greater than that of conventional boats or ships of similar size. Therefore, it is important to compare hydrofoils with other surface craft to determine where this cost can be justified in terms of operating advantages.

Comparisons are first made of characteristics of specific types for the purpose of establishing the relationships of size and speed to power required. Following this, a study based upon suitable dimensionless parameters is presented to indicate to the limits of possible hydrofoil design.

SPEED AND POWER

To obtain a direct comparison with conventional ships, a brief analysis of resistance by ship standards seems in order. First, assuming identical weight, it may be found from a comparison of hydrofoils and planing boats in the higher speeds that the frictional resistance of the foil boat is approximately one-half that of a planing hull and the residual resistance is about one-half that of a planing hull*.

A typical example is as follows: —

	Good planing boat	Comparable hydrofoil boat
Speed (knots)	44	44
Length, overall (ft.) ...	45	45
Displacement (lb.)	35,000	35,000
Wetted surface (sq. ft.) ...	238	111
Frictional effective horsepower	367	200
Residual effective horsepower	402	245
Total effective horsepower...	769	445
Lift/drag ratio	6·1	10·6
Propulsive coefficient ...	0·5	0·6
Total shaft horsepower ...	1,538	740

It should be noted that in addition to a saving in frictional resistance and in residual resistance, the hydrofoil has a slightly better propulsive coefficient since its propeller is operating in nearly open water unaffected by the proximity of hull and unequal flows. Thus the total shaft horsepower required is less than half of that required for the planing hull.

The above calculation could be refined by taking into account the net difference in weight between these boats due to inclusion of hydrofoils on one, and larger engine and fuel weights on the other. However, this weight balance can be varied by the designer by his power plant selection, type of transmission, endurance requirements, and materials used. With equal endurance and comparable power plants for the two boats in question, the following approximate weight comparison shows them to be equal. However, the foil boat would have about one-half the operating cost for smooth water operation, and in most sea conditions this margin would improve for the foil boat.

WEIGHT COMPARISON (IN LB.)

Component	Good planing boat	Comparable hydrofoil boat
Hull*	13,350	13,350
Pay load	6,500	6,500
Foil system...	0	7,600
Power plant	12,300	5,950
Fuel	3,320	1,600
Total	35,470	35,000

* Hull weights are assumed equal, although a 10 per cent variation in either is reasonable.

This so-called "typical" comparison was deliberately taken at an operating size-speed range that was reasonable for both the hydrofoil and its nearest high-speed competitor, the planing boat. What happens to this comparison if it is extended to larger and smaller sizes and to much higher and lower speeds?

* This breakdown of resistance, admittedly somewhat artificial for hydrofoil craft, can be made by first determining frictional resistance by the Schoenherr formulation applied to the wetted area of immersed foils and appendages. The frictional resistance is then extracted from the computed or trial full scale resistance, leaving a residual resistance composed of wave, induced, windage and interference drags as well as a form component of profile drag.

First, in the case of a large-sized, slow-speed craft, consider these characteristics of a small merchant ship:—

Speed (knots)	15
Length, overall (ft.)	320
Displacement (tons)	6,200
Pay load and fuel (tons)...	3,000
Wetted surface (sq. ft.)	23,250
Frictional effective horsepower	1,260
Residual effective horsepower	1,965
Total effective horsepower	3,025
Propulsive coefficient	0·6
Total shaft horsepower	4,200

An attempt to determine the characteristics of a comparable hydrofoil ship leads to immediate absurdities. Assuming a lift coefficient of 0·8 (which is well above a desirable value), one finds that about 27,000 sq. ft. of projected wing area would be required to lift the hull clear at 15 knots. Roughly, two hydrofoils of 30-ft. chord and 450-ft. span would be needed—and it would be utterly impractical to place them under anything resembling a reasonable hull. Moreover, the frictional resistance would be more than doubled, while the residual resistance would be increased fivefold over that of the conventional craft, with a resulting requirement in excess of 20,000 s.h.p. Clearly, to get any possible hydrofoil either speed must increase, or size reduce, or both.

To make a more reasonable comparison, one may consider displacement of 3,000 tons and a speed of 35 knots as a design point:—

	Displacement ship	Comparable hydrofoil ship
Speed (knots)	35	35
Length, overall (ft.) ...	400	400
Displacement (tons) ...	3,000	3,000
Wetted surface (sq. ft.) ...	18,500	8,500
Frictional effective horsepower	12,000	7,000
Residual effective horsepower	25,500	34,300
Total effective horsepower	37,500	41,300
Lift/drag ratio	19·2	17·5
Propulsive coefficient* ...	0·625	0·625
Total shaft horsepower ...	60,000	66,000

* No improvement in propulsive coefficient for the hydrofoil ship is given in this example. It is assumed that greater transmission losses will at least compensate for any improvement in propeller performance over that of the displacement ship.

Here we see the hydrofoil slightly worse than the conventional ship on a speed-power basis. Assuming nearly equal power plants, the extra weight required for foils and struts, transmission, and control would come out of pay load or range, leaving the hydrofoil inferior to the displacement ship on a smooth water basis. Rough water sustained operation might make the two nearly equal.

One must note, however, that this size and speed for the displacement ship is not one that could be considered economical in the normal concept of a ship and must be justified for other reasons, such as military requirements.

If speed were held constant and size reduced, say, by 50 per cent, the above comparison would swing sharply to the favour of the hydrofoil ship. Thus a possible future for this type is indicated where there is need to retain speed on smaller size. A general discussion of the feasibility of foil boats in these sizes is contained in a following section of this paper.

