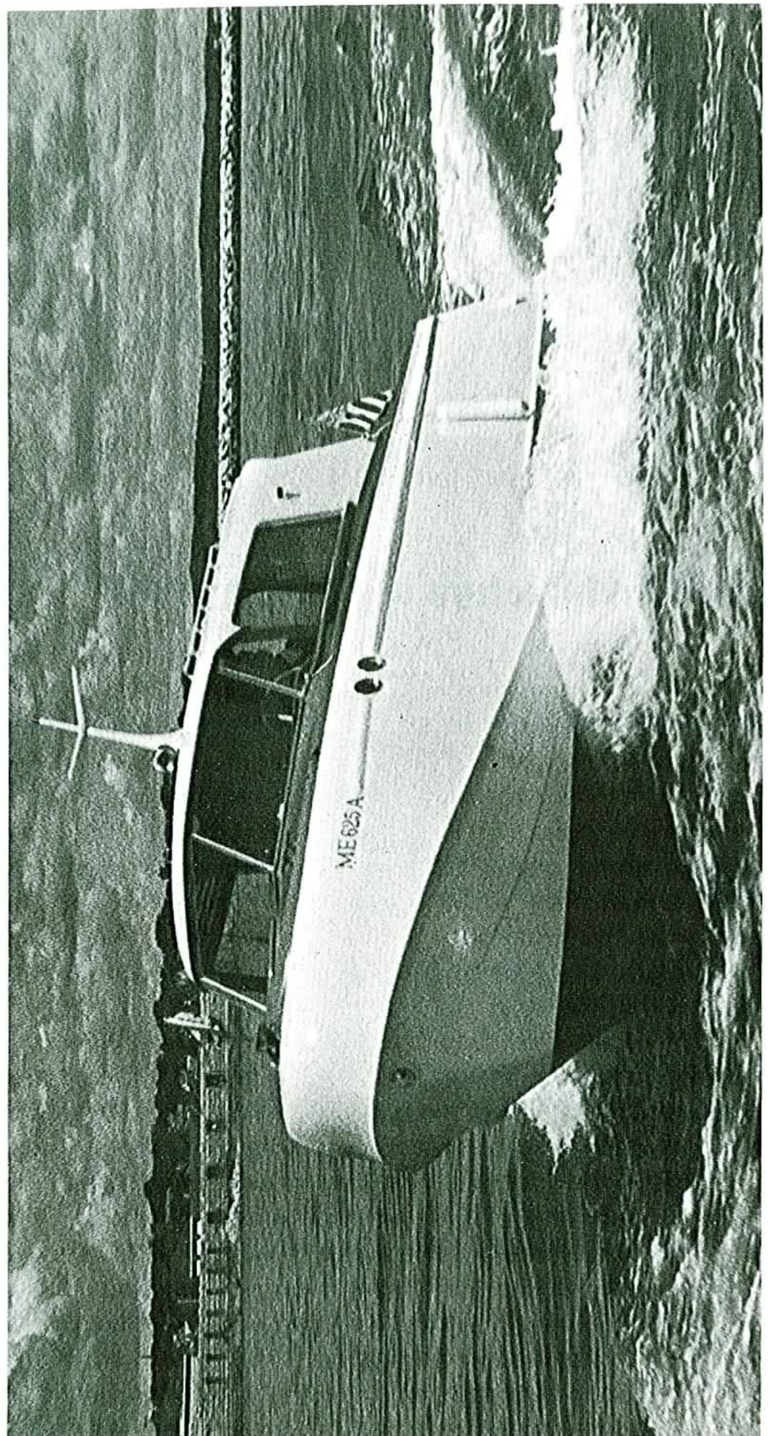

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A floating laboratory. This 30 foot cruiser has been lengthened, widened, deepened and repeatedly rebuilt and remodeled for practical trials and tests of shapes, materials and methods. She is presently equipped with a deep, box type keel containing the Interceptor engine, integral fuel, water, flotation and exhaust silencing chambers, full length capacitive ground embedded within the plastic skin and a full complement of electronic and navigating equipment.

NAVAL ARCHITECTURE OF PLANING HULLS

by

LINDSAY LORD
NAVAL ARCHITECT

THIRD EDITION

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To

William S. Newell, President of the Society
of Naval Architects and Marine Engineers,
whose lifetime of progressive thinking and constructive action has brought new lustre to the old Maine ship-building tradition.

PREFACE

I think all of us have realized from time to time that in hull design, much of the mathematics commonly used is to some extent just a refuge of the neophyte. Many a glorious vessel has been designed by rule of thumb and, on the other hand, more failures than ever are being designed by bright young graduates whose intricate mathematics alarms and confounds the alumnus of twenty years.

And yet, in this scientific day, "the attainment of sage experience to something like prophetic strain" does require that the rule of thumb have been brewed from mathematics. It would be inconceivable to attempt the design of a displacement hull without the constant application of standard formulas to check the designer's judgment. The more complicated requirements and greater expense of modern vessels make it imperative that there be a scientific basis for the design.

Yet the fact remains that in the relatively most complicated and most expensive field of all marine construction, no scientific basis of design has yet been generally applied. Naval architects have not been encouraged to study scientifically the performance of seagoing planing hulls. As a result, great numbers of these boats are, from a technical standpoint, below an acceptable standard as set by contemporary craft of the displacement type.

We have long realized that hydrostatic naval architecture is not applicable to the planing hull once it leaves its mooring. The fundamental hydraulic laws upon which standard naval architectural procedures are based simply do not apply to a hull skimming the surface. When so basic a relationship as the elemental speed-length ratio has to be discarded entirely and a new comparison found, something of the radical difference between the two types of naval architecture may be appreciated.

But during the days when planing craft were largely toys for racing, no exhaustive scientific study was economically feasible. Therefore, the design of these fast boats has been practiced largely as an art, beclouded with considerable mystery and prejudice. For the fresh-water racing machines, this guild system of design has been quite satisfactory. But the occasional attempts to create larger planing hulls, capable of operating at sea, have been until recently a series of technical failures. Just prior to World War II, the navies of the world were demanding small, fast "mosquito boats." Out of a queer hash of experience with outboard racers, battleships and gold braid, many torpedo boat designs were dreamed up and sold to the hopeful governments. Without exception they were failures. As with all other equipment, war has put the boats to uses and tests undreamed of in more peaceful times. Tactical demands are seldom based on expert knowledge of a boat's potential, or even past, performance; rather, the demands are more apt to reflect the urgent military needs of higher authority, regardless of the shortcomings or technicalities of design peculiar to some unit of equipment. Such necessary but nonetheless ruthless use of equipment furnishes the observant designer with a laboratory unavailable within even a lifetime of more peaceful years.

Ever since the days of the rumrunners, my pleasure and my work have been the design and construction of fast power craft in seagoing types. Through constant association with colleges and the teaching of naval architecture to more students, perhaps, than any other man, my tendency has always been to lift small boat design into the scientifically exact field usually reserved for big ships alone. Intimate association with the design of Army, Navy and Coast Guard small boats, and subsequent supervision of maintenance of the Pacific-based mosquito boats of all services, gave an opportunity for test and research which could happen only once in a lifetime. It is the scientific analysis of this cumulative experience and research which now forms at least a beginning for this new branch of naval architecture.

The constant angle plane, or monohedron, described in this work, was developed to meet the necessity for better mathematical correlation between a planing bottom and its designed performance. The resulting data, while still far short of the voluminous information built up over the years pertaining to displacement craft, does lay a foundation for intelligent approach to the problems of planing hull design and to the potential improvements to come.

LINDSAY LORD.

ACKNOWLEDGMENTS

To accomplish so extensive a task as the creation of a new naval architecture has been a task to which many minds have gladly contributed. I am particularly grateful to Captain Edwin D. Gibb, U.S.N., Commodore of Sea Forces, Hawaiian Sea Frontier, for his constant and invaluable assistance in every phase of the work. I wish also to express my thanks to Mr. H. A. Loveland, Master of the Pearl Harbor Boat Shop, and to Mr. Wilson Lee, Boatbuilder, for their assistance in model making and testing; to Mr. S. A. Weinberger of the Pearl Harbor Scientific Section and his able assistants Mr. Francis Aona and Mr. Edward K. N. Lum for their long hours in charting figures and data; and to Commander Robert H. Emerick and Lieutenant R. J. LeBer for their assistance as technical critics.

To Commander William J. Forrestel goes my gratitude for his recognition of the fundamental nature of this research and its importance to progress in our country's planing hull design.

L. L.

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Chapter I

THE TREND OF DESIGN

In spite of the fact that there has existed no comprehensive naval architecture applicable to the planing hull, a great mass of actual trial-and-error experience, extending back for thirty years, does exist. However, not all of it is of positive value because relatively few of the boats built have been of a genuine seagoing type. Most of the small pleasure boats and all of the racing craft have been bred to a high state of perfection for smooth-water speed alone. Ocean-going performance has left much to be desired.

World Wars I and II have each produced serious attempts at combining the antagonistic requirements of speed vs. sea-keeping ability. Best known and deservedly famous as the forerunner of them all was the Thornycroft Coastal Motor Boat. The British used these speedy little ships throughout World War I in every situation that arose. That they were much of the time unmanageable and all the time uncomfortable was of but little moment beside their occasional real usefulness. However, with the advent of peace and reduced naval budgets, experimental work was largely abandoned in favor of the always essential larger units of the fleet. Progress in planing hulls was for the most part confined to fresh-water speed alone.

Later came our Prohibition era, again with a demand for high-speed craft capable of lugging heavy loads at sea. With the normal rate of improvement to be expected from constant trial-and-error methods, high-speed hulls gradually began to appear which were capable of bringing in their cargoes in reasonably good condition even through heavy weather. But power consumption was still enormously out of proportion, largely because, as in war, the sky was the limit. Also, because no experimental methods short of testing the actual boat were available, design of the larger craft tended to become standardized along the lines of previous hulls which outwardly appeared to perform about as well as earlier experience indicated was customary.

The Sea Sled, or inverted vee bottom, is a good example of remarkable progress in design. While perfectly true that its performance in many ways is most gratifying, its radical appearance has been, perhaps, an upset to tradition too violent for popular acceptance. Large sizes of the Sea Sled hull also have presented structural difficulties which were solved only with steel truss framing, but not until after considerable damaging publicity had gained currency. However, the beautiful planing qualities and the undoubted weatherliness of these hulls render the idea worthy of further research, particularly along the lines of hull form refinements which may tend to reduce any excess of longitudinal stability. This is a characteristic of all planing hulls which has received no scientific study at all prior to the work outlined in this book. Given the benefit of such analytic study, there is no reason why present Sea Sled problems could not be resolved into hulls of both superlative performance and attractive appearance.

In 1937 a double-chine hull was introduced in England, which attempted to reduce wetted surface by a combined action of lifting out to a narrower plane plus the action of a transverse step in the narrow center plane. The reduction of wetted surface by means of double chines in which the outer chines lift clear of the surface is quite effective, as proved in Gold Cup racers, but only in calm water. At sea, wave action on double chines actually increases the wetted surface. Furthermore, in these particular boats, the transverse step in the narrow center plane was formed by recessing the after surface upward so that the inner chines completely blanked off and shrouded the step from outside ventilation, building up an enormous suction drag.

The German E-boats of World War II furnished a strange contrast to German scientific development in other lines. These boats were 106 feet long, 16 feet in beam, with 5½ feet draft. They were powered with three Mercedes-Benz supercharged diesels of 2500 horsepower each. Their top speed under favorable conditions was 42 knots, about 5 knots better than the usual British PTs. However, with a beam of 25 feet instead of their ridiculous 16-foot width, the E-boats should have made their speed with little over half their installed power. Even at the cruising speed of 31 knots, their 17 tons of fuel oil was sufficient only for a range of 600 miles. These boats are but one more example of the strange lack of design data pertinent to high-speed small craft. Assuming even the generous allowance of 75 horsepower per ton at 42 knots, they were overpowered by 2500 horsepower.

The Japanese PT boats were apparently designed in Italy. All of

them were between 59 and 61 feet overall and from 13 to 16 feet in beam. Those boats which were unchanged from the Italian design displaced only 20 tons, and with two 900-horsepower engines could sometimes touch 37 knots. Like their Italian prototypes, they were not good sea boats. They took ten boat lengths to make a turn, with speed falling off badly during the turn.

But whatever the quality of other Japanese weapons, their ship design was pitiful. Perhaps after seeing some examples of American PT boats with their shower baths and complete equipment in every department, they attempted to do their own redesigning. The result was a new class weighing 80 tons and powered with four engines of 950 horsepower each. However, they retained a 60-foot hull, and with only 16 feet of beam the weight was too great and 28 knots was their best speed.

These discouraging craft led the Japs to start building anti-PT boats: 60-foot steel hulls with heavy gun turrets, capable of only a sluggish 15 knots. But it is necessary to look elsewhere for good boat design since little of importance was ever reflected in the Jap fleet.

The Russians, in their dire necessity, produced some good examples. With the help of lend-lease Packard engines and plywood-making machinery, they produced some 70-foot hulls of great strength and lightness, longitudinally framed and of convex bottom sections. These boats were at their best in rough weather, when they frequently were capable of better than 40 knots. Because of a fortuitous combination of weights aft and good lateral plane forward, they could execute high-speed turns in less than four boat lengths. Their big flaw, in common with one otherwise good make of American PT boats, was their terrible wetness.

But all of these various characteristics, both good and bad, were largely the fruit of guesswork. In the absence of a comprehensive, scientific method for planing hull design, no exact criterion of performance, beyond existing experience and prejudice, could very well be set up. Within the limitations of the model tank, Professor K. M. Davidson alone has made resistance experiments of standard types. But since his work has necessarily been confined within the definite limits of small towed models, little has been demonstrated except that all tank results are incomplete until interpreted on a comparative basis and in the light of actual experience. Without such practical interpretation, the model tank has led to such dangerous performers as the obsolete Scott-Payne and old Vospers types which were designed with no knowledge of the relationship of the lateral plane, at speed, to weights and sea conditions. Many of these types were lost as a result of their fatal tendency toward broaching in a following sea.

Even though the Government has preferred to regard the design of its planing types as somewhat beneath the dignity of senior officers, the potentialities of fast boats loom so obvious, in spite of their faults, that developing a science of their performance has become a challenge. Both Dutch and Russian hulls have proved that John Hacker's convex forward sections can eliminate pounding. George Crouch has done much to attain higher speeds. But no customary type is now carrying at speed the loads which conservative hydraulic calculations show should be carried. The list of flaws is much longer, but behind them all there is the constant indication of a tremendous potential in usefulness as normal improvements are made.

The first Navy PTs were evolved from standard practice of the times. It will be noticed that their running lines and deadrise are typical of what was considered conservative. The forefoot was not deep and sharp in section; it was wide and buoyant to prevent plunging. Doubtless the hollow sections are conducive to outright speed in smooth water, and they do give excellent wave-flattening qualities. However, it is here that most pounding takes place and it is no secret that the early PTs invariably pounded themselves to pulp. All kinds of ingenious ideas were tried to make the hulls stronger. Boat after boat limped home with a broken back, and heavy reinforcements were patched on over decks, under decks, along chines and around engine foundations. But the more heavily the structure was stiffened up, like the immovable object, the more the sea demonstrated its irresistible force. With impact pressures of impossible magnitude, the PT was pitifully victimized by its own antagonism to the sea.

The Army has had equally bad luck. Its 72-foot crash boat features the fine entrance of old-time displacement hulls, with the result that not only does the hull plunge disastrously but even in calm water the whole boat is continuously inundated by its own bow wave. However, in the main, the fast boats of all services have been based upon the best rule-of-thumb designs available, and their excellent service in spite of performance is almost as much of a tribute to their designers as to the hardy youngsters who drove them. "Drove" must be the word since "sailed" or "handled" or even "piloted" is far too mild for the plunging, crashing rides these boats delivered.

Reputable designers, in seeking to avoid the harshness of the typical vee bottom at sea, have pointed out that a round bottom hull could be designed to go just about as fast, at least in the lower ranges of planing, and do it without pounding. There is indeed truth in the statement, but it is hardly the whole truth. The round bottom boat does ride more easily, but its dimensional ratios must be so altered that it becomes in effect simply a vee bottom with rounded chines. And there the two schools of thought become one.

Present research is delving into the phenomena of lift and suction under the plane, changes in moments of inertia at speed, wetted surface, planing angles and lateral plane effects under rough water conditions. From this scientific approach, successful adaptations and combinations are beginning to appear and high-speed hulls which can hold and even increase their speed and safety at sea are being evolved. The hulls are becoming wider and the forward chines higher. Safety is actually surpassing that of displacement types as lateral plane adjustments eliminate broaching tendencies and as the positive banked turn becomes understood. With the reduction of suction loads, planing hulls are achieving a maneuverability exciting even to young j.g.'s, and yet satisfying to men of science.

Model study in towing tanks has been disappointing for several reasons. Since there exists no tank with anywhere near the necessary length for a planing run with suitably sized models, little has been attempted. To test planing models, a run of several minutes is advisable in order to attain full equilibrium, particularly when rough water conditions are being studied. For this reason, the best results in model testing have been attained by using the ocean itself for a tank.

For any hull of importance it is advisable to have such an ocean model test to observe the performance. A model for towing may be as short as 4 feet overall, but self-propelled models of 12 or 15 feet will give more reliable data.

The models used in developing the curves for comparative performance between varying shapes of hulls, shown in this text, were of a standard series. A still-water plane area of 8.5 square feet was common to all models and test units, with variations made in beam-length ratios and in loading. The similarity of performance between such models and the full-sized boats is very close if all instrument readings and conditions are carefully interpreted.

The most obvious characteristic of the successful planing hull is its

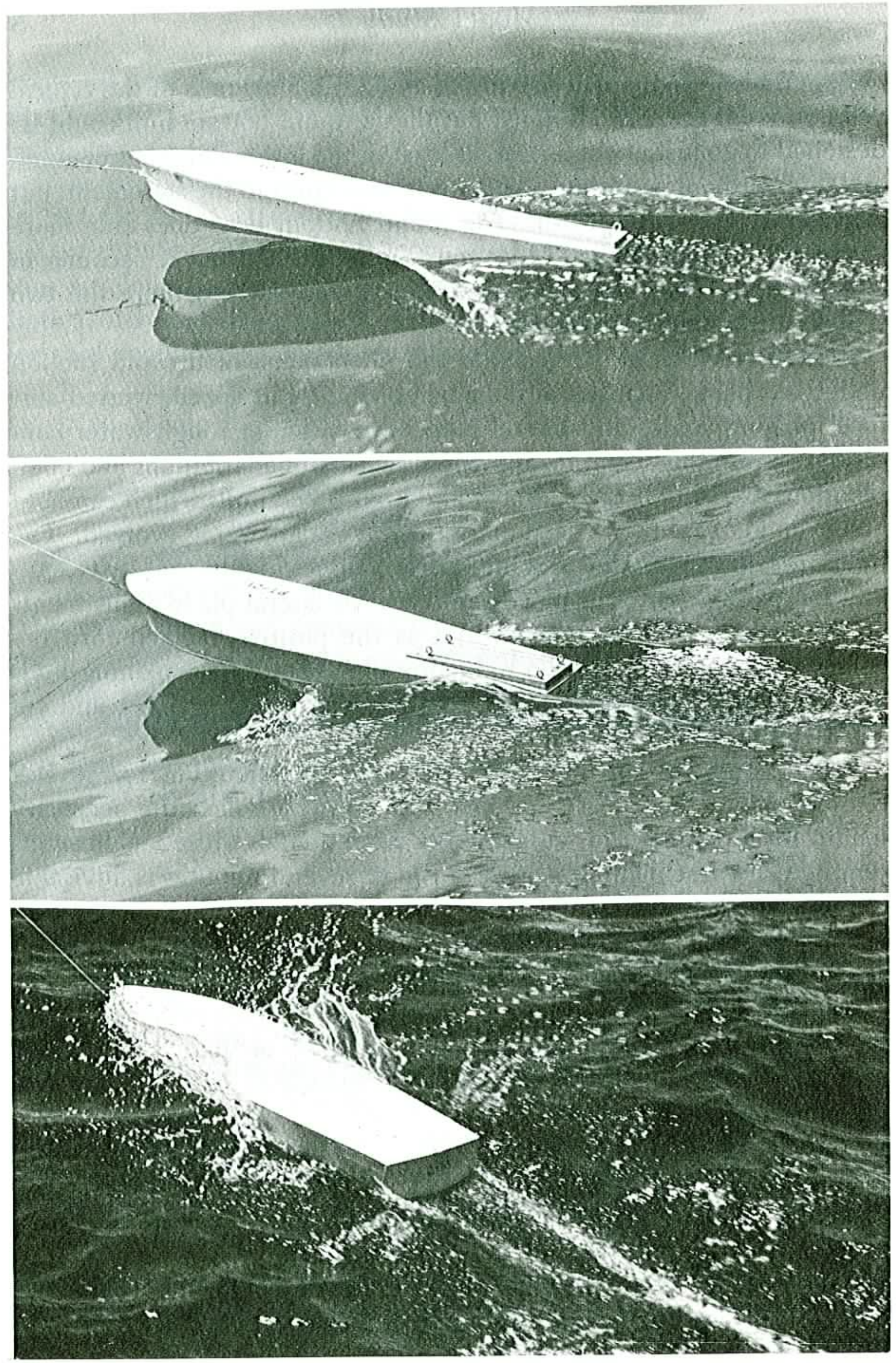


Figure 1

great beam. It is interesting to note in passing that after thousands of years, the planing hull is changing the old standard of proportions. Noah's ark was 300 cubits long and 50 cubits wide. In other words, it was 450 feet long and 75 feet in beam, which is just about the ratio that has been used ever since for ships of that general size.

Tomorrow, the planing hull of similar area will be around 370 feet long and 90 feet in beam. The shorter hull benefits from the reduced longitudinal moment of inertia together with the increased leading edge effect of the greater proportionate beam.

In Figures 1 and 2 are shown a series of model tests of two hulls identical in weight and plane area but differing materially in shape of bottom. The hull in Figure 1 is a typical vee bottom scaled from the lines of a 77-foot motor torpedo boat. In the top picture it is being towed at a speed corresponding to 44 knots on the full-sized boat. Distribution of weight is also identical with that of the full-sized boat, and it is apparent that the resulting trim is also the same. The center of gravity of the hull is directly over the forwardmost wetted surface and the amount of squat shown represents an added weight aft in the form of suction, equal to about one third of the total weight of hull.

The second picture shows the same hull being towed at a speed corresponding to 55 knots. The model would no longer tow straight, yawing wildly as shown in the photograph by the water piled up against the port side. The yawing was caused by suction under the afterbody building up to such magnitude that it is partially relieved by sidewise movement. In other words, if this hull were to be driven at 55 knots, it would take less horsepower to push it sidewise than straight ahead. Apparently, the suction aft created an excessive trim, while all lateral plane effectiveness was confined forward of midships.

The bottom photograph shows the model back at a corresponding speed of 44 knots, but in natural waves and swells equivalent to ocean swells about 100 feet long. The pounding broke all the electric instruments mounted inside which indicated trim, pressures and resistance. In full-sized boats, such pounding exceeds the strength of any manmade fabric.

In Figure 2 is shown the M-1, a bottom designed to minimize suction and retain lateral plane effectiveness aft. It is wider and shorter than the PT model. In the top picture, the towing speed corresponds to 44 knots. Its trim is in marked contrast to that shown at the top of Figure 1.

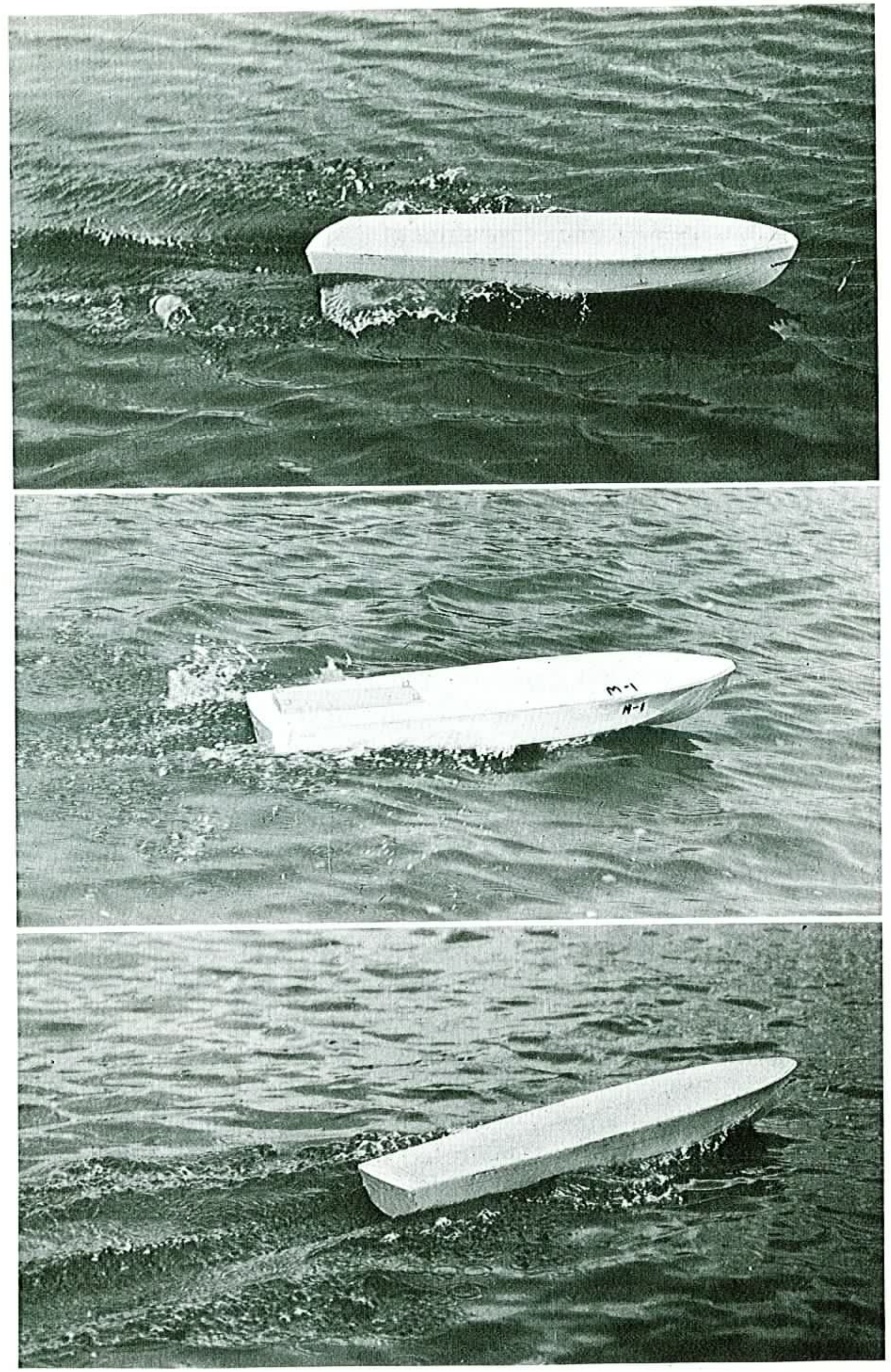


FIGURE 2

The middle picture shows the model being towed at a speed corresponding to 55 knots. The forefoot is where it belongs: just in contact with the surface. The freeboard aft shows little or no effect of downward pulling suction.

At the bottom, the clean wave pattern and wake of a sea-kindly hull are fully apparent. The interesting manner in which the M-1 lowered its overall resistance in rough water is shown on the chart in Figure 3.

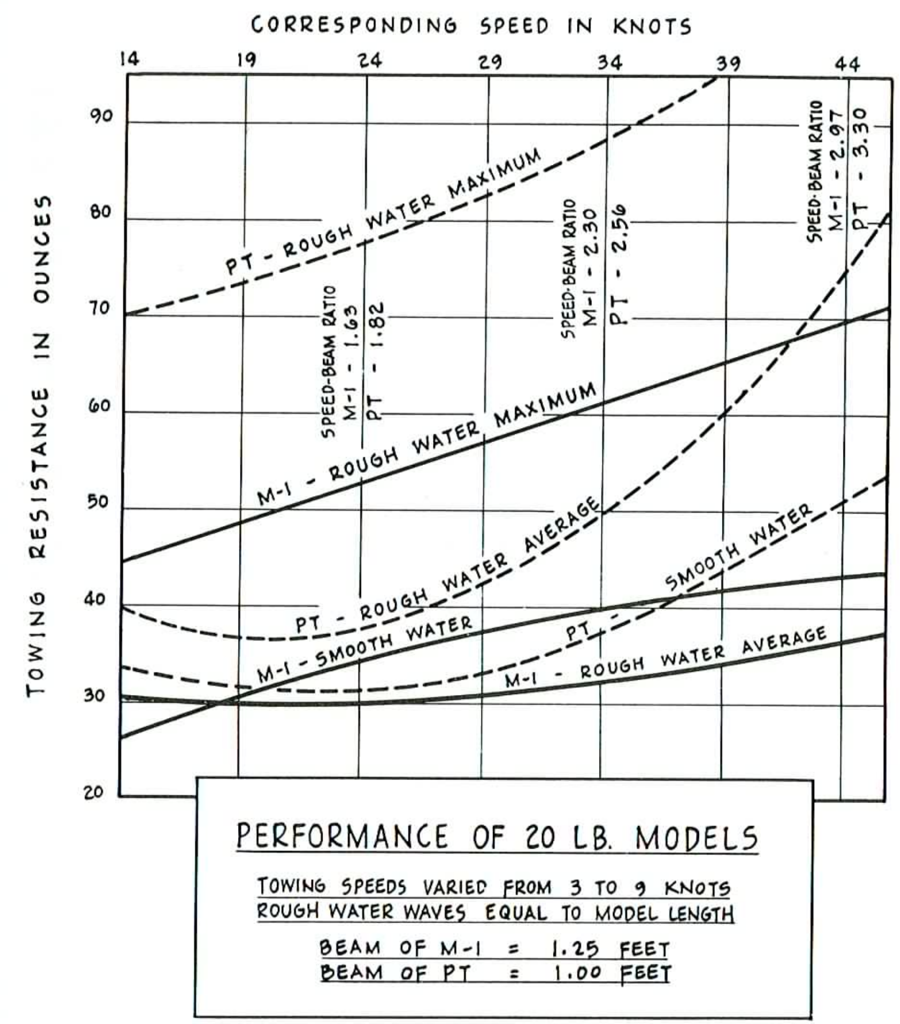


FIGURE 3

The curves labeled "Rough Water Maximum" indicate the maximum points to which the electric resistance scale moved when the hulls plunged into waves. All solid curves refer to the M-l and broken lines to the PT.

Transferring the basic design features under comparison in these models to full-sized boats, several preliminary conclusions are justified:

First, of two planing hulls similar in area, the wider one maintains its speed with appreciably less horsepower.

Second, of two similar planing hulls, the wider one is capable of planing with a greater load.

Third, fine lines forward, such as given by a typical round bottom design, are soft riding but are impossibly wet. On the other hand, planing hulls with a wide chine kept low forward are dry, but are impossibly hard riding.

Fourth, all present-day planing hulls of the seagoing type are carrying an unseen and heretofore unmeasured suction load of appreciable magnitude.

Fifth, the dynamic forces induced by planing are beyond the scope of static naval architecture, and for this reason the standard calculations for weight distribution, stability and moments are relevant only for preliminary design. However, all of the present fast boats include certain desired features, and the design problem is not only to combine as many good points as possible, but also to know in advance the exact effects of the combination when the finished boat is planing at sea.

Chapter 2
ASPECT RATIO

The lift and the drag, or the carrying power and the resistance, of a simple plane are combined functions of the effective planing angle, together with those characteristics inherent in the particular plane's beam-length proportion, or aspect ratio.

For bodies which are submerged, such as the hulls of displacement craft, proportions are determined by experience with lines of flow. According to Newton's Second Law of Motion, the resistance encountered is equal to the time rate at which the body changes the momentum of the fluid. That is, the rate at which the driving force does work is equal to the rate at which kinetic energy is imparted to the fluid.

However, in ship forms moving at relatively high speeds, momentum can be imparted to the fluid not only by the forward onrush of one bulk displacing another, but also by the resulting turbulence and suction according to the density of the medium. Such a resistance, demonstrated by dragging a flat plate crosswise to the stream, is expressed by Newton's equation:

$$R = KApv^2$$

where:

K = A constant for plates having similar shape
A = Area of plate
p = Density (lb./ft.^{3}/g)(lb. sec.^{2} ft.^{4})
v = Speed in feet per second

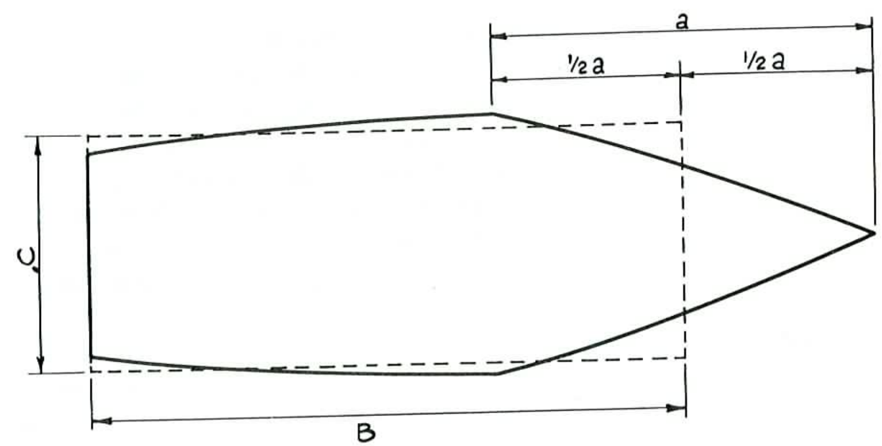
In the case of plates being moved in their own planes, the resistance rises less rapidly than the square of the speed, and the constant also has to be diminished as the physical size of the plate is increased. This situation is closely approximated by Froude's equation:

$$R = f'Apv^n$$

where symbols are the same as before, except that n is a little less than 2 and f' decreases as the length dimension of area increases.

This difference in behavior is fundamental in the differing performance between displacement hulls and planing hulls when both are being driven at high speeds. Obviously, the submerged body, moving with sufficient rapidity, increases the turbulence and the resulting suction drag, soon reaching a speed at which the viscosity of the liquid prevents further increase of speed regardless of practical increases in power.

With the plate moving in its own plane, this type of suction drag due to viscosity of the liquid is not a factor in the performance. Rather, the



ASPECT RATIO = $\frac{C}{B}$

WATER PLANE SHAPE AT REST

FIGURE 4

resistance, aside from skin friction, is largely due to the simple transfer of kinetic energy at the leading edge. Thus it becomes apparent that the leading edge of the plane at once accounts for a major portion of both drag and lift. But since lift rises as the square of the speed, and drag increases at less than the square of the speed, every proportionate increase in leading edge increment becomes successively more and more worth while.

In other words, while increasing speeds require the displacement hull to become progressively narrower, the planing hull moving at high speed requires the widest possible beam. To simplify still further, the

displacement hull can improve its speed only with added length; the planing hull requires added beam.

The effect of beam-length relationships on displacement hulls has been well standardized by generations of trial and error. A similar standard for optimum performance of the planing hull is needed. The first step toward such standardization of beam-length relationship must, of course, be an acceptable method of measurement which can be applied to all planing hulls with equal success. Such a measurement of the beam-length ratio is shown in Figure 4. Here is the waterplane shape of a typical planing hull. The dotted rectangle is presumed to have the same area as the waterplane. For hulls of ordinary shape, the width of the rectangle, *C*, is the median beam of the hull between midships and the transom. The length, *B*, is the distance from the transom to a point midway between the entrance of the stem and the entrance of the chine. Dividing the width *C* by the length *B* gives a percentage, or aspect ratio, suitable as a standard relationship in the comparison of planes.

The beam-length proportion is fundamental as a controlling factor in hull performance. This relationship, as indicated by the aspect ratio, is basic in the estimate of potential lifting power, resistance, stability and other qualities of sea-keeping ability. However, for convenience and for accurate standardization, the aspect ratio is always considered as of the waterplane shape while the hull is at rest.

In Figures 5 and 6 are shown a series of true aspect ratio floats, or bottoms, identical in area and weight but varying in aspect ratio from .2 as the narrowest one, to .6 as the widest of the series. The resistance characteristics of these bottoms, towed at planing speed, are shown in Figure 7.

Each bottom is a theoretically perfect plane of straight and parallel running lines with a slightly cambered cross section to assure a reasonable degree of directional stability when being towed. The necessity for parallelism in the longitudinal running lines is obvious in any plane which is expected to deliver similar and calculable lift across an entire cross section. Neither can there be any suggestion of a downward hook. The reasons for straight running lines will be developed and clarified in subsequent chapters.

The photograph at the top of Figure 5 shows the .2 aspect ratio bottom planing at an angle of .75 degree, its normal position for the entire range of planing speeds. Compared with the wider bottoms, it is in-

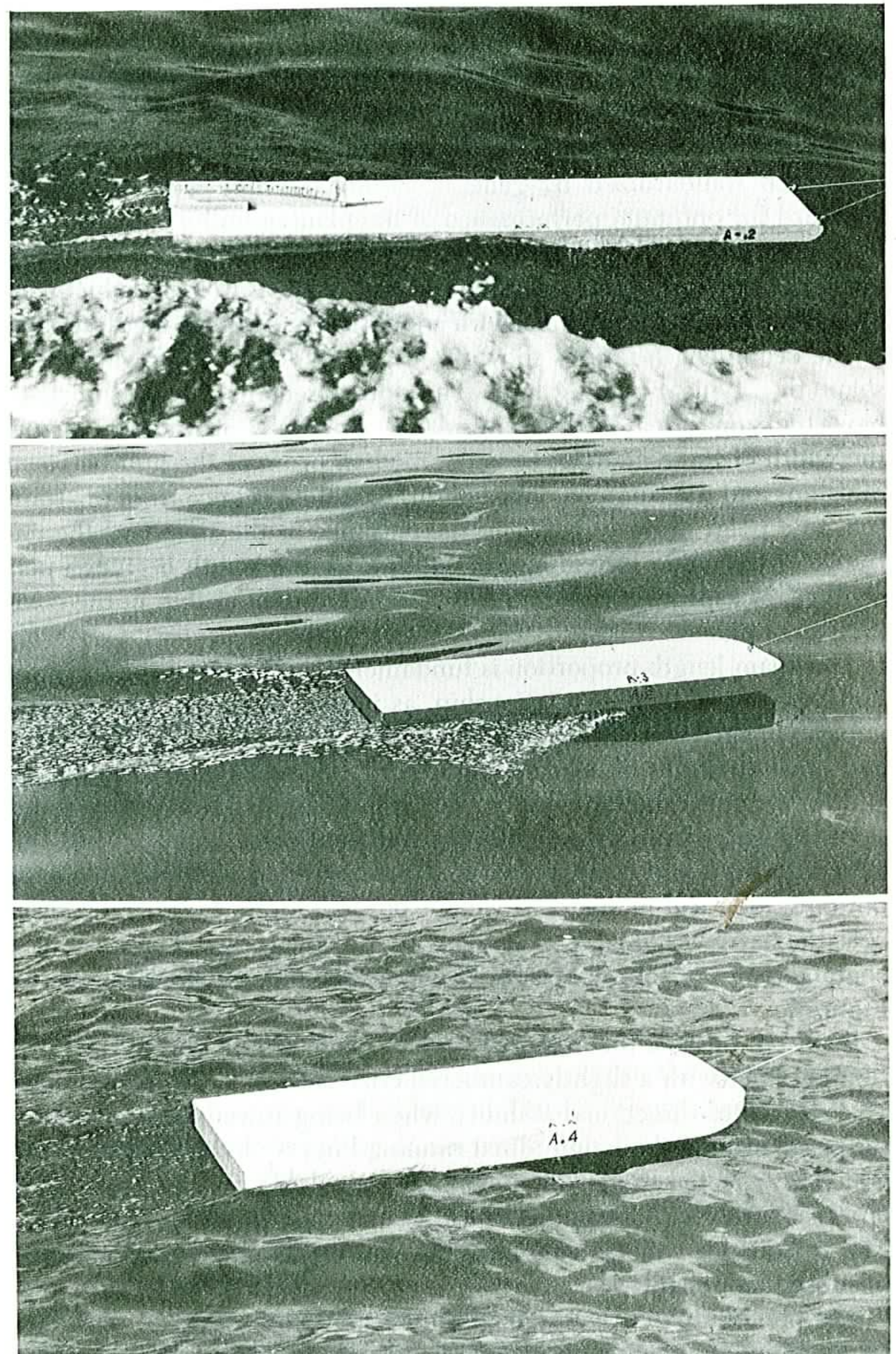


Figure 5

ferior on every count. The resistance is higher at all speeds, the lift is low, the stability poor, the longitudinal moment excessive and the wetted surface reduction almost zero. Even the wake is not "ironed out" as it should be.

All of these characteristics show proportionate improvement in successively wider bottoms up to aspect ratio .4, at which point wave-mak- 4 ing resistance in rough water ceases to decline with the higher aspect

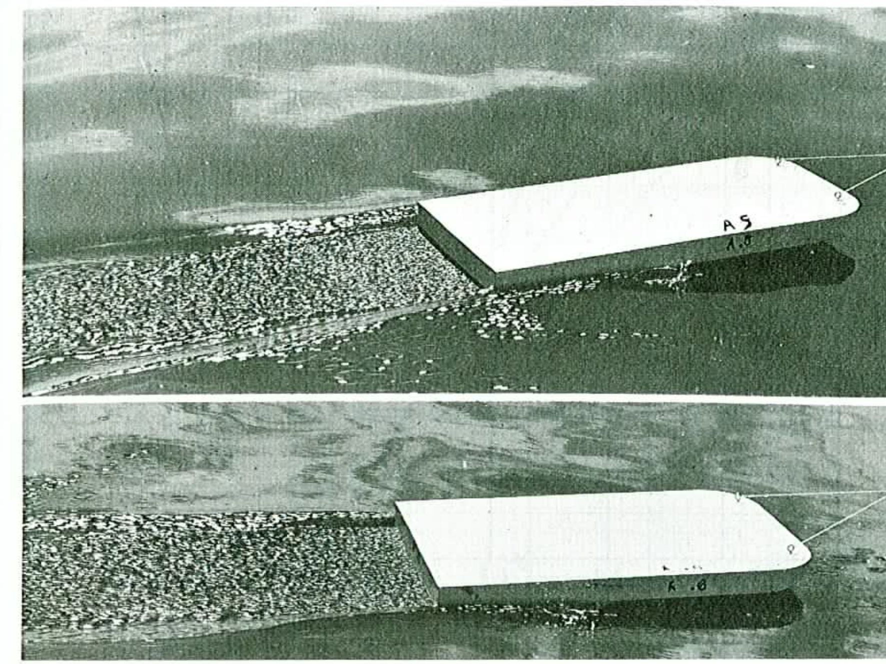


Figure 6

ratio and begins to show an increase. While this fact rules out aspect ratios .5 and .6 as suitable shapes for seagoing bottoms, it is significant to note their continued decrease in total resistance for smooth-water operation. The beautiful wake of the wider bottoms, shown in Figure 6, is also worthy of attention.

Other characteristics of varying aspect ratios are summarized in the following table of aspect ratio effects. In this table the resistance is the average for a standard total weight, or displacement, of 20 pounds. Planing angle is the normal one assumed with evenly distributed weight and a center of gravity amidships. Wetted surface reduction is shown as a percentage of still-water wetted surface. Maximum load is that total weight at which clean planing ceases in the range of normal seagoing speeds.

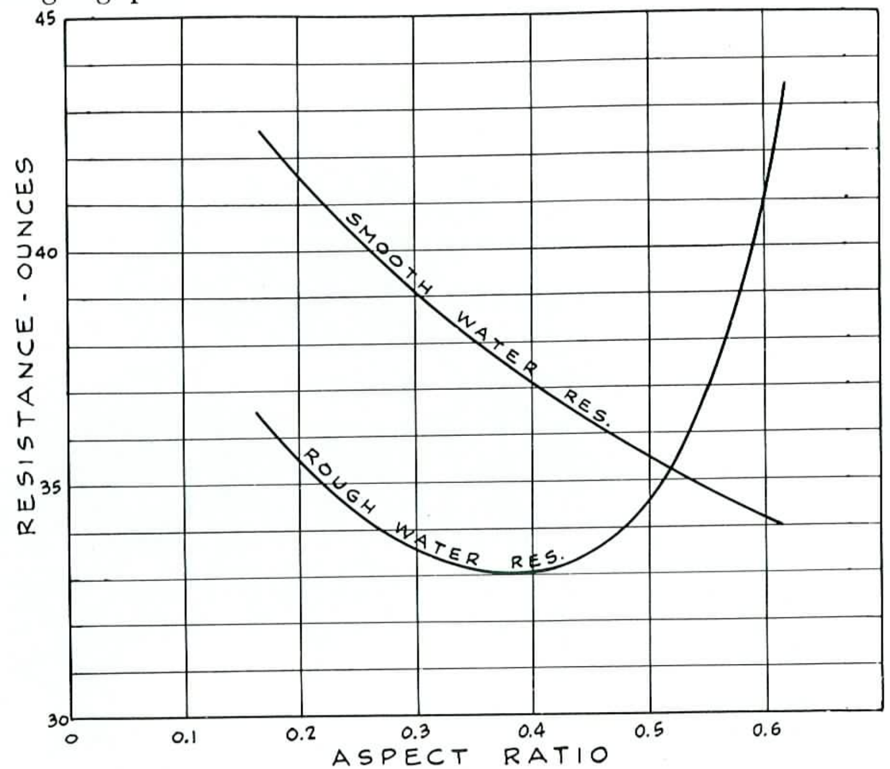


FIGURE 7

ASPECT RATIO EFFECTS

Aspect Ratio	*Resistance @ 20 lbs.*	*Planing Angle*	*BML*	*W.S. %*	*Max. Load*
.2	50 oz.	.75°	8′ 3″	90%	21 lbs.
.3	42 oz.	2.00°	5′ 2″	70%	27 lbs.
.4	37 oz.	3.00°	4′ 0″	62%	33 lbs.
.5	40 oz.	3.75°	3′ 8″	58%	38 lbs.
.6	42 oz.	4.25°	3′ 4″	56%	42 lbs.

Figure 8 shows in graphic form the load-carrying power of these varying aspect ratios and the effect of load upon resistance.

The conclusions to be drawn from these tests have been borne out repeatedly in actual boats and there can no longer be doubt as to the very real advantage of generous beam for best planing. Most high-

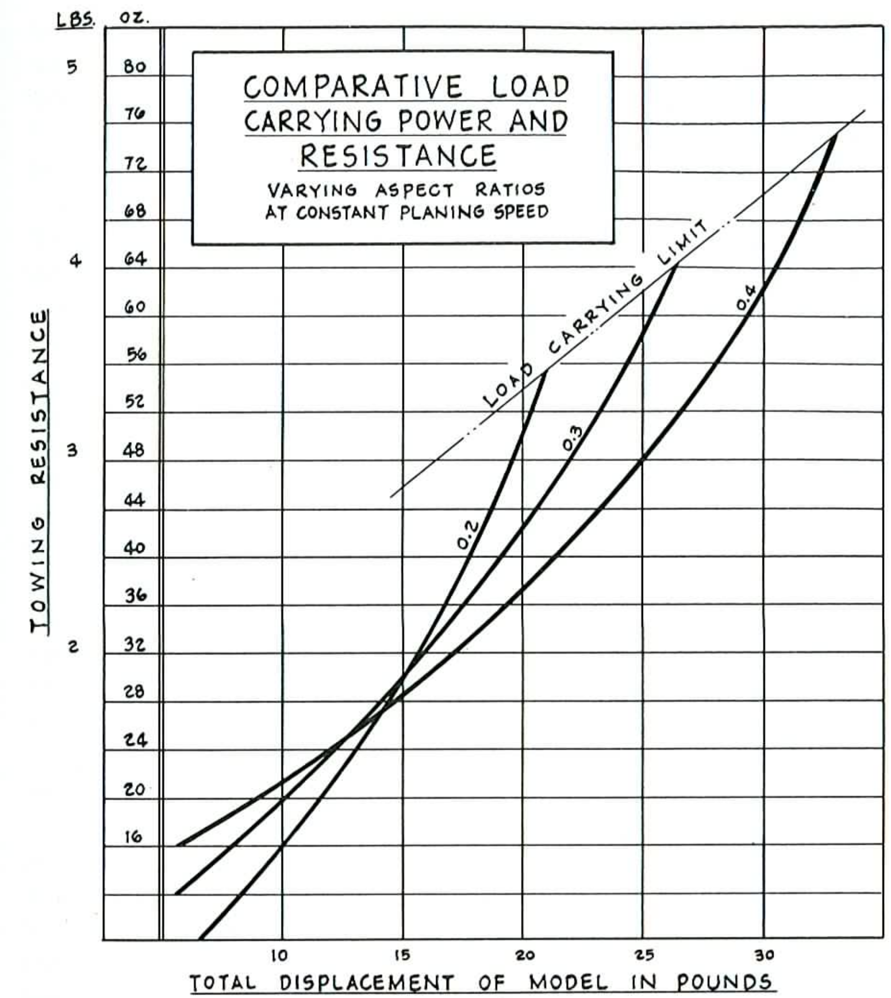


FIGURE 8

speed hulls are still around a beam aspect ratio of .2 in the larger sizes. Were such a hull sawed apart down its center line and a section inserted wide enough to double its aspect ratio, its resistance would drop significantly.

Narrow beam—that is, width similar to that of normal displacement form—is distinctly out of place on the planing type. The unfavorable resistance of narrow beam in both smooth and rough water may be attributed to several factors. First, as is well understood, it is the leading edge of an airfoil or hydrofoil which is most effective in lift.

Therefore the wider hull benefits from a greater area of leading edge lifting surface and, at the same time, correspondingly shortens its chord, or length of plane, thereby tending to reduce the extent of its area where negative pressures occur.

