R-726-1

DAVIDSON LABORATORY

AUGUST 1969

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HOBOKEN, NEW JERSEY

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Report 726-1

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Prepared for the
Office of Naval Research
Department of the Navy
Contract Nonr 263(69)
(D.L. Project 3080/079)

Background Research Initiated Under
U.S. Army Tank-Automotive Command
Contract DA30-069-ord-1763

Approved

I. Robert Ehrlich, Manager
Transportation Research Group

pages ix + 84
Figures 1-60

ABSTRACT

The results of a concept design study of high-speed wheeled amphibious vehicles originally conducted in 1956-1959 are now presented as Volume I of a two-volume study.

The basic technical and operational problems of high water speed are examined.

Six basic concepts are presented to exemplify potential mechanical solutions. Five concepts are planing hull types with retractable wheels; one example is based on the vehicle train concept.

The conclusions and recommendations which are made consider the decade elapsed since the original study.

KEYWORDS

Amphibians, Wheeled
Ship-to-Shore Operation
Planing Hulls
Coupled Vehicles

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1. INTRODUCTION

This report which is Part I of a two-part study (see Ref. 1) summarizes a design study on high speed logistical amphibians conducted for the U.S. Army Ordnance Corps in 1956-1959 under Contract DA30-069-ord-1763. Responsibility for this class of vehicle was transferred in 1959 to the Transportation Corps, and the subject Ordnance supported contract was reoriented at that time to the direct technical support of ongoing Ordnance Corps hardware projects. As a result, the design study work was never completed. In 1965 funds were provided by The Office of Naval Research under contract NR-062-374/5-3-65-263(69) to complete this study and publish a final report.

The object of the 1956-59 design study was to develop test data on, and engineering and relatived operational studies of high-speed wheeled logistic amphibious trucks, and to suggest promising new design concepts from the results.

Publishing the results of this design study at this late date raises several problems due to changes in technology and operational doctrine since the major portions of the study were completed. First, the study was conducted under the assumption that the amphibians would operate in a nuclear setting, in which dispersion of ship to inland operational dump elements was a controlling design consideration. Second, although the rate of technologic advance in the amphibious truck field and directly related areas has been imperceptible when viewed over a short period, sufficient time has now elapsed so that much of the concept engineering could stand updating, and some of the ideas which appeared new and useful in 1959 have since in fact been tried, with varying results.

In preparing this report, many of the numbers have been updated, opportunity has been taken to make limited comparisons with vehicles and test beds actually built since the study started, and most of the discursive, pre-computer age, pre-Vietnam operational analyses have been eliminated. Those few ideas which have since proved good in practice, or still look attractive-though untried-after all this time, are stressed. The basic engineering, however, has not been redone.

The 1956-59 study was limited to wheeled amphibious trucks, but all technical means to achieve high water speed were embraced except the use of hydrofoils, which were then under separate study elsewhere. While this theoretically opened the door to the study of air cushion vehicles, submersible amphibians, etc., the means actually studied were the use of planing hulls, and the use of more prosaic barge hulls in tandem train configuration. One significant advance made in the planing hull concept, suggested by Dair N. Long, was use of properly designed cavities for stowage of retracted wheels, thus eliminating the wheel-well closure doors. This suggestion was made after the design study work was completed, and no specific concept layouts were made incorporating it, although the idea was model-tested[1] and subsequently worked into the LVW test bed.

2. THE OVERALL PROBLEM

This study was conducted within the framework of operational ship-to-shore, over-the-shoreline resupply operations required in a large war involving unrestricted use of nuclear weapons in the battle area. The presumption made that the only satisfactory defense against such weapons was adequate dispersal of both ships and inland supply transfer points. Operations which involved one-way water and land distances of 25 miles or more were considered potentially necessary. The staggering supply requirements projected appeared at the time (1956-1959) to rule out helicopter-lift for the bulk of the tonnages.* The notion of a crowded beach strandline such as characterized many WWII and Korean operations, or more recently those in Cam Ranh Bay, was plainly intolerable.

The combined elements of this view of the problem forecast the need for amphibious trucks of considerably higher water speed than the 6 mph achieved by the WWII DUKW. They suggested further that neither the modest on-road nor the marginal off-road performance levels of the DUKW could be traded-off to achieve the desired water speed increase; indeed, the off-road

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*A 1960 Army study of high speed amphibian truck requirements projected that 97% of resupply cargoes would arrive overseas by ship, and 80% of this tonnage would move ashore over the beaches rather than through ports[2].

performance level of the vehicle could also stand considerable improvement. Finally, it recognized that the excellent surfability of earlier amphibians was a part of their essential performance, and hence also could not be reduced simply to get higher speed. Accordingly, a fundamental decision was early made that, although the study was specifically aimed at increasing water speeds, this would not be done at the expense of then current levels of land performance or of basic surfability.

A second starting premise was that the amphibian truck system provided a service, and that its effectiveness was measured by its influence on the entire unloading operation from conventional ships to inland transfer points, rather than by any special merits of individual vehicles. This obvious viewpoint has two important corollaries for design. First, regardless of how desirable the results of some feature may be in terms of individual amphibian performance, it cannot be tolerated if it limits, in any way, the maximum flow rate of cargo from the ship to the amphibian, as determined by the characteristics of the ship and its unloading system. Second, the amphibian system cannot be a drag on the overall operation. The amphibians should queue, not the ships. Economies which require fine-tuning of the ship-to-shore operation and cannot tolerate a clear numerical surplus of amphibians are illusory.

3. BACKGROUND

The history of modern wheeled amphibians effectively begins with the design and somewhat premature production in 1942 of the 1/4-ton 4x4 amphibian, developed through the conversion of the production WWII jeep. The amphibious jeep was not a success, primarily because there was no real military requirement for such a small machine. The valid experience from this development, both technical and logistic, was immediately capitalized upon, however, in the subsequent rapid design, production, development and deployment of the successful 2-1/2 ton 6x6 DUKW amphibian[3].

The need for the larger machine, to unload ships across unprepared beaches without loss of momentum at the surf line, was first broached in mid-April 1942. Under the extra-Ordnance Corps management of the National Defense Research Committee, the first DUKW was swimming by early June 1942. First productions models were delivered in November 1942, and DUKWs were first used in quantity in the July 1943 landing in Sicily. By December 1943, production had reached 1500 per month[3,4].

The extraordinary WWII success of the DUKW is accounted for not only by its general technical quality, but also by

1) its timeliness -- it was both available and needed;
2) the considerable (though still far from technically optimal) effort which went into operational training, an effort which was till growing at the war's end; and
3) the extensive and continuing development to which it was subject. From the moment the first DUKW floated until the end of the war, alterations shown by field experience to be necessary were rapidly made, both in the field and on current production models [3,4].

By August 1945, 21,000 DUKWs were produced and 6000 more were on order. Even so, there were never enough available to meet much more than the basic over-the-beach landing requirements, and few of the secondary uses proposed (pontoon bridges, mobile ferries, etc) were ever widely tried in the field[3].

Despite its overall success, the DUKW was early criticized as being too small for reasonable cargoes, difficult to unload, too slow in the water, too prone to bogging in muddy conditions, and helpless in exiting from the water except over reasonably good sand beaches. In 1952, Stephens, speaking for a special NRC committee convened to review and suggest the proper exploitation of wartime over-the-beach landing experience, was unenthusiastic about the possibilities for economic technological solutions to the water speed and soft soil mobility problems. He pointed out the fundamental difficulties involved in increasing water speed and suggested that, in place of attempting to develop exotic new high-speed machines, emphasis should rather be placed upon evolutionary solution to the many solvable problems (such as size, mechanical reliability, maintainability, etc.)

of essentially DUKW-like amphibians. Based on his then recent, wide, personal field experience with the DUKWs, Stephens felt that serious efforts to improve operational doctrine and methods, and more thorough training of operating personnel at all levels in the exploitation of their equipment, offered far more potential for overall improvement in amphibian over-the-shoreline operations than did any feasible, radical technical improvements in the vehicles themselves[5].

In the years immediately following, wheeled amphibian truck development _per se_ was pursued by the Ordnance Corps, consciously or otherwise, within this framework, although the important operational and training aspects which Stephens made concomitant were neglected.* A parallel line of development, that of medium sized amphibious lighters,** was begun in 1959 by the Transportation Corps[7].

Experimental amphibious military trucks of the period 1950-1956 were the XM148 GULL (5-ton 6x6, fiberglass-reinforced plastic hull)[8], the XM147 SUPERDUCK (4-ton 6x6, steel hull)[9], and the XM157 DRAKE (8-ton 8x8, aluminum hull)[10]. Their leading characteristics and, for comparison, those of the DUKW are summarized briefly in Table I. Characteristics of the present-day LARC V (5-ton 4x4, welded aluminum hull)[11], and LARC XV (15-ton 4x4, also welded aluminum)[12], whose development began at the end of this period, are also given. Figures on the GULL are not included because it went so far overweight (40,600 lb empty) that its performance could not be seriously checked, and it was so far off its design point that, even if reliable performance figures were available, indices of performance based on them would be technically meaningless.

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*So much so that by 1966, when the first landings were made at Cam Ranh Bay in South Vietnam, the necessity to operate trucks over the sand beaches at reduced tire pressures had passed out of general military knowledge, resulting in a minor "mobility" flap.

**The distinction between an amphibious truck and an amphibian lighter is presumably that the latter is more of a boat- and less of a truck- than the former. Although a 1957 review of military amphibians estimated that use afloat accounted for only 15% of their total operating time[6], a 1960 Transportation Corps study projecting that 80% of combat resupply would be "over-the-beach" implied that far more water operation would be required in the future[2].

TABLE 1

LEADING CHARACTERISTICS
OF WHEELED CARGO AMPHIBIANS, 1942-1960

	DUKW	SUPERDUCK XM147E3	DRAKE XM157	LARC V*	LARC XV*
Year	1942	1954	1956	1960	1964
Rated payload, short tons**	2.5	4	8***	5	15
Length overall, ft	31.0	33.7	42.0	35.0	45.0
Width overall, ft	8.2	9.0	10.0	10.0	14.5
Height overall, ft	8.8	9.3	10.8	9.2	13.7
Curb weight, lb	15,000	19,900	32.700	21.000	45,200
Gross veh. wt (W), lb	20,000	27,900	48,700	31.000	75,200
Wheel configuration	6x6	6x6	8x8	4x4	4x4
Tire size	11.00x18	12.50x20	14.75x20	18.00x25	24.00x29
Ground clr., in	11	13	16	23	29
Max gross horsepower	91	155	310	300	600
Propeller diam., in	25	31	2-31	29.5****	35****
Hull material	Steel	Steel	Alum.	Alum.	Alum.
Max still water speed (V_m) @ GVW, mph	6.1	6.7	7.8	8.6	8.3
Max level highway speed @ GVW, mph	50	48	45	25	29

*Production model

**All machines have some overload capability (25-50%) under favorable conditions

***Raised to 10 tons in 1959 in a last ditch effort to sell the vehicle, but reliable performance figures at 10 tons are lacking

****With shrouded propeller

The relative soft ground mobility of the same vehicles is indicated in Table II by the following indices:

1) The Eklund Mobility Factor -- MF_E (general mobility)[13]
2) The WES Vehicle Rating Cone Index required for 50-pass trafficability -- VCI (applicable to fine-grained soils)[14]
3) The ATAC Nominal Unit Ground Pressure -- NUGP (applicable to fine-grained soils)[15]
4) The Freitag-Knight Sand Index -- G_{FK} (applicable to coarse-grained soils)[15,16]

Higher values of the Eklund Mobility Factor posit better mobility in weak soils. The other three indices represent the relative minimum soil strength on which a vehicle may maintain straight, level, unaccelerated motion, and hence lower values for these indices predict higher soft soil mobility. The figures of Table II demonstrate that only the LARC V represents any noticeable mobility improvement over the DUKW, and even it is still no marsh buggy. A successful effort in 1964 to design a practical load-carrying vehicle on tires to work in soft soil areas such as are regularly found along tidal rivers, for example, aimed for traffic or 50-pass operation at a VCI of only 25.[17] The LARC XV is clearly limited to off-road operations on sandy beaches which is probably appropriate for a vehicle of this size.

A gross comparison and reconciliation of the maximum still-water speed at gross vehicle weight for each of the five machines of Table I is shown in Table III. The "effective resistance" (R_e) is readily calculated from published figures for installed gross horsepower, maximum still-water speed, and gross weight. The value of R_e is the computed resistance to motion if the conversion of gross installed horsepower to towrope horsepower were 100 percent efficient. Thus it mashes together power diverted to accessorites, drive line losses, propeller losses, and true propulsion resistance into one unfactorable lump. Nonetheless, it is revealing, as can be seen in Table III, for its range over the several vehicles is small, especially after the modest spread of speed-length ratios involved is roughly "corrected for" by constructing the coefficient $(R_e/W)/(V/\sqrt{L})^2$.*

*This coefficient is the equivalent of the coefficient C familiar in naval architecture, but with the effective resistance, R_e, used in place of actual towrope resistance. The quantity $(V/\sqrt{L})$ is the "speed-length ratio" of naval architecture, where V = speed in knots (1 kn = 1.152 mph) and L is the waterline length in feet[18]. In dealing with slow speed amphibians it is usual, and adequate, to use the overall vehicle length for L rather than the waterline length.

TABLE II

SOFT SOIL MOBILITY INDICES AT GVW,
WHEELED CARGO AMPHIBIANS, 1942-1960

	General	Fine Grained Soils		Sands
	MF_E [13]	VCI [14]	NUGP, (psi) [15]	G_{FK} [15,16]
DUKW	77	71	14.3	3.9
SUPERDUCK	79	79	16.0	3.9
DRAKE	78	142	17.0	3.8
LARC V	111	62	13.1	2.2
LARC XV	99	214	18.9	2.5

where:

$$NUGP = \frac{2W}{bd}, \text{ psi}$$

W_l = unit wheel load, lb

b = undeflected tire section width, in

d = undeflected tire outside diameter, in

$$G_{FK} = NUGP\ (36/b.d)^{\frac{1}{2}}$$

TABLE III

COMPARATIVE STILL WATER SPEED AND RESISTANCE AT GVW
WHEELED CARGO AMPHIBIANS, 1942-1960

	$V_K/\sqrt{L}$	R_e/W	$\frac{R_e/W}{(V_K/\sqrt{L})^2}$
DUKW	0.95	0.28	0.32
SUPERDUCK	1.00	0.31	0.32
DRAKE*	1.04	0.31	0.29
LARC V**	1.26	0.42	0.27
LARC XV**	1.07	0.36	0.32

*With propellers extended, wheels partially retracted

**Production model with shrouded propeller

where:

V_K = max still water speed at GVW, knots

L = overall length, ft

W = GVW, lb

R_e = effective resistance, lb

= $325 \times HP/V_K$

HP = installed gross horsepower

All five machines have full wide, deep, scow-like overall forms which are basically poor for the speeds achieved ($V/\sqrt{L} \approx 1.0$), carry their land running gear as extensive exposed appendages, and are driven by propellers of limited size operating under poor conditions in ridiculously bad propeller tunnels. The inescapable hydrodynamic facts concerning displacement amphibians were outlined by Nuttall and Hecker in 1945[19] and by McEwen in 1947.[20]

In 1955, Witney reported the results of towing tests on a number of available full-size amphibians, including the DUKW and the WWII amphibious jeep.[21] He presented his results in terms of towing resistance/gross weight, (R/W), vs. speed-length ratio, $V/\sqrt{L}$. The range of these tests are summarized in Fig. 1. It is disturbing to note that Witney measured towing resistances for the DUKW and the AMJEEP which are some fifty percent greater than the already high values predicted from tests of relatively detailed scale models, with all appendages, made at the time these vehicles were designed.[3,19,22] However, Davidson Laboratory retests of the DUKW model in 1956, in connection with speeds in the hydrofoil take-off range[1] seem to be in good agreement with Witney's tests in the small range where they overlap. Based on the results of his towing tests plus those of self-propelled speed trials, Witney concluded that the overall propulsive coefficients* for successful propeller driven amphibians lay in the range from 20 to 25 percent.[21]**

Although Roach in 1960 quoted a propulsive efficiency (effective horsepower/shaft horsepower) of 42 percent for an early experimental LARC V when fitted with a partial propeller shroud,[7] the actual performance of the production machine appears fundamentally little different from that of its cohorts.

The relative insensitivity of the performance of this type of amphibian to minor design details, in the face of the high fundamental loading,*** poor forms, and propeller limitations, is illustrated by the values of the

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*As used in this report without qualification, the propulsive coefficient is the overall efficiency defined by the ratio: towrope horsepower/gross installed horsepower; the towrope horsepower includes both air drag and the resistance of all appendages.

**In shipwork the propulsive coefficient is of the order of 60 percent or more.

***One accepted measure of loading is the displacement-length ratio $\Delta/(L/100)^3$, where Δ is the displacement in long tons, L, the waterline length in feet. For the amphibians under discussion, this ratio lies between 300 and 400. Well-designed ships for operation in the same range of speed-length ratios have values from 50 to 130.[18]

overall "effective drag" coefficient $((R_e/W)/(V/\sqrt{L})^2)$ in Table III. These are all of the same order, despite the fact that the several vehicles differ considerably in installed power, in hull refinement, in the absolute extent of appendages, and in the sophistication of propeller arrangement. The hulls of the LARCs, for example, are pleasingly faired, their land running gear exposure is cleanly arranged, and their single propellers are fitted with partial, low tip clearance shrouds. The DRAKE was able to improve performance by using an arrangement whereby, during deep water operation, its two propellers were extended down and away from the hull, partially out of the tunnel, and by exploiting the air-suspension of its eight wheels to achieve some modest wheel retraction.

The clear lesson from this considerable experience is that there can be no substantial increase in the water speed of military amphibians without a radical change in their mode of operation and hence of their form.

4. THE BASIC TECHNICAL PROBLEM OF HIGH WATER SPEED

Achieving high water speed poses difficult problems in boat design,* even without the multitude of constraints added by the amphibian features and by the definition of the military problem accepted in the early part of the study. These technical problems are well understood in principle. Fig. 2, taken from the most recent edition of the Society of Naval Architects' Principles of Naval Architecture,[18] illustrates the fundamental, first-order problem of the drag of a boat as a function of its water speed, weight and length. Comparable data for a 165 ton hovercraft from a recent (1968) paper have been added to generalize the picture further.[23] In this figure, typical drag per unit of weight (R/W) is shown as a function of the speed-length ratio $(V/\sqrt{L})$ for well designed craft of four basic types:

1) displacement boats, which are supported in the water essentially by hydrostatic forces;
2) planing boats, which are supported, once the speed-length ratio exceeds about 2, largely by hydrodynamic forces on its bottom;

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*Although the ACV or GEM type of amphibian could have been studied, that concept was barely invented -- by others -- at the time (1957-59) this study was conducted. In an excess of practicality, only "boat" types of configuration were in fact investigated.

3) hydrofoil craft, which, when flying ($V/\sqrt{L} > 2$), are supported by hydrodynamic forces upon submerged foils ; and

4) for an order of magnitude comparison only, a large, modern hovercraft which is supported by a layer of positive pressure air.

In order to see clearly the meaning of these quasi-dimensionless curves in the present context, consider a boat of the length and weight of the 6-mph WWII DUKW; i.e., 31 feet long and weighing 20,000 pounds. Consider additionally that "high speed" means 30 mph (26 knots). This arbitrary craft would operate at a speed-length ratio of 4.7. If it were a good displacement boat, it would require a towrope pull of 4600 pounds to maintain speed; a planing boat, 2900 pounds; a hydrofoil craft, 2200 pounds.

While the spread between the displacement boat and the hydrofoil at this speed is over 100% of the latter, even the hydrofoil resistance is intrinsically high. Moreover, the towrope power (pull x speed) for the hydrofoil is about 175 HP. Due to various drive and propulsion inefficiencies, such a unit would require 350-400 installed horsepower to achieve this speed; the good displacement boat would require about 900 installed horsepower. In comparison, the installed power in the DUKW was 91 HP.

The problem of allocating weight and space between powerplant and cargo in a fixed envelope was discussed by Todd, from whose 1958 paper[24] Fig. 3 is taken. At the zero power end of the scale, maximum cargo capacity is achieved, but speed is zero, while at the other end, all carrying capacity is expended to the powerplant, resulting in a hot rod with no useful cargo capacity at all. Obviously the proper answer must lie somewhere between. Todd suggested that the point of maximum cargo momentum was optimum. Even this simple criterion, however, is a function of the mission profile; i.e. land and water distances, cargo priority, hatch rates, etc.

The strong dependence of drag upon the speed-length ratio shown in Fig. 2 immediately suggests that a significant lengthening of the effective hull might help matters. For example, coupling six DUKW-size displacement hulls together into a single 30-mph unit would reduce the operating speed-length ratio of the coupled configuration to only 1.9. If the resulting shape were clean and efficient, the drag ratio of the coupled configuration would drop accordingly to 0.075, and the average drag per coupled unit, to 1500 pounds.

Fig. 2 is concerned with good ordinary boats, in which no compromises have been made with features necessary in an amphibian. Fig. 4 (which includes data from Fig. 1) presents a more realistic picture of the amphibious truck problem[21,25]. As noted in the previous section, current and past wheeled amphibians operating in the displacement mode do so with their wheels and sometimes other parts of their land running gear partially exposed. Because of overall size limitations, they are relatively heavy, and hence badly shaped, as compared with boats of the same length. The result is that their drag is usually 4 to 6 times that of the corresponding boat at the same speed-length ratio. The heavily loaded fair planing hull suitable for an amphibian[1], shown in Fig. 4, has a drag at operating speed which is 30-40% higher than a good boat, (Fig. 2) and even the experimental Flying Duck hydrofoil amphibian,[1] once it is flying with its wheels clear of the water surface, still has a drag some 60-80% higher than that for the naval architects' idealized hydrofoil craft.

This stuc concentrated on exploring the practical possibilities for wheeled amphibious trucks utilizing planing hulls, and for operating simpler displacement units coupled when at sea to form a single long hydrodynamic body. At the same time the potential for hydrofoils was concurrently under examination at another facility[26] and was specifically excluded from this study.

Because they involve quite difference considerations, the planing hulled concepts and the train ("Sea Serpent") concept are considered separately in most of the following sections. The planing hulled machines are treated first.

5. GENERAL APPROACH

The 195[illegible]-59 study began with a simple, broad operational analysis to determine the water speed range of potential interest; i.e., to answer the question "What is high speed?" Concurrently, U.S. Army and U.S. Navy organizations and facilities then active in various aspects of amphibious warfare -- doctrine, training, operation, equipment specification and design -- were contacted in an effort to develop a reasonable definition of the jobs to be done by high speed amphibian trucks and of the basic constraints within which designs must be conceived.

Fundamental relationships among design and performance features, on land, on water, and between, were cataloged and examined for interactions and explicit and implicit limitations. The opening operational analysis, the interviews, and the fundamental technical relationships together created an envelope of design targets and constraints which was expressed as a series of guidelines for the ensuing study designs.

There followed a series of preliminary designs of amphibious trucks within those guidelines, using a number of ideas then relatively new in detail. These preliminary designs generated requirements for various towing tank studies, whose results, along with those from other related ongoing tank tests, were fed back into subsequent interations of the study designs. The detailed results of these tank studies are summarized in Vol. II of this report[1].

After a number of cycles, the study designs were finalized and their evaluation on a cost basis was begun. These cost studies were not completed at the time the program was redirected. At that time, they had shown no significant operating cost differences among the several study designs. The incomplete figures comparing system operating costs using the proposed high speed amphibian trucks with system costs using competitive vehicle types -- helicopters, hydrofoil amphibians, etc. -- are now so out of date that their publication at this time, in their present form, would serve no useful purpose; updating and completing them is well beyond the intent of this effort.

6. PRELIMINARY OPERATIONAL ANALYSIS

In order to place some general bounds on the overall problem, a simple analysis was made among the following lumped system performance parameters, without regard to technical means by which these might in fact be achieved:

Average operating water speed at full load	V_W	mph
Average operating land speed at full load	V_L	mph
Vehicle cargo capacity	C	tons
Water distance, one way	D_W	mi
Land distance, one way	D_L	mi
Total distance, one way $(D_W + D_L)$	D	mi
Net hatch rate = unloading rate	R	tons/hr
Number of vehicles required per hatch	N	-

In this analysis steady state condition was assumed and all one-way vehicle loads were considered to just equal rated capacity (or to average at rated capacity). A constant time of 10 minutes was assessed per round trip for crossing the surf line. Refueling and routine service were assumed to take place concurrently with unloading. Unloading rate was assumed (purely for simplicity) to be equal to the net shipside hatch rate. Operating speed of the vehicle returning empty was assumed 25% greater than full-load speed, whether afloat or ashore. Finally, loading and unloading occupies one vehicle each.

Under these assumptions, the expression for the number of vehicles required to service a single hatch continuously is:

$$N = 2 + \frac{R}{C}\left[\frac{1}{6} + 1.8\left(\frac{D_W}{V_W} + \frac{D_L}{V_L}\right)\right]$$

Despite its great simplicity, N offers a good first-order evaluation of the size of the problem. The actual numbers of vehicles continuously in operation at one time become large when the problem of unloading several ships at a substantial distance is considered, particularly if hatch rates are significantly increased over the present ridiculously low levels. For example, unloading a single ship (3 hatches) discharging 50 ton/hr/hatch approximately 14-1/2 miles at sea and dumping 1-1/2 miles inland with vehicles capable of 10 mph on land and an extreme of 50 mph in the water, would require, continuously, 16 15-ton or 36 5-ton amphibian trucks to keep up with the unloading capability of the ship (see Table IV). At the present 5-7 ton/hr/hatch rate a ship would keep only 7 15-ton vehicles or 10 5-tonners busy.

The ratios of the numbers of units required to service a hatch fully with vehicles of different capacities and water speeds (N_{C,V_W}) is also enlightening. Fig. 5 shows, as a function of water speed, the ratio of the required number of 5-ton vehicles of a varying water speed to the required number of 5-ton, 5-mph vehicles ($N_{5,V_W}/N_{5,5}$). This relationship is shown for two hatch rates (R = 10 and 50 tons/hr/hatch), each at two total distances (D = 4 and 16 mi). Route breakdown in Fig. 5 (and also in Fig. 6) is 90% water, 10% land. Average land speed (assumed partially off-road) is taken at 10 mph in all cases.

Fig. 6 is the same picture ($N_{15,V_W}/N_{15,5}$) for 15-ton carriers.

TABLE IV

NUMBER OF AMPHIBIANS REQUIRED IN CONTINUOUS OPERATION TO SERVICE A SINGLE HATCH

Steady State Assumed

V_L = 10 mph

$D_L = 0.1 \times D$, $D_N = 0.9 \times D$, $D_W = 0.9 \times D$

Hatch Rate	5 tons/hr/hatch				50 tons/hr/hatch			
Cargo Capacity of Amphibian (C)	5 tons		15 tons		5 tons		15 tons	
Total One-way Distance (D)	4 mi	16 mi	4 mi	16 mi	4 mi	16 mi	4 mi	16 mi
Amphibian Water Speed (V_W)								
10 mph	2.9	5.1	2.3	3.0	10.9	32.5	5.0	12.2
30 mph	2.5	3.3	2.2	2.4	6.6	15.2	3.5	6.4
50 mph	2.4	3.0	2.1	2.3	5.7	11.7	3.2	5.2

7. WHAT IS HIGH SPEED?

For a logistic amphibian in a military operation, speed has various kinds of relatively intangible values, against which it usually incurs substantial direct dollar costs. Positive values are the reduction in vulnerability of the unit to enemy fire and the decrease in time required to respond to varying battle requirements. Increased operating speed may also reduce the number of some direct operating personnel (drivers) but may also increase the number of others (mechanics). Many of the costs of speed will be substantially more tangible. For example, the total installed horsepower required will increase with the speed of the individual units, simply because the power required increases with speed. Other costs of speed, tied more-or-less directly to horsepower requirements are the important elements of initial price, fuel consumption, and maintenance.

The simple lumped parameter analysis of the preceding section gives some more concrete guidance in answering the question of what is "high speed" for logistic amphibians. Figures 5 and 6, for example, show a decided leveling off in the number of machines required at water speeds of the order of 20-30 mph, regardless of delivery distance and hatch rate.

Figure 7, derived from Figures 5 and 6, shows the same thing in other terms. It presents the percent reduction in number of machines achieved by doubling their water speed from the value shown on the abscissa. Thus, as shown in Figure 7a, the doubling of the speed of a 5 ton/5-mph unit operating over a 16-mile distance with a hatch rate of 10 tons/hr/hatch (to 10 mph) reduces the required number of machines by 30%, whereas further doubling its speed (from 10 mph to 20 mph) reduces the number required by a further 20%, and the next doubling (20 mph to 40 mph) by 16%. The total reduction from 5 to 40 mph is thus only 53%.

These curves, considered together with first iteration estimates of power required, propeller performance and power plant weights for vehicles in the 5- to 15-ton payload range, led to the decision that the study designs for high speed amphibians should be targeted upon a full load, still water speed of 30 mph. A 1960 Army study[2] reached the same conclusion by essentially the same route, but current Marine Corps targets are tougher. These call for the same order of speed in sea state 3, which is characterized by waves up to five feet high.

8. WHAT SIZE HIGH SPEED AMPHIBIAN TRUCK?

World War II experience with the 2-½ ton DUKW (which in favorable situations often carried up to 4 tons) clearly showed that a machine with a larger cargo capacity was desirable. Transportation Corps experience in post-World War II years with experimental vehicles so large as to be unroadable under all but the most carefully controlled traffic conditions, on the other hand, showed, that acceptability in general on-road traffic is also highly desirable. Accordingly, it was early determined that at least some of the study designs should be of the maximum size which could still reasonably be considered roadable.

In specifying the envelope dimensions for roadability, experience in traffic with such machines as long distance buses and rubber-tired construction equipment was consulted, rather than statutory limits. This led to the still somewhat arbitrary election of target overall planform limits of 40 feet long by 10 feet wide and a first iteration estimate that a 30-mph planing hulled amphibian of this size would have a gross vehicle weight of 25,000 pounds and a rated net cargo capacity of 5 tons.

In addition, it was decided to explore the feasibility of a larger planing machine with no roadability restrictions. A review of figures from an ongoing ORO study, subsequently reported in 1957[27], indicated that some 75% of resupply cargoes were packageable in units of 4 tons or less. Ten percent were in units of 4-14 tons, and the remainder of the tonnage consisted of such outsize units as tanks, bridging equipment, etc. These figures, plus a companion search for available components in the light of the first estimate power and running gear requirements, led to the selection of 15-ton net cargo capacity as the target for the largest of the sample designs.

The lumped parameter analysis was again used to examine the potential of this size of vehicle relative to the roadable (5-ton payload) machine. The results are shown in Figure 8, which gives the ratio of the number of 15-ton carriers of a given speed required to service a hatch to the number of 5-ton carriers of the same speed to do the same job ($N_{15}, V_W / N_5, V_W$) as a function of total distance. Two hatch rates (R = 10 and 50 tons/hr/hatch), each at two water speeds (V_W = 5 and 30 mph) are shown.

Figure 8 indicates that, at currently accepted hatch rates (< 10 tons/hr/hatch), it takes at least six 15-ton/30-mph machines to do the same job (in terms of tons of small lot cargo) as ten 5-ton/30-mph machines, out to total one-way distances of the order of 30 miles. It also indicates that the larger size machines are more advantageous at slow water speeds and large distances. Both situations improve with hatch rate, but in balance it appeared from this rough analysis that emphasis in the further design study should be placed upon the maximum roadable vehicles. The 15-ton machine was accordingly not considered in the same detail as the smaller units, but was rather worked out to demonstrate scale effects upon the planing concept.

9. THE IMPORTANCE OF IMPROVING HATCH RATES

Stephens, in his 1952 report previously referred to[5], pointed out that the rate of ship-to-shore delivery of a given number of existing slow-speed amphibian trucks could be increased many times without any need for new technology simply by using them properly and by increasing hatch unloading rates. At late as 1960, accepted hatch rates for general military cargo were in the leisurely range of 5-10 tons/hr/hatch[2]. Stephens demonstrated during his WW II field work with the DUKWs that with proper shipside and boom rigging, good job organization, and effective hatch gang motivation, over-the-side loading of DUKWs in favorable sea conditions from ordinary cargo ships could regularly proceed at rates of the order of 30 tons/hr/hatch.

It may be shown by a minor alteration to the basic lumped parameter equation that, out to a total one-way distance of the order of 6-10 miles, increasing the intermittent hatch rates from 5-10 tons/hr/hatch to 30 tons/hr/hatch will improve the total daily delivery of a given number of 5-mph amphibians as much as or more than increasing their water speed to 30 mph and continuing to load at the lower hatch rates. Thus, from a system performance viewpoint, first priority should be given to raising hatch rate standards and targets to new but realistic levels, and to providing the training, incentives, and detailed equipment necessary to make them workable.

Figures 5-8 show clearly that hatch rates must be substantially improved in order to make any high speed amphibian truck system reasonably effective in relation to the current 5-7 mph systems,'expecially at modest transport distances. Thus, from Figure 5, at a total distance of 4 miles, six 5-ton/30-mph amphibians are required to do the work of ten 5-ton/5-mph vehicles if the hatch rate (for both) is only 10 tons/hr/hatch, but only four of the high speed units are required to replace ten of the slow ones if the hatch rate (for both) is 50 tons/hr/hatch.

It is clear that high-speed amphibians will not show to advantage within the logistic resupply operation unless hatch rates are greatly improved. In fact,. the influence on the operation of improvements in hatch rate alone cannot be overemphasized. (This point was also stressed in the 1957 ORO study[27] and the 1960 Army study[2] already referred to.)

10. BASIC GUIDELINES FOR THE STUDY DESIGNS

Once the full scope of the amphibian truck problem began to emerge from the initial studies, iteration of major interrelated factors produced a list of apparently feasible and consistent performance objectives and general constraints to guide the 5-ton planing amphibian study designs.

These self-imposed guidelines, some of which have already been touched upon, were essentially as given in the list following. Although they were always considered subject to change as the work progressed in detail, the option was not widely exercised. The guidelines were broadly interpreted in transferring them from the 5-ton planing to the 15-ton planing and to the Sea Serpent concepts, however.

1. General: The machines were to be designed for flexible use in unloading conventional cargo ships lying at sea, conveying their cargo across exposed sand beaches or prepared shoreline areas, over reasonable off-road terrain and/or on roads as available, to inland transfer or dump points. They were to be full amphibians, designed for extensive, effective operation ashore as well as afloat in unprotected waters. They were to be based upon current and conservatively projected state-of-the-art, so far as mechanical components were concerned.

2. Size: The basic 5-ton planing amphibian designs were to be of maximum roadable size, as follows:

   Overall width in on-road configuration: 10 ft desired maximum
   12 ft absolute maximum

   Overall length: 40 ft maximum

   Height: vehicles were to be suitable for rail shipment. If overall width exceeded 10'-4", the notion was that they could be shipped on beam end with wheels retracted, provided that the overall height in this configuration was reducible to 10'-4" or less.

   Payload: maximum consistent with size and other requirements and constraints. Estimated -- 5-ton net.

3. Water Speed: 30 mph at gross vehicle weight in still water.

4. Static Water Stability: Units were to have 24 in. metacentric height* (minimum) when loaded with a "full and down" 5-ton-size CONEX container[28] having a gross weight equal to the net payload capacity of the vehicle. To achieve this, the beam was to be increased as necessary up to 12 feet. If this still did not do the trick, payload was to be reduced. Minimum range of stability in the same unfavorable load condition was to be approximately 50°.

5. Cargo Provisions: Cargo was to be carried on a clear, flat, self-bailing deck ("wet deck") providing at least 25 sq.ft. of cargo area per net payload ton, and of a size and shape to carry a single 5-ton-size CONEX container. The deck was to be unobstructed for overhead loading, and to be suitable for unloading and loading by a large off-road forklift when ashore. Minimum static freeboard at the cargo deck was (quite arbitrarily) to be 20 in. when loaded to rated capacity.

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*Approximately the minimum value considered "safe" in generalizing field experience with the WWII DUKW.[4]

6. Off-Road Mobility: No reduction was to be tolerated in the level of soft soil mobility below that of the better amphibian trucks and all-wheel-drive military cargo trucks then current. Soft soil mobility, and soft sand mobility in particular, were to be improved if at all possible. There was no expectation that sufficient increases in mobility could be made simultaneously with the jump in water speed or to make any signficant change in the extreme mud operation or riverbank egress limitations of then current amphibians and off-road trucks. Accordingly, an Eklund Mobility Factor of 100 was set up as the design minimum.

The basic dimensional envelope was to have the following features:

ground clearance:	18 in. minimum
angle of approach:	30° approximate minimum
angle of departure:	25° approximate minimum
break angle:	10° approximate minimum

Minimum gradeability was set at 60%. Reasonable wheel suspension for off/on-road ride and conformance to major terrain irregularities was considered desirable.

7. On-Road Performance:

Speed on good level pavement:	40 mph desired minimum 30 mph absolute minimum
Minimum turning radius:	35 ft. desired

8. Surfability: At least to DUKW capabilities. This was considered to dictate a minimum beaching speed of 6 mph, and a hull with high lift bow and stern sections closed against swamping.

9. Slow Speed Water Performance: When afloat, the vehicles were to have controllability at slow speed and when stalled on a spring line (as during shipside operation[4,5,7])at least to the standards of the DUKW, for adequate surfability and for good shipside maneuvering and manners.

In relation to the 15-ton planing study design, dimensions were to be the minimum consistent with the payload capacity assumed. A rear loading ramp and other provisions to make the unit suitable for the carriage of military vehicles up to 15-ton GVW were to be provided. Stability was to be evaluated with three 5-ton CONEX units or with one 15-ton GVW vehicle aboard.

In concepting the Sea Serpent units, the 5-ton net payload was assumed rather than maximum roadable dimensions in order to permit ready comparison with the planing machines. Overall dimensions were, accordingly, to be minimized. Target water speed was, for the coupled units, approximately twice that of current conventional amphibians -- i.e., 16-18 mph -- when assembled into practical length trains.

11. INTERACTION MATRICES

Design is a process of continual compromise among competing requirements and constraints. The more varied the operations a machine must perform, and the more varied the environments in which it must perform them; the more numerous, complex, and interrelated are the compromises involved. The design of a high speed amphibian truck is, by any standard, a complex design problem.

A series of simple interaction matrices among major performance and design features at two upper tiers of design delineate the gross areas where compromises must be expected. The first of these, Figure 9, shows two levels of interaction (1 = primary, 2 = secondary) between general design features (considered as the independent variables) and general performance areas (dependent). Figure 10 presents the broad picture of interrelations between pairs of general design characteristics, again at two levels. In this matrix, characteristics across the top are considered independent; those down the left side, dependent. Thus there are two entries for each pairing of features. Reading down a column indicates the extent of the influence of the column feature on each dependent row feature; reading across the row for the same feature shows the extent to which it is affected by the other features when they, in turn, are considered as the independent (column) features.

Figures 9 and 10 forecast questions of wheel retraction and wheel-well doors which are not discussed until the next section, but they are essentially self-explanatory.

Each independent design element also has similar interaction problems. As some of these second tier matrices are presented it will become clear that the several matrices, together with the general design guidelines, define a close-fitting envelope about possible solutions.

12. PLANING HULL DESIGN

Wheel Retraction, Yes or No

There is no practical possibility for a planing hulled amphibian in which, during high-speed water operation, the wheels and land running gear are not fully retracted into the basic clean hull envelope clear of the water flow over the hull bottom. Figure 11 illustrates the magnitude of the drag increment at low speeds chargeable basically to exposed wheels. The curves summarize the results of towing tests on a scale model of the XM157 DRAKE 8x8, amphibian in which the model was tested complete and then with its wheels removed and the wheel cut-outs filled in to produce a fair hull[1]. The increment in the drag of the wheeled version over the fair body is substantial, resulting in a drag coefficient $\left(\frac{R}{(\rho/2)\cdot A\cdot V^2}\right)$, for the total drag of all eight wheels only, which is of the order of 2 over the speed range tested.* Similar tests run on the LARC V model[25] showed increments which, normalized on the same arbitrary basis, also give drag coefficients of the order of 2.

Despite their crudity, these figures may be used to form a first-order estimate of the increase in drag which might reasonably be expected on a 5-ton planing hull if its tires and wheels are not retracted. At 30 mph (using $C_D = 2$) this is a staggering 13,000 pounds - nearly 40% of the originally projected gross weight of the entire machine, and more than twice the expected basic hull drag. Enough said.

*For simplicity, the area, A, is taken as the projected frontal area of the two leading tires exposed below the fair hull line.

The wheels may be retracted out of the flow into wells in the hull and the wells provided with closures or wheel well doors to make the hull fair. However, the wheel well doors present profound structural problems. Due to impact pressures while running in a seaway, forward doors and their supporting structure and hardware must be designed for loadings of the order of 2000 lb/sq.ft.[29]* On land, running cross-country, the opened doors are exposed to all manner of potential abuse unless retracted completely within the hull structural envelope. An alternative to using wheel wells with closures, which was not appreciated until after the study design work was completed, is to retract the wheels into open-sided recesses in the fair hull, and to so shape the hull in the area of these recesses that, when planing, the water flow separates cleanly at the leading edge of the recess and realigns smoothly with the fair hull line aft without generating massive drags. Figure 12 shows one of the towing tank tested scale models of this concept. While these tests indicated that the hull discontinuities increased specific hull drag by about 25% as compared to a clean hull[1], the tradeoff (added power for reduced complexity and vulnerability) appears attractive. However, due to reasons given in the introduction, no study designs exploited this concept within the framework of the initial guidelines were made; therefore the full impact of this approach upon vehicle stability and beam requirements, structure, weights, tire sizes, and general performance was not consistently evaluated.

General Planing Hull Form Considerations

Planing hulls are used primarily to achieve low drags at high speeds. The basic factors affecting drag of a V-bottom hull at a given planing speed are the hull deadrise angles, the bottom loading, the longitudinal location of the center of gravity, and the basic length-to-width ratio of the hull[30,31,32]. In order to achieve the lowest drag at full speed in still water, the hull deadrise in the planing area should be small. On the other hand, in order to reduce impact forces and drag increments when

*Note that stowage may be so arranged that the retracted wheel helps to support the wheel doors.

operating in rough seas, and hence to be able to maintain a reasonable portion of the boat's still-water speed under these conditions, deadrise should be relatively large, especially forward. Lowest drag up to modest planing speeds is achieved with a relatively long, narrow hull and a distinctly sternward center of gravity location. For lowest "hump" drag (the transitional speed range over which lift changes from essentially hydrostatic to hydrodynamic, just before planing begins in earnest) the longitudinal center of gravity should be a little further forward. Low drag is generally favored by low bottom loading.

The kind and extent of the relationship between some of the major hull design parameters and overall performance are summarized in Fig. 13. In this matrix, the hull parameters are considered independent, and their reactions upon performance are indicated by "+" and "-". A "+" indicates that performance in the column category will generally be improved by an increase in the corresponding hull feature; a "-" indicates that the performance will generally be degraded. Figure 14 indicates the first and second order inter-relationships between hull design features only. The columns, as in Fig. 10, are considered independent variables, the rows dependent. Interactions of land running gear, which is the primary interface between land and water design features, with hull design features are also suggested.

These gross generalizations serve to crystallize some of the problems peculiar to the design of a planing hull for an amphibian. The severe dimensional constraints imposed by land operations, combined with the high gross weights resulting from the dual purpose structure and the necessary carriage of land running gear, lead to bottom loadings of 100 to 200 lb/sq ft, which only begin where pleasure and work boat experience generally leave off (40 to 100 lb/sq ft)[30-34]. The dimensional constraints also limit the scope for accommodating the important rough sea problem. For instance, ground clearance and roll stability on land are favored by low (or even negative) deadrise. Deadrise at the bow is tied to problems of tire size and number, wheel retraction, angle of approach and the generation of bow buoyancy in surf operation. Again, the longitudinal center of gravity (and/or the axles) must be so located as to provide proper tire loadings as well as a hydrodynamically favorable longitudinal center of gravity location.

Three Basic Hull Types

Three basic planing hull types were considered in the study designs. These will be called the Lo-V, Hi-V, and W hulls. The Lo-V hull (Concept 1) is characteristized by low deadrise and a chine carried low until well forward, in order to permit full housing of the wheels with a relatively small retraction distance (Figures 15, 16, 17 and 21a). The W hull (Concepts 2, 4 and 5) is an inverted-V hull, with vertical sides which permits more favorable accommodation of the land running gear than the Lo-V form (Figures 18, 19, 20, 21b, 24, 25, 26 and 27). The Hi-V hull (Concept 3) is a more normal appearing boatlike hull with desirable high deadrise forward, in which the chine forward is deliberately raised to permit the front tires, when fully extended for land use, to be operated and steered completely clear of the wheel wells (Figures 22 and 23).

Although scale model tests showed some possible propeller aeration in the W hull layout,[1] later tests of the one-half scale model of this type of hull in 1960 did not reveal any such tendency.[35,76] The 1960 test bed utilized twin, over-the-stern propellers however, so the question is not fully resolved, for the study designs incorporate large, single screws.

Examples of each type of hull were scale-model tested in the towing tank early in the program, generally at lower gross weights, and hence lower bottom loadings, than the final study designs. The results of these tests are summarized in Fig. 28. It appears from this figure that when compared to the "good" boat of Fig. 2, the first-order compromises used to adapt these hull types to the amphibian problem have increased the basic hull drag over the planing speed range by 30% or more, and have generally increased the hump resistance even more. The compromises made were, essentially, that the hulls be short and narrow for their displacement; that they not taper in beam from amidships to the transom as on properly designed boats; and that their forefoots cut back in varying degrees to achieve reasonable angles of approach for land operation.

The more than proportionate increase in hump drag is typical of short, overloaded boats, but is undoubtedly aggravated in some cases by the bow compromises.

As the study designs were developed by successive iterations to meet the basic guidelines, displacement increased, beams increased, longitudinal centers of gravity were adjusted, and numerous details of the hull shape were further altered. The final resistance curves used to calculate the performance of the completed designs were estimated by extrapolation from the earlier tank curves; guided by tank test data from other amphibian programs and basic reference works on planing hulls, such as the paper by Murray[30] and more recently those by Clement and Blount[31] and Savitsky[32].

The final study design resistance curves are shown in Figs. 29 through 33. These estimates include allowances for appendage and air drag as well as bare hull resistance. Shifts in longitudinal center of gravity with loading condition are also accounted for. In general, this effect was to shift the LCG slightly forward of the optimum position in the light running condition and aft of the optimum in the overload condition. The magnitude of these shifts was a function of the overall layout and the corresponding location of the cargo space.

13. WATER PROPULSION

Selection of an appropriate water propulsion system for the planing amphibians involved design for two distinct modes of operation: high speed operation during the main transport phase and low speed operation with good maneuverability when alongside the ship, when loitering awaiting a load, and when passing from land to water or water to land through the surf zone, where a speed of 6-10 mph is adequate. In the beaching operation especially, the propulsion gear must operate in a protected position.