At the other extreme of the size-speed range, it can be shown that the advantage for hydrofoils for very high speeds on small size is an increasing one. Although cavitation will undoubtedly affect the performance adversely, inspection of lift-drag ratios attainable for sections under full cavitation indicates that, even here, the hydrofoil should be able to surpass the ratios attainable with planing boats[11]. Although practical experience with fully cavitating foils is lacking, intuitively, it would seem that a hydrofoil under full cavitation would have a lift per unit area roughly equal to the difference between dynamic pressure on the bottom of the foil and vapour pressure on top, while a planing surface has dynamic pressure on the bottom and atmospheric pressure on top, thus giving the hydrofoil an advantage. However, many unknowns may affect this simplified picture. The possibility of stabilization difficulty has already been mentioned. Thus to stay within the bounds of reasonable experience, compare a displacement planing boat with a hydrofoil at 50 knots, which is believed to be practically attainable by both. The characteristics are as follows:—

Displacement	Planing hull	Hydrofoil
Weight (lb.)	50,000	50,000
Speed (knots)	50	50
Shaft horsepower	3,600+	1,200

Thus, even with a liberal allowance for foils and struts, a three-to-one advantage for the hydrofoil boat in power would result in a substantial increase in range or pay load.

This set of comparisons, while bracketing the areas of advantage and disadvantage for hydrofoils, leaves considerable gaps to be filled to place the hydrofoil in a definite relation to other forms of water transportation. As an aid in this problem, an attempt has been made to compare ships, boats, planing boats, and hydrofoils on a fair non-dimensional basis. A "transport efficiency" expressed as maximum speed times pay load divided by shaft horsepower was selected as one parameter, and the ordinary Froude number based on maximum speed and overall length was selected as the other. Plotting these values for various specific types caused these types to group themselves into definite patterns (Fig. 13). As would be expected, at very low Froude numbers characterized by slow ships or very large ships, extremely large values of "transport efficiency" appear. In the limit, of course, a non-propelled barge would have an infinite value on this plot.

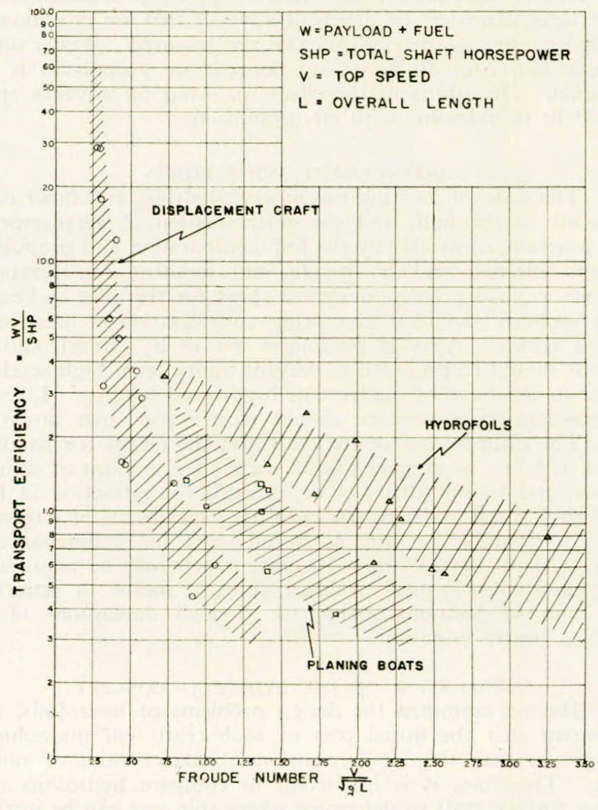


FIG. 13—*Transport efficiencies of surface craft*

As speeds increase and sizes moderate in the vicinity of Froude numbers between 0·6 and 0·7 the band for displacement ships has dropped sharply to relatively low values. Continuing this process the band for displacement types drops to near zero, which indicates an upper limit of feasible Froude number since the useful load is calculated including fuel.

A second band originating around a Froude number of 0·6 is discovered for planing forms. This band drops at a slightly lesser slope, indicating higher possible Froude numbers for this type of craft.

Further to the right (higher Froude number and higher efficiency) the family of hydrofoils appears, showing considerably less droop and much higher efficiency than other types in the higher Froude numbers. Thus from this form of limited analysis, the conclusion is indicated that from the standpoint of speed, power, size, and load capabilities, the hydrofoil is superior to other forms of water transportation above Froude numbers of 0·6 to 0·7, and that below this limit, if considered at all, a hydrofoil must be considered on other specialized grounds.

No very definite data were available to the authors on the design particulars of foil boats at supercavitating speeds. Therefore, the extreme end of the hydrofoil band and its slope at very high Froude numbers is questionable. However, it would appear that other considerations, such as stability, control, or structural integrity set the probable limits on the attainable Froude number with hydrofoils.

SIZE LIMITATIONS

From the previous discussion of speed and power, it is found that hydrofoil boats are uneconomical in comparison to other types below a Froude number of about 0·6 to 0·7. This fact indicates a minimum speed for each size of hydrofoil boat. By reference to Fig. 14 it may be seen that this relation indicates a minimum speed of about 40 knots for a 400-ft. ship, bearing in mind that length, as used herein, is merely a measure of size or bulk, assuming a hull of normal coefficients and weight loading for fast ships as shown in Fig. 15.