Second, with bottoms of constant section or reasonably straight running lines, the extremely flat planing angles normally assumed by the narrower floats, do not sufficiently reduce wetted surface. The lift component at these fractional degree planing angles is insufficient to cause bodily lift of the hull at any speeds feasible under open sea conditions. The natural tendency of the wider floats is to plane at more effective angles. The greater length of the narrow hull, together with its tendency toward excessive flatness of trim, creates so high a longitudinal metacenter that the hull becomes most obstinate in its tendency to plunge through rather than ride over waves. The effect is similar to a large moment of inertia caused by heavy weights forward. The shorter, wider hulls allow the most effective reduction of this longitudinal moment. A major factor in the ability of a planing hull to operate successfully at sea is that it shall be free to trim with the wave slopes. This requires the smallest possible longitudinal moment of inertia to foster lightness of the bow, a condition in which the long, narrow hull can in no wise compete with the wider, shorter hull.

Third, the effect of suction under the after end is inversely proportional to aspect ratio. In bottoms of non-uniform cross section, the suction effect can reach disastrous proportions. Non-uniform cross section —that is, a varying angle of deadrise—augments the pressure reduction, already excessive on narrow bottoms of even the constant section type.

It is obvious that resistance due to wave making must increase as the angle of planing increases. On the other hand, the increased component of net lift developed by the steeper plane angle results in a natural tendency to reduce the hull's wetted surface.

At higher speeds the frictional resistance of wetted surface becomes greater than for any other single resistance; therefore wave-making resistance may safely be increased by steeper plane angle up to that point where additional lift no longer reduces wetted surface. The point at which this planing angle occurs varies inversely as the speed, but in seagoing types should be from 1½ to 2 degrees. While steeper angles at seagoing speeds of around 40 knots for 70-footers may for some types actually show slightly less total resistance, the lessened lateral plane of

the hull makes it more subject to the action of cross winds. In general, no hull should be designed to plane with daylight showing under its forefoot. The ideal design would, of course, be that bottom which at 1½ degrees lifts out bodily to achieve the same reduction of wetted surface ordinarily obtained at 3 or 4 degrees.

The normal planing angle increases regularly from less than 1 degree for aspect ratio .2 to over 4 degrees for aspect ratio .6. Although the total resistance for the narrow bottom is greatest, its trim is the flattest. The total resistance decreases for aspect ratio .3 and .4 while plane angle increases. By subtracting the skin friction, which has decreased as the plane angle went up, it is apparent that the residual resistance climbed in direct proportion to the plane angle.

At aspect ratio .5 the normal angle is so steep that residual resistance has finally built up to a decidedly unfavorable proportion. By shifting the center of gravity forward, the plane angle may be reduced to 2 degrees or so with excellent effects on total resistance. However, the wide, short hull, loaded arbitrarily to keep the bow down, has achieved smooth-water efficiency at the expense of seagoing ability. This is not to say that rough-water performance requires a high-riding bow. Such an aspect is fundamentally wrong, since it allows each wave to strike with impact force. Rather, the forward position of the center of gravity creates an excessive moment which contributes toward plunging instead of following the wave contours.

An actual comparison of resistances among models of varying aspect ratio but of similar waterplane area and weight, as shown in Figure 8, indicates that under light load conditions corresponding to a very light vessel stripped down to fixed equipment only, the narrow hull operates with lower resistance. On a basis of actual weight per square foot of bottom, this is approximately the load condition under which many small racing craft operate. However, these are toys and not of seagoing size or type, and since the potential load-carrying power per square foot of bottom increases with total area and aspect ratio, larger seagoing hulls are inherently greater load carriers and must be so considered.

In the higher ranges of loading, the larger aspect ratio again becomes definitely superior. The three models tested were of identical bottom area, but the fact that the bottom of aspect ratio .4 lifted out and planed with nearly double the load carried by bottom .2 is most significant as indicating the trend of future design.

Chapter 3

SPEED RATIOS

Speed-Length Relationship

It was William Froude who propounded the law of comparison, a beautiful and fundamental relationship which states that ship speeds are proportional to the square roots of their linear dimensions. Without benefit of the criterion later developed by Osborne Reynolds, and knowing little of the mechanism of skin friction, Froude's speed-length ratio is scientifically sound and works out in practice within the range of ship speeds contemplated at the time.

Within this limited range of speeds, Froude's relationship showed that the tangential and the normal components of the total forces are essentially separable parts of a total resistance, and that when the size is changed, each part follows its own laws of change independently of the other. The expression is written as:

$$S/L \text{ ratio} = \frac{V}{\sqrt{L}}$$

Thus a 900-foot liner advancing at 30 knots has a speed-length ratio of 1, while a 36-foot lobster boat wallowing along at 6 knots also has a speed-length ratio of 1. Comparatively, then, both vessels are advancing at speeds requiring the same proportionate expenditure of power.

Down through the years, this fundamental relationship, like other well proved assumptions, has become so embedded in all naval architectural thinking that its limitation—in fact, its outright fallacy at high speeds—has gone unheeded. Obviously, the design of fast boats has been rule-of-thumb rather than scientific.

No correct analysis of today's planing hulls can proceed far without facing the fact that a speed-length relationship does not apply and that its arbitrary use for comparative purposes leads only to discouraging errors of design. From his personal observation, Froude pro-

pounded a relationship which applied reasonably well to ship forms of the displacement type, advancing at speeds which, had he known it, were below the Reynolds number of turbulence. This is a tremendous limitation and involves not only flow patterns but the shapes and streamlines of the bodies around which the flow takes place. At low speeds, the viscosity of water is such that reasonably clean laminar flow can take place around a body floating at the surface. But this laminar flow becomes generally turbulent as speed exceeds the viscous resurgent capacity of the water.

Thus, it has become apparent that it is possible to drive displacement hulls only up to speed-length ratios as high as perhaps 2. Beyond this general range, displacement hulls cannot go. And for very substantial reasons, no amount of power can drive the displacement hull faster. In fact, the speed-length relationship itself ceases to exist as a comparative value at any ratio above an approximate 2.0.

Limiting Factors

The observed fact is that at a speed-length ratio of approximately 2, a hull must have begun to show either one of two possible tendencies, according to its shape. Canoe-shaped sterns will have begun to settle, and the application of increased power will only increase the draft, even to the point of outright sinkage. Sterns with slightly more bearing aft resist the squatting tendency a fraction longer. But the tendency to swamp exists in all hulls where the buttocks, or running lines, curve upward to the water surface.

The other possible tendency is for the hull to lift and skim the surface. Flat buttocks ending at a broad stern form the characteristic hull shape fostering the tendency to lift and plane. Therefore, in that critical range of speed-length ratios between, say, 1.6 and 2.0, the hull with straight running lines and wide bearing aft will resist sinkage. Instead, it will lift and the flow beneath the hull will continue its laminar pattern for some distance aft of the hull.

This persistence of the flow pattern beyond the last point of hull contact is the first indication of new conditions beyond the scope of the Froude relationship Instead of water from below rising immediately to the surface and tumbling against the transom clear to the boot topping, the new planing tendency has ironed out the wake for some considerable distance. The flow pattern existing beneath the hull has extended itself beyond the direct influence of the hull. When this phenomenon occurs, the shape of the transom is plainly discernible in the depressed water surface for some distance aft. The water has broken clean from the transom and the resulting flow pattern is formed as if by an imaginary extension of the hull itself.

Herein is the key to the flow pattern as it now exists. It is the same as if the hull were longer and did exist over the depressed wake surface visible aft. Were it possible to say just how much added length of hull would exactly fill the wake where the flow is still coming up to the surface, this "induced length" might be added to the tangible hull length and the total used as a factor for the speed-length relationship.

However, there are two insurmountable reasons against the use of even this corrected length factor. First, there is the practical difficulty of measuring the true extent of induced length. Unfortunately, the wash from the sides quickly rolls in and covers the perfectly formed depression. Further aft, the slipstream boils up to obliterate any last trace of the true flow pattern. It is therefore impossible to observe to just what distance aft exact induced length does extend.

The second reason against using a corrected length figure is that it is no longer comparative. That is, the speed-length ratio of all clean planing hulls is 2. In other words, a correct estimate of the true induced length would always be sufficient to return the speed-length relationship to 2.0. For example, from observation together with a Reynolds number check, the induced length of a 36-foot waterline Chris-Craft moving at 18 knots, was 44 feet, making a total of 80 feet of flow length. The speed-length ratio thus becomes 2.0. Upon increasing the speed to 22 knots, the induced length has become 84 feet, making the theoretical corrected length about 120 feet. Again the speed-length ratio is 2.0. Obviously, any value as a comparative relationship has been lost.

One further difficulty remains to be pointed out in the way of any practical use of speed-length ratios for planing hulls. That is another practical impossibility in measuring the length, not only as induced aft, but also as existing forward under a hull which may have lifted half of its length clear of the water. Clearly, any workable figure of length is not to be obtained for the planing hull.

The underlying principle upon which the Froude relationship is based, applies only to the hydraulics of floating bodies surrounded by lines of flow which are too slow, relatively, to have acquired any velocity head. But since the modern, fast hull is designed with a shape which fosters lift, its dynamics more nearly approach that of the plane.

Instead of a hydraulic limitation, the properly designed hull benefits from a lifting tendency theoretically proportionate to the square of the speed. This does not necessarily presume any reduction or "hump" in the power expended for attainment of the higher speed. Rather, the planing tendency indicates that the hull has been so shaped as to have an inherent ability to receive and absorb higher powers without succumbing to the hydraulic laws which would cause a sucking under of hulls of finer waterplane.

Conversely, the broad-sterned hull *must* receive and expend greater horsepower, not only above the speeds at which submerged bodies can operate, but also within the submerged body range. Thus, at low speed-length ratios, even the planing hull is actually operating as a displacement craft and is therefore subject to the laws of bodies submerged at the surface. As such, its lines of flow are, at best, poorly shaped for low resistance at low speeds. However, some inefficiency at low speeds is a necessary corollary of any hull operating outside the scope of its fundamental design. In other words, the hydraulics of bodies submerged at the surface is a field apart from the dynamics of planing.

General Effect of Gravity

Consideration of these phenomena occurring within the range of speed-length ratios between 1.6 and 2, leads to the inescapable conclusion that for boat speeds involving any tendency toward lift, the flow characteristics surrounding the plane, and not those of purely submerged bodies, must be employed. This necessity has been fully recognized in aeronautical practice, and such a speed ratio has been developed by the National Advisory Committee on Aeronautics, which will be discussed later.

When a body moves at the surface of separation between water and air, waves are created. The densities of water and air are so different that for all practical purposes the effect of the air on the waves is negligible, the density of sea water being 1.99 and that of average air at sea level being .0024 per cubic foot. Therefore, wave surfaces are apparently acted upon from above by uniform hydrostatic pressure. But since the fact is that waves actually are created, the elevated water particles must be acted upon by pressures less than ordinary hydrostatic pressures. Similarly, the particles which are depressed must be acted upon by greater hydrostatic pressures. The significant fact in regard to the effect of gravity on wave-making resistance, is that the magnitude of the wave slopes is proportional to the specific weight, pg, of the fluid, and not to the density, p, alone.

The motion of submerged bodies does not set up forces on fluid particles which depend for their magnitudes on the attraction of gravity. Consequently, this attraction on the conditions of geometric similarity of flow patterns, and the acceleration due to gravity, on which it depends, does not appear among the variables in the dimensional analysis of submerged bodies.

The motion of surface bodies, on the other hand, creates waves which introduce gravity restoring forces. For planing hulls, therefore, flow patterns are not directly comparable unless the gravity restoring forces are proportional to the other forces present.

Hence, a range of speeds is quickly reached where performance can no longer be considered in the one case as a function of tangible length, nor in the other case as free from the effects of gravity.

Significance of Reynolds Numbers

The Reynolds number is:

$$\frac{vLp}{\mu}$$

Where:

v = Feet per second
L = Length in feet
p = Density
μ = Viscosity

This number is dimensionless in any consistent system of units. It defines the ratio of Inertia Force to Friction Force and sets up a criterion for the geometric similarity of flow patterns. When actual similarity does exist, the Reynolds numbers are also alike. Thus it is entirely logical to assume that differing Reynolds numbers indicate dissimilar flow patterns.

The magnitude of the shearing stress within a fluid depends upon the magnitude of the absolute velocity difference between adjacent particles. Thus in any discussion of modern high speeds, the wide variation in absolute speeds may be taken as positive evidence of wide variation, not only in proportionate resistance, but also in absolute resistance, regardless of geometric similarity.

In power calculations, the disproportionate increase in so-called skin friction at high speeds is brought into proper focus by separating the

calculations of friction. But a satisfactory speed relationship coefficient should be a general indication of total absolute resistance as far as may be consistent with its original proportional nature.

Dimensional analysis is essentially a means of capitalizing partial data when other details may be too obscure to permit of exact analysis. Dimensional analysis may be applied successfully to the problem if only those variables which govern the result are known. To apply it to the flow problem under consideration, it is necessary to know only that the total force R depends upon p, L, v and, when shearing stresses are considered, μ. With a correct choice of basic variables, the dimensional solution upon which correct experiments can be based points the way toward determining a general empirical solution.

There is a fundamental difference between the action of submerged bodies and of surface bodies. The resistance coefficients of similar submerged bodies will be alike whenever the Reynolds numbers are alike. But the resistance coefficients of similar surface bodies are not alike unless both the Reynolds numbers and the Froude numbers are alike. The Reynolds numbers of two submerged bodies can be made alike by moving the bodies at speeds which are inversely proportional to their relative sizes. Or the same sort of agreement may be obtained by increasing the density of the medium surrounding the smaller body. This helps to avoid high model speeds and, incidentally, is the purpose of variable density wind tunnels.

On the other hand, there is no way of making both the Reynolds number and the Froude number of a surface body equal to those of a full-size hull except by testing the model in a fluid having a very much lower coefficient of kinematic viscosity. There is no fluid known which would permit more than a slight difference in hull sizes and still allow accurate comparison. As long as the smaller hull planes on water, the conflicting demands of the Reynolds number that it move faster, and of the Froude number that it move more slowly, than the larger hull, to achieve similitude, make it impossible to ensure geometrically similar flow patterns.

With planing hulls, then, there is no theoretically sound procedure by which the total resistances of one hull can be directly compared to the total resistances of another hull radically different in size.

A major characteristic of any planing surface is the higher proportionate lift of the leading edge, as has been discussed in the preceding chapter. It is, however, not entirely the proportionate transverse extent of this athwartship area, but also its absolute extent, which influences the flow pattern and hence the lift. The absolute dimension does, as a matter of fact, largely govern the variables of a workable dimensional ratio applicable to planing hulls. It follows that a relationship based upon beam, among sizes not widely divergent, is not only correct theory but also has the practical advantage of providing a dimension which remains relatively constant under all conditions of speed. The controlling characteristic, then, of the modern, fast boat is its freedom from the flow patterns of submerged bodies operating within the non-turbulent speed flow ranges. The fast hull in skimming the surface develops a resistance in proportion to the kinematic viscosity. Acceleration due to gravity must therefore be included.

Standard Speed-Beam Ratio

There are several formulas which may be applied for fairly satisfactory comparison among modern, fast boats. The following formula shows the speed-beam relationship as promulgated by the National Advisory Committee on Aeronautics:

$$S/B = \frac{\text{Ft. per sec.}}{\sqrt{g \times b}}$$

Feet per second may conveniently be set down as knots times the constant 1.689, or miles per hour times 1.46, which converts speed into feet per second. Acceleration due to gravity is usually considered as 32.2 and beam is stated in feet. As an example, the speed-beam ratio of a hull with 10-foot waterline beam, planing at 30 knots, is:

$$S/B = \frac{30 \times 1.689}{\sqrt{32.2 \times 10}}$$
$$2.71 \text{ *}$$

Because the speed-beam relationship in this form has already been given some currency and is generally recognized, its use will be continued throughout this work as the standard ratio. There are, however, other relationships which can be similarly useful for comparisons among individual groups of hulls. Since none of them is suitable for comparison between widely differing hull sizes, one formula is perhaps as good as another. However, only in the standard ratio is gravity visibly introduced. While for strictly comparative purposes, its constant omission would be no different than its constant inclusion the very fact of its presence is a reassurance of sound derivation.

* All calculations have been worked out by slide rule, as is customary.

Direct Speed-Beam Ratio

A simple but not entirely accurate comparison ratio can be obtained merely from dividing speed by beam. Mathematically, this direct ratio suffers from slight distortion in its proportionate indication of relative effort required to drive hulls. Around standard speed-beam ratios of 3, the direct ratio may be considered as fairly reasonable, but above this point the direct speed-beam ratios rise out of proportion to comparative hull effort, and below this point the ratios drop away more rapidly than actual hull effort.

Use of the direct speed-beam formula is suitable only for non-mathematical comparisons such as those which are not to be used in connection with power or model calculations.

Basic Speed-Beam Ratio

A ratio derived from speed in knots over the square root of beam is mathematically correct and may be used for exact comparison throughout the range. While it is somewhat simpler to find than the standard speed-beam ratio, the larger numbers involved in the final ratio are out of line with the natural concept of relative speeds as generally understood on the basis of displacement craft speed-length ratios.

The basic speed-beam ratio is approximately the standard speed-beam ratio multiplied by the 2.83 root of acceleration due to gravity, or about 3.38.

The following table shows the comparative values of standard, direct and basic speed-beam ratios:

Type	*Speed Loaded*	*W.L. Beam*	*Speed-Beam Ratio* *Standard*	*Direct*	*Basic*
33′ Target Boat	44	7	4.95	6.25	16.6
42′ Rescue Boat	27	10	2.54	2.70	8.5
55′ Cruiser	40	11	3.60	3.62	12.1
65′ Crash Boat	35	12	3.01	2.91	10.1
70′ Rumrunner	37	19	2.53	1.95	8.5
72′ Rescue Boat	38	13	3.14	2.92	10.5
77′ Torpedo Boat	34	14	2.72	2.43	12.5
85′ Patrol Boat	36	22	2.29	1.64	7.7

Unless specifically indicated otherwise, the standard speed-beam ratio is understood. Use of the direct ratio, since it is inaccurate, should be discouraged. The basic ratio, as familiarity with the design of planing craft becomes more general, might very possibly come into common use and may even supplant the standard ratio.

Comparison of Speeds

The whole purpose of a speed ratio is to allow comparison among various hulls. However, there are limitations to any such comparison as hulls become widely dissimilar in size. Small differences in size such as between 40- and 50-foot hulls do not seriously affect the accuracy of comparative speed ratios, but the differences introduced as between models and full-sized hulls become highly misleading until weighted with further test data on the increasing lift per square foot of which wider planes are capable. Thus a small hull loaded to a draft presumably equivalent to a similar large hull, fails to lift out and plane as soon as the larger hull.

In other words, speed ratios for planing hulls are directly comparative only for somewhat similar sizes, a situation differing materially from the speed ratios applying to displacement hulls. A further consideration in the comparison of speeds is the fact that lift, or the planing tendency, varies with the absolute speed so that a large hull, even at a given speed-beam ratio, develops higher lift because of its higher absolute speed.

At identical speed-beam ratios the larger hull will therefore benefit not only from greater beam which gives more lift per square foot of bottom, but also from higher absolute speed which again adds to the lift per square foot of bottom. To what limits this dual advantage of size may be carried is as yet entirely unknown, but within the limits of present construction, the curve of lift continues to rise faster than the increase of beam. It would therefore seem reasonable that the greatest advantages of planing hulls are still to be realized.

There is a need for a new coefficient which will combine into one final value the whole interrelationship of relative lifts, size and shape of plane and absolute speed. Such a coefficient will be offered and discussed in the following chapter.

Planing Speeds

How fast a normal hull must be driven to "get over the hump" or to attain planing speed, while largely a function of aspect ratio, is also directly related to bottom loading. For the curves shown in Figure 9, identical loading per square foot of bottom has been arbitrarily assumed as applying to all sizes. To a considerable degree, this necessary assumption is responsible for the extremely low planing speed.

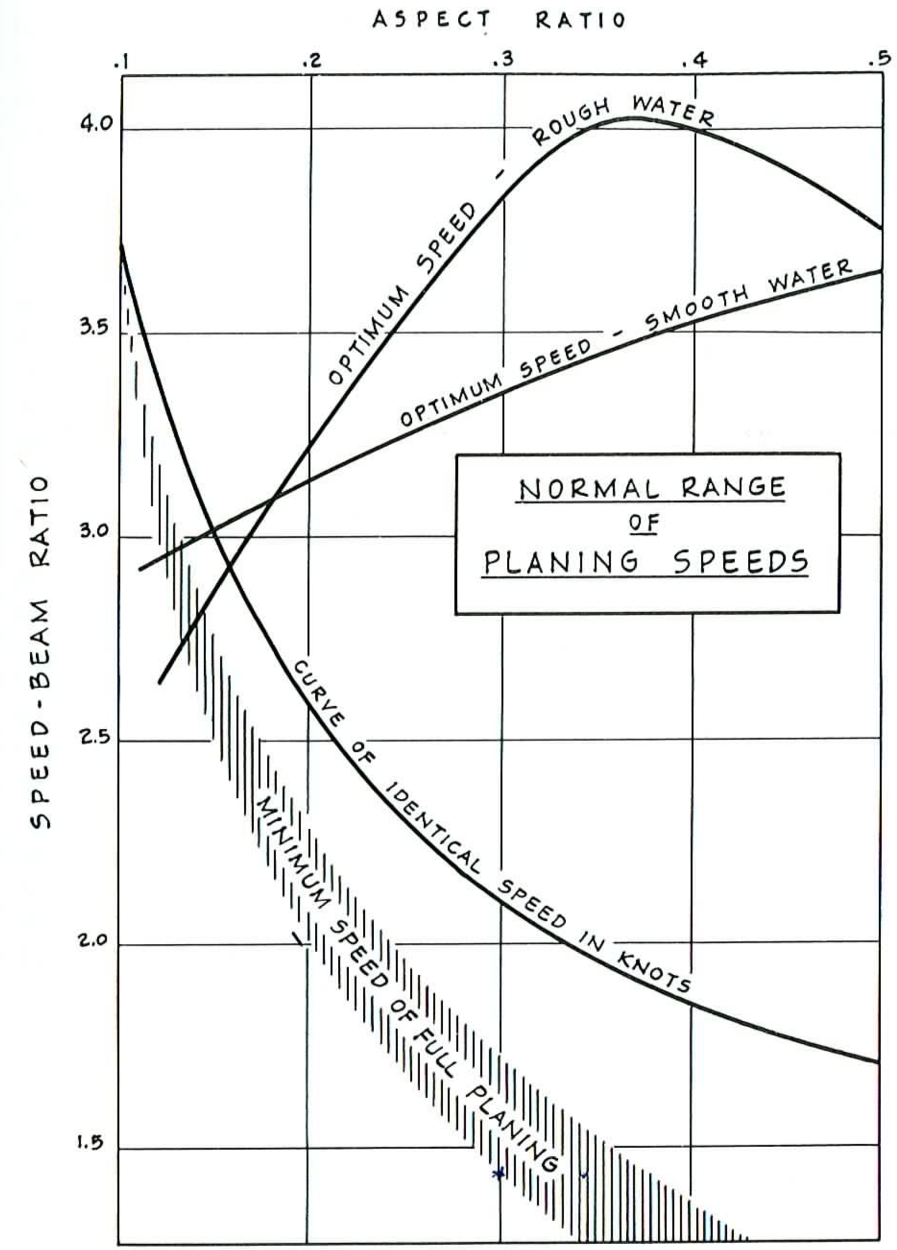


Figure 9

shown to be attainable by the wider bottoms. It is seen that with equal loading per square foot, a hull of .35 aspect ratio may start to plane at a speed-beam ratio of as low as 1.25 while the narrow hull of .2 aspect ratio would probably not develop any pronounced planing tendency until well over a speed-beam ratio of 2.

The curve of identical speed in knots has been inserted to give graphic illustration of how the wider hulls, having greater lift, tend to get up and plane at appreciably lower speeds than narrower hulls.

The two curves of optimum speed are particularly interesting. For strictly smooth water, the inherent speed of planing hulls increases fairly constantly with beam, at least within the possible range of normal marine proportions. However, for rough water or any kind of seagoing service, the optimum curve is of an entirely different character. The speed potential of all hulls throughout the range of aspect ratios between .2 and .5 has been increased due to the lessened skin friction of hulls skimming over wave crests and air bubble cushioning. However, the shape of this rough-water curve clearly indicates that bottoms wider than an aspect ratio of around .4 begin to suffer from the impact of plunging head-on into steep waves. The amount of this loss naturally depends upon particular sea conditions. But it is obvious that a bottom of .5 aspect ratio can have its great smooth-water speed potential at least nullified by head sea impact.

To take the fullest advantage of the optimum planing potential at any given speed-beam ratio below 4, the seagoing hull should obviously be proportioned close to an aspect ratio of .35. The best absolute speed for this hull will be at any point desired which is above a speed-beam ratio of 2 and below 4. Lower speeds are not worth while with a true planing bottom, and higher speeds indicate a hull too small for seagoing requirements.

Chapter 4

LOADING OF PLANES

One of the major gaps to be bridged in the attainment of scientific planing hull design is the application of correct data to bottom loading. The old criterion of loading a hull to an arbitrary Plimsoll line determined by static displacement, ignores two important characteristics of the planing hull: first, the phenomenon of dynamic lift, and second, its corollary in the form of load augment due to induced suction. Both factors arise out of the planing action and are functions of relative speed. The magnitude of each is further modified by the particular pattern of pressures beneath the plane. It is the sum of these various upward and downward components, rather than the static displacement alone, which determines the correct load-carrying power of a plane.

Unfortunately, airfoil or hydrofoil data is of limited value as an approach to this problem. The boat's bottom operating at the boundary between two mediums, one of which is approximately 800 times as dense as the other, allows but one working face of the plane. Furthermore, while this one face should ideally be subjected only to positive pressures, certain configurations of the average bottom lead to varying degrees of transient negative pressures which may detract seriously from the net dynamic lift of the plane.

Lift and Suction

Explanation of decreasing lift toward the aft end of the plane is not difficult. It is in the change of directional momentum of the water particles striking the plane that kinetic energy is transferred. Since any change in the thermodynamic conditions must obviously be negligible, the entire energy impressed by the moving hull on the water is kinetic. Ultimately, of course, all work degenerates into heat. But this takes time, and in the immediate vicinity of the forefoot of the hull, practically all of the work can be accounted for in changes of the kinetic energy of the water.

Translated into terms of useful work, lift occurs only at those points where kinetic energy is being impressed upon the water. The amount of lift at these points will vary approximately as the cosine of the true planing angle at the given point.

The points at which resistance is being turned into kinetic energy are those points at which the effective horizontal flow of water is deflected to some other direction. Since such a deflection occurs at the point of first contact between hull and water, it is obvious that the forwardmost area of the plane deflects water flow most sharply and therefore develops the most lift. Succeeding areas aft encounter a water flow already deflected from the horizontal and therefore tending to reduce the potential angle of deflection of more deeply submerged particles of water. As smaller units of the resistance force are changed to kinetic energy, lift decreases, tending eventually to become negative.

The pattern of pressures under a plane advancing at given speed varies with the aspect ratio and the degree of warpage. The usual test bottoms, or floats, are unwarped surfaces having all buttock lines parallel. On the other hand, vee-bottom hulls show considerable warpage with buttocks varying widely in their inclination and in the typical shape of hull, the chine slopes downward as it runs aft and the keel slopes upward. The intermediate buttocks follow a variety of angles between these two extremes. Obviously, the pressure characteristics, the loading and drag per square foot, vary widely across the bottom.

Figure 10 shows typical pressure patterns below the bottom of a planing hull. Even slight warpage of the planing surface results in a considerable area of suction together with a boundary area of zero pressure. Cases have been known where the suction load alone practically equaled the lift and made planing impossible. More moderate suction loads equal to one third of the weight of the boat are still not uncommon.

A block whose fore-and-aft running lines are straight and parallel can run without any large area of suction, the lift simply decreasing uniformly toward the aft end but remaining, in general, constant athwartship. This favorable resistance and lift characteristic of constant section planes is ideal for smooth-water racers, and while it must be compromised in part for the larger, seagoing types, the basic principle is sound.

The bottom of the ocean-going hull may be so designed that a major portion of its true planing area is formed of parallel running lines, but these lines must be faired into an entrance that is sea-kindly, into a keel

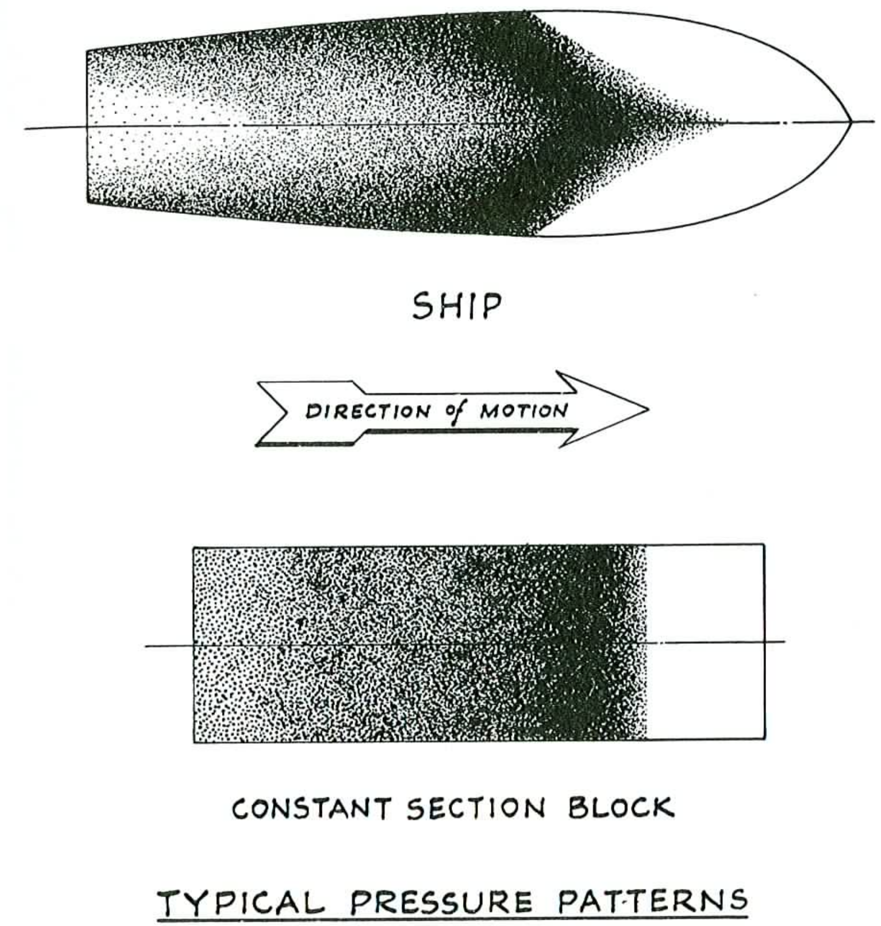


FIGURE 10

for directional stability and into a deadrise and stern that banks properly for steering and at the same time does not tend to broach in following seas. All of these features are essential to sea-keeping ability and, fortunately, their successful combination does not involve the mutual antagonisms of traditional displacement hull design. Buttock lines that are parallel aft of midships incorporate easily into the soft-riding bow and are conducive to the wide stern with its superior lateral plane characteristic. These very details also foster minimum wetted

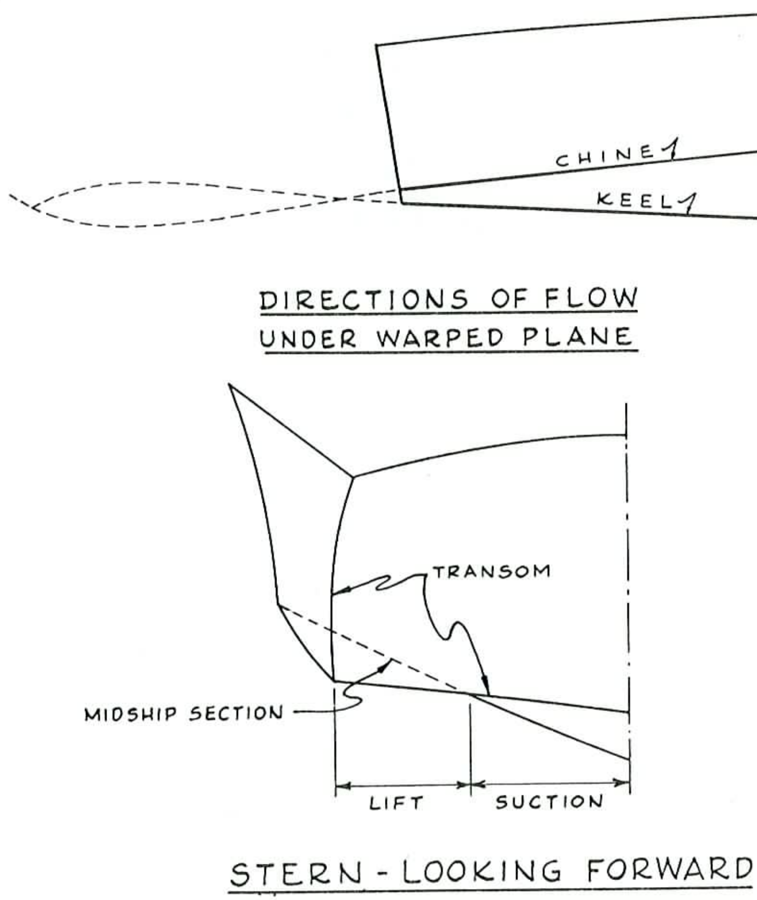


Figure 11

surface resistance. The final wetted area should not greatly exceed in extent that portion of bottom where planing may take place at a uniform angle.

Figure 11 shows the outline of a typical warped plane viewed while running. The lines of flow, following the buttock contours, appear aft of the transom in a pronounced "rooster tail" formation. This so-called rooster tail of the wake has no connection with propeller action; it is

largely the result of a hull form which imparts a non-uniform angular velocity to the water particles across the wake, the sides of the wake being depressed more than the center. The resurgent water then piles up on an already high center.

In the sectional view of Figure 11, this same hull, in planing position, shows clearly the approximate outboard area which is inclined in the direction for attaining a positive lift component. The inboard area, having the opposite inclination, can deliver only a negative lift component.

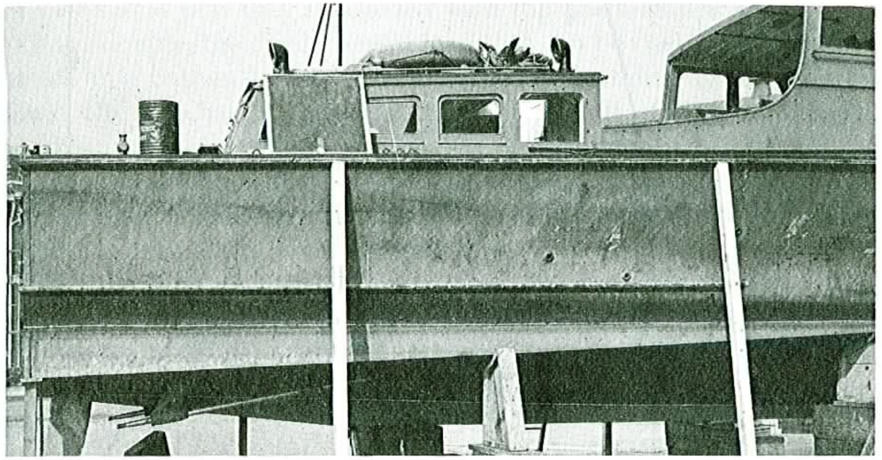


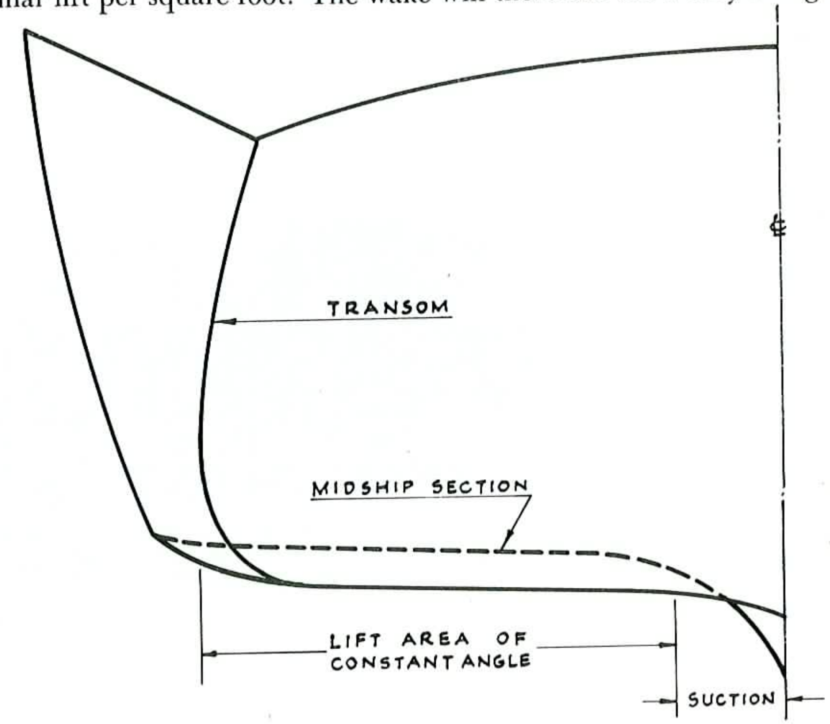
Figure 12

Figure 12 shows a typical example of the warped plane bottom. At speed, these 45-footers develop a suction load which exceeds 25 per cent of their static displacement.

Buttock lines, which are, of course, the lines of flow, have a positive lift only toward the outboard sides while those inboard allow a flow tending upward and release the rooster tail in the wake. This hull, trimmed to plane at a steeper angle, would reduce the area of suction but would create needless drag in its excess planing angle at the chine. The excess squat would also cause more water closing in laterally abaft the transom and further increase the rooster tail. Just how the true planing angle of such a bottom might be measured is impossible to say. There is no single angle.

In Figure 13 is shown the closest practical approximation to the constant section ideal which a seagoing bottom can ordinarily attain.

Some downward suction still remains due to the necessity of fairing-in the keel and the proper entrance lines, but the major portion of the planing area shows running lines all inclined at the same angle. Each square foot of any arbitrary band athwartship will develop reasonably similar lift per square foot. The wake will therefore show only a negligible rooster tail and will be a relatively smooth trough until covered by the lateral resurgence of water from the sides.



PLANE OF MINIMUM SUCTION DRAG

STERN VIEW, LOOKING FORWARD

FIGURE 13

In connection with this preliminary discussion of the differing angular characteristics of "monohedron" versus "multihedron" planes, the Sea Sled, or inverted vee bottom, must be mentioned. While the warpage of the typical vee bottom is conducive to most of its suction load,

the same warpage in reverse on the inverted vee bottom of the sea sled does not increase suction. The reason for its freedom from suction is that the least angle of its plane is outboard at the chines where it not only never operates at a negative planing angle, but also, being outboard, is self-venting. Neither of these favorable characteristics applies to the standard vee-bottom hull.

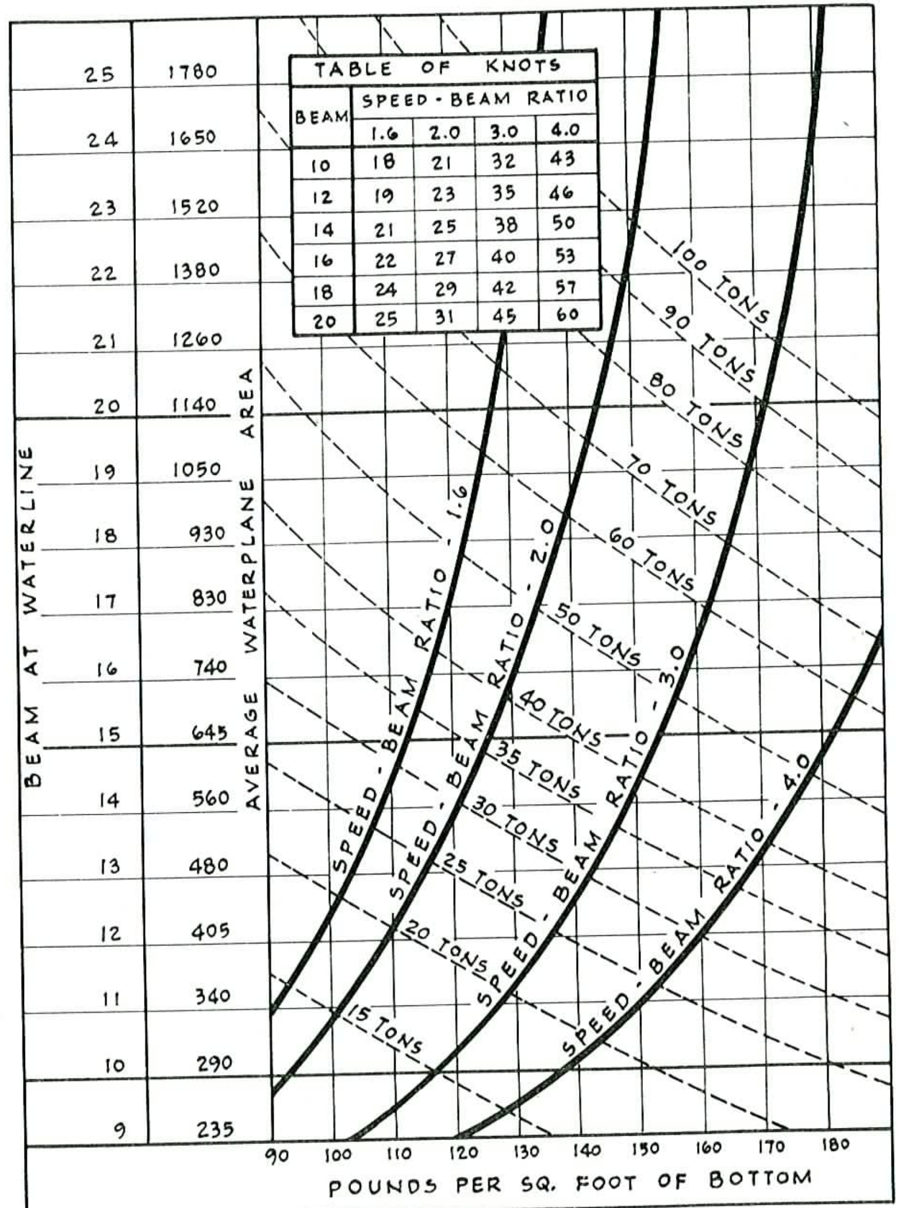
Recommended Loadings

The chart in Figure 14 is a guide as to the loading per square foot of plane area that may be carried with top performance at sea.

For example, the displacement load that can be continuously carried at sea on a bottom 15 feet wide at a speed-beam ratio of 3.0, is as follows: On the chart, it appears that such a craft of 15-foot waterline beam and .35 aspect ratio will have in the vicinity of 645 square feet of plane area and will efficiently carry about 153 pounds per square foot of plane. Therefore this bottom may be successfully loaded to a total displacement of about 44 long tons.

This is by no means the greatest possible load with which a well-designed hull can plane. It is considerably below that point by perhaps an average of 50 per cent. The Higgins PT, for example, exceeds this recommended maximum by at least that percentage and still attains some measure of planing action at sea. However, a large proportion of its weight is obviously not carried by planing action alone but by actual displacement, and therefore the power cost in relation to speed is excessive if considered from a strictly dollars-and-cents efficiency point of view. The loading figures given in Figure 14 are intended as a conservative maximum for clean planing. Also, the loads given are the static displacement weight of the vessel and cargo, a minimum load of suction being automatically taken into consideration. Excess suction loads are therefore in addition to the charted loads.

For example, the Navy's 63-foot crash boats have proved to be one of the least inefficient types in quantity production. The complete weight of one of these boats is 30 tons, which appears to be about the limit of useful load at which her performance is sufficiently agile. By means of pressure taps installed through the bottom, it has been determined that some 2.1 tons of suction load is also being carried and that about 15 per cent of her plane area is operating at or near zero pressure. This area would have been capable of a lift of around 5 tons if allowed to attain positive pressure by operating at a correct angle. Thus it is



BEAM	SPEED - BEAM RATIO 1.6	2.0	3.0	4.0
10	18	21	32	43
12	19	23	35	46
14	21	25	38	50
16	22	27	40	53
18	24	29	42	57
20	25	31	45	60

MAXIMUM LOADING
FOR BOTTOMS OF .35 ASPECT RATIO

Figure 14

seen how even this well-designed hull is actually carrying a weight of 32.1 tons and has an unused potential lift of 5 more tons. Since this boat, if perfected, could actually carry 37.1 displacement tons without increase of power, the hull is 81 per cent efficient in useful lift.

At the other end of the scale were some of the early British PT models. Here a vessel weighing 48 tons with all equipment, developed a total suction of 13 tons and dragged at zero pressure a plane area with potential lift of another 15 tons. With a lost opportunity to achieve lift of 76 tons, these first PTs staggered along with less than 63 per cent of the net load which might have been carried with equal power on a properly designed bottom.

Overloading the Plane

The effect of an overload on a planing bottom is to cause an increase in planing angle. This increase in angle not only attains a direct increase in lift, speed remaining constant, but usually brings more area of bottom into effective use. The additional trim gives positive angle to much of the area formerly operating at zero pressure, and also reduces the area carrying suction. In this way additional lifting power is acquired. In models being towed at constant speed, the increase of trim to gain lift for added weight is most pronounced.

In full-sized craft, the added weight is similarly compensated by increased trim, but the greater resistance of the steeper trim immediately adds its own increased wave-making resistance, and speed falls off correspondingly. With a reduction of speed, lift can be maintained only by further increase of planing angle. There is thus an easily observable point at which further loading makes true planing impossible and the hull is carrying its weight entirely by displacement, with speed no longer a major factor.

The operation of naval vessels under their usual overloads is due largely to the almost daily advance of technology in new equipment and instruments which overnight become necessary to fighting efficiency. Therefore, just as it is an axiom among yachtsmen that the perfect 30-footer is 40 feet long, so should the 50-ton displacement load in design be assumed as 60.

Underloading

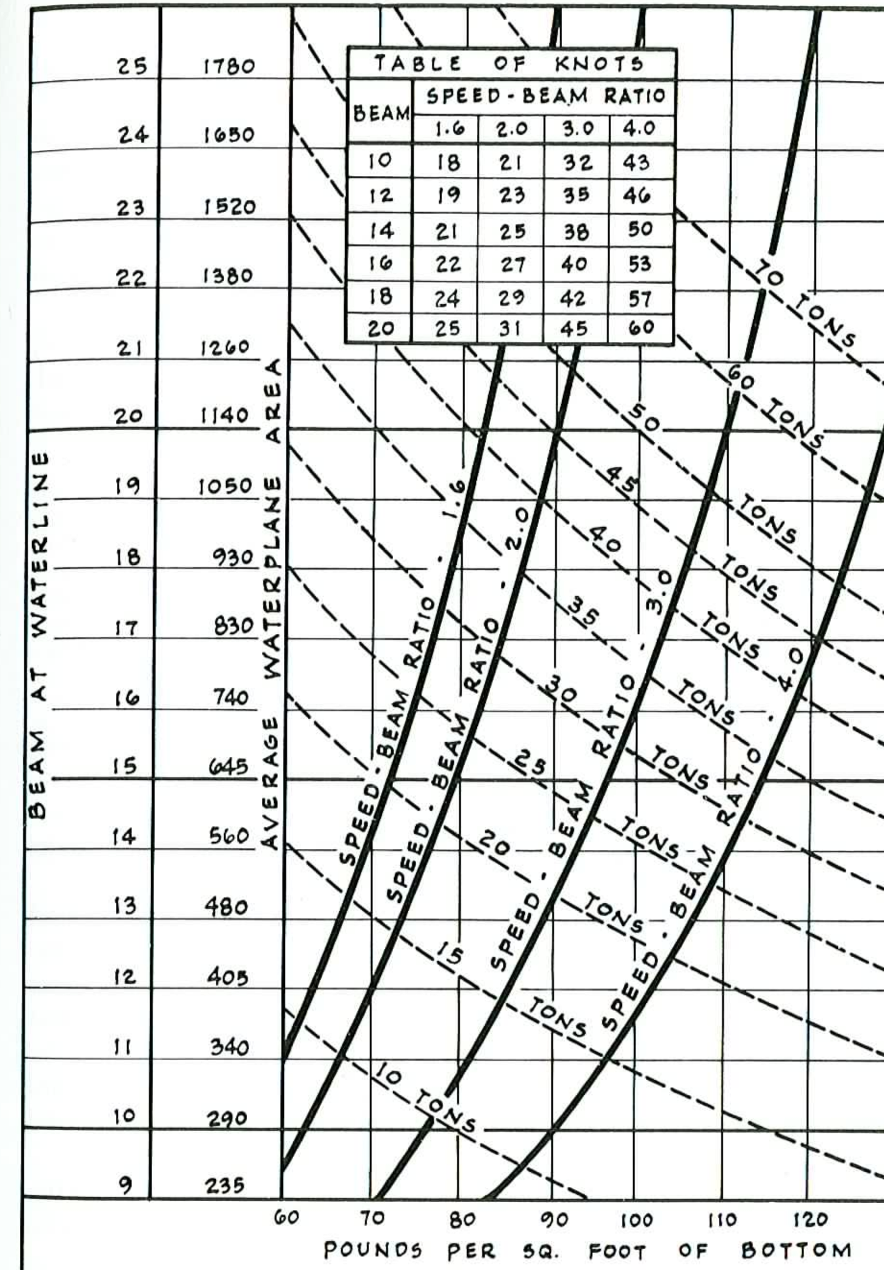
While by no means as common as overloading, underloading of the plane is fully as bad. The temptation to design an oversized plane with low unit loading is very great, offering as it does a substantial reduction in wave-making resistance and, at low planing speeds, some reduction in wetted surface friction. However, there are limits both ways on load per square foot if seagoing performance is to be retained.

The plane operating at sea without sufficient loading is dangerous in the extreme. It is in general comparable to an empty cargo vessel operating without ballast. The constant angle type, or monohedron, planing as it does with relatively little suction load, is more liable to the effects of underloading than are warped plane types with their ever-present ballast of suction. Furthermore, the true monohedron, with its wider aspect ratio, is inherently a load carrier and must be so operated for safety at sea. Otherwise, dynamic forces due to wind and wave action assume a magnitude disproportionate to the balancing forces.