It was immediately apparent from the thrust and towrope powers involved at high speeds that strenuous efforts were required to obtain respectable propulsive coefficients. The 20-25% values[4,7,19,20,21] obtained at much lower powers in the extreme propeller tunnels of slow speed amphibians were clearly out of the question. Thus if a screw propeller was to be used, some scheme whereby it was operable in a normal boat environment for high speeds, and in a protected location at low speeds, would be required.

Alternatives to screw propulsion were examined briefly at the outset of the study. The most obvious alternative was some form of water jet propulsion. Preliminary calculations, based upon scant information on such important parameters as inlet efficiencies, indicated that a net jet area equivalent to a 24" diameter outlet would be required to achieve reasonable propulsive efficiencies.*

A brief and unrewarding exploration was also made of the use of a smaller, high velocity jet system with cascaded static jet pump elements to improve efficiency. The state-of-the-art in water jet propulsion in 1956 did not include working installations of the size and power apparently required. Work on water jet propulsion since that time does not yet appear to have caught up with the basic requirements as then set forth (cf. Ref. 42). Experimental installations reported in the literature are still only toys in relation (cf. Ref. 36-39). The decision was taken to proceed on the basis of the well-documented screw propeller.

Use of the screw propeller still involved many problems aside from the development of a reasonable dual-operation retraction scheme. Propeller loadings and space limitations, which, with tip clearance and shaft angle considerations, dictated the upper limits of propeller diameter, were such that it did not appear that cavitation could be avoided. Accordingly, propeller performance estimates were made on the basis of data on cavitating propellers. The original calculations were rechecked using recently published data on supercavitating propellers[40] and some minor adjustments made. Calculated net thrusts for supercavitating propellers (3-bladed, 33" diameter x 20" pitch for the 5-ton vehicles, 55" x 33" for the 15-ton machine) are superimposed upon the gross resistance curves for the several final study designs shown earlier in Figures 29 through 33.

*These have since been verified by recalculations using relatively more up-to-date component efficiencies[36,37].

A number of propeller retraction schemes amenable to the required dual operation were considered, and two selected for elucidation. The first scheme is illustrated in Figure 34. A single screw with its struts and an appropriate torque reacting high speed rudder are mounted on a retractable tunnel roof. For high speed operation, this roof is lowered hydraulically to complete the fair hull of the unit, and propeller and rudder are in the normal position for a planing boat.

The necessary constant velocity universal joint in the propeller shaft operates at a small angle in this maximum torque mode. Joints of this type are widely used in automotive work, but would require special development to carry the high torque and thrust loads involved, and to live in the marine environment.

The cavity above the tunnel roof drains once the vehicle is planing, but will be filled during "takeoff." This will add some 2000 pounds of apparent weight to the vehicle and hence is reflected in an increase in "hump" resistance as compared to a completely fair hull.

For beaching, loitering, and shipside operation, the tunnel roof is retracted, placing the propeller in an inefficient but protected position, and bringing into effective use a larger rudder mounted in the permanent tunnel roof aft of the retracting roof section. Should this prove advisable provision may be made elsewhere for automatically limiting power available in this configuration in order to protect the constant velocity joint when operating at the large angle involved.

The second arrangement of screw propellers studied was a variation of the right angle "over the stern" drive, in a basic arrangement which dates back to an Ericsson auxiliary sailing ship of 1845.[41] This arrangement is illustrated in Fig. 35. As finally proposed two propellers were used, which could be swung in a transverse plane so as to operate beneath the fair hull line for high speeds, or protected in a shallow tunnel for low speeds. Tractor propellers were initially selected in order to reduce cavitation (in the original propellers) and to permit the incorporation of rudders on the propeller nacelles, since the side swing arrangement did not readily adapt to steering by swiveling the propellers. It is recognized that the rudders shown in Fig. 35 are too small for the purposes intended and that a better design than that illustrated would have to be employed. It was envisioned that these rudders would be incorporated on the after-side of the lower hub drive housings, and means and suggested to keep these vertical throughout the swing range of the drive. The layout incorporated a dual gear drive to keep the hub diameter reasonable in spite of the high torques which were to be transmitted and a differential to insure load sharing between the input gears. Such a dual shaft arrangement had recently been successfully constructed for the Navy by the Waste King Corporation of Los Angeles[64] and a similar concept has since been proposed for still higher power installations.[65] Although two supercavitating propellers of 24 in. diameter appears adequate for the job, propellers of up to 28" diameter could be accommodated on the study design which employed the final version of this basic arrangement.

14. LAND RUNNING GEAR

Figures 36 and 37 present matrices illustrating, respectively, the interplay of major land running gear design features and gross performance, and the mutual interactions of the design features. In the interfeature matrix, hull design is also included, as a lumped unit, because it is the principal interface between land running gear design and the total vehicle.

The most fundamental source of conflict, of course, if the size and number of tires required to insure the desired level of off-road performance. The problem is aggravated by the absolute requirement that the tires be retracted for high speed water operation. Note that the "maximum roadable" guideline under which the 5-ton study designs were developed effectively ruled out retraction schemes (since used on some Navy test beds) which increase the width of the vehicle on land over that when afloat. It was found that the beam required to obtain the desired level of roll stability when afloat was generally greater than the "desirable" 10-foot limit, so that there was, by this self-imposed rule, no room to work outside the hull beam when ashore.

It was also apparent that the necessarily large wheel wells (or side cavities in a doorless arrangement) seriously affected roll stability by reducing water plane area at or near the beam limit. This effect was lessened somewhat in some layouts by holding the wheel track to normal road vehicle dimensions and providing polyfoamed buoyancy cells outboard of the wheel wells. This arrangement also allowed the stowage of single wheel well doors (under the stability cells) outboard of the wheels, where they belonged, if at all.

If the wheel wells must be big enough to allow pivoting of the large tires within the wells for steering, the static roll stability situation deteriorates further. Such big wheel wells also increase the size, and hence the complexity and vulnerability, of the wheel well doors required. For these reasons, consideration was given in several of the study designs to steering on land by means of frame articulation, as on the then upcoming Army GOER vehicles.[43] Yaw motion only was incorporated, however, to minimize the difficulties in maintaining a fair hull for high speed planing.

Wheel retraction for high speed planing is absolutely necessary. At the time the study designs were completed, the need for wheel well closures was also accepted. A high degree of complexity relative to existing vehicles was unavoidable. Accordingly, one overall design object was to keep the mechanical systems as simple and rugged as possible without sacrificing those refinements essential to the desired performance. To this end, special effort was made to keep wheel well doors small, single, and operable by a simple rotary motion (Fig. 17). It was planned to pressurize the wheel wells* with bleed air from the power plant to avoid carrying any significant amount of entrained water, but the use of seals on the wheel doors was not planned.

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*Nonetheless, for safety, static roll stability was calculated on the basis of the freeflooding water plane.

The wheel retraction itself involved many relatively new problems, despite extensive prior art in the aircraft field, for amphibian truck wheels must all be driven. At the present time the most obvious solution to driving wheels which must retract is to use in-wheel hydrostatic motors. This possibility was scouted in the beginning of the 1956-59 study, and appeared feasible when and if suitable components were developed (as they now largely have been). However, it was felt that the problem could also be solved mechanically without depending upon further hydraulic component developments, and that this should be illustrated. Accordingly, all of the study designs utilize some scheme of mechanical wheel drive. The only exception is the 15-ton machine, upon which in-wheel hydrostatic motors and integral reduction gears, intended for intermittent use, are shown on the front wheels.

Tires

Tire size was determined by the guideline objective of obtaining an Eklund mobility factor of 100 or more. The Eklund load formula relates tire dimensions and tire loading as follows:[13]

$$MF_E = 100 - 50\left[\frac{W_1}{1.6d_r^{0.91}b^{1.94}} - 1\right]$$

where

W_1 = single tire load, lb.

d_r = nominal rim diameter, in.

b = undeflected tire section width, in.

Because it depends strongly upon tire loading, tire selection became an iterative process as the study design weight estimates developed, and requisite tire sizes were fed back into the envelope dimensions, stability was checked, etc. The final tire sizes selected are shown in Table V. Although it will not be readily apparent from the reduced scale layouts presented later in this report, these final sizes are all a few inches larger in diameter than those drawn. To accommodate these, the decks must be raised, and this has been accounted for in the final dimensions, weights, and stability calculations, but the drawings were not redone. Use of the optional larger tire listed for the 15-ton payload 6x6, however, would require more extensive changes.

It was planned that all of the vehicles would incorporate an integrated central tire inflation system designed to permit rapid alterations in operating tire pressure from the cockpit. Where off-road performance is a major problem, overlooking such a direct means for extending the range of performance is shortsighted in the extreme. A schematic for accomplishing a fail-safe, integral system has since been proposed in a more recent study.[44]

It will be noted in Table V that flexible, low tread sand service tires were specified. The possibilities for using the then-new, wide, low pressure rolligons[59] and terra-tires[45] were briefly explored, but they did not lend themselves to the layout requirements, which distinctly favored narrow tires to accommodate the propeller(s) and to simplify the structure, drive, suspension, and wheel well door design.

Use of folding tires such as were then under beginning study for STOL application by the Fairchild Aircraft Company[46] was also examined in hopes of reducing the problems of stowing the large retracted wheels. Although the Fairchild development did not look suitable, recent developments in folding passenger car spare tires[47] and large aircraft tires[60] which reduce their stored diameter by some 130% of their section height, suggest that this line of inquiry might profitably be reopened.

Advantage was taken, however, of the tire collapsing idea to the extent of making the height of the stowage wells less than the tire diameter. By partially deflating the tires through the central tire

TABLE V

TIRE SIZES SELECTED FOR HIGH SPEED AMPHIBIAN STUDY DESIGNS

Payload	5-ton		15-ton	
Layout	6x6	4x4	6x6	optional 6x6
Tire size	18.00-25	21.00-25	29.5-29	33.5-33
Ply rating	8	8	12	10
Approximate inflation,				
Highway, psi	22	20	25	20
Off-road, psi	12	15	15	15
Overall dimensions				
Diameter, in.	61	67	75	87
Section width, in.	19	24	31	33
Tread		Low Skid Sand Service		
Weight per tire, lb.	370	410	1010	1380

inflation system during the retraction operation, they may be stowed at low inflation in a considerably deflected (rather than folded) condition. It was planned that necessary air for rapid reinflation up to about 10 psi would automatically be bled from the gas turbine compressors, and that higher pressures would be obtained from the basic air brake system.

Running Gear Layout, Steering, and Retraction

The combined requirements of the planing hull design and the tires, and the proper loading of each, dictated much of the study design running gear layout. Two basic arrangements were considered: 4x4 and 6x6.* In addition, two basic schemes for land steering were studied: ordinary powered Ackermann steering of the front wheels (which involved either large wheel wells or raising the fair hull line clear of the tire in its land position) and steering by frame articulation (which required the fair and proper structural joining of two watertight, structurally sound hull sections). Although the combination of these features in the study designs was somewhat arbitrary, the pros and cons of these alternatives, which involve basic wheel retraction methods and suspension objectives as well, are most easily outlined by describing the study layouts.

6x6 with Steering by Articulation: Five-ton Concepts 1 and 2 (and the 15-ton Concept 5) illustrate the 6x6 arrangement utilizing frame articulation for land steering (see Figs. 16, 19 and 27). The wheels are retracted and stowed by pivoting in fore and aft planes without excessive or extraneous wheel motion. All wheel well closures are single doors of minimum size and complexity and can be arranged to stow inside the hull envelope when the vehicle is operating on land. Steering is accomplished hydraulically under full servo control about a king pin over the front axle. A positive dead-ahead lock is provided for use during high speed water operation. The operator's cab may be on either the front unit (Concepts 1 and 5), or on the rear unit (Concept 2) (see Figs. 38 and 39).

*A tricycle gear with a single wheel under the forefoot was also briefly looked at, but at the time (and perhaps unfortunately) it was considered to raise more problems than it solved.

The design of adequate lightweight structure at the joint, and the preservation of a fair body when afloat, present obvious but not unsolvable problems. Use of a small step in way of the hull joint to mitigate the fairing problem was tank tested, but the results indicated that the cost in drag was substantial.[1] Accordingly, the use of a rugged inflatable seal to prevent internal circulation losses, was envisioned to close the arced bottom joint. Operation of the seal was to be automatic and interlocked with the high speed joint locks (see Section 16, Controls, p.46).

It was imperative in the 6x6 layouts that wheel suspension be provided in order to assure complete ground contact, and hence proper flotation and traction, in reasonably uneven off-road terrain. This had considerable influence upon the selection of suitable wheel retraction schemes, as will be seen.

4x4 with Ackermann Steering: Concepts 3 and 5 explore two possibilities for use of Ackermann steering with the still larger tires required on a 4x4 vehicle. In concept 3 (Fig. 23), the fore part of the hull has high deadrise and a high chine so that when in the land operating position, the front wheels may be steered under, and clear of, the hull. As a result, the front wheel wells and doors are only of the size required to house the front wheels at one steering angle only. In the study design, accomplishment of this arrangement cost considerable wheel retraction motion, double wheel well doors forward, and the elimination of front wheel suspension.

In the four study designs just described, vertical wheel retraction was achieved by moving the wheels in a fore and aft plane on links. Concept 4 (Fig. 25) illustrates an arrangement whereby retraction is accomplished by rotating the wheel, its final drive and basic support, as one unit, about a centrally placed fore and aft pivot line, so that in the stowed position the assembly is upside down with the final drive outboard of the wheel (Fig. 40). In this arrangement the wheel well closure is a rugged fender integral with the wheel assembly. It swings naturally into the proper position when the wheel unit is rotated 180° for storage, and is completely out of the way during land operation.

This greatly mitigates the wheel well closure problem, and so makes it look practical to steer the front wheels within the wells, which must in any event be wide to permit rotation for stowage.

In the study design the space required to swing the wheel assemblies governed the overall with of the vehicle rather than loss of waterplane inertia to the large wheel wells, with the result that this is the widest of the 5-ton concepts, and has the greatest static roll stability. While it would not be inconceivable to incorporate a reasonable wheel suspension in the rotating stowage assemblies, it was decided that this might be one complication too many on a study design already replete with unusual machinery.

Wheel Drive, Retraction, and Suspension

Wheel drive, retraction, and suspension were treated as the performance requirements of a single integrated subsystem. Two such subsystems were devised using wheels mounted on trailing arms, and still another was outlined to meet the special requirements of mechanical drive in the "flop-over" wheel retraction concept just described.

Although all of the arrangements necessarily involved extensive new components, these were all within the current engineering art. Because the drive-retraction-suspension subsystems all started with a relatively clean slate, it was possible to devise them within the following common set of detailed guidelines:

1. Despite the great powers which were to be installed for high speed water operation, it was decided that the land drive system should be scaled completely to the much lower power and torque levels required for an on-road speed of 40 mph and a full load gradeability of 60%. This implied some method for insuring that the land drive train was never subjected to the full available installed power.

2. In order to reduce torque transmission requirements through the retraction-suspension linkages and thus to reduce stresses and weights, it was decided that the required new wheel drives should incorporate a substantial in-wheel final drive reduction.

3. For safety, service brakes were to be directly at the wheels. Exposed single air/oil disc brakes, which had performed well in the 1956 field trials on the XM147E3 Superduck, were to be used throughout.

4. All wheels were to be driven at all times. Differentials were to be incorporated as necessary to insure proper torque distribution among the wheels and to guard against drive-line "windup." These differentials were to be self-locking or lockable under driver command.

Chain Case Drive: In the first wheel drive subsystem examined, each wheel was carried on two parallel trailing arms which allowed sufficient vertical wheel motion for both retraction and springing, and which transmitted drive and braking torque reactions to the hull (Fig. 41). The upper arm doubled as a case for a chain drive. A 3:1 spur gear final reduction of the type then used by the Walter Truck Company[48] was provided in each wheel. While it was recognized that the Walter layout presented potentially more serious sealing and gearing problems than did available, coaxially-driven planetary wheel reductions, it appeared to lend itself better to the double linkage chain drive layout, wheel stowage, and inboard power train layout.

The use of swing half-shafts as an alternative to chain case drive was ruled out in the study designs using this type of retraction-suspension linkage, because the in-hull space required between the wheels was pre-empted in the stern by the retractable propeller tunnel system used. This is an example of interaction between this drive-retraction system and the propulsion system.

A second major interaction existed with the hull. For proper functioning, the inboard pivots of the trailing arms had to be relatively low and attached to substantial hull structure, suggesting use either of a W or a Lo-V hull. When the basic chain case drive scheme was utilized with a Hi-V hull, it appeared necessary to stiff-leg the front wheels, eliminating the torque reacting arms and springing (Concept 3, Fig. 23).

The chain case drive-retraction-suspension subsystem included the use of a single hydraulic cylinder at each wheel for retraction. These cylinders, used with appropriate accumulators and throttling valves, became fully adjustable load carrying hydropneumatic springs and shock absorbers when the wheels were in their land operating configuration.

Chain case drive was incorporated in the 15-ton study design. In this much heavier machine, it was thought advisable to relieve the suspension links and chain cases of some of the moments generated by tire side loads. This was accomplished through the use of an inboard hub-end slipper captured on an arcuate rail on the hull behind each wheel (see Fig. 27).

Friction Roller Drive: A second, more radical wheel drive scheme involved friction drive to the surface of the tires (Fig. 42). This arrangement was suggested by the successful operation of William Albee's Rolligon vehicles[49], which were at the time the subject of widespread interest. While Albee's wide Rolligon bags appeared inappropriate for high speed planing amphibians, the notion of friction drive was attractive for several reasons:

1. Drive to retractable wheels could be accomplished with all mechanical drive elements stationary in the hull, so that the driven wheels were to all intents and purposes undriven so far as the retraction mechanism was concerned.

2. A low torque, high speed 4x4 drive train layout with the roller drive providing a simple final drive reduction could be used to power four, six, or eight wheels.

3. Although the system essentially utilizes only tire deflection for springing, deflection under load occurs on two sides of the tire, so that twice the effective travel of an ordinary unsprung tire is available. The axle motion allowed by the upper tire deflection may be damped. This system is only reasonable for wheels which are not steered individually, so that its use in a design dictates the adoption of steering by frame articulation.

Of course, friction drive proposals immediately raise the bogey of slippage between the drive roller and the tire. While it appears that most conditions where this might occur are already immobilizing conditions even when the wheels are positively driven, it was thought that proper design of the drive roller could do much to alleviate what might remain of this problem. As a starting point, an openwork "squirrel cage" construction was suggested which permitted water and mud to work through it from the drive surfaces to an open hub end. Cleaning action could be further augmented by a static internal screw. In addition, the selective use of the wheel retraction hydraulic system (in a hydro-pneumatic spring configuration) to increase or decrease drive roller-to-tire contact forces and use of the central tire inflation system to control contact areas were envisioned as regular parts of the operating procedure.

<u>Flop-over Wheel Drive</u>: The flop-over wheel retraction scheme invites the use of hydrostatic wheel drive even more than any of the other arrangements. A mechanical drive such as the 4x4 layout illustrated (Concept 4, Fig. 25) nonetheless appeared (barely) feasible also. Features of this layout are the drive and steering couplings which are retracted hydraulically to disengage the drive from the wheel assembly prior to retraction.

As already pointed out, the flop-over layout requires beam in proportion to tire diameter (and, if it is extreme, tire width), and this requirement increases if the wheels steer. In the study design, the mechanical drive arrangements pre-empted the space where a retracting tunnel propeller might have gone. As a result the dual side-swinging, right-angle drive propeller arrangement was adopted. However, the large spaces occupied by the two rear wheel systems prevented use of the mechanical arrangement for the swing propellers as originally conceived (Fig. 35). The same general objectives were achieved by the adapted configuration shown in Fig. 25, which employs retractable rudders mounted to the hull in place of the rudders mounted on the propeller nacelles.

15. THE POWER PLANT AND TRANSMISSION SYSTEM

The overriding fact in assembling a suitable power plant for these machines was that, for high speed water operation, continuous real shaft power required would be in the range of 1000-1500 HP, even for the 5-ton units. In addition, weight was critical. The briefest review of potentially available power plants shows that the only reasonable choice is a gas turbine.

The advantages of the gas turbine in vehicles were already recognized. For a given power, they are light and compact. They are self-cooling, and may use a variety of fuels. They have generally proved to be highly reliable. The first cost of gas turbines per horsepower is gradually being reduced, along with specific fuel consumptions, which latter are now of the order of 0.4 - 0.5 lb/SHP/hr for the latest units over the power range from about 40% to 100% of full power.[50] Gas turbines having a free power turbine have "steam engine" torque characteristics which eliminate the need for hydrodynamic torque converters in land drive systems.[61] In marine use the free turbine will drive a fixed pitch propeller at essentially constant horsepower regardless of changes in loading, giving the propeller some of the advantages of controllable pitch when it is overloaded, but potentially the system has dangerous overspeed propensities when the propeller is momentarily free.[51]

Some of the basic problems with gas turbine installations per se were also well known. They require large air flows as compared to internal combustion engines, and are particularly sensitive to back pressure. They require careful protection from dust and spray ingestion. (More recent Navy experience has shown that they are also prone to erosion and corrosion due to the ingestion of salt water particles of almost colloidal size which are ever present in the marine environment [51,62,63,66].) Finally, the idling and low-load fuel consumption of gas turbines is high.

The low-load fuel consumption problem appeared most serious. A maximum of about 250 HP was all that could be utilized properly by the 5-ton amphibians on land and in water operations with the propellers in protected position for beaching. Further, in actual field use, a fair part of their running time would be spent loitering or loading at shipside. The concept of a dual power plant, since successfully used in the Swedish "S" tank,[52] appeared to offer a satisfactory solution.

A basic layout was developed in which a turbine with the high power required for planing was teamed with a 250 HP unit (Fig. 43). The simple, rugged turbine of the Boeing 502, 520, 550 series[53,61] was selected for the smaller unit rather than a gasoline engine, for example, because of basic fuel, space, and weight compatibility. Ev though this turbine did not have a low specific fuel consumption, it was estimated at the time that within its power range fuel savings would be from 60% at idle to 10% at full power when compared to a large, single unit.

This arrangement also provided ready means to insure low power at the proper times, and a source of emergency power, either to get home on, to get over the "hump" with, or to produce an extra burst of speed.

Some penalties were of course incurred. The first cost and the weight of two turbines is each greater than for a single turbine, especially if it is decided to do without the overload power capacity provided by the smaller unit. In addition, the collector gear case for distributing power to the water, land, or land-and-water propulsion systems will probably be somewhat heavier and more complex in the dual turbine system.

The complete dual turbine power package included the collector which also was a speed reducer, a normal marine reverse and reduction gear,* and the powershifting portion (without orque converter) of the Allison HT six-speed automatic transmission.[54] Clutches for selectively disconnecting the two turbines from the drive were also to be included. Possibilities for reducing windage losses of an unfired turbine to acceptable levels, so that these would not be needed, were not explored.

The package was compact and reasonably light. It fitted handily in the space available under the "wet" cargo deck as shown in Figures 16, 19, 23 and 25. The problem of emergency access to the engine room when loaded was not satisfactorily solved. Although in some of the layouts it appeared possible to provide an unobstructed crawlway, the extensive duct work required to handle efficiently the large a volume required by

* Size and weight was assumed as for the unit used with the big Packard engine on World War II P.T. boats.

the turbines would probably make any meaningful repair work in the restricted space unlikely.

The dual engine scheme was used with minor modifications in all five of the planing study designs. In the 15-ton design a more efficient 300 HP diesel engine replaced the small turbine.

Power distribution from the power plant to the wheels utilized normal universal jointed automotive drive shafts except for the drives to the front axles of the articulated designs. As laid out, these required use of a constant velocity joint at the differential. Standard automotive self-locking differentials were employed at each axle. Torque splitters, lockable under driver control, were incorporated in the land drive transfer case and axles as necessary. It was considered that the possibility for creep between the drive roller and tire obviated the need for this in the 4x4 cum 6x6 friction-drive arrangement, however.

Attention was given to feasible means for protecting the gas turbines from solid water and spray. The general scheme adopted was to duct incoming air into the bilges away from the turbine air intakes (Figs. 38 and 39). The turbines would then draw their air from the engine room, the presumption being that gravity and distance would separate out the particulate water. The bilges were to be baffled and screened to prevent the bilge water from being splashed about in the compartment before it could be removed by the bilge pumps.

Oversized folding exhaust stacks were carried high to reduce chances of water reaching the hot end of the turbine, and snorkel-type closures were fitted as an added protection. For crew protection, all exhaust ducting was surrounded by a ventilated air space. Necessary sharp bends in the ducts were assumed to incorporate properly designed diffusers to keep pressure losses to a minimum.

16. CONTROLS

It was considered that a key to the technical feasibility of planing amphibians lay in providing the capability to change its configuration radically to suit the particular mode of operation of that moment. By the same token, the key to operational feasibility appeared to lie in making these changes as automatic and foolproof as possible. To this end, all of the planing machines outlined were to provide the driver with the following simple controls

1) Operation selector
2) Steering wheel
3) Land controls
   a) transmission range selector
   b) foot throttle
   c) foot brake
   d) third differential lock -- as needed
4) Combined throttle-reverse gear marine control
5) Static controls
   a) hand brake
   b) winch controls
   c) light switches

The operation selector was conceived as a single-lever control having six sequential positions:

a) ROAD
b) OFF-ROAD
c) BEACHING
d) WATER MANEUVER (for use in close quarters generally, alongside ship, and while loitering)
e) WATER CRUISE
f) WATER MAX SPEED

The positions should be arranged that the lever could only be moved one position at a time, and only to a position next to the one it was in. Feedback interlocks were to be provided so that it could not be moved to the next position until all of the actions called for by the position it was originally in were satisfactorily completed. The principal functional characteristics of this control are shown in Fig. 44, which is essentially self-explanatory. The control system was visualized as employing air valves and air-operated slave elements to actuate prime hydraulic and air control valves located directly at the units to be controlled, or at the appropriate power source, as most convenient in each case.

In addition to the main featues shown in the figure, the following operations were also to be automatically controlled:

1) Wheel door operation, to be part of the wheel retraction sequence.

2) Land steering control to be disengaged and the land transmission put in neutral as the lever passed from BEACHING to MANEUVER.

3) In going from MANEUVER to BEACHING, the land steering control to be reengaged and synchronized with the rudder and steering wheel position.

4) Likewise, rudder control to be engaged and synchronized in passing from OFF-ROAD to BEACHING, and disengaged when the control lever was moved from BEACHING to OFF-ROAD.

5) Bilge pumps to be engaged (or disengaged) and drain cocks closed (or opened) as the selector passed between the OFF-ROAD and BEACHING positions.

During the shift between MANEUVER and CRUISE the propeller would be extended before the large turbine was fired, and the large turbine killed before retraction began, in order to protect the marine propeller shaft universal joint from high torque loads while at large angles. In addition, in switching from one turbine to the other, the currently operating turbine would not be cut out until the other was operating satisfactorily. An inconveniently located auxiliary control, permitting operation in the CRUISE configuration of the small turbine only, would be provided for emergency use in the event of a malfunction of the main turbine while at sea.

The steering wheel would control the vehicle heading in all modes of operation. The element or elements actuated -- land steering system, rudders, or both -- would be automatically selected, engaged, and synchronized by the operation selector, as just described. Steering ratios in each case would be selected to give, insofar as possible, essentially compatible heading response rates in all three basic steering modes, land, beaching/maneuver, and cruise/high speed. Both land and water steering would be hydraulically powered under hydraulic servo control.

The basic land controls would be all air-servo operated.

The marine speed control combined throttle and marine forward-neutral-reverse operation in one continuous single-lever operation as is now universally used in small boats. Selection of which turbine (or both) was thus at the driver's command but would be made automatically, as part of the operation selector function. The marine speed control also used a simple air servo as the control linkage.

Static controls would be mechanically or electrically linked directly to the driver's station where possible.

The planing amphibian control system outlined is relatively complex, but so is the operation. Accordingly, it is not considered a luxury but rather a necessity. It lets the engineer determine and direct the sequence of all changes of operating configuration in a system with many options, most of which are potentially destructive to some or all of the machine, and perhaps to its occupants as well. In effect, the operation selector provides a built-in set of instructions and a built-in check list. Such integrated controls are essential in order to reduce driver skill and training requirements and, if properly accomplished, reduce accidents and hence reduce downtime and increase availability.

Such a control system would obviously have to be fully engineered and developed, and carefully manufactured, if it were not itself to become a major bottleneck. While many usable bits and pieces are available off-the-shelf for a possible test bed, a final system would undoubtedly require the design and development of special components involving consolidation of functions into rugged, field replaceable modules adapted to a marine environment. These would be special, and not cheap. But it was considered that a reliable system functioning generally as outlined would be essential to the practical success of any machine of this kind in the hands of the troops.

17. HULL STRUCTURE AND MATERIALS

Gross weight is a critical characteristic of any planing boat, for the power required to achieve a given planing speed is essentially directly proportional to it. It is doubly important in an amphibian, because the size and weight of the land running gear and its supporting systems are also direct functions of the total weight. Studies have shown that in a fully rationalized passenger car design, a one pound increase in the weight of a single on-board component will result in a total increase of 1.5 to 2 pounds when all other structural, power train, and running gear elements are properly readjusted.[67,68] This "cascade" effect is undoubtedly compounded in an amphibian, because two separate support, propulsion, and dynamic load-carrying systems are involved.

The hull structure of an amphibian vehicle, which accounts for about one-third of its empty weight, is the principal area where the designer can exert significant direct influence upon total weight without embarking upon a major component redevelopment program. By the same token, realistic projections of hull weights are essential during preliminary design studies.

A review of materials and structures potentially applicable to planing amphibians was made at the outset of the study in 1956. Although fiberglass-reinforced-plastic (FRP) materials had already broadly penetrated small boat construction, then-recent experience with an FRP hull on the XM148 5T 6x6 GULL slow-speed amphibian suggested that FRP technology was not sufficiently advanced to count upon the early use of FRP for the hull of a planing amphibian. The final combined hull-frame structure of the GULL weighed a remarkable 17,500 pounds, as compared to less than 5,700 pounds for its steel-hulled contemporary, the XM147-E2 4T 6x6 SUPERDUCK. At the time this was attributed in part to the necessity to reinforce the FRP hull in so many places to take the land-borne load concentrations and modes.

Thin-skinned steel construction of the type used on the SUPERDUCK was not suitable either, particularly for the bottom of a planing amphibian, because of the local strength required to withstand pounding loads when operating at planing speeds in small seas. In a 5-ton planing machine these loads will regularly reach 2000 lb/ft^2 or more when running at 25 to 30 mph in 2- to 3-foot waves (sea state 3).[29]

Sheet stiffening by means of external framing, which accounts in part for the low weight of the SUPERDUCK hull, was also unsuitable because at planing speeds, the drag increment which would be incurred was unacceptable.

Successful Navy experience with a series of experimental 85-foot aluminum-hulled motor torpedo boats built in the late 1940's indicated that the planing amphibian hulls should be of welded aluminum alloy. Recent figures for planing work boats of relatively simple construction show that the steel hulls weigh 60 to 80 percent more than comparable aluminum hulls.[34] Despite their favorable strength-weight characteristics which had made aluminum alloys essential in aircraft structures for years past, their marine application had been delayed by high material costs, joining problems, and concern over corrosion. The WWII growth of aluminum production facilities, and the development of corrosion-resistant and weldable alloys opened the way. And the Navy PT's, plus numerous other post-WWII aluminum boats, demonstrated that control of galvanic corrosion by isolating dissimilar metals, avoiding stray electrical currents, and using sacrificial anodes, was practical and reliable.

Preliminary designs for the planing amphibian study concepts in welded aluminum alloy were calculated and sketched to form a realistic basis for weight estimates and to check out first-order possibilities for efficiently integrating structural and mechanical layout. Special problems were raised by the many discontinuities in the fair hull for wheel wells, propeller retraction, steering articulation (where used), etc. Th general structural solution envisioned was the use of two essentially c inuous longitudinal bulkheads, one inboard of the wheels on each side, as main structural members. The hull outboard of these was considered more nearly as flotation tanks than as primary structure, although sight was not lost of the large essentially localized planing, cargo, and shipside loads that would have to be carried by these outboard structures. The proposed hulls were longitudinally framed, with web frames and bulkheads placed so as to support alternately either hydrodynamic hull loads or various concentrated loads such as at wheel suspension attachment points and machinery foundations.

Basic plate thicknesses used for the 5-ton planing hulls were as follows:

Bottom and cargo deck	-	3/16"
Sides, foreweather deck and transom	-	1/8"
All other decks and bulkheads	-	3/32"

In estimating cargo deck structure, allowance was made for 5g cargo loadings which might arise while running in rough water, or during loading alongside ship in a seaway. Gunwale and side structure estimates also included allowances to survive the special beating to which a boat is subject in a shipside environment.

Although the basic plating thicknesses used are in general accord (on the light side) with present practice for planing aluminum work boats,[29,34] the resulting hulls, due largely to necessary redundancy arising from cutouts, etc., are still not light. The lightest, that of the nonarticulated 5-ton Concept no. 2 (Fig. 19), is estimated to be approximately 25 percent heavier than might be expected for a planing aluminum work boat of the same size, while the articulated hulls are up to 50 percent heavier than those on comparable boats.[34] Even at this, efficient design with close attention to weight-saving details would be required to stay within the final hull weight allowances (see Table IX, Section 19).

In the decade since the concept design decision was made in favor of welded aluminum alloy for the hulls, this material has in fact been widely and successfully used in larger and larger ship hulls[69] and in amphibians, from the slow-speed production LARC V's and XV's[11,12] to the experimental 5-ton planing LVW-X1[57] and 5-ton hydrofoil LVHX2[56,70]. In the same period, the use of FRP in ship hulls has also been extended from small pleasure craft to 120-foot fishing boats[55] and projected to deep submergence vessels[71] and Navy minesweepers up to 200 feet in length.[72,73] In these applications, all-up FRP hull weights have been found to be essentially on a par with those of comparable aluminum hulls. Accordingly, the question of FRP versus aluminum for future high-speed amphibian hulls should be re-examined.

In 1960-61, the U.S. Army Tank-Automotive Command designed and constructed an experimental 2-½ ton 8x8 floating cargo truck based upon an integrally bonded aluminum honeycomb hull-frame structure.[74,75] The empty weight of this vehicle the XM521 "Honeybear", was less than 40 percent of that for the standard M34 2-½ ton 6x6 truck. While this remarkable weight reduction was not made entirely in the hull structure, both the material used and, perhaps more importantly, the philosophy used, invite study in relation to future high speed amphibians.

18. THE PLANING STUDY CONFIGURATIONS

Five planing configurations were elaborated in preliminary layouts, weight, and performance estimates. Four were 5-ton payload machines, the fifth a 15-ton machine.

Leading characteristics of the four 5-ton designs are given in Table VI; of the 15-ton unit, in Table VII. Characteristics of recent production and experimental amphibious trucks, comparable in varying degrees, are included for easy comparison.

Renderings of Concepts 1, 2, 3, 4 and 5 are shown in Figures 17, 20, 22, 24 and 26, respectively, and reduced scale layouts of all five are shown in Figures 16, 19, 23, 25 and 27. Photographs of table models of Concept 2 are shown in Figure 45; those of Concept 3 in Figure 46. Note that Concept 3 shows the front wheel doors, to be exposed, whereas, in fact they should retract into the wheel well, to be out of harm's way.

TABLE VI

LEADING CHARACTERISTICS OF 5-TON PLANING STUDY DESIGNS

	CONCEPTS (1959)				EXISTING VEHICLES (1962-64)		
	1	2	3	4	LVWX1[57]	LVHX2[56,70]	LARC[11]
Wheel configuration	6x6	6x6	4x4	4x4	4x4	4x4	4x4
Hull configuration	Lo-V	W	Hi-V	W	Hi-V	Hi-V**	Scow
LOA - ft	40.0	40.0	40.0	40.0	36.6	38.0	35.0
WOA - ft	11.0	10.8	10.7	12.0	11.7	10.5	10.0
HOA - ft	9.7	10.8	11.3	10.8	12.4	11.3	9.2
Reducible for shipping to	6.9	7.7	8.2	9.5	-	-	-
Ground clearance, in.	18	18	24	22	22	20	23
Curb weight, lb.	29,100	30,100	26,500	28,900	34,000	29,210	21,000
Gross vehicle weight	39,100	40,100	36,500	38,900	44,000***	39,210	31,000
Tire size	18.00-25	18.00-25	21.00-25	21.00-25	18.00-25	18.00-25	18.00-25
Max gross HP, marine	1300*	1400*	1175*	1300*	1500#	1120	300
Prop diameter, in.	33	33	33	2-24	2-24	26	29.5
Hull material	Alum	Alum	Alum	Alum	Alum	Alum	Alum
Max still water speed @ GVW, mph	30	30	30	30	41##	42	8.6
Boating speed @ GWV, props retracked, mph	7.5	7.5	8.5	8.0	-	-	8.6
Beaching speed @ GVW, mph	7.0	6.9	7.2	7.0	-	-	8.6
Max level highway speed @ GVW, mph	40	40	35	35	35	41	25

* Includes 50HP allowance for auxiliaries; emergency use of power plant

** With hydrofoils

*** Design target 38,000

\# Continuous

\#\# @ 39,500 lb.

TABLE VII

LEADING CHARACTERISTICS OF 15-TON PLANING STUDY DESIGN

	CONCEPT 5 (1959)	EXISTING LARC XV[12] (1964)
Wheel configuration	6x6	4x4
Hull configuration	W	Scow
LOA - ft	54.0	45.0
WOA - ft	14.0	14.5
HOA - ft	12.3	13.7
Reducible for shipping to	9.0	-
Ground clearance, in.	24	29
Curb weight, lb.	74,100	45,200
Gross vehicle weight	104,100	75,200
Tire size	29.5-29*	24.00-29
Max. gross HP	4100**	600
Prop diameter, in.	55	35
Hull material	Alum	Alum
Max still water speed @ GVW, mph	30	8.3
Boating speed @ GVW, props retracted, mph	7.5	8.3
Beaching speed, as above	6.0	8.3
Max level highway speed @ GVW, mph	22	29

* Alternate: 33.5-33

** 100 HP allowed for auxiliaries

Various systems used in the study configurations have been discussed in earlier sections. Except for a few interactions among major systems, pointed out along the way, the combination of such features as cab location, stack layout, and even propeller retraction scheme in each study layout was largely arbitrary. The manner in which the major features were combined in the several concepts is presented in Table VIII.

In addition, all of the study designs have the following details in common:

1. Self-locking or driver-lockable differentials, or a functionally equivalent system to prevent single-wheel spinout.

2. Central tire inflation control, linked to the "operation selector" control, utilizing turbine bleed air for rapid, low pressure tire reinflation, and brake air for higher pressures.

3. Air operated hydraulic disc brakes working on exposed single discs at each wheel.

4. Hydraulic power steering from a common hydraulic power supply used also for wheel retraction, wheel door actuation, etc.

5. Major operating configuration controls integrated into an "operation selector."

6. No on-board spare tire. "The best place for the spare tire is in the motor pool." (E.T. Todd[24])

7. Life lubricated bearings and/or central lubrication, and outside check/drain/refill access to all machinery fluids.

8. On-board fuel for three hours of water operation at full speed.

9. Low pressure, low flow turbine bleed to wheel wells when planing to reduce entrained water weight.

10. Two 50-gpm electric bilge pumps in each hull unit. Inaccessible spaces foam-filled. Drain cocks. Automatic pump and cock operation via "operation selector."

TABLE VIII

COMPARISON OF FEATURES AMONG PLANING HULL CONCEPTS

CONCEPT NO.	1	2	3	4	5
NET PAYLOAD, tons	5	5	5	5	15
HULL	Lo-V	W	Hi-V	W	W
MATERIAL	Welded aluminum	Welded aluminum	Welded aluminum	Welded aluminum	Welded aluminum
TIRE CONFIGURATION	6x6	6x6	4x4	4x4	6x6
PROPELLER ARRANGEMENT	Retracting tunnel	Retracting tunnel	Retracting tunnel	Dual side swing	Retracting tunnel
LAND STEERING	Hull articulation (cab fwd.)	Hull articulation (cab rear)	Ackermann (steer clear of wheel well)	Ackermann (steer in wheel well)	Hull articulation (cab fwd.)
WHEEL DRIVE	Roller	Chain case	Chain case	Geared	Front-hydrostatic Rear-chain case
SUSPENSION	Double tires plus partial hydropneumatic	Hydropneumatic	Front-tires only Rear-hydropneumatic	Tires only	Hydropneumatic
WHEEL RETRACTION	Fore and aft arms	Trailing arms	Trailing arms	Flopover assembly	Trailing arms
WHEEL DOORS	Yes	Yes	Front only	None per se	Yes
MARINE POWER PLANT	Large gas turbine	Large gas turbine	Large gas turbine	Large gas turbine	Large gas turbine
LAND POWER PLANT	Small gas turbine	Small gas turbine	Small gas turbine	Small gas turbine	Diesel
CARGO DECK	Wet	Wet	Wet	Wet	Wet

11. Lightweight hydraulically operated winch -- 20,000 pounds on 5-ton units; 40,000 pounds on 15-ton unit.

12. "Wet" cargo decks, clear for overhead loading at shipside, arranged with drop coamings to permit over-the-side loading and unloading as by rough terrain forklift. The coamings are foam-filled to aid extreme roll stability, and arranged so as to prevent inadvertent off-center container rolling.

13. Normal side-loading height is approximately 6 feet on the 5-ton units; 8 feet on the 15-ton. Some of the wheel retraction systems may be utilized to "kneel" the vehicle, reducing these heights by about 2 ft.

14a) Cargo deck areas on the 5-ton vehicles provide about 30 sq. ft/rated payload ton, and will accept a full-size CONEX container.

b) The 15-ton unit provides 21 sq. ft/ton and is arranged to carry a fully loaded 2-½ ton 6x6 cargo truck, which may be loaded and unloaded by a folding stern ramp. The ramp is designed to provide buoyancy at the stern when stowed.

15. Mechanically protected, air-inflatable seals on hull joints and cargo deck engine hatches.

16. Surf-resistance cab structures, including 3/4-inch safety glass in all forward and side-facing windows.

19. ESTIMATED WEIGHTS

The estimated weights for the four 5-ton planing study designs, the 15-ton planing design, and the Sea Serpent (which will be discussed in Section 21) are summarized in Table IX. Actual measured weights for the XM147-E2 SUPERDUCK are also given for overall comparison, although they are directly comparable only to the Sea Serpent. It is apparent from the figures for the planing configurations that increased speeds involve substantial increases in the empty weights.

Despite the use of aluminum alloy, hull weights are increased by 50 to 90 percent, largely due to planing loads and structural discontinuities. Steering by articulation when on land is estimated to cost at least one net ton.

TABLE IX

SUMMARY WEIGHT BREAKDOWNS OF STUDY DESIGNS

(All Values in lbs.)

CONCEPT NO.	1 5T 6x6 LO-V	2 5T 6x6 W	3 5T 4x4 Hi-V	4 5T 4x4 W	5 15T 6x6 W	6 5T 4x4 Sea Serp.	XM147-E2 SUPERDUCK (1959)# 4T 6x6 Scow
WEIGHT GROUP							
Hull, frame, decks, doors	9,000	10,000	8,000	9,800	23,300	5,400	5,640
Cab	1,100	1,300	1,100	1,100	1,400	1,000	980
Power Train	4,900	5,200	4,800	5,100	13,200	4,300	3,080
Land running gear w/final drives	8,100*	7,600**	7,000**	6,500***	20,600**	4,000**	5,630
Land steering	700	700	300	600	1,500	400	230
Marine systems	2,000	2,000	2,000	2,500	4,100	1,800	640
Miscellaneous	1,000	1,000	1,000	1,000	3,200	1,000	1,780
WEIGHT EMPTY	26,800	27,800	24,200	26,600	67,300	17,900	17,980
Fuel, fluids	1,900	1,900	1,900	1,900	6,200	1,300	880
Crew & OVM	400	400	400	400	600	400	400#
CURB WEIGHT	29,100	30,100	26,500	28,900	74,100	19,600	19,260
Rated net payload	10,000	10,000	10,000	10,000	30,000	10,000	8,000
GROSS VEHICLE WEIGHT	39,100	40,100	36,500	38,900	104,100	29,600	27,260

* Including squirrel cage drive rollers.

** Including final chain cases and Walter reduction.

*** Including wheel final drives.

\# From actual weighed weights supplied by E.T. Todd of GMC, except for "Crew & OVM."

The weights given in Table IX for the planing amphibians reflect two tacit, hidden assumptions made when the study began in 1956:

1. that high speed amphibians might be required in large numbers; and

2. (not unrelated to the first) that to be acceptable, their first cost would have to be more nearly in line with truck costs than with helicopter costs. By the end of the study it no longer appeared that planing amphibians would be wanted in quantity, but rather that a few of them might be useful to fill a small special niche in the overall amphibious operations requirements picture. However, the corollary to this, that in this framework they would necessarily be expensive, and hence might be acceptable at aircraft prices if the net cost increment produced measurably better performance, was not examined. The cost/effectiveness of designs reflecting the kind of all-out attack on weights undertaken in the U.S. ATAC XM521 program[74,75] (see Section 17) might well favor such a more sophisticated and nominally more expensive approach.

20. PERFORMANCE EVALUATION OF THE PLANING STUDY DESIGNS

Estimated Still Water Speed

The calculated planing performances of the four 5-ton and the 15-ton payload study planing designs are summarized for three load conditions and two power settings in Table X. Propulsive coefficients and val[illegible]r the overall resistance coefficients discussed in Section 3 are a[illegible] given.

The projected still water speeds are calculated on the basis of conservative weight, drag, and propeller performance figures. The net horsepower indicated for each concept is that calculated to just achieve the initial target 30 mph still-water speed at final designed gross weights. They are not intended to correspond to any existing turbines at this time unless by coincidence. They reflect a reasonable allowance for auxiliary power requirements, but none for gas turbine power degradation with time in marine service. Accordingly, the desired initial power ratings of the main gas turbines could still be augmented at this stage to provide for power loss between turbine overhauls. A 10 percent power increase for this purpose would only increase gross weight by approximately one percent, including the cascade effect. Such an increase would not materially alter any of the performance estimates.

As noted in Section 3, longitudinal center of gravity location on a given design was generally not optimum for all loading conditions. In addition, optimum running trim in smooth water is not usually the same as in rough water. The projected speeds shown in Table X reflect this.

Since the study designs were completed, adjustable trim tabs have been successfully applied to planing hulls, making it possible to adjust their running trim while underway[77,78]. Use of such tabs offers possibilities to reduce both hump and running drags (by using different settings) and, more important, to adjust trim for different loadings and to reduce impact and drag levels when operating in a seaway. These possibilities, including that for using the retractable tunnel roof in concepts 1, 2, 3, and 5 to accomplish such adjustment, were not investigated, and no allowance has been made for possible performance benefits from such devices.