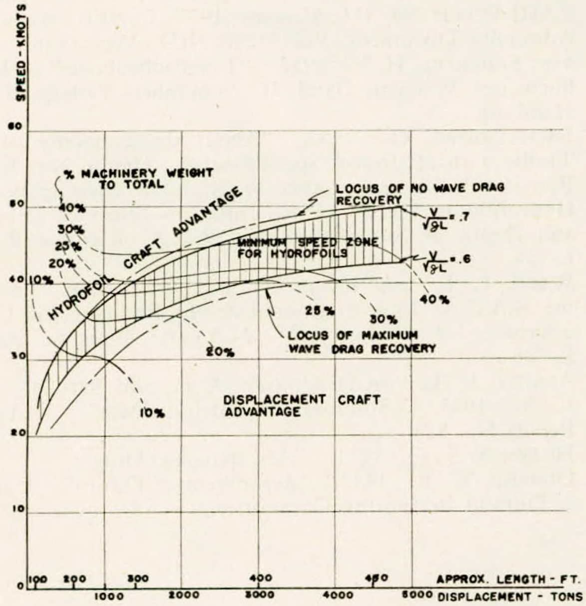


FIG. 14—*Possible characteristics for large hydrofoils*

When discussing the maximum sizes and minimum speeds for successful large hydrofoil craft, low resistance or high lift to drag ratio is most important, since this feature controls the proportion of pay load and fuel that can be carried. These factors not only contribute to the economics of the craft but

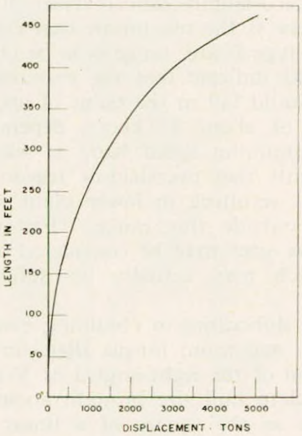


FIG. 15—*Typical weight v. length curve derived from fast displacement and planing craft*

also determine whether it will have sufficient capability to be useful at all.

Even with high lift to drag ratios in the order of 15 to 20 in mind, other limitations appear. First, while cavitation can be tolerated in high-speed small craft, it would reduce the lift to drag ratio to unacceptable amounts in a larger cargo type craft. Therefore, assuming low lift coefficients for the foils at top speed, an approximate speed limit of about 45 to 50 knots can be established for these larger craft. The intersection of this speed limit with the zone of minimum speed for hydrofoil craft on Fig. 14 indicates a maximum size in the order of 5,000 tons. Whether even this limit can be reached in practice, or would be desirable if found practicable, is open to question.

A typical ship of 500 feet will displace 15,000 to 19,000 tons while a lighter, slimmer design might go as low as 8,000 tons. Using 8,000 tons as an example, and choosing optimistic values for hydrofoil resistance and propulsive coefficient, the shaft horsepower required for 50 knots, on foils, would be over 300,000 h.p. By very rough estimating, this would leave no allowance within 8,000 tons displacement for any fuel, let alone pay load. Thus, the percentage of weight which must be allocated to machinery places an additional limit on maximum craft size, unless very radical future developments make available large capacity power plants with specific weights (lb. per h.p.) in the order of 25 per cent of those of present units.

Still another factor in the speed-size problem is the possibility of wave drag recovery by use of tandem foils. Since the spacing between foils for maximum wave drag recovery is dependent only on speed, and since in large cargo type hydrofoils a saving in the order of 20 to 25 per cent in resistance is involved, wave drag recovery controls, within limits, the relation of physical size to speed, calling for longer hydrofoil ships as speed increases. As may be seen from Fig. 14, the curve of maximum wave drag recovery falls slightly below the zone of minimum speed of economical hydrofoil craft, emphasizing the wisdom of designing to the minimum profitable speed for a cargo type hydrofoil ship.

If the best existing weight per horsepower ratios are used for plants in the 10,000 to 100,000 h.p. range, if the effects of wave drag recovery on attainable lift to drag ratios are considered, and if propulsive coefficients including shaft and gear losses in the order of 0·5 to 0·6 depending on speed are assumed, it is possible to estimate the power required and hence the machinery weight required for the larger sizes of foil craft at various speeds.

Fig. 14 also shows, in approximate terms, the relation of size to speed and to the percentage of total weight required for propulsion power. By reference to data on displacement ships at comparable speed-length ratios, about 30 per cent allowance for machinery is the maximum that could be considered useful in a cargo type if any range is to be obtained. If this is so, Fig. 14 would indicate that the maximum size of useful hydrofoil ships would fall in the range of about 1,500 to 3,500 tons at a speed of about 40 knots, depending upon which portion of the minimum speed band is selected. Of course, special requirements that overshadow reasonable efficiency, or new developments resulting in lower plant weights, may produce hydrofoils outside this range. For the present, it is believed that these sizes may be considered at least qualitative limits—ones which may actually be difficult to obtain in practice.

Some of the difficulties in obtaining even these sizes are: (*a*) limitations on maximum torque that can be absorbed in a single transmission of the right-angled or V type; (*b*) the large foil size compared to hull size brought about by the fact that foil lift increases as the square of a linear dimension while weight increases as the cube of a dimension; and (*c*) the fact that strut thickness (or the number of struts) must increase with size out of proportion to other dimensions, even when Froude scaling can otherwise be maintained.

In favour of increased size are the opportunities for making foil configurations and propulsion devices relatively cleaner, the possibilities for the use of wave drag recovery to increase efficiency at higher speed, and the fact that as long as geometrical similarity of foil configuration can be maintained with constant speed and foil loading, the stresses in the foils themselves are independent of size, although this relation does not exist for the struts.

