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Re-Evaluation of the Planing Hull Form

Daniel Savitsky*
Stevens Institute of Technology, Hoboken, N.J.
and
Jerry L. Gore†
David W. Taylor Naval Ship Research and Development Center, Bethesda, Md.

Introduction

THE planing hull form is perhaps the oldest, simplest, and most extensively employed member of the family of so-called high performance marine vehicles. It is the authors' intention to demonstrate the inappropriateness of stereotyping planing craft as small underpowered boats which must struggle to "get over the hump" and then subject the structure and personnel to severe pounding, wetness, and discomfort when operating in a seaway. It will be shown that, by appropriate application of recently developed technology, planing forms have evolved which are devoid of "hump" problems; demonstrate excellent behavior in a seaway, have substantial useful load fractions, and have a potential growth up to displacements of nearly 1000 tons.

This paper defines the planing hull form, presents a brief historical evolution of the craft, summarizes the recently developed technology and its impact upon design and application, and, finally, discusses the potential of the craft.

Fundamentals of Planing-Type Hull Design

Wave Formations

The planing hull form evolved to overcome the inherent hydrodynamic limitations associated with high speed operation of the traditional displacement hull. It is well to compare briefly both hull types and to identify what changes were brought about by the demand for higher speeds and why the shape of the planing hull evolved as it has.

Displacement hulls operate at speeds where the speed/length ratio $V_K/\sqrt{L}$ does not exceed 2 knots ft$^{-1/2}$ for fast boats; and by far the largest amount of operation will likely occur at speed/ length ratios of 0.7-0.9 knots ft$^{-1/2}$. This dimensional ratio is related to the nondimensional Froude number, *FN:*

$$FN = \frac{V}{\sqrt{gL}} = k\frac{V_K}{\sqrt{L}} \tag{1}$$

where V is speed in ft/s, g is the acceleration of gravity, and the constant k is equal to 0.298 ft$^{1/2}$/knots. For the purpose of simplicity the units for speed/length ratio have been omitted from the figures and further discussions in this paper.

The translation of the hull through the water produces surface waves which move at the speed of the hull. These waves have a fixed relationship between their length, ℓ and speed V as given by:

$$V = \sqrt{\frac{g\ell}{2\pi}} \tag{2}$$

Since the generated wave pattern remains fixed relative to the hull, the ratio of wave length ℓ to hull length L is:

$$\ell/L = 6.33\ (FN)^2 = 0.56(V_K/\sqrt{L})^2 \tag{3}$$

Dr. Daniel Savitsky has been with Stevens Institute of Technolgoy for nearly 30 years and is currently Deputy Director of the Davidson Laboratory and Professor in the Ocean Engineering Department. In 1971, he earned his Ph. D. in Oceanography at New York University. Dr. Savitsky's research activities have dealt with high performance vehicles with particular emphasis on planing hull forms. He is the author of some 65 papers and, in 1968, received the SNAME Admiral Cochrane Award for his paper "The Hydrodynamics of Planing Hulls." He was a member of Planing Hull Advocate group in the recent ANVCE study and is currently Chairman of the High-Speed Vessel Panel of the International Towing Tank Conference and a member of the H-12 Planing Panel of SNAME.

Mr. Jerry Gore earned his bachelor's degree in 1962 with dual majors in Physics and Mathematics and immediately joined the staff of the Navy Engineering Experiment Station in Annapolis. His experience over the intervening 17 years at the David W. Taylor Naval Ship R&D Center has included fluid studies, acoustics, silencing, ship and submarine tests and trials, small craft development and product improvement, ship propulsion system analysis, and ship system development and prototyping for the U.S. Navy. He was the "Planing Vehicle Advocate" for the Advanced Naval Vehicles Concepts Evaluation (ANVCE) Project Office in OPNAV, and he has recently completed an assignment in the Office of the Assistant Secretary of the Navy for Research, Engineering and Systems (ASN(RE&S)) as their Staff Assistant for Vehicle Systems. Mr. Gore's major management responsibilities, however, have been as the former Technical Manager for the Naval Special Warfare Craft Program. Currently, he is the Technical Manager of the Small Waterplane Area Twin Hull (SWATH) Ship Development Program. He is a member of ASNE and SNAME.

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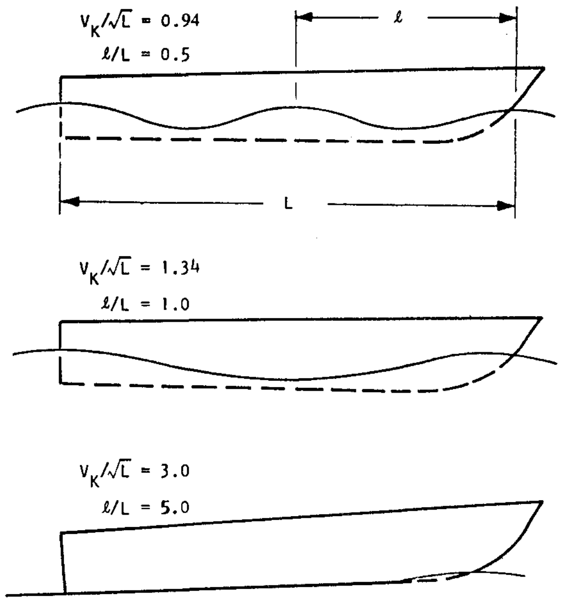

Fig. 1 Wave patterns vs speed/length ratio.

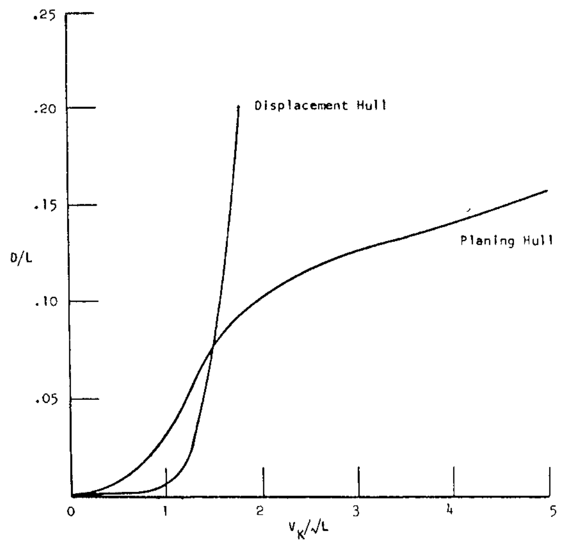

Fig. 2 Typical curves of drag/lift ratio vs speed/length ration.

It is seen from Eq. (3) that the Froude number and the speed/length ratio, which is proportional to it, really define the ratio of the length of the wave made by the hull to the length of the hull. As demonstrated in Fig. 1, at a speed/length ratio of 0.94 ($FN = 0.28$), the wave length is one-half the hull length. Such short waves exert little effect on the hull and the drag will be predominantly of viscous origin. At a speed/length ratio of 1.34, which is the speed/length ratio of every deep water wave ($FN = 0.40$), the wave length will be equal to the hull length and large increases in wave making drag will be encountered. Further speed increases will produce waves longer than the hull length so that the craft will trim to the slope of the bow waves it has generated (Fig. 1), adding considerably to the total drag. A typical drag curve for a displacement ship is shown on Fig. 2.

Dynamic Pressures

In addition to the significance of Froude number in describing the surface wave formation, FN also relates the relative importance of dynamic and hydrostatic pressures acting on the hull. If Eq. (1) is squared, and the numerator and denominator are multiplied by $\frac{1}{2}\rho$, the following is

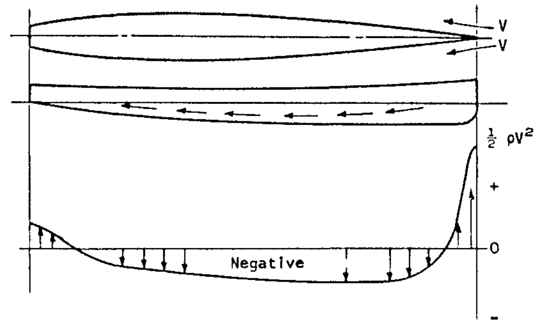

Fig. 3 Longitudinal pressure distribution for displacement hull.

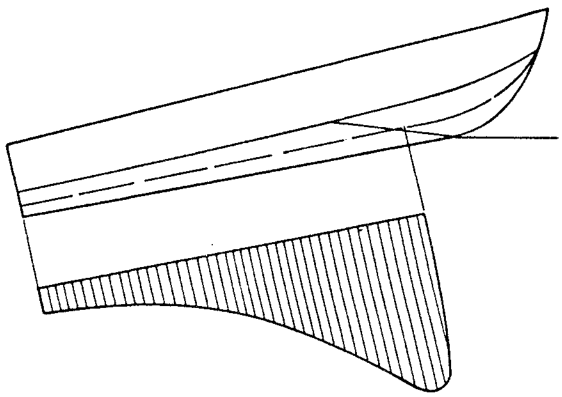

Fig. 4 Typical longitudinal pressure distribution for planing surface.

obtained:

$$(FN)^2 = \tfrac{1}{2}\rho V^2 / \tfrac{1}{2}\rho g L = 0.089\ (V_K/\sqrt{L})^2 \qquad (4)$$

The numerator is recognizable as kinetic energy—numerically equal to the stagnation pressure. The denominator represents the potential energy due to a hydrostatic head. The square of the speed/length ratio therefore defines the ratio of kinetic energy to the potential hydrostatic energy, or essentially the ratio of dynamic or lifting forces to static or buoyant force. At low values of speed/length ratio, say 0.60, the dynamic forces can be neglected and the weight of the ship will be supported entirely by buoyancy. For speed/length ratios corresponding to planing boat operation, $V_k/\sqrt{L} > 3.0$, the reverse is true—the dynamic pressures strongly predominate over hydrostatic pressures.

In the case of a displacement ship, the water flow along the bottom will essentially follow along the buttock lines which have convex curvature in order to minimize flow separation at the transom. A typical longitudinal pressure distribution for a displacement ship is shown in Fig. 3. At the stem, the velocity is zero and full stagnation pressure $\frac{1}{2}\rho V^2$ is developed. Following the streamlines along the convex buttock lines, the local velocities must become larger than the translational velocity of the hull so that, by Bernoulli's equation, the net dynamic pressure on the bottom becomes negative or, more popularly, a suction force develops along most of the hull bottom. At normal speeds for dispacement hulls, these suction forces have only a small effect upon the trim, draft, and resistance. However, as the speed increases, the negative pressures will increase as the square of the speed. This results in large trims by the stern, large increases in draft, and enormous increases in resistance. The hull actually sinks deeper and deeper into the wave it is generating—and it is this hydrodynamic phenomenon which limits the speeds of displacement hulls.