TABLE X

CALCULATED PLANING PERFORMANCE OF STUDY DESIGNS

CONCEPT	1	2	3	4	5
Configuration	5T 6x6	5T 6x6	5T 4x4	5T 4x4	15T 6x6
Hull	Lo-V	W	Hi-V	W	W
Net marine HP	1250	1350	1125	1250	4000
Still Water Performance					
Speed @ normal full power, mph					
Empty	41	41	38	39	47
With rated load	30	30	30	30	30
With 150% rated load	22	16	24	16	18
Speed @ emergency full power (land + marine turbines, mph					
Empty	45	44	42	43	NA
With rated load	37	38	35	36	NA
With 150% rated load	30	30	30	19	NA
Propulsive coefficient					
With rated load	.44	.44	.44	.45	.44
Overall resistance coefficients, Re					
R_e/W	.42	.44	.40	.42	.50
$R_e/(V_K/\sqrt{L})^2$	.029	.030	.029	.029	.044
Cargo momentum, Ton-mph					
With rated load	150	150	150	150	450
With 150% rated load	165	120	180	120	405
In Regular Head Seas, 3'x60'					
Speed, mph					
With rated load					
acceleration limited	12	12	14	12	15
power limited	(17)	(15)	(20)	(15)	15

Estimated Rough Water Performance

The all-important matter of speeds attainable in rough sea conditions had not been systematically evaluated at the time the study was redirected. Some scale-model tests had been made, however, of several of the preliminary configurations operating in 3'x60' regular waves (full-size equivalent) in which drag and accelerations were measured[1]. Drag and impact projections from these tests are normalized and summarized in Figs. 47 and 48, respectively. For comparison with good boats similar "typical" drag increments and "average" accelerations by Savitsky[79] are shown for irregular seas, state 3. Savitsky indicates an average wave height for sea state 3 of 2.5 feet. Therefore the tests in 3-foot regular waves, of length equal to 150 percent of the amphibian length, (approximately the most severe for synchronism) are considered to be somewhat more rigorous.

Figures 47 and 48 show that at speeds above 15 mph drag increments and impacts for the sample W-hull, and even for the Hi-V hull, are both considerably greater than for the typical boat. The differences between the Hi-V and W hulls seem explicable. On the W-hull, design compromises to achieve land performance objectives resulted in an oddly shaped, bluff bow, while in the Hi-V study design some land performance features were deliberately sacrificed to retain a more conventional bow. The differences between the Hi-V hull (essentially concept 3) and Savitsky's boat figures presumably reflect more subtle effects of the overall amphibian constraints.

The generalized data of Fig. 47 were used with the curves of Figs. 29-33 to estimate power limited speeds in 3'x60' headwaves. The W-hull curves (in both Figs. 47 and 48) were used not only for Concepts 2, 4, and 5, but also for the Lo-V Concept 1 because the latter had a similarly poor bow. Acceleration-limited speeds shown were selected to limit peak accelerations at a driver's stations (all perilously close to the bows) to approximately 1g.* At some expense in duplicating controls and removing

*A mean value 1g. for the one-third highest accelerations has recently been stated to correspond to a crew endurance of approximately one hour.[81]

the human as a governor, these limits could be raised to the power limits in all cases by providing an after control station for operation at high speeds on water, but this was not done in the original layouts.

In any case, the performance of any of the study designs in 3'x60' head seas falls considerably short of the current target of 30 knots (about 35 mph) in sea state 3.

Beaching and Boating Speeds

In the "boating" mode, wheels and propellers are retracted, and the smaller power plant only is utilized. This economic, low power mode is intended for loafing, queuing, maneuvering at shipside and when maintaining position under the hook on a springline for burton loading in the effective manner developed for the DUKW's during WWII[3]. In the "beaching" mode, used in the transition to or from the water, the wheels are extended and driven.

As listed in Table VI, boating speeds for all five study designs are of the order of 8 mph; beaching speeds, 7 mph. Table VII gives the estimated boating and beaching speeds for the 15-ton concept (No. 5) as 7.5 and 6 mph, respectively. These speeds are considered adequate for the purposes intended, including the operation through the surf zone. Maneuverability, handling, and shipside manners of Concepts 1, 2, 3, and 5, in which a large, slow-speed rudder is automatically and simply brought into play when the propellers are retracted, should be particularly good.

Static Freeboard and Stability

These characteristics are summarized for the five planing study designs and the Sea Serpent in Table XI, for both the designed load condition and the 50 percent overload condition. In all cases the load is considered to be a 5-ton CONEX container, full and down, which leads to a practical maximum height of load center of gravity above the cargo deck. Ranges of stability, on the other hand, do not reflect any buoyancy of the container.

TABLE XI

FREEBOARD AND STATIC STABILITY OF STUDY DESIGNS

At GVW, Loaded with 5-Ton Conex Containers, Full and Down

	GROSS VEHICLE WEIGHT			50% OVERLOAD	
	Freeboard* (in.)	GM (in.)	Range of Stability (deg.)	Freeboard* (in.)	GM (in.)
1 5T 6x6 Lo-V	20	24	60	18	16
2 5T 6x6 W	20	24	45	18	16
3 5T 4x4 Hi-V	20	24	50	18	14
4 5T 4x4 W	20	45	55	18	34
5 15T 6x6 W	24	25	45	19	10
(with M41 5T 6x6)	24	22	40	-	-
6 5T 4x4 Sea Serpent (see Section 21)	20	24	60	17	14

*Minimum, at cargo deck.

The designed load target values set up at the beginning of the study, for a minimum freeboard of 24 inches, metacentric height (GM) of 24 inches, and range of stability of 45 degrees, are all met.* The overload condition, calculated for the same high load, appears to be adequate for operations in relatively favorable sea and surf conditions only, concept 4 excepted. The width of Concept 4 was increased to 12 feet in order to accommodate internal mechanical arrangements, and therefore appears adequately stable for all reasonable sea conditions, even with a 50 percent overload.

On-Road, Off-Road and Soft Soil Performance

Dimensions. weights (and thereby axle loads), and projected highway speeds shown in Table VI, show that all of the 5-ton planing concepts are basically acceptable on-road, although Concept 4, at 35 mph and 12 feet width, might not be welcome. Experience since 1956 with large-tired, unsprung four-wheeled vehicles such as the Army GOER[43] and the LARC's[6] has shown that on-road speeds may be limited not by the available power, but by the development of excessive, largely undamped pitching motions of the entire vehicle at speeds of the order of 20-30 mph. Projected road speeds for Concepts 3 and 4 (and the Sea Serpent) were discounted by 5 mph because of their lack of suspension, but in light of the above, their practical road speeds might be still less than the limit given under some conditions where synchronous bouncing may develop.

The 15-ton unit, Concept 5 (Table VII), is clearly not for highway use, but is rather simply a "beacher."

The component of off-road behavior which received principal attention was soft soil performance. Several indices for the concept designs, first introduced in section 3, are summarized in Table XII, along with the same figures for several other amphibians in being, past and present. The figures show that, except for the 15-ton concept, the study designs represent a small to medium improvement in soft soil mobility over known levels for similar machines. By and large, the gain is most substantial in sand, where the ability to operate in sands one-third or more

*A 1968 study aimed for a minimum GM of 22 inches, and a range of stability of 60 degrees.[80]

TABLE XII

SEVERAL SOFT SOIL MOBILITY INDICES OF STUDY DESIGNS
(and comparable existing machines) at GVW

VEHICLE	TIRES	SANDS	FINE GRAINED SOILS		GENERAL
		G_{FK}	VCI	NUGP (psi)	MF_E
Concept 1 5T 6x6 Lo-V	18.00-25	1.9	66	11.2	108
2 5T 6x6 W	18.00-25	2.0	71	11.6	106
3 5T 4x4 Hi-V	21.00-25	1.7	57	11.5	121
4 5T 4x4 W	21.00-25	1.8	65	12.5	113
LVWX1 5T 4x4 Hi-V	18.00-25	2.8	80	16.0	98
LVHX2 5T 4x4 Hydro-foil	18.00-25	2.8	83	16.2	97
Concept 5 15T 6x6 W	29.5-29	2.8	316	22.5	91
with "alternate" tires	33.5-33	1.9	237	17.5	104
LARC XV 15T 4x4 Scow	24.00-29	2.5	214	18.9	99
Concept 6 5T 4x4 Scow (see Section 2!)	21.00-25	1.4	48	9.4	124
LARC V 5T 4x4 Scow	18.00-25	2.2	62	13.1	111
DUKW 2.5T 6x6 Scow	11.00-18	3.9	71	14.3	77

Note: G_{FK}, VCI and NUGP are indices of limiting low soil strength in which a vehicle will operate, and hence lower values indicate increased soft soil mobility. The Eklund Mobility Factor, MF_E, however, is formed in such a way that higher values project higher soft soil mobility (see Section 3).

weaker may be translated directly into the potentially valuable ability to negotiate dry sand slopes approximately one-fourth greater than can present wheeled amphibians.[16]

The 29.5-29 tires selected for the final iteration of the 15-ton study design (Concept 5) proved still to be smaller than may be desirable from a soft soil viewpoint, although they do not calculate to be very different from current amphibian performance levels in sands and according to the Eklund Mobility Factor (MF_E). In Table XII, an "alternate" tire size, 33.5-33, is listed for illustrative purposes which would bring G_{FX} and MF_E into line with the improved values shown for the other study designs. The design layouts shown for this concept would not accommodate this larger tire without substantial alterations, however.

The additional aspects of off-road performance may also be mentioned. Obstacle ability was for the most part maintained at the modest level of previous low-speed amphibians by providing similar ground clearance and angles of approach, break and departure, and by providing, to the same general degree, suspension conformance on the 6x6 machines. "Ride" in rough terrain may also be considered from an overall design viewpoint to be nominally equivalent to the poor levels of previous amphibians.

Comparisons with Planing Amphibian Test Beds

During the early 1960's the U.S. Navy undertook the development of three distinct designs of high speed test beds. The LVW-X1 and LVW-X2 amphibians were essentially similar planing 5-ton 4x4 vehicles. The LVH-X1 and LVH-X2, of which two each were built, were distinct designs utilizing hydrofoils.

Of interest at this point are the LVW machines. These were of essentially the same dimensions and target empty weight as the study designs, although they went approximately 20 percent over the weight targets. They were of welded aluminum construction. The four tires were sized to give an Eklund Mobility Index of 100, operated without a wheel suspension, and were hydraulically retracted into essentially open wheel wells. Water propulsion was by twin-screws in a retracting tunnel arrangement, turned by a 1500 HP gas turbine. Maximum still water speed of the LVW amphibians was 41 mph at the original design full load displacement of 38,000 pounds. In Sea State 3, power limited speed dropped to 26.5 mph. Crew-fatigue limited speed in Sea State 3, with the crew forward, was placed at 12 mph. Mechanical reliability, of the land mobility system particularly, was poor. As a result land mobility was not extensively tested.[81]

From this experience, and also from similar LVH-X1 and X2 experience[81], it may be concluded that the study concept design projections were reasonably close to reality in most respects. The fact that the LVW's went overweight,and yet appeared underdesigned for land operation,highlights the especial need in this class of hybrid, high performance machine for close, weight-conscious design throughout.

Principal Problem Areas

The 1954-56 study and subsequent experience in metal both demonstrate that amphibians (of any configuration) having high water speeds will inevitably be costly in terms of first-cost, installed power, complexity of design, operation and maintenance, training requirements, and need for such supporting equipment as navigation aids. In particular, there is no cheap, simple, easy way to achieve high speed in rough seas. Dual service, on land and sea, forces design compromises in both environments, increases tare weight, and generally degrades efficiency in either operating mode. None of these problems, evident in the beginning, has disappeared, or even become significantly more tractable in the interim. All may be expected to yield marginally, a few dollars here, a few pounds there, a knot or so somewhere else, to continued engineering research and careful, imaginative design, but any further work which demands more than such painful progress for its justification should not be undertaken.

21. THE SEA SERPENT

The Sea Serpent phase of the study was a first-order investigation of a total system to exploit the speed-length ratio effect upon hull drag to achieve higher operating speeds in the water, using essentially conventional amphibian trucks. As pointed out in Section 4 in discussing Fig. 2, the specific drag of displacement boats is highly dependent upon the speed length ratio ($V/\sqrt{L}$)at which they operate. Most simply, this is because a large fraction of total drag is generated in the creation of surface waves, primarily at the bow and stern. If a number of simple hulls could be closely joined to form one long hull, the speed length ratio for the coupled units at any given speed would be significantly reduced as compared to that for the several units operating singly, and their specific drag ratio with it. The strategem is in fact exploited in everyday Mississippi river towboat practice. The principal difference between a river tow and the Sea Serpent concept is that the former has a single major propulsion system embodied in the towboat, while the coupled Sea Serpent vehicles would form a train of self propelled units, albeit all under the control of a single driver.

While the simplistic figures given in Section 4 illustrate the basic notion, they overlook some related hydrodynamic problems: skin friction drag, form effects, and the drag generated by any unavoidable departures from a fair form, as at the joints in the coupled configuration. They also ignore many other first-order problems peculiar to a coupled arrangement, such as propulsion efficiency, steering, behavior in a seaway, structure, and feasible coupling mechanisms and procedures.

Design Guidelines

The concept of a train of self-propelled units is logically most applicable to individual vehicles of modest size. The investigation therefore accepted 5 tons of net payload per unit in a vehicle of minimum practical dimensions as a reasonable design target. Working with this exact size appeared desirable because it would make possible direct comparisons with the 5-ton planing study designs.

The water speed target for the close-coupled units was set at 16-18 mph when joined into a practical length train. On the temporary assumption that the train operation would not incur excessive additional time penalties, this promised to reduce the number of machines required to do a given job by 20 percent or more as compared to current amphibians, whenever one-way sea distances were greater than 4 miles (see Fig. 7b, Sect. 7).

In order that this modest order of reduction might yet be attractive in final trade-offs, it seemed evident that the individual Sea Serpent vehicles would have to be relatively simple and well within the current state-of-the-art. Inasmuch as some unavoidable increase in complexity was already ordained by readily foreseeable propulsion and coupling problems, for example, this applied particularly to the hull, power train, and land running gear. This resulted in the almost immediate acceptance of an unsprung 4x4 chassis with normal front wheel (optional 4-wheel) steering, and, of course, no such complex refinements as wheel retraction. Likewise, automotive power plants, transmissions and power train components, and simple welded aluminum hulls were also accepted at the outset.

A real but relatively unmeasurable advantage of a train system in which each element is self-propelled, is its evident flexibility. This was considered worth preserving, even at some slight cost. Accordingly, it was early decided that all individual Sea Serpent vehicles should be identical, and so equipped (except, perhaps, for expensive, readily transferable communications and navigation equipment) as to be immediately usable either solo or in any position in trains of from two vehicles on up to the maximum practicable number. This ruled out special bow or stern units, and implied a quick means for properly arranging control function and central control command according to vehicle operating mode and position.

Other, more detailed, self-imposed design guidelines, concerning shipside operation, stability, mobility, surfability, and cargo provisions, were outlined earlier in Section 10.

Operational Concept

It was envisioned (Figure 49) that the Sea Serpent vehicles would operate as individual units at all times when ashore, when passing inbound or outbound through the surf zone, during shipside loading operations, and in short-haul water operations. When operating in trains, a group of vehicles would work together as a team at all times, under the direction of a train leader. Barring some disability, the same Sea Serpent vehicle would lead the train operation at all times, control all vehicles while they are in the train, and carry necessary navigation and communication equipment for the entire team. Transfer of the control function and supplementary equipment to any Sea Serpent was assumed to be so arranged in the design of the subsystems as to be readily accomplished in a matter of a few minutes whenever and wherever such transfer was needed.

The number of units in a train-team (up to some technically feasible maximum) would be selected in the field so that all units could be loaded simultaneously at shipside, one to a working hatch, for one or more ships.

The operation cycle was visualized to begin on the beach with the assembly of all of the Sea Serpent vehicles assigned to a given train-team. They would proceed out through the surf singly, and make up as a train onto the rear of the train as they arrived. The train would then proceed under the control and command of the train leader to the ship loading area, where it would break up by simply slowing down and releasing units quickly, one-at-a-time from the front. Each unit would then go to an assigned hatch, be loaded, and return to an assembly area where it would make up with other units of the train-team, by the same procedure as before, for the return trip. Just outside the surf zone, the train would once more break up, again by slowing and shedding units by release from the front, and individual vehicles would go in through the surf solo. Once ashore, the train was envisioned as moving as a convoy to the unloading area, and then back to the beach.*

Principal operational problems recognized at the outset were close scheduling, and making the train hookup at sea, particularly in rough water. (Of course, the alternate always exists, of operating such a system as individual units when the seas are too rough for efficient coupling, but if this is necessary in everything but calm water, the system will obviously be marginal.)

*The concept of proceeding through the surf and some distance inland, still coupled as a train was at that time untried and not incorporated in this study. Later studies[58,84,85] investigated some aspects of this latter approach.

Technical Problems and Approaches

Resistance: Scale-model tests were conducted of trains of 1, 2, 4, and 8 simple box hulls with various treatments of the gaps between. In some tests the individual hulls were left free to pitch relative to one another at the joints, i.e., were articulated. In others the joints were made rigid.[1] A typical set of data curves from these tests is shown in Fig. 50. While rigidizing and fairing the coupling joints produced reductions in resistance as expected, both were considered impractical. Accordingly, the final projections were based on data for free to pitch, open joints.

Increments to approximate wheel drag were derived from numerous studies of single amphibian hulls in which tests were run with wheels and with faired hulls with no wheels[83] and added to the train results. The simple train tests thus "corrected" were roughly confirmed by the results of a later series of scale-model resistance tests of the tracked LVTP-5 in trains up to five vehicles[58]. Some of the LVTP-5 data are summarized in normalized form in Fig. 51. In Fig. 52 the Sea Serpent "with wheels" data are compared with the LVTP-5 data, in terms of average resistance of a single unit in trains of various lengths normalized on the basis of the resistance of a single unit operating solo.

For the final estimate of train resistance, the percent values of Fig. 52 were applied to the results of scale-model resistance tests of a good model of an early LARC V design having essentially the size, shape and displacement of the final Sea Serpent hull[83]. The resulting curves for smooth water operation are given in Fig. 53.

Water Propulsion: It was expected that problems might arise in efficiently propelling the Sea Serpent train due to wake interferences from one propeller to a following propeller and to a following hull. No tests were run to study this possibility, but a side-swinging right-angle propeller scheme was proposed for use on the Sea Serpent which would permit effective propeller operation in two positions which would not direct water against a following hull, and which could be alternated along the train length to minimize propeller wake training. This is shown schematically in Fig. 54.

<u>Longitudinal Stability of the Train</u>: The model resistance tests clearly confirmed, both visually and quantitatively, the expectation that the lead unit of the Sea Serpent water train would be pushed by following units; i.e., the lead unit coupling would be in compression (see Fig. 55). This effect was greater the longer the train, and resulted in swamping of the lead unit at successively lower speeds as the train lengthened, as shown in Fig. 50.

Tests were run which demonstrated that such swamping could be avoided by a major modification of the bow of the lead unit[1]. This, however, was not considered an acceptable solution as it created the need for two different vehicles. The solution accepted (which was costly in other coin) will use as a lead unit a standard unit run empty at all times, for trains of a length where swamping was a potential problem. From qualitative test results in regular waves[1], it was estimated that trains of two or three units only could be safety operated without this palliative.

<u>Water Steering</u>: As will be discussed in the following paragraph, the simple coupling arrangement selected permitted only pitch freedom between units. No satisfactory method of steering a long train thus rigidized, aside from selective propulsion on one side only, was proposed before the study was terminated. Later scale-model tests of water trains of a proposed high mobility vehicle[44], however, indicated that the most effective steering of such a configuration was achieved by articulating the lead unit about its joint in the yaw plane.

<u>Coupling</u>: In devising a scheme for coupling the Sea Serpent vehicles for sea-going operation the following were controlling considerations:

a) coupling must be possible at sea;

b) coupling must be possible between vehicles which may be differently loaded;

c) the connection must be quickly releasable;

d) it should permit pitch freedom between units, but (basically) restrain interunit yaw and roll, even in a quartering sea; and

e) It should be relatively simple.

(In the light of the water steering problem, some provision for controlling interunit yaw rather than simply restraining it would now be in order.)

The arrangement finally selected to illustrate one feasible system is shown schematically in Fig. 56. Each Sea Serpent vehicle has at bow and stern, incorporated structurally, one half of a long hinge running athwartships. At the stern, each carries a hollow metal, buoyant hinge pin attached to a winch cable. Operation was visualized as follows:

a) To couple: coupling is done by working up the stern of a vehicle. The vehicle ahead releases its "hinge pin" and pays out cable so that it floats off astern. The pin is manually recovered by a crew member of the following vehicle, who seats it in that vehicle's bow half-hinge, where it is clasped by a hydraulically actuated coupler. The vehicle ahead then winches the pin back to it, into its stern half-hinge, and likewise clasps the pin with its coupler. (This could possibly be done mechanically by a two-jointed front coupler.)

b) To rlease: the sternward vehicle may release the hinge pin at any time by releasing its hydraulic coupler.

Final Layouts and Performance Estimates

Leading characteristics of the 5-ton Sea Serpent vehicle, and of the LARC V, are shown in Table XIII. A breakdown of the projected gross weight was included earlier in Table IX (Section 19), static stability ranges were presented earlier in Table XI, and soft soil performance indices (all favorable), in Table XII (Section 20).

Figure 57 shows the final basic Sea Serpent layout. The vehicle is a simple, unsprung, large-tired 4x4 with Ackermann steering and non-retracting wheels. (The possibility for 4-wheel steer is illustrated but not essential.) The clean, scow hull is of welded aluminum. Cargo is carried on a wet deck amidships. Power is provided by twin automotive

TABLE XIII

LEADING CHARACTERISTICS OF A 5-TON SEA SERPENT STUDY DESIGN

(Single Unit)

	CONCEPT 6 (1959)		LARC V[10] (1960)
Wheel configuration	4x4		4x4
Hull configuration	Scow		Scow
LOA - ft	35.0		35.0
WOA - ft	7.3		10.0
HOA - ft	9.8		9.2
Ground clearance, in.	24		2
Curb weight, lb	19,600		21,000
Gross vehicle weight	29,600		31,000
Tire size	21.00-25		18.00-25
Max gross HP	550		300
Prop diameter, in.	2-20		29.5
Hull material	Alum		Alum
Max. still water speed, mph	@GVW	Empty	
Solo	12	13	8.6
Train of 2	13	14	-
3	14	15	-
10	16	17	-
Beaching speed, mph	9		8.6
Max. level highway speed at GVW, mph	30		25

engines also amidships, under the cargo deck. These were originally visualized as large displacement water-cooled V-8 gasoline engines, but in recent years automotive diesel engine ratings have crept up so that diesel engines could now be considered from a weight viewpoint. Engine cooling is by air radiators on land, bottom coolers when afloat, so that the engine compartment does not need large cooling air inlets when at sea. Engine and marine controls were to be operated by air-servos which could be controlled by the unit operator when running solo, or by the train lead unit driver when coupled. A suitably simple scheme which could readily be adapted to this service was demonstrated on the "Jeep Train" in 1964.[85]

Water propulsion is provided by twin side-swinging propellers, operable in three positions, as per Fig. 54.

The "log/cable" coupling scheme is illustrated, with a release operable from either vehicle. The scheme assumed for picking up the coupling log is manual, by a crew member in a special small forward cockpit. The present possibilities for a reasonably simple two-jointed hydraulic grab system as part of the front coupler were not studied.

The Sea Serpent approach does not require controls of the same complexity as outlined earlier for the planing amphibians, because there is no wheel retraction, and no great necessity for power sequencing. On the other hand, there are three different configurations required in a train: the lead vehicle configuration, and two propeller positions to be alternated in successive non-lead vehicles to reduce propeller wake interference. As a result, it was considered that here also an operation selector is desirable to reduce the chances for driver error, and to reduce driver skill and training requirements. Such a selector was visualized as having six essentially self-explanatory sequential positions: ROAD, OFF-ROAD, BEACHING, SEA SOLO/COUPLING/LEAD, SEA SERPENT-ODD FOLLOW, SEA SERPENT-EVEN FOLLOW, which would automatically set tire pressure (if tire pressure control were fitted), wheel drive and steering configuration, propeller operating position, power configuration, bilge pumps and drain-cocks properly for various modes of operation in the general fashion outlined earlier.

In addition to the same land and marine steering and power controls listed for the planing amphibians, the Sea Serpent driver will have two controls which, when in train configuration, release the mechanical link between his vehicle and either the unit in front of him or that astern, plus the boom-winch control for use during coupling. The scheme for manually coupling the units while afloat requires a man at the bow, so that the grappling or engaging control would be placed in the bow cockpit.

A Simple Operational Evaluation of the Sea Serpent

To obtain a first-order evaluation of the potential usefulness of the Sea Serpent concept, once the performance estimates were completed, a steady-state lumped parameter model was constructed to calculate the number of vehicles required to serve a given number of hatches at one time, as a function of sea distance and hatch rate. In this simple model it was assumed that the number of cargo-carrying vehicles in a train was equal to the number of hatches to be served, either on one ship or a small number of nearby ships, so that shipside loading of the train units proceed simultaneously, with the train disassembled.

As expected, the model was highly sensitive to the water speed increase achieved when operating in a train. The conservative effective horsepower requirements projected in Fig. 53 were used, along with an assumed overall propulsive coefficient of 40 percent. The resulting relationship between train speed (V_T, mph) and number of coupled vehicles (N_V) is approximately:

$$V_T = 12 \times N_V^{0.13} \quad .$$

Reflecting model test experience, N_V included one empty lead vehicle when the number of cargo-carrying units required to match the hatches being served was greater than 3. This deadhead was, of course, included in the total number of units needed to do a given job. Three minutes were allowed for each coupling operation between two train units; both at the start of the outward journey when making up outside the surf zone, and again at the start of the return trip, after shipside loading. No time was assessed for uncoupling.

The calculated ratio of the number of Sea Serpent vehicles, operating solo and in trains of 3 and 10 units, ($N_{\text{Sea Serpent}}$) to do a given job to the number of conventional 8 mph, 5-ton amphibian trucks to do the same job (N_{Standard}) is shown in Fig. 58. In these calculations, hatch rate was taken as 7.5 tons/hr/hatch; one-way land distance was assumed constant at 2 miles, average land speed at gross weight was taken as 10 mph, and 5 minutes was allowed for each passage in or out through the surf zone. The 50 percent water speed increase credited to the solo operating Sea Serpent over the standard machine because of its increased installed horsepower and retractable propeller arrangement, shows a substantial payoff without any help from coupled use. Operating in trains of 3 coupled units increases the basic benefits appreciably at distances beyond 10 miles, while 10-unit trains do not appear as effective as 3-unit trains even out to 100-mile one-way distances.

The value of train operation of the Sea Serpents per se is examined more closely in Fig. 59, in which is given the number of Sea Serpents operating in trains (N_{Train}) relative to the number of identical machines required to do the same job when operating at all times individually (N_{Solo}). Other assumptions are as before, except that two hatch rates are shown. The advantage from 3-unit train operation is a reduction of the order of 8 percent in the number of machines required, when sea distances exceed 30 miles, and the 10-unit train again appears to have no place. Variations in (steady) hatch rate do not significantl· change these normalized results.

The influence of train effectiveness on more sanguine estimates of the water speed advantages of coupled operation are shown in Fig. 60, for a 3-unit train and assumed hatch rate of 7.5 tons/hr/hatch. The LVTP5 model tests[55] referred to earlier suggest that, through detailed refinement of the preliminary design, the ratio of 3-unit train speed to solo speed might be increased from the presently assumed 1.15 to perhaps 1.25. However, it would take a factor more nearly like the 1.33 multiplier illustrated to make the train concept per se truly exciting.

The indication that there is no operational advantage in coupling more than 3 units greatly mitigates the potential technical problems discussed earlier, and suggests that the concept might be more appropriately applied to more simple vehicles in the future, as a useful feature rather than as a controlling design objective.

As far as the train concept has been developed here, the optimum 3-unit configuration, despite added power and retracting propellers, will proceed at the stately rate of 14 mph or so. This is clearly not a high speed for many purposes. While trains of 10 units might achieve 16 mph or a little better, even this is relatively slow, and apparently counterproductive in terms of cargo delivery. Accordingly, the advantages of the Sea Serpent concept, as it is developed here, are considered marginal, quite apart from the foreseeable technical and operational problems still unresolved at this point.

22. CONCLUSIONS

At the time the study was redirected in 1959, it had been concluded that 5-ton to 20-ton payload planing amphibious trucks capable of maximum still-water speeds of the order of 30 mph were technically feasible without extensive additional research. Their complexity and cost (in many coins), reflecting the unavoidable constraints of nature, were clearly such, however, that they were not about to replace slower, more prosaic amphibians as the workhorses of over-the-beach operations. As special mission units, on the other hand, the planing amphibian appeared to compete directly with the helicopter, offering perhaps 25 percent of the helicopter's basic speed, plus some small and incalculable gains in the face of sour weather, for well over one-half of their first cost per payload ton-mile-per-hour.

The Sea Serpent concept, as developed through 1959, did not provide the desired breakthrough to high-speed operation either. The final results did suggest, however, that if simple provision for proper 3-unit train operation could be incorporated in more standard amphibian trucks at reasonable cost, advantageous medium-speed operations would be feasible in many operating circumstances.

Cargo handling at shipside between ship hold and amphibian deck still appeared to offer room for the most dramatic improvements in over-the-beach unloading of conventional cargo ships. Improvement in hatch rates, through development of new methods and doctrine and/or inexpensive mechanical aids, appeared even more central to the economics of a high-speed amphibian system than it was to current systems.

The study suggested a number of then-novel mechanical solutions to some of the specific sub-problems of planing amphibians. Some of these have since been independently conceived from the same seeds and tried in metal, generally with favorable results. Principal among these are:

1) Retraction of wheels into open cavities within the fair-flow envelope of the planing hull, eliminating wheel well doors.

> This was utilized on the U.S. Navy LVW and the LVH test beds[56,57,58].

2) Retraction into a deep tunnel of propeller(s), shaft(s), strut(s), and rudder(s) as a unit with a bottom fairing piece.

> This was done on the U.S. Navy LVW's[81], but the contingent possibility to bring a large low-speed rudder easily into play when the propeller(s) is in the retracted position seems yet to be exploited.

3) Use of integrated operation controls to simplify driver responsibility and to increase safety and efficiency.

> This was done on the LVHX2 which provided the driver with a "mode selector" of similar concept[70].

4) Use of a dual power plant to extend the part-load economy of gas turbines.

> This stratagem has been suceessfully employed on the Swedish "S" tank[52].

As yet untried but still considered of potential value in proper applications are:

5) "flop-over" wheel retraction (undoubtedly -- today -- with hydrostatic or electric wheel motors);

6) combination of the Albee friction-roller wheel drive, articulated steering and partial hydropneumatic suspension; and

7) right-angle propeller drive with side-swing stowage permitting three or more alternate operating configurations.

As of 1969, the prior general conclusions appear still correct. Indeed they have been partially validated by actual hardware tests, by more recent studies and by the demonstrable lack of progress with high-speed amphibians during the years between. While further work has resulted in more reliable projections of sea speeds, and projections to more nearly acceptable values, the major unsolved technical problem is still to improve rough-sea performance substantially without making a crippling sacrifice in the land capability which is all that distinguishes an amphibian from a bad boat. And the march of military air developments (both hardware and doctrine) seems to have still further widened the odds against a high speed amphibian truck ever entering service.

RECOMMENDATIONS

It is recommended that any further research of high-speed amphibians be done only under the full realization that they will represent special-purpose machines fulfilling limited operational requirement. There should be no delusion that such craft will eventually replace the workhorse type of amphibian as it is now represented by the state-of-the-art.

ACKNOWLEDGMENTS

Unfortunately, and for reasons beyond our control, a decade elapsed between the original study and the final reporting. The original work was performed for the U.S. Army Ordnance Tank-Automotive Command under Contract DA30-069-ORD-1763. The final reporting is made possible through the funds provided by the Office of Naval Research, under contract NR-062-274/5-3-65-263(69). Because of the time elapsed, it is appropriate to remember those people other than the authors who were actively engaged in the actual work: Mr. Herman Nadler of U.S. ATACOM was the project monitor at OTAC and has had a continuing interest in amphibians. Of our working group, Mr. J.P. Finelli was project engineer, Mr. T.F. Helms produced the concept drawings, Mr. Dair Long was the responsible naval architect, and originated the open-wheel-well design, Mr. E. Hieber did most of the painstaking towing tank tests.

In the process of reviewing the old work, and in the task of updating and finalizing it, Dr. I. Robert Ehrlich provided his invaluable help. Our appreciation and thanks go to all those involved.

25. REFERENCES

1. "Reports on Hydrodynamic Model Tests of High Speed Amphibian Concepts," Davidson Laboratory Report 726-II, November 1966.

2. "Parameters for High Speed Amphibians, Project TCCD 60-59(NP), Final Report," U.S. Army Transportation Combat Development Group Fort Eustis, Va., December 1960.

3. "Transportation Equipment and Related Problems, Vol. I - Summary Technical Report of Division 12," NDRC, Washington, D. C., 1945.

4. "Final Report Army 2½ Ton Amphibian Truck 6x6 GMC DUKW-353 Inch," Sparkman & Stephens, 7 November 1944.

5. "Final Report of the Committee on Amphibious Operations," National Research Council," November 1952.

6. Bregnlski, R.R. and Swennes, B.A., "Research, Investigation and Experimentation in the Field of Amphibian Vehicles," Ingersol-Kalamazoo Division, Borg-Warner Corp., December 1957.

7. Roach, C.D., "Design of Wheeled Amphibians," SNAME, April 1960.

8. "5-Ton, 6x6, Amphibious," Report No. XM148-1, ACF-Brill Motors Co.

9. "Characteristic Sheet, Truck, Amphibious, 4-Ton, 6x6, XM147 E3, SUPERDUCK," Ordnance Tank-Automotive Command, 30 October 1958.

10. "Characteristic Sheet, Truck, Amphibious, 10-Ton, 8x8, XM157, DRAKE," Ordnance Tank-Automotive Command, 6 June 1959.

11. "Operators Manual, Lighter, Amphibious (LARC-V) Self-Propelled, Diesel, Aluminum 5-Ton, Design 8005," TM55-1930-205-10, Department of the Army, December 1962.

12. "Lighter, Amphibious, Resupply, Cargo (LARC XV); Self-Propelled Aluminum, 15-Ton," Military Products Division, Fruehauf Corporation, August 1965.

13. Eklund, K.F., "The Influence of Load and Inflation on the Selection of Pneumatic Tires for Military Vehicles," U.S. Army Engineers Board Report 922, 10 April 1945.

14. Rush, F.S., Temple, R.G., "Trafficability Tests in Fine-Grained Soils with Two Vehicles with 9- to 10-Ton Wheel Loads,' WES Misc. Paper No. 4-879, March 1967.

15. Nuttall, C.J., Jr., "Ground Crawling: 1966." USA Waterways Experiment Station Contract Report No. 3-162, May 1967.

16. Freitag, D.R., Knight, S.J., "Technique for Estimating the Slope Climbing Ability of Wheeled Vehicles in Sand," SAE, January 1963.

17. "A Vehicle Analysis for Remote Area Operation," USA Waterways Experiment Station," Misc. Paper No. 4-702, February 1965.

18. Comstock, J.P., Principles of Naval Architecture, SNAME 1967.

19. Nuttall, C.J., Jr., and Hecker, L.G., "Emphasizing Amphibious Vehicle Design, Supplementary Final Report on Contract No. OEMsr-154," Sparkman & Stephens, Inc., 30 March 1945.

20. McEwen, E., "Notes on the Military Amphibian," FVRDE, April 1947.

21. Witney, N.A., "Water Performance of Amphibious Craft," FVRDE TN142, February 1955.

22. Hecker, L.G., "Preliminary Summary Report on Amphibious Equipment and Vehicles Designed by Sparkman & Stephens, Inc. in cooperation with the National Defense Research Committee for the period May 1941 to August 1943," Sparkman & Stephens, 1944.

23. Wheeler, R.L., "Amphibious Hovercraft," SNAME, June 1968.

24. Todd, E.T., "The Wheeled Amphibious Vehicle," GMC Truck and Coach Division, GMC, 22 January 1958.

25. Kamm, I.O., "What Price Water Speed Improvements of Amphibians," Journal of Terramechanics, Vol. V, No. 1, 1968.

26. Votre, Louis, "Turbine-Driven Amphibians - The New Trend in Fast Assault Craft," SAE Paper 5-270, January 1961.

27. Sutherland, William H., et al, "Analysis of Means for Moving Logistic Cargo from Ship to Shore," Operations Research Office, Tech. Memo ORO-T-361, November 1957.

28. "Characteristics and Data of the Transportation Corps Marine Fleet and Terminal Service Equipment," U.S. Army Transportation School, May 1960.

29. Danahy, P.J., "Adequate Strength for Small High-Speed Vessels," Marine Technology, January 1968.

30. Murray, Allan B., "The Hydrodynamics of Planing Hulls," SNAME, February 1950.

31. Clement, E.P. and Blount, "Resistance Tests of a Schematic Series of Planing Hull Forms," SNAME 1963.

32. Savitsky, D., "Hydrodynamic Design of Planing Hulls," Marine Technology, October 1964.

33. Hadler, J.B. "The Prediction of Power Performance on Planing Craft," SNAME, November 1966.

34. Graul, T., Fry, E.D., "Design and Construction of Metal Planing Boats," SNAME, July 1967.

35. DeRight, S.R. and Szten, E.M., "Acceptance Test Report for Sky Larc Half-Scale Model," Ingersol-Kalamazoo Division, Borg-Warner Corp., December 1960.

36. Gasiunas, A., Lewis, W.P., "Hydraulic Jet Propulsion: A Theoretical and Experimental Investigation into the Propulsion of Seacraft by Water Jets," IME, October 1963, IME 1963-64 Proceedings, Vol. 178, Part 1, No. 7.

37. Brandau, John H., "Aspects of Performance Evaluation of Water Jet Propulsion Systems and a Critical Review of the State-of-the-Art," SNAME, May 1967.

38. Contractor, D.N. and Johnson, Virgil E., Jr., "Water Jet Propulsion," SNAME, May 1967.

39. Delao, Martin M., "Practical Considerations of Water Jet Propulsion," SAE, August 1965.

40. Venning, E., Jr., and Haberman, W.L., "Supercavitating Propeller Performance, SNAME 1962.

41. Saunders, H.E., Hydrodynamics in Ship Design Vol. II, page 807, SNAME 1957.

42. Kim, H.C., "Hydrodynamic Aspects of Internal Pump-Jet Propulsion," Marine Technology, 3:1, January 1966.

43. Harshfield L.G., "The GOER Concept," SAE, January 1965.

44. Dugoff, H., et al, "Coupled Mobility Devices," Davidson Laboratory Report 1164, December 1964.

45. Goodyear Tire and Rubber Co., "Terra-Tire Engineering Data," July 1965.

46. Hoess, T.A., et al, "The Design and Develooment of Laboratory Models to Study the Feasibility of High-Flotation Tires for Aircraft," Battelle Memorial Institute (for Fairchild Aircraft Division, Fairchild Engine Airplane Corporation) August 1958.

47. Sidles, J. and Cardenas, A. "A New Spare Tire," SAE, May 1967.

48. Walter Motor Truck Co., Inc., "Principles of Construction of Walter Four-Point Positive Drive," Bulletin T-22, 1 November 1952.

49. Jacobs, H.A., "They Stop at Nothing," Ordnance, November-December 1954.

50. Noake, D.A., "A New Concept in Land-Vehicle Propulsion," Armor, May-June 1967, ATAC/AVCO.

51. McCay, A.W., "Gas Turbine Propulsion for High Speed Small Craft," Marine Technology, July 1968.

52. Krondgard, S.O., "The Volvo Dual Power Plant for Military Vehicles," SAE, January 1966.

53. "Industrial Gas Turbine Engine, Model Specification," Boeing Report 551, The Boeing Co., Turbine Division, 24 April 1964.

54. "Allison Fully Automatic Application Data, Model HT-70," Allison Division, General Motors Corporation, 1964.

55. Spaulding, K.B., Jr. and Silvia, P.A., "Design and Construction of Fiberglass Boats from 60 to 120 Feet in Length, An International Survey," The Society of Plastics, Industry, Inc., 22nd Annual Meeting, 1966.

56. "Characteristics Sheet - LVHX2, Landing Force Amphibious Support Vehicle, Hydrofoil," Bureau of Ships Code 529V, 24 March 1964.

57. "Characteristics Sheet - LVWX1, Landing Vehicle Wheeled," Bureau of Ships, Code 529V, 28 August 1962.

58. Van Dyck, R.L. and Ehrlich, I.R., "Drag Studies of Coupled Amphibians," Davidson Laboratory Report 1137, July 1966.

59. Ford, C.J. and Nilsson, L.U.G., "Goodyear Rolligon Tires - A New Approach to Vehicular Mobility." SAE Paper No. 658, January 1956.

60. Digges, K.H. and Petersons, A.V., "Results of Studies to Improve Ground Flotation of Aircraft." Proc., 1967 SAE Aerospace Systems Conference.

61. Stoddard, R.L. and Harrison, R.H., "Installation Status of Gas Turbine Engines," SAE No. S217, October 1959.

62. Dalton, C.A., "Sulfur and Sea Salt Attack of Turbine Blades," Marine Technology, 2:3, July 1965.

63. Nolte, G.H., "Sea Experience with Pratt and Whitney Aircraft FT12 Gas Turbine in LCM-8," Marine Technology, 2:4, October 1965.

64. Moore, G.W., "Some New Developments in Landing Craft for the Navy," SNAME So. Calif. Meeting, 12 April 1958.

65. Cheng, H.M., et al, "Analysis of Right-Angle Drive Propulsion System," SNAME Paper No. 20, June 1968.

66. Brockett, W.A., et al, "U.S. Navy's Marine Gas Turbines," Naval Engineer's Journal, April 1966.

67. Artinian, L. and Terry, S.L., "The Total Cost of Weight," SAE, March 1961.

68. Stevens, R.T., et al, "Breakthrough in Body Structure Utilizing Glass Fibers, Polymers, and Plastics," SAE No. 274A, January 1961.

69. Holtyn, C.H., "Aluminum - The Age of Ships," SNAME Paper No. 9, November 1966.

70. Carl, D., "The Landing Vehicle Hydrofoil LVHX2," SAE No. S402, October 1963.

71. Hom, K., et al, "Glass Reinforced Plastics for Submersible Pressure Hulls," SAE Paper No. 690C, April 1963.

72. Spaulding, K.B., Jr. and Della Rocca, R.J., "Fiberglass-Reinforced Plastic Chine Sweepers," SNAME, Paper No. 8, November 1965.

73. Buermann, T.M. and Della Rocca, R.J., "Fiberglass Reinforced Plastics for Marine Structures," SNAME Paper No. 3, May 1960.

74. Bischoff, T.J., "Design of the New Experimental Truck, Cargo, $2\frac{1}{2}$-Ton, XM-521," SAE No. 273C, January 1961.

75. Pearson, S.J. and Bischoff, T.J., "Design of the Bonded Body-Frame Structure for the XM-521 Experimental Military Truck," SAE Paper No. 273B, January 1961.

76. Gyres, S.N., "Test Report for Half-Scale Model of the High-Speed Inverted 'V' Hull, Amphibious Vehicle, Wheeled (LVW-X3)," Hydronautics Report TR-742-1, January 1968.

77. Clement, E.P., "Discussion of the Prediction of Power Performance of Planing Craft, by J.B. Hadler," SNAME, 1966.

78. McGown, S.C., "The Seaworthiness Problem in High-Speed Small Craft," SNAME, New York Metropolitan Section, 24 June 1961.

79. Savitsky, D., "On the Seakeeping of Planing Hulls," Marine Technology, April 1968.

80. Miller, E.R. and Lindenmuth, W.T., "Optimum Characteristics for High Speed Amphibians," Hydronautics Report T.R. 742-2, March 1968.

81. Miller, E.R., et al, "Wheeled Amphibian Engineering Design Handbook," Hydronautics, To be published.

82. Miller, E.R., et al, "A Parametric Study of High Speed Support Amphibians," Hydronautics Report T.R. 615-1, February 1968.

83. Condert, T.R. and Finelli, J.P., "Towing Tests of a Proposed Lighter," 5-Ton Amphibious," Davidson Laboratory Report LR-701, June 1958.

84. Kamm, I.O. and Schwartz R.B. "Research on a Light Truck Highway Train," Davidson Laboratory Report No. 995, March 1964.