It should be noted, however, that in all of this appraisal of larger sizes of hydrofoil ships, further development of a reliable control device for submerged foils is a prerequisite, since at large sizes (over about 100 tons) it does not appear feasible to provide sufficient lifting area or high enough lift-drag ratios with surface-piercing foils.

Special cargo handling, docking, and launching facilities would probably be required for large hydrofoil ships since at these sizes retraction of the foils would be most unattractive. Thus, these ships would have an increased draught over conventional craft of the same size when not on the foils. Also, the foil span with relation to hull beam grows with size to the point that handling alongside would become unfeasible. Other difficulties that appear with size increase include the need for the protection of the foils against fouling and the need for provisions to inspect and repair minor foil or strut damage without the opportunity to bring them out of the water except by a complicated docking procedure.

CONCLUSIONS

It may be concluded that hydrofoil supported craft are feasible and, within certain limitations of maximum size and minimum speed, are superior to displacement or planing craft on the basis of speed and power. While further development of stability and control features can improve the seakeeping qualities of hydrofoils, they generally provide a smoother ride and maintain speed in a seaway better than conventional craft.

What the future brings depends on the continued effort that is put on this problem. Certainly many applications present themselves. Fast, comfortable, point-to-point passenger service such as a commuter's ferry would provide could be operated at lower cost with hydrofoils; pleasure craft of high speed could be placed within the reach of the average sportsman if and when mass-produced foil systems are made available; and military applications for relatively small stable high-speed craft undoubtedly exist.

REFERENCES

1. GUIDONI, A. Jan. 1928. "Seaplanes, Fifteen Years of Naval Aviation". Journal of the Royal Aeronautical Society, Vol. 32, No. 205.
2. NUTTING, W. W. 1921. "The HD-4, a 70-Miler with Remarkable Possibilities". Reprinted Smithsonian Report for 1919, Publication 2595, Government Printing Office, Washington, D.C.
3. COOMBES, L. P., and DAVIES, E. T. J. Nov. 1937. "Note on the Possibility of Fitting Hydrofoils to a Flying Boat Hull". Royal Aircraft Establishment, Farnborough, Report No. B.A. 1440.
4. SOTTORF, W. Dec. 1940. "Experimental Examination of the Question of Hydrofoils". Report No. 1319, German Research Establishment for Aircraft, Institute for Seaplane Development, Hamburg.
5. GRUNBERG, V. June 1937. "La Sustentation hydrodynamique par ailettes immerges: Essais d'un système sustentateur autostable". L'aerotechnique, No. 174, 16th year.
6. KELDYSCH, M. V., and LAVRENTIEV, M. A. Nov. 1949. "On the Motion of an Aerofoil under the Surface of a Heavy Fluid, i.e., a Liquid". Paper to ZAHI, Moscow, 1935; English translation by Science Translation Service, Cambridge, Mass., STS-75.
7. KOTCHIN, N. E. Aug. 1951. "On the Wave-Making Resistance and Lift of Bodies Submerged in Water". Transactions of the Conference on the Theory of Wave Resistance, U.S.S.R., Moscow, 1937; English translation by A.I.(T) Air Ministry, R.T.P. No. 666, March 1938, Technical and Research Bulletin, No. 1-8, August 1951; The Society of Naval Architects and Marine Engineers.
8. VLADIMIROV, A. May 1946. "Approximate Hydrodynamic Calculation of a Hydrofoil of Finite Span". ZAHI Report No. 311, Moscow, 1937; English translation, Admiralty Document, PG/53280/NID, May 1946.
9. VON SCHERTEL, H. F. 1952. "Tragflächenboote". Handbuch der Werften, Band II, Schiffahrts-Verlag, Hansa, Hamburg.
10. SACHSENBERG, G. 1952. "About the Economy of the Traffic with Hydrofoil Speed Boats". Hansa, No. 30/31.
11. BENSON, J. M., and LAND, N. S. "An Investigation of Hydrofoils in the N.A.C.A. Tank; I—Effect of Dihedral and Depth of Submersion". N.A.C.A. Wartime Report L-758.
12. WARD, L. E., and LAND, N. S. "Preliminary Tests in the N.A.C.A. Tank to Investigate the Fundamental Characteristics of Hydrofoils". N.A.C.A. Wartime Report L-766.
13. ABBOTT, I. H., VON DOENHOFF, A. E., and STIVERS, Jnr., L. S. 1945. "Summary of Airfoil Data". N.A.C.A. Report No. 824.
14. HOERNER, S. F. 1951. "Aerodynamic Drag".
15. DURAND, W. F. 1943. "Aerodynamic Theory". Volume 2, Durand Reprinting Committee.

Correspondence on "The Marine Gas Turbine"

Consequent upon the publication in the November 1953 TRANSACTIONS of the paper by B. E. G. Forsling, Civ.Ing. (Member) on "The Marine Gas Turbine", a note on the use of intercoolers in compressor units was received from Mr. G. I. Pauley which is now published with Mr. Forsling's reply.

MR. PAULEY writes: Mr. Forsling states that the thermal efficiency of gas turbine plant may be increased by the use of intercooling in the compressor unit. No doubt the use of intercoolers in conjunction with heat exchangers is meant, but this is not clear from the text, which may thus be misleading. The use of intercooling alone does not necessarily increase the plant efficiency, since, although the work of compression is reduced with a consequent increase in net output, this may be more than balanced by an increase in the heat supplied, i.e. fuel consumption, due to the lower temperature at which the air leaves the compressor.

