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Large High-Speed Surface Effect Ship Technology

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SYNOPSIS

The Surface Effect Ship Program currently underway in the US Navy is structured to achieve large displacement (2000-4000 tons), high speed (80-100 knots) ships in the next decade. The present paper outlines some of the key technical areas requiring development. Data is presented from the two current 100 ton displacement SES, namely, the waterjet-propelled SES-100A and the waterscrew propelled SES-100B and from other sources to demonstrate that progress is being made and that challenging work remains to be done. Specific attention is given to the seakeeping and ride quality of SES at high speed in rough seas, and approaches for solution are discussed. The important topics of seal characteristics and inlet design are also discussed to suggest avenues for improvement and the need for development.

Introduction

The amphibious hovercraft is continuing to make advances in high-speed transportation around the world and notably in Great Britain where it all began with Sir Christopher Cockerell's experiments in 1955. By ship standards all of the craft are small, varying in size between 5-165 tons and operating over comparatively small distances (5-25 miles) but with greatly improved speeds (50-70 knots) compared to today's ocean liner.

American interests have by and large been geared toward the large-tonnage ocean-going form of air cushion ship known as the Surface Effect Ship (SES). These interests began with the US Maritime Administration programme in 1960 and accelerated with the joint programme between the US Navy and the Maritime Administration in 1966 with the formation of the Joint Surface Effect Ship Project Office (JSESPO). This office became a complete US Navy programme in 1970 under the cognisance of SESPO (PM-17) in the Naval Material Command when the US Maritime Administration elected to defer their interests until the military impetus had developed the concept beyond the feasibility stage.

Technologists will recognise the problem of increasing size and speed to any form of transportation and this is certainly the case for the large high-speed SES. Both military and commercial mission application studies have shown that, to be cost-effective, the ocean-going SES would have to be in the 2,000-4,000 ton displacement class and have a speed capability approaching 100 knots. These studies prompted President Lyndon Johnson in his transportation message to the US Congress in 1966 to make projections for the 100 knot Navy.

The form of the SES is still evolving but one particular characteristic has influenced the configuration to a large degree, and that is the form of propulsion. Over 90% of the air-cushion-supported craft operating today are propelled by air propulsion and most commonly the air screw. When one projects this type of craft to the large size an unworkable scheme evolves. Cockerell recognised this in 1963 (ref. 1) with his designation of "the propulsion problem" where he noted that a 3,000 ton amphibious hovercraft would need 75 separate air screws, each 20 ft diameter to propel the craft at 100 knots! While this was not the initial reason, this problem has prompted the large high-speed SES to take the form utilising hydrodynamic propulsive means, either waterjet or waterscrew. Water propulsion can either mean extending a propulsive pod below an otherwise amphibious hovercraft, as exemplified by the Vosper-Thornycroft VT-1 or it can require a change in concept by departing from the familiar peripheral skirted craft and utilising catamaran hulls to house the propulsion units.

In addition to the advantage of handling hydrodynamic propulsive means, the catamaran or sidehull SES has other advantages worth exploring. The sealing of the air cushion by sidehulls (with flexible seals fore and aft) clearly reduces the lift power requirements but at the expense of propulsive power to overcome sidehull hydrodynamic resistance. For certain geometries, speeds and other configurational parameters there are lower power expenditures for the sidehull SES than for the amphibious form of air cushion craft. Stability can also be improved by taking advantage of the hydrodynamic shaping of the sidehull.

Extensive work has been done in the development of high-speed SES and the two most prominent examples are the two 100 ton class SES developed expressly for the purpose of determining the feasibility of the high-speed sidehull SES concept. One craft is the waterjet propelled SES-100A built by Aerojet-General Corporation, Tacoma, Washington, USA (Fig.1) and the other, the supercavitating semi-submerged water-propeller propelled SES-100B built by Bell Aerospace Corporation, New Orleans, Louisiana, USA (Fig.2).