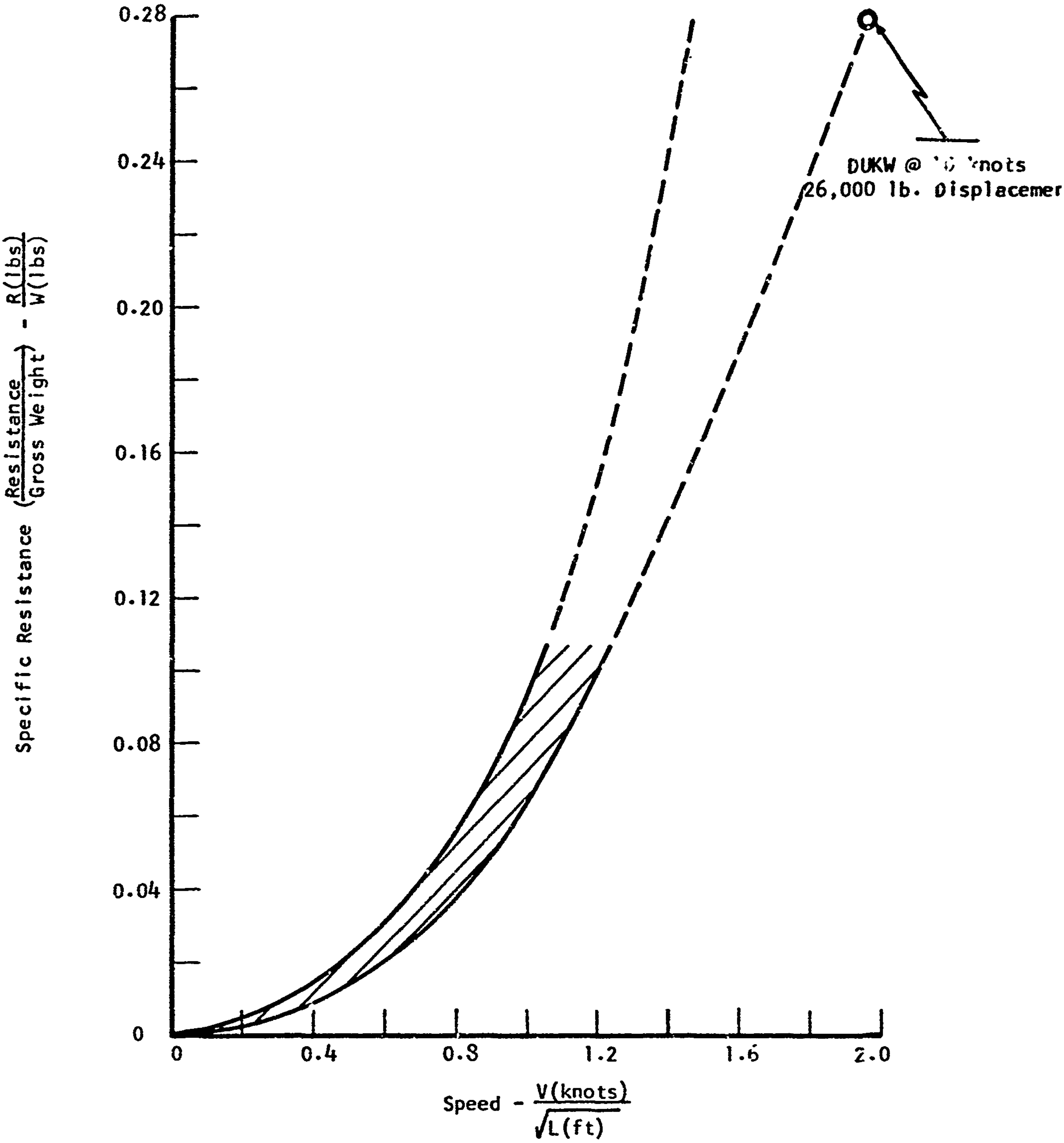


FIG. 1. SPECIFIC RESISTANCE FOR A RANGE OF WHEELED AMPHIBIANS[19]

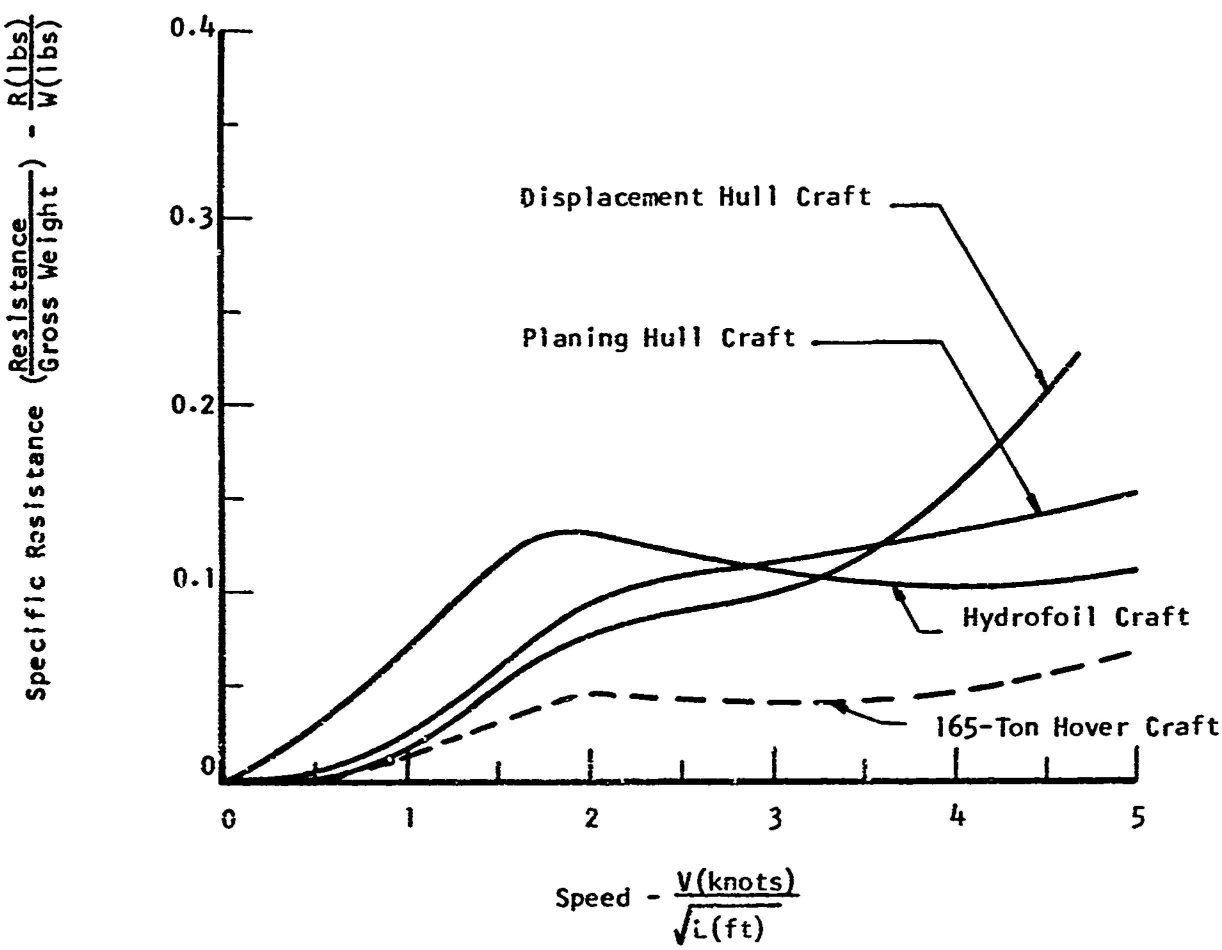


FIG. 2. SPECIFIC RESISTANCE OF WELL DESIGNED BOATS[18,23]

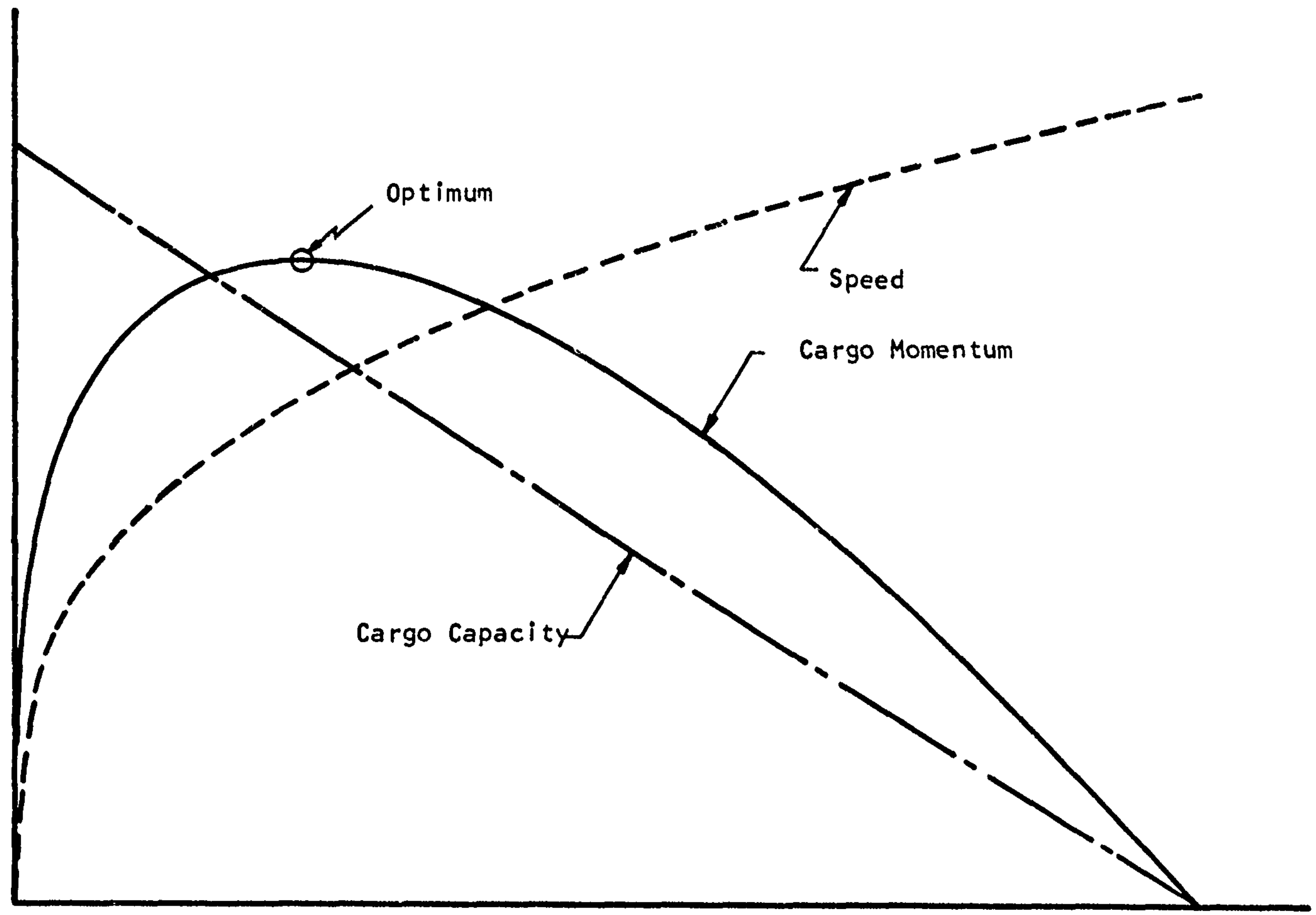


FIG. 3. GENERALIZED RELATIONSHIP BETWEEN SPEED, CARGO CAPACITY AND CARGO MOMENTUM[24]

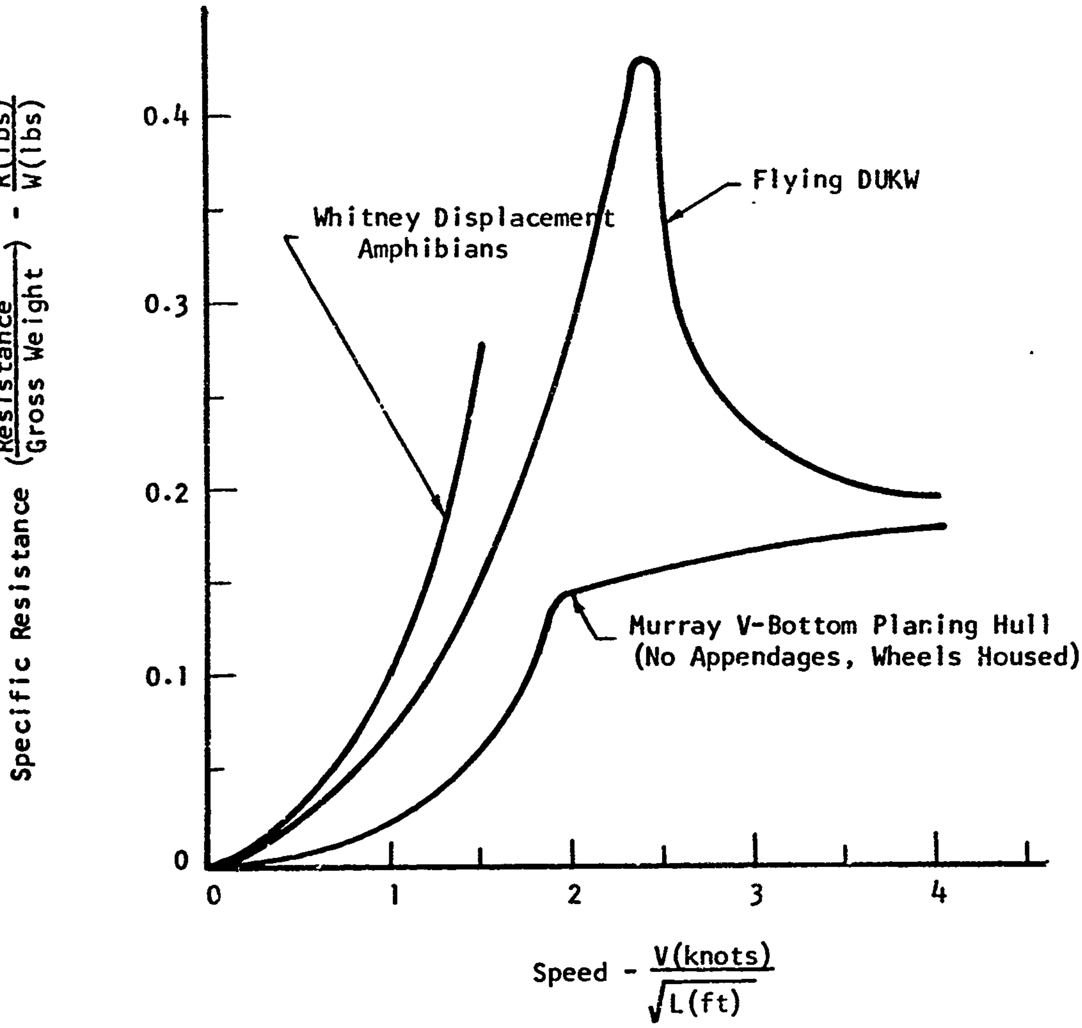


FIG. 4. SPECIFIC RESISTANCE OF VARIOUS TYPES OF AMPHIBIOUS TRUCKS[1,19-25]

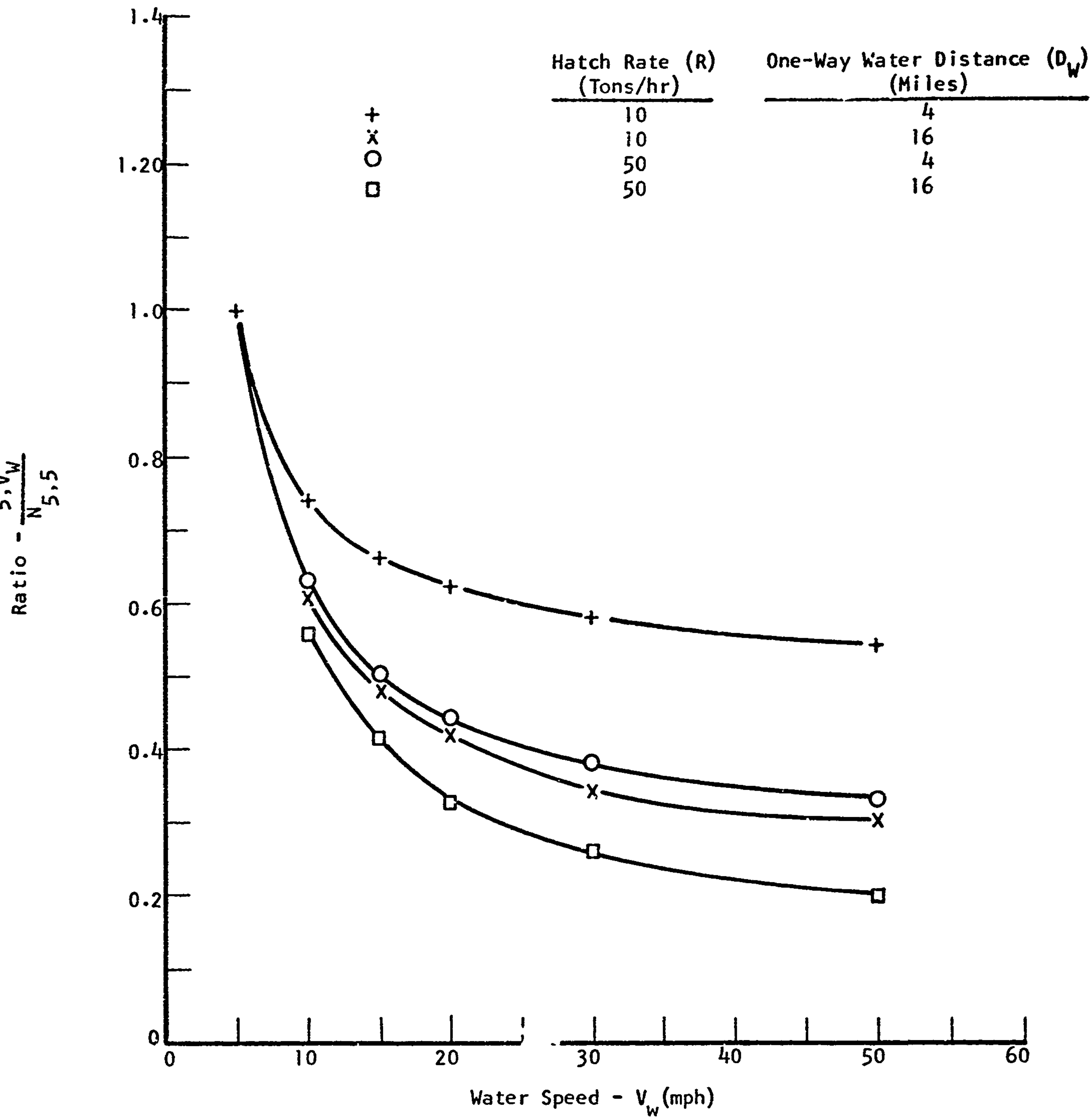


FIG. 5. REDUCTION IN THE REQUIRED NUMBER OF 5-TON VEHICLES ACHIEVED BY INCREASING WATER SPEED WHEN COMPARED TO A STANDARD OF 5 MPH

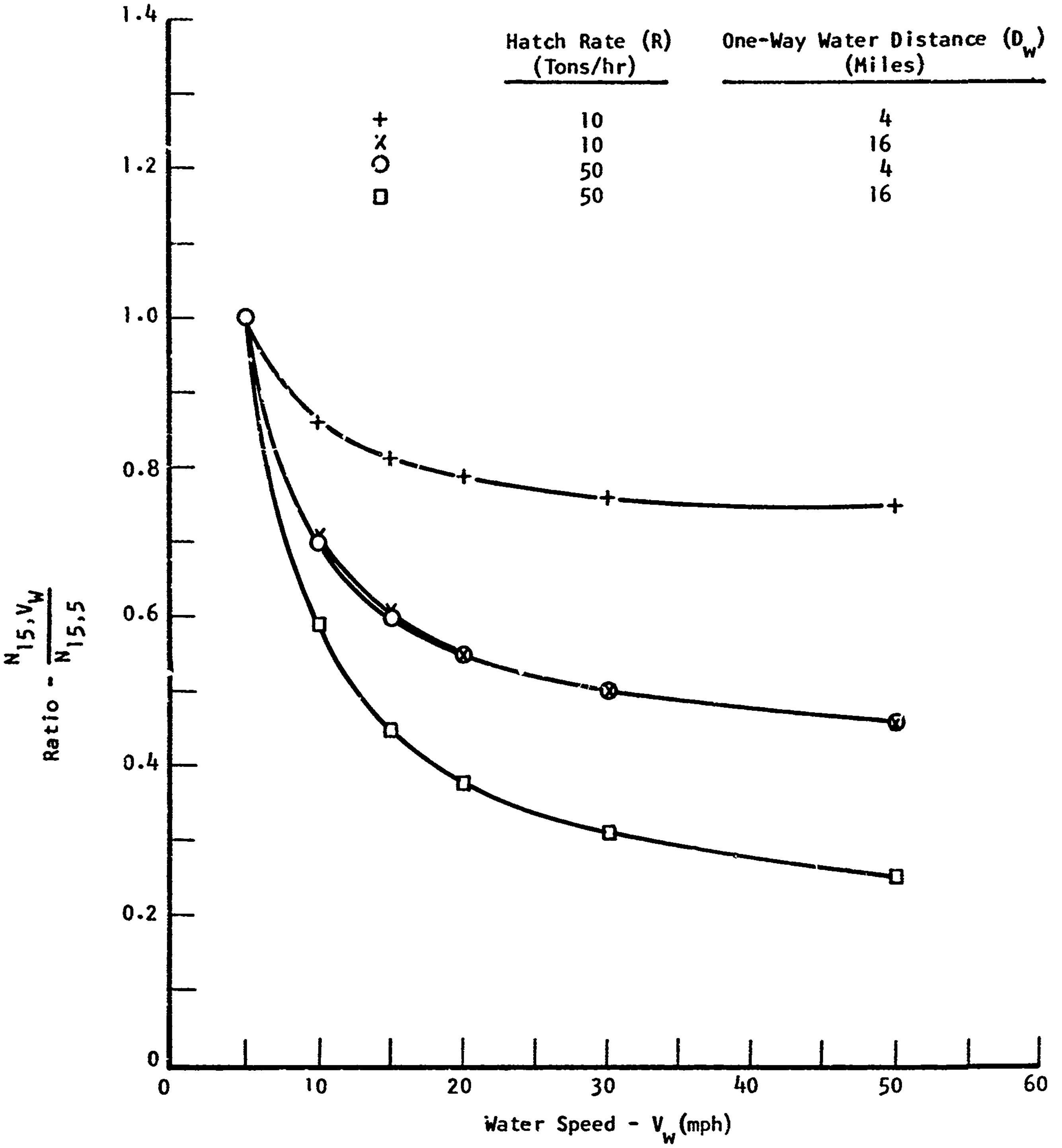
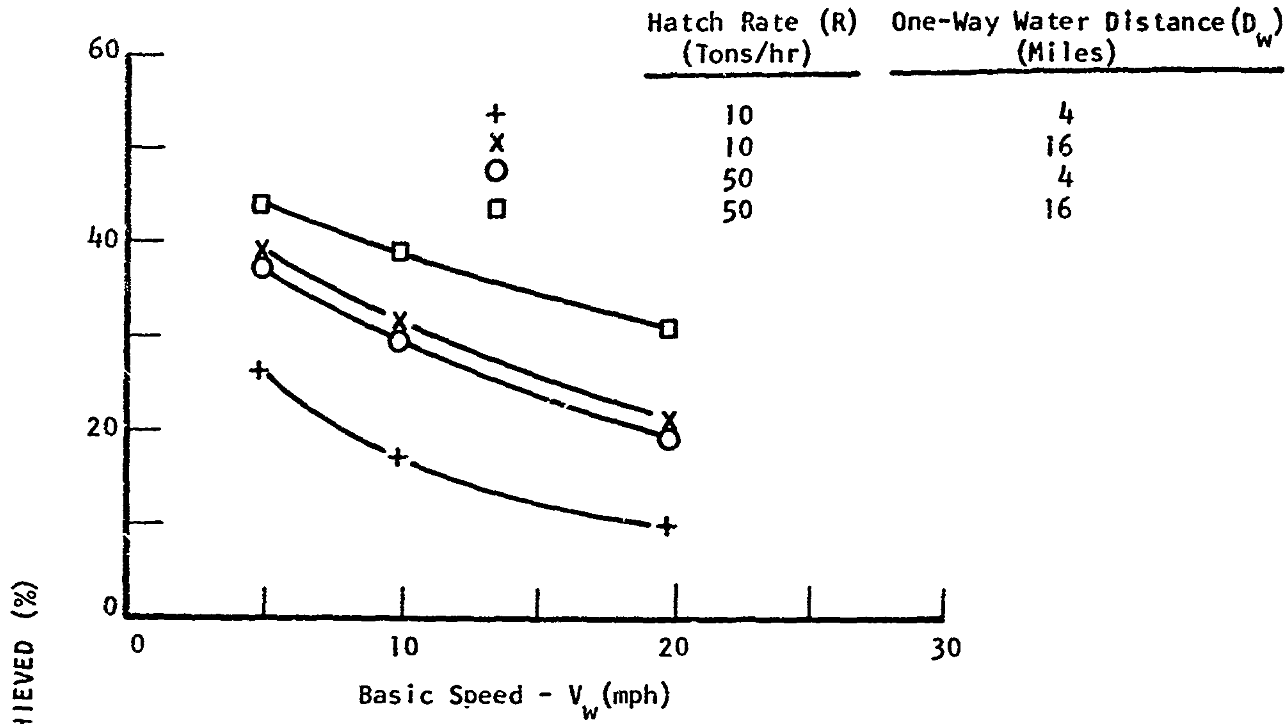
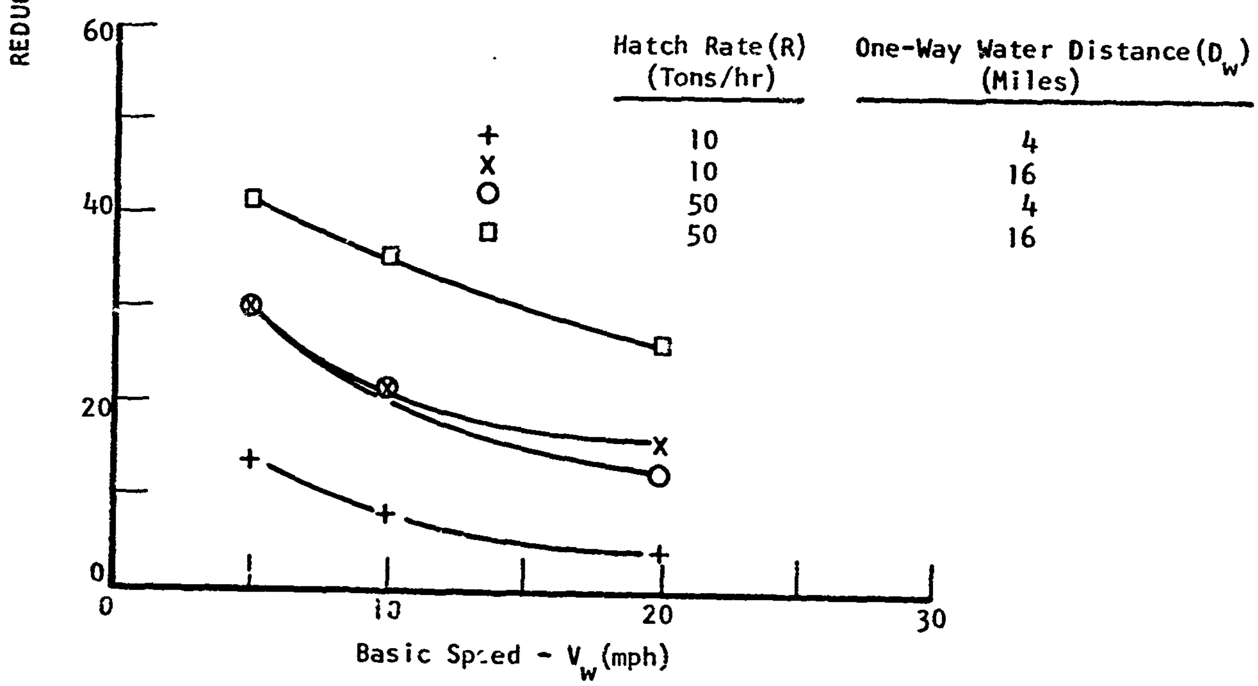


FIG. 6. REDUCTION IN THE REQUIRED NUMBER OF 15-TON VEHICLES ACHIEVED BY INCREASING WATER SPEED WHEN COMPARED TO A STANDARD OF 5 MPH



a. 5-Ton Vehicle



b. 15-Ton Vehicle

FIG. 7. REDUCTION IN THE REQUIRED NUMBER OF VEHICLES TO BE ACHIEVED BY DOUBLING A GIVEN SPEED, 5 and 15-TON VEHICLES

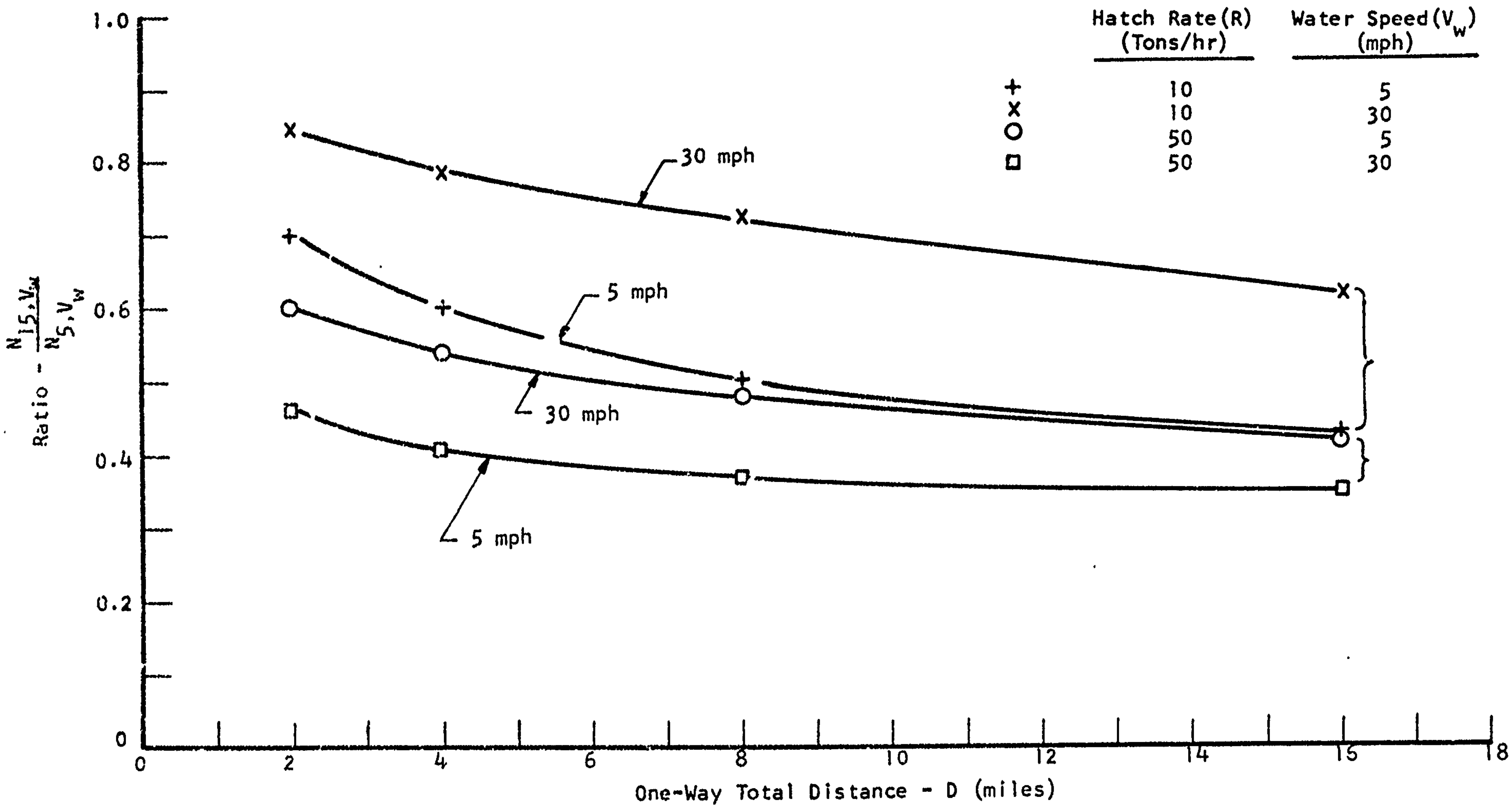


FIG. 8. RATIO OF REQUIRED 15-TON TO 5-TON AMPHIBIOUS VEHICLES AS A FUNCTION OF ONE-WAY TOTAL DISTANCE

INDEPENDENT / DEPENDENT	WATER FEATURES							LAND FEATURES					CARGO FEATURES	
	GROSS VEHICLE WEIGHT	HULL		MARINE POWER SYSTEM	MARINE PROPULSION SYSTEM	WHEEL DOORS		WHEEL RETRACTION METHOD	RUNNING GEAR		LAND POWER SYSTEM	LAND STEERING SYSTEM	AREA	WET DECK
		SIZE	FORM			YES – NO	DESIGN		SIZE	CONFIGURATION				
LOAD CAPABILITY	1	1	2	2					2	2	2		1	2
WATER PERFORMANCE	1	1	1	1	1	1			2	2				1
SURF & BEACHING PERFORMANCE	1	2	2		1		2		1	1		2		1
OFF-ROAD PERFORMANCE	1	2	2				2	2	1	1	2	2		2
ON–ROAD PERFORMANCE	2	1						2	2	2	2	2		
TRANSPORTABILITY	1	1							2					

1 PRIMARY INFLUENCE

2 SECONDARY INFLUENCE

FIG. 9. INTERACTION OF DESIGN FEATURES AND PERFORMANCE

DEPENDENT \ INDEPENDENT	GROSS VEHICLE WEIGHT	HULL SIZE	HULL FORM	RUNNING GEAR SIZE	RUNNING GEAR CONFIGURATION	POWER SYSTEM, MARINE	POWER SYSTEM, LAND	WATER PROPULSION CONFIGURATION	LAND STEERING CONFIGURATION	WHEEL RETRACTION CONFIGURATION	WHEEL DOOR CONFIGURATION
GROSS VEHICLE WEIGHT		1		1	2	2	2	2	2		2
HULL SIZE	1			1	2	2		2	2		
HULL FORM				1	1	2		2	2		1
RUNNING GEAR SIZE	1	1	2		1			2	1	2	1
RUNNING GEAR CONFIGURATION	2	2	1	1			2		2	2	1
POWER SYSTEM, MARINE	1	1	1	2				1			2
POWER SYSTEM, LAND	1			2	2				2	2	
WATER PROPULSION CONFIGURATION	2	2	2	2	2	1					2
LAND STEERING CONFIGURATION			2	1	1		2			2	2
WHEEL RETRACTION CONFIGURATION	2	2	2	1	1		2		1		2
WHEEL DOOR CONFIGURATION	2		2	1	1	2			1	2	

1 PRIMARY INFLUENCE

2 SECONDARY INFLUENCE

FIG. 10. INTERACTION OF VARIOUS MACHINERY FEATURES

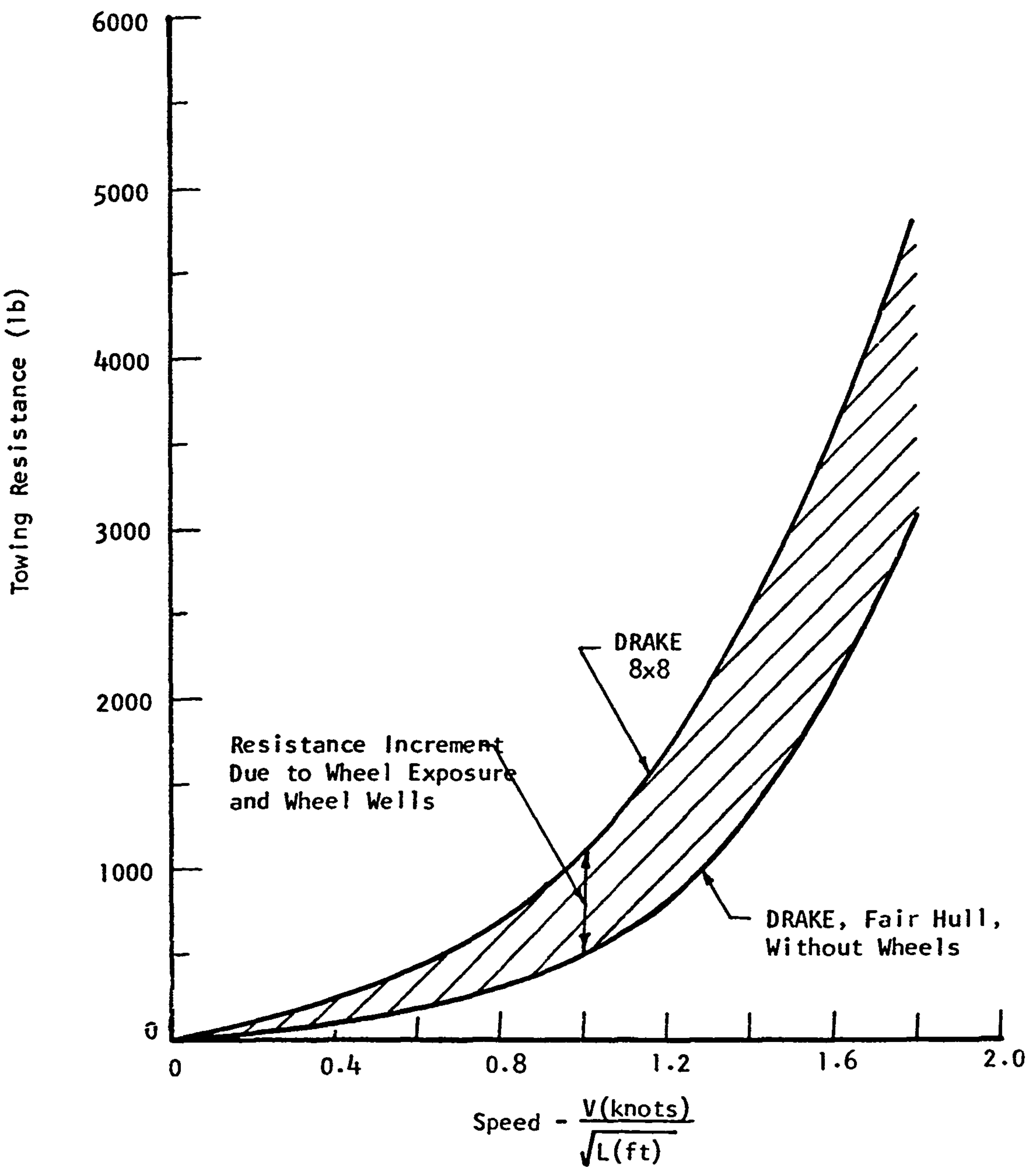


FIG. 11. EFFECT OF EXPOSED WHEELS

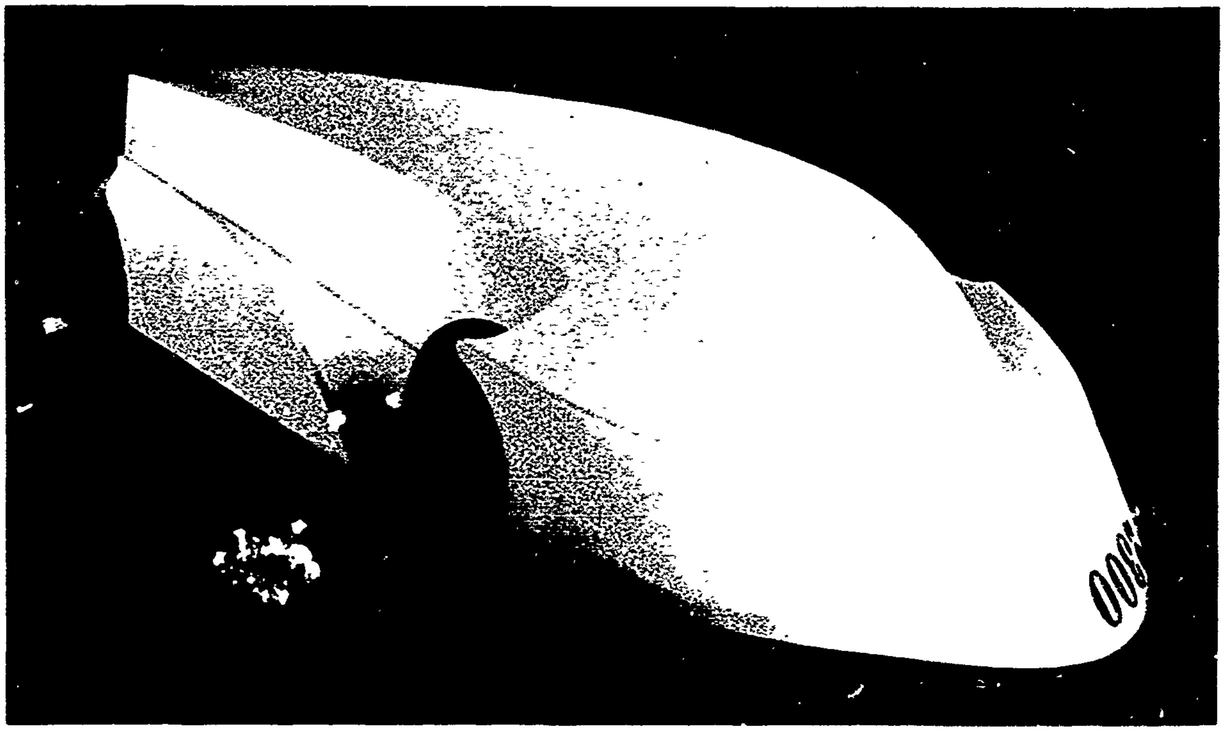


FIG. 12. THE "DAIR LONG" OPEN HULL CONCEPT, SHOWING HOW WHEELS ARE RETRACTED IN A RECESS IN SUCH A WAY THAT THE WATER FLOW SEPARATES CLEANLY AND IS THEN RECAPTURED SMOOTHLY

HULL DESIGN VERSUS PERFORMANCE + = BETTER WITH INCREASING VALUE − = WORSE WITH INCREASING VALUE	WATER PERFORMANCE					LAND PERFORMANCE										LOAD PERFORMANCE	TRANSPORTABILITY
						SOFT SOIL	OBSTACLES						SLOPE				
	RESISTANCE & SPEED	HUMP RESISTANCE	SEA KINDLINESS	STATIC STABILITY	SURF ABILITY AND BEACHING	MOBILITY	GROUND CL.	APPROACH ANGLE	BREAK ANGLE	DEPARTURE ANGLE	RIDE	ROADABILITY	SIDE	LONGITUDINAL	AGILITY		
LENGTH	+	+	+	+	+			−	−	−	+	−		+	−	+	−
BEAM (OVERALL)	+	+	·	+	+						+	−	+		−	+	−
BASIC DEADRISE ANGLE	−	+	+		+		−						−	−		−	
GROSS VEHICLE WEIGHT	−	−	+	−	+	−					+	−			−	+	−
LONGITUDINAL C.G. FORWARD OF STERN	−	−						+		−							
BOW, SHAPE, FULLNESS		−	−	+	+			−							−		
STERN, SHAPE, FULLNESS	−			+	+					−					−	+	
FREE BOARD				+												−	−
VERTICAL C.G. HEIGHT				−									−			−	

FIG. 13. INTERACTION OF HULL DESIGN AND PERFORMANCE

DEPENDENT \ INDEPENDENT	LENGTH	BEAM	APPROACH & DEPARTURE ANGLE	GROSS VEHICLE WEIGHT *	L C G	BOW SHAPE	STERN SHAPE	FREE BOARD	V C G	RUNNING GEAR
LENGTH		1		1		2	2			
BEAM	1			1		2	2		1	1
APPROACH & DEPARTURE ANGLE						2	2			2
GROSS VEHICLE WEIGHT *	1	1	2			2	2	1	2	2
L C G	2									1
BOW SHAPE			2		2		2	2		2
STERN SHAPE					2			2		2
FREE BOARD	2	2	2	2					2	
VERTICAL CG HEIGHT	2	2	2	2		2	2	2		2
RUNNING GEAR	1	1	2	1	1	2	2			

* AS AFFECTED BY HULL DESIGN ONLY

1 PRIMARY INFLUENCE

2 SECONDARY INFLUENCE

FIG. 14. INTERACTION OF VARIOUS HULL FEATURES

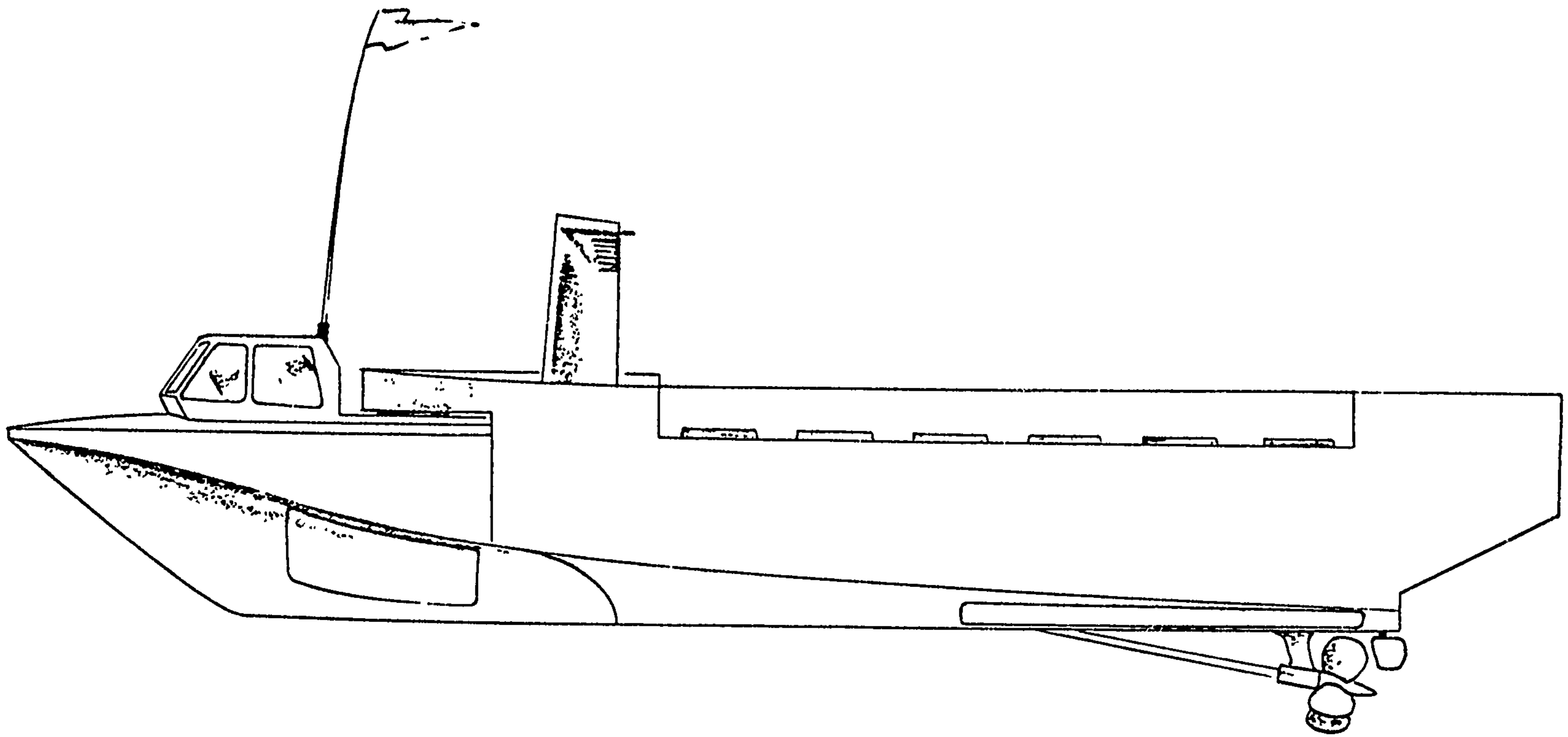


FIG. 15. CONCEPT NO. 1 - A 5-TON, 6x6, LOW-CHINE, V-BOTTOM PLANING AMPHIBIAN WITH RETRACTABLE PROPELLER AND TUNNEL, RETRACTABLE WHEELS, ROLLER DRIVE, "GOER"-TYPE ARTICULATED STEERING