To demonstrate this, curves are here given for a hypothetical gas turbine with 100 per cent efficient intercooling, i.e. down to initial temperature (Fig. 1). A pressure ratio of

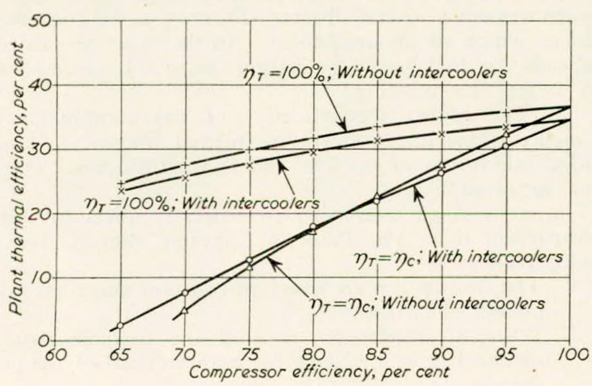


FIG. 1—*Pressure ratio 5:1*

5 : 1 is taken, and curves are drawn for the unit with and without intercooling, showing overall efficiency against compressor efficiency (i.e. adiabatic efficiency η_c) assuming (*a*) turbine efficiency 100 per cent; (*b*) turbine efficiency = compressor efficiency. Standard temperature and pressure conditions at compressor intake have been taken, and a maximum cycle temperature of 700 deg. C. The specific heats have been assumed constant with $\gamma = 1{\cdot}4$, but it is not thought that this is likely to affect general conclusions.

Over the range of compressor efficiencies shown, the use of intercoolers reduces the plant efficiency if expansion through the turbine is isentropic, i.e. $\eta_c = 100$ per cent. If, however, the turbine has the same efficiency as the compressor (about 87 per cent in practice), it would appear that the use of the intercooler increases the plant efficiency only if $\eta_c < 82$ per cent.

To determine whether increasing the pressure ratio increases the performance, similar curves are shown for a pressure ratio of 6·5 : 1 (Fig. 2). An improved performance is shown, but this is general for the whole plant with or without intercoolers. The trend of the curves is similar to those in Fig. 1, although

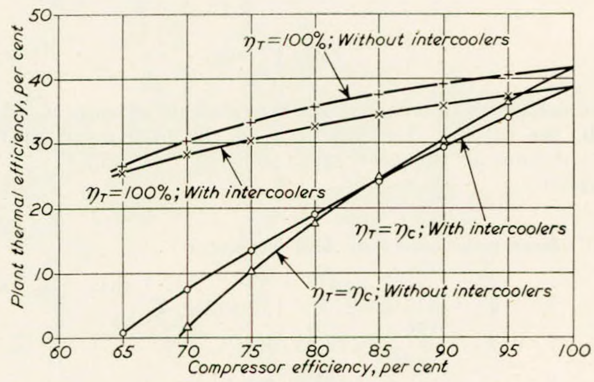


FIG. 2—*Pressure ratio 6·5:1*

with identical turbine and compressor efficiencies the intercoolers are now effective up to a compressor efficiency of 84 per cent.

Finally, in Fig. 3, curves are given for a unit having a pressure ratio of 5 : 1, operating under the same conditions as before, but using a heat exchanger of thermal ratio 0·65, a

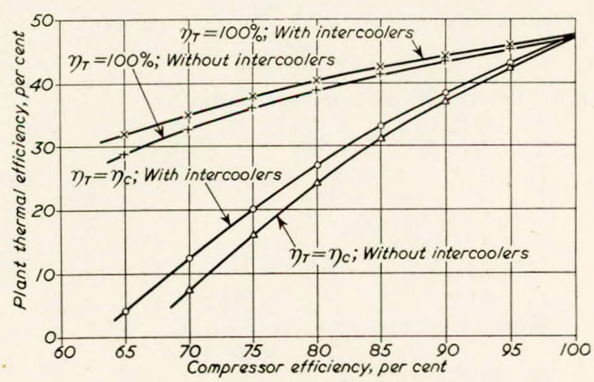


FIG. 3—*Pressure ratio 5:1*
Unit with heat exchangers

figure suggested by Mr. Forsling as being the most economical for the installation described in his paper. These show that the efficiency is increased by using a heat exchanger alone, and is *further* increased by the use of an intercooler. Thus it will be seen that the use of intercooling alone (without reheating or heat exchange) does not lead to overall improvement if the isentropic efficiencies of compression and expansion are such as may be expected today, i.e. approaching 90 per cent. A specimen calculation is given below.

SPECIMEN CALCULATION

A simple gas turbine takes in air at standard temperature and pressure and works with a pressure ratio of 5. Adiabatic efficiencies of compression and expansion are both 85 per cent and the maximum temperature in the cycle is 700 deg. C.

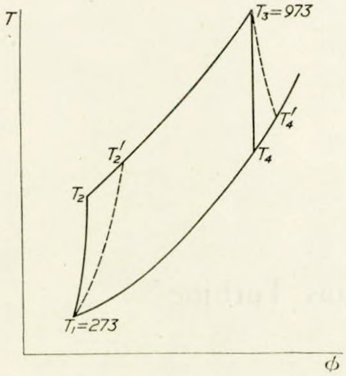


FIG. 4

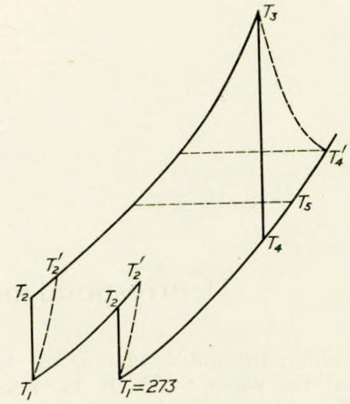


FIG. 5

Where used with intercoolers the intermediate pressure is $\sqrt{5} \times 14{\cdot}7$lb. per sq. in. The efficiency of the intercoolers is 100 per cent and of the heat exchangers 65 per cent. $\gamma = 1{\cdot}4$ throughout.