Both craft were designed and built for the US Navy SESPO as part of the overall programme to develop a

100 knot capability at sea. These test craft have been collecting the necessary technological data since the start of their test programme in 1972 and have successfully achieved stable speeds in excess of 75 knots.

Such a new form of air cushion ship, however, has required new technology that will continue to challenge designers for years to come. Moving sidehulls through the water at supercavitating speeds and at Froude numbers greater than 2.0 and cavitation numbers less than 0.10 has strained the technology. The design of the sidehulls to minimise resistance and susceptibility to cavitating flow is a technology challenge explored here. At high speeds, the stability boundaries close in, requiring a more explicit understanding of the "plough-in" experienced with certain skirt designs. The ride quality during high-speed encounter with rough seas has threatened to nullify the speed potential advantage of air cushion ships. The life of seals beating over waves at high speeds reduces rapidly with speed requiring material developments.

Each of these items require improvements in the technology base for large high-speed SES to become a reality in the near future. The present paper addresses each of these items and suggests avenues of approach for their solution.

Ride quality

One of the problems with the ride quality of the air cushion ship is the disparity between the ride quality criteria as suggested by the various researchers and agencies. A systematic study of the habitability (ride quality) acceleration criteria had been conducted by many groups of researchers over the years and summarised in the US Air Force Systems Command Manual (ref. 2). Such references have presented human tolerance requirements in terms of acceleration, amplitude, frequency and type of task that the human being is asked to do while being subjected to such an environment. The majority of the research has been in connection with programmes on space flight, low-level high-speed aircraft, automobiles, trucks and high speed trains. Virtually no quantitative data exists for use as criteria for long-duration cruise of SES or hovercraft. Data would be difficult to obtain because all existing hovercraft have very short run times in the order of minutes where high acceleration levels can be tolerated without impairment or major discomfort.

In spite of lack of definitive data, SES criteria were established to provide measurable guidelines for design use. These data designated SES-HQ-1 criteria are shown in Fig.3 together with the recent International Standards Organisation (ISO) limits (ref. 3). General agreement is seen between the criteria for the 25 to 30 minute duration but an order of magnitude difference appears for the long term (24 hour) duration.

Also included on Fig.3 are the Wesleyan University study (1945) results for possible motion sickness at frequencies around 0.25 Hz. This same study indicated that motion sickness would more probably occur at the same frequencies but at acceleration levels approximately 0.15*g* above those shown. In addition to motion sickness the efficiency of the crew would be impaired significantly by resonance of human organs at certain frequencies and acceleration levels. Visual acuity limits are indicated on Fig.3 that would hamper operation of controls and displays. These are shown according to MIL-STD-1472A (1968).

The area of concern for the SES is in the range of encounter frequencies of 0.20–2.0 Hz. It is in this region that most disagreement exists between the current SES criteria, issued by SESPO in 1968, and those proposed by ISO in 1973. At the typical SES cruise frequency of 1.0 Hz the criteria differ by an order of magnitude (0.10*g* RMS according to current design practice and 0.015*g* RMS recommended by ISO). It is interesting to note that commercial hydrofoil practice would recommend approximately 0.06*g* RMS for 24 hour tolerance. All of the data, also, have been obtained from analysis of air crew men vibrated sinusoidally. What are the true values to be used for designing for high-speed operation in an irregular sea? Insufficient information is available on this subject.

The available data for today's air cushion craft, both amphibious and sidehull, cover a broad spectrum of sea conditions and some attempt has been made to normalise the data for design purposes and overview of the situation. Fig.4 shows the acceleration levels felt by crew and passengers.

This heave acceleration data covers more than a decade of hovercraft experience and a consistency appears in the data worth noting when plotted against a roughness parameter of average wave height divided by cushion depth. It is taken that the craft speed correspondingly decreases as the roughness parameter is increased. As the wave height approaches the height of the cushion (roughness parameter approaches 1.0), slamming influences the motion giving rise to high accelerations.