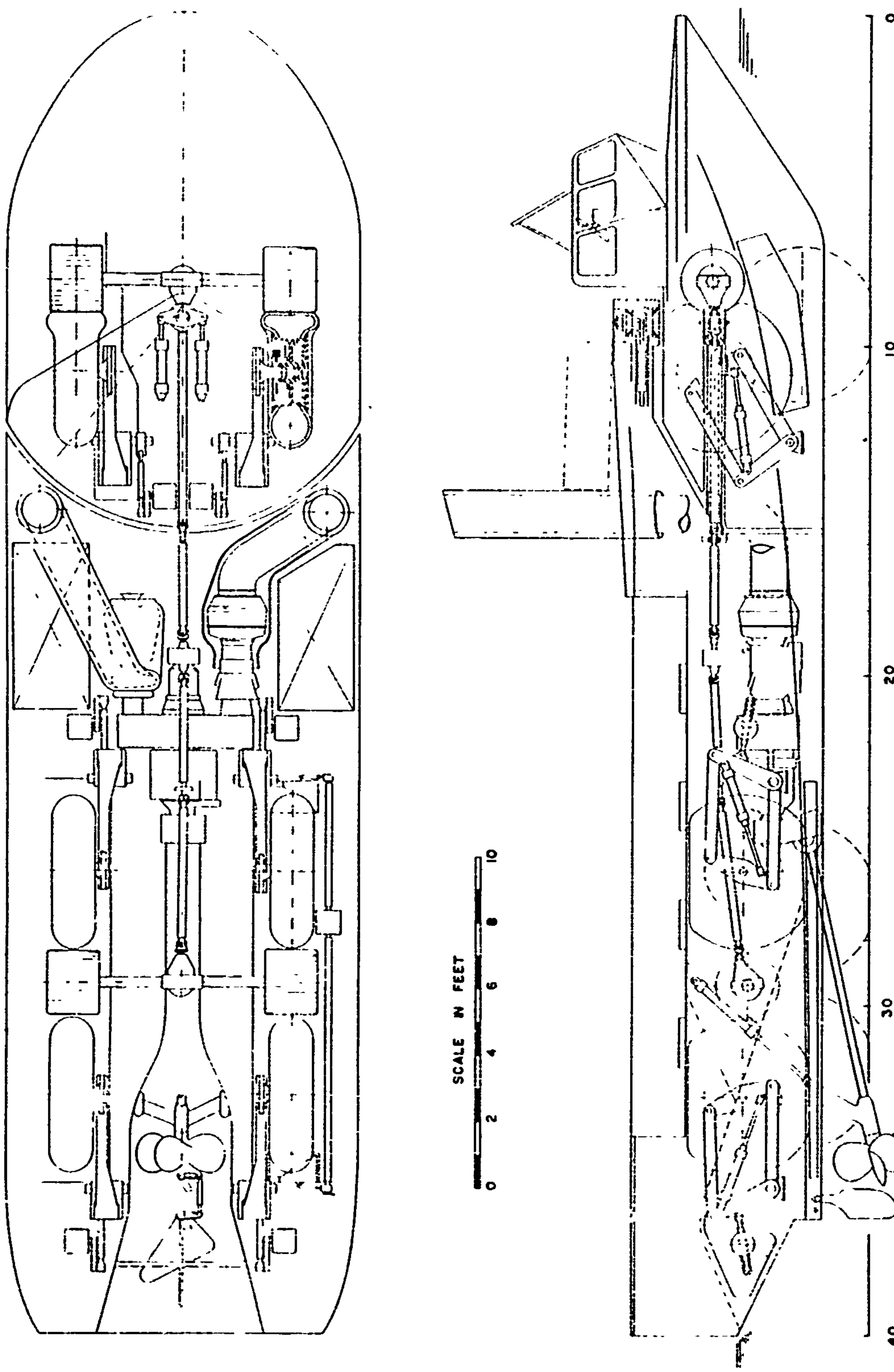


FIG. 16a. CONCEPT NO. 1 - LAYOUT DRAWINGS: STARBOARD ELEVATION AND PLAN VIEWS

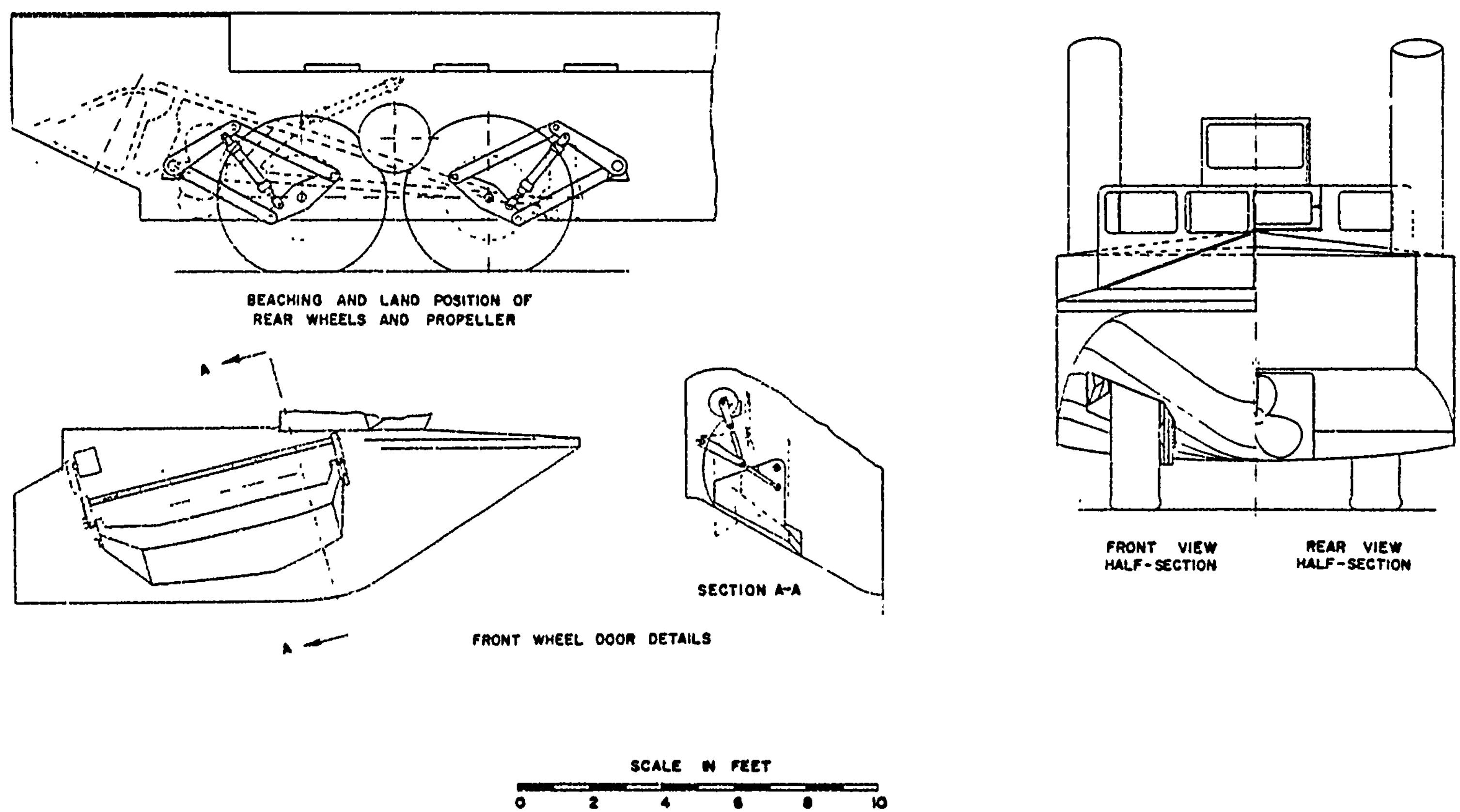


FIG. 16b. CONCEPT NO. 1 - LAYOUT DRAWINGS: FRONT AND REAR VIEWS AND WHEEL DETAILS

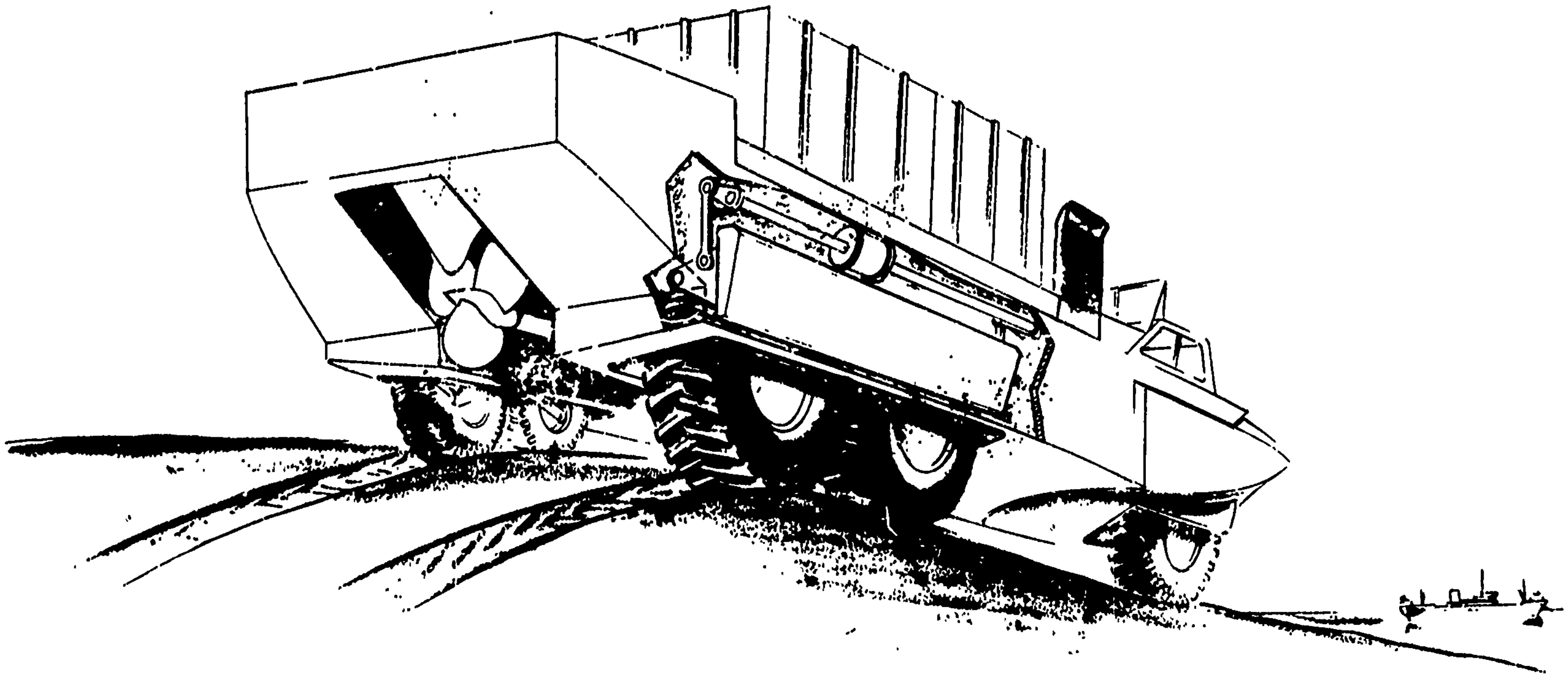


FIG. 17. CONCEPT NO. 1 - WHEEL POCKET COVERS IN STOWED POSITION, PROPELLER RETRACTED

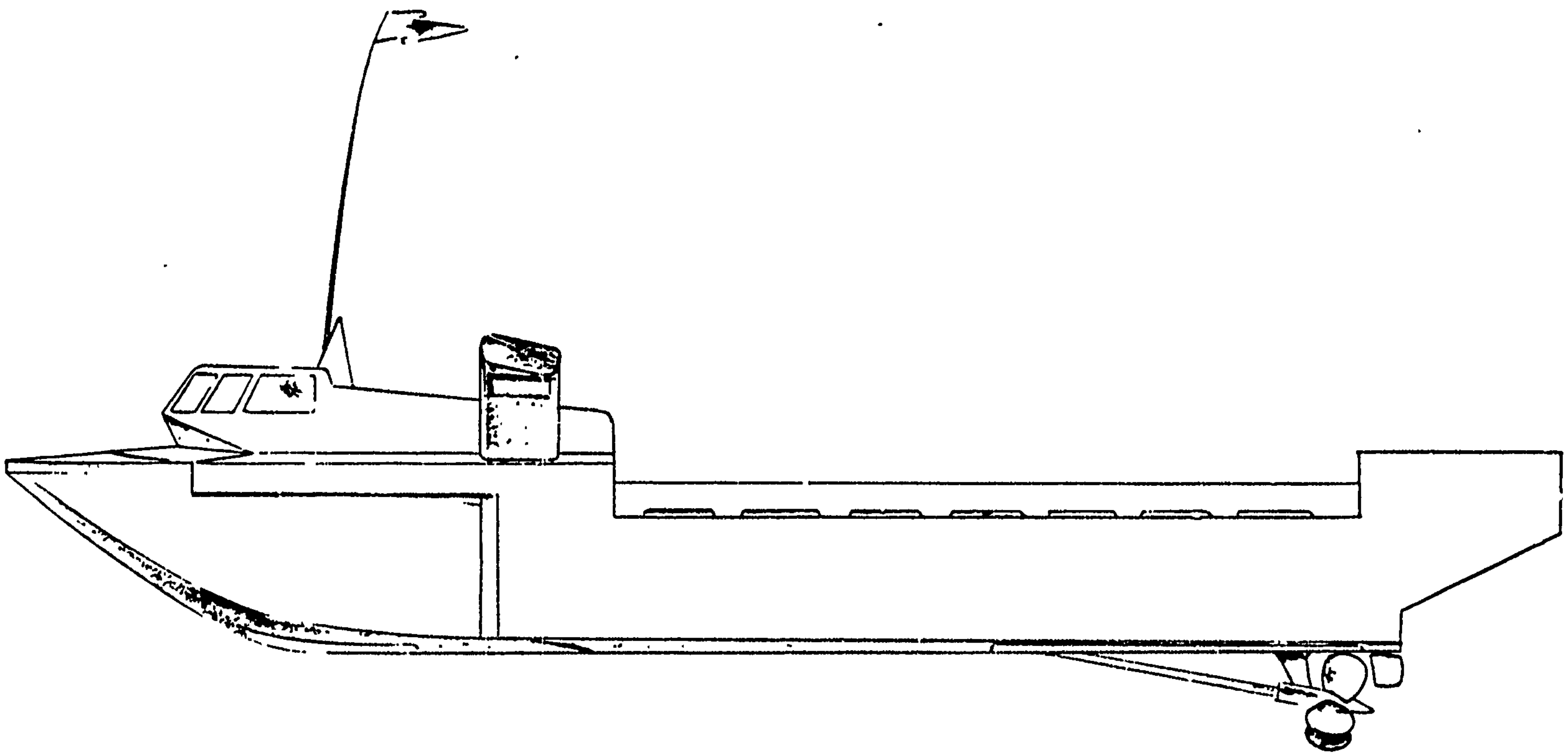


FIG. 18. CONCEPT NO. 2 - A 5-TON, 6x6, W-BOTTOM PLANING AMPHIBIAN WITH RETRACTABLE PROPELLER AND TUNNEL, RETRACTABLE WHEELS, CHAIN DRIVE, "GOER"-TYPE ARTICULATED STEERING

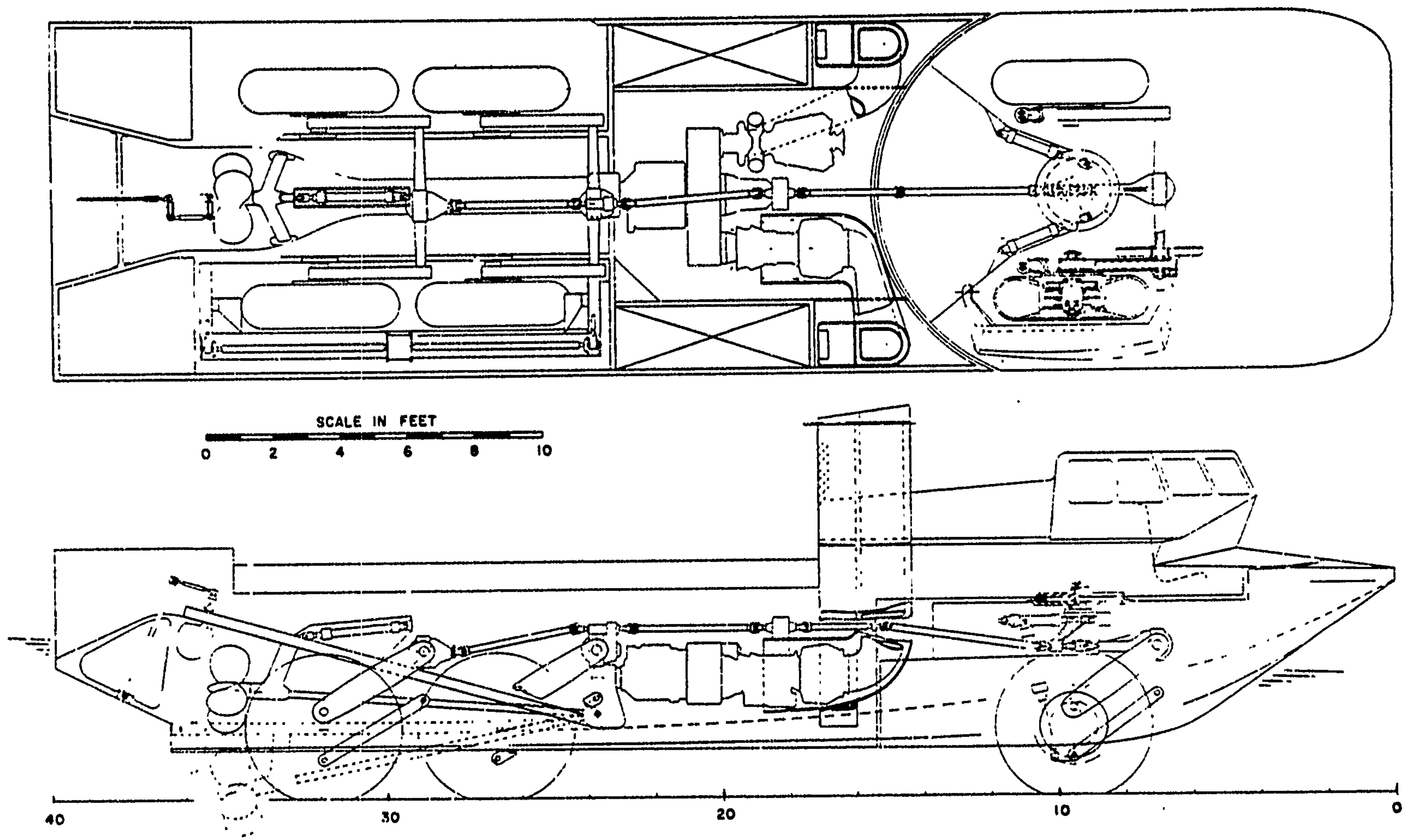


FIG. 19a. CONCEPT NO. 2 - LAYOUT DRAWINGS: STARBOARD ELEVATION AND PLAN VIEWS

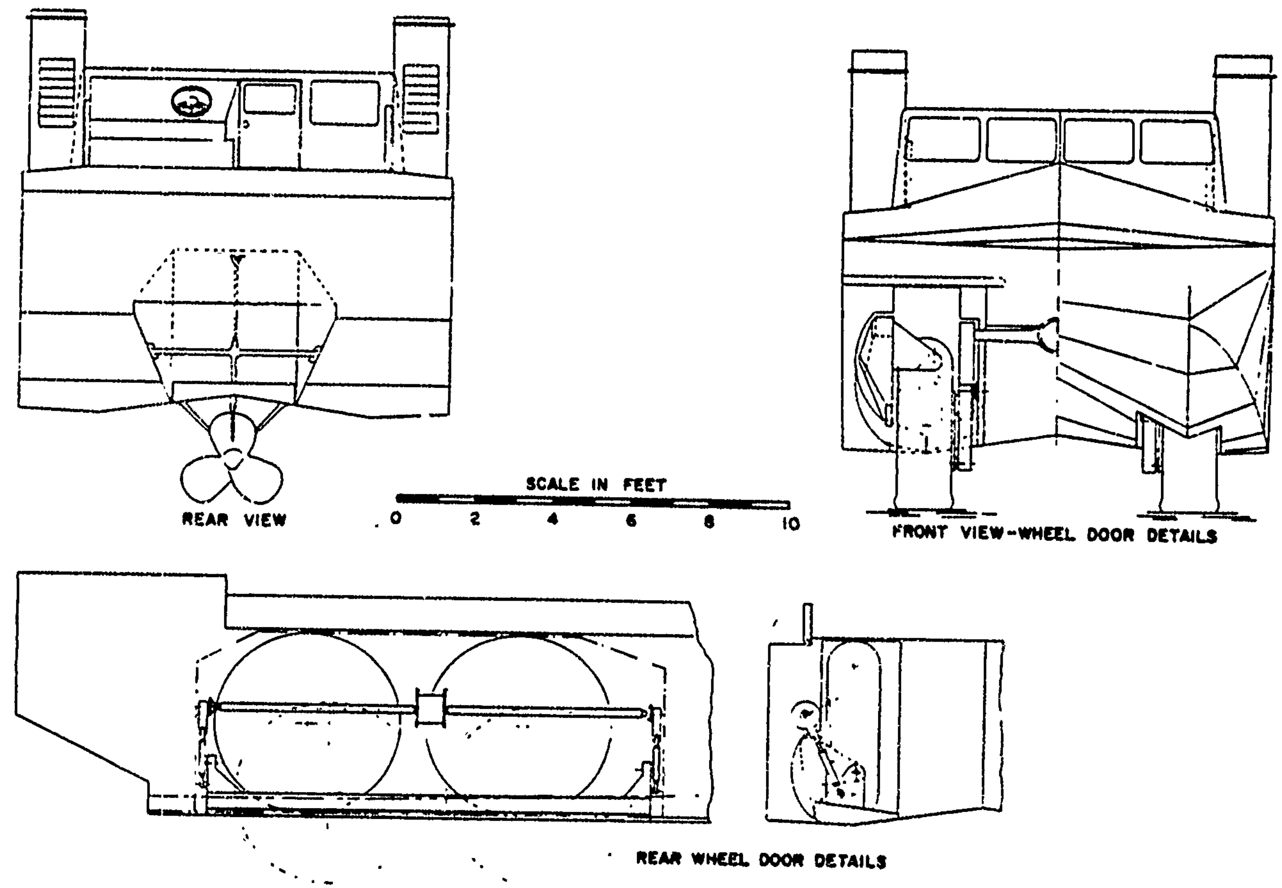


FIG. 19b. CONCEPT NO. 2 - LAYOUT DRAWINGS: FRONT AND REAR VIEWS AND WHEEL DETAILS

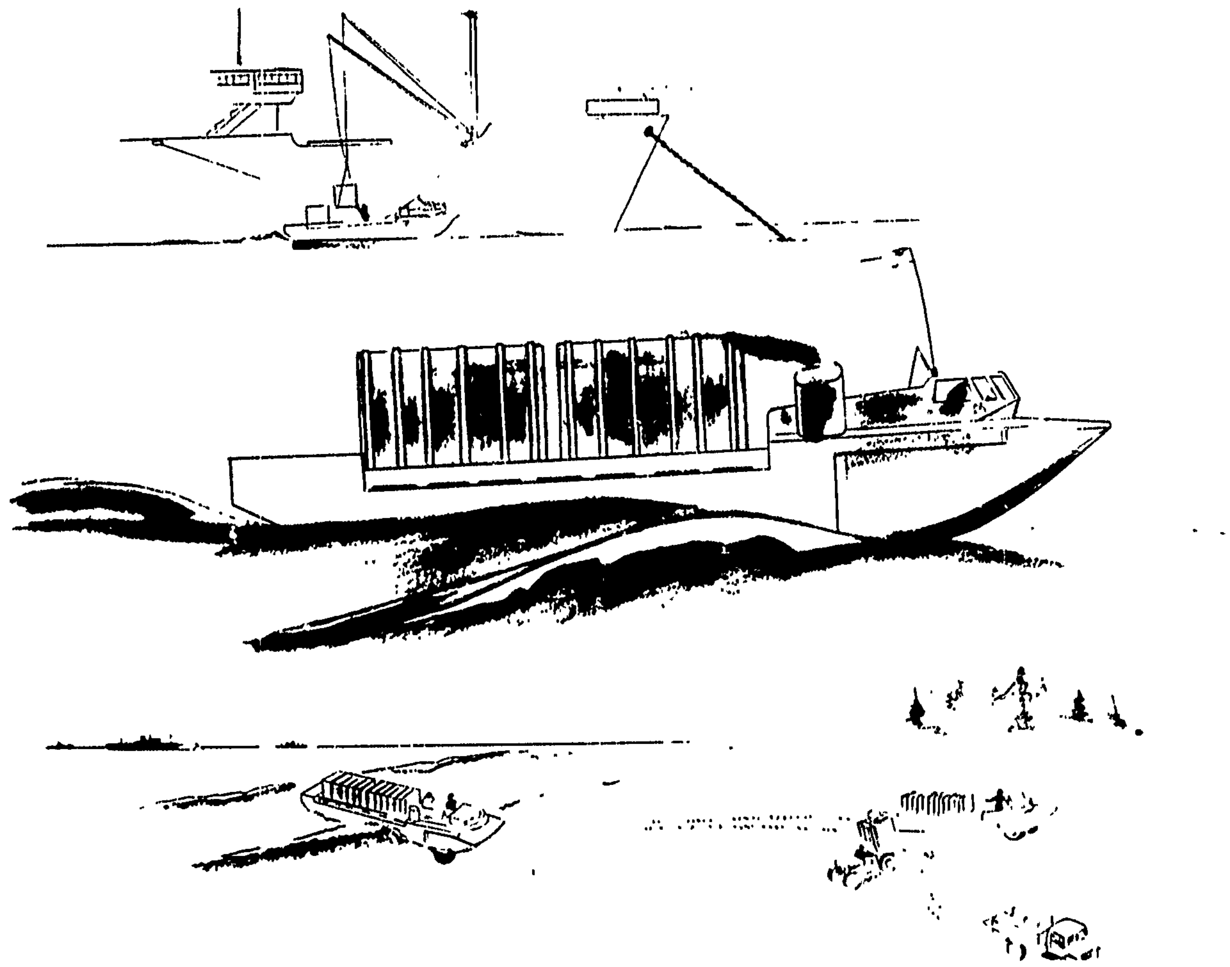
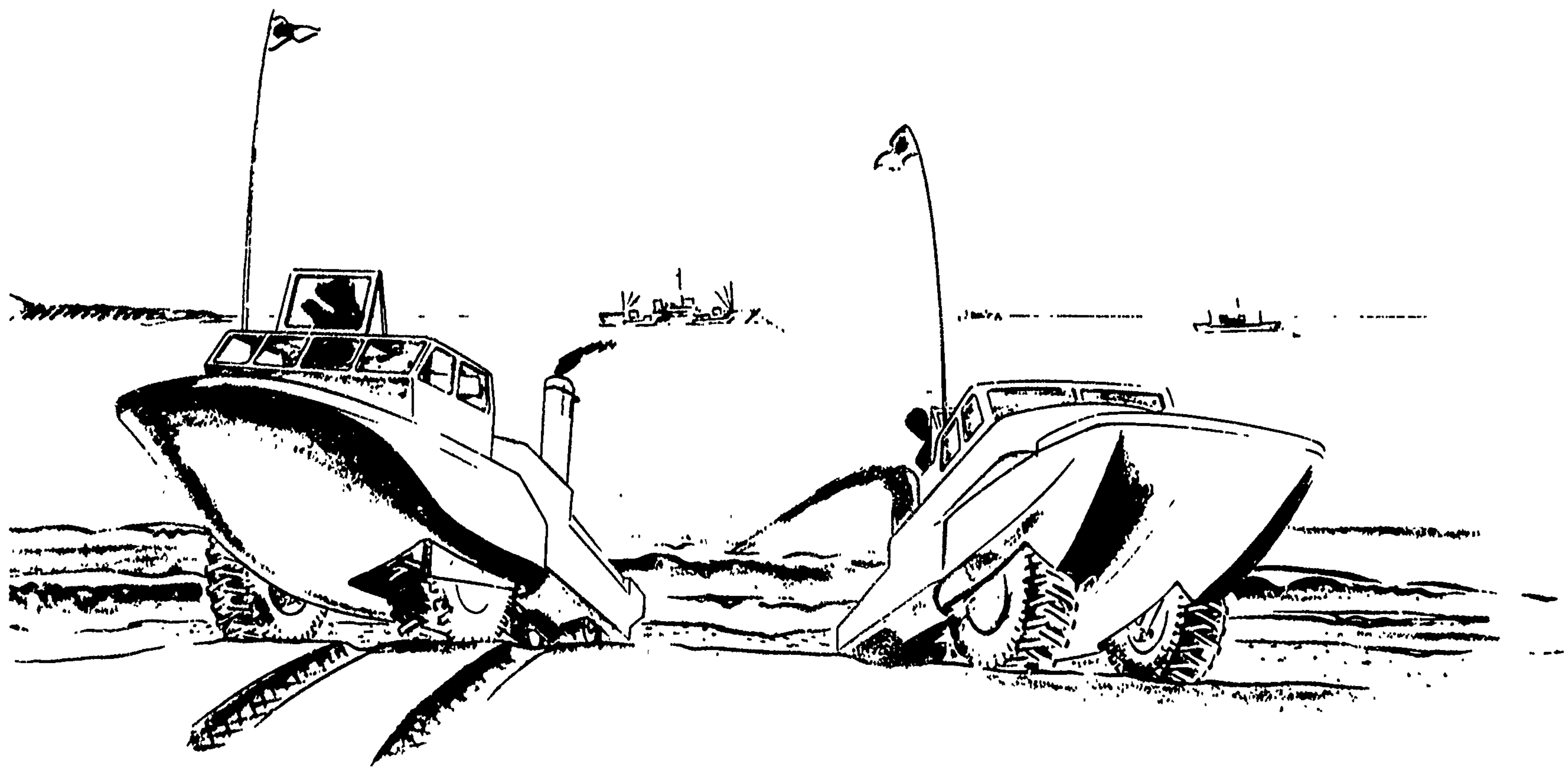


FIG. 20. CONCEPT NO. 2: GENERAL VIEWS



a. Concept No. 1
Low-Chine, V-Bottom Planing Hull

b. Concept No. 2
W-Bottom Planing Hull

FIG. 21. CONCEPTS 1 and 2

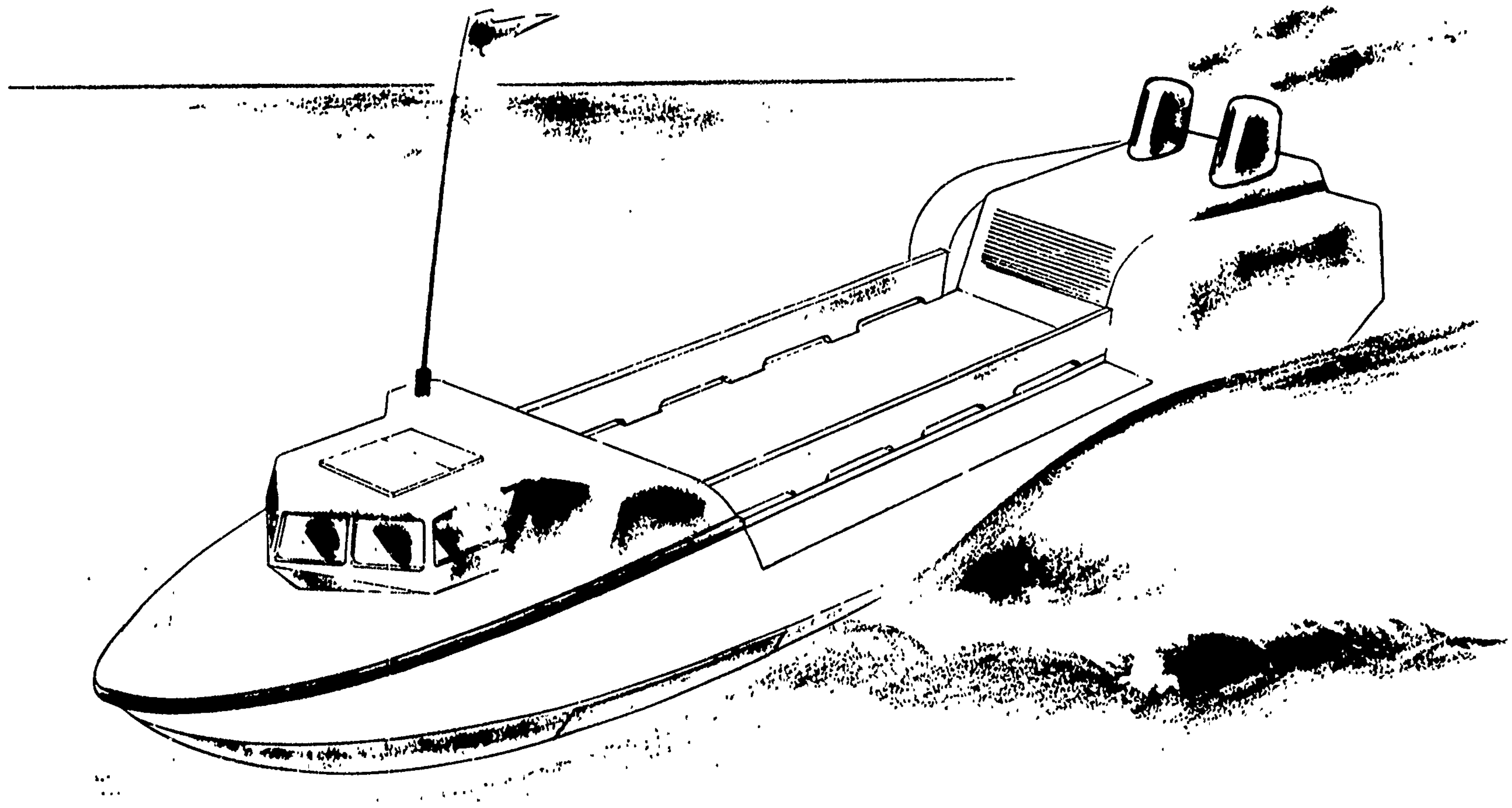


FIG. 22. CONCEPT NO. 3 - A 5-TON, 4x4, HIGH-CHINE, V-BOTTOM PLANING AMPHIBIAN WITH RETRACTABLE PROPELLER AND TUNNEL, RETRACTABLE WHEELS, CHAIN DRIVE, ACKERMANN STEERING

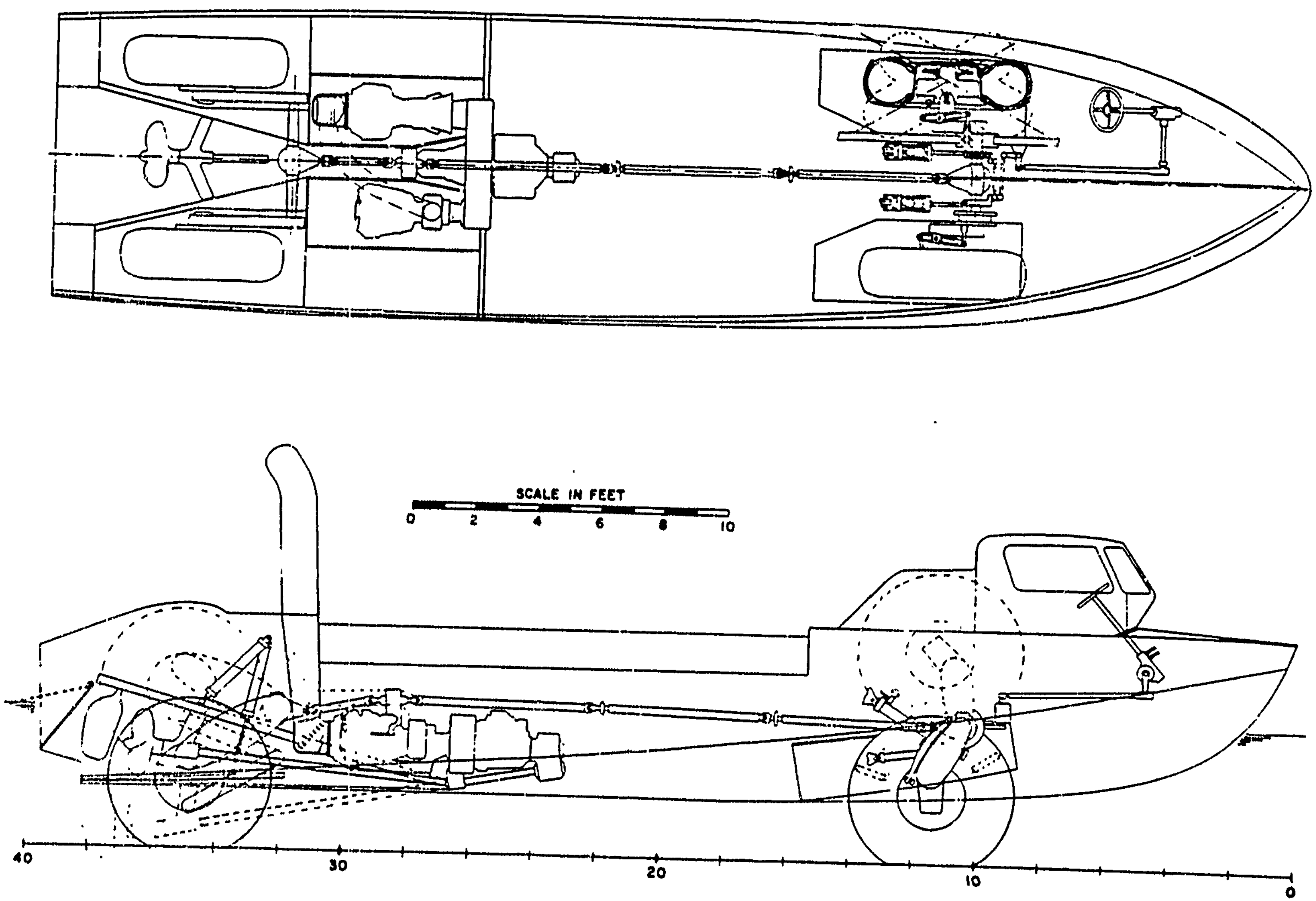


FIG. 23a. CONCEPT NO. 3 - LAYOUT DRAWINGS: STARBOARD ELEVATION AND PLAN VIEWS

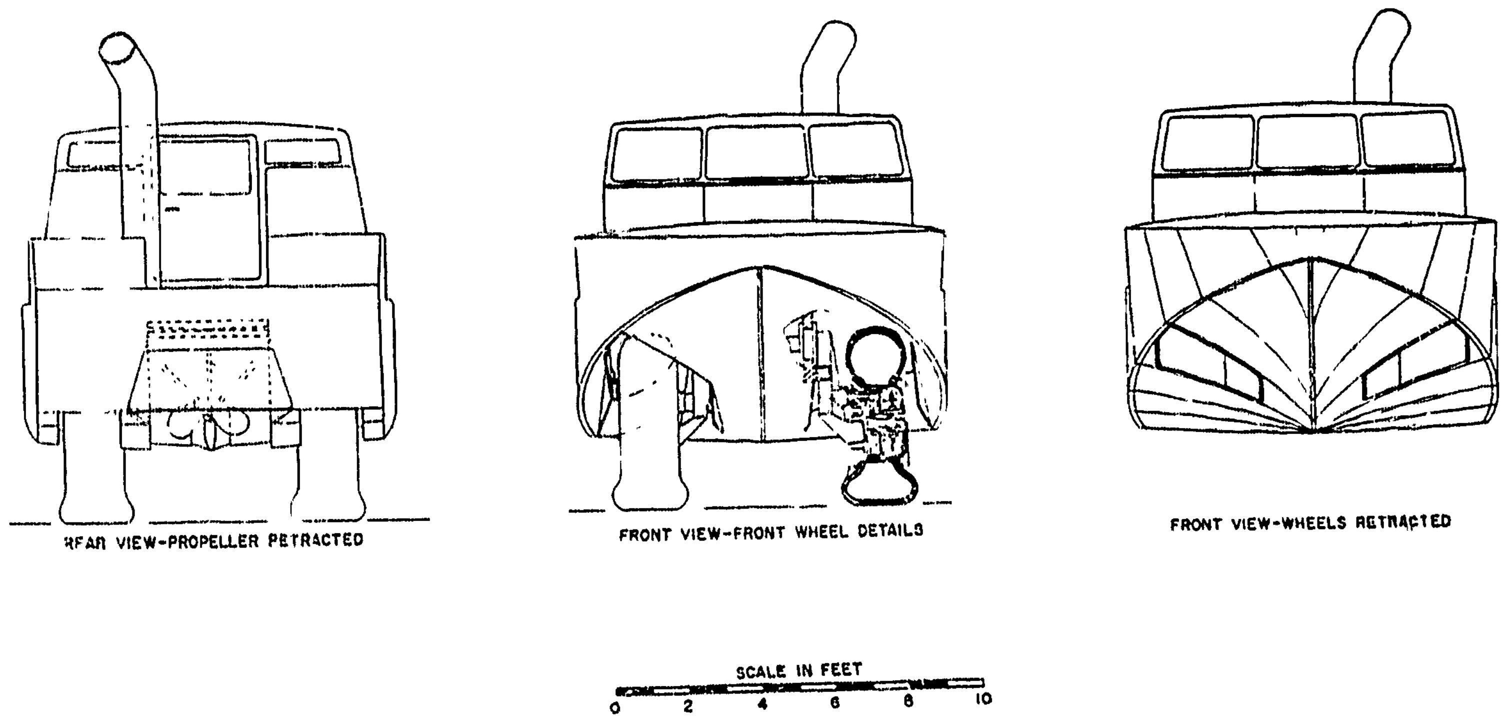


FIG. 23b. CONCEPT NO. 3 - LAYOUT DRAWINGS: FRONT AND REAR VIEWS

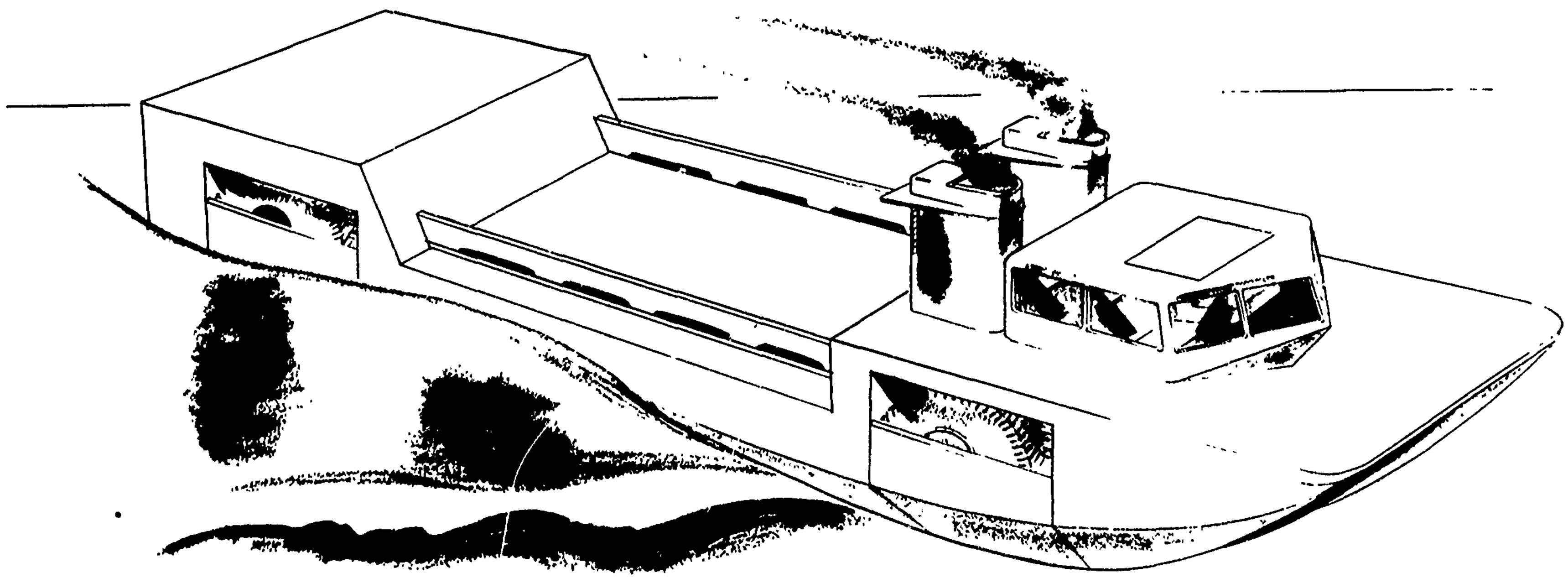


FIG. 24. CONCEPT NO. 4: A 5-TON, 4x4, W-BOTTOM PLANING AMPHIBIAN WITH RETRACTABLE RIGHT-ANGLE DRIVE PROPELLERS, "FLOP-OVER" WHEEL RETRACTION, ACKERMANN STEERING