(*a*) *Without intercoolers or heat exchangers*

$$T_2 = 273 \times 5^{\frac{\gamma-1}{\gamma}} = 273 \times 5^{2/7} = 432{\cdot}4 \text{ deg. K.}$$
$$\therefore T_2 - T_1 = 432{\cdot}4 - 273 = 159{\cdot}4 \text{ deg. C.}$$
$$\therefore T_2' - T_1 = \frac{159{\cdot}4}{0{\cdot}85} = 187{\cdot}5 \text{ deg. C.}$$
$$\therefore T_2' = 187{\cdot}5 + 273 = 460{\cdot}5 \text{ deg. K.}$$
$$T_4 = \frac{973}{5^{2/7}} = 614{\cdot}4 \text{ deg. K.}$$
$$\therefore T_3 - T_4 = 973 - 614{\cdot}4 = 358{\cdot}6 \text{ deg. C.}$$
$$\therefore T_3 - T_4' = 0{\cdot}85 \times 358{\cdot}6 = 304{\cdot}8 \text{ deg. C.}$$

Work of compression $= Cp\,(T_2' - T_1) = 187{\cdot}5\,Cp$
Work of expansion $= Cp\,(T_3 - T_4') = 304{\cdot}8\,Cp$
$\therefore$ Net work per cycle $= 117{\cdot}3\,Cp$
Heat supplied per cycle $= Cp\,(T_3 - T_2') = 512{\cdot}5\,Cp$
Plant thermal efficiency $= \dfrac{117{\cdot}3}{512{\cdot}5} = 22{\cdot}9$ per cent.

(*b*) *With intercoolers and no heat exchangers*

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\sqrt{5}\right)^{2/7}$$
$$\therefore T_2 = 273 \times 5^{1/7} = 343{\cdot}7 \text{ deg. K}$$
$$\therefore T_2 - T_1 = 343{\cdot}7 - 273 = 70{\cdot}7 \text{ deg. C.}$$
$$T_2' - T_1 = \frac{70{\cdot}7}{0{\cdot}85} = 83{\cdot}2 \text{ deg. C.}$$
$$\therefore T_2' = 83{\cdot}2 + 273 = 356{\cdot}2 \text{ deg. K.}$$

$\therefore$ Work of compression $= 2\,Cp\,(T_2' - T_1) = 166{\cdot}4\,Cp$
As before, work of expansion $= 304{\cdot}8\,Cp$
Net work per cycle $= 138{\cdot}4\,Cp$
Heat supplied per cycle $= Cp\,(T_3 - T_2') = 616{\cdot}8\,Cp$
Plant thermal efficiency $= \dfrac{138{\cdot}4}{616{\cdot}8} = 22{\cdot}4$ per cent.

(*c*) *With heat exchangers and no intercoolers*

Recoverable heat $= Cp\,(T_4' - T_2')$
$= Cp\,[(T_3 - T_2') - (T_3 - T_4')]$
$= (512{\cdot}5 - 304{\cdot}8)\,Cp$
$= 207{\cdot}7\,Cp$
Actual heat recovered $= 0{\cdot}65 \times 207{\cdot}7\,Cp$
$= 135\,Cp$
Heat supplied $= (512{\cdot}5 - 135)\,Cp = 377{\cdot}5\,Cp$
Plant thermal efficiency $= \dfrac{117{\cdot}3}{377{\cdot}5} = 31{\cdot}1$ per cent.

(*d*) *With intercoolers and heat exchangers*

Recoverable heat $= Cp\,(T_4' - T_2')$
$= Cp\,[(T_3 - T_2') - (T_3 - T_4')]$
$= (616{\cdot}8 - 304{\cdot}8)\,Cp$
$= 312\,Cp$
Actual heat recovered $= 0{\cdot}65 \times 312\,Cp$
$= 203\,Cp$
Heat supplied $= (616{\cdot}8 - 203)\,Cp$
$= 413{\cdot}8\,Cp$
Plant thermal efficiency $= \dfrac{138{\cdot}4}{413{\cdot}8} = 33{\cdot}4$ per cent.

MR. FORSLING replies: Mr. Pauley is perfectly correct in his conclusions that the addition of an intercooler to the simple gas turbine set cannot be expected to give any appreciable improvement in overall thermal efficiency at the compressor efficiencies which are obtained today. In the paper he had tried to indicate this fact briefly by stating (page 271, second paragraph under "Intercooling"):—

"The higher the efficiency of the compressor the smaller is the improvement in thermal efficiency. Under ideal conditions of no loss, the thermal efficiency cannot be improved".

The set with an intercooler is, however, somewhat better in comparison than Mr. Pauley's diagrams suggest, for the following reasons:—

1. The calculations are based on constant isentropic compressor efficiencies.
2. When an intercooler is used the pressure ratio is assumed to be the same for both sections of the compressor.
3. The total pressure ratio is assumed to be the same whether an intercooler is used or not.