The subject of structural loading, the combined effect of slamming and craft motion in rough seas at high speed is a subject not well understood or documented. This fascinating technology, so important to the success of large light-weight high-speed SES, is not discussed here.

It is possible to predict the accelerations imparted to the craft by using various degrees of sophistication to the representation of the wave pumping action and the motion of the craft. A simple pumping action, assuming that the compression of the cushion is isothermal, ie,

$$(P_A + p_c)V_c = \text{constant} \qquad (1)$$

where P_A is the ambient air pressure, 2116 lb/ft^2 and V_c is the cushion volume will yield the line labelled 'simple theory' on Fig.4 expressing heave acceleration as a function of the sea roughness parameter,

$$h_w/d_c = \text{wave height/cushion depth} \qquad (2)$$

This simple one-dimensional isothermal theory correctly predicts the trends but overestimates the actual experienced accelerations by an order of magnitude. The actual situation is a rather complicated combination of total craft motion, frequency of encounter, fan characteristics, seal motion, pressure and pressure alleviation mechanisms notwithstanding the wave surface geometry and its interaction with the craft. For the sake of brevity the complete theory is not given here. The overlay of such theoretical solutions incorporating the particular features of each craft shown would detract from the presentation of the data. Despite the varied craft designs from the amphibious SR.N5 to the sidehull XR-3 it is to be noted that a common trend exists that indicates that rather high acceleration levels are being experienced which, although much improved over displacement or planing craft of the same size and speed, nevertheless represent habitability problems for long-range duration cruise.

Of particular interest are the acceleration levels attained by the two 100 ton displacement class SES, the SES-100A and the SES-100B, in rough water operation.

The pertinent geometric characteristics of the SES-100A and SES-100B are given in Table 1, viz:

Table 1 SES 100A and B Characteristics

	SES 100A	*SES 100B*
Nominal gross weight	100 short tons	100 short tons
Length overall	81 ft 11 in	77 ft 8½ in
Breadth overall	41 ft 11 in	35 ft 0 in
Cushion L/B	1.95	2.0
Cushion density, P_c/L	1.40	1.60
Cusion pressure	95 lb/ft^2	100 lb/ft^2
Sidehull	High deadrise, 60°	Low deadrise, 30°

The SES-100A is powered by four AVCO/Lycoming TF35 gas turbine engines (3750 bhp maximum) driving two waterjet pumps and three axial flow fans in an integrated power package. The SES-100B is propelled by three Pratt & Whitney FT12A-6 gas turbine engines (4500 bhp maximum) driving two semi-submerged supercavitating water propellers. A separate lift system is comprised of three United Aircraft of Canada ST6J-70 gas turbine engines (620 bhp maximum) driving eight centrifugal fans.

The heave acceleration data taken from the full scale trials data of the SES 100A and SES 100B are shown on Fig.4. The data for the SES 100A include both with and without the ride control system activated. The ride control system is an active cushion-venting system with the vents activated by servo valves signalled by an accelerometer mounted in the deckhouse. Without ride control (heave attenuation) it will be noticed that the 100 ton SES have a ride quality equal to today's commercial hovercraft such as the SR.N6 (Winchester class) and SR.N4 (Mountbatten class) hovercraft currently plying their trade in the Solent and the English Channel. It is generally recognised that the ride quality of hovercraft diminishes to unacceptable as the average wave height approaches the cushion depth, ie, the sea roughness parameter approaches 1.0 and it is encouraging to see the marked improvement in heave acceleration attenuation with the activation of the heave attenuation system (HAS) on the SES-100A.