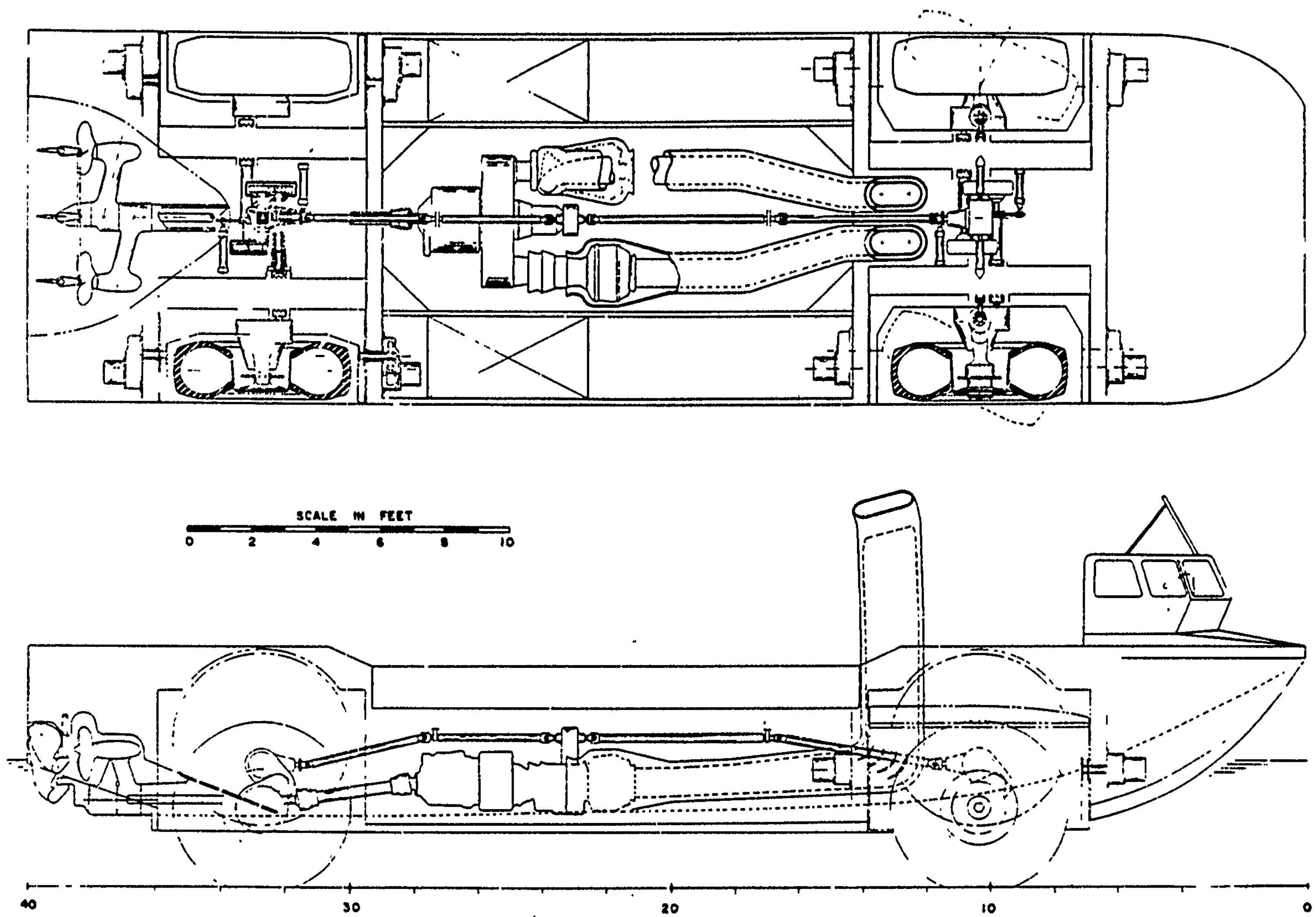


FIG. 25a. CONCEPT NO. 4 - LAYOUT DRAWINGS: STARBOARD ELEVATION AND PLAN VIEWS

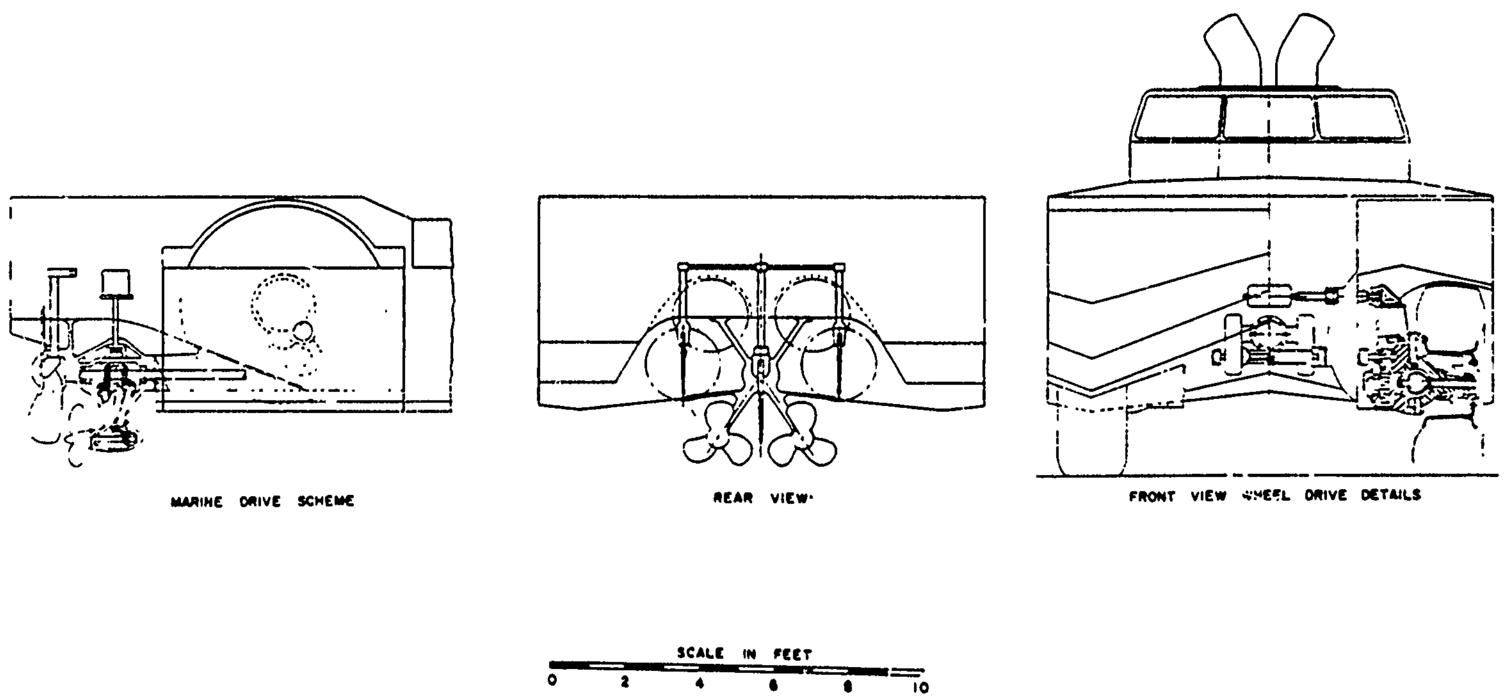


FIG. 25b. CONCEPT NO. 4 - LAYOUT DRAWINGS: FRONT AND REAR VIEWS

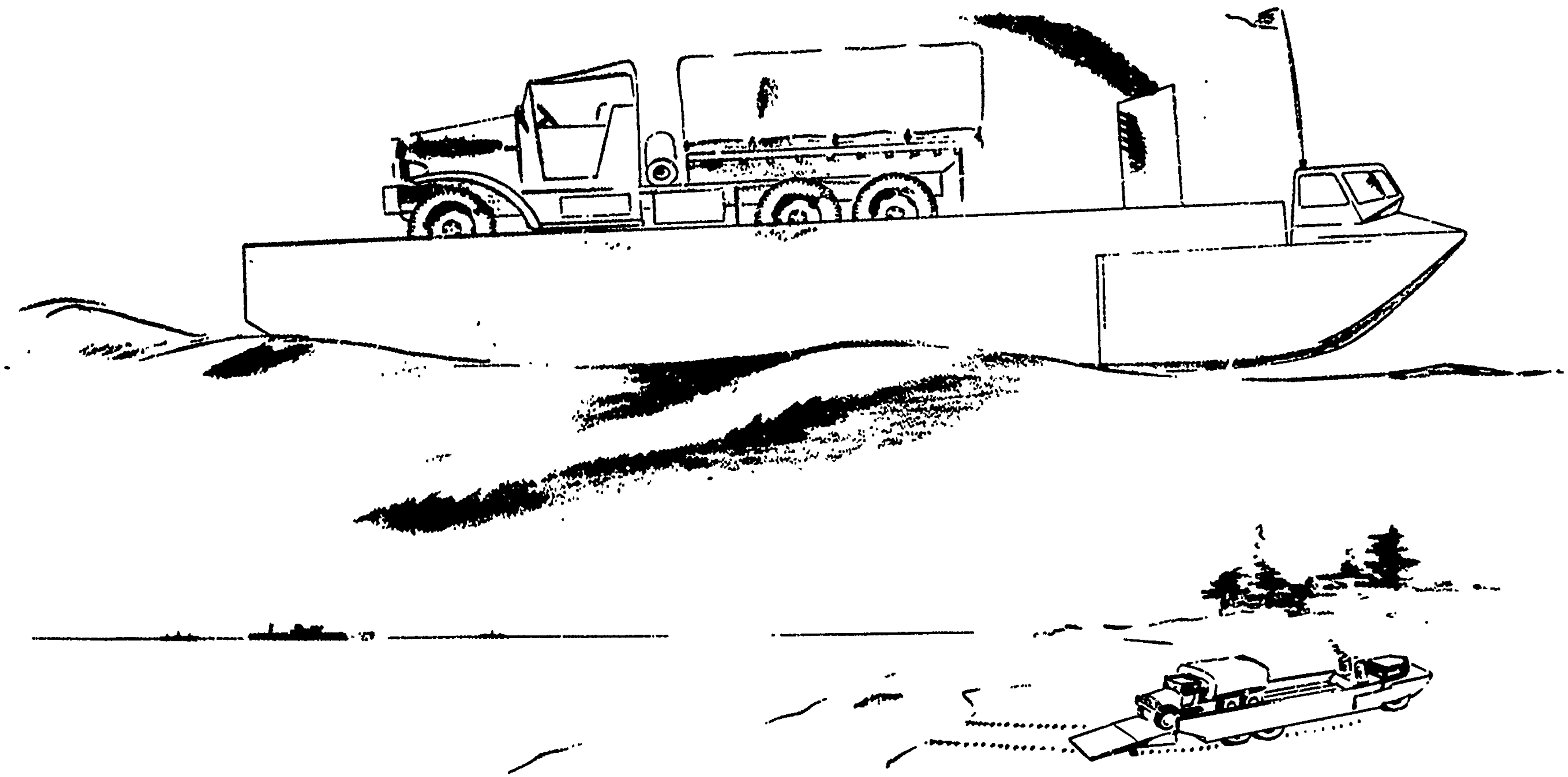


FIG. 26. CONCEPT NO. 5 - A 15-TON, 6x6, W-BOTTOM, PLANING AMPHIBIAN WITH RETRACTABLE PROPELLER AND TUNNEL, RETRACTABLE WHEELS, CHAIN AND HYDROSTATIC DRIVE, "GOER"-TYPE ARTICULATED STEERING

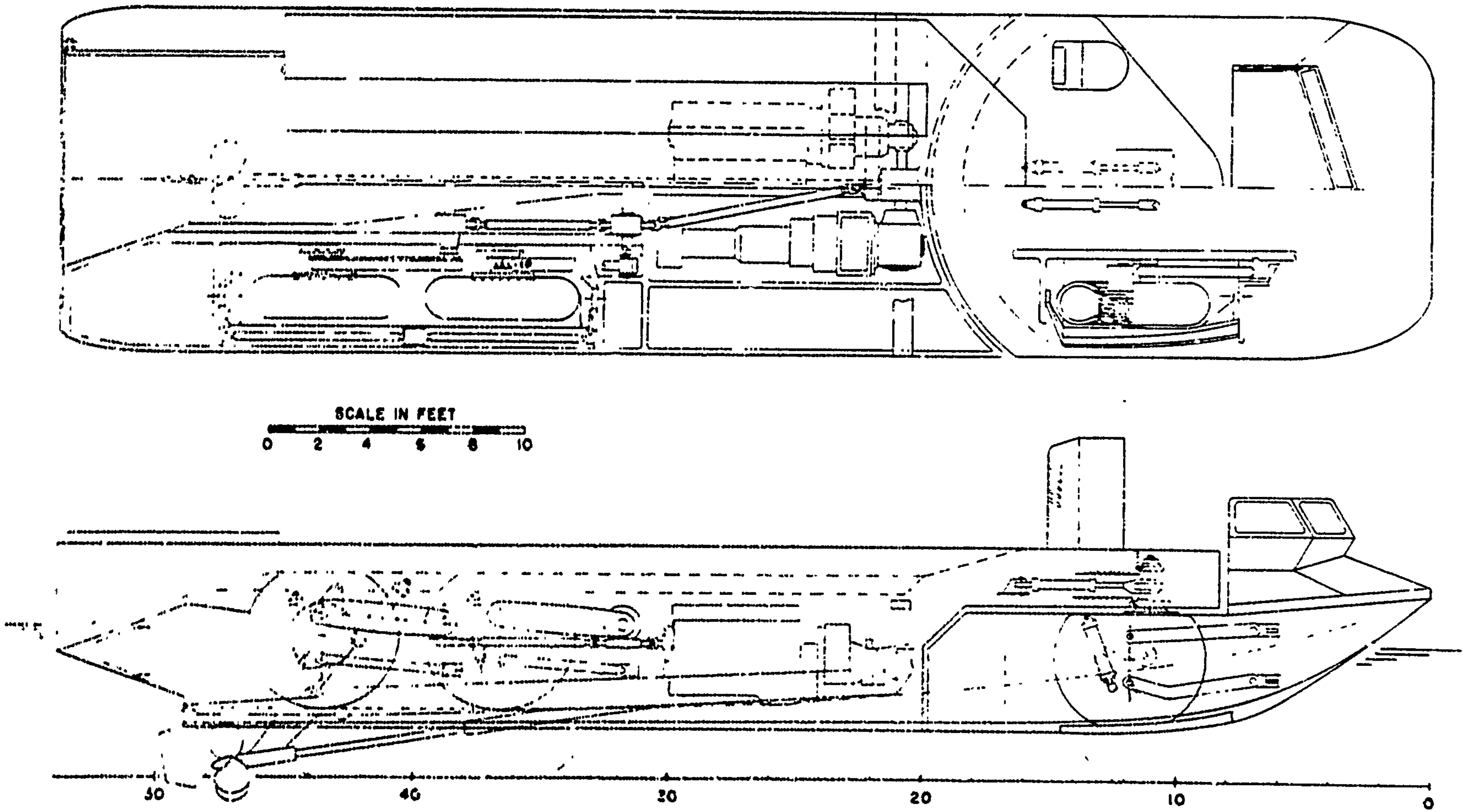


FIG. 27a. CONCEPT NO. 5 - LAYOUT DRAWINGS: STARBOARD AND PLAN VIEWS

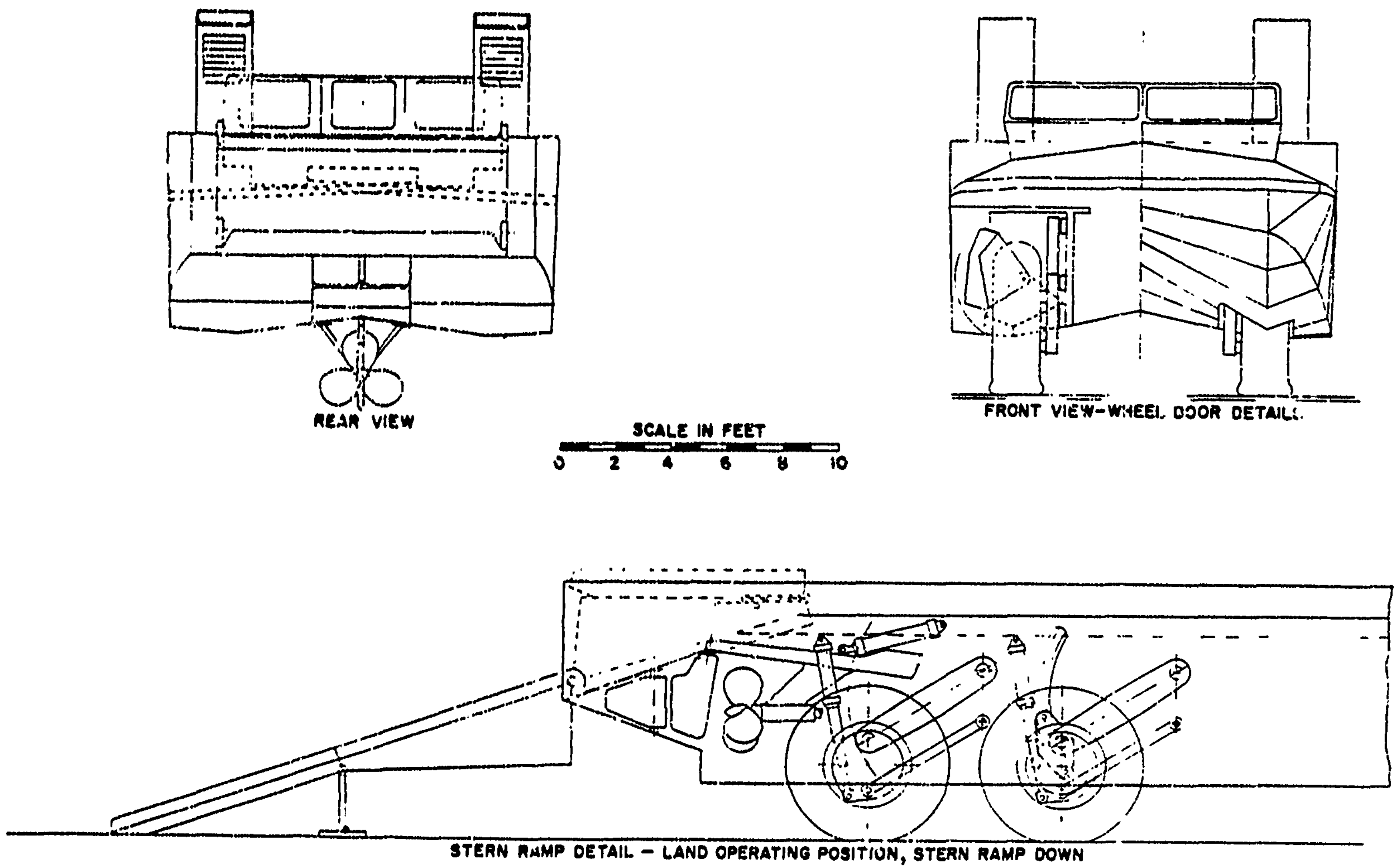


FIG. 27b. CONCEPT NO. 5 - LAYOUT DRAWINGS: FRONT AND REAR VIEWS, STERN RAMP DETAILS

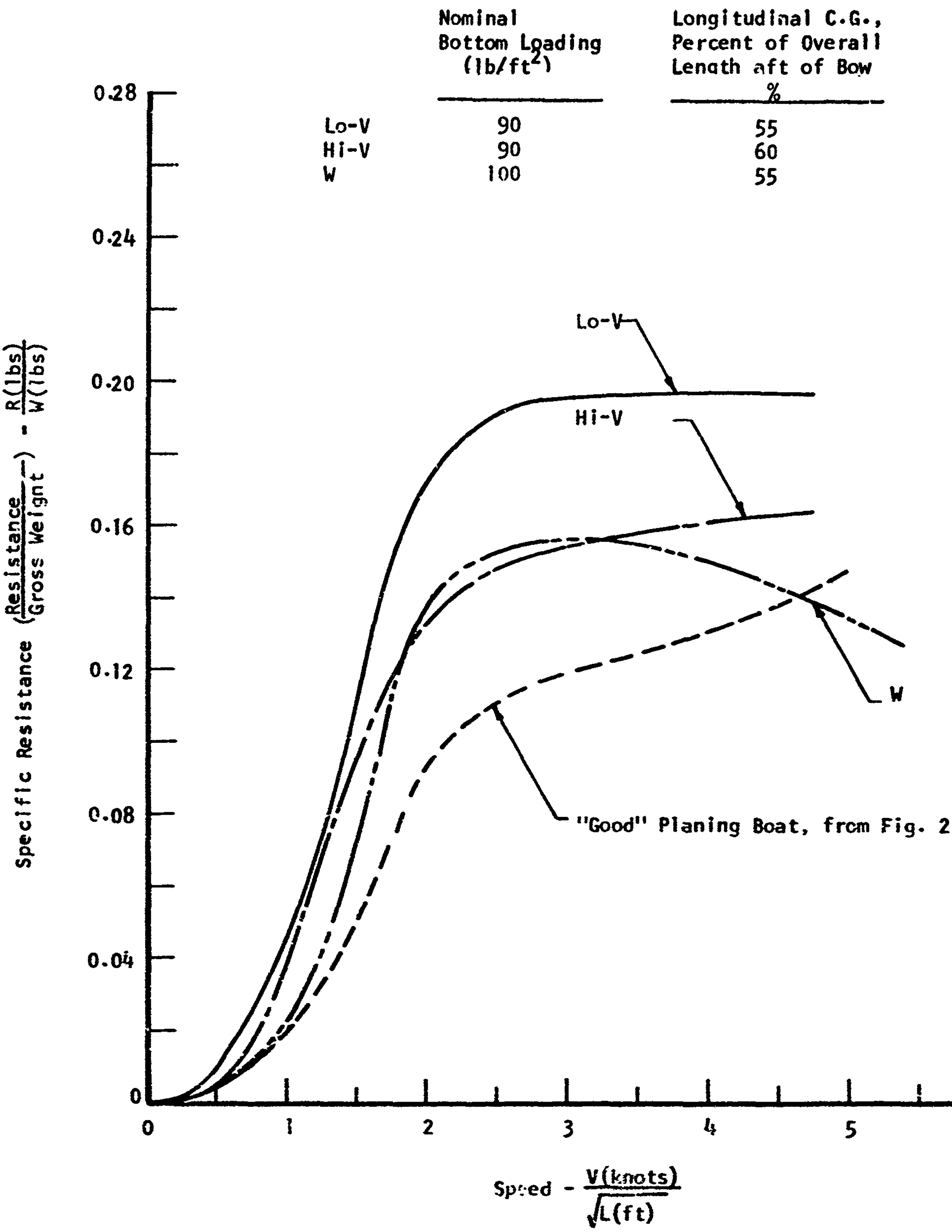


FIG. 28. TYPICAL TOWING TANK RESULTS FOR FOUR 40-FT PLANING AMPHIBIANS, BARE HULLS

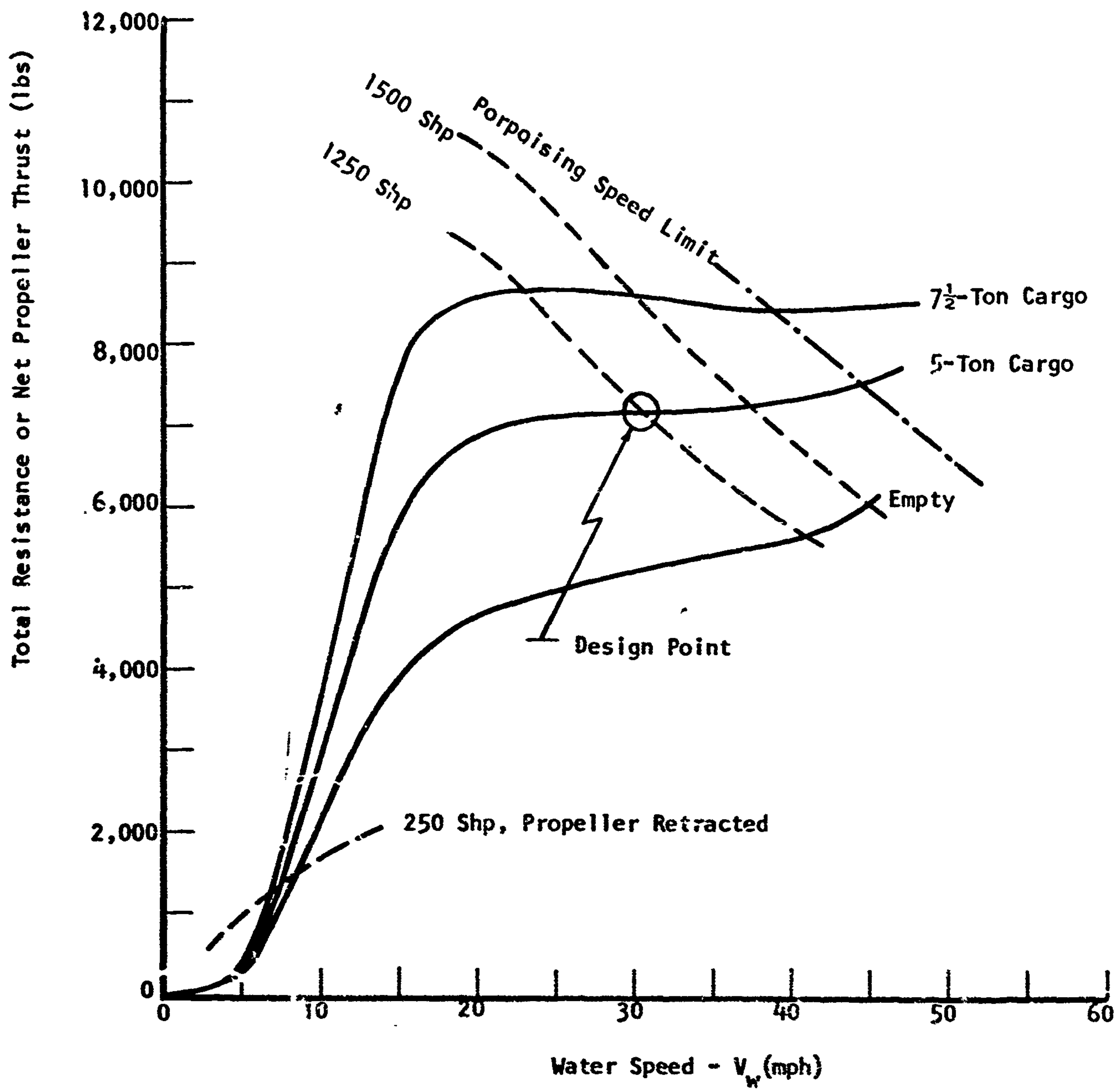


FIG. 29. CONCEPT NO. 1 - CALCULATED STILL WATER PERFORMANCE

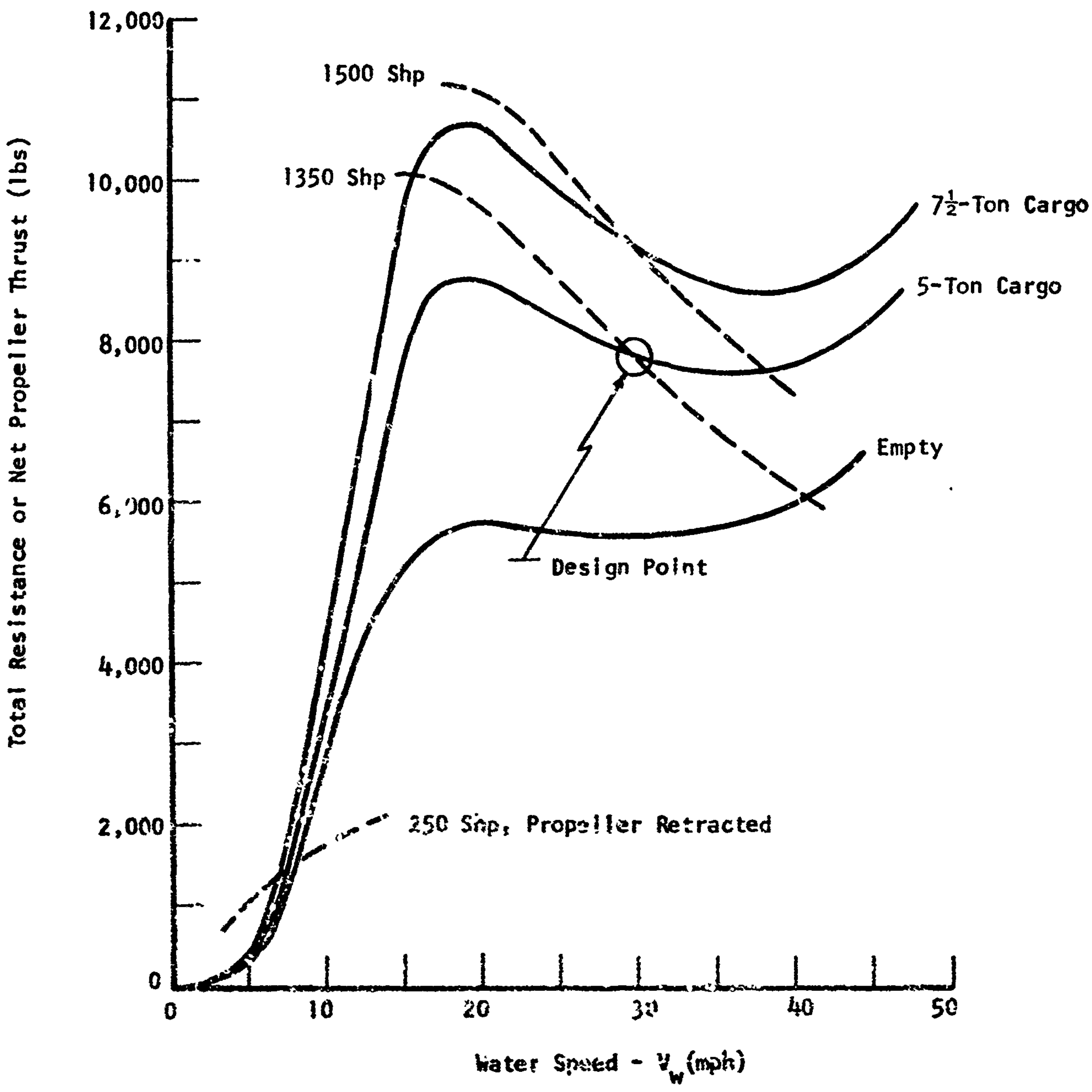


FIG. 30. CONCEPT NO. 2 - CALCULATED STILL WATER PERFORMANCE

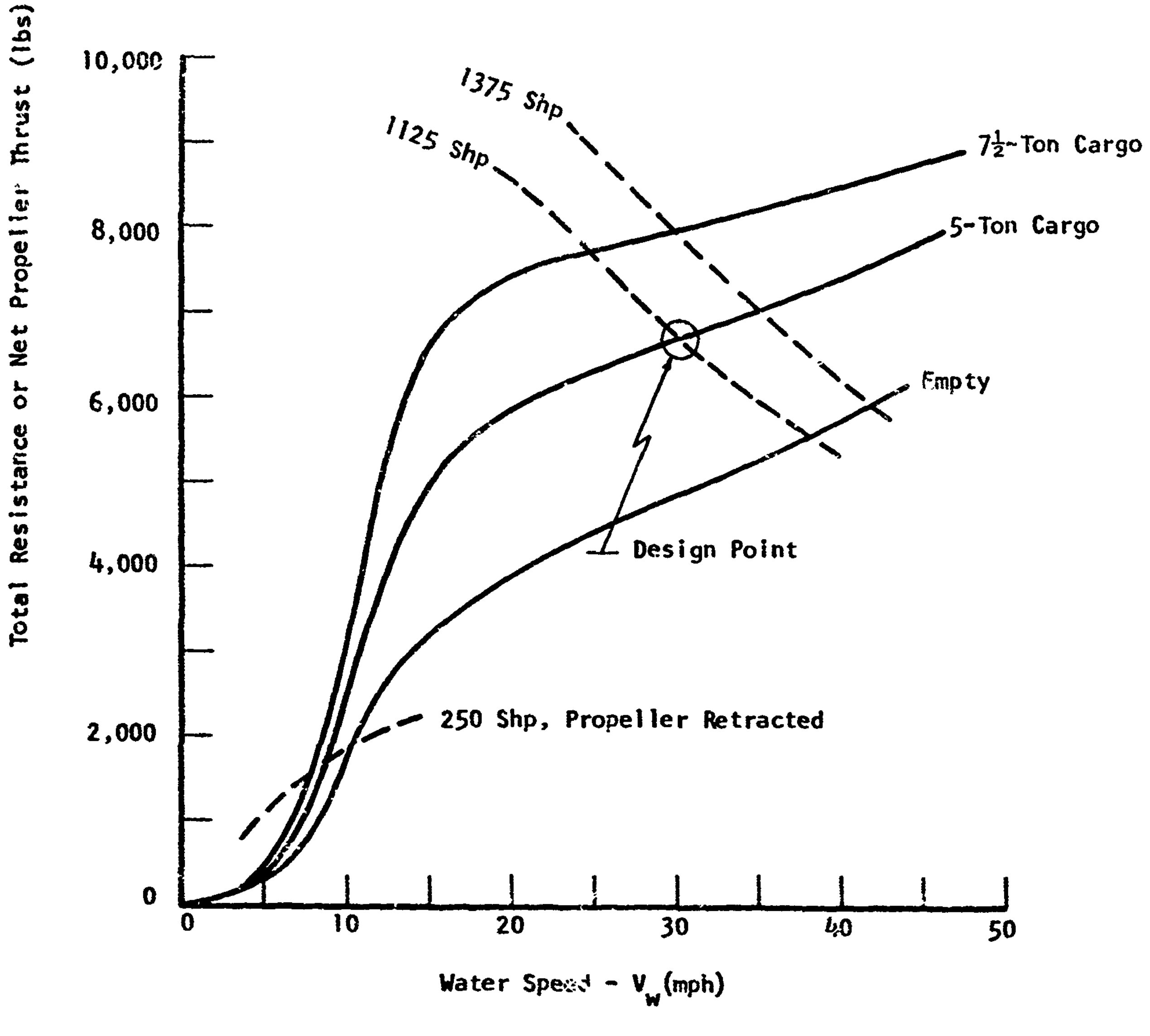


FIG. 31. CONCEPT NO. 3 - CALCULATED STILL WATER PERFORMANCE

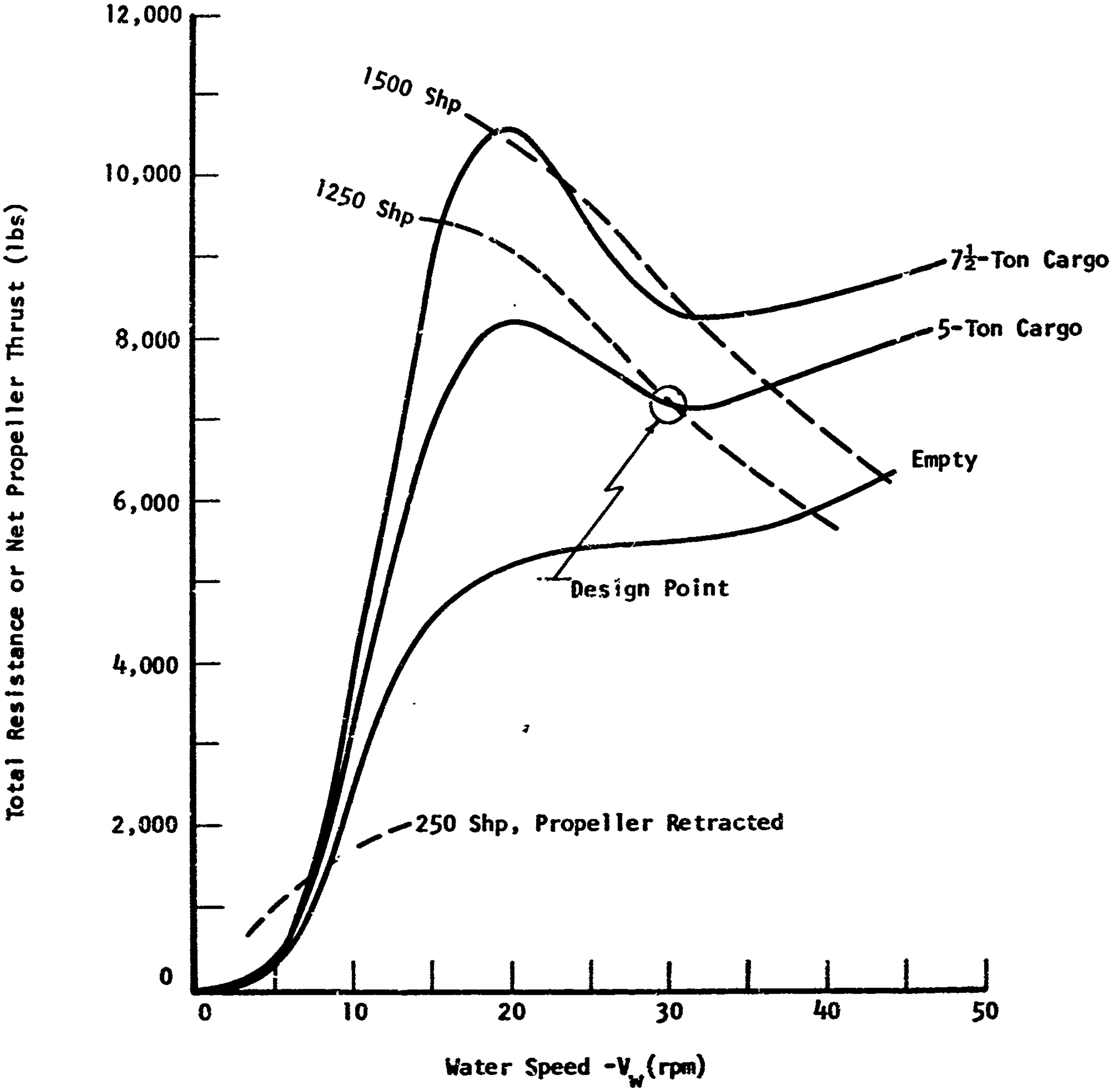


FIG. 32. CONCEPT No. 4 - CALCULATED STILL WATER PERFORMANCE

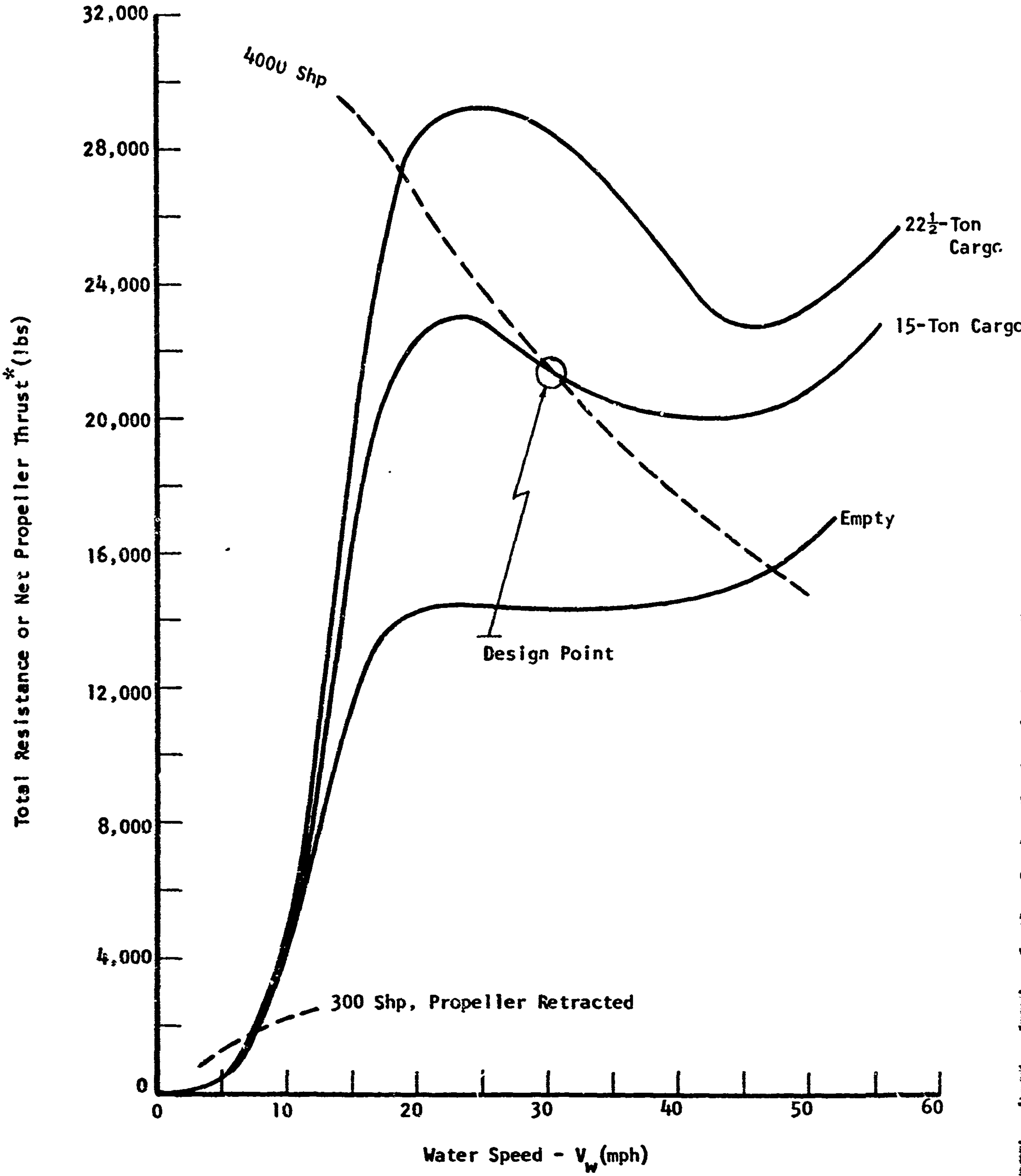
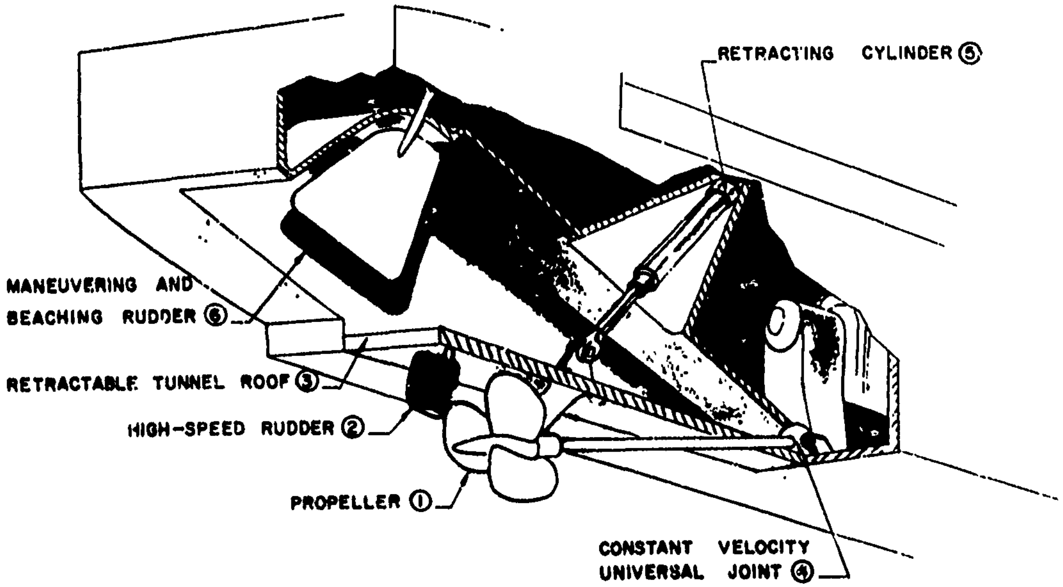
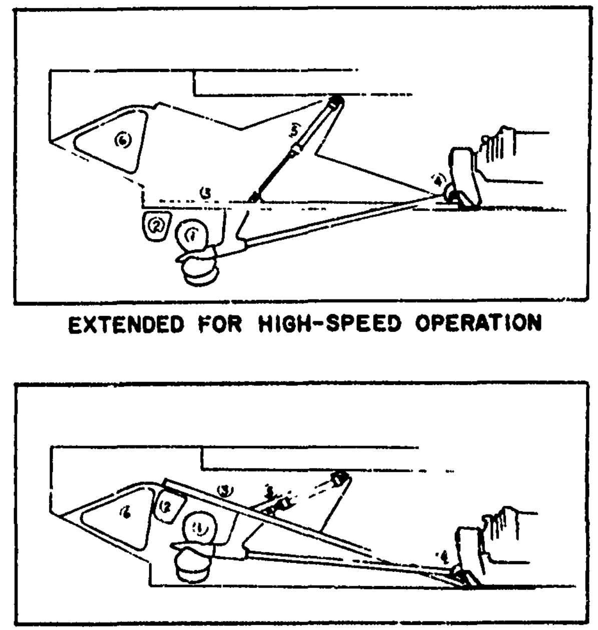


FIG. 33. CONCEPT NO. 5 - CALCULATED STILL WATER PERFORMANCE

*Note change of resistance scale when comparing with Figs. 29-32



RETRACTION SEQUENCE



EXTENDED FOR HIGH-SPEED OPERATION

RETRACTED FOR MANEUVERING, BEACHING AND LAND OPERATION

FIG. 34. RETRACTABLE PROPELLER AND TUNNEL SYSTEM

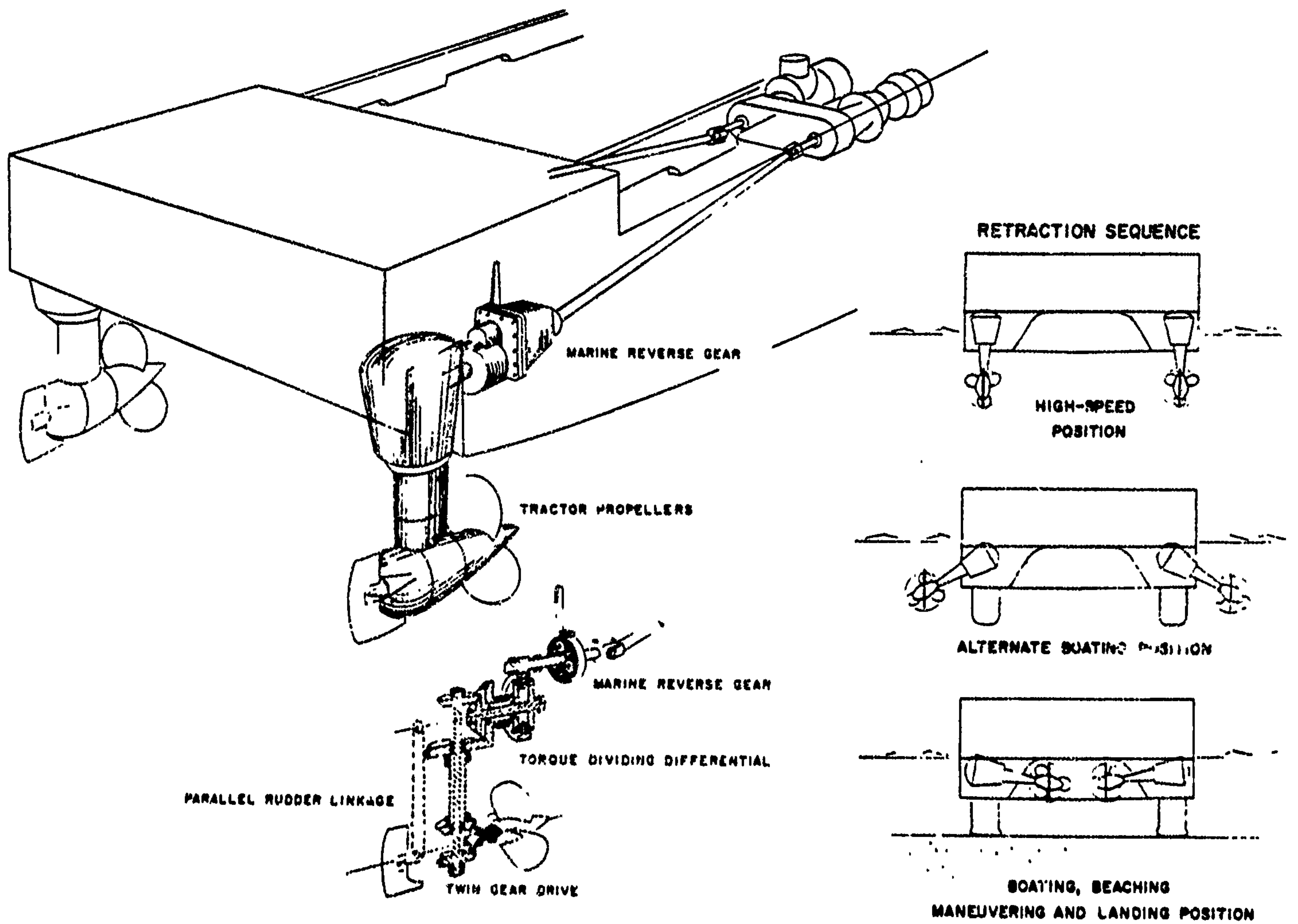


FIG. 35. RIGHT-ANGLE PROPELLER DRIVE AND RETRACTION SYSTEM

RUNNING GEAR VERSUS PERFORMANCE + = BETTER WITH INCREASING VALUE − = WORSE WITH INCREASING VALUE		LAND PERFORMANCE: SOFT SOIL: MOBILITY	LAND PERFORMANCE: OBSTACLES: GROUND CLEARANCE	LAND PERFORMANCE: OBSTACLES: APPROACH ANGLE	LAND PERFORMANCE: OBSTACLES: BREAK ANGLE	LAND PERFORMANCE: OBSTACLES: DEPARTURE ANGLE	LAND PERFORMANCE: RIDE	LAND PERFORMANCE: ROADABILITY	LAND PERFORMANCE: SLOPE: SIDE	LAND PERFORMANCE: SLOPE: LONGITUDINAL	LAND PERFORMANCE: AGILITY	WATER PERFORMANCE: SURF AND BEACHING	WATER PERFORMANCE: SPEED	WATER PERFORMANCE: SEA KINDLINESS	WATER PERFORMANCE: STATIC STABILITY	LOAD PERFORMANCE
NUMBER OF TIRES (SINGLES)		+	−		+		+	+	+	+	+		+		+	
TIRE	DIAMETER	+	+	+	+	+	+		−			+	−		−	−
	SECTION WIDTH	+					+		−			+			−	
SUSPENSION DEPTH		+					+		−	−					−	−
STEERING MODE	ACKERMAN							+			+					
	ARTICULATED FRAME	+										+			+	
WHEEL DRIVE	DIRECT	+														
	FRICTION						−								+	+
CROSS VEHICLE WEIGHT *		−									−	−	−		−	−