The planing hull form is configured to develop positive dynamic pressures so that its draft decreases with increasing

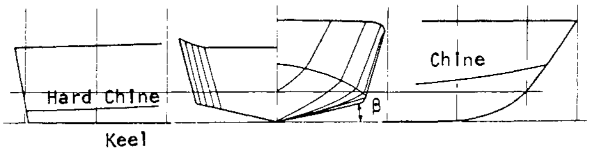

Fig. 5 Typical high speed planing hull geometry (Series 62).

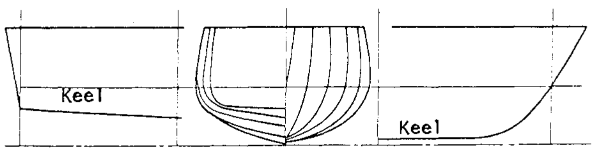

Fig. 6 Typical high speed displacement hull geometry (Series 64).

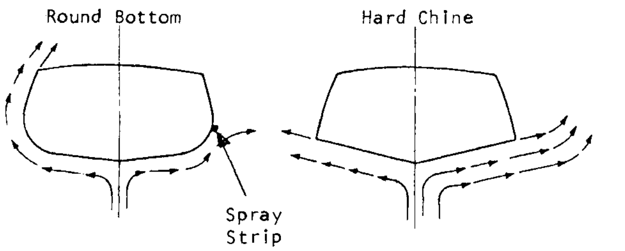

Fig. 7 Transverse flow patterns for round bottom and hard chine planing hulls.

speed, enabling it to ride higher and higher on the wave it is generating, thus avoiding the enormous drag increases associated with the very high speed displacement hull (Fig. 2). A typical bottom pressure distribution for a planing hull is given in Fig. 4.

Typical Planing Hull Form

To attain positive dynamic pressures, the planing hull form eliminates convex curvature of the buttock lines so that the keel and chines are straight in the aft part. There is, of necessity, some convex curvature of buttock lines in the bow area; but this part is above the water at high speed. Whereas in the displacement ship all means are taken to reduce flow separation at the stern and to preserve the smooth flow conducive to the recovery of pressure at the stern, in a planing boat the straight buttock lines are cut off clearly by the transom stern so as to induce early flow separation. The transverse section is typically a deadrise section with a sharp intersection of the bottom and sides to form a hard chine. Representative lines for a high speed planing hull are shown in Fig. 5.

High speed displacement boats usually have straight buttock lines, but "round" bottoms (soft chines) in transverse planes (Fig. 6). The transverse flow component around the bilges of the round bottom will produce negative pressures in these areas and, at very high speeds, will cause the water to flow up along the sides making for a "wet" boat. These effects increase the trim and resistance of the boat, but are correctable by inducing transverse flow separation through the use of a longitudinal breaker strip (Fig. 7). Quite a small stringer will "break" the suction and cause the flow to separate or detach from the hull. A hard chine hull naturally provides for high speed flow separation at the chine, generates positive pressures along the bottom, and thus becomes the recommended configuration for planing at speed/length ratios of 3.0 or more.

Further discussions of the relationship between planing hull geometric details, proportions, loadings etc., upon powering, seakeeping, and utilization will be undertaken in later sections of this paper. The important conclusion to be drawn at this point is that, for speed/length ratios in excess of 3.0, the planing hull form must be configured to develop positive dynamic pressures which lift the hull, thereby significantly reducing the wavemaking drag while also reducing the wetted surface area. In general terms, a rise in the center of gravity (c.g.) coupled with flow separation at the chines and transom are conditions which signify that the hull is "planing."

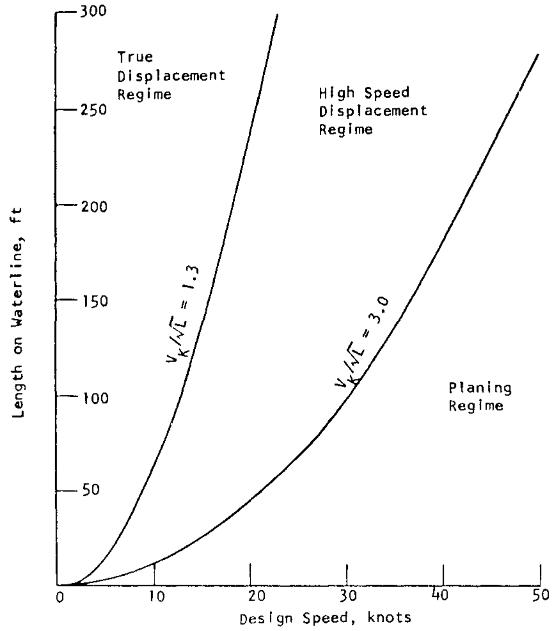

Fig. 8 Speed regimes for various hull forms.

Typical Speed Regimes for Various Hull Types

Representative speed regimes for operation of displacement vessels, high speed displacement hulls, and planing hulls are graphically depicted in Fig. 8 as a function of waterline length and speed. It is seen that, for speed/length ratios less than approximately 1.3, a displacement hull form should be used. For speed/length ratios between 1.3 and 3.0, hull forms should be of the high speed displacement type. Planing hull forms should be used for speed/length ratios in excess of 3.0. Figure 8 can be used as a guide in selecting the proper hull type early in the design process.

Although planing hull resistance will be discussed in a later section, it may be of interest to provide an envelope of lift/drag ratio vs speed/length ratio for vessels typical of each of the regimes identified in Fig. 8. Such a curve is plotted in Fig. 9 which shows significant decreases in lift/drag ratio with increasing speed/length ratio. Also spotted on this curve are typical vessel types at their appropriate speed/length ratio.

Historical Evolution of Planing Craft

Today's light-displacement, high-speed, small combatant ships and ocean-capable patrol craft have evolved beginning in the latter 1800s. As the 20th century began, the primary purpose of these vessels was either to carry the newly developed self-propelled torpedo or to defend against its use; hence, the names "Torpedo Boat" and "Torpedo Boat Destroyer" were coined and are with us still. At first these vessels were driven by piston steam engines and generally employed round bottom hull forms. As early as 1872, the Rev. C. Ramus proposed the development of a 2500 ton stepped planing ship which was rejected as unrealistic by William Froude. In 1894, Sir Charles Parsons, the English inventor, made the first marine installation of a steam turbine engine in his own specially designed 45 ton displacement hull Turbinia. Her round bottom and long slender proportions permitted her to attain a speed of nearly 35 knots in 1896 after suitable propellers were finally fitted to her triple shafts.

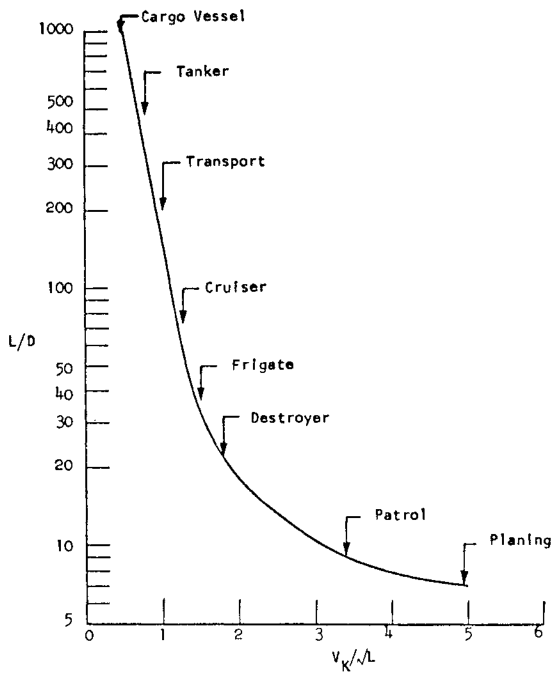

Fig. 9 Typical lift/drag ratios as function of speed/length ratio.

During this arduous development period, the British Admiralty was kept abreast of progress but showed so little interest that Parsons finally decided to demonstrate Turbinia at Spithead in the 1897 Naval Review of Queen Victoria's Fleet under the noses of the Lords and Admirals and in full view of the largest gathering yet of marine and naval experts from around the world. As Parsons raced among the assembled guest warships, various attempts were made to intercept him to no avail. In the end, he persevered. Having to resort to such tactics in public is too often required in trying to advance ideas against conventional attitudes.

The turbine continued to be developed in ever larger sizes which made the evolution of the Torpedo Boat Destroyer possible. As the original counter to the smaller Torpedo Boat it required more payload and seakeeping ability to be effective against the threat. Meanwhile, the newly invented internal combustion engine was becoming the key to regaining the speed advantage and furthering the evolution of the smaller craft due to its lightness and the range-enhancing characteristics of its fuel as compared with carrying bulky coal and water for steam. In about 1896, Britain began to issue several patents for hull designs which had all the features of a planing hull including a transom stern and one or more transverse steps. In 1910, Thornycroft, in England, was experimenting with a 25 ft long single stepped planing hull which achieved a speed of nearly 35 knots. This ultimately led to the design and construction of the 55 ft Coastal Motor Boat (CMB), a 46 knot, 14 ton single step planing hull which carried two torpedos and a crew of five and fought successfully in World War I in the very rough waters of the North Sea and English Channel. The design of a 70 ft CMB was available for production when the war ended.

The Second World War brought substantial refinement and continued development which saw the hard chine hull form evolve to equal status with the round bilge forms so prevalent earlier. Great Britain, Germany, and the United States, each anticipating their own needs, had begun to pursue in the 1920's and 1930's, various approaches which ultimately led to standardized classes of "PT" boats employed all over the world. The early parentage of the planing hull forms as we know them today was derived from the vast body of experience stemming from this war, with Great Britain, and the U.S. favoring hard chines and Germany favoring round bilges.

To capitalize on the impressive German WWII E-boat capabilities, in 1948 the British constructed two prototypes called the Bold Class. Bold (Pathfinder) was produced in round bilge form while its sister vessel used a planing hull with hard chines. Pathfinder was the last British round bilge planing boat built, all successors being hard chine designs. A succession of followup efforts was undertaken, as shown in the Small Combatant Family Tree (Fig. 10) spurred by the outbreak of the Korean War. In Britain these included the 73 ft Gay Class, a design quite similar to the World War II Motor Torpedo Boats (MTB's) and powered by the Packard gasoline engine; and the Dark Class, a 68 ft boat, capable of 40 knots and the first class of boats to use the Napier "Deltic" diesel engine. The early 1960's marked the real beginning of the high performance era with the Brave Class which ultimately attained 50 knot speeds with her Rolls-Royce gas turbine propulsion engines and transcavitating propellers.