These acceleration levels (on Fig.4) have been quoted for operation into head seas. Fig.5 shows a typical set of rough sea operation data for the SES 100A, selected for the case of almost pure heave motion (acceleration at the forward deckhouse approximately equal to the acceleration at the craft centre of gravity). The data are for the craft operating at a gross weight of 200,000 lb in 2-2½ ft average wave height seas (sea state 2) with a short wavelength of approximately 30 ft. The craft speed varied between 28-32 knots in these conditions with the HAS inactive and between 25-31 knots with the HAS active. For a simple heave attenuation system utilizing cushion venting without active fan blade or propulsion control, the loss of cushion flow causes deeper immersion of the sidehulls and a consequent forward speed drop at constant power setting. The data (Fig.5) show the expected fall-off in vertical acceleration level as the craft turns away from head sea to beam sea and, finally, following sea operation. The marked improvement in ride with activation of the HAS is clearly evident from Fig.5 with more than 50% reduction in acceleration levels shown. The problem still remaining, of course, is the amount of power expended to achieve this good ride. In a similar test run, for example, the SES-100A operated at 41 knots in 3 ft average waves (sea state 3) and experienced 0.39 g RMS heave acceleration. With the heave attenuation system activated, the heave acceleration was significantly reduced to 0.13 g RMS but the speed dropped to 35 knots. Since these results represent a 2000 ton class SES operating at speeds in excess of 60 knots in high sea states, these results are encouraging. Much work remains to be done, however, to determine optimum power utilization between lift and propulsion systems for heave attenuation.

In attempting to understand the dynamic behaviour of either the amphibious or sidehull SES, modelling techniques are used extensively. The acceleration levels (and hence motion) of models, however, depend on the proper scaling of the cushion total pressure ($P_A + p_c$) which for complete scaling would require a Froude scaling of the ambient air pressure P_A – ie, testing in reduced atmosphere facilities. Such facilities do not exist for SES use and a more complex interpretation has to be placed on the model results. This problem was first pointed out, to the author's knowledge, by Dr. J. Breslin of Stevens Institute of Technology, USA, (ref. 4). By way of illustration, the acceleration levels expressed in normalised form can be shown to be

$$\frac{1}{g}\cdot\frac{d^2z}{dt^2}\cdot\frac{1}{(h_w/d_c)} = F\left(\frac{2\pi U}{\lambda}\sqrt{\frac{d_c}{g}}\,;\,\frac{\partial Q}{\partial P_c}\right) \tag{3}$$

where d^2z/dt^2 is the vertical acceleration, g is the gravitational acceleration and h_w/d_c is scaled wave or sea roughness parameter shown on Fig.4; U is the craft forward speed; λ is the wavelength and d_c is the cushion depth or clearance height of the hard structure above the mean water level. The conductance of the cushion system, expressed as $\partial Q/\partial P_c$ represents the ease with which changes of flow (Q) occur in the cushion for a given change in cushion pressure (P_c) and allowing for cushion system effects is the inverse of the cushion fan slope. Again, assuming sinusoidal waves and linearisation are valid then it is possible to construct curves such as Fig.6 for a typical large SES (ref. 5).

The full-scale acceleration response curves have been drawn for three typical values of the parameter $\frac{\partial Q}{\partial p}$. Scaling the model by Froude scaling only would produce a set of acceleration response curves as shown. The different nature of the full-scale and model scale curves is a direct result of scaling only P_c in the term $(P_A + P_c)$, which is a major term in the equation of heave dynamics. Since the atmospheric pressure (P_A) is not scaled and has a large numerical value (2116 lb/ft^2) compared to the value of cushion pressures discussed (50-300 lb/ft^2) this has a significant effect on the model motion and acceleration. For large values (greater than 5.0) of the non-dimensional encounter frequency,

$$\omega_e = \frac{2\pi U}{\lambda}\sqrt{\frac{d_c}{g}} \tag{4}$$

the model would overpredict the acceleration levels of the full scale ship, while for small values ($\omega_e < 5$) the model scale underpredicts the acceleration levels. This result appears to be true for all values of the cushion conductance, $\frac{\partial Q}{\partial p}$. Unfortunately, for large SES of the order 500-4,000 tons, operating in sea states 4-6 will all have values of the non-dimensional frequency less than 5.0, where the model would predict optimistic values of heave acceleration in which case care must be taken to properly interpret model test values of heave acceleration.