The isentropic efficiency of the compressor is dependent upon the pressure ratio, primarily because the losses in one stage increase the work for the same pressure ratio in all the following stages. For a comparison a better basis, therefore, is to assume constant small stage efficiency, i.e. assume polytropic compression. Taking the polytropic efficiency at 90 per cent, which is a reasonable value, the isentropic efficiency works out as follows:—

Pressure ratio	Isentropic efficiency
2·236 (= $\sqrt{5{\cdot}0}$)	88·8
2·550 (= $\sqrt{6{\cdot}5}$)	88·6
5·0	87·5
6·5	87·1

In Figs. 1 and 2 Mr. Pauley has, therefore, inadvertently assumed that the efficiency of the compressor is higher when no intercooler is used, to the extent of 1·3 and 1·5 per cent respectively for the pressure ratios referred to. Although the pressure drop across the intercooler has been neglected, the assumption of constant isentropic efficiency favours the cycle without an intercooler.

The assumption of equal pressure ratio before and after the intercooler gives the minimum work of compression when the air is cooled to the ambient temperature in the intercooler. This position of the intercooler gives the maximum output of the set.

The thermal efficiency, however, can be improved at the cost of some reduction in the net output by cooling at a lower pressure. More work is then done after the cooler so that the temperature of the air after the high pressure compressor is increased and the fuel consumption is reduced accordingly. Under given conditions the division of work which gives the highest thermal efficiency can be determined.

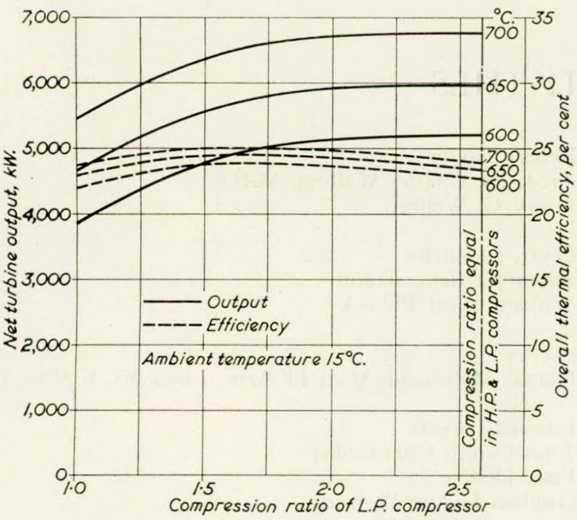


FIG. 6—*Performance of gas turbine with intercooler Open cycle—constant overall pressure ratio*

This is illustrated in Fig. 6 which shows the performance of an open-cycle gas turbine set with intercooler. The net output at the turbine coupling and overall thermal efficiency are plotted against the compression ratio of the low pressure compressor on the following assumptions:—

Mass flow (constant)	100lb. per sec.
Thermal ratio of intercooler ...	85 per cent
Cooling-water temperature (= air temperature) ...	15 deg. C.
Pressure drop across intercooler	2 per cent
Pressure drop across combustion chamber with ducting	2·5 per cent
Polytropic efficiency of compression	90 per cent
Overall pressure ratio of compression (including intercooler pressure drop) ...	6·5 per cent
Isentropic turbine efficiency ...	88 per cent for 600 deg. C. 87·5 per cent for 650 deg. C. 87 per cent for 700 deg. C.

As various losses have been neglected, the performance to be expected from an actual set would be somewhat inferior.

The diagram (Fig. 6) shows that a modest increase in thermal efficiency is obtained at a comparatively low compression ratio of the low pressure compressor (about 1·75). If the compression ratio is increased beyond this point the net output increases slowly. It is interesting to note that if the pressure ratio of the two compressors is the same the overall thermal efficiency is about the same as that for a set without intercooler.

In order to get a fair comparison between the cycles with and without intercooling, the optimum pressure ratios for both cycles should be determined. As mentioned in the paper, when an intercooler is added a higher pressure ratio should be adopted than for the simple cycle. In order to determine the influence of the pressure ratio on the thermal efficiency with acceptable accuracy, the influence of various losses must be taken into account, for instance, gland leakage losses which increase with the pressure ratio, reduced stage efficiencies in the high pressure range of compressor and turbine, due to shorter blade lengths, and the effect of relative blade speed on turbine efficiency.

In this connexion it may be sufficient to add that some improvement in the thermal efficiency of the intercooler cycle will be obtained by increasing the pressure ratio above 6·5.

On the same assumptions as for Fig. 6, the following figures are obtained for the cycle without intercooler:—

Turbine inlet temperature, deg. C. ...	650	650	700	700
Pressure ratio	5·0	6·5	5·0	6·5
Net turbine output, kW	4,850	4,779	5,545	5,565
Overall thermal efficiency	22·07	23·49	22·58	24·22

On the assumptions made the overall thermal efficiency will have a flat maximum at about 6·5 pressure ratio. Comparing these figures with Fig. 6 shows that only a small gain in thermal efficiency has been obtained with an intercooler, but the net output has gone up quite considerably (12-15 per cent) for the same air mass flow. This gain, however, is generally insufficient to compensate for the extra cost and added complications of an intercooler, so that a simple intercooler set is not attractive.

The assumption of constant specific heat taking the index for isentropic expansion at 1·4 gives a somewhat too high thermal efficiency and too low a net output, as will be seen from the following table, which is based on a 650 deg. C. turbine inlet temperature.

	Thermal efficiency, per cent	Net output, kW
Index for isentropic expansion = 1·4	23·53	4,641
Variable specific heat allowing for fuel added	22·48	4,940

This table allows for no leakage losses, whereas Fig. 6 and the previous table assumes that leakage losses equal fuel added.