When U.S. PT Boat (Patrol Torpedo) needs became obvious in the early 1940's, the British Navy's Packard-engined, Thorneycroft-designed MTB's along with the 70 ft Scott-Paine design served as parent vehicles from which the 80 ft Elco and 78 ft Higgins PT boats evolved through the war years. The U.S. Navy's post-WWII program was late in starting and consisted of developing a new class of PTs which would capitalize on both foreign and U.S. World War II experience. This program spawned a family of four aluminum hull PT boats (hull Nos. 809-812) which first saw service in the early 1950's. Each boat was different from the others (812 was round bilge, the others had hard chines) and all were capable of speeds in excess of 40 knots. These boats operated as a squadron under the Navy's Operational Development Force from 1954 to about 1959 and then became the "leftovers" from an era of unequalled intensive design and construction experimentation which included, in 1941, two long distance races held among competing designs. These became known as "Plywood Derbies"and during the second one, held in August 1941, the destroyer Wilkes was assigned to accompany the PT boats on a 185 mile run in swells of 6-8 ft with occasional waves of 10-12 ft (eventually to 15 ft).

The Wilkes was directed to run as nearly as possible at full power and ran the course in 6 h 18 min at an average speed of 29.8 knots. The Board of Inspection and Survey, who supervised the test, reported that this time was only 25 min better than PT 21 which finished in 6 h 43 min at an average speed of 27.5 knots. The Board further observed that for the assigned mission, modern destroyers possessed no sensible advantage over the motor boats under sea conditions highly unfavorable to the boats.

No requirements were put forward leading to subsequent U.S. PT boat developments until the early 1960's when events in Southeast Asia created a need for fast coastal patrol craft. At this point, the U.S. Navy surveyed domestic and other free world patrol craft available for immediate acquisition and procured the Norwegian Nasty design. In addition to outright procurement of several craft from Norway, a U.S. construction progam was initiated with John Trumpy and Sons in Annapolis. These wooden craft could achieve speeds in excess of 40 knots but pounding at high speed in waves was severe. In an attempt to acquire a more seakindly boat quickly, the commercially developed aluminum Osprey Class was placed in naval service. Both classes used the Napier Deltic diesels previously employed in the British Dark Class.

At this point (early to mid-1960s) both the British and U.S. Navies had achieved similar positions with respect to their high performance hull configurations with one exception—the British Navy had dropped the complex Napier Deltic diesel and had installed the Rolls-Royce Proteus gas turbine in the new Brave Class. This gave it "benchmark" status in performance for that era with speeds well over 50 knots.

During this post-WWII era, a similar evolution was occurring in Germany and the Soviet Union. Their programs

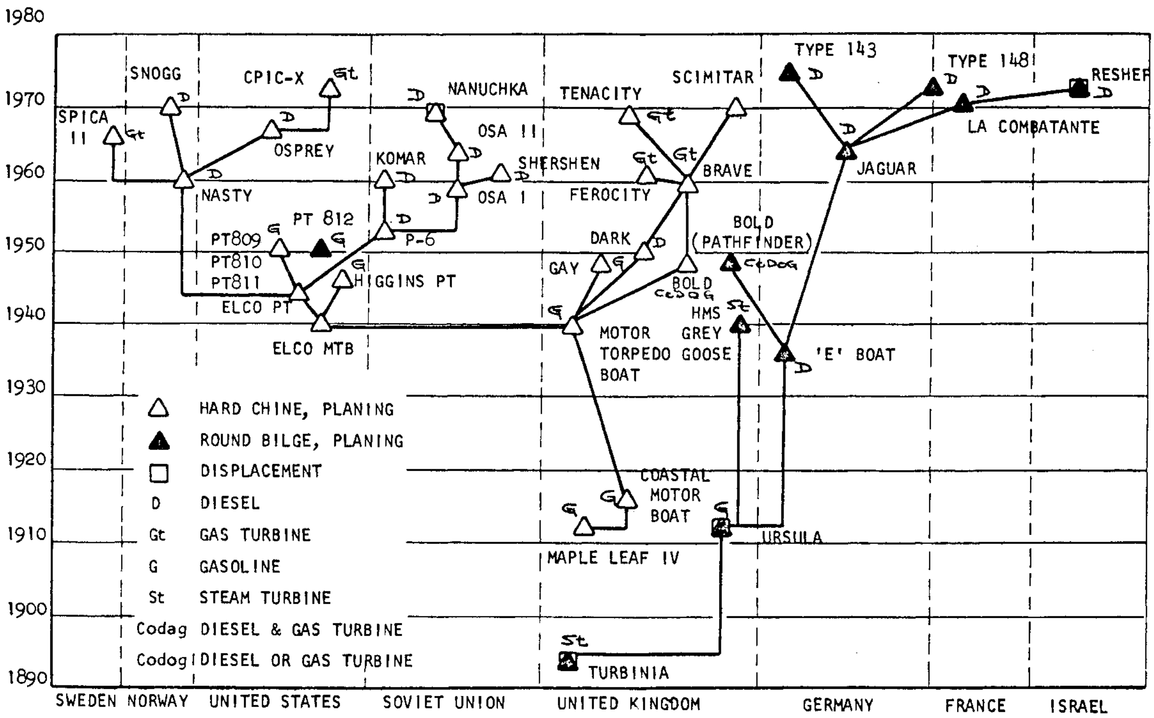

Fig. 10 Small Combatant Family Tree.

had produced the West German Jaguar Class Patrol Craft, and the Russian OSA Class (Fast Attack Craft-Missile) and Nanuchka Class (Missile Corvette), The 139 ft diesel-propelled Jaguar, with a 23 ft beam and displacing 190 tons, is round bilge forward but becomes hard chine in approximately the after one-third of the hull. The OSA is a 128 ft hard chine, 210 ton boat with a 25 ft beam. This craft has been exported by Russia. The Nanuchka, at nearly 1000 tons with an overall length of 193 ft and a 40 ft beam, is thought to be unique among the modern large high performance craft in having a hard chine hull configuration. Both Russian craft use lightweight, high-speed diesel machinery and make over 30 knots.

A Small Combatant Family Tree (Fig. 10) provides some insight into the timing and chosen paths of planing hull technology pursued by selected nations as they relate to hard chine and round bilge hull forms and to machinery selections.

The completion of the U.S. Navy's experimental PT boat development effort in the early 1950's was followed by extensive laboratory research and experimentation. During this effort various modeling techniques were developed which involved several comprehensive hull series programs in which prismatic surfaces were optimized for smooth water, with little emphasis on seakeeping. This emphasis on calm water performance generated a basic foundation of knowledge which is still useful today but probably did more harm than good by providing data that seemed to imply to the design community "don't worry about speed or motions in a seaway—whatever you get as fallout after optimizing for calm water is what you must live with." In July 1966, the Director of Defense Research and Engineering (DDR&E) directed the development of improved naval craft for use in the riverine and coastal environments of Southeast Asia. At this point the Navy reawakened to a need which was not well understood technically and rapidly initiated a series of engineering development programs intended to produce a variety of specialized inshore warfare craft quickly.

The Quick Reaction Capability (QRC) approach taken to supply various types of small craft combatants to Vietnam was doomed to take what was available. And we did—the Nasty was just the beginning. The authors could run through a list of designators for various vessels that would leave you with glazed eyes. For example, here are just a few: PTF, PCF, PBR, LCSR, LCSR(L), ATC, MON, ASPB, QFB, LCPL, LSSC, MSSC, HSSC, RUC, STAB! Furthermore, there were boats in wood, glass reinforced plastic, aluminum, steel, and ferro cement. There were diesels, gasoline inboards and outboards, and gas turbines; there were different varieties of everything used in these boats. The point is, the Navy was not prepared for this kind of situation and if it should happen again there is no program presently in place to prevent history from repeating itself. Meanwhile, a well-conceived basic hull research and development program was being undertaken in the mid-1960s to improve seakeeping and reduce resistance in the preplaning range. As a consequence of this program the development of the Experimental Coastal Patrol and Interdiction Craft (CPIC-X) became feasible and was begun in 1970 under an Advanced Development Objective for Special Warfare Craft. This program included both coastal and riverine requirements for more performance than was achievable from the then existing commercial sources. CPIC-X was designed, built, and extensively tested as a preproduction prototype. It met its intended operational specifications in all respects, but never went into series production. It has, however, become the U.S. "benchmark" design which other designers show evidence of leaning upon when making their own contributions toward meeting those conflicting demands for the best compromise of high speed and seakindliness in one hull form with minimum cost and complexity.

Summary of Technology and Impact on Design and Application

Smooth Water Drag

The smooth-water drag is predominantly dependent upon hull deadrise, trim angle, and length/beam ratio. The effects of these prime design variables are summarized below and their influence on hull design discussed.

Effect of Trim and Deadrise

Planing craft hydrodynamic technology is based primarily upon experimental data obtained from tests of prismatic planing surfaces, such as those reported by Savitsky,[1] and the

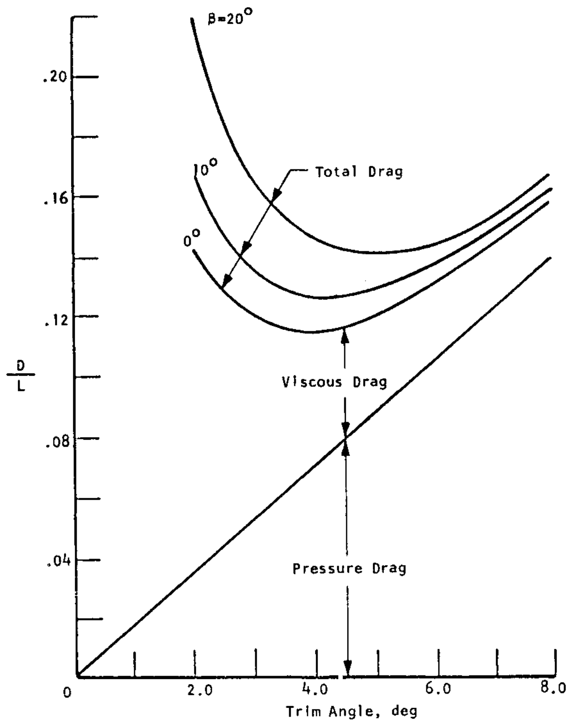

Fig. 11 Variation of drag/lift ratio for prismatic planing surfaces.