* AS AFFECTED BY RUNNING GEAR DESIGN

FIG. 36. INTERACTION OF RUNNING GEAR FEATURES AND PERFORMANCE

INDEPENDENT / DEPENDENT	GROSS VEHICLE WEIGHT *	NUMBER OF TIRES	TIRE DIAMETER	TIRE SECTION WIDTH	WHEEL DRIVE ARRANGEMENT	STEERING MODE	SUSPENSION DEPTH	HULL
GROSS VEHICLE WEIGHT *		2	2	2	2	1	1	
NUMBER OF TIRES	1		1	1	2	2	2	1
TIRE DIAMETER	1	1		1	2	1	2	1
TIRE SECTION WIDTH	1	1	1			2		1
WHEEL DRIVE ARRANGEMENT	2	1	2			1	1	2
STEERING MODE	2	2	1					1
SUSPENSION DEPTH		2	2	2	1	2		
HULL		1	1	1	2	1		

* AS AFFECTED BY RUNNING GEAR DESIGN

1 PRIMARY INFLUENCE

2 SECONDARY INFLUENCE

FIG. 37. INTERACTION OF VARIOUS RUNNING GEAR FEATURES

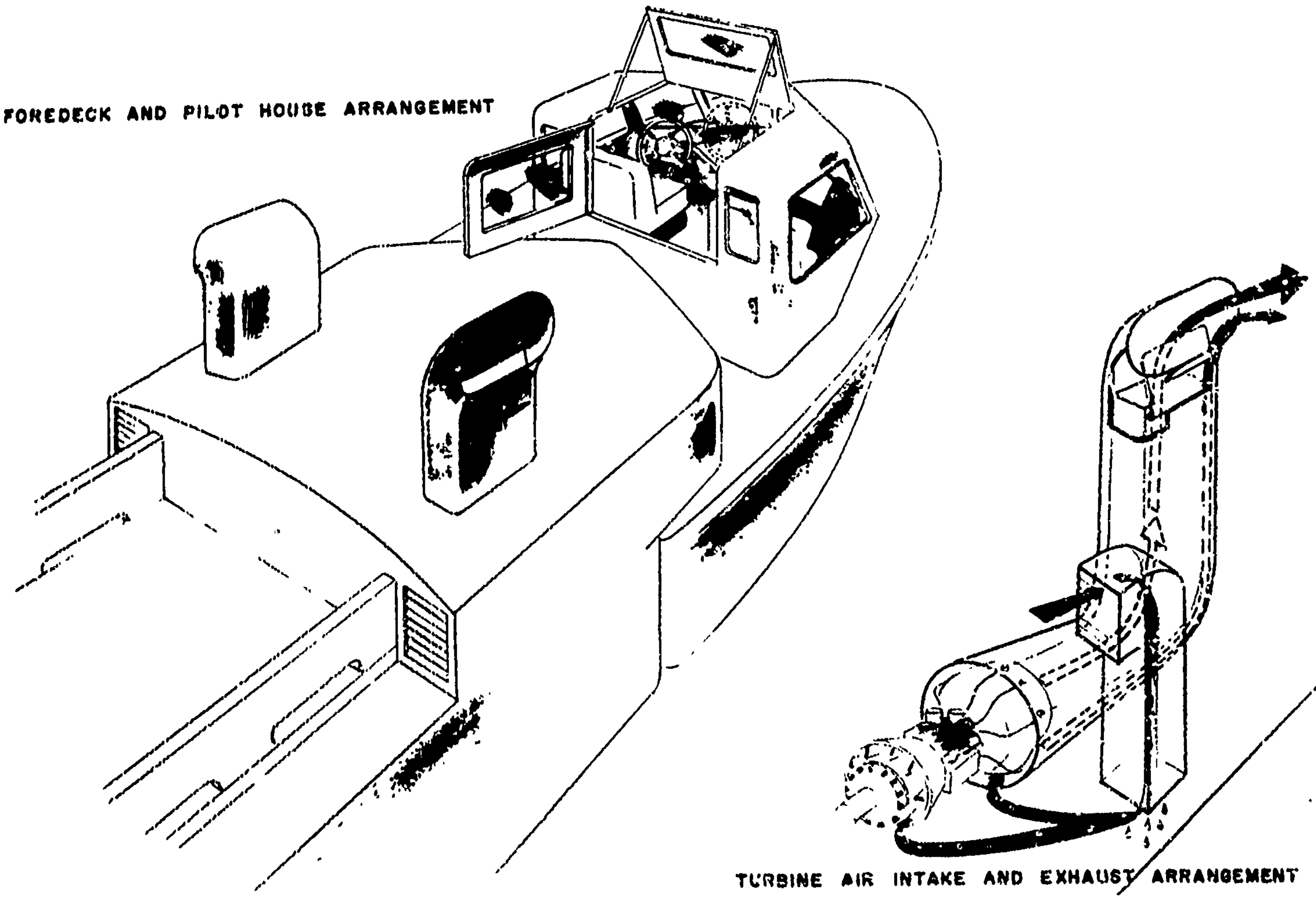


FIG. 38. CONCEPT NO. 1 - FOREDECK AND ENGINE BREATHINS ARRANGEMENTS

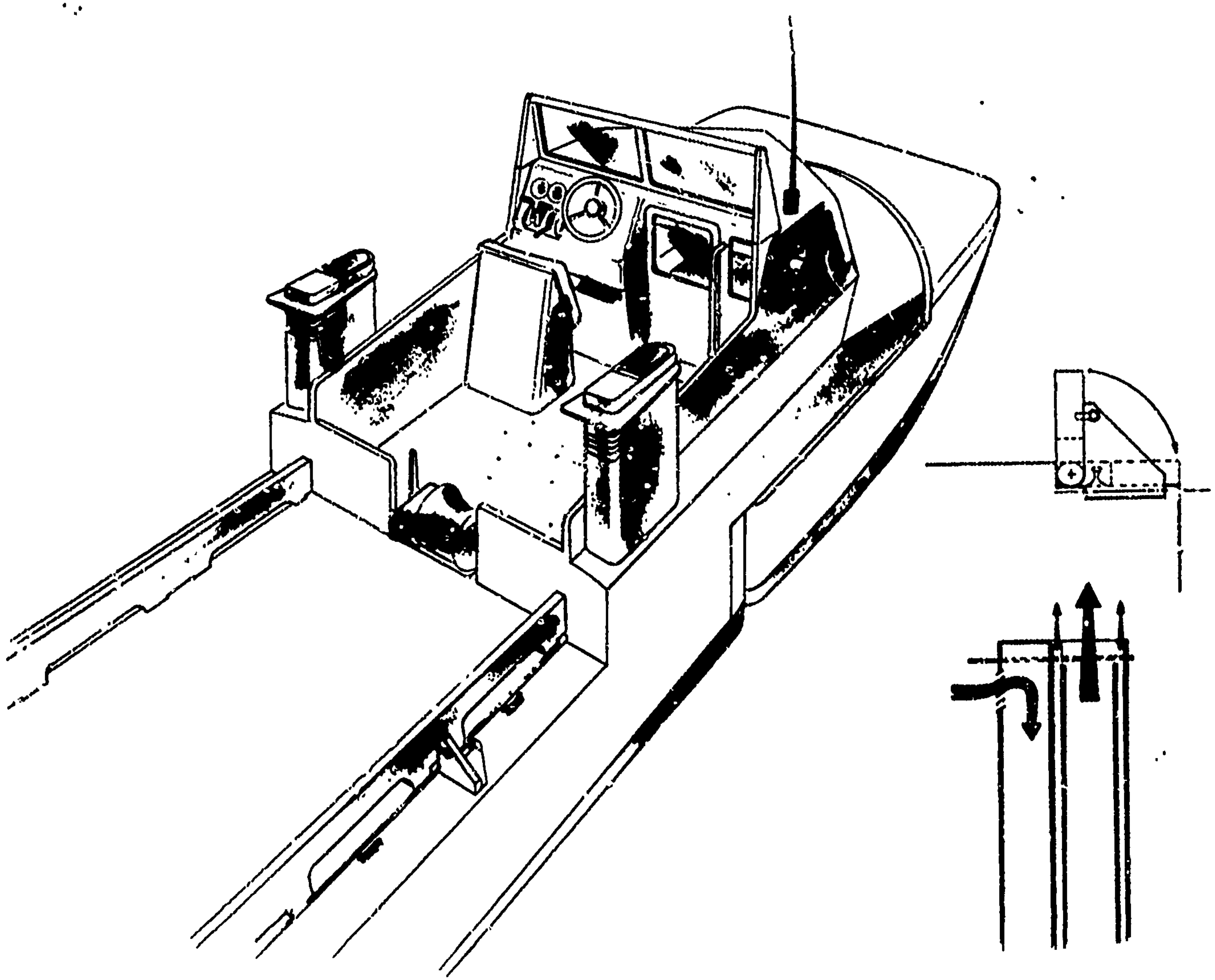


FIG. 39. CONCEPT NO. 5 - FOREDECK AND ENGINE BREATHING ARRANGEMENTS

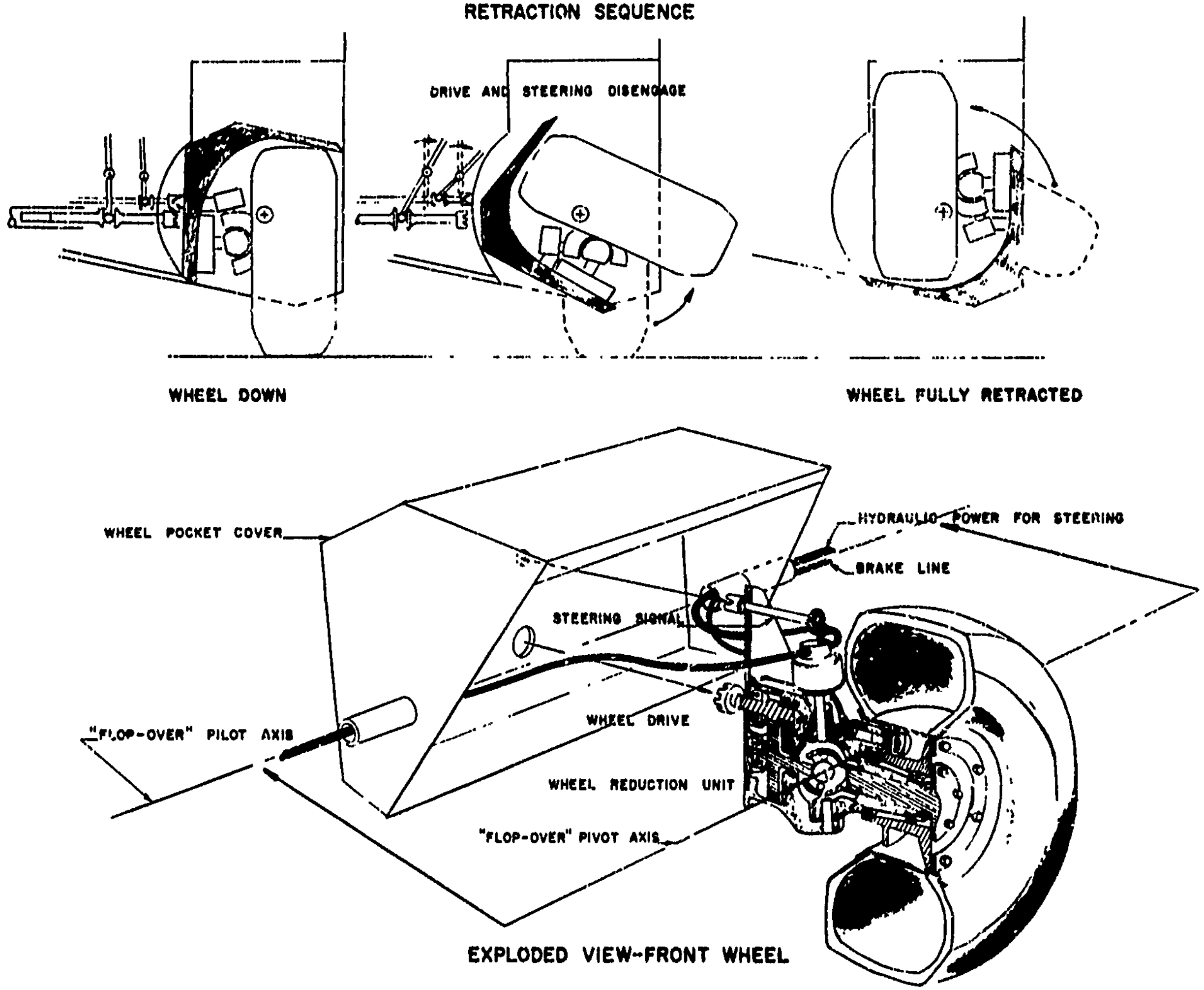


FIG. 40. CONCEPT NO. 4 - "FLOP-OVER" WHEEL RETRACTION SYSTEM

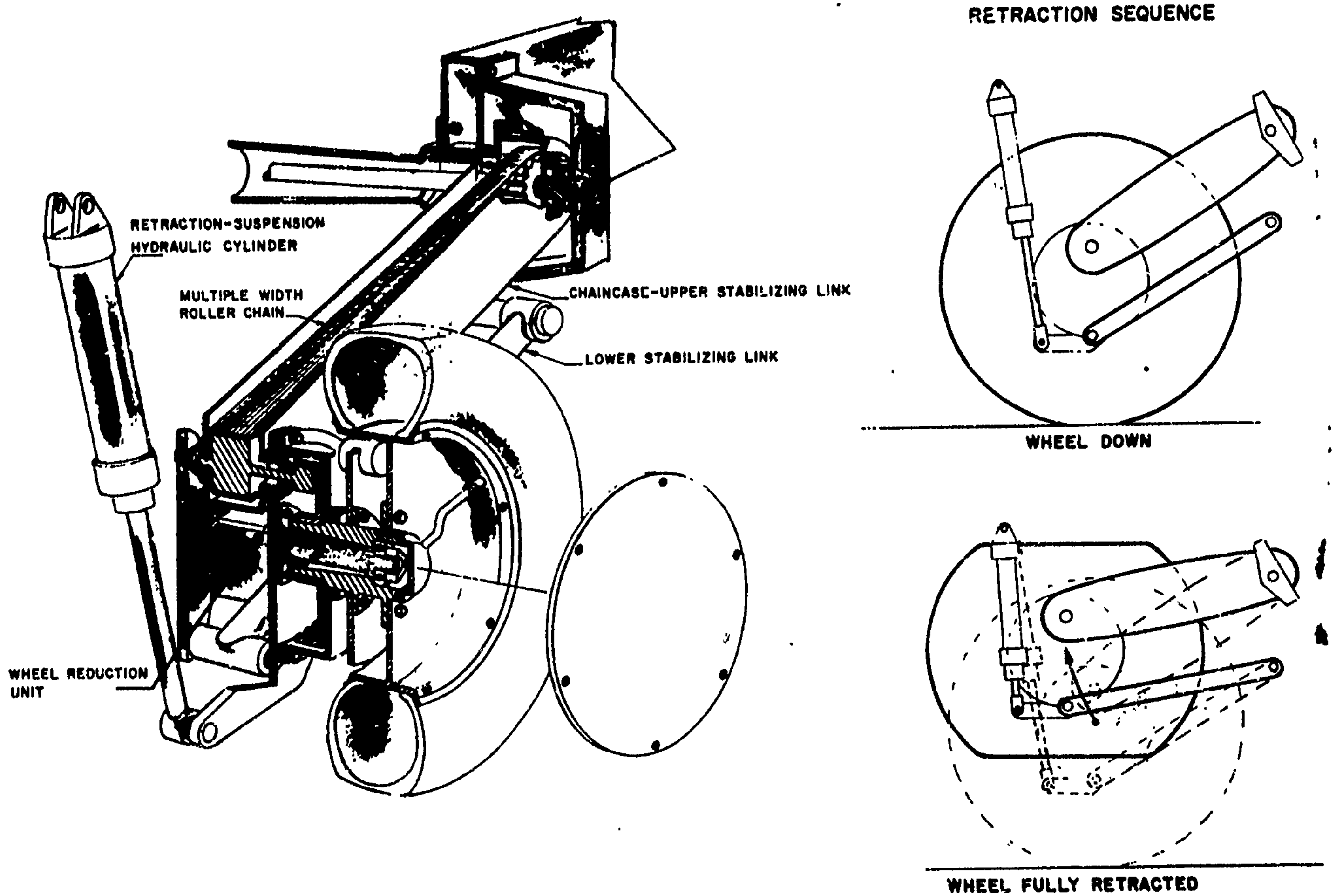


FIG. 41. CONCEPTS NO. 2 and 3 - CHAIN DRIVE AND RETRACTION SYSTEM

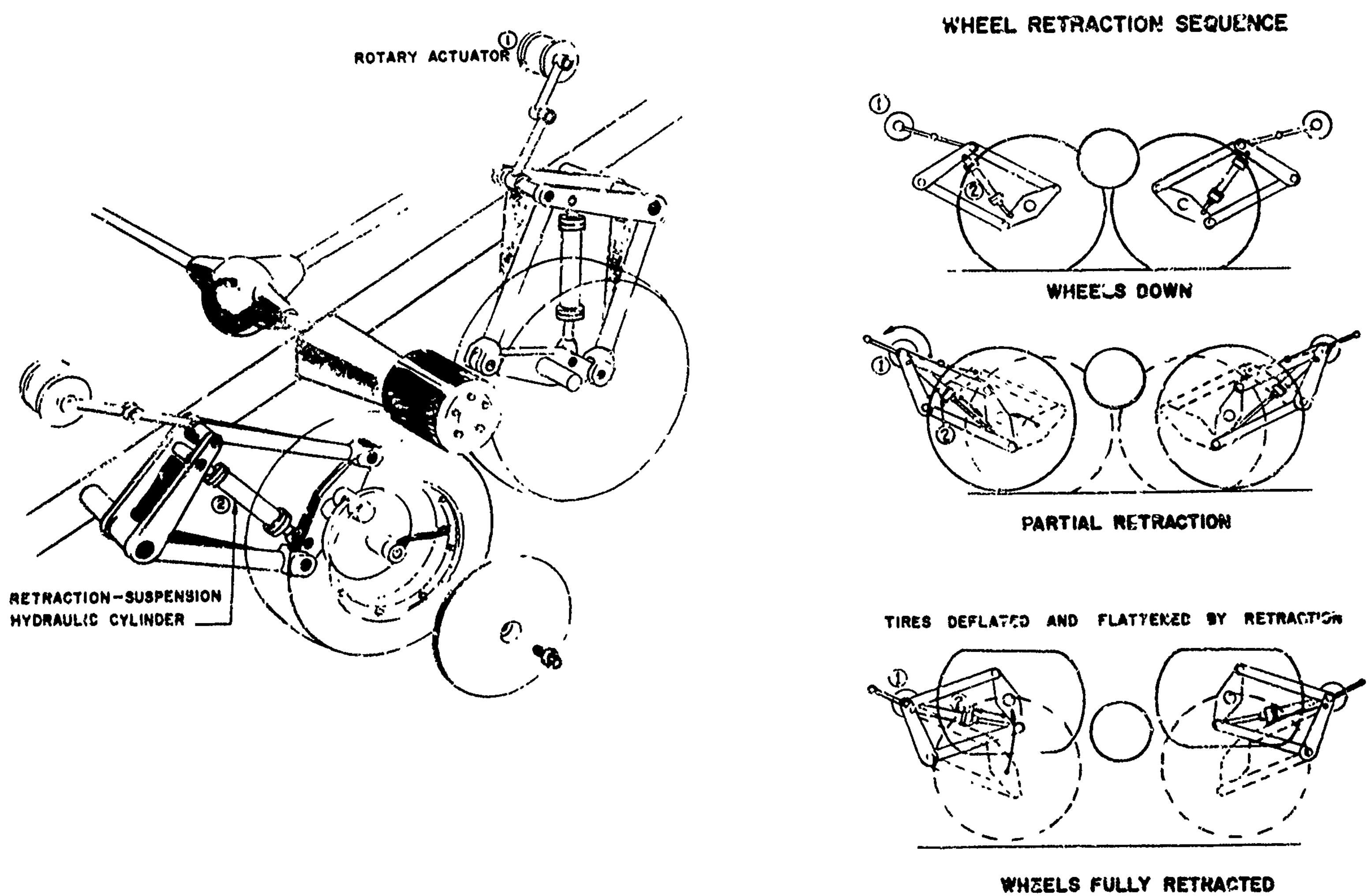


FIG. 42. CONCEPT NO. 1 - ROLLER DRIVE AND WHEEL RETRACTION SYSTEM

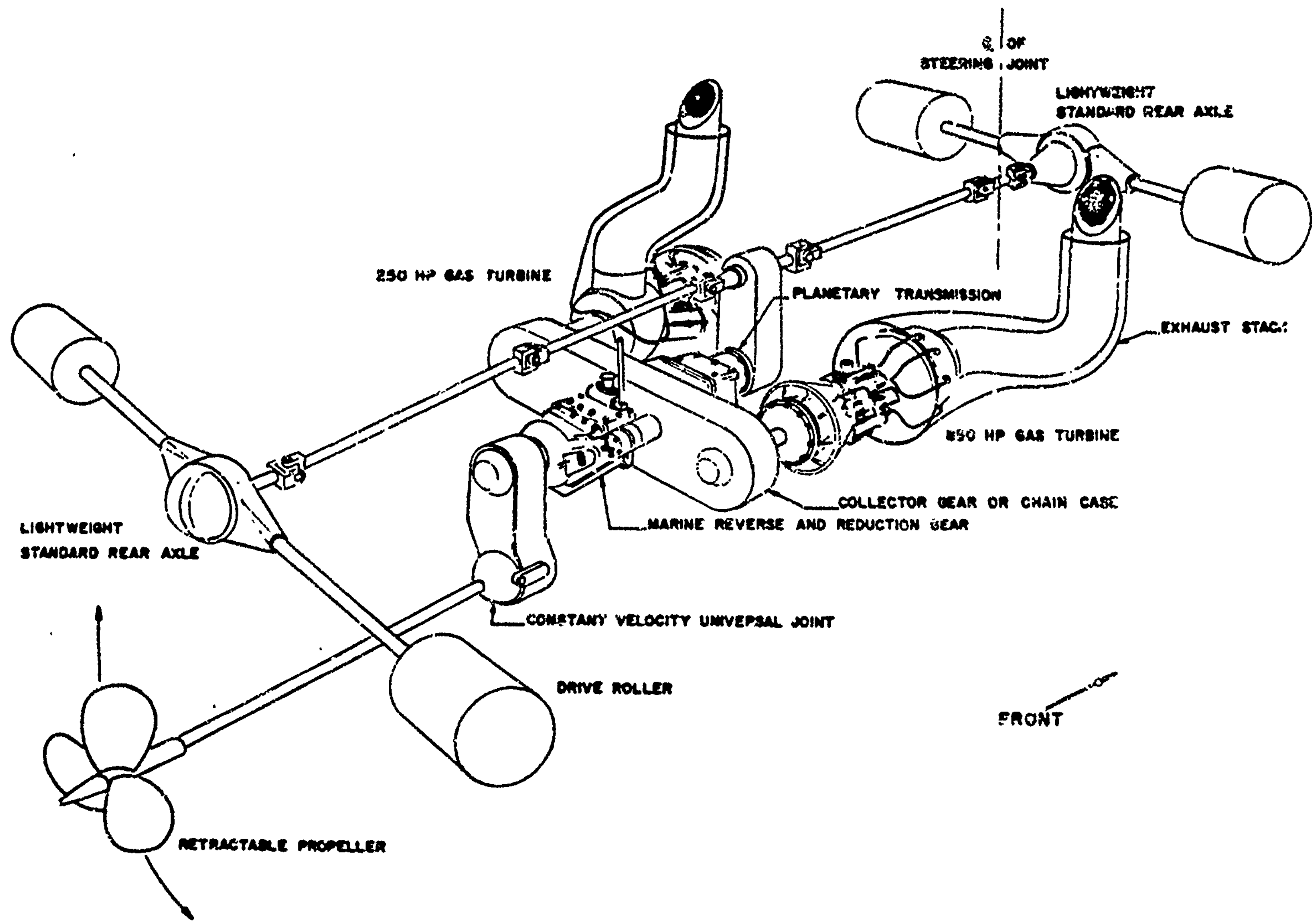
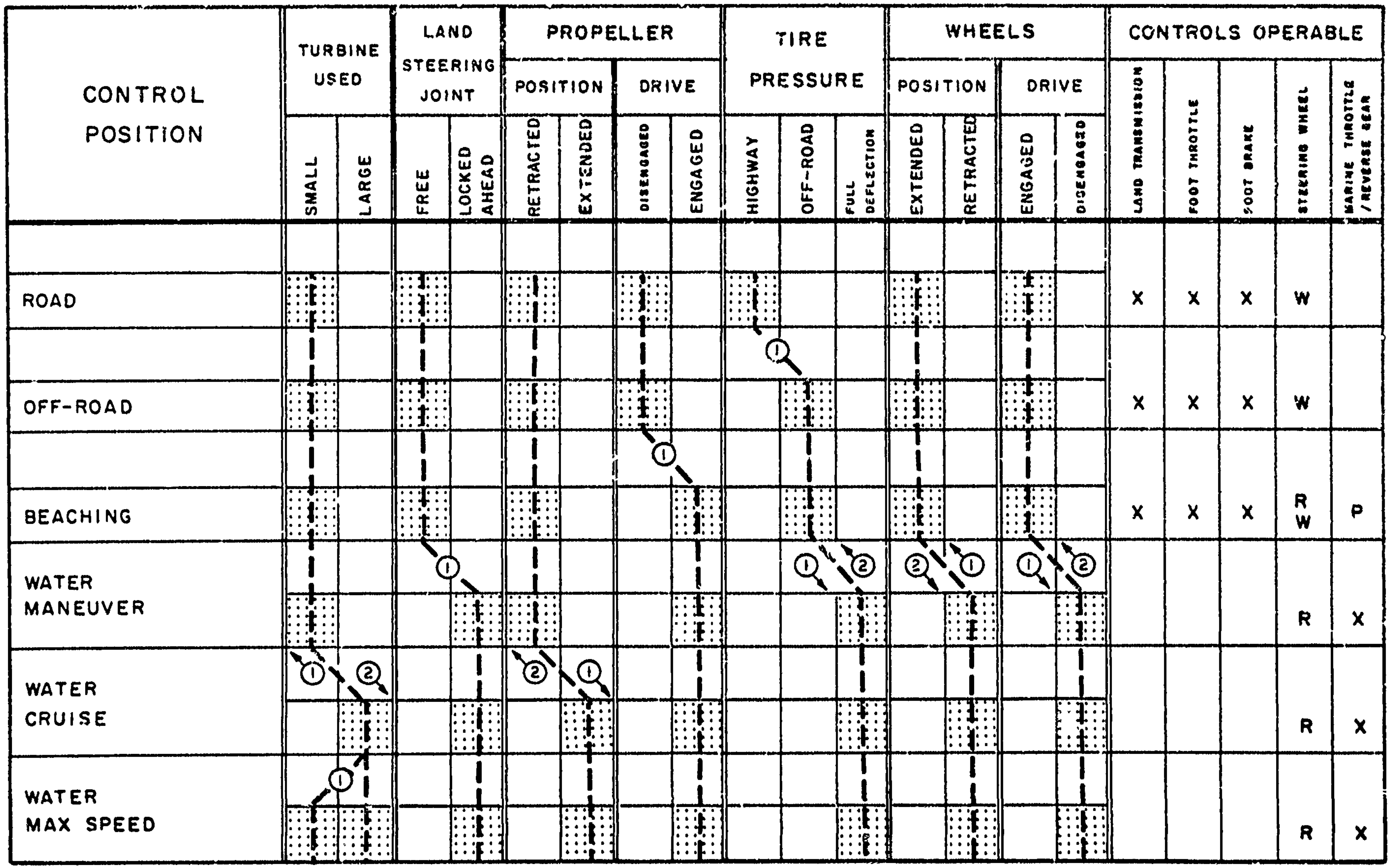


FIG. 43. TWIN TURBINE POWER PLANT ARRANGEMENT AS APPLIED TO THE ROLLER DRIVE OF CONCEPT NO. 1



①② INDICATE SEQUENCE   R=RUDDER, W=WHEELS   P=PARTIAL, ie, FORWARD/REVERSE ONLY

FIG. 44. FUNCTIONAL SCHEMATIC OF "OPERATION SELECTOR" CONTROL OPERATION FOR ARTICULATED PLANING AMPHIBIANS (OTHER PLANING AMPHIBIANS ARE ESSENTIALLY SIMILAR)

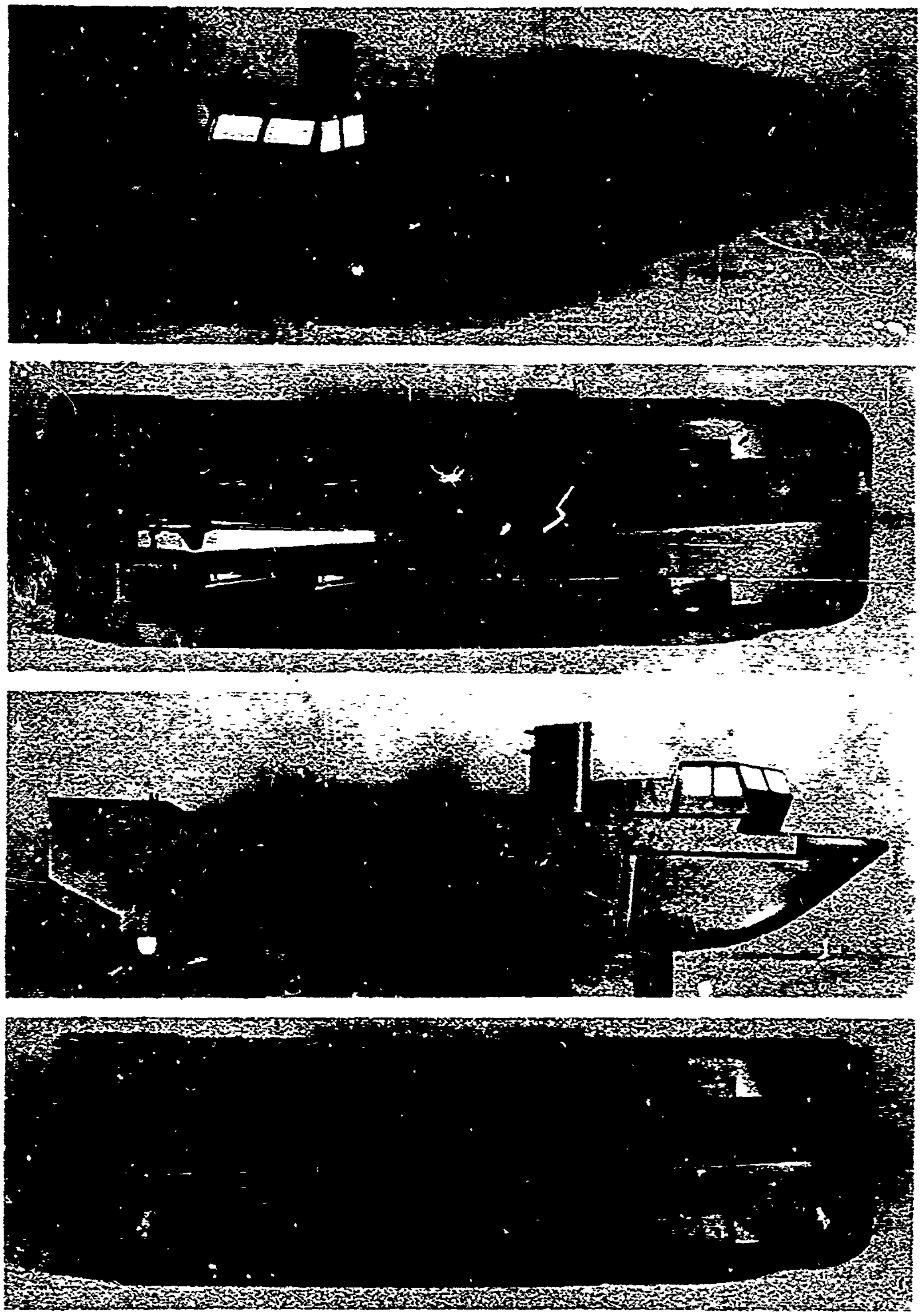


FIG. 45 - CONCEPT NO. 2 - SCALE MOCK-UP

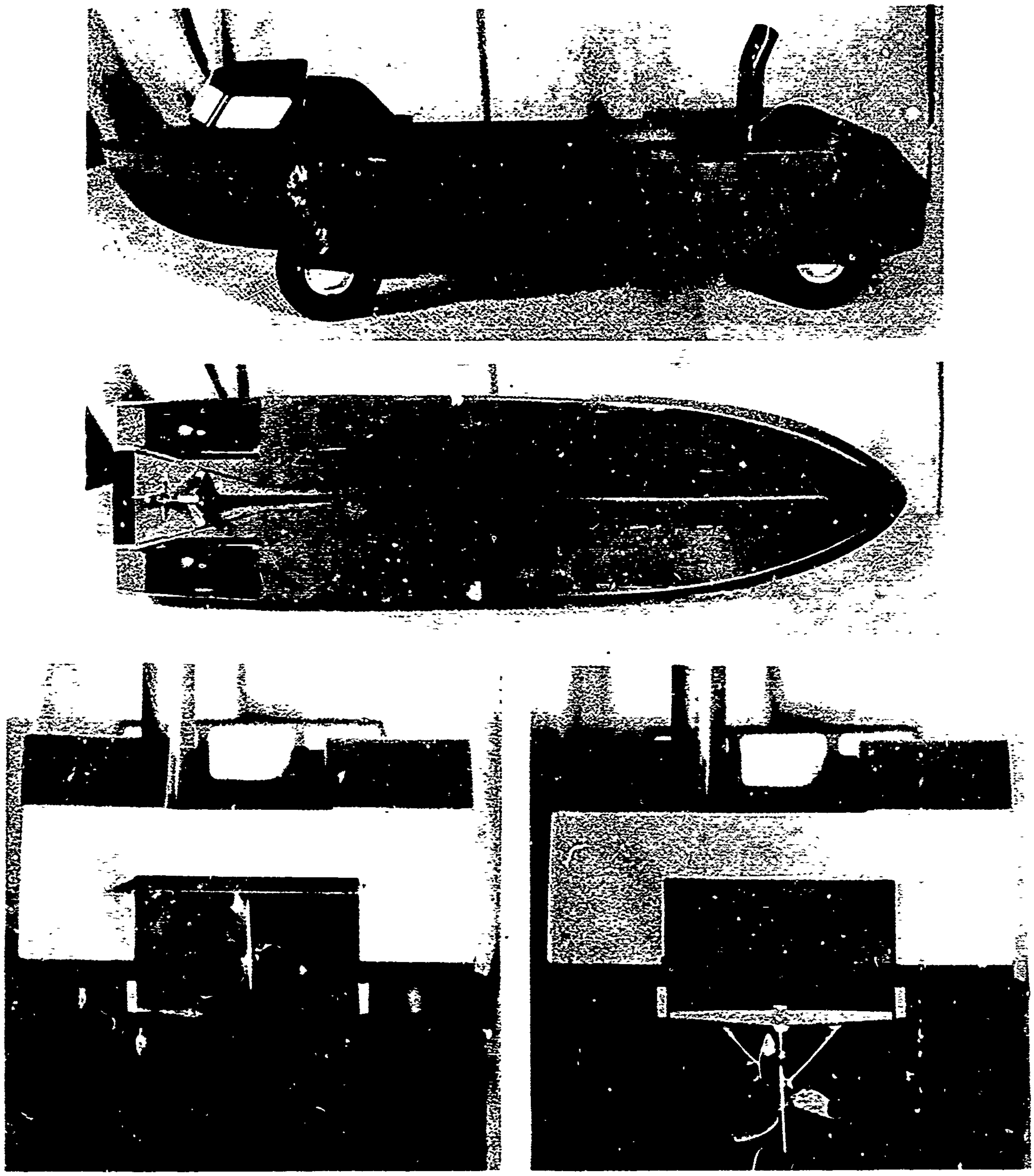


FIG. 46. CONCEPT NO. 3 - SCALE MOCK-UP

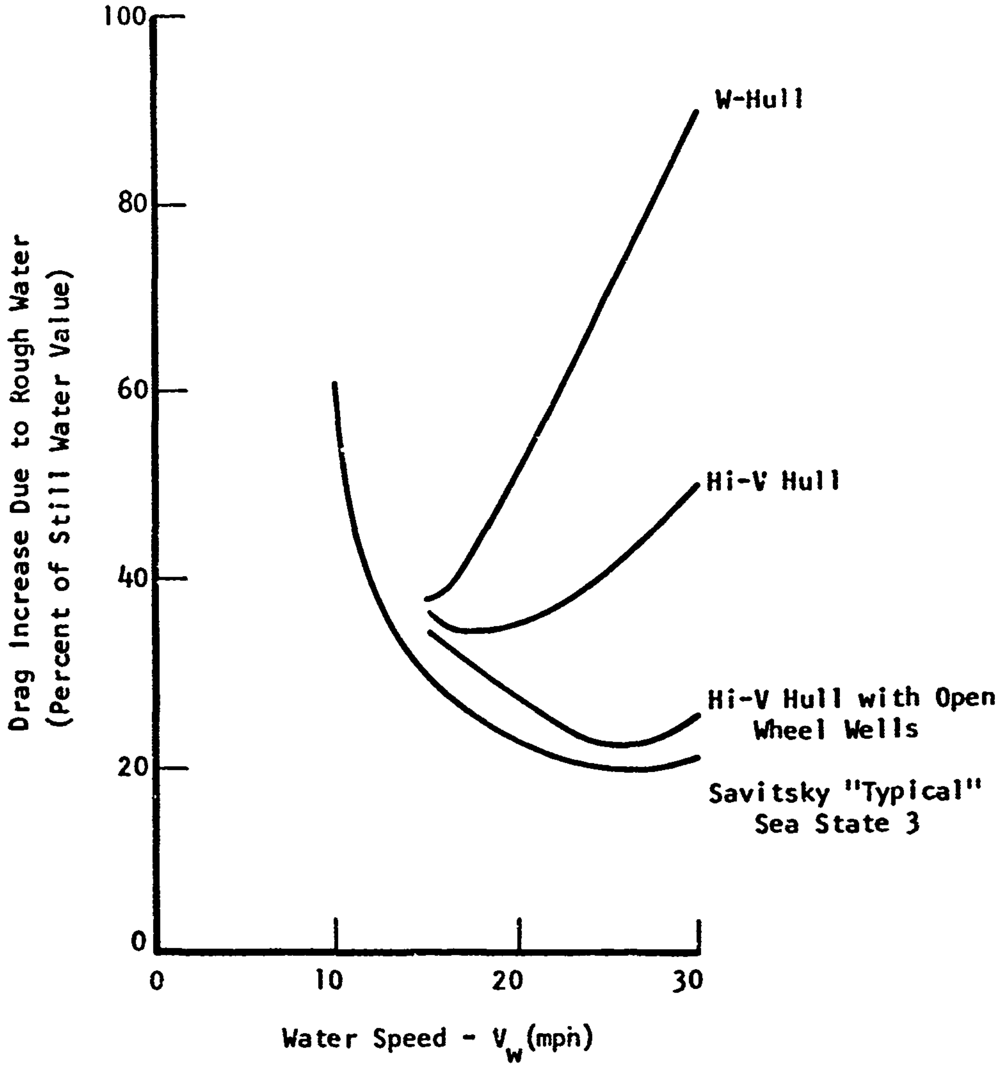


FIG. 47. INCREASED DRAG EXPERIENCED IN 3'x60' REGULAR HEAD SEAS, WHEN COMPARED TO STILL WATER DRAG[1,79]

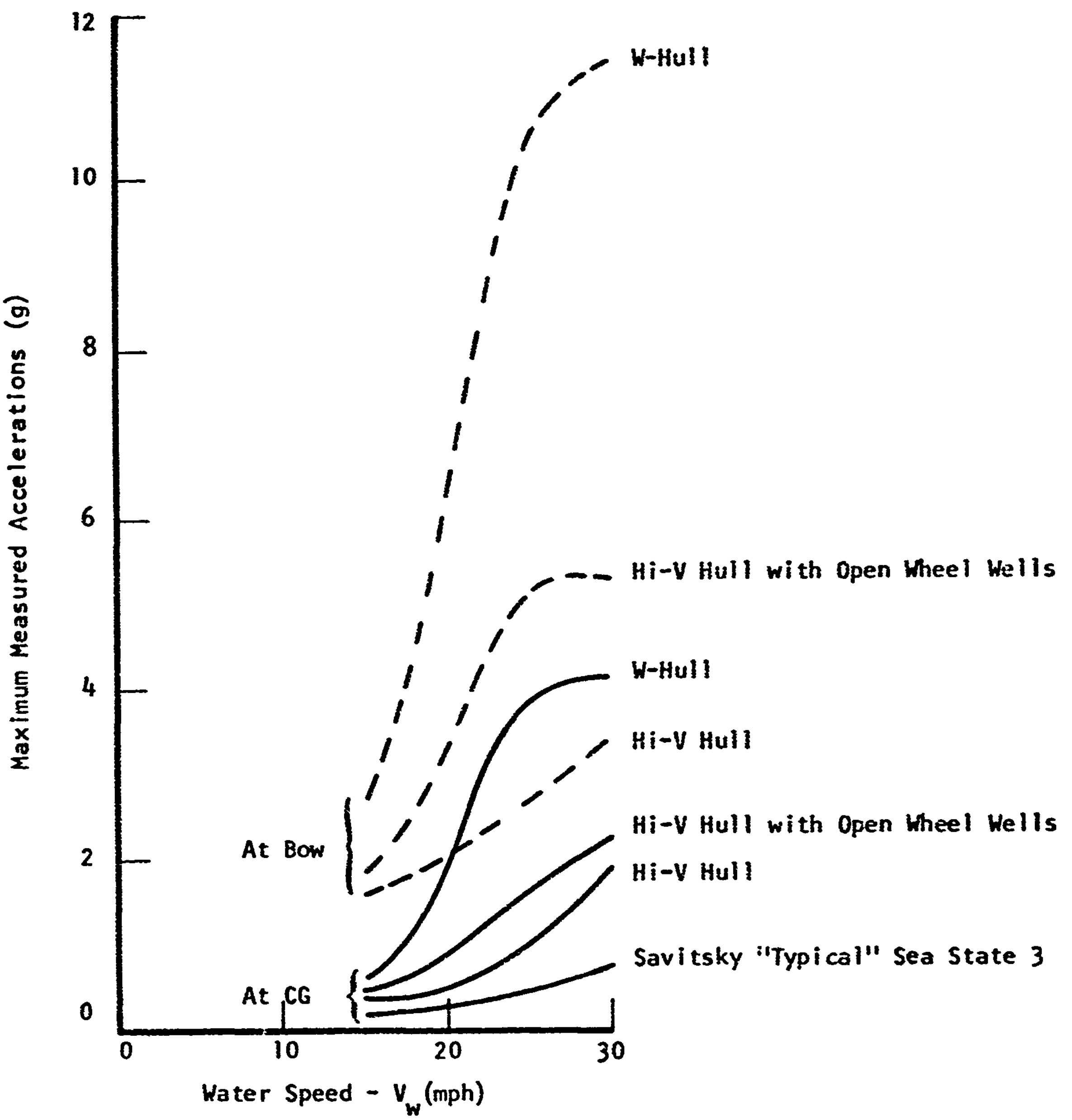


FIG. 48. MAXIMUM ACCELERATIONS MEASURED IN 3'x60' REGULAR HEAD SEAS[1,79]