A hull planing too lightly on the water is, to a degree, ready to take off. Planing hulls of racing type frequently do take off, hurling themselves through the air for a boat length or more. However good this performance may be for racing, it must inevitably lead to disaster at sea. The exposure of excess surface to a high wind broad off the bow has resulted in heeling an underloaded hull on its beam ends. Should the hull be so heeled by wind action while more or less completely out of the water, it might easily return to the surface with an angular impact sufficient to cause capsizing.

Figure 15 is a chart of the minimum allowable loading for monohedron planes. This does not mean that there is actually any dividing line between safely and unsafely loaded planes. Rather, it is a case of the more loading up to a plane's maximum lift, the safer. But since power consumption varies so closely with load, the private pleasure cruiser, in the desire to economize on power, is about as frequently underloaded as naval craft are overloaded. This condition must be carefully weighed in any discussion of sea-keeping ability.

For example, the minimum weight to which a 40-foot pleasure cruiser of 10-foot waterline beam may be designed for reasonable performance at sea is as follows: With a speed-beam ratio of 2.0 the weight must be as a minimum, around 62 pounds per square foot of bottom. Opposite the 10-foot beam, it is seen that a hull of .35 aspect ratio will have about 290 square feet of waterplane. Thus the minimum designed load for this craft should be: $62 \times 290 = 17{,}980$ pounds for the minimum performance to be called seagoing at that particular speed. For higher speeds the loading must be increased for equivalent safety.



MINIMUM LOADING

FOR BOTTOMS OF .35 ASPECT RATIO

FIGURE 15

Ideal Loading

A safe, average loading, to avoid as far as possible the dangers of partial take-offs in rough weather, and to hold to a reasonable power consumption consistent with private pocketbooks, is shown in Figure 16. According to this chart, the previous 40-footer, if designed to venture for extended runs into the open sea, should be loaded to at least around 75 pounds per square foot, or a total of 21,750 pounds.

The following table of typical bottom loadings is presented simply because it is illuminating on the frequency with which rule-of-thumb design has happened to approach apparent good loading practice and yet has been wide of the mark in allowable dynamic loads.

Boat	*Aspect Ratio*	*Speed-Beam Ratio*	*Load per Sq. Ft.*	*Ideal Load*
33′ Target, Army	.27	4.9	55	110
42′ Crash, Army	.30	2.2	125	85
45′ Crash, Navy	.31	2.5	85	100
50′ Fisherman	.35	2.4	90	90
60′ Rumrunner	.25	2.7	90	100
63′ Crash, Navy	.32	3.0	120	115
77′ PT	.26	3.9	130	120

The 33-foot target boat class, although of warped plane type, could not possibly develop sufficient excess weight of suction to bring about the total necessary for safety at sea. It may be said in her defense that she was not designed for the job but was simply a stock model runabout rushed into uniform as a quick expedient.

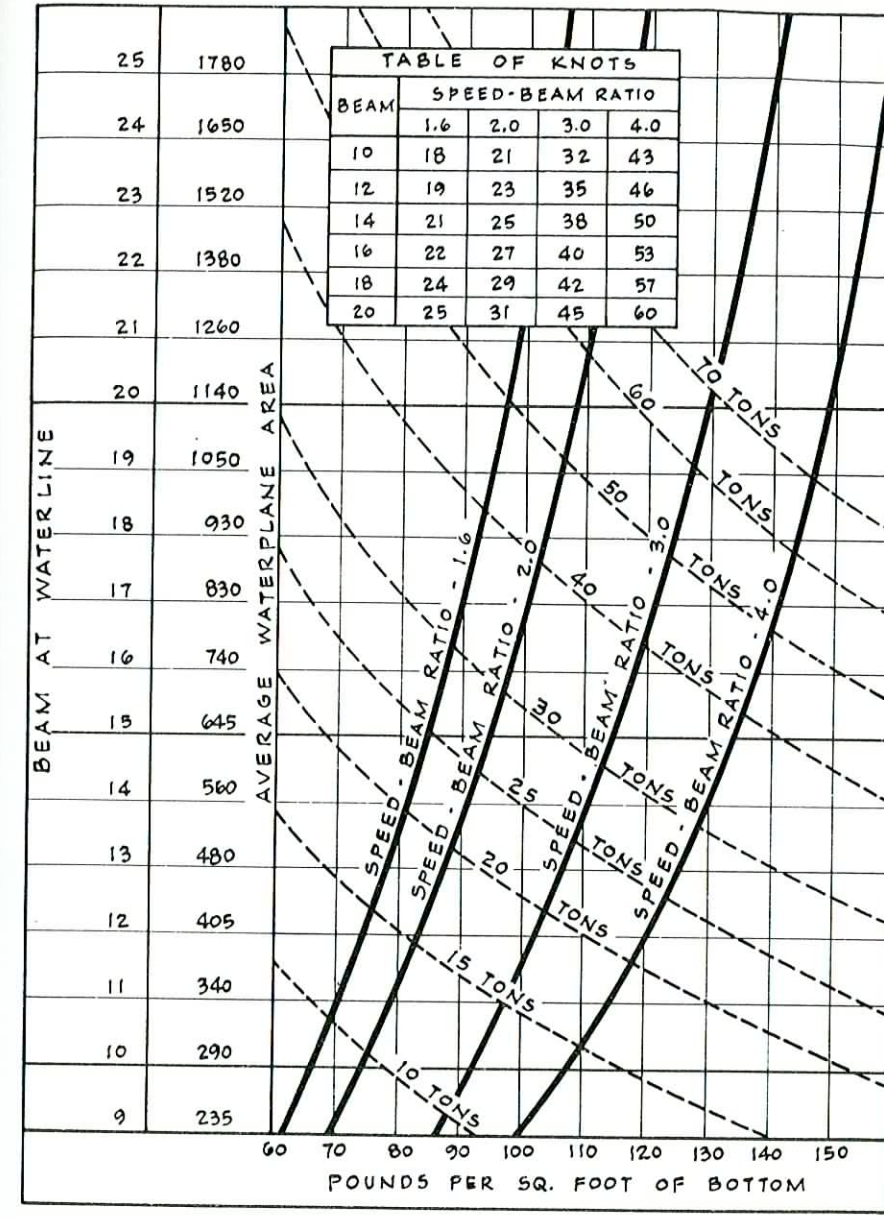
The 42-foot crash boat was too narrow and therefore carried its load largely by displacement. However, this extra loading did add to comfort in spite of its power cost.

The 45-foot crash boat appears to be somewhat underloaded, but this model is one of the worst sufferers from suction load induced by plane warpage, and the total load actually runs considerably beyond the maximum allowed by good practice. Her lift efficiency is around 64 per cent.

The 50-foot fisherman was a monohedron design by Lord, with meticulous attention given to its development. Her efficiency in lift is 93 per cent.

The 60-foot rumrunner class, designed by Lord in 1930, came as close to the ideal as the old-style, narrow hulls possibly could, and are in constant use today by the Coast Guard on Central Pacific patrol duty.

The 63-foot crash boat, a Long design, comes very close to scientific loading and is among the superior craft being mass produced today.



STANDARD LOADING
FOR BOTTOMS OF .35 ASPECT RATIO

Figure 16

The 77-foot PT, from an abandoned Thornycroft design, added such an enormous suction load that performance never justified production. While its displacement load per square foot of bottom was not greatly in excess of the recommended load, the discouraging proportions to which an unseen dynamic load may rise on poorly shaped bottoms, is indicative, among several even more important considerations, of the inaccuracy of static displacement as a guide to loading.

The Coefficient of Load

The necessity for a correlation of speed and load, as mentioned in the previous chapter, indicates the need for a coefficient by which a logical comparison between planing hulls can be made. Such a coefficient must recognize both that as absolute planing speed increases and that as actual beam increases, load potential also increases. To meet this need the following coefficient of load, C_L, is submitted:

$$C_L = \frac{\dfrac{100\,W}{\sqrt{V}}}{B \times a}$$

Where:

W = Weight in tons
V = Speed in knots
B = Beam in feet
a = Aspect ratio of plane

For example, the speed-weight ratio of a 30-ton, 35-knot crash boat which has a 12-foot beam with .3 aspect ratio is:

$$C_L = \frac{\dfrac{3000}{\sqrt{35}}}{12 \times .3} = \frac{509}{3.6} = 141$$

Were its load reduced to 20 tons with other factors the same:

$$C_L = \frac{\dfrac{2000}{\sqrt{35}}}{12 \times .3} = \frac{339}{3.6} = 94$$

Or, with the original 30-ton load, assume speed were reduced to 20 knots, thereby reducing the lift. The effect is to increase its proportionate load:

$$C_L = \frac{\dfrac{3000}{\sqrt{20}}}{12 \times .3} = \frac{670}{3.6} = 185$$

Figure 17 summarizes average good practice in designing the various types of seagoing planing hulls. That this proposed coefficient of load still leaves something to be desired for purposes of direct comparison

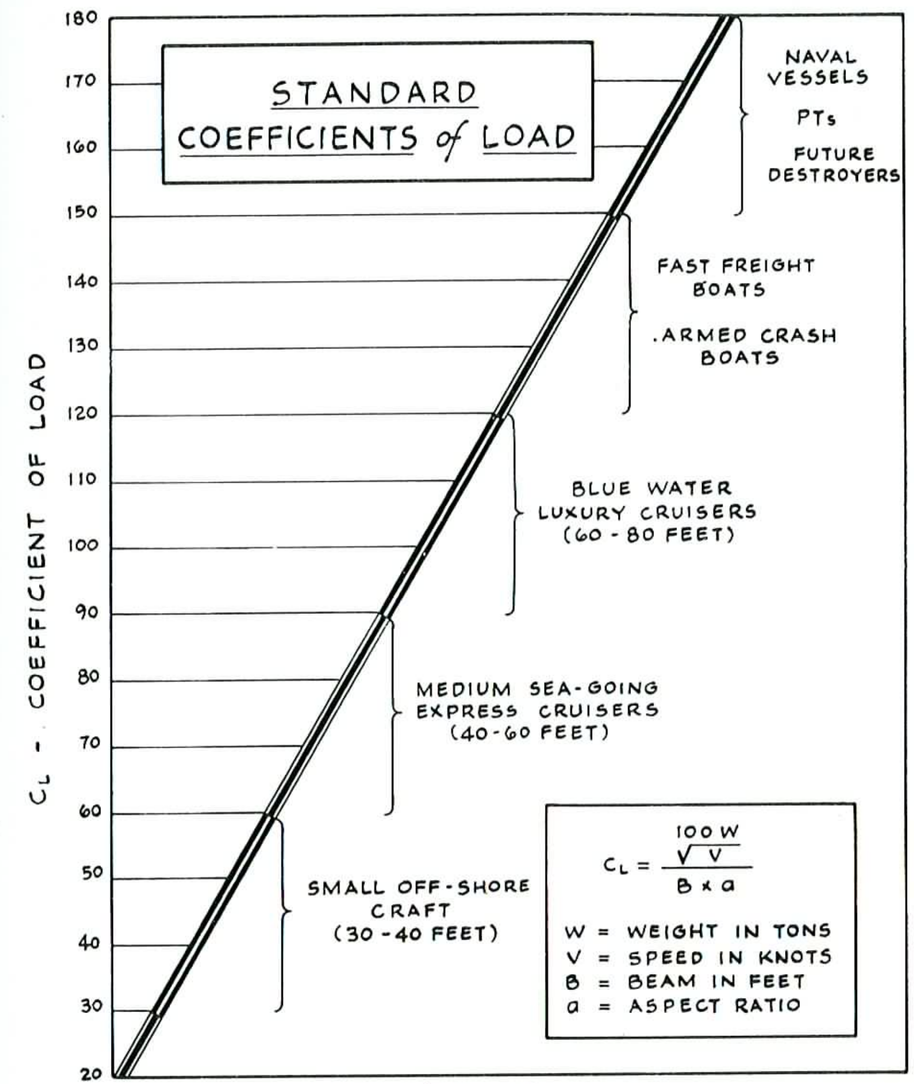


Figure 17

among any and all sizes or types of planing hulls is obvious from the fact that while larger hulls are known to be inherently greater proportionate load carriers, the coefficient does not indicate the whole comparative difference. However, this shading of final accuracy is undoubtedly less than the individual differences in shape between two bottoms, and if it is recognized that small craft should normally have lower coefficients than larger boats under actually similar conditions, the coefficient can be used with most helpful results in making an intelligent comparison between hulls and for scientifically loading a new design. In this way, it corresponds to the displacement-length ratio of displacement craft and definitely establishes the relative weight of a planing hull under the interlocked design conditions of speed and lift.

Typical Speed-Weight Load Coefficients

Type	*Speed Knots*	*Weight Tons*	*Beam Feet*	*Aspect Ratio*	*Load Coeff*	*Ideal Coeff*
33′ Target Boat	44	4	7	.26	33	55
40′ Cruiser	17	8	10	.35	56	56
42′ Rescue Boat	22	16	10	.31	110	80
53′ Cruiser	38	24	13	.33	91	90
63′ Crash Boat	35	30	12	.30	141	140
77′ Torpedo Boat	35	48	14	.28	212	170
85′ Patrol Boat	40	65	21	.35	140	160

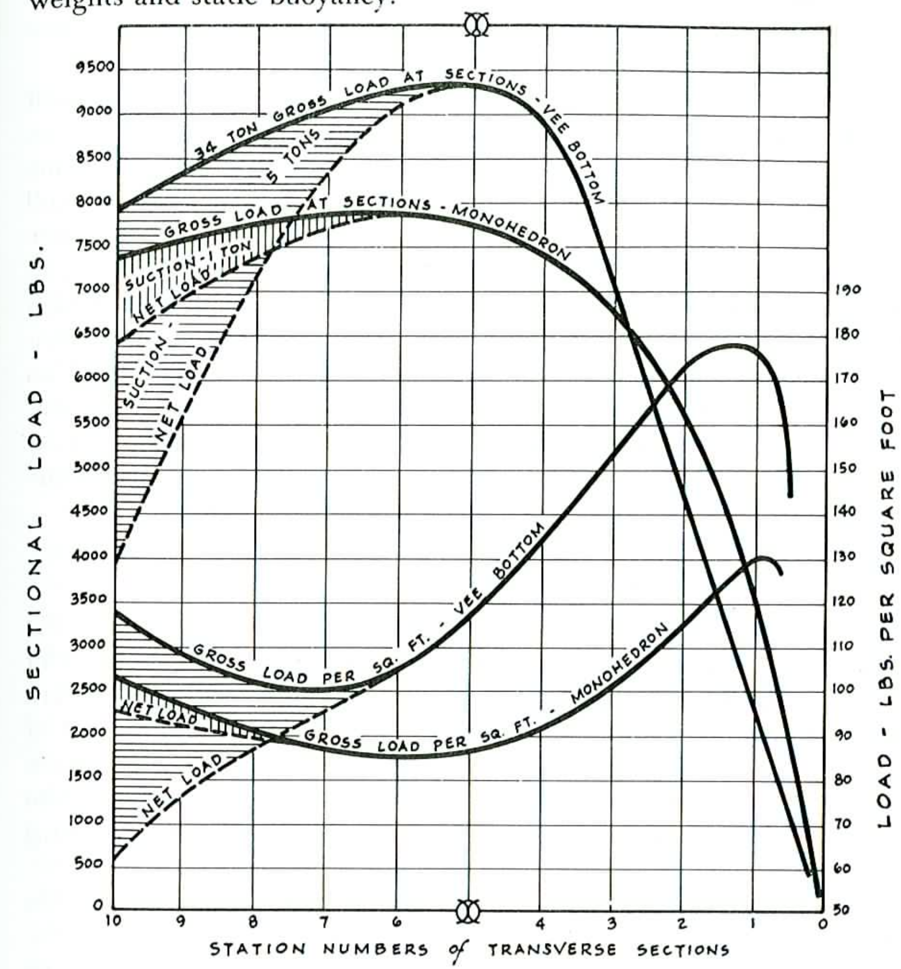
Lift Distribution

The chart in Figure 18 showing the comparative lift distribution between typical vee-bottom and monohedron types, and their dynamic loads, is worthy of careful study. The curves were derived from two hulls of approximately similar waterplane areas and with identical total loads at rest of 29 tons each. How greatly this load was increased by suction when planing is shown by the area between "gross" and "net" curves. For the vee-bottom hull, the gross weight when planing increased to 34 tons and that of the monohedron increased by 1 ton.

The two curves at the bottom of the chart show the same loads, but integrated on a parameter of lift per square foot of bottom area instead of the top curves' total lift at given stations.

Several important factors in the performance of planing hulls are illustrated by these curves. Of primary significance is the increase of load and lift aft of midships. The effect of this increase is to augment the longitudinal moment of the hull, a condition directly opposed to good seagoing performance. A high longitudinal moment caused by weight at some distance from midships leads to fore and aft stiffness that plunges the hull into, rather than raising it over, wave tops. Excess weight aft is also a cause of broaching. It is with the increased weight

and consequent lift distribution shown by these gross curves that the designer must be concerned, rather than with the half truths of static weights and static buoyancy.



LIFT DISTRIBUTION
WHILE PLANING WITH
29 TON STATIC LOADS

Figure 18

The 29-ton static load represented in these curves worked out to a load per square foot of 100 pounds. The marked change in the distribution of this lift when planing is apparent from the curves of load per square foot. High unit pressures forward, due to leading edge effect, cause more or less trim, which in turn brings a proportionate increase in unit pressures aft. Concave forward sections greatly aggravate these extremes of lift at the ends, as do also low forward chines and excessively narrow sterns.

It is obvious that the pressure per square foot under the stern must rise in direct proportion to reduction in beam at the stern. However, since any stern, wide or narrow, without compensating lateral plane invites broaching in following seas, the designer must so shape his hull and distribute his static weights that the curve of weight per square foot, together with its expected load of suction, is reasonably constant. The trend in modern design is to retain a slight increase in gross weight per square foot at the aft end simply as a shock absorber for the inevitable pitching that comes from running into head seas. Too great an increase aft, however, such as brought about by very narrow transoms, has a tendency to allow excessive squatting and therefore fails to remove the danger of broaching. The compromise is a fine one and requires intelligent analysis of comparative previous hulls.

Magnitude of Suction Load

Except by a physical test of the actual boat, the exact weight of the suction load is unknown. It can, however, be very closely estimated for design purposes by comparison with previous similar hulls. In modern hulls the load varies between 2 and 9 per cent of the static weight of the boat. Plane warpage, hull appendages and propeller cavitation are the main factors involved. Older type hulls of very narrow beam also added to the suction load because of the poor aspect ratio of planing surface.

Hulls which most nearly approach the constant section ideal in the after portion of the plane show the least suction load.

The following examples are indicative of present suction loads on typical boats:

40′ Cruiser, Convex monohedron: Aspect ratio .35; Speed-beam ratio 1.8; Static weight 12 tons. Suction load .24 tons or 2%.

42′ Rescue, Concave vee bottom: Aspect ratio .3; Speed-beam ratio 2.2; Static Weight 13 tons. Suction load 1.68 tons or 14%.

63′ Crash, Straight deadrise and convex bottom: Aspect ratio .32; Speed-beam ratio 3.0; Static weight 30 tons. Suction 2.1 tons or 7%.

80′ Huckins PT, Concave vee bottom with extremely low deadrise aft: Aspect ratio .32; Speed-beam ratio 3.1; Static weight 58 tons. Suction load 3.5 tons or 6%.

Chapter 5

PRELIMINARY DESIGN

Visualization

No design can be successful without prior analysis of the services which it must fulfill and the inherent qualities necessary to meet those service requirements. The whole success of all designs, whether for pleasure, commercial or naval craft, is equally dependent on this required service analysis.

From the broad outline of preconceived service requirements, the first tangible step in preliminary design of small craft must always be the freehand sketch of profile and arrangement plan. It is from these that intelligent thought may be applied toward reasonable adjustment between qualities desired and qualities possible.

The list of service requirements, that is, qualities desired, is, in spite of the complaints of some designers to the contrary, seldom sufficiently demanding. Only in the humorous article is the competent designer ever confronted with service requirements so mutually antagonistic as to present any real difficulty of solution. The true fact is that most designs are created and accepted with too low a standard of qualities desired. Perhaps the deadweight tradition of an art which was old when civilization dawned has set the mold too firmly. Perhaps, too, the inspired design must always remain a rarity

The traditional antagonisms of displacement hull design have always compromised either speed or roominess, beauty or cost, speed or seaworthiness. It was axiomatic that there could be no "best" boat, only a best compromise.

Fortunately, the planing hull is potentially free from tradition. It need not be bound to the antagonisms of which displacement hull designers are instinctively wary. Its roominess is a natural outcome of a shape scientifically matched to speed and seaworthiness. Its beauty need not of itself add cost since the higher order of skill needed for successful planing hull design will automatically reflect itself in masterful treatment of line and material.

From freehand sketch plans, the size and approximate weight, with consequent power requirements for the speed desired, may be roughly determined. Ordinarily, yachts will yield the most satisfaction if powered within a range of speed-beam ratios from 1.8 to 2.8, commercial vessels from 2.0 to 2.5 and naval craft from 2.5 to 3.5. This in turn has retroactive bearing on the size of hull.

From the sketch plan, showing as it does only the outline size, the approximate hull dimensions for a .35 aspect ratio may be determined from the following table:

HULL DIMENSIONS IN FEET

Length O.A.	*Minimum Beam*	*Chine Beam*	*Mean W.L.*	*Sq. Ft. of Plane*
30	10.0	8	23	184
35	11.3	9	26	234
40	12.5	10	29	290
44	13.7	11	32	352
47	15.0	12	34	408
52	16.3	13	37	481
55	17.5	14	40	560
59	18.8	15	43	645
62	20.0	16	46	736
66	21.4	17	49	833
71	22.5	18	52	936
76	23.8	19	55	1005
80	25.0	20	57	1140

Preliminary sketches are best done on a very small scale to keep the overall length of drawing within 24 to 30 inches so that the whole picture may be easily visualized while it takes shape.

Having determined the probable waterline or chine beam consistent with the size of hull necessary, a suitable speed may be determined from the following table of speed-beam ratios.

TABLE OF SPEED IN KNOTS

S-B Ratio	*Beam in Feet*					
	8	*10*	*12*	*14*	*16*	*18*
3.2	30.2	33.8	37.2	40.0	42.9	45.7
3.0	28.3	32.0	34.8	37.6	40.3	43.0
2.8	26.5	29.7	32.5	35.1	37.6	40.0
2.6	24.6	27.6	30.1	32.6	34.8	37.2
2.4	22.7	25.5	27.8	30.1	32.2	34.3
2.2	20.8	23.3	25.5	27.5	29.6	31.3
2.0	18.9	21.3	23.2	25.1	26.8	28.6
1.8	17.i	19.2	20.8	22.6	24.2	25.7

Having visualized the necessary size of hull and the appropriate speed, the correct weight to be achieved in final design may be determined from one of the charts in Figures 14, 15 and 16. Naval and commercial craft should be guided by the chart of Maximum Loading; seagoing pleasure cruisers may follow the Standard Loading of Figure 16. Only those boats designed for low cost should be as lightly loaded as indicated in the chart of Minimum Loading, for these will ordinarily not venture the discomforts of extreme heavy weather offshore.

POWER ESTIMATES

From the charts shown in Figures 19 and 20, very close estimates of the speed, power and weight relationship may be made. To use the charts, a triangle edge is laid across any two known quantities and the third read direct. For example, a hull with good monohedron characteristics that weighs 15 tons and is powered with 375 applied horsepower, may reasonably be expected to deliver 21 knots. The same weight and power in a typical displacement type hull would deliver equal speed provided the hull had a length of about 90 feet. Both of these charts were constructed from data on hulls delivering average good performance. The vee bottom chart was compiled by Paul G. Tomalin of the U. S. Coast Guard. The chart on displacement hull speeds is the product of the Marran and Shaw thesis.

The approximate size of propeller for the application of the correct power must also be known in the preliminary design stage. To assist in determining probable wheel diameters, the chart shown in Figure 21 is most helpful. This highly simplified nomograph was prepared by Mr. G. Tobil of Fremantle, Australia and gives surprisingly good results. To estimate a wheel size, simply place a straight edge from propeller revolutions to engine horse power and read the diameter in inches at the right.

In order that the other conditions of a given weight-power ratio may also have due consideration before a diameter is selected, it is wise to be fairly certain of a suitable pitch. By turning to page 199, the chart in Figure 82 will provide a good estimate. It now becomes a matter of judgement to so balance the propeller dimensions that, if at all possible, pitch will be about the same as diameter on medium speed bottoms and greater than diameter on faster hulls.

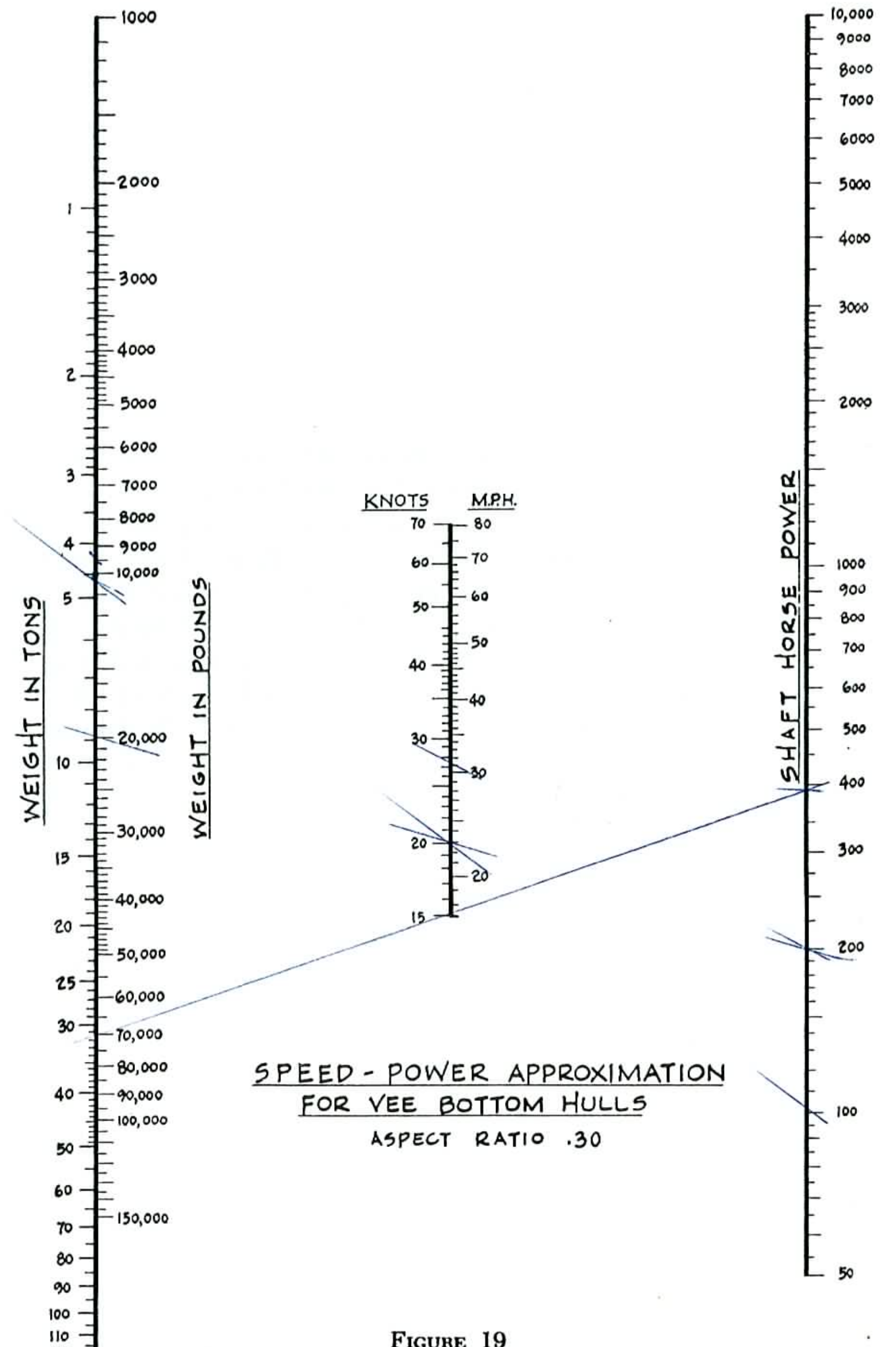


FIGURE 19

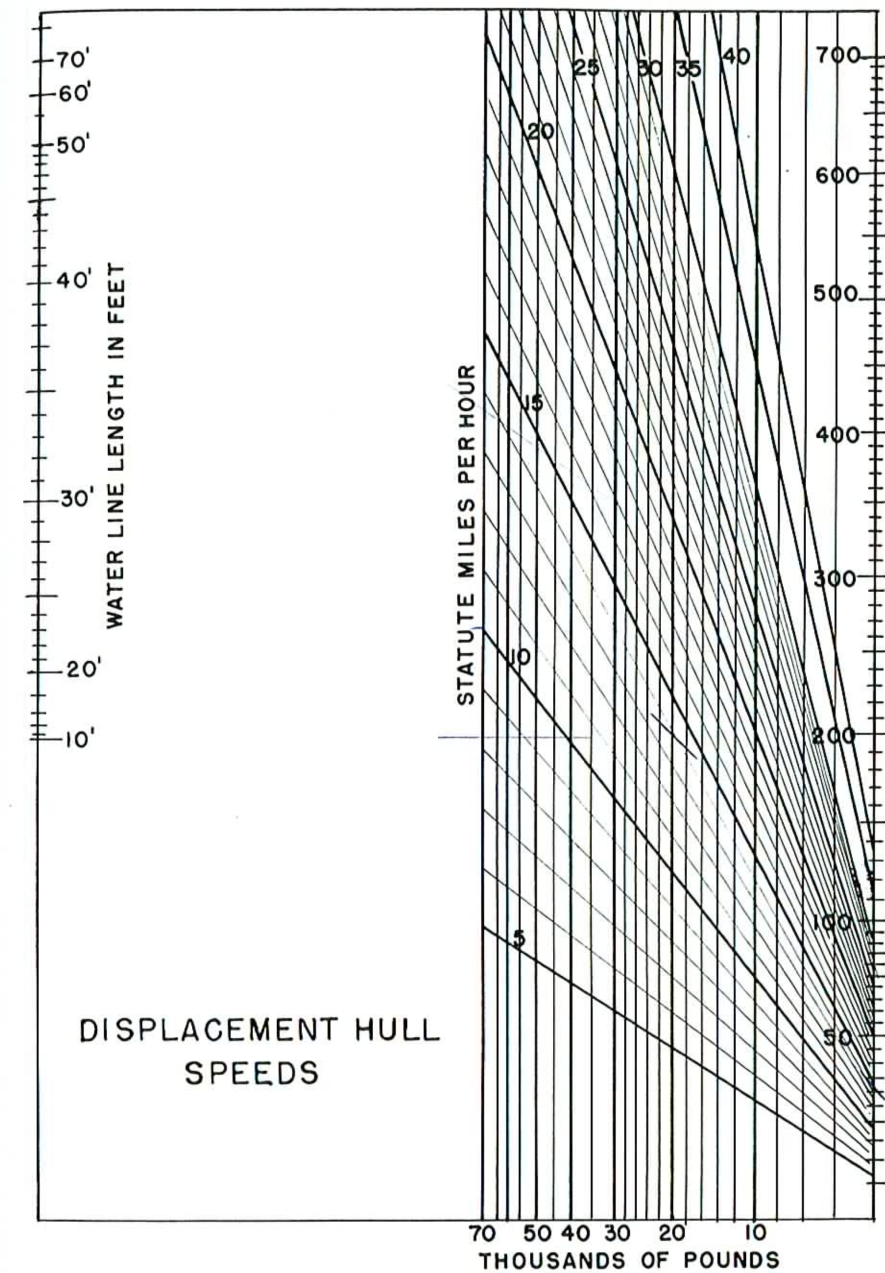


FIGURE 20

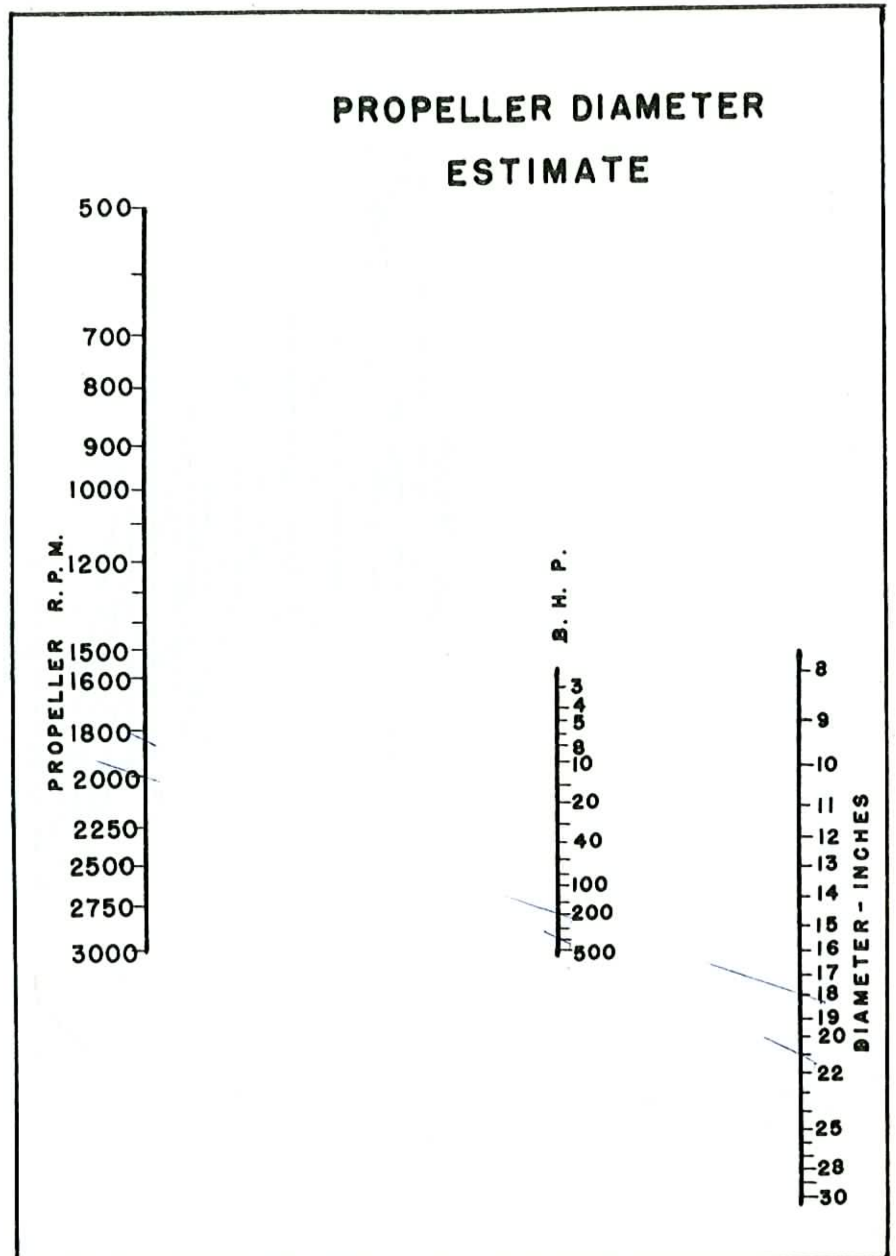


Figure 21

Contract Plans

For bid and contract purposes, five or six sheets of plans are usually sufficient. These plans include (1) Lines and Offsets, (2) Outboard Profile, (3) Deck Plan, (4) General Arrangement, (5) Inboard Profile, (6) Construction Plan and (7) Typical Sections. In addition to these plans, detailed specifications of materials and requirements not fully shown on the plans, must be drawn up. For boats up to around 40 feet overall, where a scale of ¾ inch to the foot or larger is possible, it is not usually necessary for the designer to furnish any additional plans. The few questions that may arise are easily settled in the normal course of periodic inspection, conducted as much for the designer's own protection as for the builder's information.

For larger craft or radical designs, the designer must furnish additional plans in detail to cover foundations, bulkheads, tanks, wiring, piping, ventilation, fittings and any parts of the structure which cannot be ordered simply by a catalogue number.

Shop plans, based upon the specifications, are always prepared by the builder in order that the details of trade work may be best suited to his own facilities and in order that final design may proceed with drafting, procurement and construction under a single authority and thereby facilitate the work. All drawings prepared by the builder must, of course, be approved and signed by the designer.

Because of the tremendously advanced scientific potential of the planing hull, either a page of blueprint or a few sheets incorporated in the specifications should present the "Particulars." For an accurately designed planing boat, these particulars include many times the amount of data pertinent to displacement hulls. Full explanation of these particulars and the methods of their calculation are covered in subsequent chapters, but the list of data to be presented is usually about as follows:

Length overall
L.W.L. at rest
Beam, extreme
Beam, W.L.
Displacement, bare boat
Displacement, cruising condition
Waterplane coefficient
Aspect ratio

C.G. of fixed weights aft of midships
C.B. aft of midships
C.G. of variable weights aft of midships

Center of flotation aft of midships
Dynamic center of hydraulic pressure
Resultant apparent planing angle
Virtual hydraulic planing angle
Added load due to suction
Pounds per inch immersion at rest
Pounds per inch at speed

Cruising horsepower
Shaft horsepower per ton
Speed, maximum
Speed, cruising

Speed-beam ratio
Dynamic lift, planing
Operating slip, planing
Propeller efficiency
Towrope pull, thrust

Transverse B.M. at rest
Transverse B.M. dynamic
Longitudinal B.M. at rest
Longitudinal B.M. dynamic
Area of waterplane at rest
Area of plane at speed

Wetted surface at rest
Wetted surface planing
Longitudinal moment about C.F.
Increase in virtual weight, pitching
C.G. of effective lateral plane

Turning diameter
Banking angle
Rudder lateral thrust
Increase in weight due to turning
Speed reduction on turns

Specifications

The specifications and contract plans supplement each other. The specifications contain the detail that cannot be shown on general plans. The preparation of a good specification requires both an intimate knowledge of every phase of boat building and also considerable time. The specification writer must cultivate the rare ability to discriminate between the essential and the trivial; to know from experience what to describe and what to leave out. Specifications cannot take the place of shop drawings or "know how," and disregard of this limitation leads to the entangling of important features in a mass of redundant detail.

The usual procedure in drafting the specifications is to start with a previously used set for a craft of generally similar characteristics, and to tailor the particular additions and changes to make it fit. This is quick and effective, but the writer should guard against the tendency thus induced to compound trivialities.

When a good specification is not available for guidance a proposed list of topics should be drawn up and arranged in logical sequence. The shorter the form the better, except for government contracts. Reliable builders universally prefer the latitude offered in short specifications, and the resultant saving in cost is to the owner's benefit. A typical list of headings and their contexts is as follows:

General. This section includes all principal dimensions of the boat, statements as to the builder's responsibility in buying and protecting materials, type of work expected, care of hull during construction, inspection by the designer and a list of plans to be furnished with the specifications. Shop drawings, plans and photographs to be furnished by the builder are also general, as are statements as to workmanship, care in holding down weight and type of work to be performed in addition to that shown on plans.

Materials. Special treatments, such as preservatives, galvanizing and painting, logically go into the materials section. Describe treatment of faying and finished surfaces.

Testing. Includes requirements for testing materials, tanks, piping, bulkheads and structure during the process of construction. Tests on auxiliaries, plumbing, electric, ventilating and fire-fighting systems should also be outlined. Standards of performance for the complete boat during its trials are stated in this section. Guarantees and penalties, if these are contemplated, are more logically a part of the contract proper rather than the specification.

Framing. To be subdivided into paragraphs on Stem, Keel, Knees, Frames, Floors, Keelsons, Beams, etc.

Planking. To include description of material, sizes and method of application.

Fastenings. Materials, locations and essential sizes.

Foundations. Materials, dimensions and fastenings. Rudder stock and shaft log foundations.

Bulkheads. Materials, thickness, stiffeners and construction.

Decks. Beams, carlines, deck materials and structure, fastenings and method of construction.

Guards. Materials, fastenings and method of construction.

Flooring. Materials, access panels and hatches, construction, sizes and finish.

Deck Houses. Materials, sizes, structure and description of exterior doors, windows and finish. Grabs.

Hatches. Weather deck hatches and soft patches.

Interior Joinerwork. Cabin furnishings and finish.

Metalwork. Guard caps, stem band, shoe, skeg and metal trim.

Steering System. Description of type, operation and materials.

Insulation. Type, thickness and location for heat resistance and for acoustical control.

Hardware and Fittings. Materials, sizes and finish of each item. Description of masts, staffs, airports and special fittings.

Plumbing. Catalogue numbers of fixtures, piping and tank sizes and materials.

Electric System. Description by catalogue number of fixtures, batteries, generating system, location of lights, outlets and switches, wire sizes, location and type.

Fire-Extinguishing System. Name, type and location of bottles, nozzles, and release handles.

Ventilating System. Make, model and location of blowers and fans. Duct sizes, construction and grille types. Cowls and vents.

Equipment. In this section may be grouped the specifications for stanchions, life lines, anchors, lines, galley equipment, cabin fittings and upholstery, special racks and lockers, navigating equipment, all special items desired by the owner.

Engineering. Make, model and particulars of main engines and auxiliaries. Shafting, bearings, struts and propellers. Cooling, silencing and exhaust systems. Fuel system. Controls. Pumps. Flexible mountings. Lubricating system.

Index. A complete topical index is of great convenience.

Structural Policy

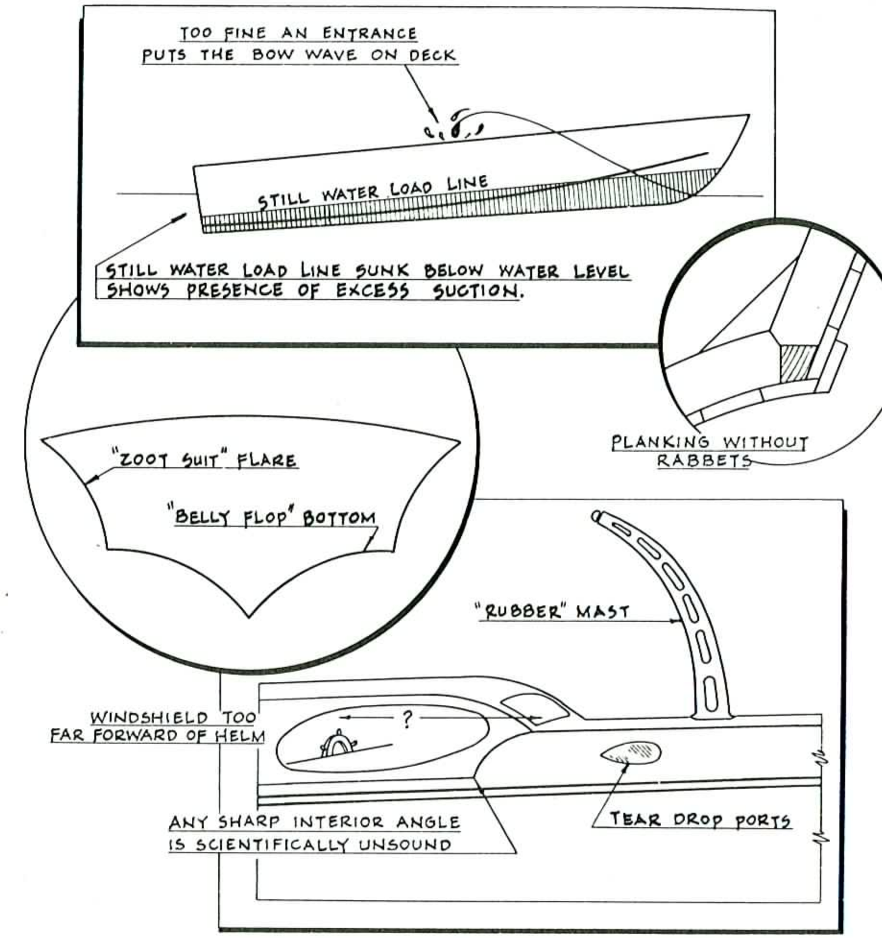
The materials to be used in construction and the facilities for their fabrication will have their effect on weight and shape. The growing importance of molded shapes and plastic materials is bound to revolutionize traditional practices in the not too distant future. For the immediate future, aluminum, magnesium and stainless steel are due to replace much of what formerly would have been built of wood. These new and better materials allow a freedom in designing hull shape which cannot immediately be grasped due to the momentum of traditional thinking along the lines dictated by slower craft of wood construction.

The hogged sheer is an engineering as well as an esthetic necessity for planing hulls. It is structurally sound and materially reduces beam windage, both prime considerations. The old-fashioned stem must be replaced by the now possible rounded and more buoyant bow. Sharp corners from sides to transom must give way to stronger and more graceful curves. Topside flare, for so many years considered beautiful, although largely without real utilitarian basis, may give way to stronger convex shaped sides. The airplane fuselage is a worthy study for the progressive designer, provided he keeps in mind that even with new materials, a practical boat must have decks that can be walked around on, sides that will not dent at a float or pier, and fittings, such as flag staffs, that will support a man's full weight when the boat is giving its wildest performance.

Materials and methods of construction are covered more fully in a subsequent chapter, but regardless of their type, it is essential that the planing hull shall have the toughness to withstand punishment far beyond that ever experienced by slower craft.

Honesty of Design

There is a temptation when dealing with a subject as modern as the planing hull, for the inexperienced designer to produce something "futuristic" in his attempt to streamline. The art of naval architecture can be achieved only by those lines, mass and proportions which are



LEE SHORES OF DESIGN

Figure 22

consistent with the structure and its intended use. Anthony Fokker, master of airplane design, was sadly disillusioned when he produced the expensive "Q.E.D." Her required design masterfully executed by Starling Burgess, she was streamlined to the extent of sliding covers

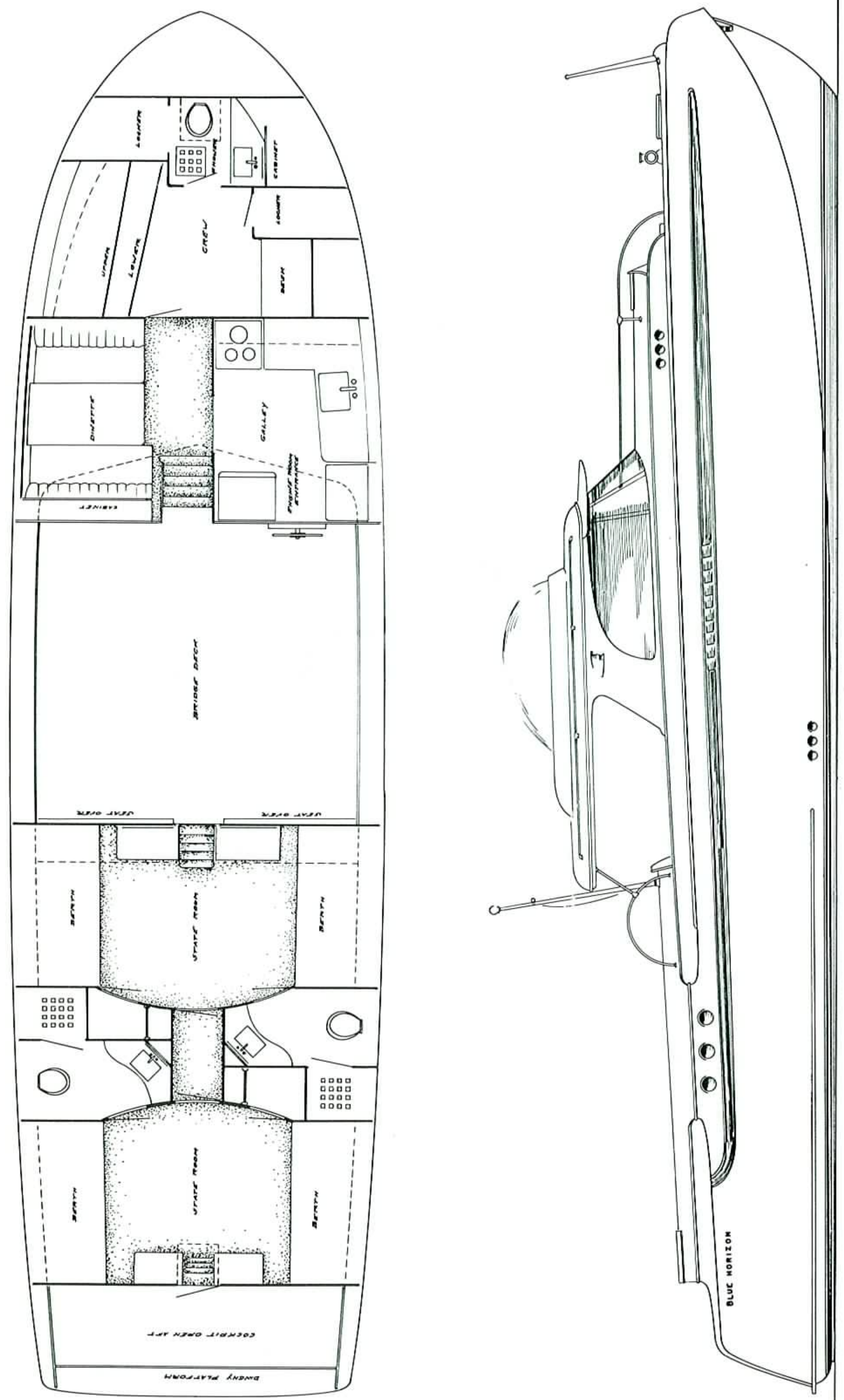


Figure 23

over the weather decks. However, she frustrated utilitarian hopes and, unmourned, was burned soon after her trials.

Masts bent like a sapling in a gale are structurally unsound and are therefore hideous to the trained eye. In a similar category are tear-drop ports, fake lines and all artificialities of design. Symmetry and balance of the mass silhouette are practical as well as artistic necessities, and are achieved only by the correct engineering approach. Any resort to fad defeats the elemental spirit of power and speed which sound design alone can achieve.