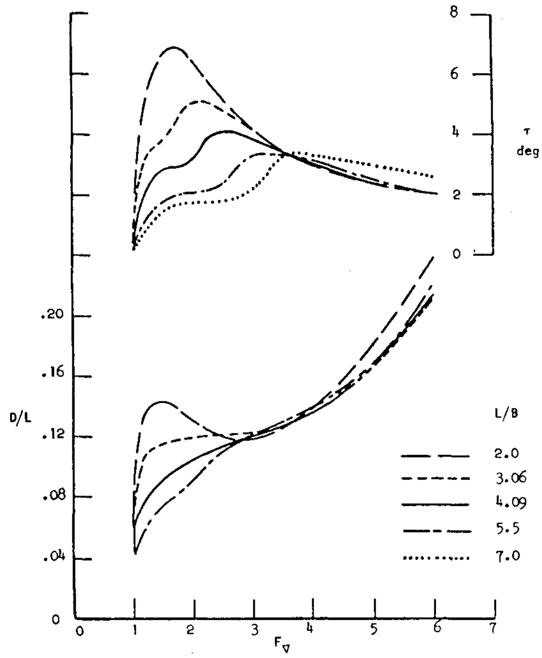

Fig. 12 Drag/lift ratio and angle of attack vs Froude number for Series 62.

results of hull series tests, such as Series 62 reported by Clement and Blount[2] and Series 64 reported by Hubble.[3] Empirical equations have been developed to define the engine horsepower of bare hulls (Savitsky[1]) and the shaft horsepower of fully appended powered hulls (Hadler[4]). The relationship between hull drag/lift ratio and the primarily planing variables at full planing speed is succinctly summarized in Fig. 11 for a hard chine prismatic hull. The drag/lift ratios are slightly dependent upon speed and mean-wetted-length/beam ratio. The important hydrodynamic characteristics to be noted are:

1) The drag/lift ratio is primarily dependent upon trim angle with the optimum trim at approximately 4 deg.

2) At trim angles less than 4 deg, the viscous drag due to bottom friction dominates, while at larger trims, pressure drag due to dynamic lift generation dominates. For typical hull forms, low trim angles will also immerse the bow, further adding to the total resistance.

3) The drag/lift ratio increases significantly with increasing bottom deadrise, especially at low trim angles.

4) For trim angles less than 4 deg, the drag/lift ratio decreases with increasing trim angle. This is a beneficial feature that reduces the drag penalty due to overloading since, all other parameters being equal, planing hull trim angles increase with increased loading.

If the sole design requirement was to provide minimum power at high speed in smooth water, then it would be concluded, from Fig. 11, that a flat bottom hull planing at a trim angle of approximately 4 deg would be the ideal combination of hull form and trim attitude. Unfortunately, this selection would be unacceptable for several practical reasons:

1) At high speed, the combination of $\beta = 0$ deg and $\tau = 4$ deg most likely will result in longitudinal instability, that is, porpoising.

2) When operating in a seaway, the flat-bottom hull will develop severe wave impact accelerations (as discussed in a subsequent section on seakeeping).

3) Trim angles less than 4 deg are desirable to reduce wave impact accelerations (as discussed in a subsequent section on seakeeping).

Early planing hull designs were guided almost entirely by the requirement for high speed in calm water so that low hull deadrise angles were used and loaded to attain optimum trim angle. Modern planing hull design is so dominated by seakeeping considerations that reasonable compromises in smooth water performance are not only tolerated but sought. Consequently, good planing hull forms will have moderate deadrise at the stern (~ 15 deg) increasing to high deadrise (approximately 50 deg) at the bow. To achieve the desirable low trim angles in rough water, provision is made to shift ballast or fuel to bow tanks. If this design feature is not possible, then transom flaps are installed to lower the trim as necessary. These trim control techniques allow for setting the optimum trim angles in both calm and rough water. Design procedures for selecting the size and deflection of trim flaps are given by Savitsky and Brown.[5]

Effects of Length Beam Ratio

One of the top three form parameters affecting planing hull performance in both calm and rough water is the length/beam ratio. Typical curves of trim and resistance vs speed as a function of length/beam ratios are given in Fig. 12 for five models of Series 62.[2] It is seen that, as speed increases, the craft trim and resistance increase to "hump" values and then decrease as the speed is further increased. There is a significant decrease in hump trim and resistance with increasing length/beam ratio such that, for length/beam ratios in excess of approximately 6, the humps are barely noticeable.

It is interesting to observe that, at volume Froude numbers (F_∇) between 2.5 and 3.5, the drag is essentially constant and independent of length/beam ratio. Consequently, increases in installed power will result in much higher speeds in this speed range than in other speed ranges. *At volume Froude numbers greater than 3.5-4.0, the drag will moderately increase as the length/beam ratio increases.*

Simply stated, when given a fixed displacement, the designer should attempt to configure the planing bottom to be as long and as narrow as possible—consistent with the requirements of internal arrangements. Fortunately (as will be shown) a high length/beam ratio hull is also very desirable for good performance in a seaway.

A review of the proportions of past planing hull designs indicates that the predominance of constructed boats had length/beam ratios between 3 and 5, with large numbers of commercial and recreational craft being in the range between

3 and 4. It is these craft which experience pronounced hump trim and high resistance characteristics—a performance pattern which even nautically oriented observers so typically associate with planing boats. In recent years, the design trend has been to length/beam ratios in excess of 5.0, even at the expense of compromising the internal arrangements. This results in a substantial reduction or even elimination of the "hump" problem, as well as a substantial reduction in drag in the preplaning speed range.

Effect of Slenderness Ratio

Further demonstration of the advantage of high length/beam ratio in reducing preplaning drag is shown in Fig. 13 where L/D vs F_∇ is plotted for a range of slenderness ratios, $L/\nabla^{1/3}$. In previous studies of planing hull resistance,[5] this nondimensional parameter was shown to have a substantial influence upon resistance. The curves, which are for a displacement of 100,000 lb., represent the state-of-the-art for efficient planing hulls and do not represent any one hull over the entire speed range. At $F_\nabla \leq 2.0$, corresponding to the cruise speed range for most naval craft, the longer hulls have substantially less resistance than the shorter ones. There is only a small effect of slenderness ratio at $FN \approx 3.0$ and a moderate increase in resistance with increasing slenderness ratio for $FN \geq 3.0$.

An acceptable planing hull design must demonstrate low drag characteristics from speeds corresponding to displacement mode up to the high speed planing regime. Further, the hull must also demonstrate good performance in a seaway. The design which best satisfies these requirements has a high length/beam ratio and a corresponding high slenderness ratio.

Rough Water Performance

Perhaps the greatest demand imposed upon today's designers of planing hulls is to develop hull forms with operational capability in a seaway.

Traditionally, planing hulls have been characterized as small boats with no rough water capability. It should be recognized, however, that such hulls were designed almost entirely for high speed in calm water, culminating in a hull form and loading combination which resulted in unacceptable seakeeping qualities in even moderate sea states.

Recent research in planing hull seakeeping technology, as reported by Fridsma[6] and summarized by Savitsky and Brown[5] have quantified the relations between hull form, loading, speed/length ratio, sea state, and the expected added resistance, motions, and—most importantly—wave impact accelerations. In fact, the designer now has the tools to optimize the planing hull for specified operational requirements in smooth and rough water. An example of such an optimization was given by Savitsky, Roper, and Benen.[7]

A brief summary of the most important seakeeping technology and its effects upon planing hull design is given below.

Wave Impact Accelerations

As shown in Ref. 5, the average impact acceleration at the CG of a planing hull operating in irregular head seas having a Pierson-Moskowitz spectrum can be simply represented by the following empirical equation:

$$\bar{n}_{CG} = 0.0104\left[\frac{H_{1/3}}{b} + 0.084\right]\frac{\tau}{4}\left[\frac{5}{3} - \frac{\beta}{30}\right]\left[\frac{V_K}{\sqrt{L}}\right]^2\frac{L/b}{C_\Delta} \quad (5)$$

where:

$\bar{n}_{CG}$ = average center of gravity acceleration, g
$H_{1/3}$ = significant wave height, ft
τ = equilibrium trim angle, deg
β = deadrise angle, deg

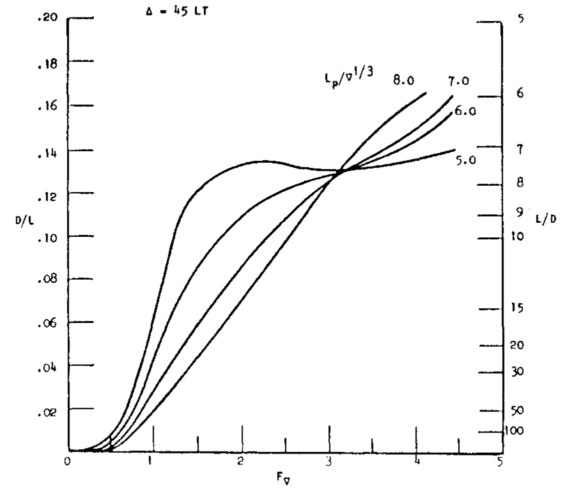

Fig. 13 Drag/lift ratio for efficient planing hulls as function of volume Froude number and slenderness ratio.

V_K = speed, knots
L = load-waterline length, ft
b = beam, ft
C_Δ = beam loading coefficient, Δ/wb^3
w = weight density of water, lb/ft^3

The average $1/N^{th}$ highest acceleration, $\bar{n}_{1/N}$ is related to the average acceleration $\bar{n}$:

$$\bar{n}_{1/N} = \bar{n}(1 + \log_e N) \quad (6)$$

Therefore, the 1/3 highest and the 1/10 highest are, respectively, 2.1 and 3.3 times the average acceleration.

The limits of applicability of these empirical equations are identified in Ref. 5.

Several interesting and useful design conclusions become obvious from an examination of the impact acceleration equation. All other conditions being equal:

1) The impact accelerations are linearly dependent upon equilibrium trim angle. Hence, the accelerations are easily reduced by a reduction in trim angle through the use of ballast transfer or trim flaps.

2) The impact accelerations are inversely proportional to the deadrise angle—large increases in deadrise result in large decreases in impact acceleration.