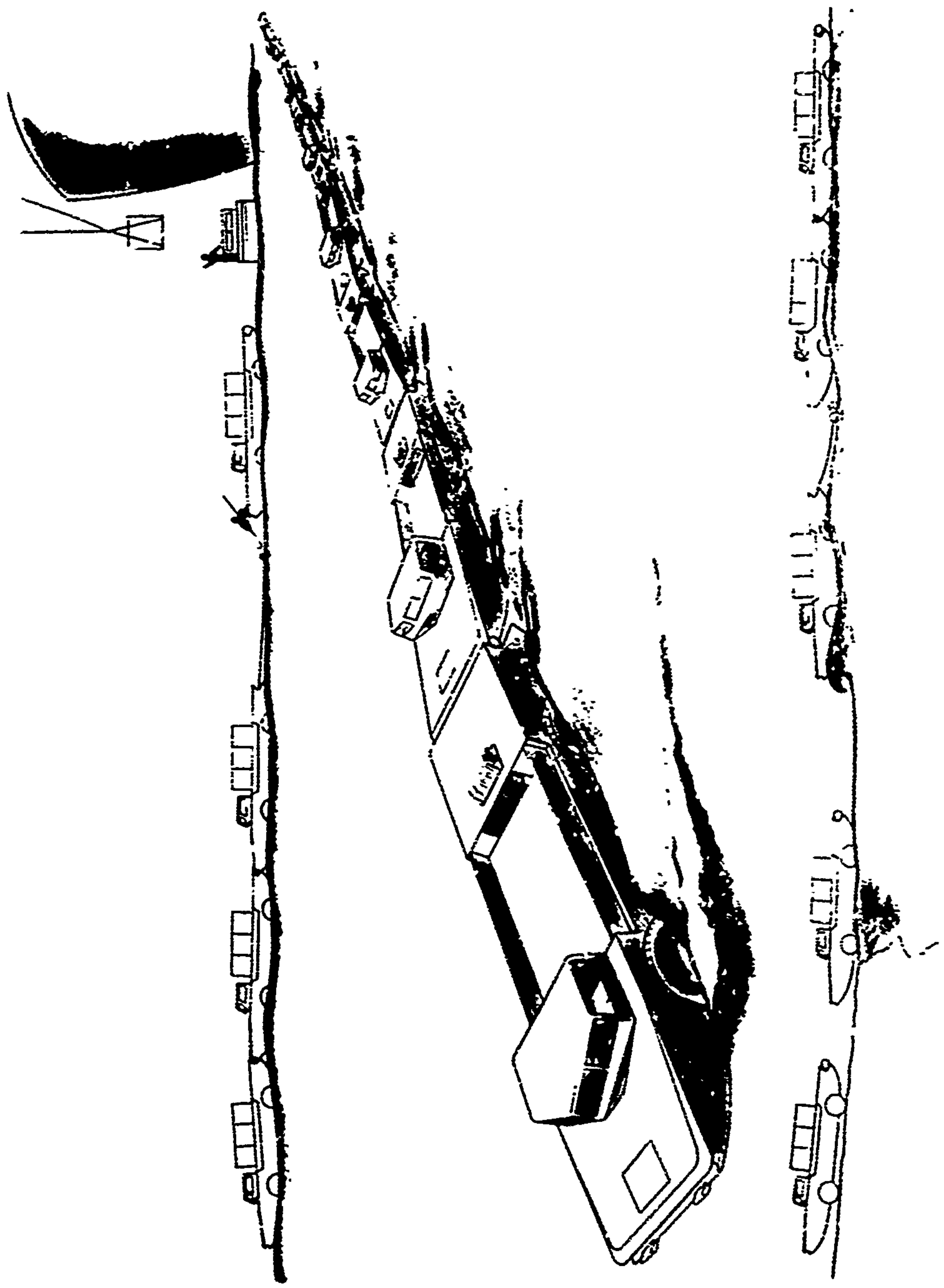


FIG. 49. CONCEPT NO. 6 - SEA SERPENT

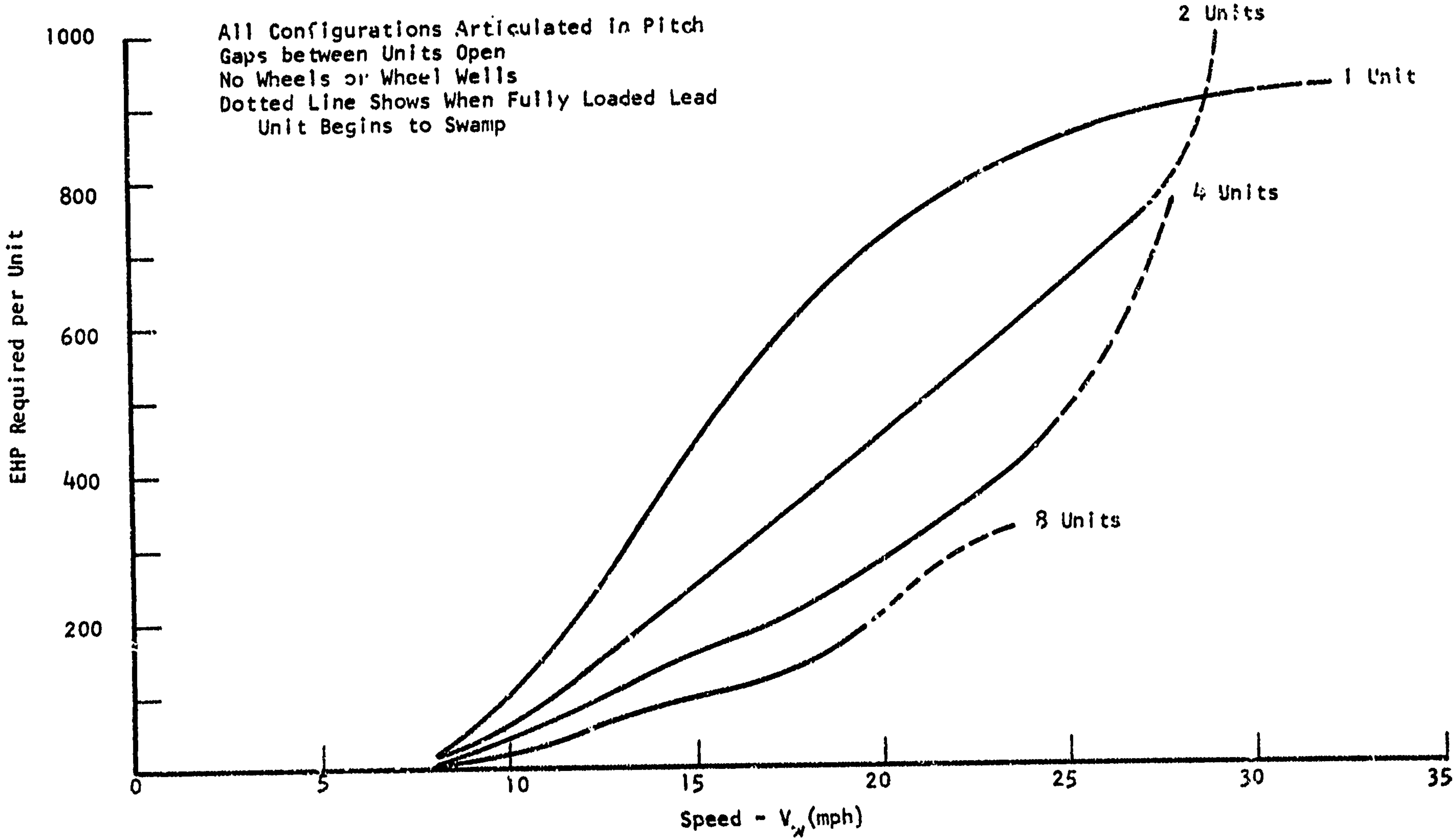


FIG. 50. POWER REQUIRED PER UNIT FOR VARIOUS SEA SERPENT CONFIGURATIONS[1]

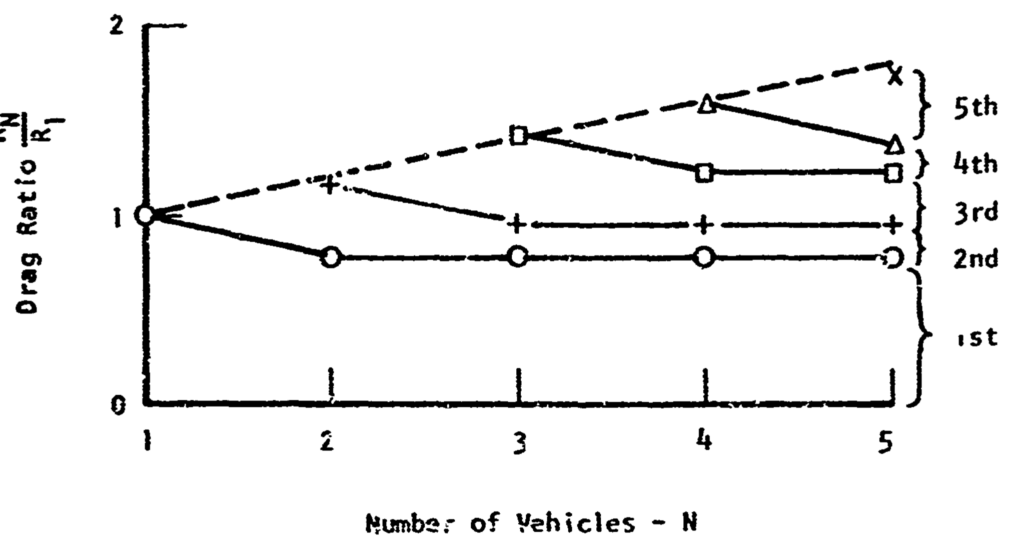


FIG. 51. DRAG BREAKDOWN BY VEHICLE LOCATION IN A TRAIN OF LVTP-5 LANDING CRAFT AT ABOUT 7 MPH[58]

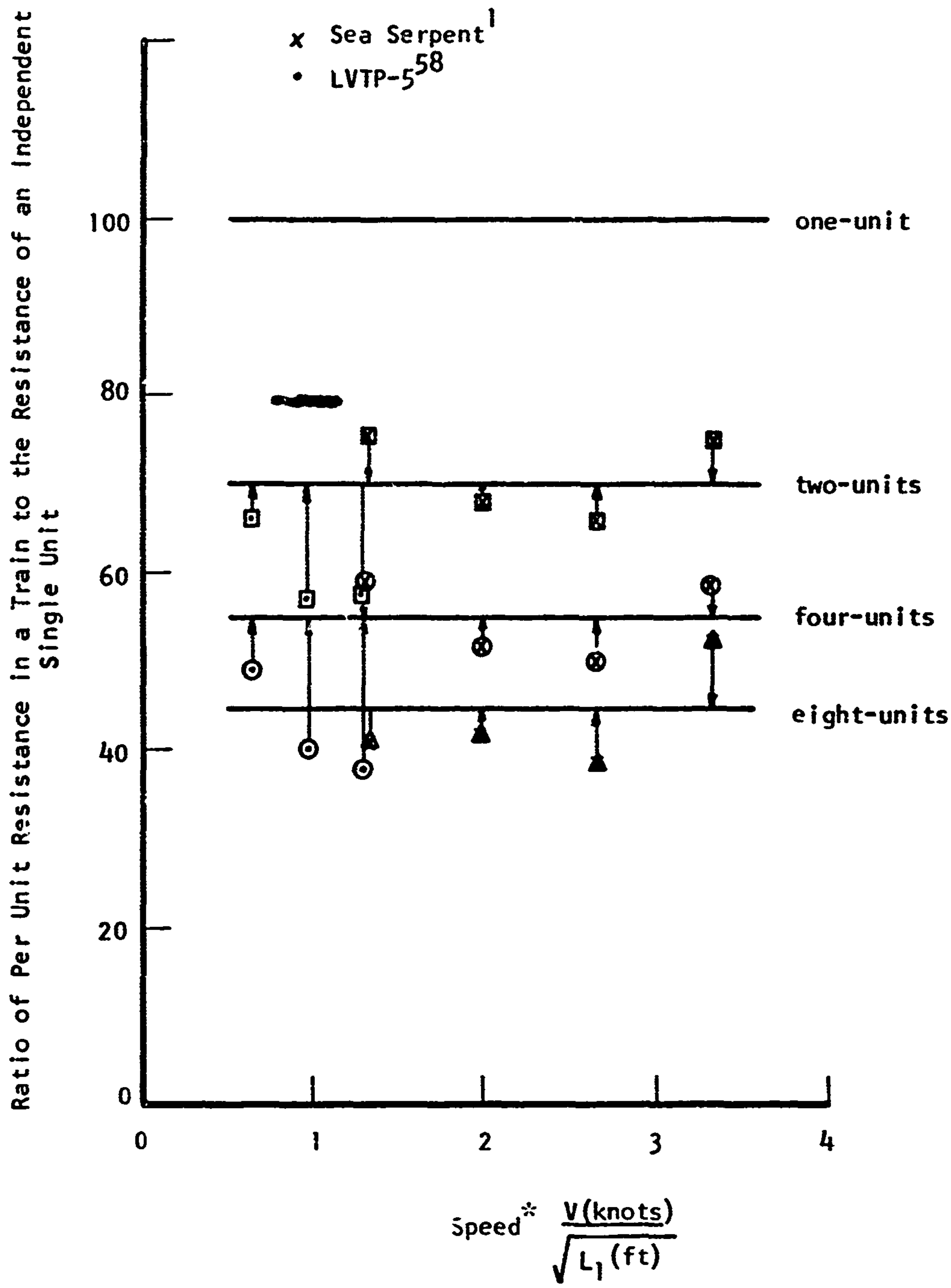


FIG. 52: PER UNIT RESISTANCE OF VEHICLES IN A TRAIN CONFIGURATION

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*L_1 = Length of a Single Unit

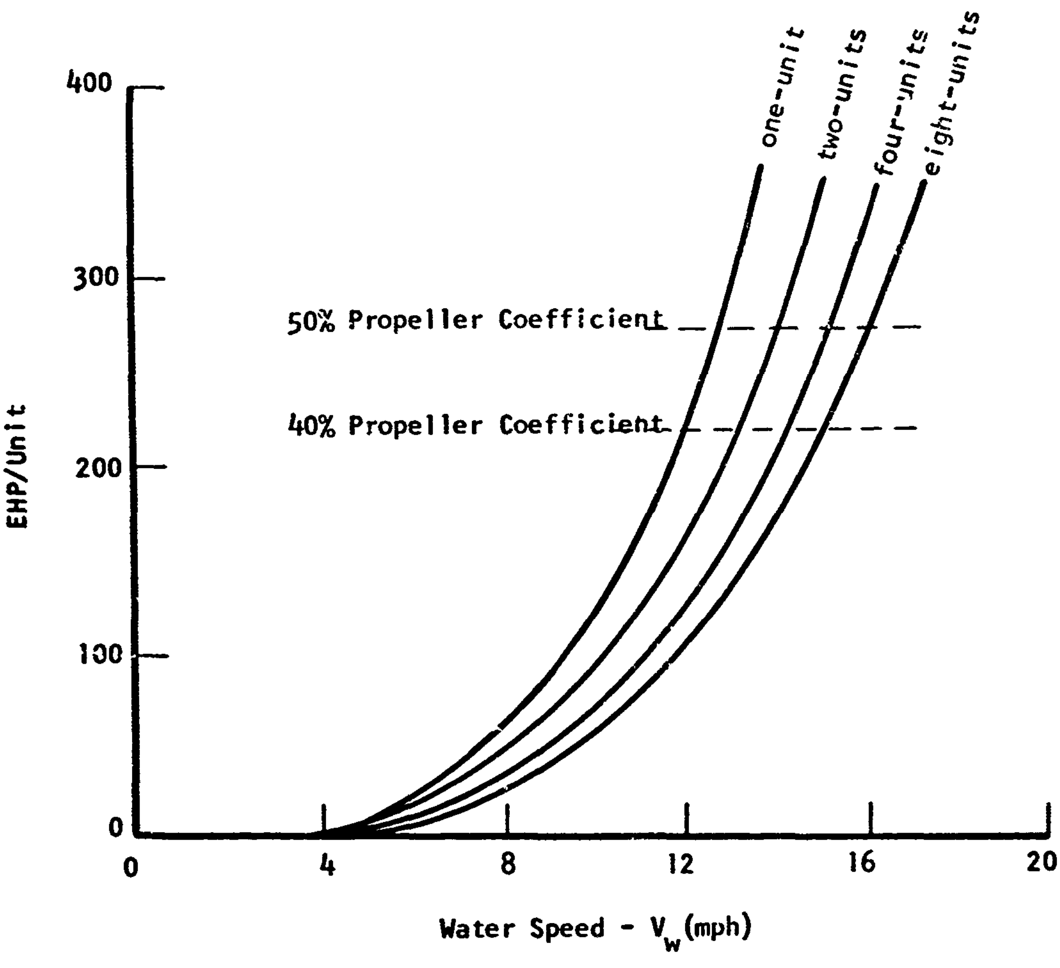


FIG. 53. ESTIMATED SEA SERPENT SMOOTH WATER EHP

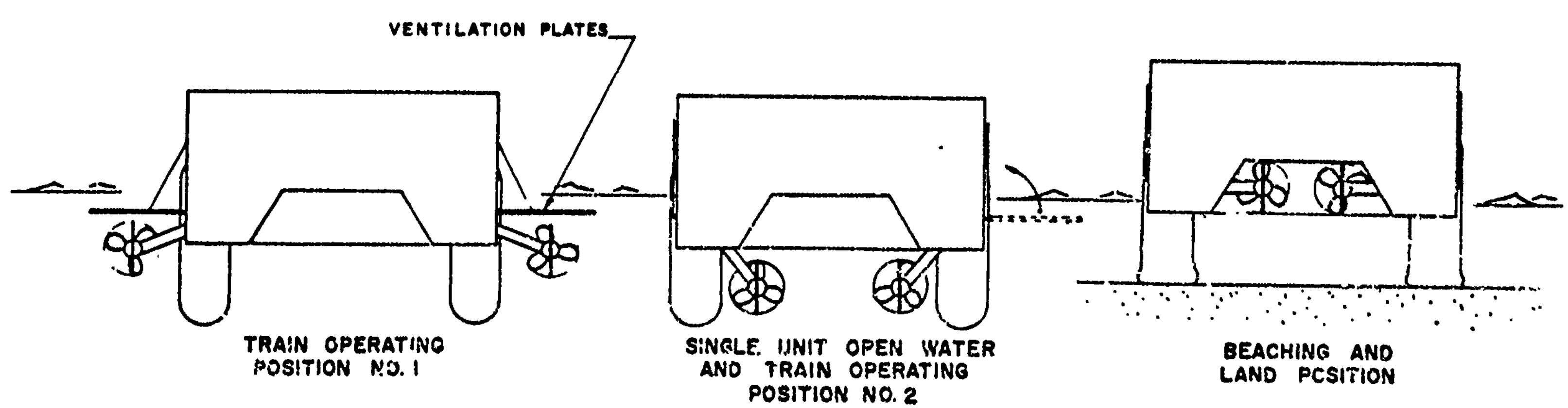
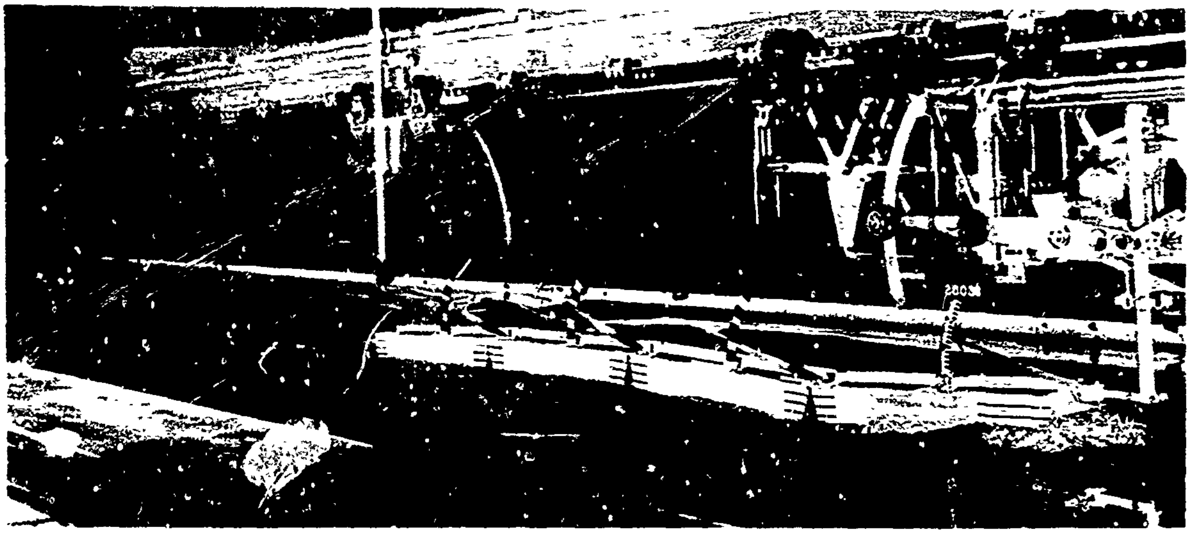
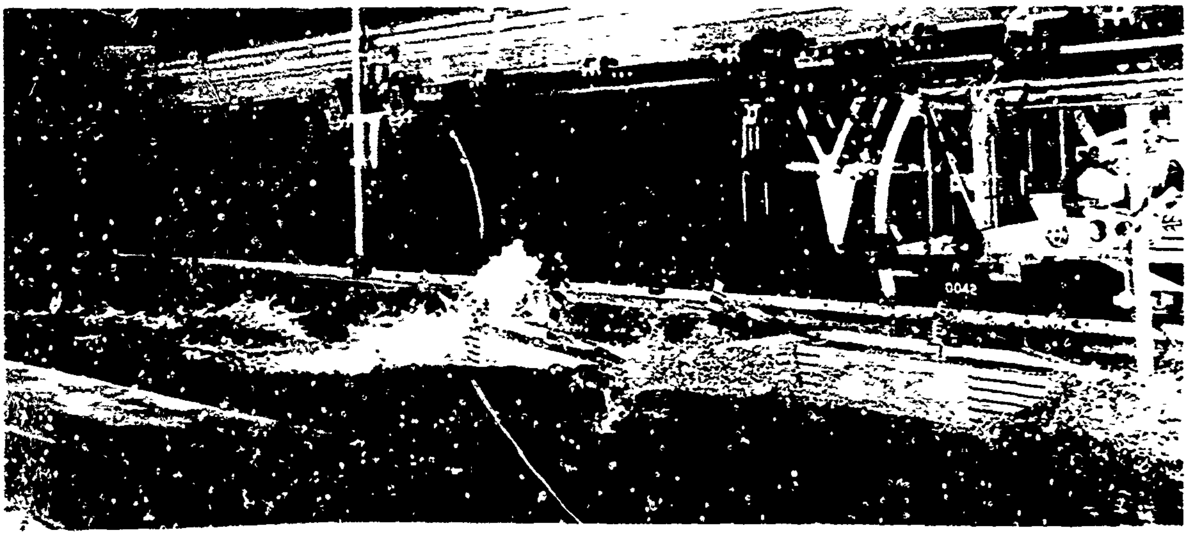


FIG. 54. CONCEPT NO. 6 - RIGHT-ANGLE PROPELLER DRIVE AND RETRACTION SCHEME AS APPLIED TO THE SEA SERPENT CONCEPT



a. Four Units Without Wheels at 12.2 Knots, 380 EHP



b. Four Units Without Wheels at 27.8 Knots, 3100 EHP

FIG. 55. CONCEPT NO. 6 - PHOTOGRAPHS OF MODEL TESTS OF THE SEA SERPENT AT 36,000 LBS PER UNIT DISPLACEMENT (ALL DATA ARE PROJECTED FULL-SCALE)

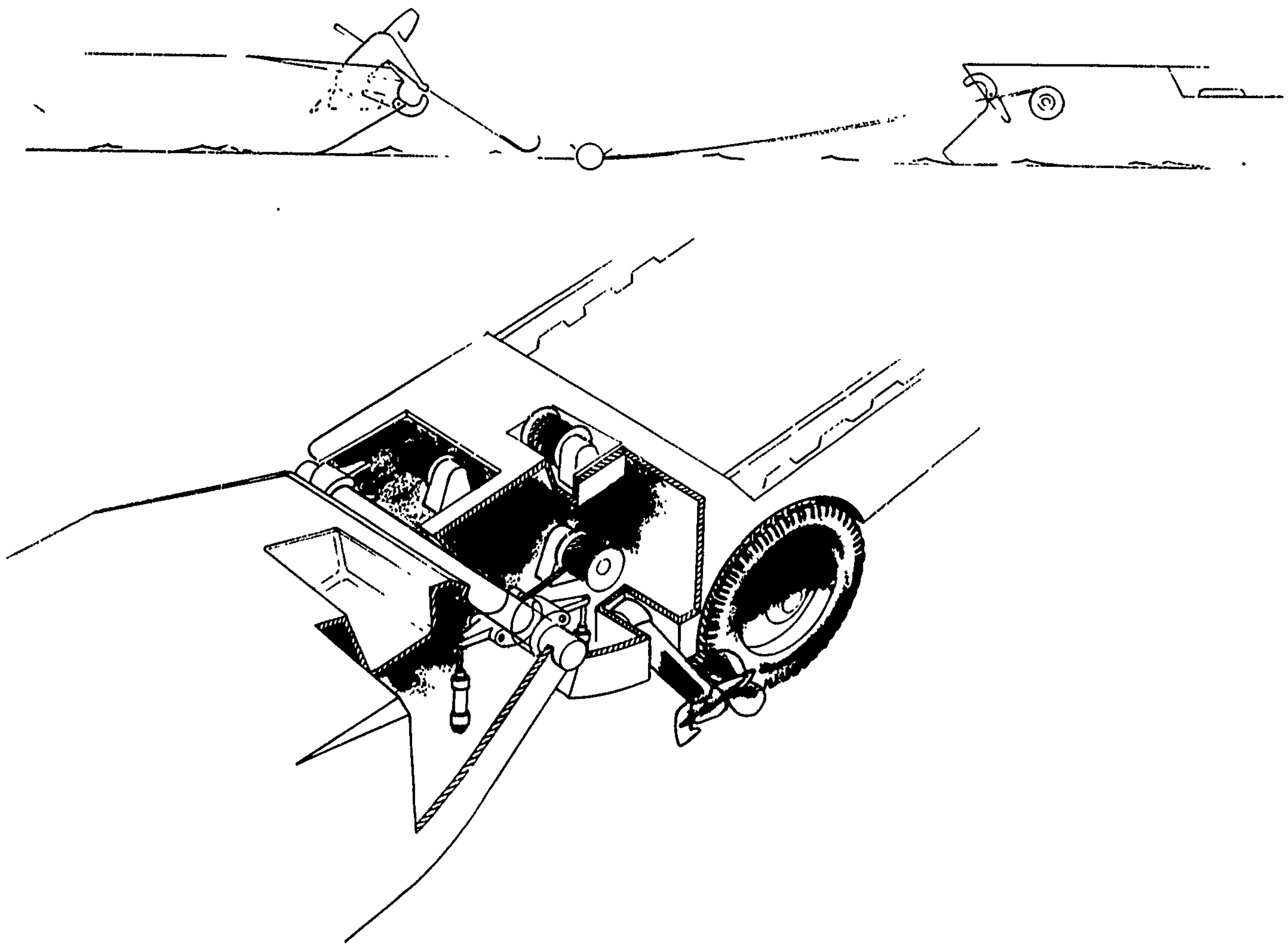


FIG. 56. CONCEPT NO. 6 - COUPLING SCHEMATIC OF SEA SERPENT

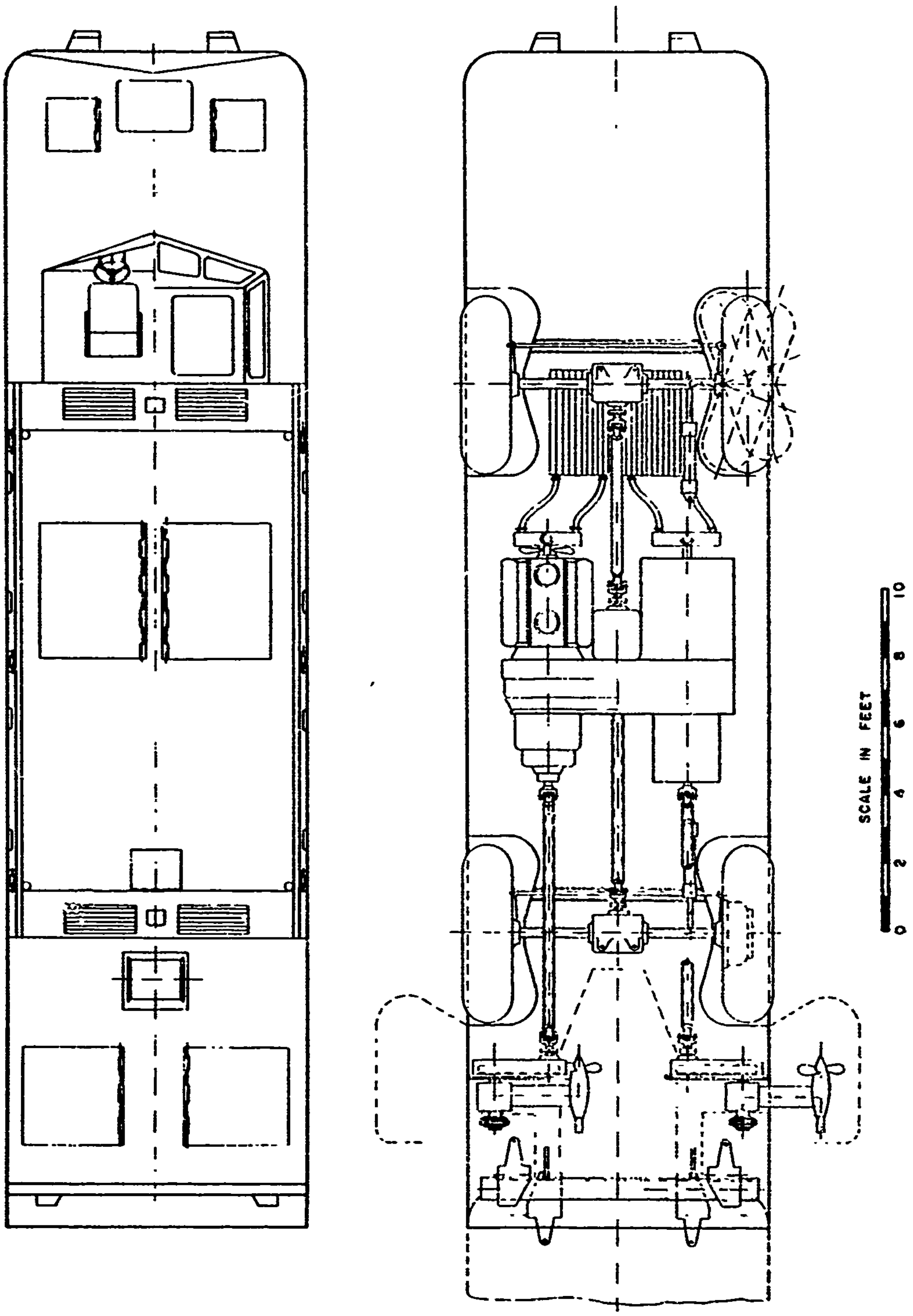


FIG. 57a. CONCEPT NO. 6 - LAYOUT DRAWINGS: PLAN VIEWS

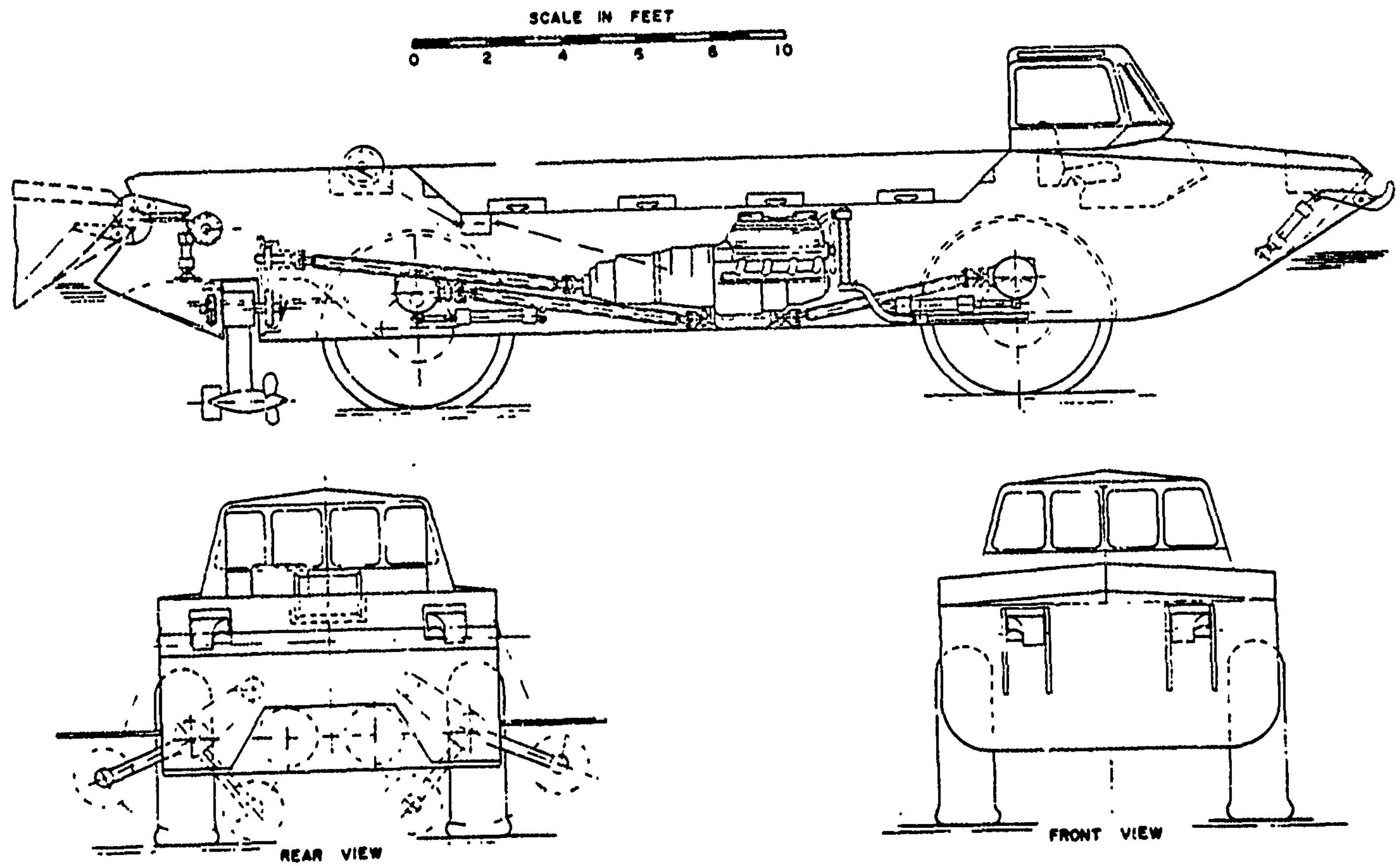


FIG. 57b. CONCEPT NO. 6 - LAYOUT DRAWINGS: STARBOARD, FRONT AND REAR VIEWS

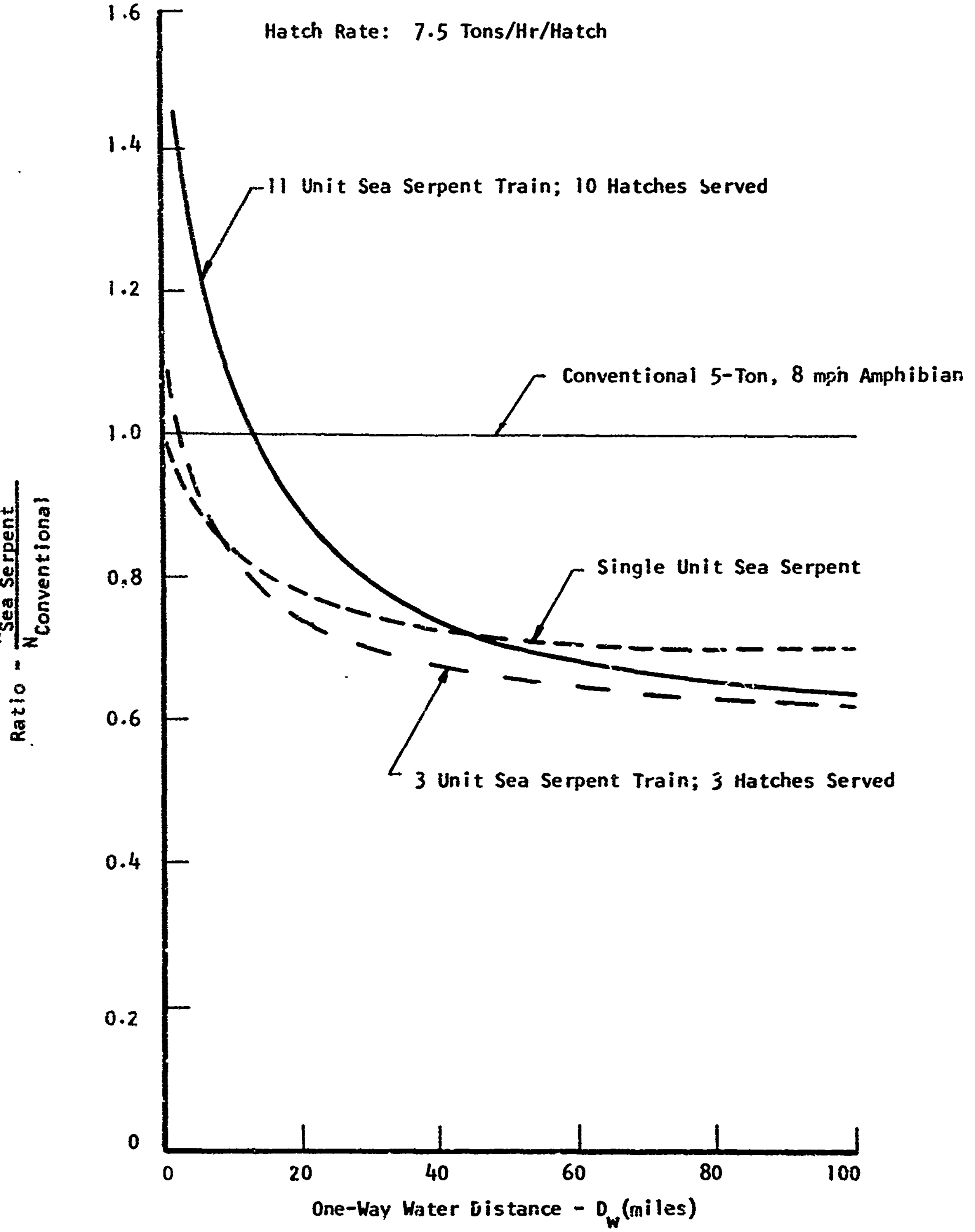


FIG. 58. REDUCTION IN THE REQUIRED NUMBER OF CONVENTIONAL 5-TON, 8 MPH AMPHIBIANS ACHIEVED BY USE OF THE SEA SERPENT

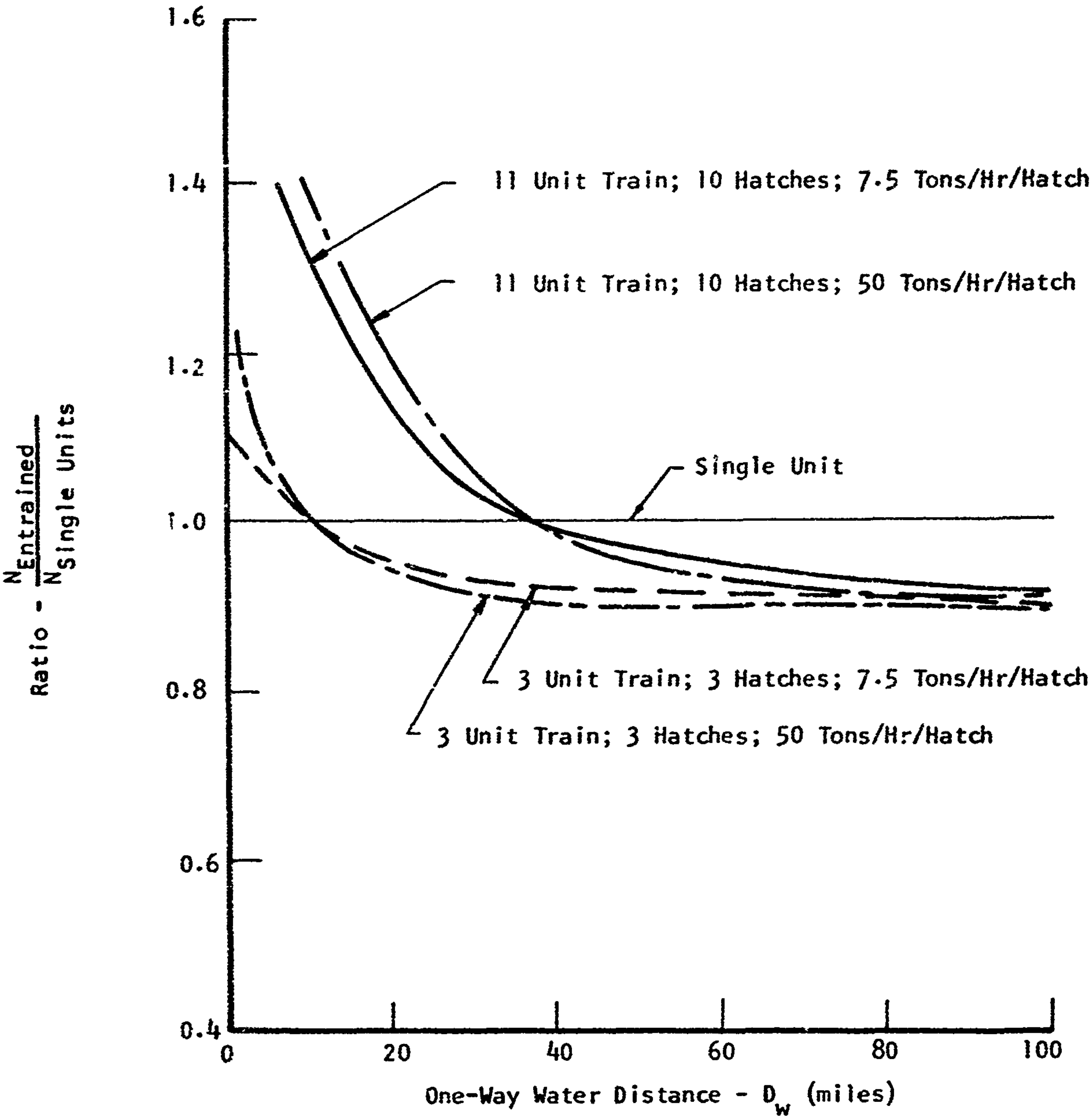


FIG. 59. REDUCTION IN THE REQUIRED NUMBER OF SEA SERPENTS ACHIEVED BY ENTRAINING

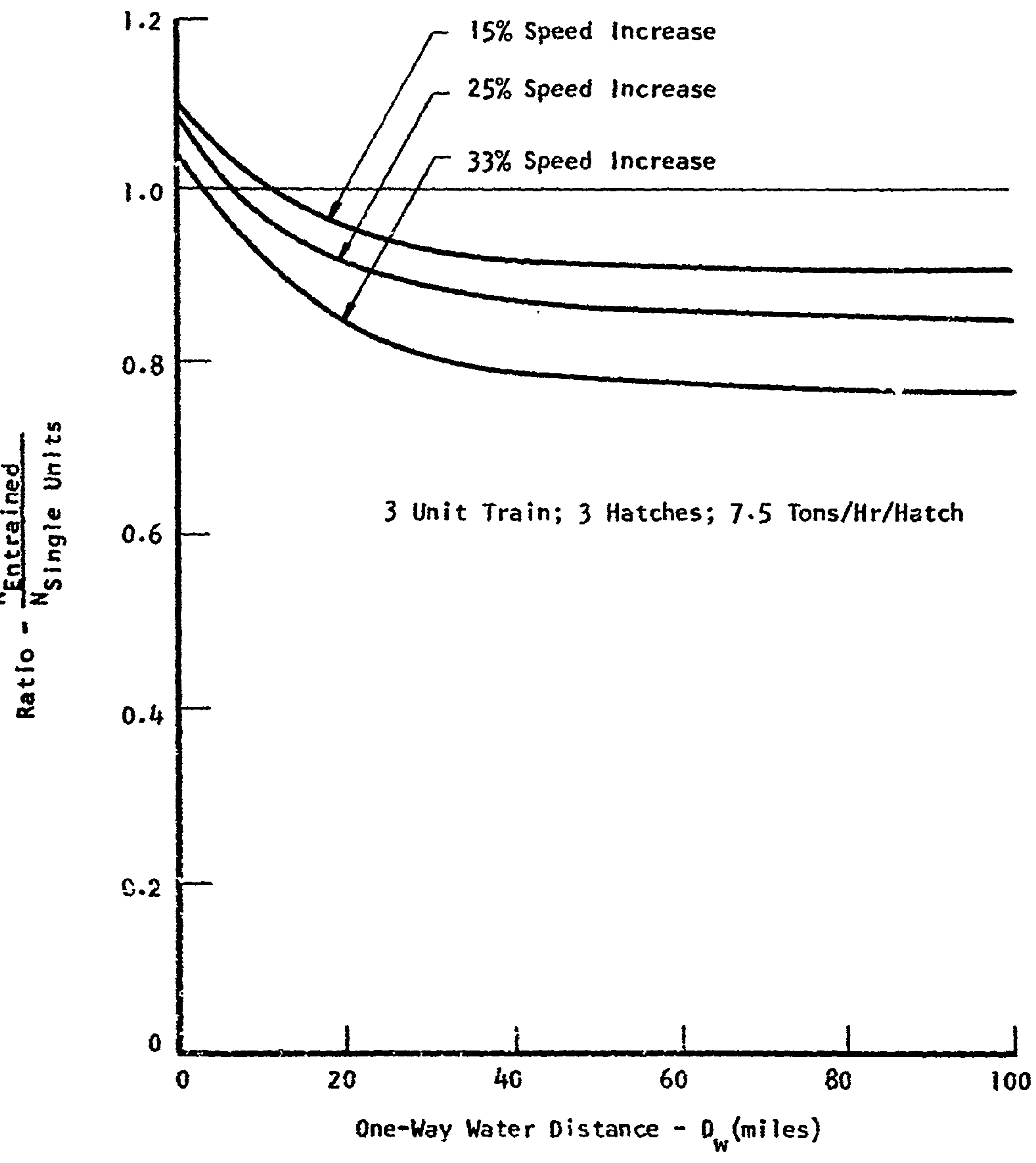


FIG. 60. REDUCTION IN THE REQUIRED NUMBER OF SEA SERPENTS ACHIEVED BY THE INCREASED SPEED OBTAINABLE BY ENTRAINING

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UNCLASSIFIED

Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Davidson Laboratory, Stevens Institute of Technology	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP

3. REPORT TITLE

HIGH-SPEED WHEELED AMPHIBIANS, A CONCEPT STUDY

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Final Report

5. AUTHOR(S) (First name, middle initial, last name)

Clifford J. Nuttall, Jr. and Irmin O. Kamm

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
August 1969	89 + 60 figs.	84

8a. CONTRACT OR GRANT NO. Nonr 263(69) b. PROJECT NO. c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) R-726-1 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)

10. DISTRIBUTION STATEMENT

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11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Office of Naval Research Department of the Navy

13. ABSTRACT

The results of a concept design study of high-speed wheeled amphibious vehicles originally conducted in 1956-1959 are now presented as Volume I of a two-volume study.

The basic technical and operational problems of high water speed are examined.

Six basic concepts are presented to exemplify potential mechanical solutions. Five concepts are planing hull types with retractable wheels; one example is based on the vehicle train concept.

The conclusions and recommendations which are made consider the decade elapsed since the original study.

DD FORM 1 NOV 65 1473 (PAGE 1)

S/N 0101-807-6811

UNCLASSIFIED

Security Classification

A-31408

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Amphibians, Wheeled						
Ship-to-Shore Operation						
Planing Hulls						
Coupled Vehicles						

DAVIDSON LABORATORY, Stevens Institute of Technology
Hoboken, N. J.

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