The difficulty of artistic design is almost inversely proportional to the size of the vessel. To achieve speed, space and seaworthiness in a small hull and then expect beauty also, is a large order. However, true beauty is but a reflection of basic engineering qualities. The profile and arrangement plans of a 50-foot cruiser shown in Figure 23 would seem to indicate that even in the smaller seagoing sizes, beauty is a natural outcome of balanced design.

Specific Requirements

The required details for which the designer is responsible will vary according to individual owners, but certain common-sense fundamentals should be included in all boats. For example, the matter of natural ventilation, along the lines put forth in the section under ventilation, needs the most careful thought from the very beginning. There must be no taking of chances with a danger as real as the explosions of bilge gases.

For pleasure cruisers, the comfort standards of typical sailboats have become archaic. Silence, dryness and ease of motion are inherent in good design but cannot be successfully engineered as afterthoughts. Even the details of comfort such as ample water, window sills that can be seen over from seats, anchors that need not be man-handled, airy quarters, topside tightness and dryness, need thought from the beginning.

Chapter 6

LINES

There is a chapter in the old Herreshoff legend which reveals that the Wizard of Bristol modeled the lines of a new one-design class after the underbody of a frozen mackerel. The resulting boats were excellent "ghosters" but not unusually fast in stronger airs. The line of flow, or run, was streamlined for the least resistance only at very low expenditures of power. To absorb and use effectively the high powers customary today requires a geometrical bottom development of the utmost refinement. While hull lines are fundamental in the overall performance of any craft, the performance of planing hulls is truly a magnification of every characteristic represented in the lines.

It is the shape of the plane which limits not only speed potential and riding qualities at sea, but also controls the vital fundamental of useful lift. In the planing hull, all other qualities shaped by the lines exist largely as accessories to the plane's major function of lift. Any shape detrimental to maximum lift at sea has no place in the lines of a seagoing planing hull. A particular form may even add speed under some conditions, but if it detracts from lift under seagoing conditions, it must be discovered and eliminated. Therefore, each part of the plane must not only be designed for its major function but must be so modified as to contribute toward lift. The forefoot must smooth the way; the forward chines must maintain the hull in a buoyant, planing position; the after chines and total deadrise must allow the plane to take a positive bank while steering; the keel and lateral plane must give directional stability; and all of these must be accomplished in a manner consistent with obtaining the greatest useful lift from the maximum plane area.

For the plane itself, that is, for that portion of the bottom aft of midships where the real work of lifting must be done, a constant section is theoretically ideal. With constant section all buttocks or running lines are straight and parallel with each other. Without parallelism among all major running lines, the plane cannot deliver uniform lift per square foot of area. With the parallel running lines of constant section, the lift is nearly uniform entirely across any given transverse section. In a fore-and-aft direction, the lift tends to increase uniformly toward the leading edge.

But to operate at sea, the constant section plane must be modified with an entrance shaped to surmount and flatten waves. A properly designed forebody makes its entrance easily and without jar; it is shock absorbing rather than shock creating. On the other hand, a high degree of buoyancy must be retained in the forebody to permit a riding over rather than into or through steep waves.

Forebody Sections

In Figure 24 are shown the four general shapes of forefoot in use today. *A* is the concave shape highly successful for small racing craft which never leave calm waters. The degree of concavity varies from the extreme shown to the type which is almost flat, but in general, any forefoot of this classification can be considered as having good wave-flattening qualities and therefore tending toward a dry boat. With wide beam at the chine, this type is highly buoyant, riding over rather than through small waves. However, in steep ocean waves, the blunt bow of a wide chine frequently renders impossible the maintenance of planing speed. If the wide forward chine is also low, that is, close to the waterline, an unfortunate sponson effect results which can be subject to extreme pounding. This pounding tends to soften in direct proportion to deadrise, or height of chine above the water. The Huckins "Quadraconic" is an example of good refinement of the concave type. Its forward chines close in to moderate beam while deadrise is fairly high. The Higgins motor torpedo boat combines unusually high chines and deep forefoot with slight concavity. Its soft riding is indeed a fact, but the extent to which spray is tossed up and tumbled on deck sometimes approaches the limit of human endurance. In general, concave forward sections focus, or pocket, the impact of waves in inverse proportion to the deadrise, and any hull so shaped has a tendency in some degree to the characteristic heavy, jarring impact of the "belly-flop" dive.

B illustrates the straight deadrise type most commonly designed in the mistaken idea that it is cheap to construct. Hulls designed to allow the use of flat plywood sheets are frequently drawn up with this straight deadrise. That flat sheets will wrap against straight frames athwartship is a fallacy. The elements of a sheet so wrapped must be straight, it is true, but the elements are not normal to the keel; they fan out from some focal point or very short curve, and the bottom edge of

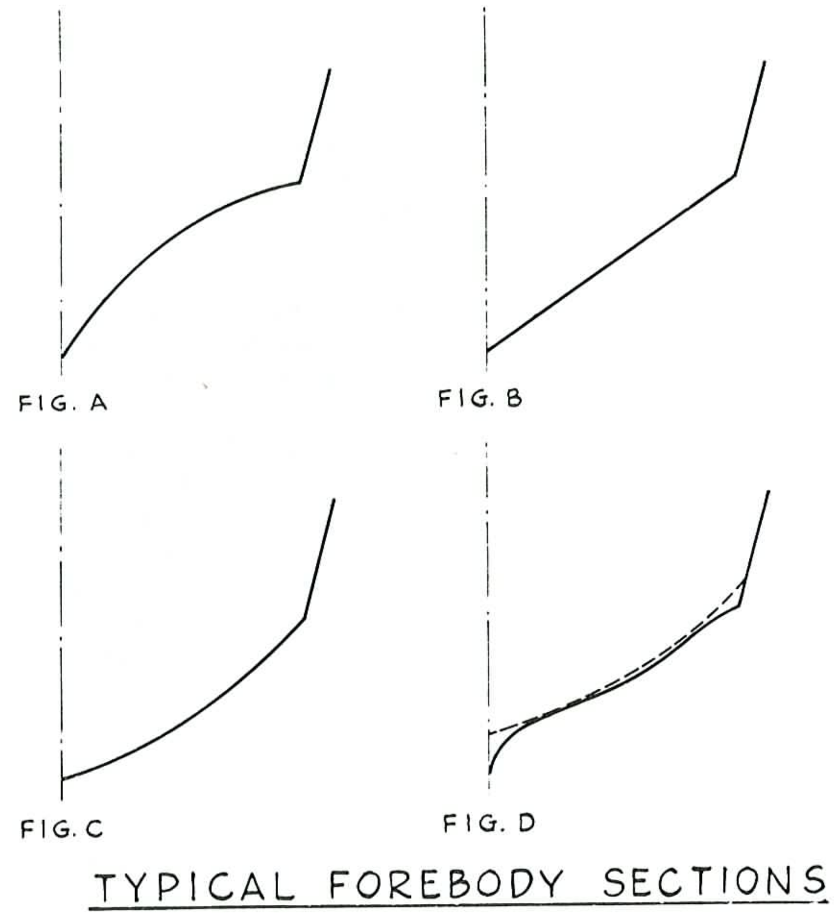


FIGURE 24

any transverse frame, particularly forward of midships, will be convex as shown in *C*.

However, when the straight deadrise type is kept straight by planking in the standard manner, its rough-water performance is frequently superior to that of concave sections of similar chine beam and height. In general, the deadrise should be about 25 degrees or more in that por-

tion of the forefoot likely to be tossed clear of the water. On the other hand, such deadrise, increasing as it must near the bow, results in a wet boat, and unless an effective, separate spray guard is fitted, its softer riding tendency is largely outweighed. Appearance also is somewhat against the boxlike straight deadrise. In its favor is the fact that waves cannot strike the slight overall convexity of its surface with the impact to which concave sections are subject.

The convex type shown in *C* is largely an outgrowth of the round bottom displacement hull form, but with greater beam at the chine level to give the necessary quick buoyancy. The shape has two primary

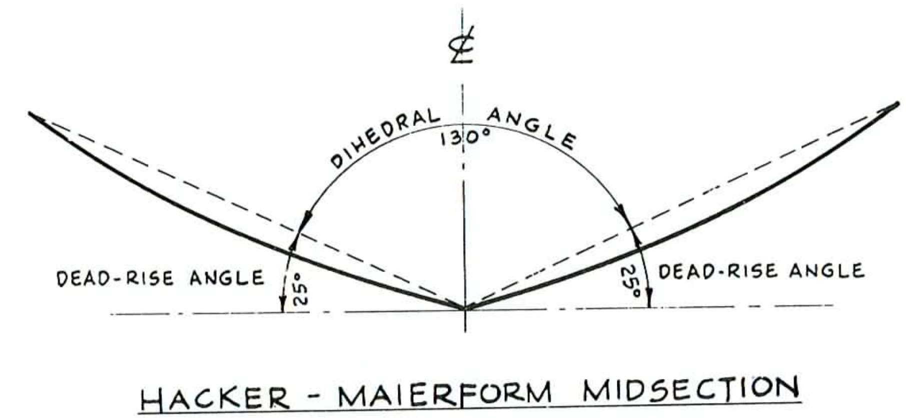


FIGURE 25

advantages. First, it presents a form inherently strong and resistant to impact. Second, it tends to dissipate and reduce the wave striking force. The fault of the convex shape is that the bow wave is not flattened outward, but instead tends to roll up and tumble entirely on deck. This fault can be minimized by the addition of spray guards but such afterthought appendages are forever being damaged by floating debris. Satisfactory spray control must be built into the original design.

However, the potential superiority of the convex shape in strength and soft riding is so basic that it is worth while to incorporate the general principle in some variant which will also maintain reasonable dryness on deck. Such a refinement is shown in *D* of Figure 24.

Tests conducted by John L. Hacker, who pioneered the wide chine, convex forefoot, have led to the conclusion that a deadrise, as shown in Figure 25, of 25 degrees or more as measured on a straight line from keel to chine, effectively changes impact to a snubbed deceleration. With this resultant dihedral angle of 130 degrees, a convex section is entirely free of the impact type of pounding. The more nearly such a section is moved aft toward midships, the softer the riding becomes. However, as this section is moved aft, the sections further forward acquire progressively more deadrise, of but little advantage in softer riding and of far less effectiveness in wave flattening. Although the

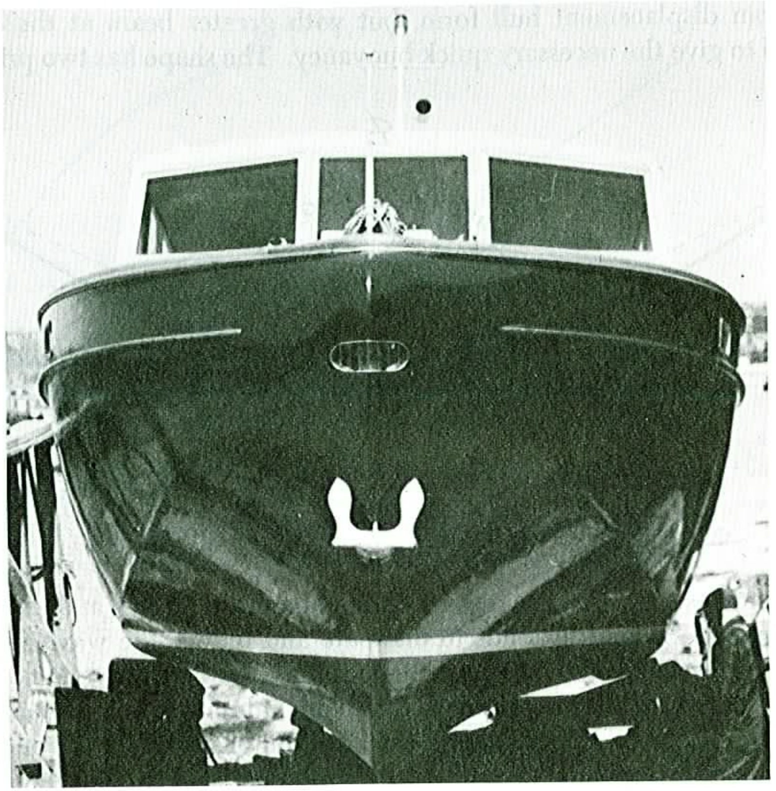


FIGURE 26

seagoing hull at high speed may frequently extend its whole forward half clear of the water in even moderately heavy weather, the moment of the striking force upon returning to the water decreases in proportion to its distance from the bow, and it is hardly worth while to sacrifice dryness forward in order to achieve a full 130-degree dihedral at midships. In Mr. Hacker's latest designs the forefoot section around 25 per cent aft of the stem shows the convex 130-degree dihedral.

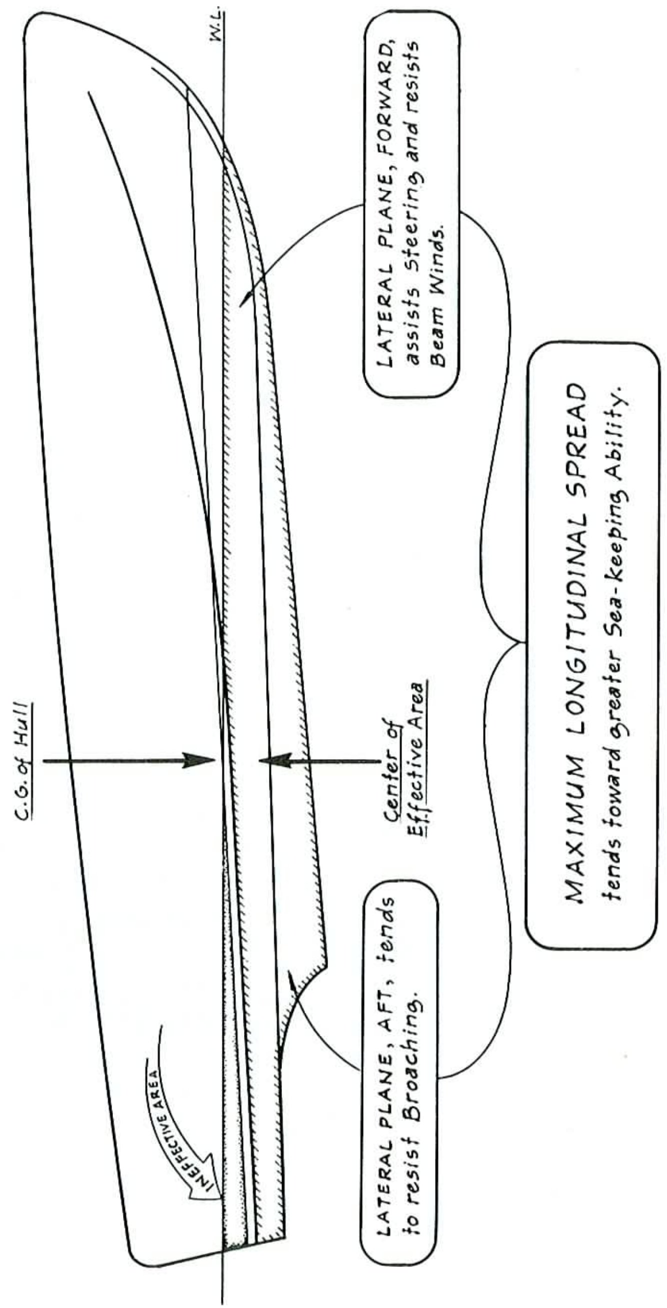
In *D* of Figure 24 is shown an essentially convex forefoot section, as indicated by the broken line. However, as a spray guard, a pronounced outward hook at the chine has been incorporated. In addition, there is included a fin section at the keel to lend the more positive lateral plane of the concave type. With these obvious refinements, the soft riding convex forefoot is given very good wave-flattening qualities and excellent directional stability against beam winds. Figure 26 shows the non-pounding forefoot with such a built-in spray guard.

A further advantage of the generally convex type is the fact, which becomes apparent when drawing up a set of constant section lines, that this is the normal section as faired out to a soft riding forefoot from an after plane free from major warpage. The resultant ease of the entire hull shape is characteristic of all those hulls whose efficient speeds not only cover fairly high brackets but also extend down into low ranges where hulls with harsh angles are out of their element and become awkward and cranky. For extreme speeds, the shape does present a very slight additional wetted surface due to the more consistent immersion of the forefoot. However, up to speed-beam ratios of 3.5 the added resistance is negligible and speeds above this ratio are usually the mark of a hull too small for a seagoing job.

Lateral Plane

A distinguishing characteristic of the hull suited to blue-water cruising is the more generous lateral plane, distributed fore and aft throughout the underbody. The important function at sea of lateral plane is twofold: first is its tendency toward maintenance of directional stability under difficult conditions. Second is its essential reaction in connection with steering. Adequate realization of both functions is of primary importance.

Implicit in the meaning of directional stability is the fact that the hull shall tend to hold a straight course not only in confused seas but also in strong winds broad off the bow. An overall high deadrise together with some keel area is usually associated with the hull which, in heavy weather, seems to be running on a track. Effective lateral plane is that portion of the underwater profile which is fully immersed in solid water with the hull in planing position. Due to the normal lift of a planing hull, a considerable profile area of the forefoot rises above solid water and reduces the lateral plane forward. At the same time, the wetted area of the hull immediately above the after chines tends to



GENERAL PRINCIPLE OF LATERAL PLANE

FIGURE 27

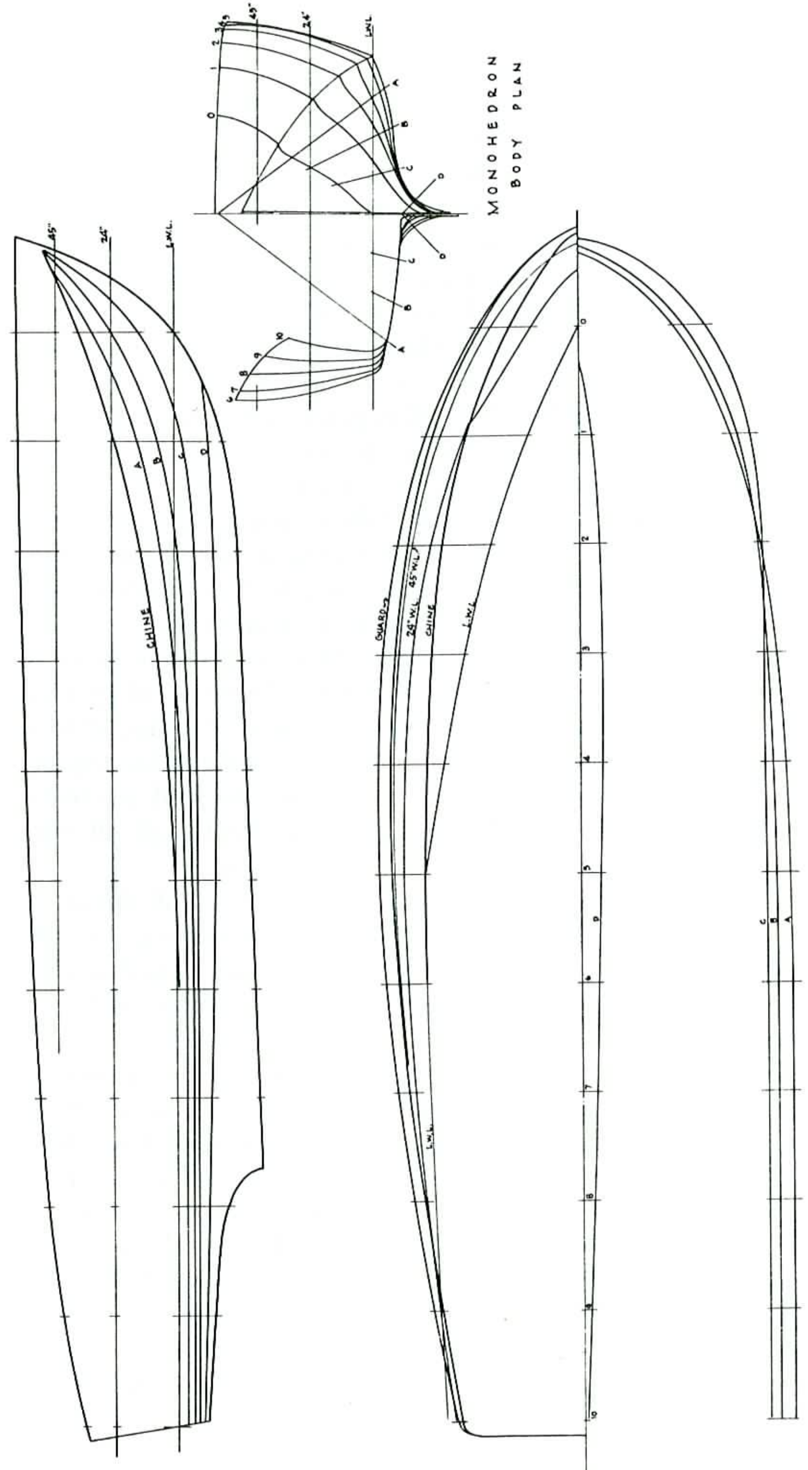
become relieved of pressure from solid water. The usual fore and aft reduction of effective lateral plane is thus approximately equivalent.

Figure 27 is a visualization of the hull in planing position, shaded to show effective and ineffective areas of lateral plane. The net remaining lateral plane and its distribution determine much of the hull's handling and sea-keeping qualities. Here it is shown that the center of effective lateral plane area should coincide vertically with the boat's center of gravity and also that the whole area should have maximum longitudinal distribution. If the center of lateral plane area is forward of the center of weights, the hull is potentially unbalanced and will tend to broach in direct proportion to the degree of unbalance. The excess of area forward of the center of gravity tends to create a pivot about which the stern, riding on a following sea, may swing into the dreaded yaw. On the other hand, an excess of lateral area aft of the hull's center of gravity leaves the bow relatively unstable and at the same time detracts from normal steering responsiveness.

However, if a theoretical lateral plane could be entirely localized in a single, concentrated area of fin directly below the center of gravity, somewhat after the fashion of the racing hydroplane, steering, at least in smooth water, might be excellent. But because of the inadequate longitudinal spread of the effective lateral plane, the hull would be practically helpless in heavy weather. The bow would fall off with beam winds and the stern would yaw in following seas.

The error has sometimes been made in past planing hull design of narrowing the afterbody in an attempt to combat broaching. In spite of demonstrated ineffectiveness, the custom has persisted, perhaps because of an understandable lack of theoretical information on the net effectiveness of lateral plane.

An increase in lateral plane aft of midships achieved by narrowing in the chines is more apparent than real. The resultant increased draft at rest, falls largely into one of the areas of reduced lateral plane effectiveness shown in Figure 27 when planing. The failure to maintain adequate waterline beam aft of midships is a major cause of broaching in that the essential lateral plane aft is relieved of hydraulic pressure. Furthermore, narrow sterns in stepless bottoms tend to squat, and this in turn reduces the longitudinal spread of the lateral plane. The combination of results due to narrow sterns can be directly opposite to the popular conception of the cause of broaching. Actually, the wide stern,



40 FT. MONOHEDRON CRUISER LINES

FIGURE 28

if greater area of lateral plane aft of midships remains, is less subject to yaw.

Intelligent design approach to the problem begins with study of the lateral plane normally provided by deadrise alone when the hull is in planing position. This basic area is beneficial to seagoing performance in direct proportion to its extent, and may usually be substantially increased and balanced by the proper design of a docking keel. The total of the effective lateral area so provided should be distributed and balanced as shown in Figure 27. The optimum net area of lateral plane, that is, the effective area after deductions for lift forward and low pressure above the after chines, appears to be in the neighborhood of 23 per cent of the area of horizontal waterplane at the still-water load line. More than this appears to be of decreasing value, but less area definitely accounts for a proportionate reduction in those seagoing qualities derived from the reaction of lateral plane.

Monohedron Lines

The word *monohedron* is not intended to indicate a bottom shape peculiar to any one designer nor exclusive with any one builder. Rather, it is a theoretical characteristic incorporated in greater or lesser degree in all planing bottoms. Its derivation is from *hedron*, meaning a geometrical figure having any number of planes. Since the theoretically ideal shape for planing on the water surface is a figure of constant section all in one plane or with monohedral angular characteristics, *monohedron* becomes a logical term.

In order that a plane shall have as nearly perfect monohedral relationship to the water as possible, it is necessary that a maximum of its running lines shall be straight and parallel with each other. Referring to Figure 28, it is evident that all sections, or station lines, of the afterplane area between diagonals *A* and *C*, coincide throughout their major lengths. Parallelism among station lines on the body plan indicates similar parallelism among the running lines on the profile. The area inboard of diagonal *C*, being a submerged body, may be hydrofoiled for least resistance as indicated by the shape of diagonal *D*. The docking keel thus formed has, by its hydrofoiled running lines, transformed an area of potential warpage, and therefore of suction, into shape which, because of being submerged, adds only a proportionate skin friction to the total resistance.

In the profile is shown again the monohedron characteristic of parallel running lines. The diagonals have been plotted both normal to their own planes and also as viewed in profile, better to illustrate the true monohedral angular characteristic of the planing surface. The entire pressure surface of the plane and the whole keel appendage are therefore operating under conditions most conducive to constant lift

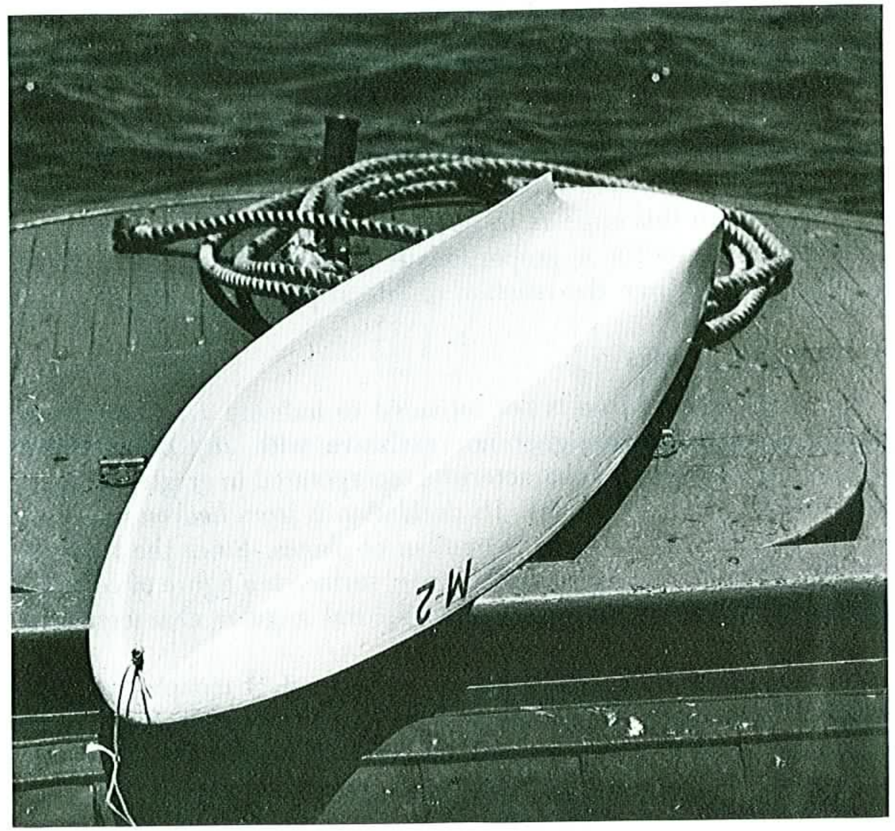


FIGURE 29

and least suction. Figure 29 is a photograph of the model built from these lines.

A general procedure for drawing monohedron lines is somewhat as follows: With aspect ratio and beam determined, draw the chine in plan view, keeping the widest beam between stations 5 and 6 with the beam at station 10 about 85 per cent of the maximum. In the case of

hulls with engines astern, the maximum chine beam may be at station 7.

Instead of plotting the running lines as buttocks, a truer picture may be obtained by plotting diagonals in profile. While the resultant running lines from station 6 to 10 will be identical, whether buttocks or diagonals, the diagonal viewed in profile recognizes to some extent the lateral displacement of water along the forward sections. A logical placement of diagonals is to establish on the body plan a point one half the total waterline beam above the still-water load line. This is the focal point of all diagonal planes.

Using the still-water load line as a base, mark off on the body plan its half beam at station 6 into four equal parts. Through these three points draw three diagonals from the single point on the center line. Stations 6 to 10 should form a common line the entire distance between diagonals *A* and *C* to develop a plane of minimum warpage.

To incorporate the shock-absorbing bow, draw the profile of the outboard diagonal. At the same time, draw the chine, crossing the still-water load line in the vicinity of station 5, and fairing up its forward end close under the deck.

Transom, or station 10, and stations 6, 5 and 2 are next sketched in on the body plan. Fairing of all diagonals or buttocks which are within the area of constant section, is done from the predetermined straight runs aft, through suitable points on station 2. At no point does the profile of one of these diagonals or buttocks ever drop below the height of its after end. From station 6 forward, the curves should display a constantly increasing increment of rise. When compromised by structural considerations, or for hulls designed expressly to operate only in the lower speed-beam ratios around 2, the area of constant section may safely be curtailed between the confines of chine and quarter-beam buttock. In this case, the profiles of diagonals and buttocks inboard of the more restricted constant section area will show at midships a slightly deeper draft than at their after ends. However, by maintaining some area of constant section along the outboard edges of the plane, the low resistance characteristics of a non-dragging chine are retained and planing speeds within the lower and more economical range will not be impaired. While there is, of course, no definite line of demarcation as to the exact extent of the constant section area, it can be readily appreciated that the more closely it approaches the keel, the higher is the hull's speed potentiality. But since the worst drag in the initial stages of planing occurs at the chines, it is important that parallelism among running lines be attained first along the outboard edges, and that this constant angle area be extended inboard in a proportion consistent with the proposed speed-beam ratio of the hull. Thus the tendency within the central area of the plane for suction to increase with speed is largely counteracted.

For topsides, wide flare forward and midships is of some advantage, although real wave deflection must be done below the chine. Carrying flare aft is a matter of individual preference. Where great deck space aft is useful, a flare is indicated. Otherwise the stronger and more beautiful tumble home may be incorporated. As for dryness of the after deck, a flare tends to fend off the spray and a tumble home tends to avoid the path of the spray. In practice, one is as miserably wet as the other unless the forefoot is of a wave-deflecting type.

The matters of keel, deadwood and skeg call for some compromise. They are highly desirable protective features, but do add excess resistance in the upper brackets of speed. This resistance, however, is never of sufficient magnitude to warrant the omission of an adequate docking keel or to sacrifice well-balanced lateral plane. Single-screw pleasure boats should always be fitted with full-length keel and skeg below the wheel. Twin screws cannot be so fully protected, and largely for this reason, the smaller pleasure craft should be single screw. Engines are usually less liable to failure than are small propellers mounted in twin-screw locations.

Monohedron Performance

The photographs in Figures 30 to 33 show clearly the characteristics to be expected in the most modern designs.

In Figure 30, the model is being towed at a speed corresponding to 25 knots. The forefoot is in contact with the water, wave-flattening is satisfactory, wake is clean and the hull is running at a trim of 2 degrees which is entirely due to lift, not from suction forcing a squat. The full freeboard aft indicates the absence of squat.

In Figure 31 the towing speed has been increased to correspond to 35 knots. The apparent planing angle has dropped to one degree and fifty minutes together with an overall increase in lift. There is a slight sternward shift of the center of this lift.

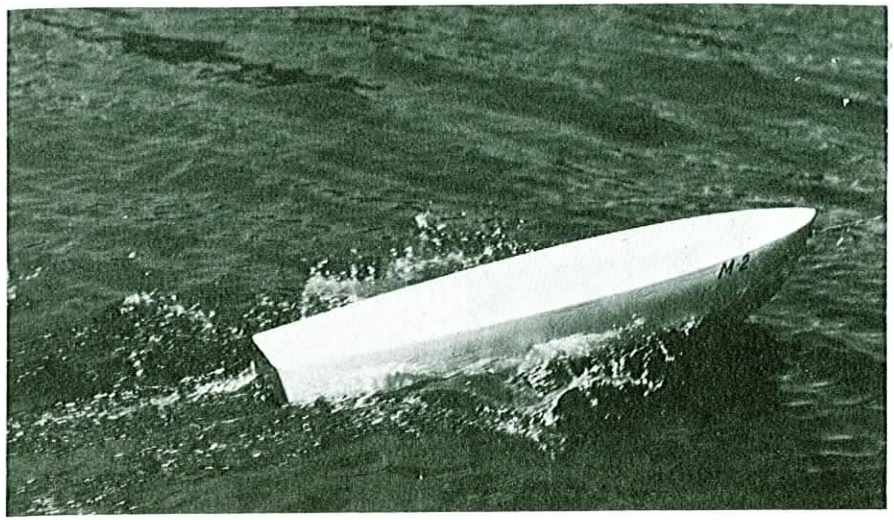


Figure 30

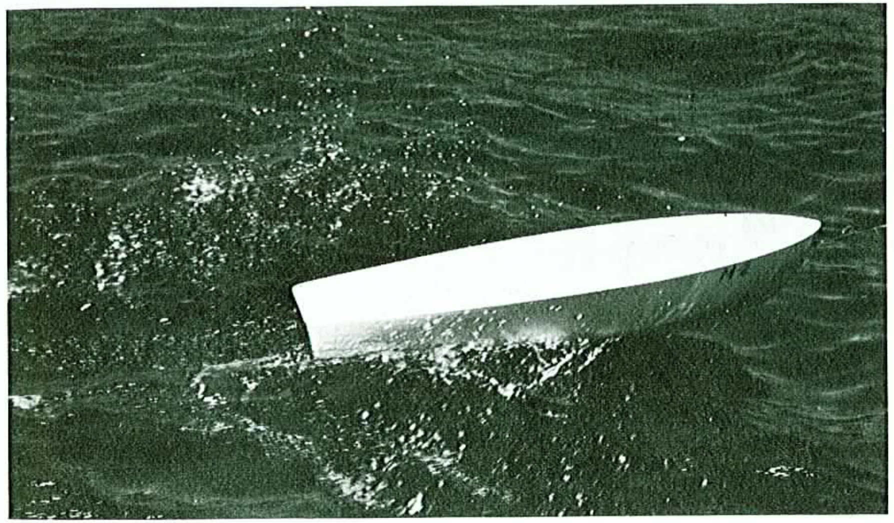


Figure 31

FIGURE 32

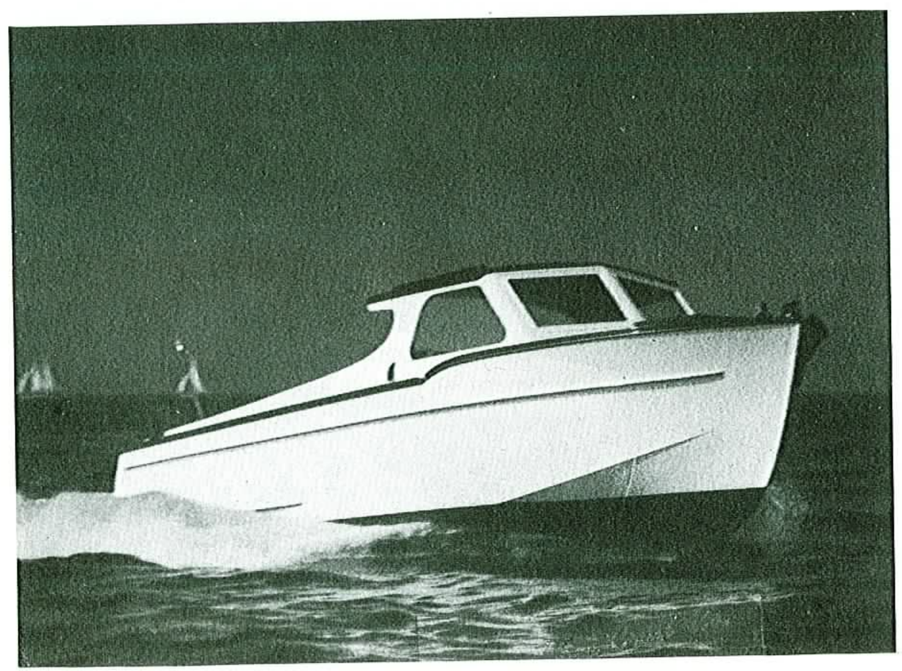
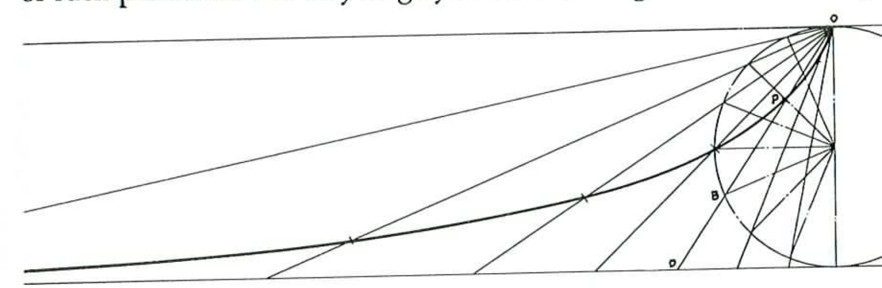


FIGURE 33

Figure 32 shows a small cruiser designed by Richard Cole running at 25 knots. It is interesting to note that the apparent planing angles remain substantially the same between model and cruiser in this size range. However, as length increases, the apparent planing angle must become less for reasons fully explained on Page 170. But regardless of how small or how large the cruiser, the ideal trim puts the forefoot in contact with the water while not appreciably reducing the freeboard aft. A hull does have actually less skin friction with its bow lifted clear of the water and will therefore make more speed, but only in calm water.

GENERATION OF LINES BY FORMULA

There have been serious attempts to generate hull lines by mathematical formula, but for practical purposes it is evident that the merit of such procedure lies very largely in the resulting similitude for dupli-



CISSOID CURVE

OP = BD

FIGURE 34

cation in other sizes according to whatever shape was first worked out, rather than in achieving any possible virtue inherent in a mathematical formula. The Huckins "Quadraconic" is a case in point, and is further discussed under the section of Developable Surfaces. These bottom lines represent much refinement through years of trial and error in the famous Fairform Flyers. By judicious application of logical equations to the development of the more important running lines of a known hull, good similitude is retained throughout a range of sizes, and assures predictable performance.

A typical formula for the generation of lines is one based on the cissoid curve, shown in Figure 34. It is undoubtedly a beautiful and

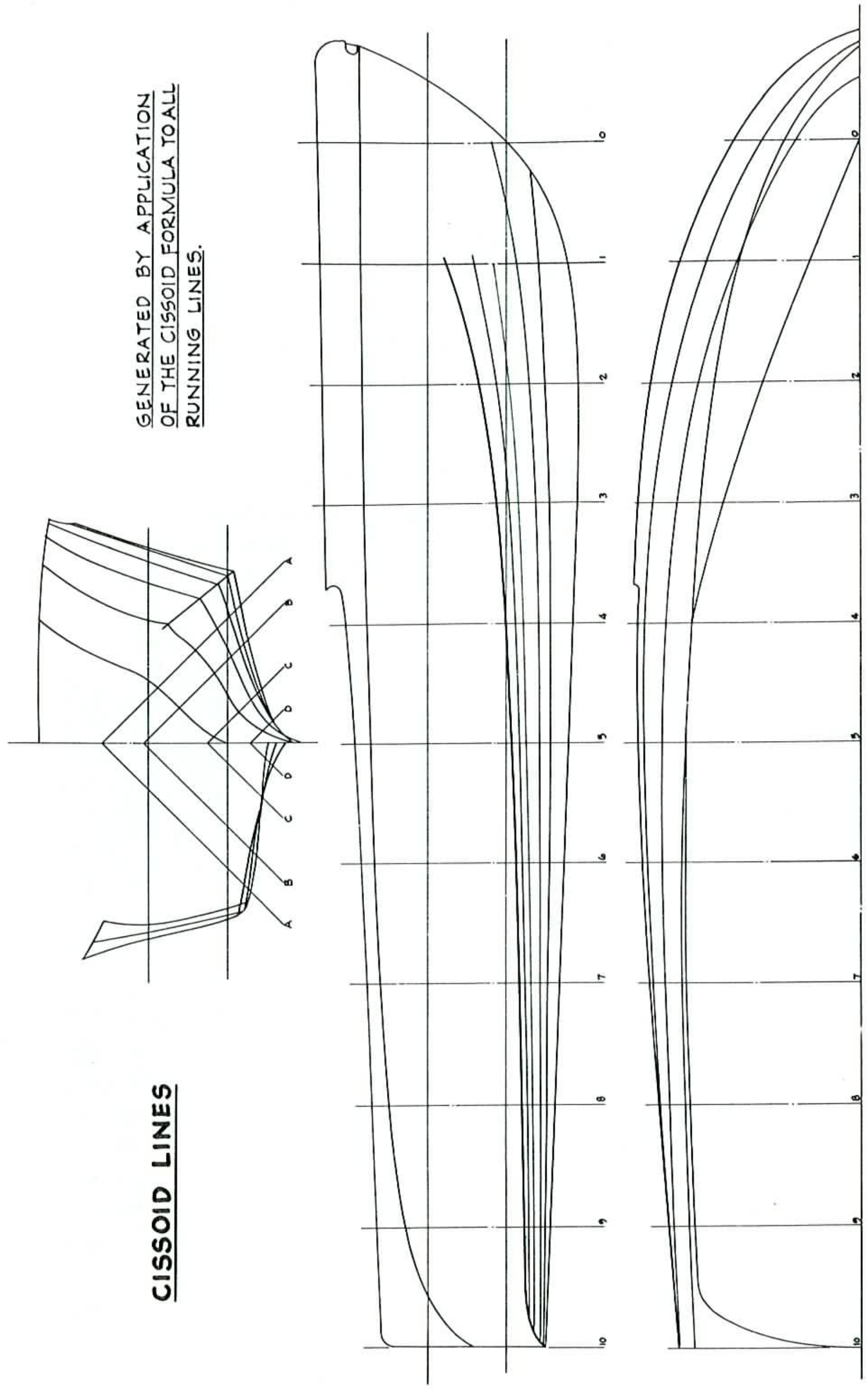


FIGURE 35

practical curve, and as such is a good model for basic running lines. By arbitrary application always in the same manner, hull sizes may be readily varied without change of essential characteristics. Figure 35 shows a set of lines so developed.

The order of procedure is somewhat as follows: Draw on a separate sheet of tracing cloth a cissoid whose length is the overall length of the proposed hull drawing, and whose spread or radius is developed from a circle having a diameter of 125 per cent of the maximum half beam at the chine.

Mark diagonals on the body plan at standardized locations and, locating the cissoid under the hull profile, trace a profile view of the diagonal. Obviously, the system must be based on arbitrary rules for placement of the cissoid curve. Once such rules are determined, subsequent hulls may be drawn with good similarity.

The cissoid lines shown in Figure 35 indicate a bottom plane of fair parallelism among running lines, a soft riding entrance and good wave deflection. But from the body plan, it is evident that an unnecessary plane warpage still exists which could have been faired out had the lines not been held to a rigid formula.

In Figure 36 are shown the essential lines of a Higgins PT. While not strictly developed by formula in regard to the overall shape, all transverse bottom sections are laid out from a characteristically similar curve. It is possible that this leads to certain minor economies in construction.

A good feature of the lines is the high forward profile of the chine. However, the excessive concavity of sections allows serious pounding just forward of midships when the going is rough. Aft of midships the plane warpage becomes so extreme that suction loads up to 8 tons are not uncommon, a full explanation of the poor trim and balance of these vessels.

Incidentally, in the Higgins patent, Number 417,690, the tendency of the bottom to create and carry a heavy suction load is acknowledged in these words: "It will be understood, of course, that there is no void beneath the hull, but that the vacuum is satisfied by the maintenance of solid water drawn up from the depths." In other words, so proud are its makers of lugging 8 tons of "solid water drawn up from the depths" that they patent their handicap. Sinbad the Sailor also had an old man of the sea.

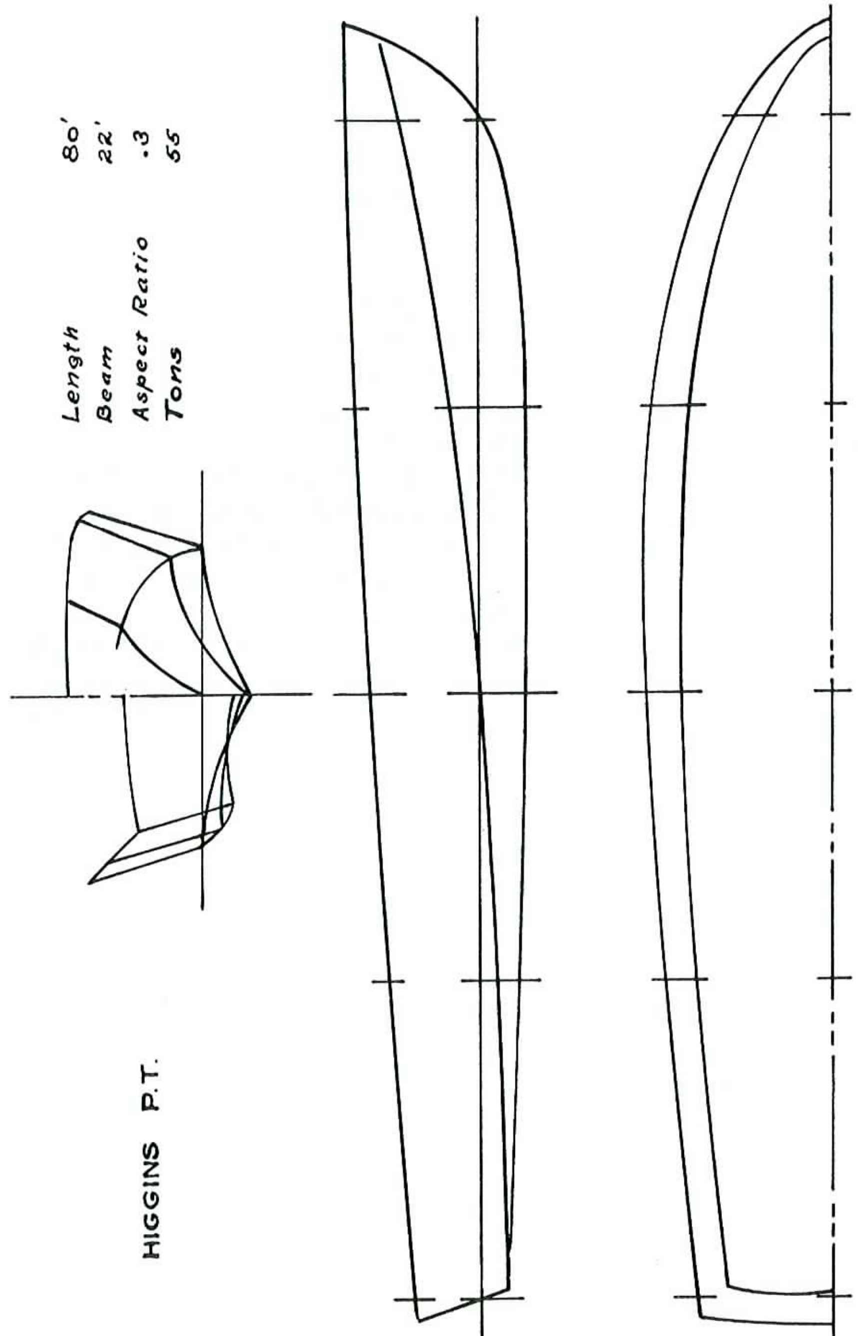


FIGURE 36

DEVELOPABLE SURFACES

When the bottom or topsides of a hull are to be sheathed with single sheets of plywood or other stiff material which cannot be bent into compound curves, a development of the surface in question must be made coincident with the lines drawing. Unfortunately, it is impossible to achieve the most efficient or beautiful lines with any developable surface for the very good reason that the elements of the developable surface are of necessity the inflexible straight lines of either a cylinder or a cone.

However, for smaller craft in the lower ranges of planing speed, the potential economy of construction may outweigh other losses. It is probable that up to speed-beam ratios of 2.2 the loss of efficiency may be negligible, although cost should be carefully analyzed since savings may be more apparent than real.

Figure 37 shows the method of development used to determine the lines of a small utility hull where full-length sheets of plywood were to be used for the shell. The bottom shape has been developed from a single cone, while topside configuration is from two cones. Any number of cones may be used; the method of development remains the same. The number of cones to be used depends entirely on the desired relationship of elements.

After preliminary visualization of the proposed elements, the exact location of the apex of the cone presents the only problem. For the bottom elements, an apex located as shown will produce running lines about as satisfactory as can be expected with any fixed formula. The procedure is as follows:

Draw the chine in plan and profile, and using it as a directrix, extend elements from each station intersection to the chosen apex on both plan and profile views. From the intersection of the various elements with station lines, plot the body plan. Fair up in the usual manner with diagonals and buttock lines.

For topsides which tumble home aft and flare forward, the double cone per side is a correct solution. Pick first some station line suitable for use as an element common to each cone. Station number 3 is ordinarily suitable, hence the apex of the cone generating the after tumble home will lie on station 3 above the profile, while the apex of the cone generating the forward flare will lie on station 3 below the hull. Draw a tentative station 3 on the body plan, using a straight line, and locate

FIGURE 37

the vertical height of each apex by fairing and checking the intersections of elements and sheer on plan and profile.

It is reasonable to incorporate developable surfaces in any low-priced hull, regardless of shell material, in the interests of maximum economy of construction. But where initial economy is not the prime consideration, compound curves should be introduced as may be needed to attain more ideal shapes.

Discussion of Lines

In Figure 38 are shown the lines of an older type vee-bottom hull of British design, complete with narrow, squatting stern and pounding forefoot. The downward slope of the chine and the upward trend of the keel mark the warped plane. Its running lines operate at varying angles, some overloaded and some dragging suction. The aspect ratio is on the order of .2, although the excessive squat induced by suction under the afterbody changes the trim by an additional 3 or 4 degrees and technically improves the aspect ratio while planing to perhaps .3, but does so at the expense of plane angle and effective lateral plane.

The stern has been narrowed or "tucked in" with the intention of reducing the tendency to broach. That broaching is not stopped has been frequently demonstrated and is a further indication that the deep chine aft can be fully as subject to broaching as a shallower and wider chine with unbalanced lateral plane. In addition, the tucked in chine reduces the average aspect ratio of the plane. That, in a hull with a high-riding forefoot, is necessarily more subject to the whimsy of wind and wave.