3) The impact accelerations vary inversely with beam loading coefficient $C_\Delta = \Delta/wb^3$ or as the cube of the beam. Thus, even a 10% decrease in the beam is expected to reduce the accelerations by nearly 30%. A recent planing hull design incorporated a double chine hull as shown in Fig. 14. The upper chine provided the beam necessary for roll stability at low speed and the lower chine, which caused flow separation during the impact process, provided the narrower beam desirable for reduction of wave impact loads. Full-scale test results for the hull form were presented by Blount and Hankley.[8]

4) Although it appears from the impact equation that accelerations increase with increasing L/b, the ultimate effect is to reduce the accelerations. Increasing L/b for a given hull displacement leads to a reduction in beam which, in turn, increases C_Δ by the cube of increasing L/b, thus resulting in a significant reduction in impact loads.

5) Accelerations are proportional to the significant wave height in irregular seas and increase as the speed squared.

Figure 15 presents a graphical representation of the trim and deadrise effects upon the 1/10 highest impact accelerations expected to be experienced by a 200 ft planing hull running at 50 knots in a 10 ft significant wave height. If a reduction in impact acceleration were the only operational

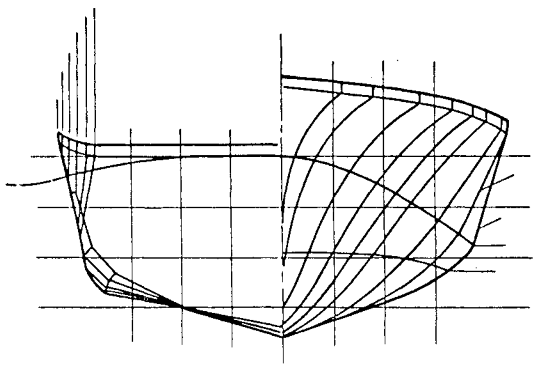

Fig. 14 Body plans for modern double chine planing hull.

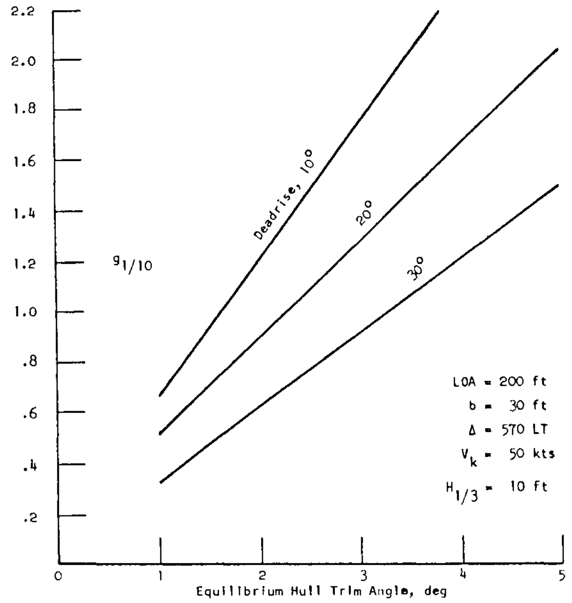

Fig. 15 c.g. impact acceleration in head seas.

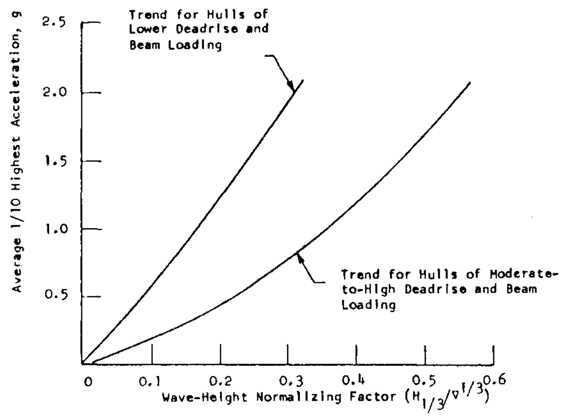

Fig. 16 Typical c.g. accelerations for planing hulls $F_\nabla \simeq 3.0$.

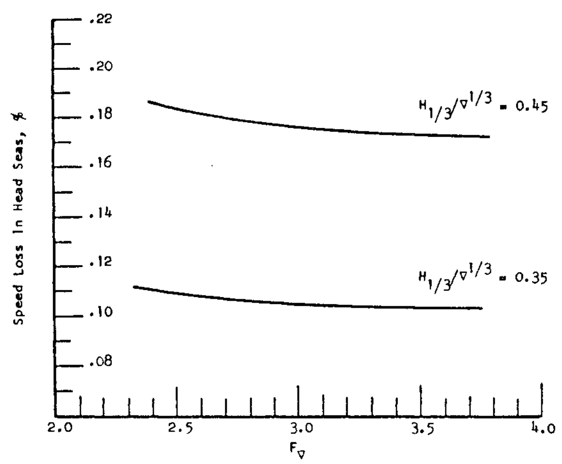

Fig. 17 Speed loss in seaway for typical high length/beam ratio planing hull (at constant power).

consideration, a planing hull would be designed with high deadrise, a longitudinal weight distribution such that the craft would run at a very low trim angle, and a narrow beam to obtain a high beam loading. Unfortunately, while this combination of design and operating parameters would indeed yield small impact accelerations, it will also develop large hydrodynamic resistance and have reduced internal volume. An acceptable design must establish the best compromise between resistance, impact acceleration, and total useful volume. The planing hull technology for developing a design philosophy of effective trade-off studies is at hand. An example of one such philosophy is given in Ref. 7 where, for the specified performance requirements, a double chine hull form appeared to be the optimum configuration.

The actual average 1/10 highest acceleration levels as a function of nondimensional wave height obtained in full-scale trials of planing hulls operating at $F_\nabla \approx 3$ is shown in Fig. 16. The upper curve is representative of hulls with lower deadrise and beam loading, typical of planing craft designed a decade ago. The lower curve shows the trend for modern, more useful planing hulls designed with moderate-to-high deadrise and beam loading. It is seen that recent hull deisigns experience less than one-half the acceleration levels measured on earlier planing forms. For the 20 deg deadrise hull specified on Fig. 14, $H_{1/3}/\nabla^{1/3} = 0.37$ and, from Fig. 16, it is expected that at 50 knots the 1/10 highest c.g. acceleration will be approximately $1g$—a rather modest load for a 50 knot speed capability. It is expected that future seakeeping research should result in additional reductions in g loadings while still maintaining an acceptable hull form.

Speed Loss in Seaway

In addition to demonstrating reduced impact accelerations, it is also essential that the speed loss in waves be acceptably small. The results of recent model tests of a modern planing hull, such as shown in Fig. 14, have indicated only modest resistance increases in irregular seas. These data have been used to predict the speed loss in waves at constant engine horsepower and the results are shown in Fig. 17. It is seen that, for $H_{1/3}/\nabla^{1/3} = 0.35$ (corresponds to a 10 ft wave for the 200 ft planing hull in Fig. 15), the speed loss is approximately 10% over the entire planing speed range. For a 13 ft significant wave height ($H_{1/3}/\nabla^{1/3} = 0.45$), the speed loss is approximately 17%. Although Fig. 17 indicates a small reduction in speed loss with increasing F_∇, there are combinations of hull loading and form which result in moderate increases in speed loss with increasing F_∇. Mostly, however, for high length/beam ratio planing hulls with moderate deadrise, the speed loss in a seaway is primarily dependent upon significant wave height and, to a much smaller extent, upon planing speed.

Relative to the effect of geometric form, it has been found from model tests that the speed loss in waves increases with decreasing deadrise angle and/or decreasing trim angle, particularly if substantial bow immersion is associated with low trim.

Pitch Motions in Seaway

The pitch and heave motions in a seaway are usually largest in the displacement speed range when the wave encounter period is most likely to be equal to the natural period in heave and/or pitch. At planing speeds, the motions are essentially constant with speed, being approximately one-half those in the displacement speed range. For high length/beam ratio

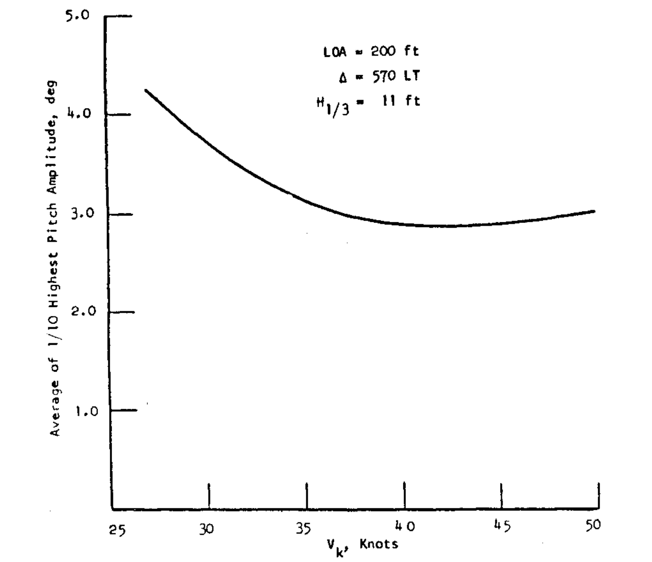

Fig. 18 Pitch motions of high length/beam ratio planing hull in head seas.

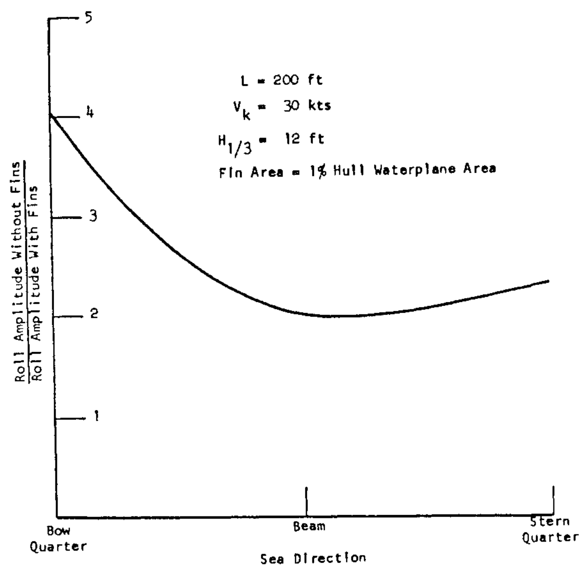

Fig. 19 Effect of activated roll fin stabilization.

hull forms, the pitch motions are expected to be tolerably small. Figure 18 shows the expected 1/10 highest pitch amplitude vs speed for a 200 ft planing hull operating in seas with an 11 ft significant wave height. These plots are based on the results of recent model and full-scale tests[8] scaled to a 200 ft planing craft. It is seen that, for speeds in excess of 35 knots, the 1/10 highest pitch amplitude is only approximately ± 3 deg.