This problem of model motion scaling is not limited to SES but applies to the amphibious ACV or hovercraft, and some interesting comparisons between predicted full scale response and Froude scaled model tests for a fully skirted craft may be found in ref. 6.

It is to be emphasised that to achieve acceptable ride quality in high-speed SES it will be necessary to incorporate heave attenuation systems that convert the basic steep pressure-flow characteristic ($\frac{\partial Q}{\partial P} < 400$) to a more generous pressure-flow characteristic ($\frac{\partial Q}{\partial P}$ approaching 1000). Such high values of cushion conductance must be obtained through active systems such as venting (described above) or throttling, either through variable pitch fans or other means of relieving the pressure variations as the waves sweep through the cushion. Accumulator systems have also been studied and under certain conditions may relieve the acceleration levels.

Performance

The large high-speed SES in the 2,000-10,000 ton displacement class, as envisioned by the USN and in the early sixties by the Maritime Administration, must be capable of operating efficiently at high speed or its very existence is challenged. The key performance parameter by which one can compare various means of transport is the transport efficiency which expresses the ratio of the work done by transporting a craft of displacement W at a certain speed V to the *total* power P expended; viz:

$$\text{Transport efficiency} = \frac{WV}{P} = \eta_p\left(\frac{L}{D}\right)_{EFF} \tag{5}$$

where W is in lb, V is ft/sec and P is the total brake horsepower expressed in ft lb/sec.

It has become common practice to discuss performance efficiency in terms of the effective lift-drag ratio,

$$\text{Effective lift-drag ratio, } \left(\frac{L}{D}\right)_{EFF} = \frac{W}{D + \frac{P_c Q}{V}} \tag{6}$$

where D is the resistance (lb), P_c is the cushion pressure (lb/ft^2) and Q is the cushion flow (ft^3/sec). This term is only strictly valid in its relation to the transport efficiency for $\eta_p/\eta_L = 1$ where η_p is the propulsion system efficiency and η_L is the lift system efficiency.

Without the effective drag term of the cushion flow it is usual to quote the lift drag ratio as,

$$\text{Lift drag ratio, } \frac{L}{D} = \frac{W}{D} \tag{7}$$

The performance efficiency expressed in the above terms for the SES-100A and SES-100B is shown on Fig.7 for calm sea conditions. The data for the SES-100B have been taken from ref. 7. The effect of the two basic different hull forms between the SES-100A and SES-100B can be seen from Fig.7.

The SES-100A has a catamaran hull form of low-resistance high-deadrise (60°) geometry requiring hydrodynamic stabilisation (skegs) and variable immersion with speed. The SES-100B also has a catamaran hull form but with a low-deadrise (30°) geometry obtaining hydrostatic stability without need of auxiliary hydrodynamic stabilisation. These differences result in the SES-100B having a lower minimum resistance (higher lift-drag ratio) than the SES-100A but a higher resistance (lower lift-drag ratio) at the maximum speed capability which is approximately 80 knots for both craft.

On the basis of the performance efficiency parameters discussed above it is seen that it is possible to achieve L/D of 20 and transport efficiencies of 10 by proper choice of hull form. These values occur at a Froude number of 1.40 which for large SES, say in the 2,000 to 4,000 ton displacement class, would correspond to 60 to 70 knots. It is still difficult to obtain high performance efficiency, however, with the configurations selected in the 100 ton displacement class since the values of transport efficiency $\left(\frac{WV}{P}\right)$ achieved are still comparable with todays' high speed hovercraft (eg, $\frac{WV}{P} \leqslant 5$ in the 60-80 knot speed regime; see ref. 8).