In this particular design, the stern has been tucked in to only 61 per cent of the maximum chine beam. The greatest beam is at station 3, abnormally far forward, a custom still occasionally persisting in the mistaken notion that hydrofoiling applies to a waterplane area. Obsolete also is the low chine forward, barely making an appearance above water, due no doubt to the prejudice of a sailboat era that there was something obscene about a visible chine. The combination of these two ideas gives this hull a bluntness in meeting steep head seas which renders the maintenance of planing speed an impossibility. Its quick buoyancy is of some advantage only when not in the least needed, that is, among waves too small to cause serious plunging. In addition to these very obvious faults, the low chine increases the average waterplane length and thus tends to decrease further an already

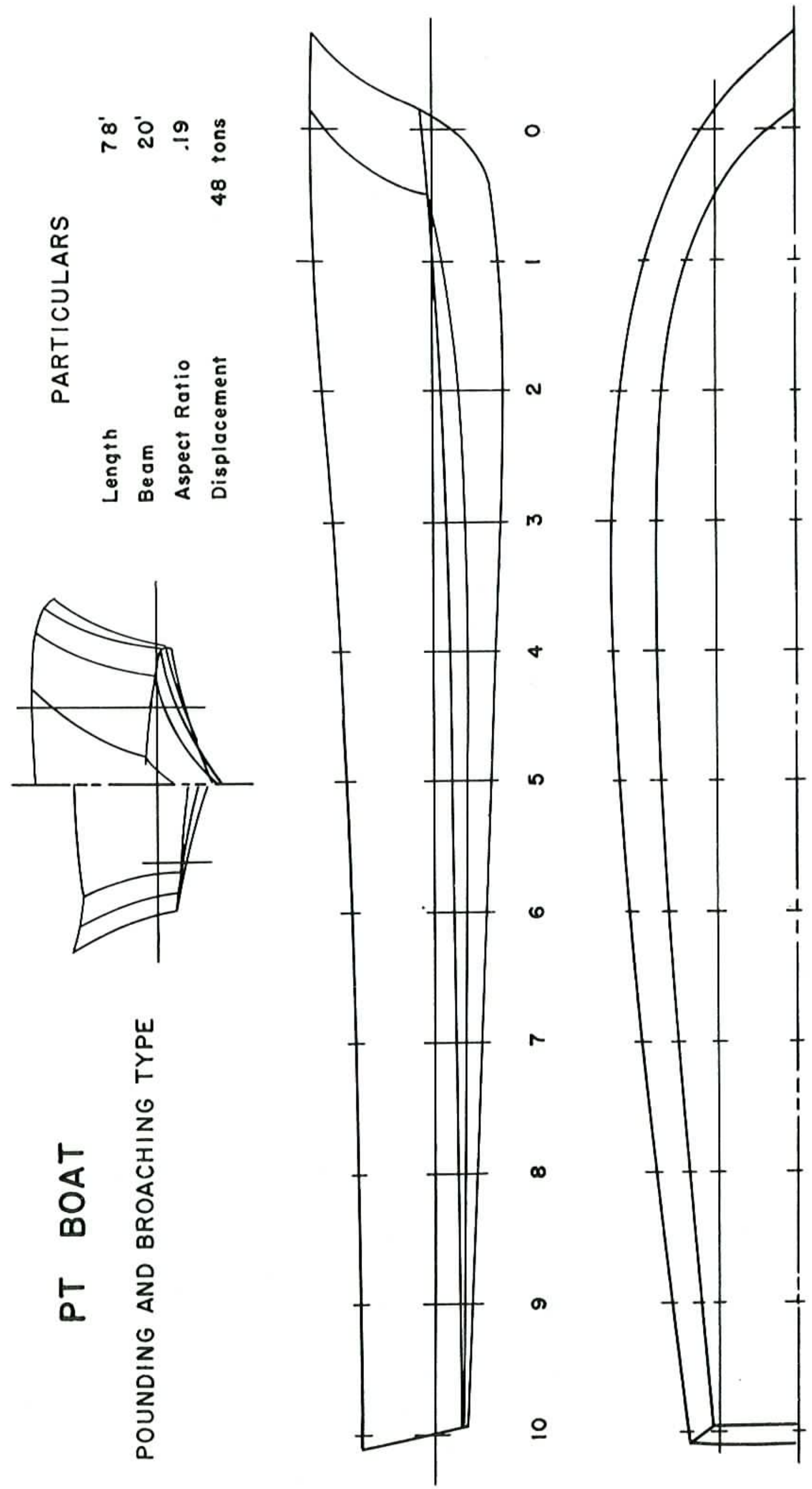


FIGURE 38

low aspect ratio. The deep draft at the stern, due partially to an excessively narrow after waterplane, fails to provide adequate lateral plane aft of midships, because the hull area above the after chines does not maintain pressure against solid water when planing. But it is from such misguided beginnings that the offshore hull has evolved.

In Figure 39 is shown a set of lines developed after long experience for a fleet of rumrunners. Many of these craft are still serving the Coast Guard and, although of concave section, after a dozen or more years they show no pounding tendency serious enough to loosen fastenings or start seams. The forefoot is very slightly concave with high deadrise. Designed around 1930 by Lord, the advantages of greater beam had not at the time been fully appreciated, but plane warpage was obviously reduced to a minimum for the concave forefoot type. Her widest water line is around station 6 and the chine beam at the transom is about 80 per cent of the maximum. The clean running of these hulls has inspired the design of many successful commuters and sport cruisers during the ensuing years. The rounded bilge aft avoids the needless drag of a chine that digs in, and also adds a smartness to steering which is noticeably absent with a harsh after chine. Properly loaded hulls built from these lines bank some 16 degrees inboard on fast turns, aided materially in this desirable quality by the fairly steep deadrise carried throughout the entire length. This deadrise is all highly effective lateral plane and is also of great benefit in ease of planing among waves. The combination of smart banking and directional stability in seagoing hulls has not received its proper share of design attention.

Plane warpage on this 60-footer is very slight, and up to speeds of 30 knots, the suction load developed is not appreciable. Above this range a moderate suction does make its appearance and at 40 knots is apparently sufficient to induce some squat. The amount of squat at this high speed is roughly equivalent to that which would be added at the 30-knot speed if a 2-ton load were to be placed in the after hold.

Were these lines widened to give a plane aspect ratio of over .3, say around 15 feet, the overall performance would compare favorably with the best hulls being produced today.

Of a similar type, but with the modern wider beam and higher useful lift potential, are the lines shown in Figure 40 of the Huckins Quadraconic. The greatest chine beam is just aft of midships and the transom beam is 87 per cent of maximum. That this great width aft does

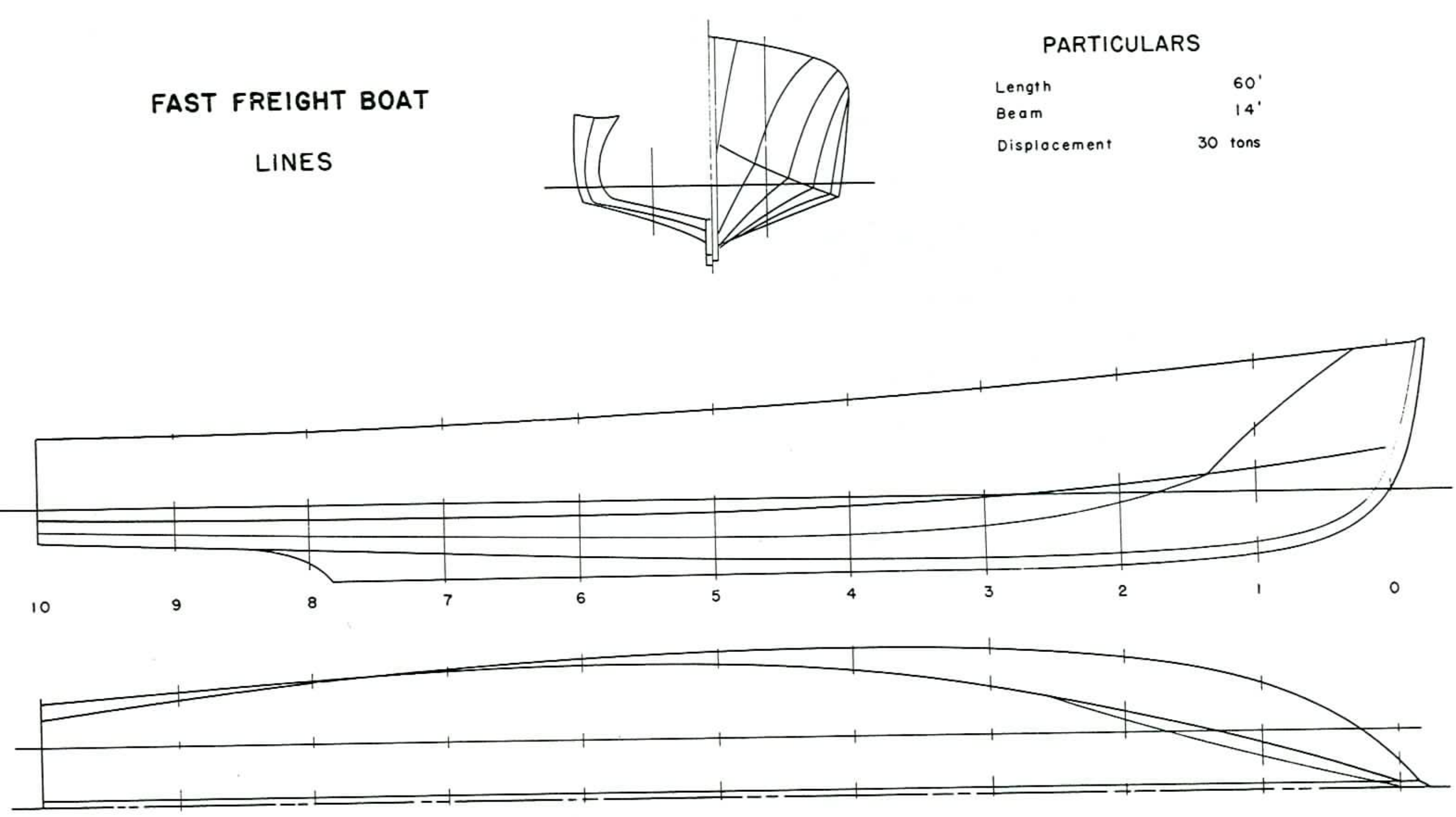


FIGURE 39

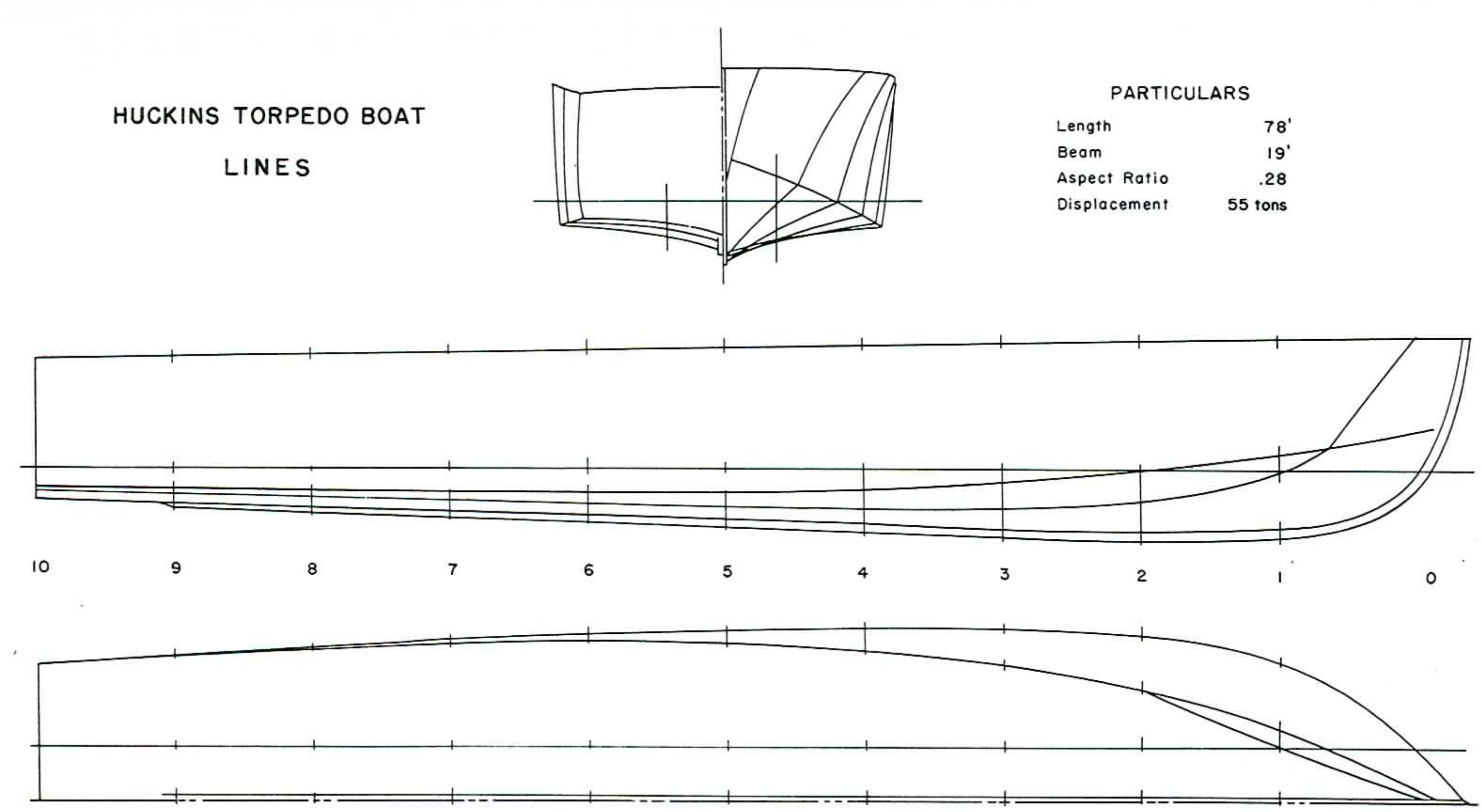


FIGURE 40

not necessarily cause unmanageable broaching has been demonstrated. The absence of excessive resistance forward and a balanced underwater lateral plane tend to maintain good directional stability, although additional deadrise would be a distinct improvement in this connection and also give a greater inboard bank on turns. In profile, the buttock lines aft of midships are all in very close parallel relationship, and as a consequence lift distribution is fairly constant and a minimum planing angle is retained all the way up into the highest speed ranges. The Huckins forward chine is moderately high and, in planing position, comes out of water just forward of midships. This lift, together with its forefoot in contact with the water, is indicative of a hull potentially suited to offshore cruising.

The lines of one of the Navy's most successful planing hulls, the 63-foot Miami-built crash boat, show practically straight deadrise with very high chine forward, fairing into slight convexity. The buttock lines from midships aft are very nearly parallel and the plane aspect ratio is about .32. At planing speed the lift is unusually high and the marked deadrise all the way aft contributes to smart maneuverability. In general, these Miami boats are a very close approach to the monohedron ideal.

Figure 41 shows the body plan lines of a 13.8 meter cruiser powered by twin Diesels aft. In this case the Center of Buoyancy must be around 60% aft and the Center of Flotation at least 70% aft. To achieve this balance all possible excess buoyancy must be taken away from the forefoot. Sectional convexity forward must be replaced with steep deadrise from a deep keel. Concave sections sharpened by deepening the keel rather than by narrowing the chine beam ride softly and flatten the waves without sacrificing final buoyancy when the hull plunges into steep seas at high speed.

Figure 42 shows the essential lines of the 65-foot patrol boats designed for Caribbean duty, one of which is pictured on page 244. These vessels are powered with twin Diesels totaling 800 horsepower giving a top speed of 22 knots.

The forefoot is deep and slightly convex. High deadrise is maintained the full length of the hull with consequent good lateral and directional stability under rough sea conditions. That high deadrise is essential to good seakeeping ability has long been an accepted fundamental and when this high deadrise is carried all the way aft as a monohedron constant section, the resulting hull is both highly seaworthy

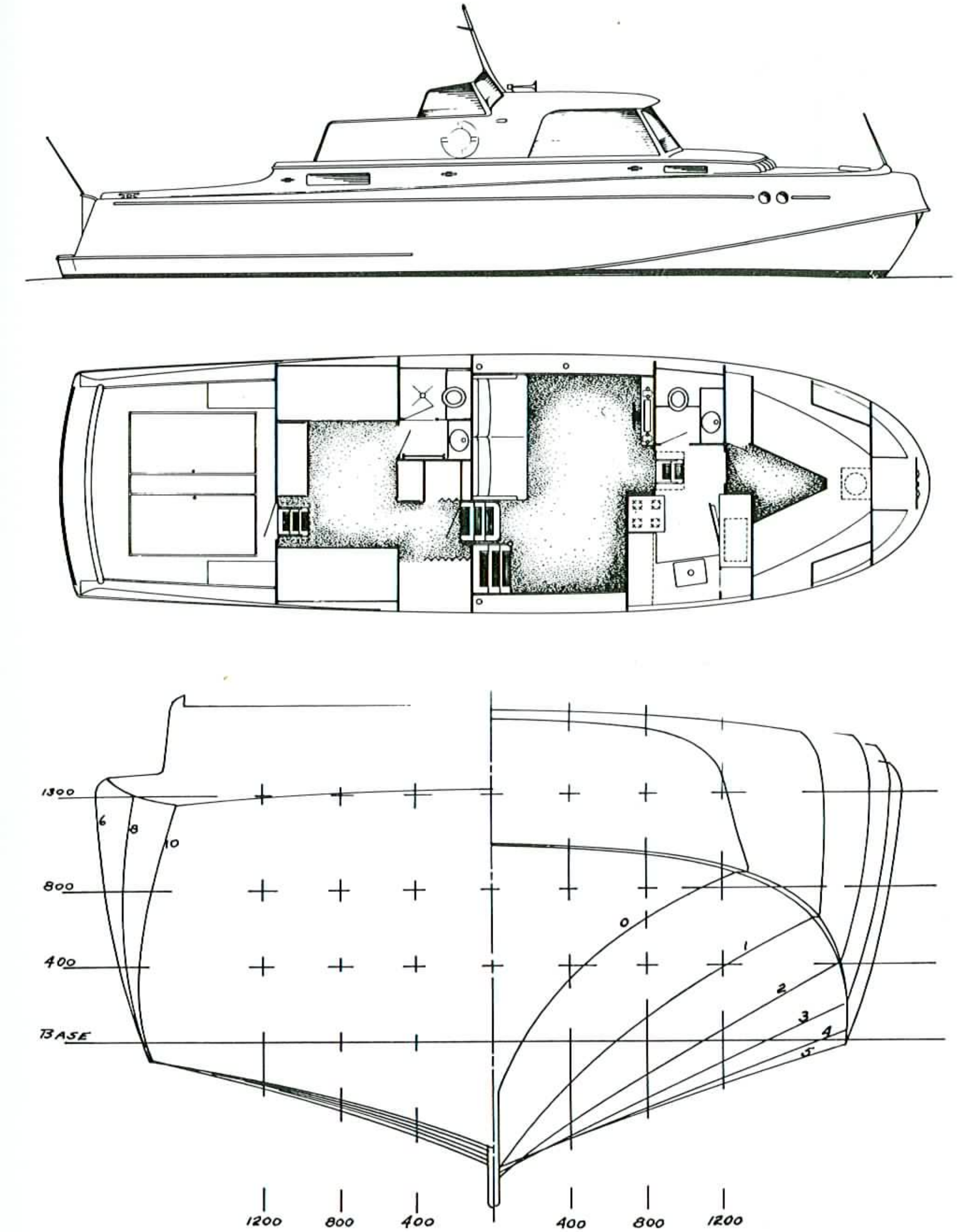


FIGURE 41

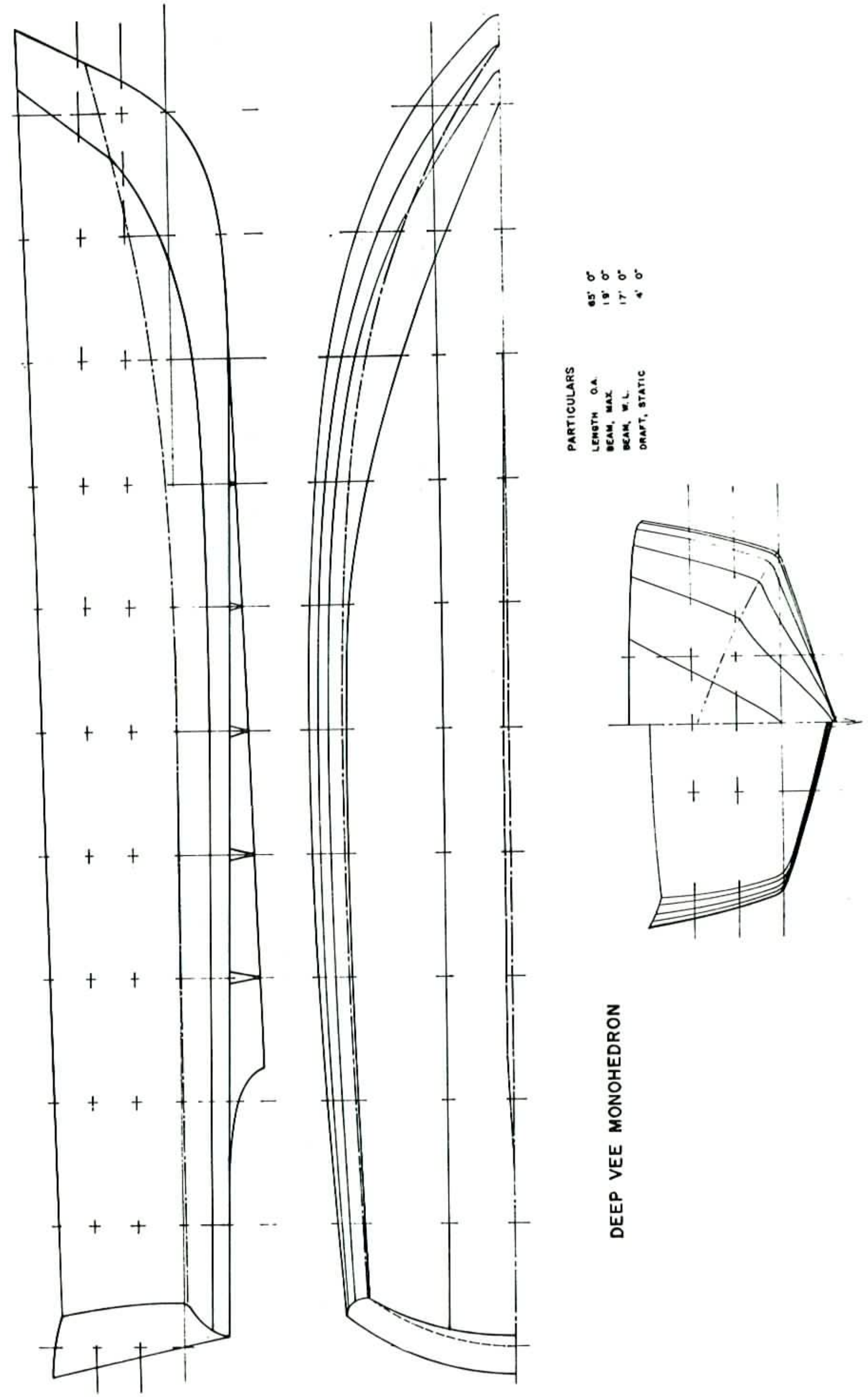


FIGURE 42

and capable of absorbing tremendous horsepower. It is a misconception to think of such a bottom with its straight run and deeply immersed transom as "fast" in the sense that it has low resistance. As a matter of fact, its resistance is rather high but its speed potential, which is very real, comes from its ability to absorb and use high horsepower. Without high power these so-called fast bottoms are actually sluggish.

Experiments with these hulls by Thornycroft of England indicate a very slight loss of top speed due to the rounded chines but a compensating improvement in sustained rough weather speed.

Figure 43 shows the essential lines of a 31-foot cruiser designed particularly for a single Diesel engine set down deep in the hull. By shaping the keel to what old-timers call a "planked down" effect, the engine can be set below flush hatches in a properly low cockpit. Fairing this keel out results in a really effective lateral plane deep enough forward so that yawing and leeway are practically eliminated and the tendency of most small cruisers to sail back and forth at their mooring is minimized. This great depth of forefoot so eases the deadrise that the concave sections tend to ride easily.

An interesting feature of these lines is the chine plus spray guard shape which has proved remarkably effective in quick buoyancy and spray flattening unequaled by appendage type spray guards.

Structural features such as this double chine are possible only in reinforced plastic construction. Afterthought spray guards, sometimes added in multiples, are usually the result of desperation in an attempt to make a wet boat somewhat drier. But such appendages are both noisy and vulnerable as well as actually adding to skin friction. Features which are built-in, on the other hand, need not compromise one quality to gain another.

In Figure 44 are shown the curves of sectional areas of the 60-foot vee bottom, its modern counterpart as developed by Huckins, and the theoretically ideal monohedron. While interesting from the standpoint of hydrostatic design as representative of underwater volumes at the still-water load line, they have little bearing on the displacement in planing position.

Effect of Steps

Steps may be either transverse, longitudinal or fan shaped. In any case a step is simply a clean break in the planing surface put there for the purpose of reducing skin friction. Like the transom, a step ends a

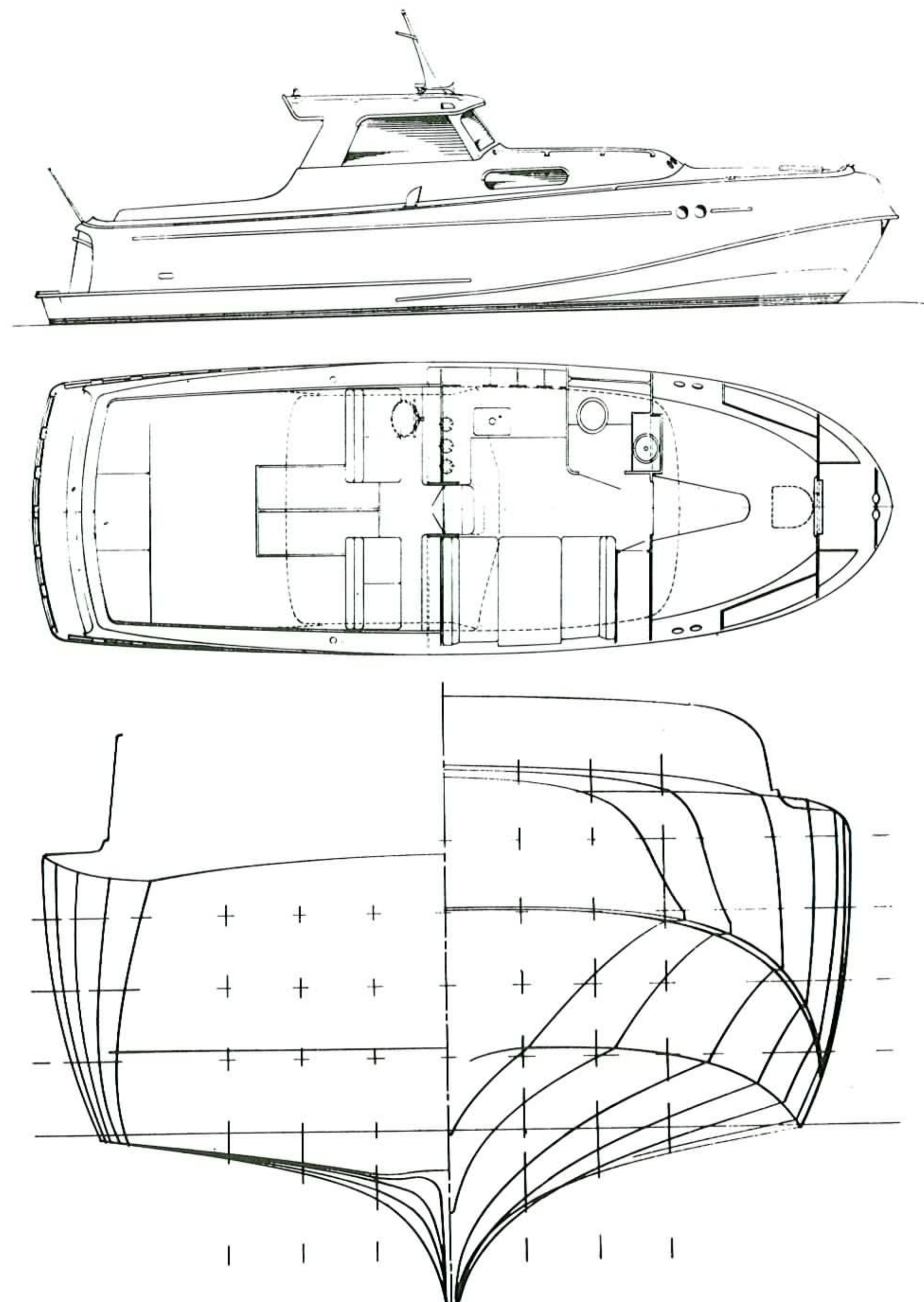


FIGURE 43

planing surface, and the theory is to cause the water to miss contact with the forward portion of the following plane. Substantial reduction in wetted surface can be achieved in this way under relatively smooth water conditions.

Varying numbers of transverse steps have been tried in seagoing hulls, from the early multi-step jobs designed by Fauber to the single-step British Coastal Motor Boats used in World War I and back to the multi-step hull of the smart performing Elco 80. The principle is three-fold: to cut down wetted surface, to break the suction under the after-body and to increase lift by means of a series of leading edges.

The term "longitudinal steps" has sometimes been mistakenly applied to appendages which are only spray guards. Particularly noticeable in this respect is the high deadrise or "deep vee" hull which, if lacking built-in spray deflection below the chine, resorts to multiple appendage type spray guards. Flat sheet plywood bottoms tend to assume a total convexity which requires added spray guards but planked hulls, and especially plastic hulls, if so equipped indicate that the original design proved excessively wet.

Lapstrake hulls do achieve a slight step effect together with some spray guard benefits but, in wood construction, their generally good performance is due largely to light weight. Perhaps the same may be said of plastic hulls where lapstrake is simulated for added strength in an otherwise delicate shell.

Steps are commonly vented by having the sides open to atmosphere above the waterline. The air can then rush in to fill the void which would otherwise be a partial vacuum causing extreme eddy-making resistance. Venting is also accomplished by introducing air ducts through the bottom of the boat. The suction in such pipes has been successfully applied to exhaust gases, acting as a powerful exhaust supercharger.

The increase of lift comes from the greatly improved aspect ratio of the individual sections of the plane. In effect, a multiple series of leading edges have been achieved. The suction load inherent in stepless hydroplanes can be entirely eliminated in a properly designed bottom incorporating fully vented steps. Skin friction of wetted surface can be reduced to as little as half that of stepless bottoms.

The disadvantages of steps are the costly construction necessary to overcome the structural weakness due to a sudden change in sectional area, and the unpredictable performance of steps in rough weather. Since continuity of structural elements of the plane is essential in order to withstand the severe longitudinal bending moments, steps should

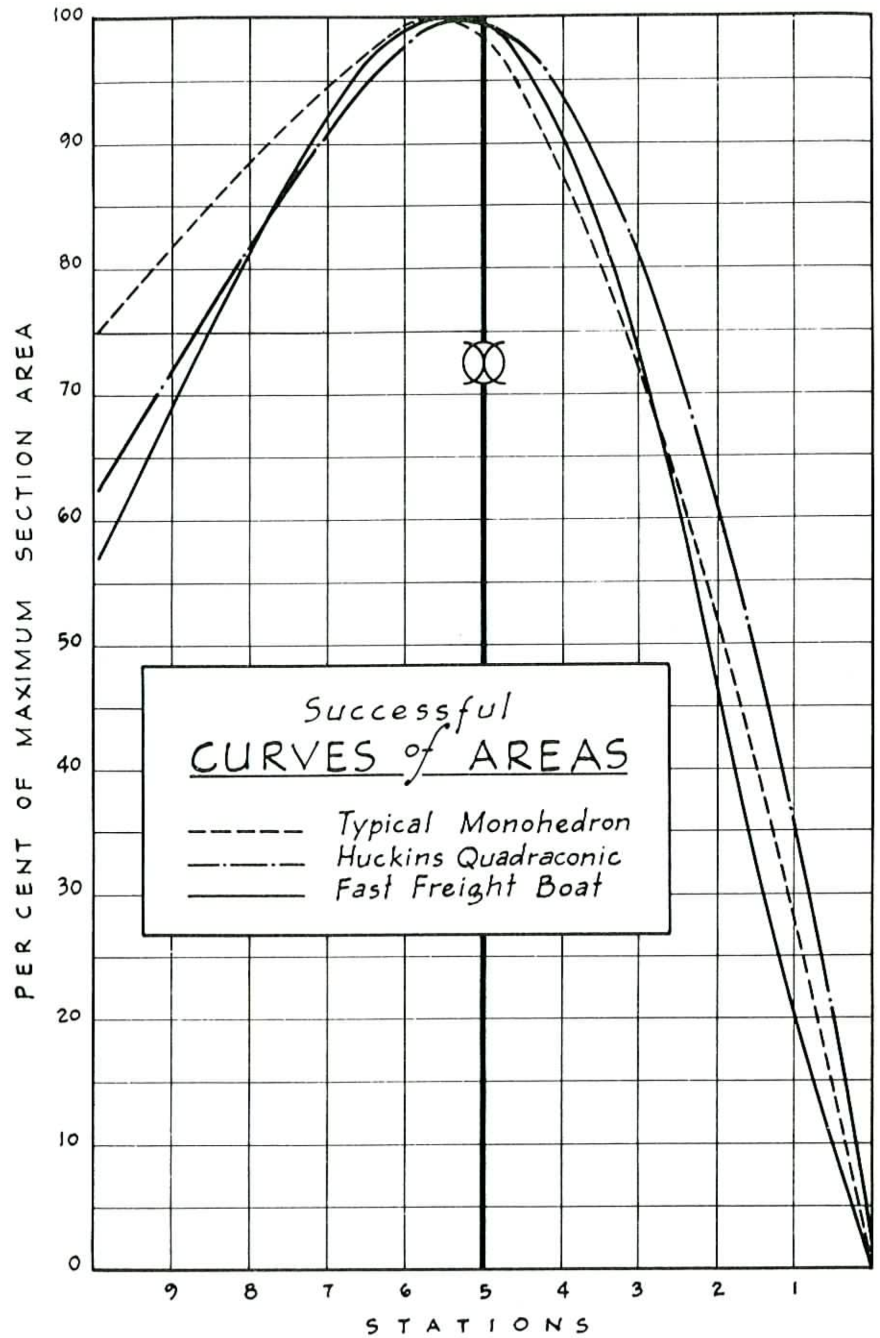


FIGURE 44

be added as an accessory to an otherwise continuous and properly shaped bottom. The framing and surface of these added steps must be strong enough to withstand impact and the dynamic forces set up by planing. Watertightness is not necessarily a factor, but if the steps are allowed to leak, fairly large openings should be provided to act as self bailers.

In rough water, wave action frequently piles solid water above the side vent and thus suddenly changes the action of the step from reducing resistance, to a suction which enormously increases resistance. If such a sudden change occurs along only one side at a time, as shown in the photograph at the bottom of Figure 45, the wildest kind of steering will result. Stoppage of vents along both sides; as in the middle picture, may cause loss of planing speed. In either case the hull tends toward crankiness and may even become unmanageable. Step venting through air or exhaust ducts is less susceptible to blocking off from wave action, but presents other difficulties of structure and arrangement.

The top photograph in Figure 45 illustrates the second great disadvantage under which stepped hulls labor at sea. Due to the high lift of each step as a new leading edge, the hull tends to resist pitching and therefore the bow is driven directly into the waves. The causes underlying this stubborn motion are fully explained in Chapter 10, in the section on Dynamic Moments. But in general, present-day sizes of stepped hulls cannot be considered as ocean-going vessels. However, as hull lengths more nearly approach the length of storm waves, say 200 or 300 feet, the high longitudinal moment of inertia induced by steps will be of relatively less importance, and the high lift capacity of the type will remain as an asset worthy of renewed consideration.

Below speed-beam ratios of 3.5 in craft up to 20 feet in beam, steps are not worth their cost and hazard. In larger craft the considerations begin to change, but in general, the average reduction in wetted surface, about 35 per cent better than for stepless bottoms, is not worth the disadvantages involved until speeds are to exceed 45 knots or thereabouts.

Step design starts with a normal type of stepless bottom. In the single transverse type, the forwardmost step is located about midships. Its depth at the keel should be about 1 inch per foot of chine beam up to 10 or 12 feet of beam. Wider hulls may reduce the proportionate depth slightly but never to less than ¾ inch depth per foot of beam.

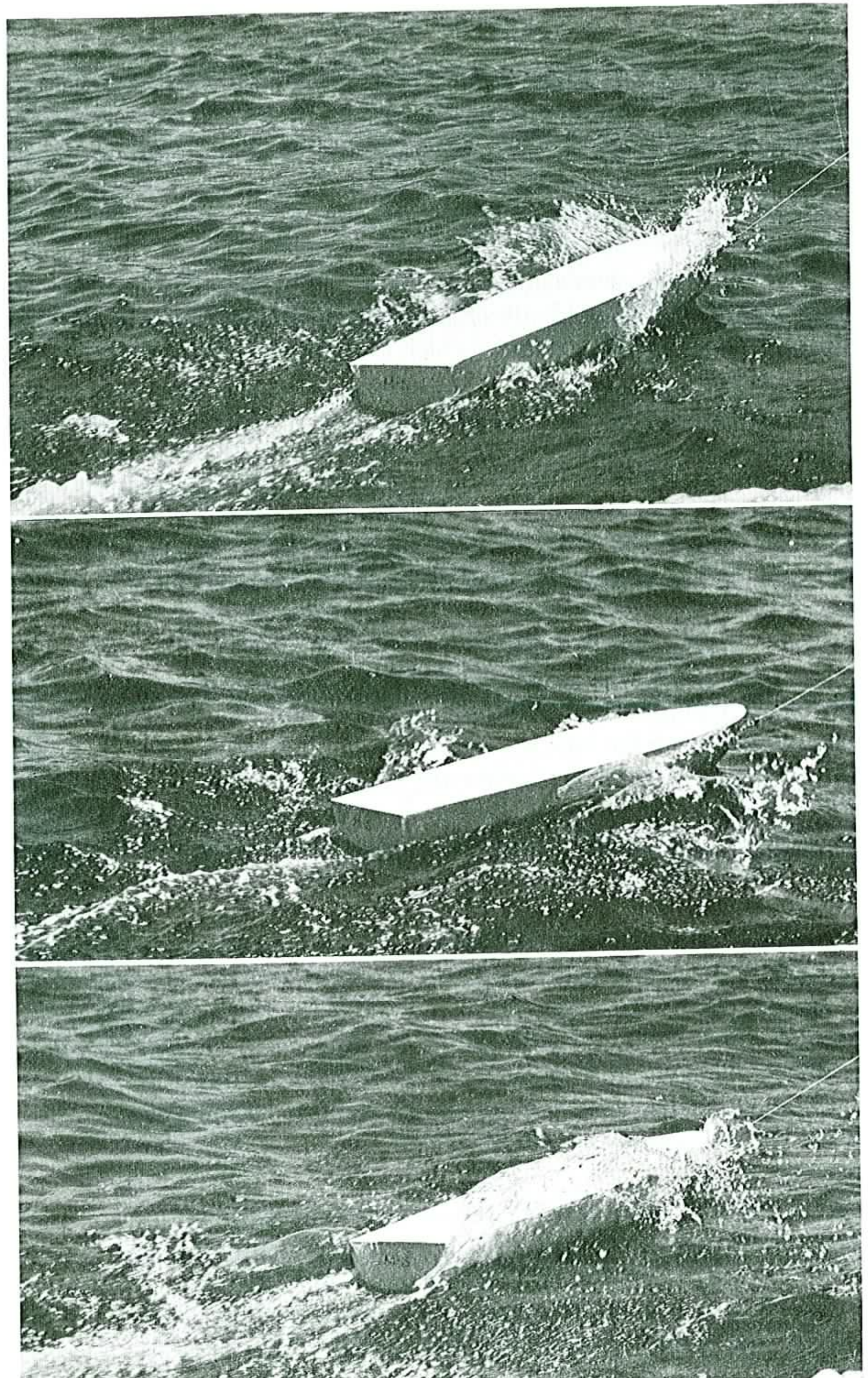


FIGURE 45

Depth at side should be about 125 per cent of depth at keel. Where venting is by natural draft ducts from inside the hull, the duct areas should total to the square of the depth that otherwise would have been used at the side. Such natural draft ducts will carry air at speeds of about 2000 feet per minute.

If exhaust pipes are introduced behind steps, the volume of gas from the engine may be calculated satisfactorily from the piston displacement in cubic feet times one half the revolutions per minute. Half again as much gas plus air, due to reduction in volume by cooling, must be introduced behind each step as would normally be admitted by an air duct alone drawing air at 2000 feet per minute at boat speeds around 50 knots. At higher boat speeds, the amount of air drawn into the steps is in direct proportion.

The angle of all steps abaft the forward plane should be kept as small as possible, usually on the order of 1½ degrees from horizontal.

Maximum lift is attained with multiple steps, either transverse or fan shaped, since such a bottom is essentially a series of leading edges. Some highly successful applications of the fan-shaped multiple step have been made on small runabouts, particularly the Eddy "Aquaflow" boats, where the size and spacing of steps is comparable to clapboarding. On lightly loaded bottoms where the side openings can be relied upon to remain above the surface, the lift and wetted surface reduction are most gratifying.

The total elimination of suction load by using multiple steps, or its partial elimination with a single-step bottom, results in flat trim accompanied by a longitudinal moment of inertia so high as to preclude small sizes from ocean-going work.

There have been two notable approaches toward reduction of excess longitudinal metacentric radius in single-step hydroplanes. One was the British Coastal Motor Boat of Thornycroft design and the other was the Italian torpedo boat built for Mediterranean patrol service. Both hulls were basically of the typical straight-run, high-speed, round bottom type, with the addition of a wide chine forward ending at a single, deep step amidships. This gave a planing surface forward with good wave-flattening qualities, but left an essentially displacement type of afterbody to drag along as best it could. One result, although entirely unwitting, was a faster lifting bow in head seas due to a reduction of lift at the stern. That this tended toward a slight reduction in longitudinal *BM* was never analyzed by the designers, and the lack of fore-and-aft balance apparently discouraged further experiment even by trial and error.

The conclusion seems inevitable that until stepped hulls are inherently powerful in the sailboat meaning of the term, that is, until they have physical size approaching waves at sea, their performance will not come up to minimum standards for blue-water cruising. Their future may hold promise, at least for much interesting speculation. But not only must the high-speed performance of stepped bottoms be adjusted to the existing facts of a timeless ocean, but the sluggishness and poor maneuverability caused by steps encumbering the flow lines at low speed must also be overcome.

Up to the present time, no full program of research and true scientific analysis of all the facts and effects underlying step hydroplane performance has been undertaken. Herein lies a field of potential future development.

Sea Sled Lines

The inverted vee bottom, or Sea Sled, developed by Albert Hickman, has much to recommend it from the standpoint of efficient, low-resistance operation. That the type is no longer commonly produced is due largely to structural difficulties and not to lack of speed or lift. A minor disadvantage of the type is the difficulty of good interior arrangement caused by the inverted vee, particularly forward. However, it is possible that the type may be revived when new materials and structural methods give promise of sufficient added strength.

Essentially, the Sea Sled lines may be compared to those of a soft-riding, convex-section planing hull of more conventional design which has been split along its keel and the two halves reunited with their former outboard sides together and decked over to complete the hull. The sides are straight and parallel with the after chines beveled to facilitate turning without tripping. Forward, the keel meets the deck, sloping aft with a very slight rocker to the flat transom. The basic feature of its performance is that the bow wave is tossed up within the inverted vee and the hull rides on somewhat aerated water, with consequent reduction in wetted-surface friction. The downward convexity of the inverted vee sections effectively snubs out any serious pounding tendency, and its dryness is most gratifying.

Suction under its bottom is practically nonexistent because, while its planing surfaces are considerably warped, the running lines of least angle are the chines, and these are adjacent to undisturbed water not blanketed by the hull. The consequent lift of the Sea Sled is comparable to that of the multi-step hydroplane. This very lift, however, together with its full distribution longitudinally, has rendered the action of large sleds at sea far too stiff. The theory of excess longitudinal metacentric radius is fully developed in Chapter 10. For the present, it is sufficient to point out that this tremendous longitudinal stiffness is an important factor contributing to the structural difficulties of the Sea Sled form in ocean-going sizes.

Lines of Tenders

While the towing of tenders by planing boats is decidedly out of the question as a regular procedure, the towing qualities of tenders in general are a matter of at least academic interest.

Tenders being towed are subject to the same laws of speed-length or speed-beam ratio as are any other hulls. The failure of many tenders to tow straight, their tendency to yaw and dart about from side to side, is the direct result of a hull shape being driven beyond its capacity for speed. For example, a round bottom tender with narrow stern, which would be the easiest and smoothest hull for rowing, is one of the worst possible hulls for towing. If it is, say, 9 feet long on the water, its top speed regardless of power will, as a displacement hull, be 6 knots, and blood, sweat or tears will not increase it.

However, if the irresistible towline does move faster, displacement characteristics must cease and one of two conditions will result. Either the tender will sink, or, as usual with lightweight boats, it will attempt to plane and in so doing try to present a broader leading edge. In other words, the plane will present less resistance in an angular direction than fore and aft with a narrow planing surface.

Modern tenders should be extremely wide; aspect ratios even up to .6 are excellent. The bottom should be shaped with planing in mind, as previously described for larger hulls. Such a shape is admittedly only fair for rowing, but good for any type of motor drive, and essential for proper towing.

Chapter 9

SUBDIVISION AND FLOODING

Floodable Length

In its strict interpretation, the word *bulkhead* means a watertight subdivision capable of withstanding any pressure of water likely to be brought against it. A true bulkhead cannot be fitted with openings, not even so-called watertight doors. For this reason, the subdivision of vessels up to around 100 feet in length must, for practical reasons of ventilation and accessibility, forgo complete watertight subdivision in its true meaning. A collision bulkhead abaft the forepeak is usually the limit of fully watertight subdivision in present planing hull sizes.

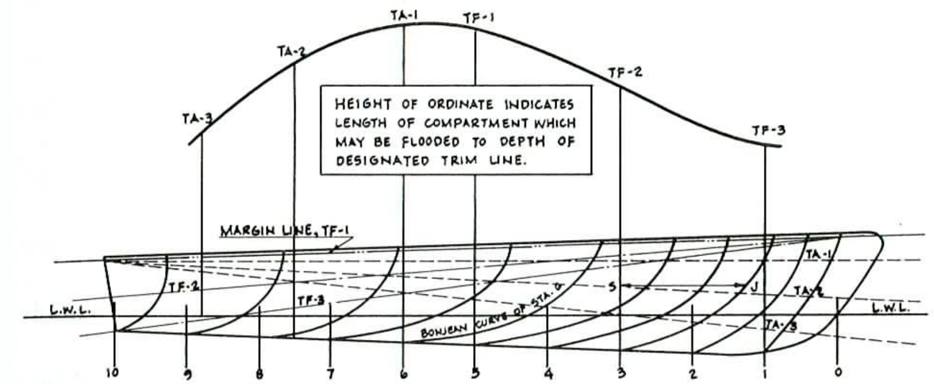
The floodable length of a vessel is the maximum length of compartment which may be flooded without putting any point of the deck below water. In general, forward compartments whose flooding would cause trim by the head, must be considerably shorter than compartments aft of midships whose flooding would cause only parallel sinkage.

If subdivision is to be incorporated as a design feature, bulkhead spacing must be so calculated as to obtain at least one compartment that may be flooded without putting the deck under. Bulkheads spaced at any distances greater than the floodable length are more of a hazard than no bulkheads at all, for flooding can then cause enough trim to put the deck at a large angle, and foundering is greatly hastened. In a one-compartment ship, there is the danger that rupture will be suffered on a bulkhead, flooding two compartments and usually causing a plunge sinking or else a sinking by capsize. All sinkings are bad, but plunges and capsizes can be so rapid that personnel may not have time for escape.

However, in the final analysis it is the total amount of water entering a hull which is of vital concern. The method whereby this water may be confined to a non-disastrous location and extent is the problem. In small craft, three-compartment subdivision may result in bulkhead spacing so close as to be entirely impractical from the standpoint of ar-

rangement. Two-compartment subdivision, if attainable, is superior to one-compartment subdivision or none at all. However, for practical purposes, careful spacing of bulkheads for one-compartment floodability is the ideal to which small craft may usually aspire.

High-speed craft suffer damage most frequently from running into or over floating obstructions. A waterlogged telephone pole with its climbing spikes may be just awash and almost invisible. The planing bottom running at 40 knots over this obstruction would suffer a rupture which would nullify any kind of bulkhead compartmentation. Only



FLOODABLE LENGTH CURVE

Figure 58

an excellent double-bottom construction would keep the vessel afloat. However, in ordinary wood construction, a watertight double bottom cannot be fitted because of the certainty of dry rot in unventilated spaces. Where molded construction using plastics is employed, or where the hull is of metal, the double bottom is feasible and should be incorporated.

Floodable Length Curve

The floodable length at any section of the hull is calculated most easily with the aid of Bonjean curves as shown in Figure 58. The Bonjean curve itself is simply an area curve of a transverse section, drawn on a hull profile. To plot the curve, the area of a section up to each of several waterlines is measured by planimeter and the area figure then set off on the profile drawing to the right of the station in question, as a linear distance along the particular waterline. Half a dozen such distances set off from the transverse station are sufficient for the fairing of a Bonjean curve. The linear distance is set off to the right to any convenient scale which confines the curves within limits that do not cause a confusion or undue overlapping.

Displacement is then easily calculated to any trim line by putting the indicated sectional areas through Simpson's rule. The sectional area up to the level of the trim line is simply the linear distance from the intersection of trim line and station, measured horizontally, to the applicable Bonjean curve.