High speed planing motions are attenuated with increase in hull deadrise and/or decreases in hull trim. As discussed earlier in this paper, such a combination of deadrise and trim will have excessive resistance.

Roll Motions in Seaway

Recently, attention has been paid to reducing the rolling motions of a planing craft in the preplaning range in order to provide a more stable platform for military systems and to improve habitability. The problem has been to increase the hydrodynamic roll damping which is inherently small even for hard chine planing hulls. Active roll fin stabilized systems have been used with good success at speeds in excess of 10 knots when roll stabilization was necessary. The effectiveness

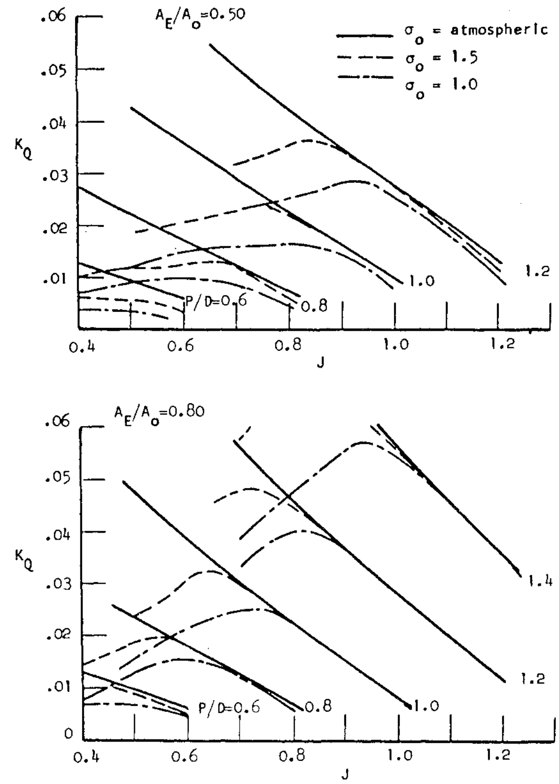

Fig. 20 Cavitation characteristics of Gawn-Burrill propeller.

of active roll fins, whose area was approximately 1% of the hull waterplane area is demonstrated in Fig. 19. These results are based upon recent full-scale trials for a ratio $H_{1/3}/\nabla^{1/3} = 0.50$. It is seen that, in beam seas, the roll motions were reduced by a factor of 2, in stern quartering seas by a factor of 2.2, and in bow quartering seas by a factor of 4. Such large attenuations in roll motions obviously improve the mission effectiveness and the crew's efficiency. The speed loss due to the added drag of the roll fins is easily accepted for the added stabilization and comfort it provides. At planing speeds, the fins can be retracted to eliminate this appendage drag.

Planing Hull Propulsors

The most commonly used and most economical means of propelling planing craft is the screw propeller mounted on an inclined shaft. Subcavitating propeller types are used for speeds below 35 knots, while transcavitating propellers have demonstrated excellent performance characteristics in the speed range of 35-55 knots. Although a number of other propulsor types (i.e., ventilated propellers, partially submerged propellers, waterjets, etc.), do indicate some promising performance features, their application to planing craft have been limited, and operational experiences are limited.

Subcavitating Propellers

Conventional subcavitating propellers of commercial manufacture are most commonly used on planing craft up to speeds of approximately 30 knots. Above 30 knots, these propellers have had serious erosion problems. Through custom design and close tolerance manufacturing, the useful speed of these propellers may be increased to approximately 35 knots.

Propeller characteristics are obtained from standard series propeller charts, such as the Gawn-Burrill series.[9] This series covers a range of blade/area ratios and pitch/diameter ratios for a series of cavitation numbers. The developed blade outlines are of elliptical shape and the sections are ogive (flat face, circular arc back, and sharp leading and trailing edges). While demonstrating good performance characteristics in the fully wetted condition, these sections sustain serious thrust breakdown and losses in efficiency when cavitation occurs. Figure 20 demonstrates the thrust breakdown for expanded blade/area ratios of 0.50 and 0.80 for cavitation numbers down to 1.0 (28 knots). It is seen that large propeller diameters and large blade/area ratios are required to reduce the propeller loading (K_T) and, hence, delay thrust breakdown at high speeds. This is an impractical solution for the designer since the large diameter will require large shaft angles, long support struts, low rpm, and large reduction gears—especially if a gas turbine powerplant is used.

The Gawn-Burrill series test data do not extend to design speeds beyond 38 knots. However, as indicated by DuCane,[10] it is believed that, even for the highest blade/area ratios, cavitation will no longer be avoidable and severe thrust and torque breakdown accompanied by efficiency losses will occur, spreading gradually to higher advance ratios (lighter propeller loadings) with reduced propeller cavitation numbers.

Fully Cavitated Propellers

Since planing hulls will operate at speeds in excess of 35 knots, propeller cavitation will be unavoidable. Fortunately, there is a developed propeller series available which is designed to accommodate cavitation without the serious performance deterioration associated with the ogive propeller. This is the Newton-Rader[11] propeller series which has cambered sections such as shown in Fig. 21. For typical advance ratios at design speed, the propeller develops a cavity which extends over more than 85% of the blade surface and beyond the trailing edge. They are frequently referred to as "fully cavitating" or "transcavitating," as distinct from the supercavitating propellers. Figure 22 compares the efficiencies of the Gawn-Burrill and Newton-Rader propellers at a cavitation number of 0.50. At the usual design values of advance coefficient $0.7 \lesssim J \lesssim 1.0$,† the shaded area represents the gain in efficiency associated with the Newton-Rader propeller. At $J = 0.80$, for example, there is a 22% gain in efficiency even though the blade/area ratio of the Newton-Rader propeller is only two-thirds that of the Gawn-Burrill propeller. Further, there is no significant compromise in efficiencies at low speeds when the propeller is fully wetted.

The use of a fully cavitating propeller permits an increase in loading, resulting in smaller propeller diameters and higher rpm. Although this usually causes a reduction in propeller efficiency, the Overall Propulsive Coefficient (OPC) may actually increase due to the reduction in appendage drag associated with reduced shaft angle and shorter strut lengths. In addition, there should be a weight reduction associated with smaller reduction gears, propellers, shaft, etc.

Most Newton-Rader propellers installed on fast patrol boats have been constructed of high tensile, nickel-aluminum bronze and have had blade/area ratios of approximately 0.7 and a working stress level of less than 15,000 psi. This compares with 80,000 psi ultimate tensile strength of the material. Blade erosion has been minimal even at extended service at speeds up to 55 knots. The American Bureau of Shipping has recently certified a Newton-Rader propeller designed for a high speed planing yacht. These propellers have been fabricated by foundries that normally produce small boat propellers in large quantities. The price has been very modest. Design procedures useful in selecting the optimum Newton-Rader propeller are given by Blount and Hankley.[8]

Habitability

Criteria for evaluating the ride quality for performance effectiveness of crew members in high speed marine vehicles

† $J = v/nD$, where v is boat speed, ft/s, n is propeller revolutions/s, and D is propeller diameter, ft.

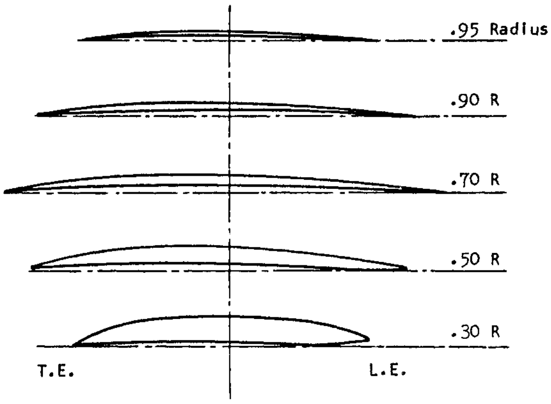

Fig. 21 Blade sections for Newton-Rader fully cavitated propeller.

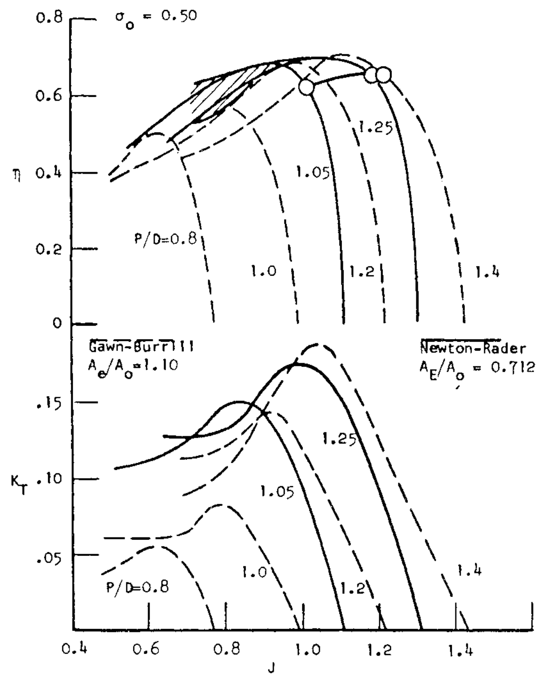

Fig. 22 Comparison of Gawn-Burrill and Newton-Rader propellers at low cavitation number.

continue to be reviewed and, as yet, there is no consensus of agreement on any one standard. For the purposes of this paper, reference is made to the International Standards Organization standard reported in MIL-STD-1472B and by Von Gierke. This criterion uses vertical accelerations and frequencies of occurrence as a measure of human tolerance. The criteria are shown in Fig. 23 where curves of a 1/3 octave RMS g are plotted against center frequency of 1/3 octave and for tolerance levels corresponding to 1, 2.5, 4, and 8 h durations. The ISO standard is for center frequencies greater than 1 Hz and corresponds to levels of fatigue decreased proficiency. Von Gierke's criteria are for center frequencies less than 1 Hz and correspond to 15% motion sickness incidence.