For a marine gas turbine set the intercooler would be used in combination with a heat exchanger. Under these conditions a considerable increase in thermal efficiency is obtained

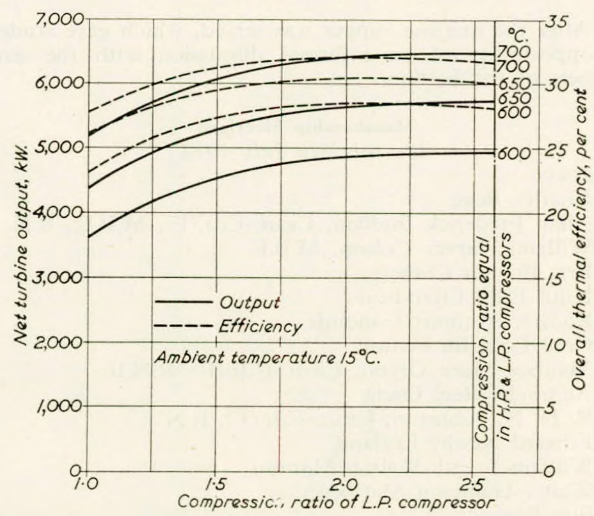


FIG. 7—*Performance of open cycle gas turbine with intercooler and heat exchanger: constant overall pressure ratio*

by the intercooler, as is shown in Fig. 7. The thermal ratio of the heat exchanger has been assumed to be 65 per cent and pressure drops of 4·25 per cent have been allowed for the heat exchanger (air and gas side), including ducting. Otherwise the same assumptions as for the plain intercooler cycle (Fig. 6) have been used. By adding a heat exchanger the optimum pressure ratio of the low pressure compressor is increased to about 2·0, i.e. the intercooler is more effective.

INSTITUTE ACTIVITIES

Sydney Local Section

General Meeting

A general meeting was held at Science House, Sydney, on Thursday, 15th July 1954, at 8.0 p.m. Engineer Captain G. I. D. Hutcheson, R.A.N.(ret.) (Local Vice-President) was in the Chair and seventy-two members and guests were present. A paper on "Electronic Equipment and Its Application to Industry" was presented by Mr. N. H. Hicks, and the lecturer demonstrated a number of different types of electronic equipment under actual operating conditions. Messrs. Weymouth, Sime, Lawrence, Mercer, Ridgley, Buls, Hall, and the Chairman, took part in the discussion which followed.

A vote of thanks to the lecturer was proposed by Mr. H. G. Ferrier and carried with acclamation.

Students' Lecture

On Tuesday, 29th June 1954, at 8.0 p.m., a students' meeting was held at Science House, Sydney. There was an attendance of eighty-one, comprising fourteen members and sixty-seven students and apprentices; Engineer Captain G. I. D. Hutcheson, R.A.N.(ret.) (Local Vice-President) was in the Chair.

Mr. L. Bateman delivered a lecture on "Marine Diesel Engines" and illustrated his remarks with lantern slides. Numerous questions were asked by the students in the discussion which followed and these were very fully answered by the lecturer and by the superintendent engineers present. A vote of thanks to Mr. Bateman was proposed by Mr. G. E. Arundel, seconded by Mr. J. Munro, and carried by acclamation.

After the meeting supper was served, which gave students the opportunity of an informal discussion with the senior members of the Section.

Membership Elections

Elected 28th July 1954

MEMBERS

Charles Begg
Allan Frederick Budden, Lieut.-Cdr.(E), M.B.E., R.N.
William Harvey Colson, M.B.E.
Eric Bowyer Corbett
Knut Eskil Erlandson
Roderic Anthony Esmonde
Cecil Longdin Fenton
Maurice Mark Glynn, Lieut.-Cdr.(E), R.N.R.
Alexander Reid Greig
R. H. N. Johnston, Lieut.-Cdr.(E), R.N.
Edward Newby Leyland
William Joseph Walton Manton
Walter Gameson Matthews
Ilija Pejovic
Alan Philliskirk
Alan Blythe Robison
Robert Taylor
Frederick Charles Walling, M.B.E.
Harry G. Webber

ASSOCIATE MEMBERS

Raymond Bruce Dann
William Scott Pollock

ASSOCIATES

Gamal Mahmoud Abou El Azm, Lieut.(E), E.N.F., B.Sc. (Eng.)
Edward Barnett
John Dennis Christopher
Peter Dobbie
Ghulam Halder Durrani
Arthur Ernest Fitton
Kenneth Alfred M. Goodyear
Sailendra Nath Das Gupta
Kenneth Kirtley
Harry Leah
David Allison Lock
Louis Gordon Lowe
Sydney John Maude, M.B.E.
Henry Chalmers Morris
William Edward Naylor
Walter Percy Noble
Vadakkemadam Manikkam Raghu
Andrew McCrae Russell
Suryakant Yadneshwar Shrikhande, Sub. Lieut.(E), I.N.
Herbert James Tate
William Whyte
Wilfred Ernest Wolfenden

GRADUATE

Jawaher Lal Gupta, Sub. Lieut.(E), I.N.

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER

Stephen Kirk Pearson

TRANSFER FROM ASSOCIATE TO MEMBER

Clifford Angus Hardy
Joseph Stanley Townsend
Henry Stephen Wood

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Alfred Johnston

TRANSFER FROM STUDENT TO GRADUATE

Iain Ramsay McLeod

TRANSFER FROM PROBATIONER STUDENT TO STUDENT

David Wyndham Brinsdon