Allowable sinkage may be calculated to an arbitrary "margin line" 3 or more inches below the deck line. To sink to this margin line, the hull must be trimmed forward, hence the designation TF-1. From the aft end of this margin line, strike in a line of parallel sinkage, designated TA-1 for trim aft.

A convenient method is to divide the hull depth on station 10 into equal parts and draw TF-2 and TF-3 from the point of intersection between station 0 and the margin line. On station 0 below the parallel sinkage line at TA-1, draw TA-2 and TA-3 with similar spacing.

The volume of the hull sunk to any trim line is easily calculated by measuring to scale the horizontal distances S-J, which are the areas of stations up to the trim line intersection. Putting these areas through Simpson's rule gives total volume and center of gravity of the volume.

The volume of water displaced at each trim line which is in excess of the ship's original, or intact, volume is *damage water*, and its center of gravity is X distance from midships:

$$X = \frac{V}{Vd}(y + b) + b$$

Where:

V = Volume in cu. ft. of intact displacement
Vd = Volume in cu. ft. of damage water
y = C.B. of V aft of midships
b = C.B. of $V + Vd$ forward of midships

As an example, assume that the normal displacement of a boat is 70 tons or 2460 cubic feet. The volume of damage water to sink it to trim line TF-2 is 3250 cubic feet. The C.B. in intact condition is 9 feet aft of midships and the C.B. of the entire displacement in damaged condition is 12 feet forward of midships. The center of gravity of the damage water is then:

$$\begin{aligned} X &= \frac{2460}{3250}(9 + 12) + 12 \\ &= .755\ (21) + 12 \\ &= 15.85 + 12 \\ &= 27.85 \text{ Ft. forward of } ⊠ \end{aligned}$$

Next find the sectional area of the hull 27.85 feet forward of midships below the trim line and divide the damage water volume by this area to obtain the approximate length of the damaged compartment at 100% permeability. Actually, the compartment will contain furnishings or stores which will reduce the permeability and thereby extend the actual floodable length. Crew's quarters may be considered 90 per cent permeable, engine room 80 per cent and store room 70 per cent.

If the compartment ruptured is mainly crew space, consider the calculated length as 90 per cent of the actual floodable length and from point X erect an ordinate whose height is 100 per cent of the full floodable length.

The same procedure followed with each of the trim lines will give six ordinates whose tops can be faired into the floodable length curve. Adjust the position of each compartment so that any ordinate dropped from the curve falls at the actual center of gravity of the space between bulkheads.

The height of an ordinate of the floodable length curve is the floodable length, L_f, and is also found from the formula:

$$L_f = \text{Lost } BM \times \frac{12 \times 35 \times \text{cu. ft.}}{\text{Beam}^3 \times \text{Perm.}}$$

Stability Flooded

Due to the large metacentric height of planing hulls, considerable metacentric radius may be lost before the vessel becomes unstable. However, if a large center compartment is ruptured, it is possible that the immersed volume may be so increased that BM, which equals $I \div V$, becomes very small, GM becomes negative, and the vessel capsizes.

Since the floodable length curve indicates the longest compartment as near midships, this one area should be assumed as flooded and a stability calculation performed to determine how much, if any, to shorten the compartment.

For an example, assume the following dimensions and characteristics:

Beam = 20 feet
Molded draft, H = 3.6 feet
C.B. below L.W.L. = .9 feet
Length of compartment = 36 feet
Cross section to L.W.L. = 46 sq. ft.
Permeability = 80%
Surface permeability = 90%
Original waterplane = 1420 sq. ft.
Original GM = 9 feet

The problem may then be set down in fourteen steps:

(1) Lost buoyancy in tons $(36 \times 46 \times .80) \div 35$	= 37.8
(2) Lost waterplane, tons per inch $(36 \times 20 \times .90) \div 420$	= 1.54
(3) Original waterplane ÷ 420 Tons per inch	= 3.37
(4) Remaining waterplane (3) − (2), tons per inch	= 1.83
(5) Sinkage in feet $(1) \div [12 \times (4)]$	= 1.73
(6) Mean height new layer $H + [\frac{1}{2}(5)]$	= 4.46
(7) Rise of lost buoyancy $[\frac{1}{2}(5)]$ + [C.B. below L.W.L.]	= 1.76
(8) Original displacement Tons	= 72
(9) Rise of C.B. in feet $(7) \times [(1) \div (8)]$	= .92
(10) Lost moment of inertia, I $(36 \times 20^3 \div 12)\ .9$	= 21600
(11) Lost BM, feet $(10) \div [35 \times (8)]$	= 8.58
(12) Net lost GM (11) − (9)	= 7.66
(13) Original GM, feet From original data	= 9
(14) Remaining GM After damage (13) − (12)	= 1.34

The conclusion to be drawn from this example is that a planing type hull subdivided for one-compartment floodability will not ordinarily capsize due to loss of stability. However, had the center compartment been longer than assumed for a one-compartment floodability, that is, a non-subdivided hull, or had the original GM been smaller, negative GM would have been reached before sinkage to deck level and therefore capsize would have resulted.

During two years of war after Pearl Harbor, none of the two-compartment mosquito fleet was reported sunk. Of the one-compartment craft sunk after bulkhead rupture, a majority foundered by plunging and the remainder by capsize. Of the non-subdivided group sunk, sinking was by capsize in every case reported. For non-subdivided pleasure craft, the lower center of gravity would prevent capsize. Sinkage would be on an even keel to deck level, where the ordinary pleasure boat would float awash.

Bulkhead Strength

To calculate the strength required for a bulkhead to withstand flooding on one side, the required section modulus is:

$$Z = \frac{\text{Moment}}{\text{Stress}} = \frac{M}{f}$$

Main compartmentation bulkheads of steel may be designed for a stress of 27,000 pounds per square inch, while 1-inch plywood may be stressed to 3000 pounds per square inch. All bulkheads in planing craft extend the full depth of the hull and should be calculated for a head of

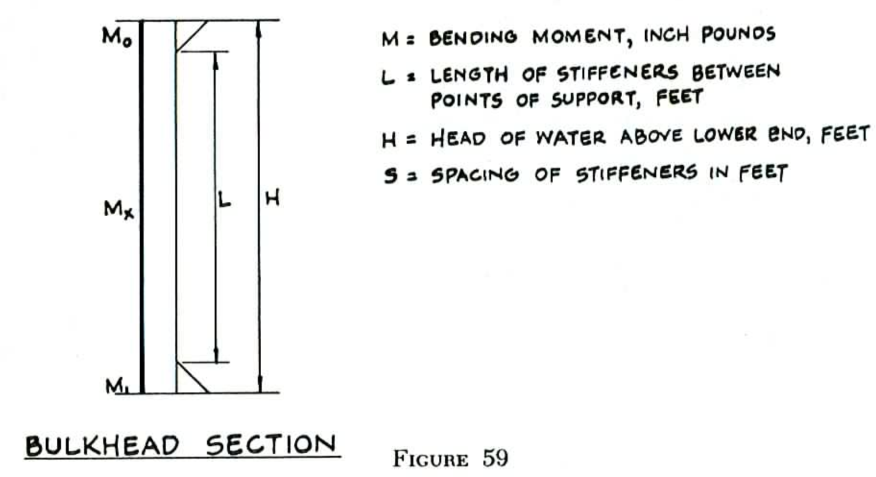


Figure 59

water pressure equal to their height, H, as shown in Figure 59. For this condition, with vertical stiffeners bracketed at both ends, the moments are:

$$M_1 = 38.4L^3S$$
$$M_o = 25.6L^3S$$
$$M_x = 16.4L^3S$$

With vertical stiffeners but without brackets, the moments are:

$$M_x = -49L^3S$$

Vertical stiffeners bracketed at bottom only:

$$\begin{aligned} M_1 &= 51.2L^3S \\ M_o &= 0 \\ M_x &= -22.9L^3S \end{aligned}$$

Vertical stiffeners bracketed at top only:

$$\begin{aligned} M_1 &= 0 \\ M_o &= 44.3L^3S \\ M_x &= -32.4L^3S \end{aligned}$$

For horizontal stiffeners bracketed at both ends:

$$\begin{aligned} M \text{ at ends} &= 64HL^2S \\ M \text{ middle} &= -32HL^2S \end{aligned}$$

REQUIRED THICKNESS OF PLATING

HEAD IN FEET	S = 18"		S = 21"		S = 24"		S = 27"		S = 30"		S = 36"	
	INCHES	LBS	INCHES	LBS	INCHES	LBS	INCHES	LBS	INCHES	LBS	INCHES	LBS
6	.0725	2.9	.0840	3.4	.0960	3.9	.1080	4.4	.1200	4.9	.1440	5.9
8	.0800	3.2	.0892	3.6	.1019	4.2	.1146	4.7	.1274	5.2	.1528	6.2
9	.0825	3.3	.0922	3.8	.1054	4.3	.1185	4.8	.1317	5.4	.1580	6.5
10	.0850	3.4	.0955	3.9	.1091	4.5	.1227	5.0	.1364	5.6	.1636	6.7
12	.0900	3.6	.1050	4.3	.1200	4.9	.1350	5.5	.1500	6.1	.1800	7.4
14	.0970	3.9	.1128	4.6	.1290	5.3	.1451	5.9	.1612	6.6	.1934	7.9

$R = \dfrac{\text{LENGTH OF PANEL}}{\text{WIDTH OF PANEL}}$; ASSUMED EQUAL TO OR GREATER THAN 3.0.

WHEN R IS LESS THAN 3.0 MULTIPLY HEAD IN FEET BY FACTOR K AS IN TABLE BELOW AND USE RESULTING REDUCED HEAD.

R	1.00	1.25	1.50	2.00
K	.65	.80	.90	.96

FIGURE 60

Figure 60 shows the required thickness of plating for steel bulkheads with stiffeners spaced for 18 to 36 inches apart at heads from 6 to 14 feet.

For example, calculate the required section modulus Z for a bulkhead 8 feet high with vertical stiffeners spaced 1.5 feet on centers and bracketed at both ends:

$$\begin{aligned} M_1 &= 38.4 \times 8^3 \times 1.5 \\ &= 29{,}500 \text{ ft. lbs.} \\ M_o &= 25.6 \times 8^3 \times 1.5 \\ &= 19{,}700 \text{ ft. lbs.} \\ M_x &= 16.4 \times 8^3 \times 1.5 \\ &= 12{,}600 \text{ ft. lbs.} \\ M_1 &= 29{,}500 \times 12 \\ &= 354{,}000 \text{ inch pounds} \\ Z &= \frac{M}{f} \\ &= \frac{354{,}000}{27{,}000} \\ &= 13.11 \text{ in.}^3 \text{ required} \end{aligned}$$

In all probability the same weight of plating and stiffeners will be carried from bottom to top; hence what is strong enough for M_1 will be more than enough for points higher up.

As an example, assume that a forward collision bulkhead has a moment of 70,000 inch pounds and is to be built of fir plywood with stiffeners spaced 18 inches on centers. The required section modulus is:

$$\begin{aligned} Z &= \frac{70{,}000}{3000} \\ &= 23.3 \text{ in.}^3 \end{aligned}$$

If the plywood is to be 1 inch thick and the stiffeners are 2 by 6 inches, the calculation is as follows:

	Area	*y*	*M*	I_x $bd^3/12$	*r*	Ar^2
Plywood	18	0.5	9	1.5	1.4	35.28
Stiffener	12	4.0	48	36.0	2.1	52.92
	30		57	37.5		88.20

$$\begin{aligned} \text{Neutral axis, } y &= M/A = 57/30 = 1.9 \text{ in.} \\ I_{na} &= I_x - Ar^2 \\ &= 37.5 - 88.2 \\ &= 125.7 \text{ in.}^3 \\ \text{Extreme fiber, } c &= 7 - 1.9 = 5.1 \text{ in.} \\ Z = I/c &= \frac{125.7}{5.1} \\ &= 24.65 \text{ in.}^3 \end{aligned}$$

Flotation

Prior to construction in reinforced plastics, safety flotation was achieved by means of sealed metal tanks fitted to shape and securely strapped in place to resist being torn out should their buoyancy be called into play by flooding. At least once a year the tanks had to be removed for inspection—a major operation. Their weight and bulk was considerable. For this reason, only lifeboats carried flotation tanks.

Safety flotation today is provided by expanded unicellular plastic foams such as polyurethane or blocks of polyvinyl chloride. These non-structural foams may weigh only two pounds per cubic foot and may be either foamed in place in odd-shaped voids or may be cut to shape from blocks and used as core material behind structural skins.

The cubic footage of all material in the boat displaces its own bulk in water when submerged. Assuming that a certain 30-footer weighs 10,500 pounds and its total impermeable bulk is 150 cubic feet, it is apparent that the weight per cubic foot is 70 pounds. Sea water at 64 pounds per cubic foot would require 164 cubic feet to balance the weight. Therefore, if no increase of boat weight is involved, 14 cubic feet of foam flotation would maintain the boat awash. Whether she would float upright or upside down would depend on how the Vertical Center of Buoyancy had been affected by the placement of foam flotation.

Built-in flotation is almost universal today in plastic hulls and it is practical to add it in wood and metal hulls. Space out under the weather decks is usually the best location since this is likely to be above the Center of Gravity but low enough to keep the deck awash, at least.

Subdivision in Pleasure Boats

Since it is impractical to subdivide the usual type of cabin cruiser for even one-compartment floodability, the answer to proper subdivision must come from an analysis of several other factors. For example, the advantage of a watertight collision bulkhead as a safety feature is apparent. However, if it is watertight, it will be necessary to pierce it for a bilge suction line, which adds some complication to an otherwise simple bilge pump system. Also, separate ventilation of the forepeak must be provided. If this is accomplished by means of air ducts piercing the bulkhead, watertightness of the bulkhead is completely lost. If

it is done by means of a watertight hatch, ventilation will be indifferently attended to and any wood construction will inevitably suffer from dry rot. Furthermore, the hatch may not be closed when most needed.

The question of the collision bulkhead has been well answered in a new 36-foot cabin cruiser of molded construction. Since the material is not wood, rot is not a consideration. The bulkhead is molded into the shell but at its lowest point a limber hole ⅜ inch in diameter will drain off any condensation into the bilge. In the event of rupture forward of the bulkhead, water would of course pour in through the limber, but its small size would restrict the flow to an amount easily within the capacity of the single suction bilge pump. Lack of ventilation in this case, due to the impervious materials of the molded structure, will result only in condensation even though no hatch is provided.

Fortunately all the old complications formerly inherent in true watertight subdivision have been completely eliminated in modern plastic hulls. With this versatile new material, rot is no longer a problem and any number of compartments and voids may be sealed off without regard to ventilation. Some of these compartments may form what amounts to a double bottom and will be used to carry fuel and water while others will be left void as convenience dictates. Subdivision of the plastic hull is thus almost automatic and compartmentation may be achieved for withstanding any set of circumstances the architect desires.

Compartmenting below cockpit and cabin soles into voids a foot or so in each direction is structurally good and goes a long way toward achieving the unsinkable ship. From a practical standpoint, it is better that these small voids created by multiple subdivision be sealed off as empty spaces rather than being filled with foam. Subsequent access for alterations or any other reason is then much easier. The slight amount of condensation which will occur in sealed voids is of no consequence in plastic construction.

Chapter 10

HYDRODYNAMIC CALCULATIONS

It is in the dynamic phase of planing that the greatest field for research still remains. Planing hulls the world over, being relatively new, are still roaring along on their expensive missions without the benefit of the displacement hull's generations of experience. The fundamental changes which take place when the throttles are opened are only beginning to be reduced to science.

However, while almost unlimited research remains to be done, several of the basic phenomena of stability changes, centrifugal forces and metacentric variations due to planing have recently been measured and reduced to formulas of average performance. It is from this pioneering, though still preliminary, research that planing hull design has gradually approached and demonstrated its capacity for a truly seagoing type of performance, with not only elimination of such elementary flaws as pounding and broaching but also a more complete realization of the planing hull's tremendous potential in lift, stability and comfort.

Dynamic Moments

The metacentric radius of any hull, that is, the distance from the center of buoyancy up to the metacenter, is the quotient of waterplane inertia and underwater volume, thus:

$$BM = \frac{I}{V}$$

For hulls without motion the formula holds good, but in the upper ranges of speed possible for displacement type hulls, around a speed-length ratio of 2, there occurs a noticeable increase in stiffness, indicating a larger *BM*. Since the underwater volume has not always decreased but has, in fact, sometimes increased due to squatting, the waterplane moment of inertia has obviously increased.

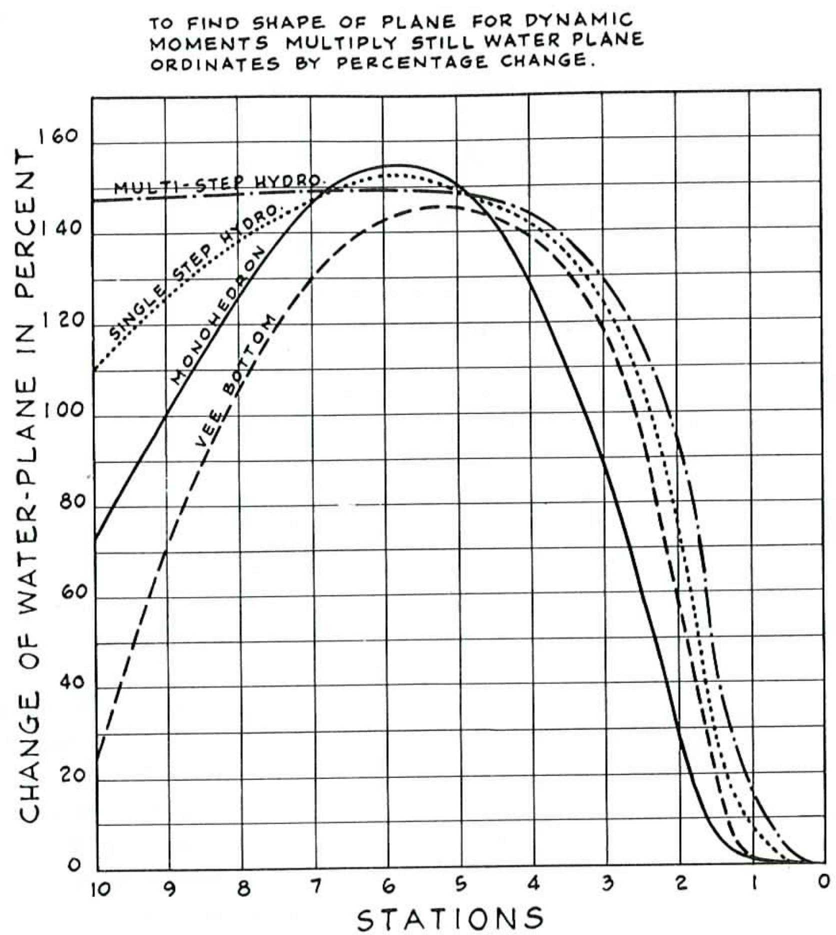
For planing hulls, the increased *BM* first became definitely apparent during the process of loading torpedoes on board in the side tubes and subsequently handling and firing the same torpedoes with the hull at planing speed. The degree of heel with the boat at rest, caused by empty tubes on one side and full tubes on the other, was about double the heel with the same weight distribution when planing. So marked an increase in *BM*, or transverse stability, could be due only to the dynamic action of planing.

Tests on change in longitudinal *BM* have also shown the effect of dynamic forces, although in this direction, *BM* decreases in good seagoing types of hulls and increases only in stepped types. The effect of a decreased longitudinal metacentric height is to improve the seakeeping qualities of the hull. The higher the longitudinal *BM*, the more the hull tends to plunge through rather than ride over steep waves. Conversely, a low metacenter allows change of trim with sufficient rapidity for the hull to accommodate itself to normal wave slopes.

Perhaps the paramount reason for a step hydroplane's lack of seaworthiness, at least in present-day sizes, is its excessively high longitudinal *BM*. From the chart of Effective Waterplane Change for Dynamic Moments in Figure 61 can be drawn the effective waterplane shape for calculation of dynamic moments of inertia. Figure 62 illustrates this effective change. The waterplane shape at rest is that of a monohedron hull, and from this shape are calculated the static moments of waterplane inertia. The ordinates of this static shape are then multiplied by the percentages indicated in the chart of effective change in Figure 61. A calculation for longitudinal and transverse *BM* made directly from these new ordinates gives the metacentric radius while planing.

It will be seen that the width of the first three stations of monohedron type hulls is reduced. The midship stations are wider. Number 9 remains at its still-water width and then number 10 shows a decrease. The planing angle and the pattern of pressures thus effectively increase the transverse moment of inertia and, by shortening the spread of longitudinal pressures, decrease the longitudinal moment. Examination of all the curves shows that the true monohedron attains the greatest transverse stability and also benefits most from reduced moment fore and aft.

The waterplanes of a single-step and a multi-step hull are shown in Figures 63 and 64. The ordinates of the effective dynamic shape are



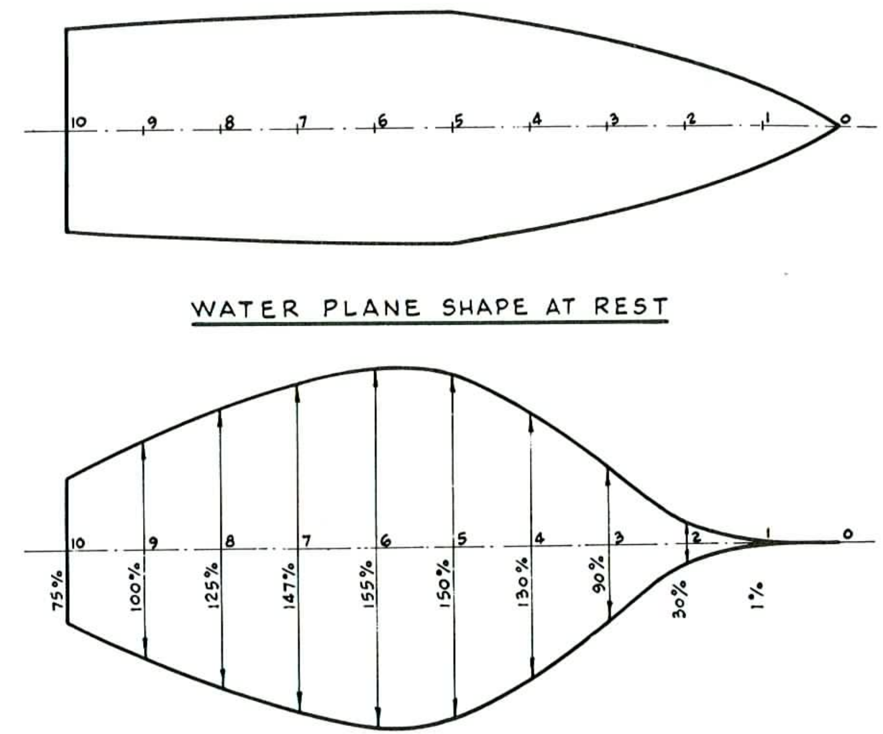
EFFECTIVE WATER-PLANE CHANGE FOR DYNAMIC MOMENTS

BASED ON AVERAGE PERFORMANCE AT SEA
SPEED BEAM RATIO = 3

FIGURE 61

lengthened or shortened according to the percentages shown by the proper curve in Figure 61. Both types carry the excess beam of their "effective" shapes much too far aft.

The chart in Figure 61 is worked out as an average for hulls of each of the four types noted and at speeds of around a speed-beam ratio



WATER PLANE SHAPE AT REST

WATER PLANE AREA OF DYNAMIC MOMENTS

FIGURE 62

of 3. For other speed-beam and aspect ratios of monohedron type hulls, the increase of transverse *BM* may be read direct from the chart in Figure 65. For example, a hull of aspect ratio .3 running at a speed-beam ratio of 3.5, will increase its transverse *BM* to the 1.39 power of its static *BM*. Assuming the static *BM* to be 14 feet, the dynamic *BM* will be: $14^{1.39} = 39$ feet.

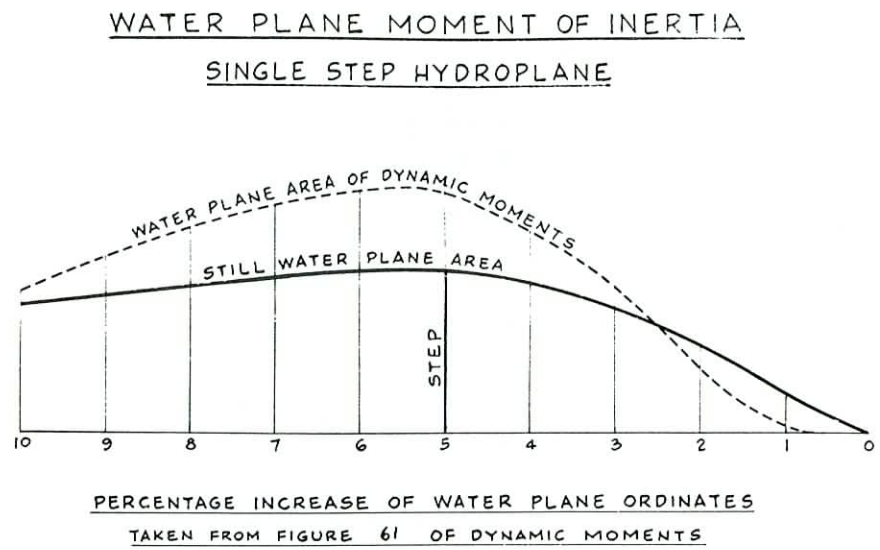


FIGURE 63

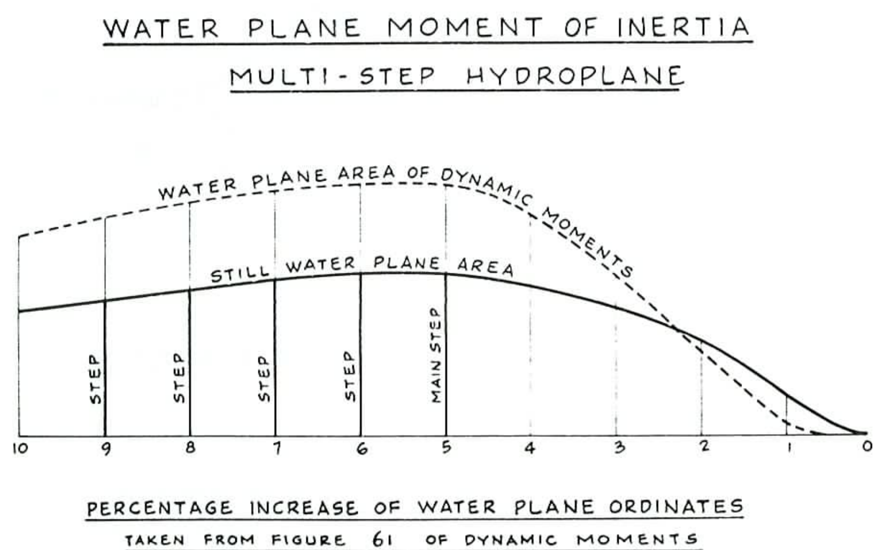


FIGURE 64

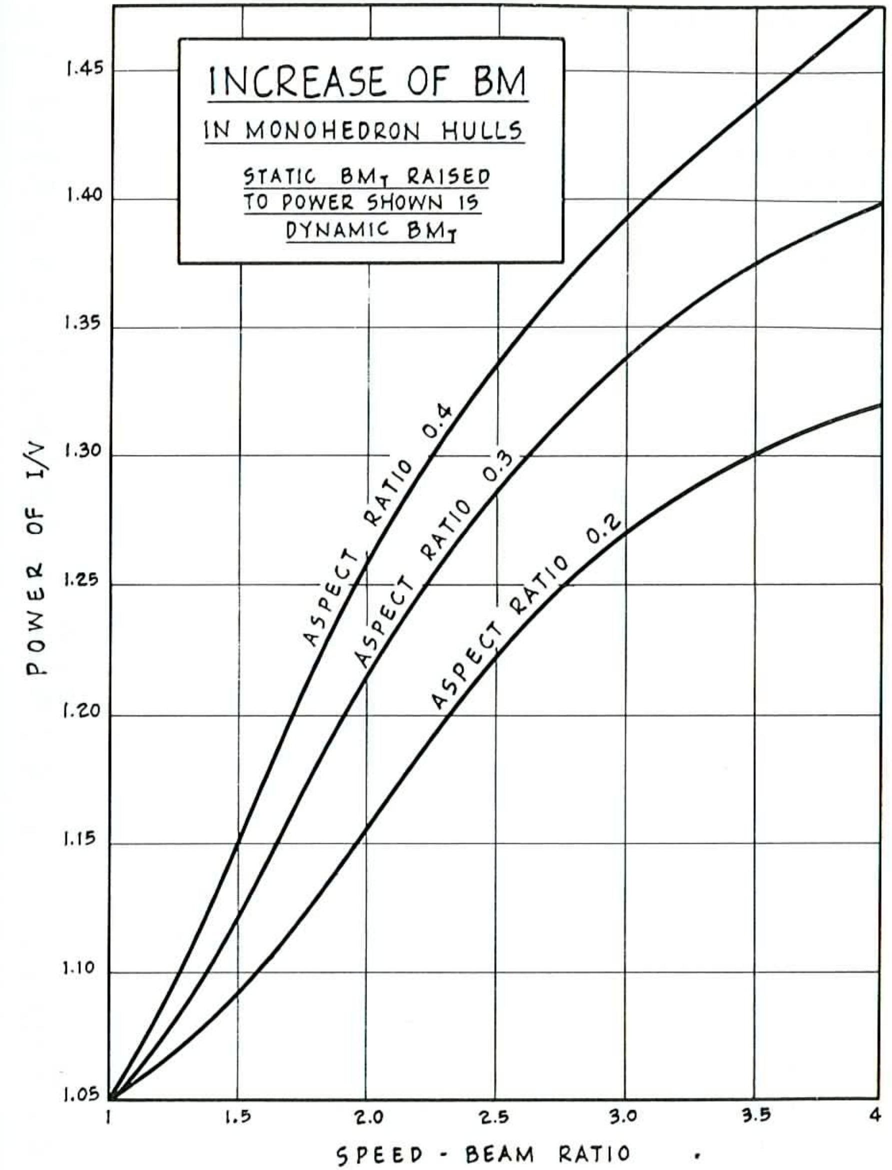


FIGURE 65

The following table is indicative of normal metacentric radii:

		Static BM		Dynamic BM	
		Trans.	*Long.*	*Trans.*	*Long.*
Monohedron,	40′ × 13′	9.3′	106′	20′	66′
Monohedron,	86′ × 22′	19.8′	313′	51′	161′
Vee-Bottom,	77′ × 16′	18.6′	275′	38′	194′
Vee-Bottom,	65′ × 15′	17.2′	205′	39′	163′
Single-Step,	55′ × 14′	12.1′	180′	36′	175′
Multi-Step,	32′ × 9′	6.5′	92′	17′	132′
Round-Bottom,	65′ × 17′	6.2′	95′		
Round-Bottom,	83′ × 16′	9.5′	225′		

From the foregoing table, the reason for the monohedron's characteristic sea-kindliness becomes immediately apparent. Also of interest is the static *BM* of the two round-bottom hulls listed. At full speeds there may be some increase in transverse stiffness but there is certainly no appreciable change in the longitudinal moment. At sea speeds, therefore, the 83-foot round-bottom hull plunges into waves with the stubbornness of a 225-foot *BM*, while the 86-foot monohedron is bucking a dynamic *BM* of only 161 feet.

On the other hand, the 65-foot round-bottom hull carries a longitudinal *BM* of only 95 feet compared to the 65-foot vee-bottom hull's *BM* of 163 feet. With such unfortunate longitudinal stiffness, the vee-bottom hull is forced to reduce below planing speed at sea. Herein lies the key to the fundamental theory of correct design approach to genuine seaworthiness in planing hulls.

Excess Metacentric Radius

While the dynamic *BM* of some warped plane vee-bottom hulls is excessive for seagoing performance, that of stepped hulls is far worse and it is obvious that their prime difficulty at sea lies in their excess radius of longitudinal metacenter. Because maximum lift is attained with stepped bottoms and because the type offers the greatest reduction in wetted surface, its performance in still or relatively quiet water is most gratifying. The total elimination of all suction load from the afterbody results in a trim which is practically flat. But this one characteristic of flat trim is its undoing at sea.

As will be seen in the chart of Dynamic Moments in Figure 61, the longitudinal moment of inertia of the waterplane is antagonistic in the extreme to sea-kindliness. The high longitudinal moment of inertia shown by the area of dynamic moments, results in a hull performance

devoid of ability to climb over waves. Its only method of travel seems to be through rather than over.

There remains, however, the very good possibility that on hull sizes much larger than any now in existence, this objection may be overcome. It seems reasonable that hulls 200 or 300 feet long would negotiate at speed any seas normally encountered. The high longitudinal moment of inertia would be largely counteracted by the proportionately greater volume of the hull, resulting in a more balanced longitudinal *BM*. But in sizes below 100 feet, the multi-step hull of present day shape is definitely a hazard at sea. In general, the single-step type evidences similar characteristics but in lesser degree. The lift does taper off somewhat toward the aft end, with the result that the single-step hull is not quite as stubborn in performance as the multi-step, but still has considerably higher longitudinal metacenter than does a stepless bottom.

Planing Action Among Waves

The action of a fast boat in rough water consists essentially of heaving and pitching. Rolling is not a factor and, at planing speeds, the yaw and yaw-heel rarely need occur. On the other hand, planing hulls of harsh, flat or awkward forms which are forced into abnormal speed reductions, do have every known motion to an even dangerous extent.

For hulls designed to attain a high degree of useful lift, cruising speed can be maintained and even increased under difficult sea conditions. At reduced speed, their motions should be comparable to those of displacement craft. At high speed, the augmented transverse *GM* due to planing, stabilizes the tendency to roll, while the speed itself in a hull of proper lateral plane, is usually sufficient to outrun following seas which would otherwise buoy up the wide stern and tend to carry the hull into a yaw.

Since the rapidity with which waves equal to the boat's length are crossed, the tendency is to force an equally rapid period of oscillation on the vessel. The more this tendency is resisted by the longitudinal *GM* of the vessel, the more the hull will tend to plunge through instead of over the seas. Large dispersion of longitudinal weights, particularly forward weights, also tends to cause plunging.

However, a craft with extremely small moment of inertia forward tends to shoot its forefoot farther out into space after skimming the crest of a wave, and the resultant drop is of considerable moment.

Waves of approximately hull length or slightly longer cannot be crossed at right angles without this "taking off" and landing effect.

A planing bottom will make its best speed in waves of appreciable size. An increase of around 4 per cent over calm-water speed may usually be expected with wave heights from one quarter to one half the boat's beam. With waves of a height equal to the beam, the distance between crests ordinarily equals or exceeds the length of the hull, and speed is reduced back to calm-water conditions.

Where the distance between crests is around half the length of hull, or less, a considerable air cushion operates beneath the plane, unless the hull is overloaded, effectively cutting down frictional resistance. This reduction in friction is pure gain up to the point where the size of waves causes sufficient plunging to add new wetted surface plus added wave-making.

The form of hull best suited to minimizing plunging is extremely full and blunt at and immediately below the chine in the forward sections. Waterplane area combats plunging, and sectional fullness should be incorporated as rapidly as possible immediately above the forefoot but not at or below the still-water load line. Fullness on deck is of little value except for the convenience of room to work, for by the time the bow has plunged to the deck, the greatest resistance has already been set up.

The tumbling aboard of green water broad off the bow is always indicative of one of two possible design errors: either the chine has been formed inadequately to act as a wave deflector, or the hull is incapable of maintaining sufficient speed under the given sea conditions. With proper speed and wave deflection, the forward sections are of necessity ahead and above the point of contact except among waves at right angles and steep enough to make some plunging inevitable.

In hulls with concave forward sections of very low deadrise, the landing impact is disastrous to structure, reaching a recorded magnitude in excess of 14 G. Such hulls must either be slowed down or driven on a different course.

In a series of experiments conducted by John L. Hacker, it was found that the crashing type of impact could be totally eliminated with a forward section of modified Maierform lines. The essential feature was sectional convexity. Examination of the dihedral angle, formed at the keel between chords of the arcs from keel to chines in the vicinity of stations 2 and 3, indicated that the impact type of pounding was effectively eliminated at any angle smaller than 130 degrees. At about 130 degrees the characteristic pound of impact had changed substantially to that of a snubbed deceleration. The smaller the angle, provided the general convexity was maintained, the softer the snubbing effect became.

Forces Due to Pitching

The natural period of pitching of planing craft in present-day sizes very frequently coincides with the period forced by waves. For example, the natural period of a 60-ton 85-footer was 2.6 seconds. At her designed speed of 42 knots, she would travel in 1 second 42×1.689 or 71 feet. In 2.6 seconds this is 185 feet. Therefore, waves of about 185 feet from crest to crest caused her the widest amplitude of pitching. Shorter waves, namely 60 to 80 feet, would force a pitching period of 1 second, but at greatly reduced amplitude.

The axis of pitching is through the center of pressure beneath the plane. In the process of crossing waves of any appreciable magnitude, this center must move forward and aft far too radically to be accurately located for all conditions. However, it can be very closely approximated by observation of the performance of similar types.

For example, it is quite obvious that the upward component of the center of pressure, like the center of buoyancy, must be directly below the downward component of the center of gravity. However, the planing phenomenon, as previously explained, involves an additional weight of a magnitude varying according to the type of plane. This added weight is the suction load of the particular plane, and is always toward the aft end of the hull. The effect is to move the downward component of the center of gravity farther aft.

Within the range of speeds suitable for seagoing hulls, the cleanest monohedron types are usually subject to a sternward shift in center of gravity of about 5 per cent of the static waterline length. Warped plane types commonly experience a shift of 8 to 12 per cent aft. The 45-foot Navy rescue boat operates with a 14 per cent sternward shift in the dynamic center of gravity.

It is therefore more accurate, when some reasonable estimate of this shift can be made, to assume a point somewhat farther aft than the static center of gravity as the point about which pitching and strength calculations are made. Such a point may then logically be called the *pitching fulcrum*.

The period of pitching, unlike the period of roll, is considered for one direction only, and may be treated like the problem of a pendulum. The time in seconds for this period in one direction is:

$$T = \pi\sqrt{\frac{GM}{g}}$$

Assuming a 40-footer with a planing GM of 61 feet, the time of its natural period is:

$$T = \pi\sqrt{\frac{61}{32.2}}$$
$$= 4.2 \text{ seconds}$$

This natural period, however, is of only occasional consequence since the period of pitching is usually forced by the waves in conformity with their own period. Hence, for practical purposes, the period of pitching is a function of the distance between wave crests and the speed with which they are crossed.

The amplitude of pitching can be most conveniently expressed in radians instead of degrees. Radians are equal to:

$$\frac{\pi \times \text{degrees}}{180}$$

The weight of each item is increased from its static condition to a "virtual weight" according to the time of the pitching period and the amplitude. The formula for determining the virtual increase in weight is:

$$W_v = \frac{W}{g} \times D \times \frac{4\pi^2}{P^2} \times R$$

Where:

W_v = Virtual weight
W = Original weight
D = Distance of W from C.G.
P = Period in seconds
R = Radians

Assume that a 90-footer is being forced to pitch in a period of 1.2 seconds and that she carries a 100-pound anchor in a bow hawse pipe 60 feet forward of her pitching fulcrum. The amplitude is 5 degrees. To determine the virtual weight of the anchor, first find the numerical value of the radians:

$$\text{Radians} = \frac{\pi}{180} \times 5$$
$$= .087$$

The virtual weight is then:

$$W_v = \frac{100}{32.2} \times 60 \times \frac{39.5}{1.44} \times .087$$
$$= 445 \text{ pounds}$$

Structure and all parts pertaining to the securing of this anchor must be designed with a virtual weight of 445 pounds in mind. Each mass of weight throughout the hull should be similarly evaluated in order to make certain of adequate structural strength for foundations, girders and fastenings.

For stresses set up by rolling action, it is necessary to know the transverse reaction. For example, assume that a 100-pound tender is mounted on deck 8 feet above the waterline. What is the transverse reaction on the gripes if a 24-degree roll is completed in 1.1 seconds? A completed roll is from port to starboard and back to port.

$$\text{Radians} = \frac{\pi}{180} \times 24$$
$$= \frac{\pi}{7.5}$$
$$= .419$$

Maximum tangential force is then

$$Ft = \frac{100}{32.2} \times \frac{4\pi^2}{1.1^2} \times .419 \times 8$$
$$= 3.1 \times 32.6 \times .419 \times 8$$
$$= 338 \text{ lbs.}$$

There is also a direct transverse component which is original weight times radians.

$$\text{Transverse Component} = 100 \times .419$$
$$= 41.9 \text{ lbs.}$$
$$\text{Maximum Reaction} = 338 + 41.9$$
$$= 379.9 \text{ lbs.}$$

While a considerable strengthening of the gripes is already indicated, it must be remembered that if the center of gravity of the tender is above the gripes, their foundation also bears a transverse moment equal to the maximum reaction times the height in feet of the C.G. above the gripes. This must always be added, especially with tall erections such as masts or guns.

Assuming that, in the foregoing example, the C.G. of the tender is 1.2 feet above the gripes:

$$379.9 \times 1.2 = 455 \text{ ft. lbs.}$$

The gripes must then be strong enough to withstand a total transverse component of 455 pounds.

Momentum of Boats

The force necessary to bring a vessel to rest in a given distance is somewhat less than that required by the theoretical body moving in frictionless space. This is due to the damping influence of water friction, and in the case of planing hulls, the straight running lines and broad stern cause this damping friction to be disproportionately high. It is therefore realistic to compute planing hull momentum on the basis of speed as 80 per cent of actual. The formula for determining this force is:

$$F = \frac{W \times V^2}{g \times 2D}$$

Where:
F = Pounds of force necessary to bring vessel to stop
W = Weight of craft in pounds
V = Speed in feet per second
g = Acceleration due to gravity
D = Distance in feet in which stop is made

As an example, if the boat weighs 20,000 pounds, is moving at the time at 2 knots and is brought to a stop within a distance of 3 feet, what is the strain on the line between the bitts on deck and the bollard on the pier?

$$F = \frac{20{,}000\ (2 \times 1.689 \times .8)^2}{32.16 \times 6}$$
$$= 758 \text{ pounds}$$

Longitudinal Strength

In calculating the longitudinal strength of a vessel, the simple beam theory proves entirely satisfactory provided the weights assumed are virtual weights. The relationship for any cross section is then:

$$M = Sp$$

or:

$$p = \frac{M}{S}$$

Where:
M = Bending moment
p = Stress in tons per square inch at extreme fiber, C
S = Section modulus

Section modulus is defined in the Load Line Regulations as follows: "The longitudinal modulus, I/C, is the moment of inertia, I, of the midship section about the neutral axis divided by the distance C measured from the neutral axis to the top of the strength deck beam at side, calculated in way of openings . . . Areas are measured in square inches and distances in feet. Below the strength deck, all continuous longitudinal members . . . are included. Above the strength deck, . . . the extension of the sheer strake is the only member included."

The strength calculation for a given hull is based upon the size and strength of each longitudinal member which is continuous throughout a major portion of the hull. For example, consider an 80-foot hull bridged on wave crests 70 feet apart, or 35 feet each away from the center with no support between. The moment may be figured for (1) load increasing uniformly to center or (2) a straight uniform load. Obviously, the weight of the boat is not exactly either, but the assumption of a load increasing uniformly to the center is on the safe side and is therefore considered advisable for the unpredictable stresses set up among waves. The weight is 60 tons, and it will be assumed that the craft is pitching through 5 degrees in a period of 1.2 seconds, its forefoot having just landed on the forward wave. The first calculation is to determine the virtual weight acting as a sagging moment.

$$W_v = \frac{60}{32.2} \times 35 \times \frac{4\pi^2}{1.2^2} \times \frac{\pi}{180} \times 5$$
$$= 1.86 \times 35 \times 27.4 \times .087 = 155 \text{ tons}$$

The sagging moment is then

$$M = \frac{W \times D}{6}$$

for the uniformly increasing load to be assumed. For a straight uniform load, the denominator would be 8 instead of 6. D is the distance in feet from supports.

$$M = \frac{155 \times 35}{6} = 905 \text{ foot tons}$$

Since the structural members of the hull are fir, an allowable fiber stress of 1500 pounds per square inch may be used. Converting 905 foot tons to inch pounds:

$$905 \times 12 \times 2240 = 24{,}300{,}000 \text{ inch lbs.}$$

Dividing the inch pounds by the allowable fiber stress, the required section modulus is obtained:

$$S = \frac{24{,}300{,}000}{1500} = 16200 \text{ in.}^3$$

The strength calculation is as follows:

	(1)	(2)	(3)	(2) × (3)	$\frac{BD^3}{12}$
Member (one side)	Area″	n''	Mom	$f(I)$	$f(I)$
Deck Stringers	25.6	120	3,070	370,000	
Deck Planking	340.0	120	40,800	4,900,000	
Clamp	20.5	110	2,250	247,000	197
Side Planking	80.0	41	3,280	134,100	44,400
Longitudinal Bulkhead	80.0	41	3,280	134,100	44,400
Chine	16.5	22	364	8,000	
Engine Stringers	28.7	16	457	7,300	360
Bottom Planking	365.0	9	3,280	29,600	
	956.3		56,781	5,830,100	89,357
				89,357	
				5,919,457	

$$
\begin{aligned}
\text{Neutral Axis, } n, &= \frac{\text{Mom}}{\text{Area}} \\
&= \frac{56{,}781}{956.3} \\
&= 59.3 \text{ inches above base} \\
An^2 &= 956.3 \times 59.3^2 \\
&= 3{,}355{,}000 \\
I &= 5{,}919{,}457 - 3{,}355{,}000 \\
&= 2{,}564{,}457 \text{ in.}^4 \\
\text{Max. Fiber, } C &= 120'' - 59.3'' \\
&= 60.7'' \\
S &= \frac{I}{C} \\
&= \frac{2{,}564{,}457}{60.7} \\
&= 42{,}300 \text{ in.}^3 \\
\text{Times 2 for both sides} &= 84{,}600 \text{ in.}^3 \\
\text{Safety Factor} &= \frac{84{,}600}{16{,}200} \\
&= 5.2
\end{aligned}
$$

Such an excess of strength has frequently failed in boats with bad pounding bottoms. On the other hand, properly designed hulls have always had sufficient strength with a safety factor as low as 3.

Dynamic Lift

The displacement hull is limited to carrying loads exactly equal to the weight of water displaced. While a planing hull technically can do no more, it can be loaded to a draft which would leave too little freeboard for safety on a displacement hull, and then by its dynamic

lift, add appreciably to its load-carrying power. Furthermore, it will carry this increased load with greatly increased stability over that possible in any displacement type. That is why good planing hulls have the potentiality to mount tremendous military weights on deck without becoming topheavy or losing their priceless ability to bank inboard on turns.

At low planing speeds the greater portion of the load may be carried by displacement. The proportion changes as speed increases until at speed-beam ratios of 3 and upward, dynamic lift alone should support the entire weight. While the conformations of various planing bottoms always add to the known weight some degree of unknown suction load,

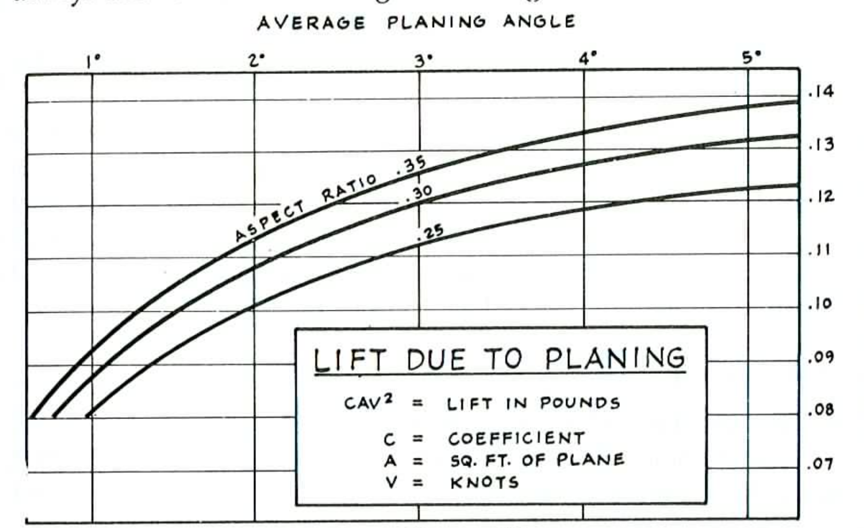


FIGURE 66

the lifting capacity derived from planing action, as demonstrated in present-day good practice, may be readily approximated from the chart of dynamic lift shown in Figure 66. Bear in mind that this is not a chart of maximum or even recommended loads. It is simply indicative of the proportionate weight ordinarily lifted by the planing action alone, exclusive of any weight further supported by the buoyancy of displacement.

For example, a 40-foot cruiser having 290 square feet of plane area and powered for a minimum planing speed, say 18 knots, is presumed to trim up by the bow so that her running lines present an average of approximately 2 degrees of virtual planing angle. If her aspect ratio is .35, the coefficient shown on the chart is .115, and the lift in pounds due to planing alone is:

$$L_p = .115 \times 290 \times 18^2$$
$$= 10{,}800 \text{ pounds}$$

Obviously, this cruiser can weigh much more. From the chart of Maximum Loading in Figure 14, it is apparent by interpolation that she can be successfully loaded to 10 tons or 22,400 pounds. The difference of 11,600 pounds is being carried by displacement.

On the other hand, at a speed-beam ratio of 3, or 32 knots, this same cruiser has a dynamic lift of:

$$L_p = .115 \times 290 \times 32^2$$
$$= 33{,}500 \text{ pounds or } 15 \text{ tons}$$

which is exactly the total maximum loading shown in Figure 14 for that speed.

For larger craft at high speed-beam ratios, the dynamic lift due to planing alone may commonly exceed the recommended maximum loads shown in Figure 14. This is because at high speeds, the potential dynamic lift becomes so great that, were it fully matched with actual load, the craft when not planing might be burdened with a load beyond the best limits of her buoyancy alone for safe and agile operation.