Superposed on this curve are measured acceleration levels for a high length/beam planing hull of moderate deadrise operating at speed/length ratios of approximately 2, 3, and 4 in an irregular wave having a significant wave height of approximately 30% of the hull beam. It is seen that, using these criteria, the accelerations encountered at high speed indicate a tolerable ride up to 4 h duration. This evaluation is substantiated by personnel aboard even though the visual appearance of flying spray to observers not on the boat seemed severe. This demonstrated significant improvement

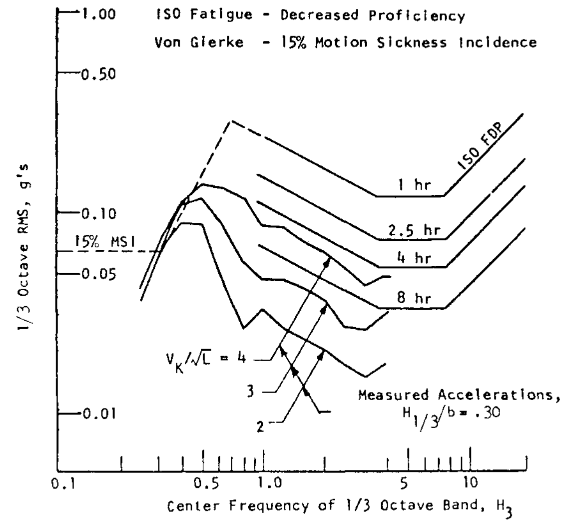

Fig. 23 Limits of human tolerance to vertical accelerations.

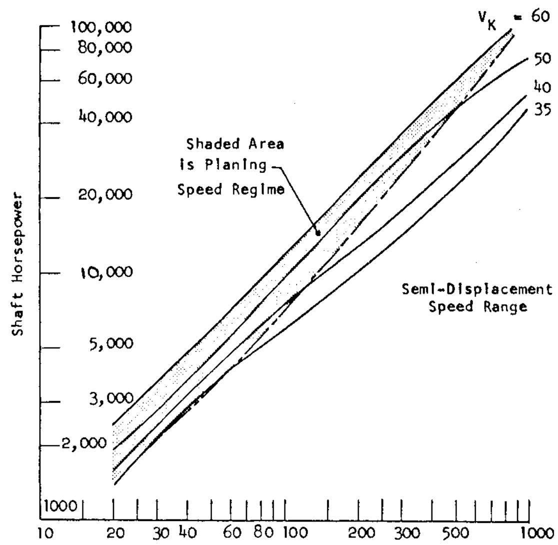

Fig. 25 Total shp vs displacement for various speeds.

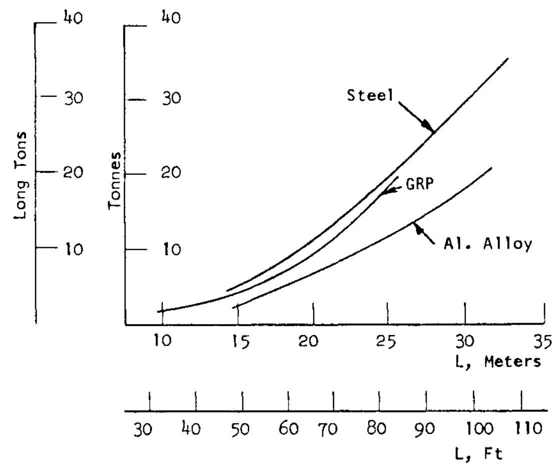

Fig. 24 Weight of hull and deck vs overall length.

over that traditionally associated with previous planing hull designs.

Materials

Three groups of materials have been found to be practical for planing hulls: marine aluminum (primarily 5000 series), Glass Reinforced Plastics (GRP), and mild and high tensile steel.

Weldable marine aluminum can produce a structural weight low enough to make high speed planing hulls feasible at low cost. Light weight is essential for planing hulls in order to reduce the relatively large power requirements (or to increase the useful load). This observation, of course, applies to all high performance vehicle types. In the United States, welded marine aluminum is the predominant material for planing hulls used as naval craft, crew boats for offshore service, and more recently fishing vessels. Further, the necessary aluminum welding skills appear to be increasingly more available in many boatyards throughout the country.

GRP are widely used to construct high speed planing craft up to approximately 50 ft in length, especially in the recreational boat industry. GRP are the least costly of materials when built in large quantities for small size hulls.

High tensile steel has a strength/weight ratio similar to typical marine aluminum alloys. Because of minimum gage constraints, however, it may not be attractive for small craft since it will result in heavier hulls relative to other materials. Recent studies by R. Allen of the David Taylor Naval Ship Research and Development Center indicate that high tensile steel may indeed be attractive for large, high speed planing hulls with displacements in excess of 500 tons. Because minimum gage, and not strength, is the governing consideration for small hulls, mild steel is used primarily for these hulls. Relative to cost considerations, it appears that, although heavier, steel hulls will be cheaper than aluminum. Planing hulls built of steel are more widely available abroad, particularly for speeds less than approximately 35 knots.

A comparison of structural weight versus overall length as a function of hull material was made by Sharples[12] and is presented in Fig. 24. It is obvious that the steel hulls are substantially heavier than aluminum hulls. Their justification is, of course, based upon lower cost and their fire resistant qualities.

Structure

Whatever material is used, the hull structure should be designed for the lightest possible weight consistent with the applied loads, since reductions in structural weight can be traded off for increased payload and/or fuel. Wave impact loads usually dominate the structural design. Methods for predicting loads on planing hulls are generally available and are sufficiently accurate for use in new designs. These methods are based upon expected loadings for the specific hull being designed. The procedure is to obtain CG accelerations either from model tests or estimates using Eq. (5) and then apply Heller-Jasper[13] or Allen and Jones,[14] converting these accelerations to design pressures for plating, stringers, and frames, and logitudinal bending moments. The predicted stresses using these loadings have been in good to excellent agreement with full-scale measurements.

Classification societies' procedures generally are not based upon loadings for the specific craft and usually result in heavier structures when compared with the tailored design procedure described above.

The magnitude of the impact acceleration is of course the key to design since it affects not only the structural loads but also the basis for evaluating the habitability of the design. From an accumulation of experiences by R. Allen on full-scale planing hulls, it appears that the limiting value of crew tolerance is obtained when the average of the 1/10 highest acceleration exceeds 1.5 g at the manned stations. The maximum acelerations associated with these 1/10 highest levels will be approximately 3.0 g. In the past, crew habitability has had little effect on hull structural design and almost all hulls were capable of withstanding much higher g levels than the crew could tolerate. The lower deadrise and low beam loadings of the older designs resulted in much higher impact accelerations compared to the modern designs which have greater deadrise and higher beam loadings, run at lower trim angles, and have higher length/beam ratios. The lower g loadings used to design modern planing hulls are now very nearly consistent with crew tolerances, resulting in a more balanced and efficient hull design.

Powering

Figure 25 has been prepared to illustrate the approximate relationship between displacement, speed, and shaft horsepower for a hard chine hull in the speed range of 35-60 knots. For the purpose of this illustration, a different design was optimized for a particular speed to establish the shaft horsepower at that speed for a given displacement. The shaded area represents the planing regime where $F_\nabla > 3.0$. In this speed regime a lift/drag ratio of 7.0 has been assumed. In the preplaning speed range, $F_\nabla \leq 3.0$, the maximum values of lift/drag ratio as given on Fig. 12 have been used. An OPC equal to 0.50 has been assumed (OPC = ehp/shp). Any specific design should be evaluated either by the analytical procedures of Ref. 1 or by appropriate model tests.

Figure 25 can be used to delineate those speed-displacement combinations which require total shaft horsepower in excess of the capabilities of existing high speed marine diesels. For example, if 3000 shp is taken to be the maximum horsepower available from a single diesel, then the use of four diesels (total shp = 12,000) can be considered only for designs whose combination of displacement and speed is less than approximately 160 tons and 42 knots, respectively. For larger speed-load combinations, gas turbines would have to be used. These conclusions can be adjusted in accordance with the number and availability of diesel engines used and the expected horsepower of these engines.

Considering for a moment a nominal 1000 ton planing ship, it is estimated that nearly 120,000 shp would be required to propel the vessel at a planing speed of 60 knots. This is a prodigious amount of power which would require the use of three 40,000 hp gas turbines. The cost of these engines would probably dominate the price of the boat. If the speed were reduced to 50 knots, then two 40,000 or three 27,000 hp gas turbines would be sufficient. Hence, while the hydrodynamic and structural technologies exist to design a 1000 ton, 60 knot planing ship, the cost of propelling the craft at such speed may have to be justified.

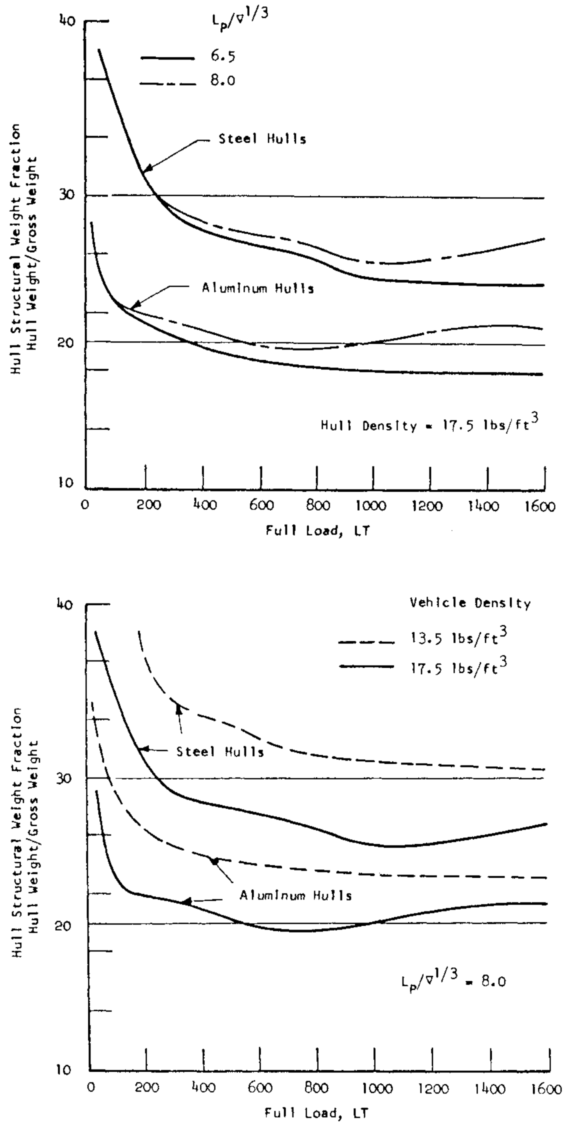

Fig. 26 Hull structural weight fraction as function of size, vehicle density, and slenderness ratio.