The Speed of Planing

The question thus arises as to the exact definition of true planing, since it now becomes obvious that many fast boats with good planing bottoms actually derive as much or more of their load-carrying power from displacement as from dynamic action. From a purely mathematical standpoint, planing action begins under a suitably shaped hull, at that point where speed-length ratio becomes imponderable. That speed-length ratio (speed over the square root of length) is apparently 2. But this is only where the dynamic action begins. As indicated in Figure 66, the dynamic lift, or planing action, once developed, continues to increase as the square of the speed, other factors remaining the same. It is thus theoretically possible to compute the exact speed at which dynamic lift matches the weight of the boat and at this point to have an apparently 100 per cent planing hull.

However, other factors do not remain the same. The planing angle varies. The suction load and the effect of weight distribution are also variables. In fact, the net effect of these variables is undoubtedly to combine so that 100 per cent pure planing becomes a physical impossibility. To be technical, therefore, a planing hull is simply one so shaped that a degree of dynamic lift is added to its natural buoyancy during the time when its speed of advance exceeds that rate at which solid water can close in abaft of it.

In practice, the tendency toward dynamic lift, with hulls of suitable shape, becomes pronounced at speed-length ratios of 2 or speed-beam ratios of 1.6. This, then, is the minimum speed of planing tendency, and since hulls must be particularly adapted in shape to react to the lift, such hulls are properly classed as planing hulls.

Gyroscopic Effects

The high rotative speeds toward which modern engines are tending produces definite gyroscopic effects, increasingly pronounced as higher speeds are attained. At the present time, the usual range of rotative speeds, weights and radii of gyration are not great enough to have more than moderate effect on the hull performance in an overall degree. However, even the present magnitude of gyroscopic effects is definitely a factor of hull performance during the execution of sharp turns and while pitching. The required strength of engine mountings is also affected.

The characteristic of a gyroscope which is responsible for the effects noted on planing hulls is that of *precession*. That is, when an external force is applied to the end of the axis of rotation, the point of application does not tend to move with the force, but instead moves at right angles to it in the direction of rotation.

A high-speed crankshaft becomes a very effective gyroscope and, when mounted longitudinally in a hull, resists pitching or rudder action according to the law of precession. Thus a left-hand engine, that is, one rotating counterclockwise when looking at its forward end, will tend to swing to starboard when the bow of the boat is lifted on a wave. Similarly, in a sharp turn to starboard, the forward end of the engine will tend to bear downward. In single-screw boats this precession is transmitted to the hull and can have noticeable effect on performance. With twin screws of opposite rotation, precession in one engine cancels out that of the other and only internal stresses and engine foundations are subjected to additional strain. In PT boats, where it was common to install three engines all of left-hand rotation, the effect of precession was most pronounced. All of these boats showed the tendency to level off on starboard turns and to lift their bows and squat even more on port turns. For this one reason, port turns were sluggish in comparison to starboard turns. Furthermore, the difficulty of holding a course was in a large measure dependent upon the type of sea encountered. For example, in long swells where the bow climbs to a crest and then, shooting out into space, drops suddenly, it tended to veer off to port and land in the water somewhat to the left of its original course. This tendency caused by precession necessitated continuous right rudder in addition to the right rudder normally carried to compensate for the right-handed wheels. The extra rudder drag accounted for a 3 to 4 per cent loss of speed. Had two of the engines been of right-hand rotation, therefore turning left-hand wheels, and only one left-hand engine retained, a better overall balance would have been achieved. For best results, only twin screws with two or four engines should have been installed.

Typical Gyroscopic Calculation

For purposes of estimating the torque produced by any gyroscopic couple, the following relationship is true:

$$T = Iw_1\, w_2$$

Where:

T = Torque
I = Moment of inertia of a uniaxial body, about the axis of symmetry and through the center of gravity
w_1 = Angular velocity of the uniaxial body
w_2 = Rate of precession

If the torque, T, is measured in foot pounds, the dimensions of the moment of inertia, I, must be in slug feet squared, while angular velocity and rate of precession are measured in radians per second. A slug is simply weight in pounds divided by 32.16. A radian is pi times degrees over 180.

As an example of the method of computing the torque and its reaction on the bearings of a crankshaft of an engine mounted longitudinally in a planing hull during a normal turn, the following data applies:

(1) Weight of rotating member : 500 lbs. = 15.5 slugs.
(2) Speed of rotation : 2400 R.P.M. = 252 radians per second.
(3) Radius of turn : 120 feet.
(4) Speed of turn : 50 feet per second.
(5) Angular velocity of hull : 50 ft./sec. ÷ 120 ft. = 0.416 radians/second.
(6) Radius of gyration of crankshaft and flywheel : 1 foot.
(7) Rotation of crankshaft : counterclockwise from bow.
(8) Moment of inertia : 15.5 slug feet^2.

Substituting this data in the equation for torque:

$$\begin{aligned} T &= 15.5 \times 252 \times 0.416 \\ &= 1630 \text{ foot pounds of torque} \end{aligned}$$

In this example, it should be noted that since the radius of gyration was 1 foot, the moment of inertia, slug feet squared, is of course the same as slugs, the crankshaft weight divided by gravity. This torque is exerted as a couple by the hull on the crankshaft and is the torque necessary to cause the crankshaft to precess as the hull makes its turn. If this turn is to port, the reaction forces of the crankshaft on the hull are upward at the forward bearing and downward at the after bearing. With a turn to starboard, the forces are reversed.

It thus becomes obvious that in the interests of best performance at sea, all high-powered, high-RPM engine installations should be equally balanced, opposite rotation units. It is also advisable in these installations involving high rotative speeds, to check the torque of precession and allow adequately for its reaction in the entire engine mounting system.

peller is to be turned at 2000 R.P.M. by a 1000-horsepower engine. The horsepower per hundred revolutions is:

$$1000 \div \frac{2000}{100} = 50$$

Coming in from the right of the chart at 50 horsepower per hundred revolutions, it is seen that a monel or high tensile bronze shaft should be 2 3/16 inches in diameter, or a tobin bronze shaft should be 2 7/16 inches in diameter.

Dropping straight from the point of intersection, it is found that bearing spacing of around 7 feet is recommended. These proportions do not follow exact calculations for any theoretical safety factor. Rather, they are based on actual performance of present-day equipment in general war service.

For strut bearings, cutless rubber performs very well. It is probably of some help in isolating vibration due to minor cavitation, but mainly its advantage lies in the protection afforded from the scoring of shafting. Molded plastics such as Micarta and Formica also perform very well.

Muff couplings outside the hull should be strictly avoided and struts forward of the propeller should be minimized. No strut forward of a wheel should be closer than one and one half inches for each knot of designed speed. Thus a 30-knot hull should be fitted with no strut ahead of the wheels closer than 45 inches. The main strut should be aft of the wheel for best efficiency.

Chapter 13

STEERING

Among other shortcomings of planing hull design in the past, steering has been one of the most obvious. It has been almost axiomatic that high-speed craft steered indifferently to port, sluggishly at low speeds and not at all when backing down. These are serious flaws to have been encountered so frequently, especially since the planing hull has potentially the highest type of maneuverability. Therefore the study of steering and the design of steering equipment merits unusual attention.

Particularly bad, because of its danger at high speed, is the tendency of some rudders to "let go" after starting a sharp turn. These rudders give a fast initial turn but after 20 to 90 degrees of change in course, they simply lose their grip and cease to give directional stability, leaving the boat floundering out of control.

This letting go, or burbling, is due in some degree to the location of the blade in an area of low pressure such as occurs under warped plane bottoms. A further cause is incorrect shape of the blade itself, although even the most perfect rudder has a maximum angle of swing beyond which burbling will start.

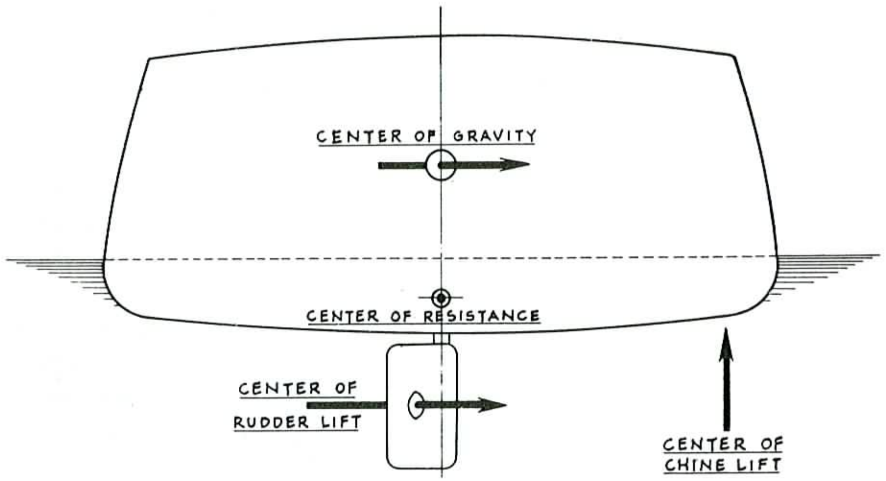
Given a bottom which is free from undue pressure variations, the steering of a planing hull can be made more responsive and far safer than was ever attained with displacement craft. The planing hull's great power, when directed into a properly controlled turn, gives a handling assurance unmatched by slower vessels.

Rudder Forces

In Figure 88, Direction of Forces in Turning, is shown the result of laying the rudder blade of a planing hull. When the blade is laid to port, it sets up a force in the water which reacts on the hull and throws the stern to starboard. This reaction is translated into angular motion which in turn causes the starboard chine to become a leading edge of the plane.

The hydraulic force on the rudder acts at the center of rudder pressure. The distance between the center of lateral resistance of the hull and the center of rudder pressure is a lever arm tending to bank the hull inboard on the turn. Opposing this tendency is the lever arm upward to the boat's center of gravity. The moment of centrifugal force, acting at the center of gravity, tends to give the hull an outboard list in proportion to the square of the speed and tightness of the turn. In displacement hulls of every type, the moment of centrifugal force commonly exceeds rudder force to such an extent that turns are sickening and the outboard list sometimes becomes dangerous.

DIRECTION OF FORCES IN TURNING



SHOWING LEFT RUDDER AND THE RESULTANT LEADING EDGE EFFECT OF THE STARBOARD CHINE

FIGURE 88

That the opposite is true of a planing hull, since turns are made up against the safe inboard slope of a self-created wave, is due to the great lift of the leading chine. This lift is frequently ten times that of the rudder pressure alone and should always be enough to overbalance centrifugal force on any properly loaded planing hull. Rudder force and chine lift therefore operate together and, in conjunction with each other, should provide the necessary banking force to allow safe and sharp turns under any sea conditions.

HULL REACTIONS IN STEERING

When the planing hull banks on its turn, the stern rides up on its own self-created wave along the outboard chine. The pattern of pressures and suction which are created varies according to the contour of the plane and shape of appendages, but in general acts according to hull type. In Figure 89 is shown a diagram of the forces which are set up when an excessively flat vee-bottom hull turns to starboard. The transom swings to port and banks inboard, supported on an area of high pressure along the leading chine and a smaller pressure area along the inboard chine. Both pressure areas taper off sharply to an area of partial suction along the middle of the plane. Sea conditions will vary this pattern momentarily according to the whimsy of wave

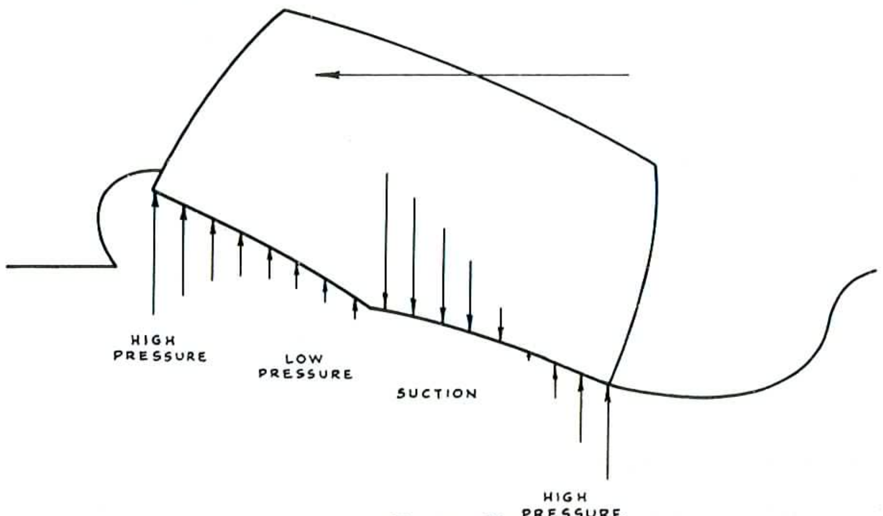


FIGURE 89

contours, and herein lies a potential danger. The moments of its widely spaced points of support are so great that should one of them be wiped out by wave action, capsize is a possibility.

The sharp chine also tends to plough into its banking wave and cause the hull to trip, or roll over with centrifugal force. Elimination of the tendency to trip has been overcome in some hulls by beveling the after chine. Another way has been to cramp the beam of the afterbody. But a narrow afterbody reduces the angle of attack of the leading chine which in turn cuts down its leading edge action and power to lift.

In Figure 90 is shown the afterbody of a plane having good deadrise and generally monohedron characteristics, banking on a turn. The entire after-plane area has lift and therefore resists squatting with its consequent drag on speed. The rounded chines avoid any tripping tendency, and the usually wider beam of this type aids the outboard chine in benefiting from leading edge lift. Without the successful combination of these desirable features, that is, the wide, suction-free stern and the rounded, high-lift chines, turning may be well banked but the excess drag of the stern will cause an unnecessary loss of speed with a consequent sluggishness comparable to that of displacement craft in maneuvering.

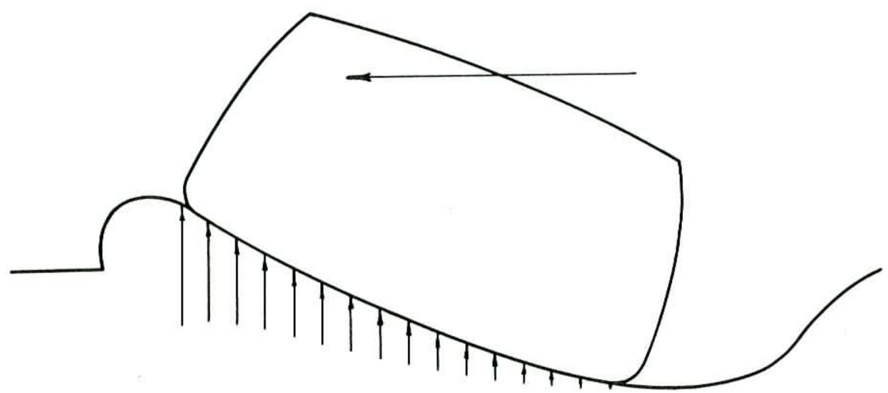


FIGURE 90

SPEED WHILE TURNING

Even in trailing position the rudder causes resistance and this resistance increases markedly as the blade is put over. An appreciable reduction in boat speed is thereby induced on turns. A further speed reduction results from the angular motion of the plane against some of its own lateral plane resistance. The final factor in speed reduction may be attributed to the increase in virtual weight of the boat. This effective increase of weight may in turn be separated into two causes: first, the action of centrifugal force which pushes the hull into its banking wave; second, the scoop action of a rudder mounted on the usual vertical stock. This downward scoop action occurs as a result of the stock being mounted normal to the still waterline. Therefore, when a given planing angle is attained, the stock assumes a corresponding

rake and the rudder delivers a definite downward component, proportional to the sine of the virtual planing angle.

Obviously, weight increase due to centrifugal force against the banking wave must be accepted as inherent in the turning process.

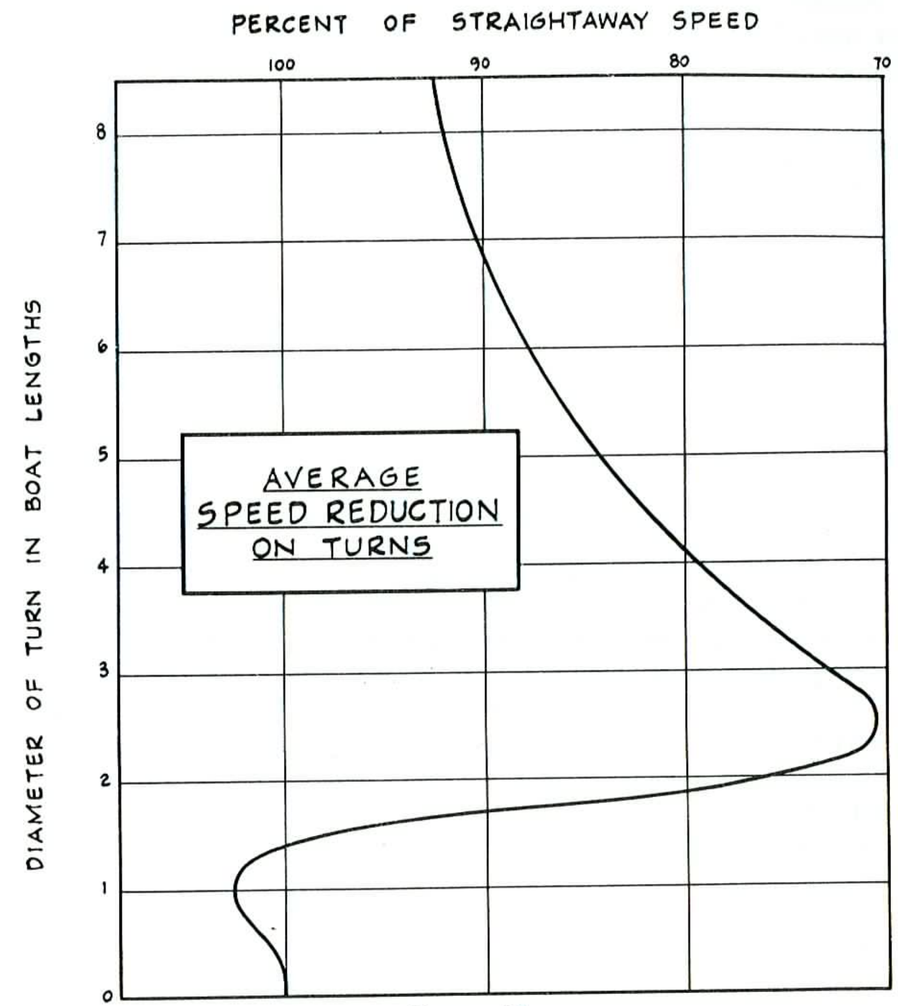


FIGURE 91

However, the weight added by scoop effect can definitely be eliminated in design by the simple expedient of eliminating the rudder stock rake assumed during planing. Since the virtual planing angle of the hull is known, this angle, properly less than 2 degrees, should also be given to the rudder stock. In other words, the profile of the stock will present a reverse rake with its head slightly forward of its bottom end while the hull is at rest, or vertical while the hull is in planing position.

Assuming a rudder normal force of 2740 pounds on a given rudder set normal to the still waterline, the downward component at a 2-degree planing angle would be:

$$2740 \times \sin 2° = 95 \text{ pounds}$$

which represents unnecessary weight added during turns.

The average speed reduction on turns of the best hulls is shown in Figure 91. The curve is based on the diameter of the turn in terms of boat lengths and shows the final or equilibrium speed after making a full about turn or half circle. It will be seen that making such a turn in two and a half boat lengths results in a speed reduction to 70 per cent of the original straightaway speed. This percentage is the best average performance recorded to date.

Improperly balanced hulls such as those which have their centers of gravity aft of their true centers of lateral plane are frequently subject to making snap turns. Hulls with stepped bottoms, due to the very lift which makes them fast on the straightaway, have the least lateral plane aft and are usually the worst offenders in making snap turns on the dead flat. But if even a slight banking wave can be induced, a turn can be made in perhaps two boat lengths entirely without skid and in perfect safety. The flat skid, or snap turn, if completed, is the result of improperly related forces in a precarious state of equilibrium, allowing the stern to be flung about as in a bad yaw which winds up with the boat making sternway. During the excitement of such a skid the boat's speed, as far as the stern is concerned, has momentarily increased by grace of smooth water, as shown by the curve opposite the one boat length turn.

Good planing hull performance calls for sufficient rudder power, integrated with chine lift, to make a full about turn in two and a half boat lengths, together with smart banking on turns of any diameter. The instantaneous lift of the outboard chine with the first movement of the rudder is essential to the fast seagoing hull.

In Figure 92 is shown a diagram of the hull position on a turn. Assuming full right rudder, the hull skids out to port, making the port chine a leading edge and receiving from it tremendous lift, tending to cause an inboard bank. As the turn is made, the hull continues to

maintain its relative angularity with the turning circle. The tactical diameter is the distance between two straightaway courses in opposite directions. The final diameter is the smaller distance resulting from a full circle turn.

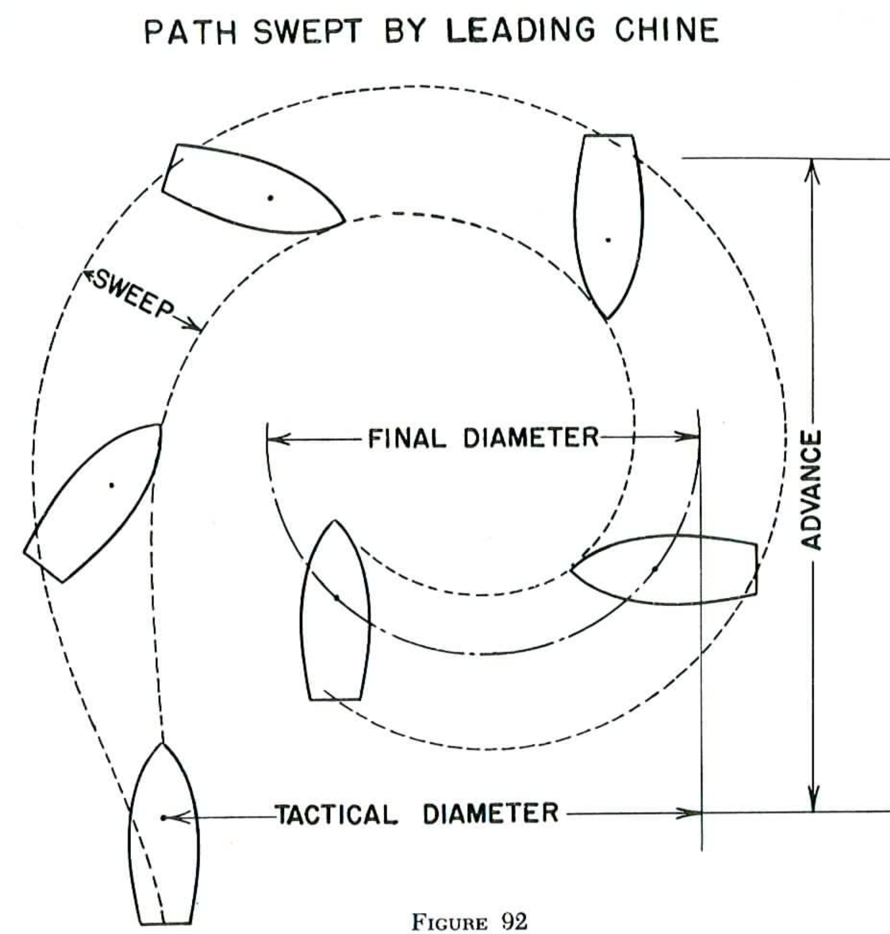
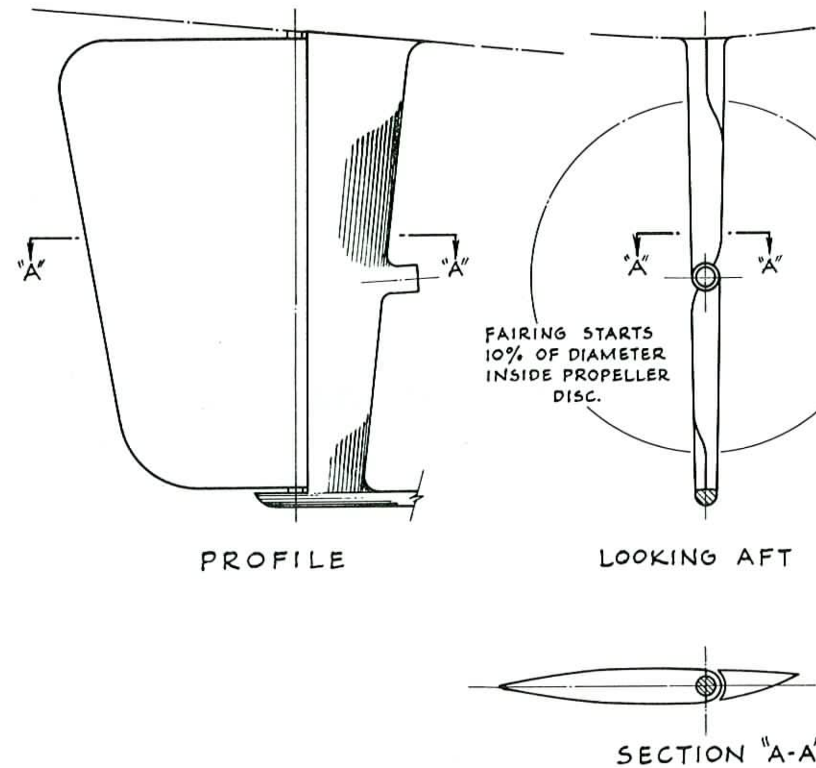


FIGURE 92

RUDDER SHAPES

Probably no other type vessel has been subjected to so desperate an assortment of rudders as have planing craft. This is, of course, due to endless cut-and-try attempts to make rudders perform against hulls that protest. However, there are no secret phases of rudder design other than the basic "feeling" which the designer must have for every part of his creation.

Blade area must, of course, be adequate, but it should also be proportioned efficiently. Like the sail or airplane wing, high aspect ratios (span divided by chord) give higher lift values per unit of area. For single-screw hulls, the rudder shown in Figure 93 has much to recommend it. The blade is deep with an aspect ratio of nearly 2.

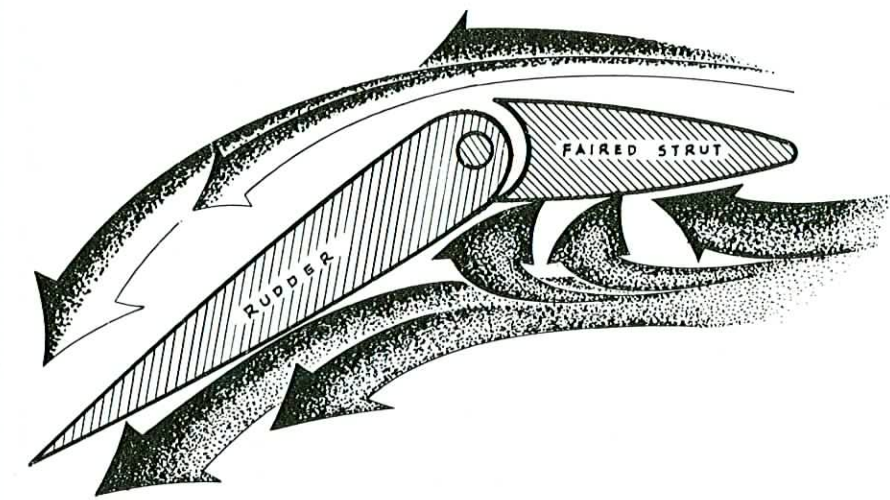


RUDDER & STRUT FAIRED
for RIGHT HAND SCREW

FIGURE 93

By fairing the blade into the strut, a peculiar advantage is obtained in that the lateral area of the strut adds as much turning effort when the rudder is laid as if its area were actually in the blade itself. This advantage in turning effort is available only when strut and blade are faired together without space between, thus restraining the water from

flowing around and abaft the leading edge. Strut area up to at least 20 per cent of the total area of strut and blade combined seems to add about as much lateral thrust, or turning tendency, as if the strut area were an integral part of the blade. This action is illustrated in Figure 94. Here it is shown how an appreciable proportion of the fast-moving water striking the blade tends to be deflected forward across the ship's center line and will add lateral thrust if there is deadwood or a strut for it to react against. By appreciably reducing the required area of



REACTION OF CLOSE COUPLED STRUT OR
DEADWOOD IS TO ADD 20 TO 25 PERCENT
TO LATERAL THRUST OF RUDDER

FIGURE 94

movable blade, the net result on effort to lay the rudder is similar to that of a larger blade with 20 per cent or so of balance.

The lateral area of the strut in a strut-rudder combination may be anything required to achieve reasonable strength, but bearing in mind that strut area beyond 20 per cent of strut and blade combined is of diminishing value in added lateral thrust, the chord length should be held within the shortest possible dimension consistent with a suitable hydrofoil section.

There has been considerable experiment to determine the best sectional shape for high-speed rudders and wide differences of opinion exist. As in all such cases where experimental results conflict, extension of the scope of agreed hypotheses is indicated. Commander Peter Ducane in England adapted the old-fashioned "fishtail," or tugboat, rudder to planing craft and felt that it was superior to either a flat plate or to a hydrofoil section type. Later, Frank Huckins in this country perfected the type and his results have demonstrated one of the best of the balanced type rudders to date.

In section, the fishtail rudder is a narrow wedge with the point forward. The theory is that in splitting the water at high speed, the wedge section, while admittedly leaving a bubble of suction behind its broad after edge, has parted the water at such a gentle angle that the bubble is only the exact width of the rudder's after edge. On the other hand, a hydrofoil rudder blade is much thicker toward its leading edge, causing a wider angle of split in the water, and since a rudder may be quite short in its chord length, the water might not close in fast enough to follow its relatively fat contour. The resultant bubble in such cases is therefore said to be larger than that left by the fishtail type in outboard installations.

But, to examine the facts fully, it must be noted that large ships with speeds of 30 knots and upwards are invariably fitted with rudders of hydrofoil section because of their demonstrated lower resistance. The apparent gap between these two schools of thought must be bridged by the process of relating the speed of the vessel to the rate of resurgence, or closing in, of the water. Obviously, the higher the speed the longer must be the tail of the hydrofoil. This relationship, within the range of planing speeds, has been closely approximated by a series of practical tests conducted beneath the clear waters of a Hawaiian lagoon. This series of tests was made with six full-sized rudders varying only in thickness. One rudder at a time was suspended 2 feet below a surfboard and towed at varying speeds on an outrigger from a PT boat. The observers, in diving helmets, noted the resurgence, or lack of it, as indicated by any marked trail of appended bubbles. The results are plotted in a curve of least resistance in Figure 95, and are applicable to submerged rudders.

According to this curve, a hull moving at 35 knots in warm, tropical water may expect to avoid any appreciable bubble of suction behind its appendages if they are hydrofoiled to a maximum thickness ratio of

13.7 per cent. Cold water, being more dense, would require a very slightly thinner section.

On the smaller craft, with high aspect ratio rudders, the optimum percentage of thickness over chord would seldom allow sufficient mass thickness in the blade to give adequate strength. But by increasing the chord length through fairing the strut into a hydrofoiled entity with the rudder, adequate mass may nearly always be attained. In this way,

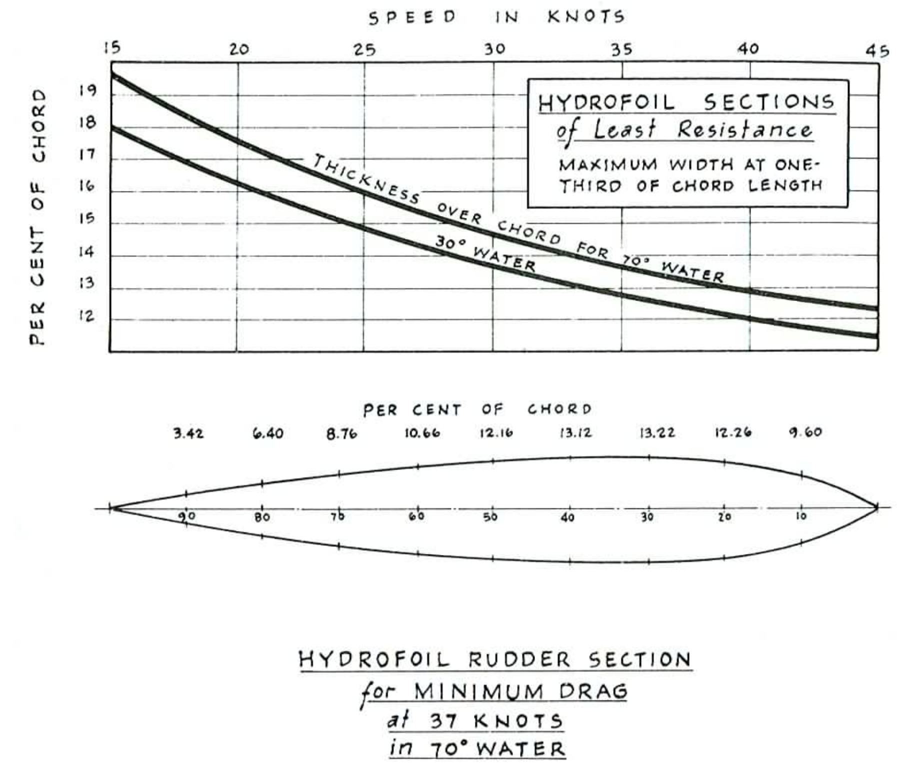


FIGURE 95

the strut-rudder combination, faired as a unit hydrofoil appropriate to the speed, offers the possibility of an appendage unit of adequate strength together with the shorter chord necessary for a high aspect ratio of the blade itself.

For outboard rudders extending above the surface of the water, the fishtail section shows a slightly lower resistance. For spade rudders, a wedge angle of about 4 degrees for the fishtail has proved satisfactory.

Balanced Rudders

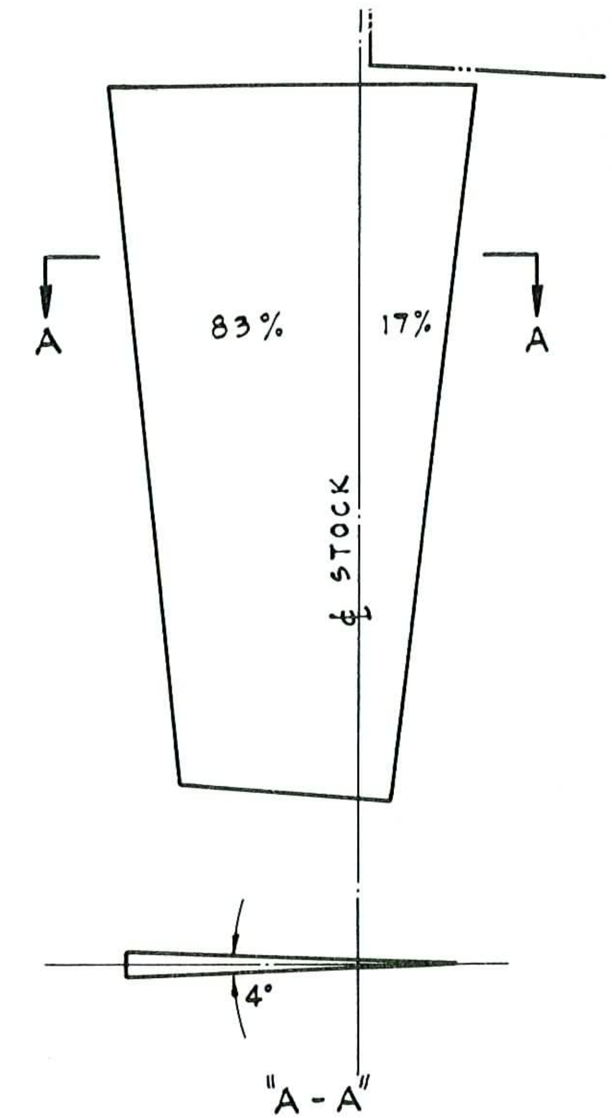
Dagger, or spade, rudders are usually balanced with a small part of their area forward of the stock and therefore cannot be incorporated with a faired strut as previously described. On fast liners and large naval craft a balanced type of rudder is hydrofoiled into a strut-rudder unit by having the strut extend down only to the shaft. This puts the lower pintle about a third of the way up from the bottom of the blade and allows a considerable area of the bottom portion of the blade to extend forward of the stock and thus balance the after area. But such a rudder in small planing craft invites fouling between what are in effect scissor blades, and it may become impossible to return the rudder to midships.

The Huckins balanced rudder is shown in Figure 96. Due to the fishtail section, or wedge shape, the center of pressure when trailing is somewhat farther aft than with other sections. Hence a balancing area of 17 per cent, which would be excessive with the flat plate type and completely out of the question with hydrofoils, is easily carried and contributes greatly to the ease of steering.

The maximum area of balance which may be incorporated depends upon the fore-and-aft location of the center of pressure, both when the blade is trailing and when it is hard over. Obviously, a center of pressure forward of the stock would impart a negative turning moment, making the steering of a straight course dependent upon the whimsy of backlash in the steering gear. Large ships are sometimes deliberately provided with rudders slightly overbalanced in order to reduce the torque on the stock when backing down. However, the comparative speed at which planing hulls move astern renders any such consideration quite pointless. On the other hand, their high speeds forward make rudder steadiness of great importance.

As shown on the curves at the bottom of Figure 98, the center of pressure on a 4 per cent wedge section rudder moves from 22 per cent aft of the leading edge at trailing to 42 per cent aft at hard over. If, for example, the center of the stock were at a point 25 per cent aft of the leading edge, the rudder would tend to oscillate away from midships, a tendency that would have to be restrained by gear linkage necessarily subject to more or less backlash.

With flat plates the center of pressure is slightly farther forward, especially at trailing, with the consequence that even less balance area may be incorporated. Hydrofoil sections will develop a center of pres-



HUCKINS BALANCED RUDDER

WEDGE-SPADE TYPE

Figure 96

sure around 10 per cent aft of the leading edge while trailing, and the resultant permissible balance is extremely small. It is the location of the center of pressure at trailing and not its location at hard over which must be considered when determining the maximum balance area.

In general, use of the customary balanced rudder on high-speed hulls has three disadvantages. The first is that a balanced rudder ordinarily forces the use of a strut forward of the propeller. The second is its excessive resistance when turning, due to the fact that as much as one quarter of the deflected water stream is directed somewhat forward and therefore against the direction of turn. The third disadvantage of the balanced rudder is its customary lack of effectiveness when backing down. It seems to be an invariable characteristic of high-speed hulls fitted with balanced rudders that directional control when moving astern must be accomplished either with the engines or with the spasmodic "kick ahead" routine. On the other hand, a greatly improved control while going astern with unbalanced rudders has been repeatedly demonstrated. To hazard a guess as to the causes underlying this performance, it may be due in part to the fact that when going astern the rudder has no beneficial reaction from a powerful slipstream, but must react entirely on relatively still water. Merely to deflect the path of some of this slow-moving water is inconsequential in a turning reaction. But if this turning reaction, together with all added resistance to straight-line motion, is applied entirely toward the inboard side of the turn, the total effect is usually enough to give positive control.

Rudder Location

It is obvious that a rudder should be so placed as to derive maximum benefit from the slipstream. This means that it must be directly behind the center of the propeller. Steering to either side, whether there is way on or not, will then be properly effective.

Of great importance is another consideration of rudder location, affecting especially the so-called outboard rudder. Since these rudders extend above the surface, the flow conditions are not the same as for rudders deeply submerged. As has been previously explained, the benefit of high aspect ratio on a fully submerged blade is considerable. This influence is due in part to the formation of vortices at the top and bottom of the rudder with consequent pressure losses. If the surface of the water around an outboard rudder would remain level, the pressure loss at the top would be entirely eliminated. But unfortunately, this is not the fact. At high speeds, any rudder angle so depresses the water level on the after face that an enormous and disproportionate loss in rudder lift is suffered.

For this reason, the area of outboard rudders, and therefore their drag, must be considerably in excess of that suitable for submerged rudders. It should also be borne in mind that the effective area of an outboard rudder cannot extend above the level of the bottom of the transom, regardless of the static waterline.

Contra-Propeller Fairing

When a rudder is set behind the exact center of the screw, which is where it should be, it is in a stream of water whose relative speed is not only considerably greater than that of the boat, but whose direction is a well-defined spiral. Thus a blade streamlined for minimum fore-and-aft resistance only, such as a plain hydrofoil, is actually operating under conditions outside the scope of its design.

With the rudder behind a right-hand wheel, the slipstream is impinging upon the top half of the strut from the port side and on the lower half from the starboard side. Obviously, a rudder streamlined as a single section is causing about the same resistance as if 5 degrees or so of rudder were being carried.

There are two methods whereby this non-streamlined resistance may be reduced. The simpler way is to cut the span of the blade so it does not project below the shaft center. Such a rudder behind a right-hand wheel will be carried slightly to the right for its normal trailing position, as may be observed on small runabouts where the type is in common use. It is thus removed from cross currents and may settle at the angle of least resistance.

However, there are two disadvantages to the short rudder. One is the necessity for increasing chord to attain adequate area. The resultant less favorable aspect ratio does not achieve the positive steering of a rudder which deflects more of the propeller race. A second disadvantage, applying more to large craft with higher centers of gravity, is that the center of rudder pressure is not deep enough to counterbalance the moment of centrifugal force and give the hull a positive inboard bank.

The better method of fairing calls for the careful design of strut and blade faired into a winding streamline section which at every level will part the slipstream spiral evenly. This was illustrated in Figure 93.

A 40-foot hull which originally had a spade rudder hung entirely above the shaft center had a top speed of 26.2 knots and failed to answer smartly to full right rudder. It was found that the blade normally trailed 4 degrees to starboard due to the slipstream spiral. Furthermore, with left rudder, the burbling point was reached when the blade swung to 25 degrees off the center line. Beyond this angle it frequently "let go" and the boat floundered out of control. When the original rudder was replaced with a contra-propeller strut-rudder combination as shown in Figure 93, steering became fully positive to either side and speed increased .4 knot. While the speed increase was due largely to getting the strut aft of the wheel, the equalization of steering was due to contra-propeller fairing.

Rudder Area

The rudder, like the propeller or any other appendage, is a necessary additional resistance, and as such should be made as small as possible within the limits of achieving satisfactory performance. Enough area to turn the boat within a diameter of two and a half times her own length is ordinarily considered highly satisfactory. To turn in a tighter circle than this is of little tactical value but may necessitate a disproportionate increase of rudder area and resistance. The rudder's immediate function is to exert a lateral component of thrust on the order of 4 pounds per horsepower in 50-ton, 40-knot hulls, up to 10 pounds per horsepower in 10-ton, 20-knot hulls.

The area of rudder to exert the correct lateral thrust has been determined from experience and is plotted in Figure 97 for rudders placed directly in the propeller race. For example, a hull of 15-foot beam designed for 30 knots should ordinarily perform at its best with 5.3 square feet of pressure surface. For balanced spade rudders, this total area must be in the blades alone; for strut-rudder combinations, the blades should contain not more than 80 per cent of the required pressure surface, the other 20 per cent of the necessary pressure being automatically delivered by the strut. While the profile area of the strut may be considerably greater in proportion than its arbitrary 20 per cent of lateral thrust would indicate, experience has demonstrated the conservatism of the 80-20 division of lateral thrust. The blades of strut-rudder combinations so sized perform well.

The final criterion of correct rudder area and design is its ability not just to steer but to steer with an inboard bank. To achieve this potentiality of the planing hull, the moment of rudder force plus chine lift must exceed the moment of centrifugal force.

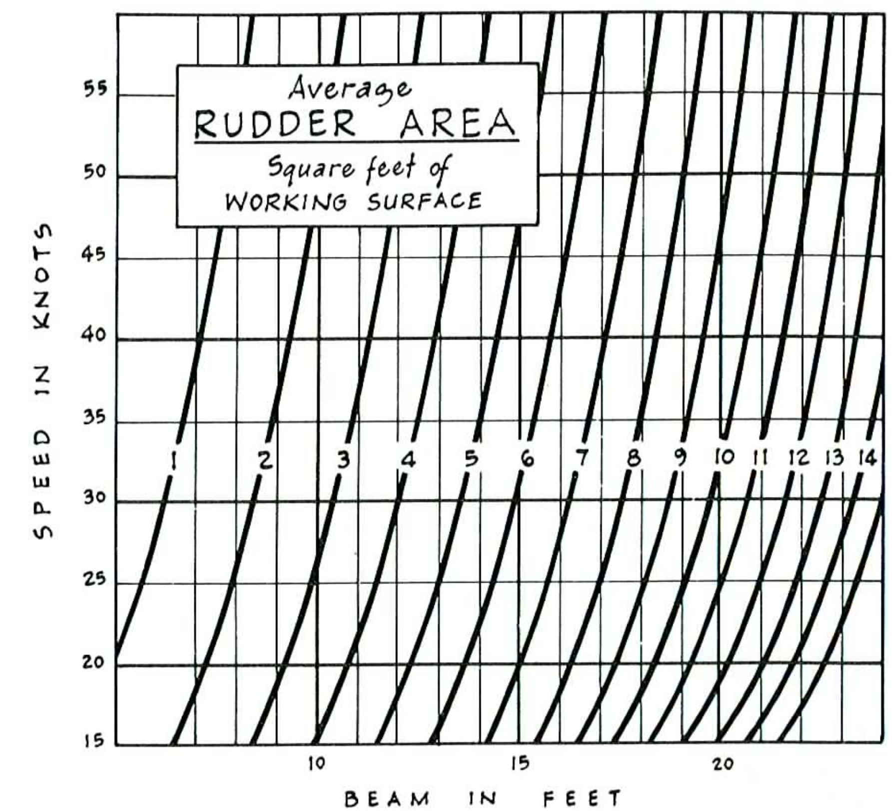


Figure 97

Rudder Forces

The curves shown in Figure 98 are the result of exhaustive experiment in ascertaining the true values of lift and the actual location of centers and burbling points under a planing bottom. It has long been apparent that airfoil data, of which a great deal has been carefully compiled, is not fully reliable when applied to hydrofoils. Rudder design from airfoil data has led to an exaggeration of the normal force,

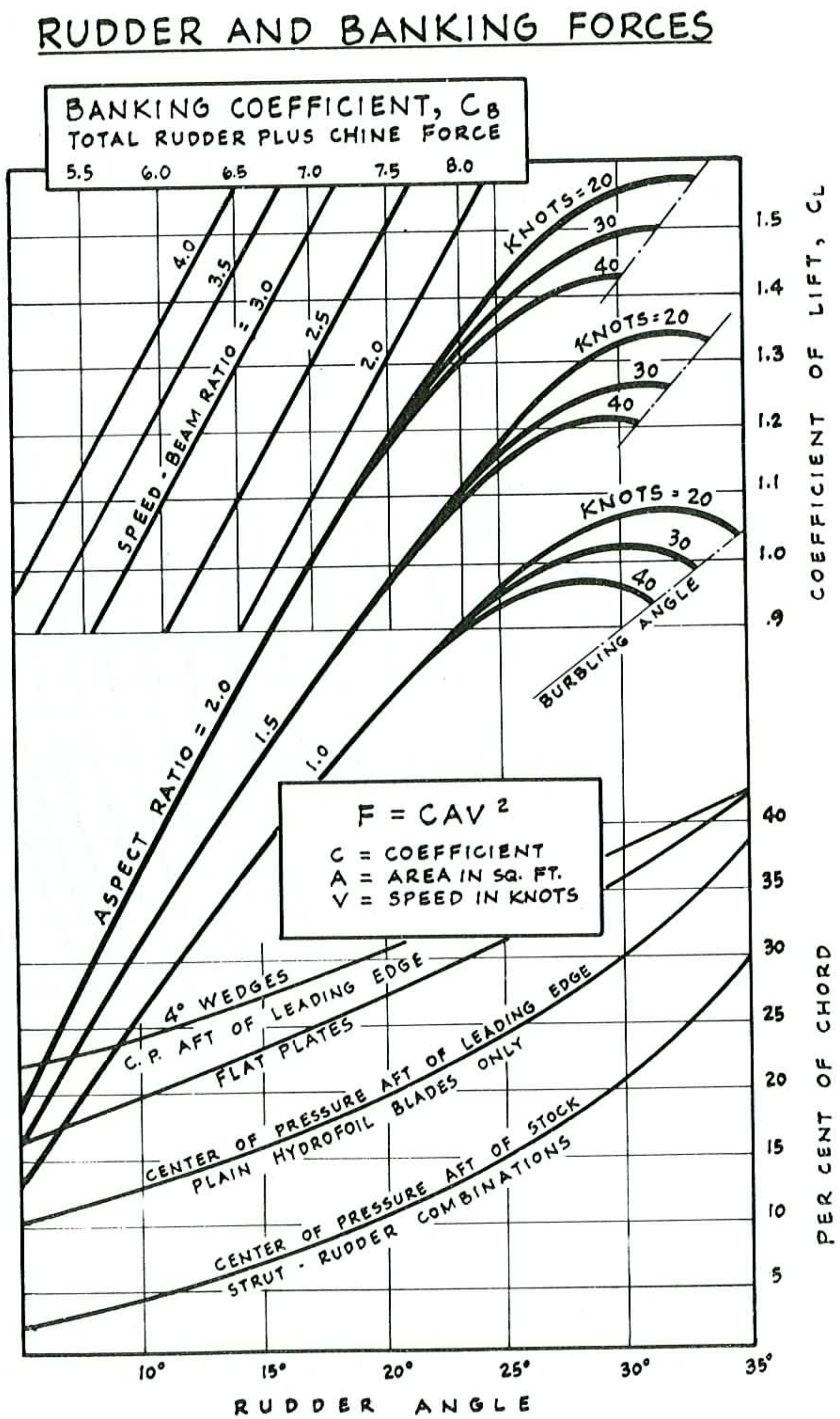


FIGURE 98

a slightly delayed angle of burble and, finally, centers of pressure materially farther aft than is true in actual operation under a planing bottom.

To find the rudder normal force of a blade with 3 square feet and 1.5 aspect ratio under a hull moving at 30 knots, find opposite the top of the 30-knot curve in the 1.5 aspect ratio group, the coefficient of lift, 1.27. Rudder normal force is then:

$$Fn = 1.27 \times 3 \times 30^2$$
$$= 3429 \text{ pounds}$$

It will be noticed that the curve stops at a rudder angle of 32 degrees, the point where a burbling tendency may be expected to begin. The lateral force component will be the normal force times the cosine of the angle, .848 for 32 degrees:

$$Fl = 3429 \times .848$$
$$= 2908 \text{ pounds}$$

If the blade is a balanced wedge of 15-inch chord length, the center of pressure at 32 degrees will be 39 per cent aft or 5.85 inches aft of the leading edge. Since at trailing the center of pressure will move up to 22 per cent, the stock cannot be more than 3.3 inches from the leading edge. The lever at hard over is then 2.55 inches and the twisting moment is:

$$Mt = 3429 \times 2.55$$
$$= 8744 \text{ inch pounds}$$

The comparison of this balanced rudder with a hydrofoil strut-rudder combination is most interesting. The equivalent blade area is 80 per cent of that of the balanced rudder or 2.4 square feet and its chord length is 12 inches, although the hydrofoiled unit is still approximately of 1.5 aspect ratio and the total area is still considered as 3 square feet. Therefore the total lateral force remains the same but normal force is reduced in proportion to actual blade area:

$$Fn = 1.27 \times 2.4 \times 30^2$$
$$= 2740 \text{ pounds}$$

This reduction in normal force represents a direct reduction in resistance to ahead motion by approximately 20 per cent. And from the lower curve in Figure 98, the center of pressure at 32 degrees will be about 24 per cent aft, not of the leading edge, but of the stock. The 24 per cent represents 2.88 inches:

$$Mt = 2740 \times 2.88$$
$$= 7891 \text{ inch pounds}$$

which indicates nearly 10 per cent less torque, or easier steering than the equivalent balanced rudder alone.

Banking Forces

As shown in Figure 88, the direction of rudder and chine forces combine to oppose the centrifugal force set up by turning. The resultant difference between these forces determines whether or not the hull banks properly on turns. In practice, the actual lifting force contributed by the outboard chine varies widely according to the conformation of the planing surface. However, for seagoing types having monohedron characteristics, the banking force can be calculated quite closely from the coefficients shown at the top of Figure 98.

As an example, the banking force of a PT which has 7.5 square feet of aspect ratio 1.5 rudder and slows down to 30 knots, or a speed-beam ratio of 2.2 on full about turns, is as follows:

From the end of the 30-knot, 1.5 aspect ratio curve, move left to the diagonal of 2.2 speed-beam ratio. Above this point find the banking coefficient, C_B, 7.2. The banking force is then:

$$\begin{aligned} F_B &= 7.2 \times 7.5 \times 30^2 \\ &= 48{,}500 \text{ pounds or } 21.6 \text{ tons} \end{aligned}$$

And its moment is this pressure times the combined distance from the center of rudder pressure up to the center of lateral resistance of the hull in planing position, plus one half the waterline beam, thus:

$$\begin{aligned} M_B &= 21.6\,(2.5 + 8.6) \\ &= 240 \text{ foot tons banking force} \end{aligned}$$

The formula for centrifugal force opposing this banking force is:

$$Fc = \frac{W \times V^2}{g \times R}$$

Where:

W = Weight in tons
V = Speed of turn in feet per second
g = Acceleration due to gravity
R = Radius of turn

The boat in question weighs 58 tons and makes a turn of about 4 boat lengths or a radius of 140 feet:

$$\begin{aligned} Fc &= \frac{58\,(30 \times 1.689)^2}{32.2 \times 140} \\ &= 33.7 \text{ tons} \end{aligned}$$

And since the center of gravity is 7 feet above the center of lateral resistance, the moment of centrifugal force is:

$$\begin{aligned} Mc &= 33.7 \times 7 \\ &= 236 \text{ foot tons} \end{aligned}$$

Since the moments are practically identical, no banking tendency is to be expected, a condition borne out by actual performance and due largely to the heavy weights which PT boats carry above their decks.

A fast freight boat which normally banks about 16 degrees inboard, has the following characteristics:

Total rudder area	5.5 sq. ft.
Aspect ratio of rudders	2.0
Speed on turn	23 knots
Speed-beam ratio on turn	2.05
Weight loaded	18 tons
Radius of turn	90 feet
½ beam at waterline	5.6 feet
Center of rudder pressure up	2.8 feet
Center of gravity down	3 feet

$$\begin{aligned} F_B &= 8.2 \times 5.5 \times 23^2 \\ &= 23{,}600 \text{ pounds or } 10.5 \text{ tons} \\ M_B &= 10.5\,(2.8 + 5.6) \\ &= 88 \text{ foot tons} \\ Fc &= \frac{13\,(23 \times 1.689)^2}{32.2 \times 90} \\ &= 9.5 \text{ tons} \\ Mc &= 9.5 \times 3 \\ &= 28.5 \text{ foot tons} \\ \text{Net Mom} &= 88 - 28.5 \\ &= 59.5 \text{ foot tons} \end{aligned}$$

The degree of bank to be expected from this 59 foot-ton inboard tendency is found by the following formula:

$$\operatorname{Sin} \theta = \frac{M}{W \times GM}$$

Where:

θ = Angle of bank
M = Net moment
W = Weight in tons
GM = Dynamic metacentric radius

For the freight boat, the angle of bank is:

$$\begin{aligned} \operatorname{Sin} \theta &= \frac{59.5}{18 \times 13} \\ &= .254 \\ \theta &= 15 \text{ degrees (about)} \end{aligned}$$

Power to Swing Rudder

While present planing hull sizes do not require motor driven steering gears, it is essential to calculate the steering load both in order to size accurately all elements of the gear and also to take advantage of the fastest possible time to lay the rudder. Ten or a dozen turns on the wheel to either side are of no particular consequence in displacement hulls, but the quick maneuverability of planing craft makes it advisable to reduce to a minimum the number of turns on the steering wheel.

The load on the spokes of the steering wheel which a helmsman can conveniently handle should not exceed 30 pounds at hard over. To

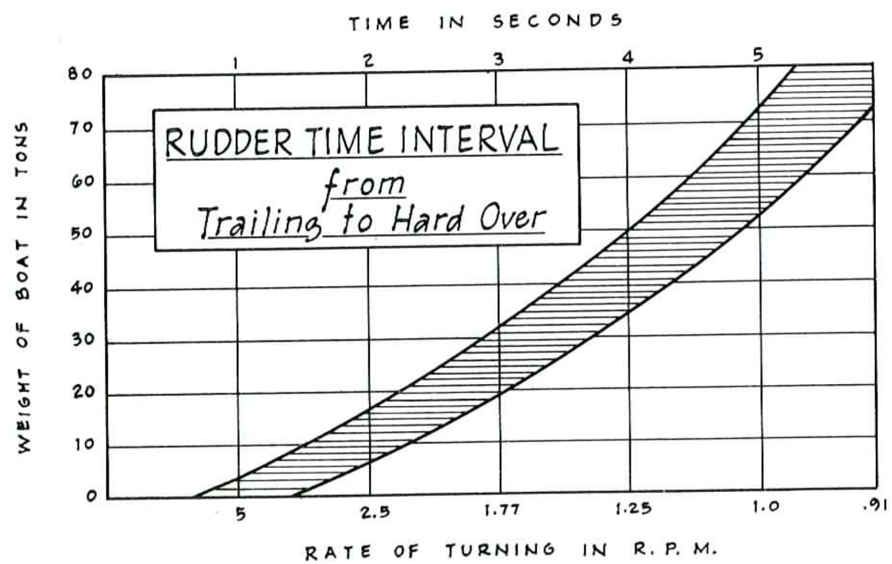


Figure 99

keep below this range, the rudder force must be known and then controlled through proper gear ratios.

For example, the blade of a hydrofoiled strut-rudder unit has a normal force of 2740 pounds. This load is a static force and must have a time element introduced in order to arrive at the power required for turning. The approximate time allowance for laying the rudder is shown in Figure 99. Here it is shown that a 50-ton craft may properly take around 4.5 seconds to lay its rudder from trailing to hard over. Figures along the bottom of the chart indicate that an average amplitude of swing within 4.5 seconds will be at the rate of about 1.15 revolutions per minute. The speed of the load in feet per minute is then:

$$S = \frac{\text{Radius} \times 2\pi \times \text{R.P.M.}}{12}$$
$$= \frac{2.88 \times 6.283 \times 1.15}{12}$$
$$= 1.74 \text{ feet per minute}$$

The horsepower to swing the blade is then the load times feet per minute:

$$\text{H.P.} = \frac{2740 \times 1.74}{33{,}000}$$
$$= .144$$

Or stated in a combined form:

$$\text{H.P.} = \frac{\text{Load} \times R \times 2\pi \times \text{R.P.M.}}{33{,}000 \times 12}$$
$$= \frac{2740 \times 2.88 \times 6.283 \times 1.15}{33{,}000 \times 12}$$
$$= .144$$

or simplified:

$$\text{H.P.} = \frac{\text{Load} \times R \times \text{R.P.M.}}{63{,}000}$$
$$= \frac{2740 \times 2.88 \times 1.15}{63{,}000}$$
$$= .144$$

And torque is:

$$T = \frac{63{,}000 \times \text{H.P.}}{\text{R.P.M.}}$$
$$= \frac{63{,}000 \times .144}{1.15}$$
$$= 7890 \text{ inch pounds}$$

In order that this hard-over pull shall not exceed 30 pounds at the rim of a hand steering wheel, reduction on the order of 260 to 1 must be provided. This is easily accomplished by a worm gear reduction of 22 to 1 plus a steering wheel radius of 12 inches. The gear reduces the rudder torque to around 360 pounds and the 12-inch-radius wheel brings it down to 30 pounds at hard over. With a gear reduction of 22 to 1, the steering wheel will turn slightly more than two and a half revolutions while bringing the rudder from trailing to hard over. Additional reduction to allow for friction losses must be estimated according to the type of gear.

Steering Linkages

The ideal linkage for transmitting motion from the steering wheel to the tiller arm should be semi-reversible, free from backlash or slack, low in friction and permanent in relative setting.

A semi-reversible gear is non-reversible at and around its mid point, but becomes reversible toward the hard-over positions, exactly in the manner of automobile steering gears. The old-fashioned marine system employing tiller rope on a drum is at best but a makeshift for fast boats. Not only is it subject to slack but, being reversible, the tendency toward backlash and "springiness" can be quite annoying.

With a correct worm ratio, the rudder maintains a tendency to return to midships but will unfailingly hold a straight course without guidance from the wheel. If the worm, or cam and roller, is mounted on the rudder stock, the motion to be transmitted from the steering wheel is rotary and requires either bevel gears or chain and sprockets at its forward end. It is ordinarily the simplest installation to mount the steering wheel directly on the worm shaft and use only a push-pull linkage to reach the tiller. This motion can also be very neatly transmitted hydraulically with the advantage of only a tubing connection which may be led around any obstructions.

Twin Rudder Design

Twin rudders are essential for twin-screw installations, but are the source of some excess resistance unless designed to function with a slight degree of mutual independence. It is obvious that the inboard rudder operates within a smaller radius than the rudder on the outboard side of the turn and therefore requires a greater angularity if needless reduction of speed on turns is to be avoided.

The smallest turning circle should be laid out with radii normal to the rudder blades to determine the exact additional angle necessary for bringing the inboard blade into adjustment with the outboard. This will usually be on the order of 5 degrees. That is, with the outboard rudder, swinging inboard, stopped at the burbling angle indicated in Figure 100, the inboard rudder must be allowed an additional 5 degrees of swing before being stopped.

The additional 5 degrees will not cause burbling around an inboard rudder because the angular motion of the hull, keel and appendages preceding it on the turn have already imparted a directional wake effect

to the water stream lines reaching the blade. On the other hand, if the inboard rudder is not given 5 degrees of additional angle, its effect is to set up a force in opposition to that of the outboard rudder. Stops must therefore be so placed that the inboard swing to either side is limited to the burbling angle shown in Figure 98, according to the shape and speed of the blade. Stops for the outboard swing to either side should allow the additional 5 degrees.

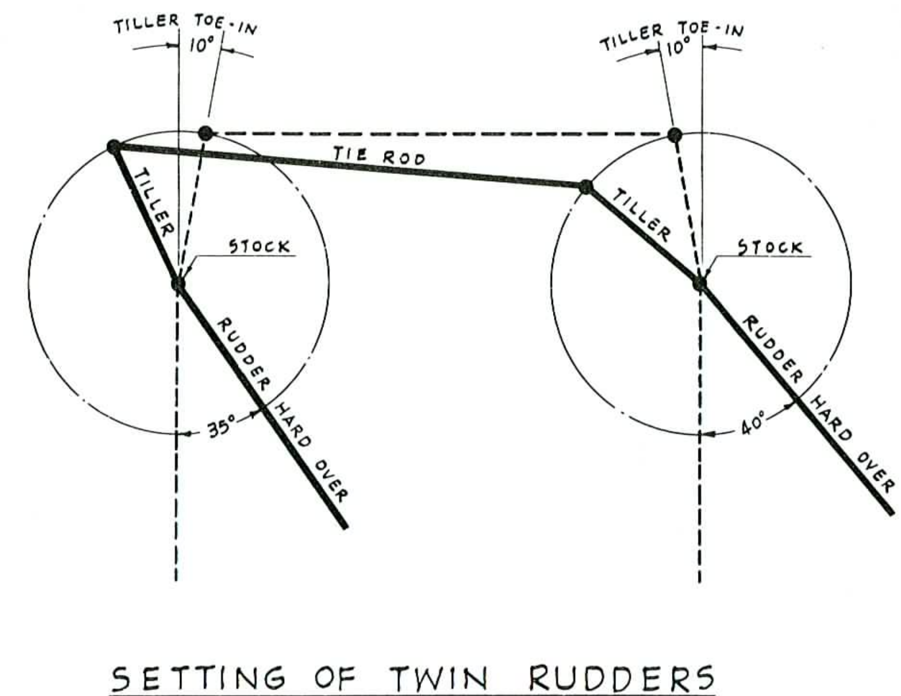


Figure 100

Figure 100 illustrates the necessary tiller setting to maintain rudder angles normal to their radii. When tillers are set forward of the stocks, as in the diagram, a "toe in" is required. If the tiller arms are set aft over the blades, a "toe out" would be used. In either case, the tiller arm must be set at an angle twice as great as the additional angularity to be added to the swing of the inboard blade. In other words, with a required 5 degrees of additional inboard rudder angle, the tiller arms must each be set 10 degrees off the rudder blade center line. With forward tillers, each will have a 10-degree toe in from the center line.

With tillers aft of the stock, each will toe out 10 degrees from center or 20 degrees from each other.

The two tillers should be connected with a tie rod in which a turnbuckle has been incorporated at one end. While planing hull rudders do not operate in the converging wake that closes in behind displacement hulls, there is frequently a minor deviation of currents due to the propeller race. With spade rudders or rudders which do not extend the full depth of the propeller race, the normal trailing position will be 1 or 2 degrees off center, the exact amount to be determined on the trial trip.

For trailing adjustment, the tie rod should be disconnected during a full power straightaway run, the deviation of the free rudder noted and the tie rod turnbuckle adjusted accordingly. Reconnecting the tie rod in this position assures the trailing position of least resistance.

Differential Tie Rod

In order to equalize under all conditions the pressure on each blade of twin rudders, the proposal made by Captain Edwin D. Gibb, U.S.N., merits consideration. While rudders can be theoretically balanced for equal pressures by the methods previously discussed, it is obvious that wave action will cause greater loads on one blade than on the other. Also, on planing hulls, the inboard bank plunges the inboard blade into water of much greater density than that surrounding the outboard blade.

Were the blades free to equalize the total pressure between themselves, their relative angularity would be constantly shifting and a minimum of resistance due to excess rudder angle would result. The differential tie rod accomplishes this constant balance.

Essentially, as shown in Figure 101, the differential consists of an equalizer link set fore and aft with its ends connected by pins to a two-piece tie rod. Motion from the wheel is applied transversely to the center of the equalizer link. Experimental installations of the differential have indicated that the radius of the equalizer link should be held to a minimum consistent with the size of pins required. For example, with a tiller radius of 10 inches, an equalizer link radius of 1¼ inches is sufficient to allow adequate differential operation while moving ahead and yet holds fair control over the rudders when backing down.

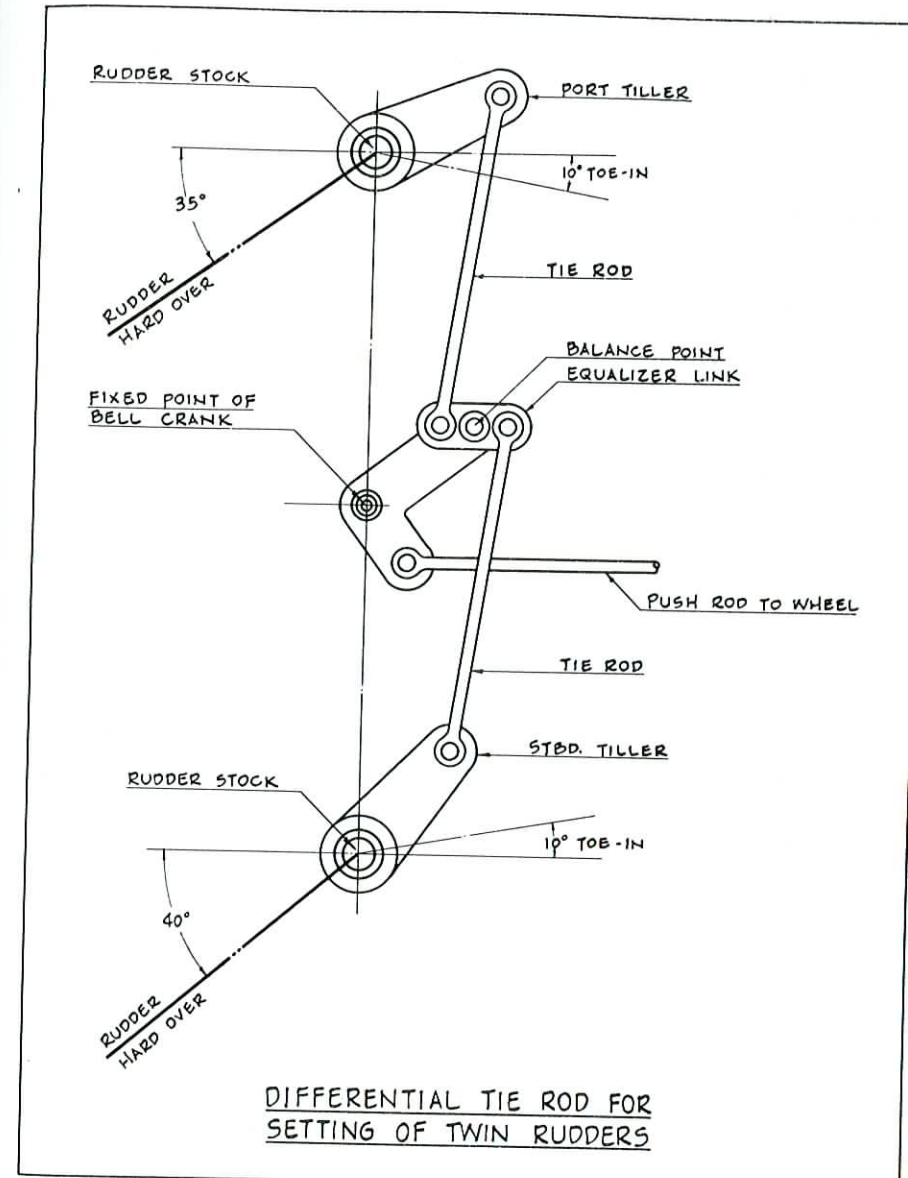


Figure 101

The mechanism allows the blade momentarily subject to the greater pressure to slack off and in so doing, to transfer greater pressure to the other blade, thereby maintaining perfect balance at all times.

Strength of Rudder Stocks

The hollow stock is usually to be preferred over solid stocks due to its great saving in weight. In strut-rudder combinations, the slightly larger diameter of hollow stocks is of relatively slight concern. With the lower pintle bearing thus provided, the stock need be calculated for torsion stress only, according to the following formulas:

$$\text{For hollow stocks, } M_t = \frac{\pi}{16} f \frac{D^4 - d^4}{D}$$

$$\text{For solid stocks, } M_t = \frac{\pi}{16} f\, D^3$$

As an example, the required diameter of hollow bronze stock for a rudder subject to 3000 pounds of normal force in torsion, is as follows:

Conditions:

$$\begin{aligned}
Fn &= 3000 \text{ pounds} \\
\text{Radius of C.P.} &= 4 \text{ inches} \\
\text{Allowable stress} &= 15{,}000 \text{ p.s.i.} \\
\text{Torsion} &= 3000 \times 4 \\
&= 12{,}000 \text{ inch pounds} \\
12{,}000 &= \frac{\pi}{16} \times 15{,}000 \times \frac{D^4 - d^4}{D} \\
&= 2960 \times 4.05 \\
\therefore d &= 1.5 \text{ inches} \\
\text{and } D &= 1.9 \text{ inches (approx.)}
\end{aligned}$$

One and one half inch iron pipe size bronze is therefore satisfactory.

The calculation of stock size for a spade rudder involves both torsion and bending, as follows:

Conditions:

$$\begin{aligned}
Fn &= 3000 \text{ pounds} \\
\text{Radius of C.P.} &= 3 \text{ inches} \\
\text{Bending arm} &= 9 \text{ inches} \\
\text{Allowable stress} &= 15{,}000 \text{ p.s.i.} \\
\text{Stock type} &= \text{Solid} \\
\text{Torsion} &= 3000 \times 3 \\
&= 9000 \text{ inch pounds} \\
\text{Bending} &= 3000 \times 9 \\
&= 27{,}000 \text{ inch pounds} \\
\text{Combined Moments} &= \sqrt{(.4M_t)^2 \quad (.4M_b)^2} \\
&= 17{,}600 \\
&= \frac{\pi}{32} \times 15{,}000 \times D^3 \\
&= 1470 \times D^3 \\
D^3 &= 11.9 \\
\text{and } D &= 2.48 \text{ inches}
\end{aligned}$$

Chapter 14

HULL STRUCTURE

The history of planing hull structure has demonstrated the urgency of sound design and expert workmanship together with adequate materials for the purpose. What may have proved adequate in displacement hulls frequently fosters an entirely misleading concept of adequacy in the planing hull, for here stresses of a magnitude greater than those of any other type of craft are the rule. And while progress in structural design must always come mainly from experience with previous similar hulls, the field for originality with new materials and methods is greater than ever.

In studying the structural failures of various hulls, it should be borne in mind that certain unfortunate shapes of bottom doom any type of structure, no matter how strong, to premature failure. On a pounding bottom there are no structural methods or materials known to man which can long stand the punishment.

But assuming that the designer has created a correct shape of bottom, the structure must still achieve great strength and do it with extremely light weight. Strength without lightness is futile and, as in all sound engineering, the simplest methods are likely to produce the most nearly perfect results.

The hull is in effect a box type girder which speed and wave action will combine to destroy. Its points of support may be anywhere, and the stress directions are from everywhere. Its virtual weight is multiplied many times by acceleration and impact. To meet these conditions there is no finer example than the egg shell which translates in engineering structures to the so-called "stressed skin."

The hull of a boat is nearly ideal in shape for stressed skin construction. The shell of planking and decking is already the extreme fiber, and intelligent use of its potentiality will lighten and strengthen the structure. As an example of this well-understood principle, a heavy beam running along or close to the hull's neutral axis adds far less

ADDENDA

THE WEIGHT-SPEED-POWER RELATIONSHIP

By Prof. Harold A. Thomas

This paper is concerned with an analysis of weight-speed-power relation for fully-planing boats. While the general principles explained are applicable to fully-planing hulls of all types, the specific numerical applications given here are limited to V-bottomed hulls having constant deadrise angle and straight run in the wetted portion of the bottom, because this is the only type of planing hull for which complete up-to-date test data is available in published form. For reasons which are ably presented in Dr. Lindsay Lord's book on *Naval Architecture of Planing Hulls,* it is believed that the best planing hulls of the future will be of this type.

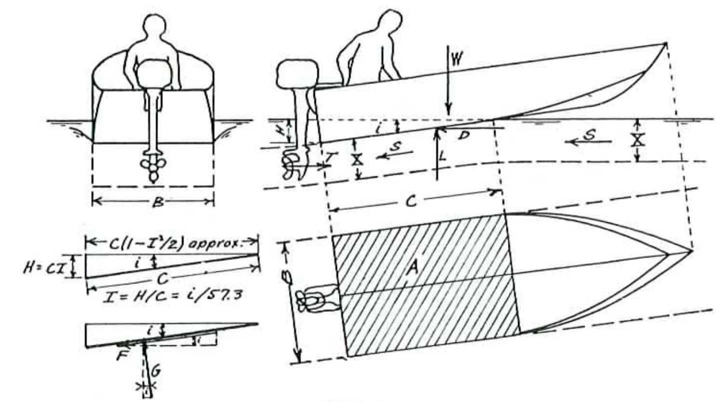


FIG. 1

In the flat-bottomed boat the deadrise angle is zero and the wetted area is therefore a plane rectangle

For hulls in which the deadrise angle is not constant but increases from the stern forward, as in many designs of the past, the numerical data of this paper may be reasonably accurate if the boat is operated with lifted bow, so that water does not touch the forward part of the hull where the deadrise is steep. In this case the effective deadrise angle is taken as the mean value over the wetted portion of the bottom of the hull, and the effective hull inclination or angle of attack is taken as the inclination of the equivalent plane bottom, as indicated in Fig. 2. However, in this case there is no easy way to determine the exact error of the computation. If such a boat is operated with lowered bow, so that the sharp forward part of the hull plows through the water and raises a high bow wave, then the boat is no longer fully-planing but is semi-displacement. The term fully-planing as used in this paper in connection with the types of hulls under consideration does not imply that the buoyant part of the lift is zero but merely means that the transom is dry and the deadrise angle at entrance is not steep enough to raise a high bow wave.

For hulls in which the floor lines of the transverse sections are straight, the equivalent-plane-bottom line in the side view lies half way between the chine line and the keel-rabbet line, as indicated in Fig. 2. If these floor lines are not straight, then the equivalent-plane-bottom line in each transverse section is adjusted to equalize the shaded areas above and below, as indicated in the figure.

In this paper the symbol i is used to designate the hull inclination in degrees (see Fig. 1), while the symbol I is used to express the same angle in radians. The relation between these is $i = 57.3\ I$. The use of the latter symbol adds materially to the conciseness of the equations. Frequent use is made of the approximations: $\sin i = I$ and $\cos i = 1 - I^2/2$. (see Fig. 1.). For small values of i these approximations are sufficiently accurate for most purposes.

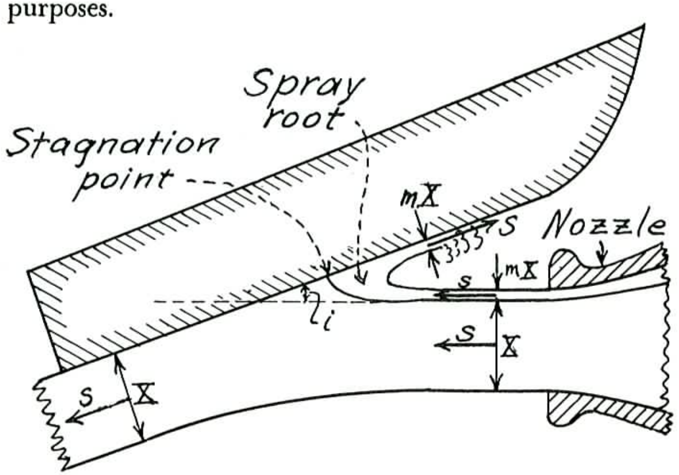


FIG. 1a

Sometimes when a motorboat is attempting to move upstream in a swift river, the power of the engine is just sufficient to hold the boat stationary against the current, so that no progress is made. The principles which govern the weight-speed-power relation for a fully-planing boat can be explained most conveniently by reference to a system of this kind where the boat is stationary and the underlying water is moving. Such a system is shown in Fig. 1. In the flat-bottomed boat in this figure the deadrise angle is zero and the wetted area on the bottom is therefore a plane rectangle of breadth B and chord-length C, its area A being the product BC. If the deadrise angle is not zero, then the wetted area is not rectangular but is somewhat pointed at the forward end, and in this case the distance C is the mean length of the wetted area. Following aviation practice, the term aspect ratio, designated by the symbol a, is used to denote the ration of the breadth B to the length C of the wetted surface.

We will first make an analysis of the performance of a flat-bottomed boat such as that shown in Fig. 1. This is worth while because the performance of a V-bottomed hull does not differ greatly from that of the equivalent plane-bottomed hull, particularly if the deadrise angle is not large. After completing this we will take up the modifications necessary to make the analysis applicable with precision to hulls having deadrise.

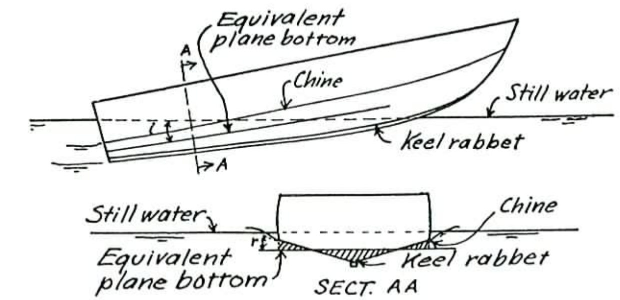


FIG. 2

If the deadrise angle is not constant then the effective hull inclination is evaluated as the equivalent plane bottom

The water approaching at speed S is deflected downward as it flows under the stationary flat-bottomed boat of Fig. 1. The individual water particles actually move downward and outward in diverging paths as they leave the stern, but it is convenient to think of an imaginary case where the flow under the boat is in a stream of rectangular cross section of width B and depth X. This stream is deflected downward at inclination I as it passes under the hull, while the speed S remains unchanged. The stream depth X required to make the theoretical uplift computed under this assumption exactly equal to the true dynamic uplift can be determined by experiments on similar hulls in a towing tank. The ratio of X to C is denoted by N.

Towing tank experiments show that the value of N for flat-bottomed hulls is independent of size and speed and depends only on the aspect ratio and the inclination angle i. For a technical discussion and bibliography of such tests the reader is referred to an article entitled *The Hydrodynamics of Planing Hulls* by Allan M. Murray in the Transaction of the Society of Naval Architects and Marine Engineers, Vol. 58, 1950. Among other things, this article gives the results of an extensive series of tests on planing V-bottomed hulls with uniform deadrise angles, made at Stevens Institute of Technology for the U. S. Navy (Preprint No. 244, entitled

"Wetted Area and Center of Pressure of Planing Surfaces" by B. V. Karin-Kroukovsky, D. Savitsky and W. F. Lehman). From the above-mentioned test data the following formula may be derived: $N = .344\ a^5\ i^{.1} + M$, where M is the friction factor explained in a subsequent paragraph. This is applicable to flat-bottomed hulls operated at inclination angles up to about 13 degrees.

In the following paragraphs it will be shown that the vertical uplift exerted on a planing boat by the underlying water consists of three parts: (1) the dynamic lift, (2) the skin-friction lift and (3) the static or buoyant lift. Similarly the horizontal drag exerted on a planing boat by the underlying water is made up of four parts: (1) the dynamic drag, (2) the friction drag, (3) the static drag and (4) the parasite drag. Since each of these seven forces can be found by the application of familiar elementary principles, the method of doing this will be explained in detail.

The following explanations are simplified and the equations are shortened by use of the word slug, which is the engineer's term for the unit of mass. One slug of material weighs 32.2 pounds, this figure being denoted by g. The density of a liquid, denoted by d, is its mass per unit volume. The density of fresh water is 1.94 slugs per cubic foot and the density of sea water is 1.99 slugs per cubic foot.

Referring to Fig. 1 we note that the number of slugs of water passing under the boat per second is BXSd, which is equivalent to NASd. This water leaves the stern at a speed S which is inclined at a slope I below the horizontal. Therefore the vertically-downward component of this velocity is SI. Since momentum is the product of mass and velocity, the vertically-downward momentum created per second by the downward deflection of the water flowing under the boat is NASd times SI, or NAS^2dI.

The weight W of the boat and its contents is the vertical force which produces this downward increase of momentum in the underflowing water. According to Newton's second law of motion a change of momentum per second is numerically equal to the force which produces it. Therefore (neglecting corrections for skin friction and buoyancy) $W = NAS^2dI$. The uplift exerted on the bottom of the boat by the underflowing water is, of course, exactly equal and opposite to the downward weight W. The value L of the uplift computed from the equation $L = NAS^2dI$ is called the dynamic lift and is somewhat different from the true lift which includes corrections for skin friction and static buoyancy.

The dynamic lift L may be thought of as the vertically-upward force which would be exerted on the bottom of the boat by the underflowing water if the contact were frictionless and there were no variation of pressure with depth, just as if the stream were a jet issuing into the open air from a rectangular nozzle, as indicated in Fig. 1a.

Similarly, the dynamic drag D is the horizontal force which would be exerted on the bottom of the boat by the underflowing water if the contact were frictionless and there were no variation of pressure with depth. Since

the resultant dynamic pressure must be normal to the frictionless bottom of the boat, the ratio of its horizontal component D to its vertical component L must be tan i, which is approximately equal to I. That is: $D = LI = NAS^2dI^2$.

The following alternative derivation of the foregoing formulas for dynamic lift and drag throws light on an important factor which affects the performance of a planing boat. Floating on the surface of the water approaching the stationary hull there is a thin layer of water (not shown in Fig. 1) of thickness mX which does not flow under the boat but turns upward and flows forward along the bottom of the hull, soon breaking into spray as indicated in Fig. 1a. The thickened portion of this layer where it makes the turn is called the "spray root". At ordinary inclination angles this layer is very thin and produces but an insignificant effect on the dynamic lift. However, it produces an extremely important effect on the dynamic drag, increasing it by 100 percent.

In the case of a flat-bottomed boat let L_1 and D_1 be the dynamic lift and drag due to the main stream of depth X and let L_2 and D_2 be the dynamic lift and drag due to the surface stream of depth mX. Using the principle that the force exerted in any direction by a flowing stream is equal to the mass of water flowing per second times the change of velocity component in that direction, we obtain the following relations: $L_1 = BXS^2dI$ and $D_1 = BXS^2dI^2/2$; $L_2 = mBXS^2dI$ and $D_2 = mBXS^2d\ (2 - I^2/2)$. Since the flow can exert no force on the frictionless hull in the direction parallel to its bottom, the following expression for m is obtained by equating to zero the sum of the changes of momentum per second in that direction: $m = 1/(4/I^2 - i)$. Combining these results we find that $L_2 = mL$, and $D_2 = D_1$. Therefore the total dynamic lift is $L = L_1 + L_2 = (1 - m)\ BXS^2dI = NAS^2dI$ approximately, and the total dynamic drag is $D = D_1 + D_2 = 2BXS^2dI^2/2 = NAS^2dI^2$. These expressions for L and D are the same as those previously obtained.

For an inclination angle $i = 5$ degrees, or $I = .0873$ radians, the value of m is .0019. Thus the foregoing approximation of considering $1 - m$ equal to 1 in computing the dynamic lift is sufficiently accurate for most practical purposes.

Innumerable experiments have been made on hydraulic skin friction and these show that frictional resisting force F varies as the water density d, the wetted area A, the square of the speed S, and an experimental coefficient M whose value increases greatly with surface roughness and decreases somewhat with the speed S and with the length C of the wetted surface. Thus the expression for the skin-frictional dragging force along the bottom of the hull takes the form $F = MdAS^2$. This is not the place for a discussion of the complexities of the modern skin-friction formulas. For the purposes of this paper one of the older and simpler formulas is satisfactory. The following formula for M for smooth varnished surfaces can be deduced from the famous experiments of Wm. Froude, the father of modern naval architecture: $M = .00324/\ (C^{.15}\ S^{.15})$. For surfaces as rough as medium sandpaper the values of M become about twice as large.

Referring to Fig. 1 it is noted that the vertical component L' of the skin friction force F is the product of F and I. That is: $L' = -MAS^2dI$. This is called the friction lift, the minus sign indicating that it acts downward. Similarly it is noted that the horizontal component D' of the friction force is the product of F and $(1 - I^2/2)$. That is $D' = MAS^2d\,(1 - I^2/2)$. This is called the friction drag.

If the pressure distribution in the moving water under the flat-bottomed hull were exactly the same as in the surrounding undisturbed water, then the static or buoyant force G acting upward on the bottom of the hull would be the product of the wetted area A, the depth $CI/2$ of the center of this area below the level of the undisturbed water, and the weight gd of a cubic foot of water. Experiments described in the previously mentioned reference show that the actual buoyant lift is somewhat less than this by a ratio R the value of which varies slightly with the inclination angle i. From data in that reference the following formula is deduced: $R = .545\ i^{.1}$. Thus the expression for the buoyant force G acting on the inclined bottom of the hull is given by $G = RgdACI/2$. Referring to Fig. 1 it is noted that the vertical component L'' of G is the product of G and $(1 - I^2/2)$, but at the flat inclination angles encountered in the high-speed operation of fully-planing hulls the term $(1 - I^2/2)$ does not differ appreciably from 1. That is: $L'' = RgdACI/2$. This is called the static lift. Similarly it is noted that the horizontal component D'' of G is the product of G and I. That is $D'' = RgdACI^2/2$. This is called the static drag.

If the hull has submerged appurtenances such as propeller struts, rudder, fins, skeg, or the gear case of an outboard motor, these produce additional drag which is called parasite drag and may be represented by a term ZdS^2. Values of Z suitable for various appurtenances can be found in technical publications or can be determined by special towing tests. This term can also be used to cover the drag caused by windage.

Combining these results we find that the weight W of the boat and its contents is equal to the sum of the dynamic lift L, the friction lift L' and the static lift L''. That is: $W = AS^2d\,(N - M)I + RAgdCI/2$. Similarly the propeller thrust T is the sum of the dynamic drag D, the friction drag D', the static drag D'' and the parasite drag ZdS^2. That is: $T = AS^2d\,([N - M]I^2/2 + M) + RAgdCI^2/2 + ZdS^2$.

In deriving the equations of this paper the propeller thrust T is considered horizontal, as in Fig. 1. If desired, the correction for inclination of the propeller shaft can be made as follows. If the inclination of the propeller shaft below the equivalent plane bottom is j degrees or J radians, where $J = j/57.3$, then the vertical component of the propeller thrust is approximately $(I + J)\,T$ and the horizontal component is approximately

$(1 - [I + J]^2/2)T$. Therefore if it is desired to modify the foregoing equations to allow for inclination of the propeller shaft, add $(I + J)\,T$ to the right-hand member of the equation for W and change the left-hand member of the equation for T to $(1 - [I + J]^2/2)\,T$. However, these terms are not carried in the equations of this paper, because they add to the length of those equations.

Since power is the product of force and speed, the theoretical power required to drive the boat is the product of the thrust T and the speed S. This is in turn equal to the product of the engine power P and the combined efficiency E of the propeller, shaft bearings and gears (if any). That is $TS = PE$ or $T = EP/S$. Inserting this value of T in the foregoing equation for thrust, we obtain the following expression for the power P required to drive a fully-planing flat-bottomed boat in still water:

$$P = (AS^3d/E)\ \ ([N - M + RgC/S^2]\ I^2/2 + M + Z/A)$$

Dividing each member of this equation by the corresponding member of the foregoing equation for W, and collecting terms, we obtain the following alternative expression for power:

$$P = (WS/E)\ \ ([N - M + RgC/S^2]\ I/2 + [M + Z/A]/I)/\,(N - M + RgC/[2S^2]).$$

This is the desired weight-speed-power-relation for a fully-planing flat-bottomed boat. For certain purposes it may be desirable to express this relation in other alternative forms. For example, a form not containing the symbols A or C can be obtained by letting $C = A/B$ in the foregoing equations for W and T, and then eliminating A between these equations.

In using the equations given in this paper, lengths should be expressed in feet, weights and forces in pounds, speed in feet per second, density in slugs per cubic foot, and power in foot pounds per second. If it is desired to express speed in miles per hour, denoted by S', or in knots, denoted by S'' and power in horsepower, denoted by P', replace S by $1.467S'$ or by $1.689S''$, and replace P by $550\ P'$.

In solving various problems by use of the foregoing equations it may be necessary to know the relation between the bottom-inclination angle i and the location of the center of gravity of the boat and its contents. In Fig. 3 let nC be the distance from the transom to the point of application of the total uplift W on the bottom of the hull. The previously mentioned references gives the following formula for computing n for a flat or V-bottomed hull: $n = (.84 + .015r)\ a^{.05\ +\ .01r/i^{.125}\ +\ .0042r}$. This formula is applicable up to inclination angles of about 13 degrees and deadrise angles up to about 30 degrees. It is noted that this formula does not contain the speed term S. Therefore the formula is accurate only at speeds sufficiently high to make the dynamic lift large in comparison with the buoyancy lift. Since $Ty = Wx$, the location of the center of gravity of the hull and its contents is found by: $x = Ty/W$.

In using the equations of this paper it will be noted that, since the value of N depends on the aspect ratio B/C and the hull inclination I, the value of M depends on the wetted length C and the speed S, and the value of R depends on the hull inclination I, these equations have to be solved by successive approximation in any problem where either C, S or I is unknown at the start. Using assumed values of M, N and R compute the other quantities by the proper equations, and also compute M, N and R. If the computed values do not check the assumed ones, revise the assumption and repeat until a satisfactory check is obtained.

Alter Trim by Shifting Ballast

The crew of a boat can alter its trim by shifting ballast and their own weight, and there is a particular bottom inclination which minimizes the power required to produce a given speed. In the last equation for power a member of the Sophomore math class can easily show that the power required to drive a fully-planing flat-bottomed boat at a given speed in still water is a minimum when $I = \text{sq. rt. } (2M/[N - M + RgC/S^2])$.

The term "clean" is used to describe a hull without appurtenances which produce parasite resistance. It is interesting to examine the special case of a clean flat-bottomed hull operated at speed so high that the buoyancy terms (those containing g/S^2) become negligible. For this case the equation for power reduces to $P = (WS/E)\ \ (I/2 + 1)[(N/M - 1])$. and the most efficient planing angle becomes $I' = \text{sq. rt. } (2/[N/M - 1])$. Inserting this value of I' in the equation for P, we find the following expression for the power required to drive a clean flat-bottomed hull under high-speed conditions at its most efficient planing angle: $P = WSI'/E$. Thus if the propulsive efficiency E is considered constant, and if we consider I' as approximately constant, it is seen that the required power P varies directly as the boatweight W and directly as the speed S. In other words: If the power is constant a given reduction in weight produces the same proportional increase in speed. If the speed is constant the required power varies directly as the weight. If the weight is constant then the fuel consumption in gallons per hour varies directly as the speed, and in gallons per mile is independent of the speed. These findings are in great contrast with the corresponding ones for a displacement boat, where there is a limiting condition beyond which a great increase of power produces only a slight increase of speed.

The foregoing equations are strictly applicable only to flat-bottomed hulls. The following procedure is used to make the equations apply with precision to V-bottomed boats with constant deadrise angle r. The equation previously given for the weight of a flat-bottomed boat is: $W = (N - M)\ AS^2dI + RACgd/2$. Noting that $A = B^2/a$ and $C = B/a$, this becomes $W = (N - M)\ B^2S^2dI/a + RB^3gdI/(2a^2)$. This can be written: $W = cdS^2B^2/2$ where $c = (I/a)\ \ (2[N - M] + RBg/[aS^2])$. The factor c is called the "coefficient of lift".

The equation for the weight W′ of the corresponding V-bottomed boat with uniform deadrise angle r is: $W' = c'dS^2B^2/2$ where $c' = c - .0065\ ri^{.6}$. This relation is based on experiments and analysis described in the aforementioned reference. It is applicable for deadrise angles up to about 30 degrees. The drag D of the V-bottomed boat is given by: $D = W'I + MA'S^2d\ \ (i + I^2/2) + ZdS^2$. This is equal to the thrust T of a propeller with horizontal shaft. The power required to drive a V-bottomed boat with such a shaft is given by: $P = TS/E$.

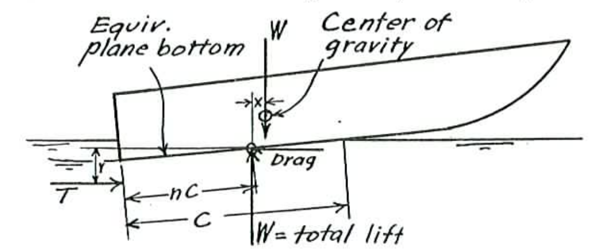


FIG. 3

This drawing shows the relation between the bottom-inclination angle and the location of the boat's center of gravity

The area A′ in the foregoing formula is the actual wetted area, which is slightly greater than BC for three reasons: (1) In the V-bottomed hulls of uniform deadrise angle, to which the numerical data of this paper apply, there is no large bow wave, but there is a rise of water, known as the "spray root" along the entering edge. This makes the wetted area somewhat greater than would be the case if the water surface at this location were perfectly flat. If C is the mean wetted length which would exist if the water surface were flat, then the corrected mean wetted length C′ is given by: $C' = C + .0908\ Br/i$. This equation is based on data given in the previously mentioned reference. (2) There is some wetted area above the chines in the after part of the hull, the upper edge of this area being usually covered with spray and somewhat vaguely defined. In the absence of published test data on this topic it is suggested that this additional area A″ be estimated by $A'' = C^2I$. This area may be included in computing the friction drag but not in computing lift. (3) In a V-bottomed hull the actual wetted area on the bottom is slightly greater than BC, due to the transverse slope of the bottom. The true area of the sloping bottom is $(1 + r/6750)$ BC.

In practical applications of the equations of this paper, the most difficult term to evaluate is the efficiency E of the propeller, shaft bearings and gears (if any). In the reference previously mentioned, Mr. Murray states that in designing high-speed boats, naval architects often do not count on getting values of E better than .50, and values lower than this may be encountered if there is serious trouble with propeller cavitation. However in carefully-engineered designs for boats of considerable size, propeller efficiencies of from 70 to 80 percent are known to be obtainable in single-screw installations and from 60 to 70 percent in twin-screw ones.

In order to give the reader some idea of the relative magnitudes of the nine forces acting on a fully-planing hull, a numerical example is given on facing page.

While the equations which have been given are correctly deduced from the given assumptions, there are, of course, several problems which will waylay any person who tries to apply these results in building a very fast boat. The first one is the fact that a boat becomes unstable and porpoises when the attempt is made to carry its entire weight on a narrow transverse strip of water under the stern. A second is the fact that in single-step, inboard motorboats it is practically impossible to get the center of gravity far enough aft to make the boat ride at its most efficient inclination. These first two problems can be licked by expedients such as the three point suspension. A third one is the fact that waves and rough water play hob with the concept of a steady rectangular or somewhat-pointed wetted area on the after part of the bottom of the hull, so that rough-water performance is not the same as still-water performance. A fourth is the difficulty of building a small, light hull which can carry a large, powerful motor and lots of fuel, and is strong enough to stand the terrific buffeting of rough water. A fifth is the difficulty of preventing cavitation and rooster-tailing from cutting the efficiency of the propeller down to a very low value. Others have to do with pounding, nose diving, spray throwing, stability in turning, and evaluation of parasite resistance, windage, etc. Most of these problems can be licked only by practical tricks learned while building and operating fast boats.

EXAMPLE.

GIVEN: Hull breadth: $B = 4$ ft. Mean wetted length: $C = 5$ ft.
Mean deadrise angle of wetted area: $r = 5$ degrees.
Hull inclination angle: $i = 3$ degrees. Speed: $S = 50$ ft per sec. (34.1 mi per hr.). Operation in fresh water, density: $d =$ 1.94 slugs per cu. ft. Parasite drag and windage factor: $Z = .008$.
Efficiency of propulsion: $E = .50$. Varnished surface.
Horizontal propeller shaft.

COMPUTED: Wetted area of equiv. flat bottom: $A = BC = 20$ sq. ft.
Aspect ratio: $a = B/C = .800$. Hull inclination: $I = i/57.3 = .0524$ radians. Skin friction factor: $M = .00324/(CS)^{.15} = .001415$.
Stream depth factor: $N = .344\ a^{.5} i^{.1} + M = .345$.
Buoyancy factor: $R = .544\ i^{.1} = .553$.

FOR EQUIVALENT FLAT-BOTTOMED HULL:

Dynamic lift: $L = NAS^2 dI =$ 1752 lbs.
Friction lift: $L' = -MAS^2 dI =$ -7 lbs.
Static lift: $L'' = RgdACI/2 =$ 90 lbs.
Weight: $W = L + L' + L'' =$ 1835 lbs.
Dynamic drag: $D = NAS^2 dI^2 =$ 92 lbs.
Friction drag: $D' = M\ AS^2 d(1 - I^2/2)$
$+ MS^2 dC^2 I = 137 + 9 =$ 146 lbs.
Static drag: $D'' = RgdACI^2/2 =$ 5 lbs.
Parasite drag: $D''' = ZdS =$ 30 lbs.
Total drag: $D + D' + D'' + D''' =$ 282 lbs. = propeller thrust = T.
Coefficient of lift: $c = (I/a)(2(N-M) + RBg/aS^2) = .0473$.

FOR VEE-BOTTOMED BOAT:

Coefficient of lift: $c' = c - .0065\ rc^{.6} = .0421$.
Weight: $W' = c'dS^2B^2/2 = 1633$ lbs.
Friction area: $A' = B(C + .0908\ Br/i)(1 + r^2/6570) + CI^2 = 22.54$ sq.ft.
Total drag = propeller thrust = $T' = W'I + MA'S^2 d(1 + I^2/2) + ZdS^2 =$ 279.3 lbs.
Engine power: $P = T'S/E = 27930$ ft.lbs. per sec. or 50.8 horsepower.
Location of center of gravity (assuming $y = 1.2$ ft. in Fig.3):
$n = (.84 + .015r)a^{.05+.01r} / i^{.125+.0042r} = .776$. $x = T'y/W' = .21$ ft.
Distance from transom to center of gravity: $nC + x = 4.09$ ft.
Most efficient planing angle for equivalent flat-bottomed hull:
$i = 57.3$ sq. rt.$(2M/(N - M + RgC/S^2)) = 4.95$ deg. Since the given inclination angle of 3 degrees is less than this, the boat's speed could be improved by moving the center of gravity farther aft.

The example above gives the reader an idea of the relative magnitudes of the forces acting on a fully planing hull

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