Effect of Planing Hull Proportions on Design

A recent study by Hadler, Hubble, Allen, and Blount[15] has produced a computerized Planing Hull Feasibility Model which can be used to examine ship proportions, size, and construction materials on various performance parameters. This model is an excellent tool for optimizing the hull form to meet specific mission requirements. Several of the conclusions of that study are reported herein since they compliment and reinforce some of the technology developments already discussed.

Structural Weight

Saving structural weight can provide the designer with trade-off leverage which will have significant effect upon increasing the useful load capacity of a ship; the authors of Ref. 15 calculated the hull structural weight fraction (structural weight/full load) vs. vessel displacement for several values of vehicle densities (full load/enclosed volume) and slenderness ratios. The results are summarized in their Fig. 7 which is reproduced here as Fig. 26.

It is seen that:

1) The use of aluminum instead of steel will result in a 30-40% weight reduction, with the larger reductions occurring with smaller size craft.

2) There will be a large reduction in structural weight fraction with increasing size to 500 tons and only a slight reduction with further increases in displacement.

3) The structural weight fraction will decrease significantly with increasing vehicle density. Weight fractions can be achieved as small as 20% for the denser and larger vehicles.

Useful Load Fraction

The trends for useful load fraction (as well as weight fractions for structure, machinery, and other fixed weights for four existing planing hulls are shown in Fig. 27. These trends are extrapolated to hull sizes up to 1000 tons. It is seen that the useful load fraction increases with increasing displacement. For a gross weight of 1000 tons it is predicted that the useful load will be nearly 50% of full load displacement. The term useful load includes military payload, ship's fuel and potable water, ship's complement and effects, and stores.

Hull Shape Details

Having specified the hull proportions, loading, average bottom deadrise, etc., the designer must provide such details as section shape (vee bottom, concave, inserted bell, inverted vee, etc.), the longitudinal shape (stepless or stepped hull, warped bottom, forefoot contour), and planform shape (chine shape, aspect ratio, transom shape). An excellent guide for selecting these details is given by Koelbel in Ref. 16.

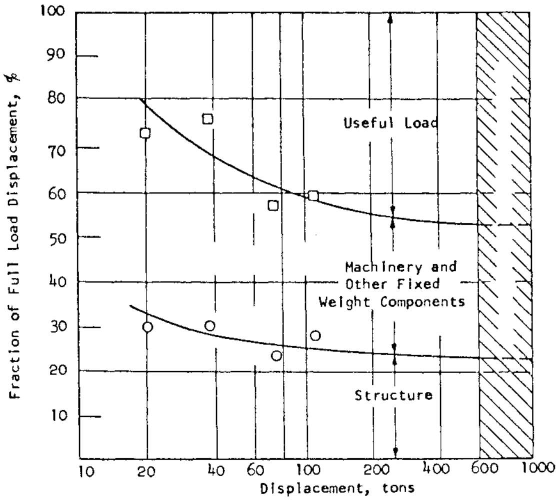

Fig. 27 Trends for various load fractions for four military planing hulls.

Costs

Quantitative projections of costs of planing hulls are impossible to discuss in these inflationary times. However, there are major considerations which should reduce the cost of planing craft relative to other members of the advanced vehicle family. These are:

1) The number of shipyards worldwide which are capable of building planing hulls is relatively large and continues to increase. This should result in more competitive bids and a favorable price to the customer. In contrast, there are only limited numbers of manufacturers capable of constructing ACV, SES, hydrofoil, etc.

2) The required structural technology is in hand and hull construction can follow normal shipyard practice. In fact, many of the traditional builders of displacement ships are easily expanding into the fast patrol boat market.

3) There are no special control or operational systems nor special support or maintenance procedures required in planing hulls.

4) With the elimination of hump speed characteristics through proper hull design, it appears that constant pitch, fully cavitated propellers can be used throughout the speed range. These propellers are easily fabricated in existing foundries.

5) The absence of hump will also enable economical slow speed operation on one relatively small engine and, with the ability to bring on line (in sequence) multiple engines, the result will be an operating profile where engines can be set to run at their best fuel rate.

The final decision on cost will be dependent upon careful analysis to establish trade-offs between capital costs, operating costs, maintenance costs, and value of the mission to be performed. A reduction in maximum speed, for instance, can result in a reduction in the number of engines required—a decision which will have a significant impact on cost, especially if high powered expensive gas turbines are being considered.

Potential Applications of Planing Hull Forms

The design of high speed craft has recently become one of the most active areas of naval architecture. The 200 mile fishing limit, recognized since Jan. 1, 1977, by virtually all nations, imposes national coastal state jurisdiction over nearly 10% of the world's ocean areas. These areas have become the Exclusive Economic Zone of the coastal states who wish to protect and exploit their potential off-shore wealth, which includes fishing as well as oil and other natural resources. It has been estimated that 90% of both living and natural resources in and under the area are within the 200 mile limit and that world demand for patrolling coastlines could require up to 600 high speed vessels. Such factors, when coupled with the mounting aspirations of emerging nations, have given rise to a worldwide interest in planing craft which are capable of acceptable operations in a seaway. A confirmation of this interest was demonstrated at the March 1978 RINA Symposium on Small Fast Warships and Secondary Vessels which attracted over 350 delegates from 32 nations, making it the largest and most successful symposia ever held by the RINA. A resumé of that symposium is given in Ref. 17.

From a totally military point of view, high-speed planing hulls armed with powerful surface-to-surface missiles, self-protected with surface-to-air missiles and close-in defensive weapons and countermeasures, and fitted with modern electronics systems will be entering service in the world's navies in ever-increasing numbers. This enthusiastic interest in the use of small, fast, patrol craft with devastatingly capable missile systems was indeed precipitated in 1967 by the sinking of the Israel destroyer Eilat by Styx missiles from fast patrol boats of the Egyptian Navy. Since that time, second generation antiship missile systems have appeared which operate from lightweight fixed launchers. In addition, gun armaments have experienced rapid developments with the introduction of effective and accurate fire control, increasing rates of fire, and high-precision munitions. Various caliber guns are available which have been effective even against aircraft and incoming missiles and which are compatible with planing hulls.

As discussed by Dorey,[18] sensors, computation and display facilities, and electronic warfare systems now form an integral part of the weapon fit of any warship, and their availability in forms compact and light enough to be installed in high-speed planing craft can make this class an effective warship.

Developments now in the technology pipeline using microminiturization for all forms of electronics equipments will have a dramatic effect on the "packing factor" of the black boxes which comprise the weapons systems of today. When the effects of such change are ultimately felt in all facets of the combat system design for small warships, the day of the multimission small warship will have arrived, overcoming a long standing objection to smaller ships which stems from a perception that they are unable to carry the weight and volume required. This perception does not, in our opinion, reflect the state-of-the-art now available, let alone reflect what will be possible in just 5 years.

Having demonstrated the feasibility of producing large planing craft of at least 1000 tons with a significant useful load fraction, it appears attractive to consider these craft as alternatives for various applications.

Hull mounted sonar transducer systems may be incompatible with high speed planing hull forms, especially when operating fast, in high sea states, due to surface noise and quenching effects. However, towed variable-depth sonars streamed aft would be feasible in a large planing hull of about 1000 tons. She would have the ability to carry a military payload of about 125 tons and enough fuel (320 tons) for open ocean deployments and transoceanic transists to forward operating areas. For example, with a Combined Diesel Or Gas turbine (CDOG) plant such a fuel load could result in a range at 10 knots of about 14,000 n. m. on twin diesels or a range of about 1200 n. m. at 44 knots on twin turbines. Assuming an installed power of about 100,000 shp distributed on three shafts, the estimated top speed would be about 55 knots in calm water or about 52 knots in sea state 3.

Such performance would result in a ride quality characterized by 1/3 highest c.g. accelerations of approximately 0.5 g at 50 knots in head sea waves of $H_{1/3} = 15$ ft.

These large planing hulls, equipped with roll fins for reduced speed cruising can now be designed with acceptable seakeeping characteristics as defined by the standards shown on Fig. 23. Specifically, a 570 ton planing craft operating at

50 knots in a 10 ft significant wave height would provide an acceptable ride quality for nearly 3 h duration; operations at slower speeds would of course produce the ride quality improvement commensurate with that speed. Such planing ships can now serve effectively in concert with the traditionally larger naval displacement ships. Together they can complement each other in providing an effective military operation in heavy seas.

As postulated by Ranken,[19] the patrol tasks facing maritime states falls into the following broad categories: surveillance, fishing patrol, protection of offshore resources, and customs and immigration.

The maximum desirable speed and size of craft are different for each operational category. It is estimated that a 25 knot patrol boat may be sufficient for fishing patrol while a 45-50 knot craft would be necessary for customs and immigration duties. It is not the objective of this paper to define these operational limits, but rather to indicate that once such an operational need is established, the technology now exists to design an optimum planing hull to meet these needs in sizes up to at least 1000 tons where installed power limitations and the cost of propulsion will influence the decision to stay "on the surface" or accept the cost and complexity of foils or air cushion support.

Hopefully, the myriad of apparent uses for small fast warships around the world will stimulate interest and awareness in our own U.S. Navy for a similar capability.

The onset of the worldwide political changes of recent months and years points again and again to the need for more affordable and more plentiful naval assets with the punch to counter aggravated assault on the high seas without risking the $100 + million to $1 billion combatants which are our first line of defense.

Just those political changes which have occurred in areas accessible from the eastern Mediterranean and the Indian Ocean alone are sufficient to increase the jeopardy of our highly valued ships in these waters. The authors believe that the state-of-the-art for planing ships, with multimission capabilities, adequate range and endurance, and sufficient bang for the buck, has reached the appropriate level for a U.S. Navy funded prototype development effort for a low cost, more "riskable" alternative to more of the same types of ships we now have in the fleet.

Conclusions

The recent developments in planing hull technology summarized in this paper demonstrate that high speed hulls can now be designed with the following characteristics:

1) Impressive seakeeping characteristics in comparison with the older designs when hull form proportions and loadings are properly selected for the sea state and speed of interest.

2) Structural weight fractions as low as 22% of full load displacement.

3) Useful load fractions approaching 50% of full load displacement.

4) Elimination of the traditional "hump" trim and resistance penalties.

5) Simplicity of design which permits ease of fabrication, the use of available propulsion systems, readily available engines, and proven propellers capable of speeds up to 60 knots.

6) Avoidance of special control systems.

7) Full load displacements of at least 1000 tons.

There is a growing worldwide capability to construct these craft in existing shipyards. Such experience will result in a more competitive industrial base compared with other types of advanced marine vehicles.

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