Optimum hull forms incorporating the best performance together with good seakeeping and good propulsion system installation (blisters, inlets, pods and appendages, etc) are still under exploration by SES designers. Indeed, to make any further improvements in performance efficiency may require radical departures from current practice. Fig.8 illustrates the problem confronting the SES designer.

Two basic parameters affecting the configuration (once the concept has been chosen, ie, amphibious, sidehull, etc) are the length-to-beam ratio (L/B) and the cushion density (P_c/L). Fig.8 shows an example for a given displacement ship showing how the conflicting constraints of speed, range, hump margin in rough seas, roll stability and ride quality all influence the ship design. The current state of the art on geometry gives L/B of 2 or less. The largest hovercraft in the world today, the SR.N4 (Mountbatten class) (ref. 9) has a beam of approximately 77 ft which if Froude-scaled to say 4000 ton displacement would require a beam in excess of 200 ft which is clearly unmanageable in most of the shipyards of the world and on many sea routes. Further, today's hovercraft are lightly loaded with cushion densities (P_c/L) less than 1.0. For the large high-speed SES to be a feasible vehicle it must change its basic form by increasing cushion density and L/B. As Fig.8 shows, however, increasing L/B (beyond 2) can decrease roll stability and increasing cushion density (beyond 1.6 of the current SES designs) decreases hump thrust margin. Fig. 9 illustrates some of these effects.

Since for the large SES the propulsion is most likely to be marine propulsion for several years to come then the characteristic cavitation boundaries of both supercavitating propellers and waterjets further compound the selection of L/B and P_c/L in order to maintain sufficient accelerating thrust margin.

Fig.8 further shows an example of the range curves as a function of L/B and P_c/L for a given displacement SES and how these parameters must be traded to optimise a ship for range. The geometric requirements for good ride quality are in conflict with the performance requirements. Good seakeeping characteristics would call for a long ship by hovercraft and SES standards and this influences the minimum craft size for operation at sea. Sea State 3 or higher occurs on most sea routes a large percentage of the time and the wavelength of the maximum energy wave in Sea State 3 is approximately 160 ft. Accordingly, to avoid violent heaving and/or pitching, large SES should avoid having a waterline length much less than 160 ft. for operation on these sea routes. Coupled with L/B and cushion height to beam ratio this helps set minimum size SES for trans-oceanic operation.

The impact of increasing these parameters on SES design is considerable in terms of payload carrying capability (high L/B increases structural weight fraction) and lift systems (high P_c/L may require multi-staging of fans to provide the necessary cushion pressures for large SES use).

Seals

The sealing of the cushion has received development to varying degrees of sophistication since the advent of the first skirt on hovercraft in 1960. By far the most developed skirt or seal is that used by British hovercraft, namely the 'bag and finger' type. Lesser developments, in terms of operational experience and design improvements, occur with the French jupe system and the American hinged seals and fingers used on some of the experimental sidehull SES.

Seal development at Aerojet-General Corporation has encompassed material research, laboratory testing and exploration of different seal concepts on the SES-100A to provide necessary operational data at high speed on a large scale. Fig.10 shows two of the seal concepts explored for the SES-100A, the deployable seal comprised basically of rod-stiffened rubberised fabric deployed by adjustable downstops and cushioned on pressurised rubberised fabric bags and the multi-membrane seal. The multi-membrane seal is a derivative of peri-cell concept comprised basically of a loop attached to the main structure and a series of discrete cells attached to the loop and main structure.

Each type of seal has its advantages and disadvantages. The deployable seal is simpler and lighter but tends to be less stable at nosedown attitudes than the multi-membrane. Tuck-in and plough-in (ref. 9) are still occurring on hovercraft and the problem becomes of concern at the high speeds of the SES (80 knots). The design of the multi-membrane seal is an approach to solving this problem.

Two problems that confront the SES designer today designing for the large displacement SES: strength and life of the seal and specifically life of the fingers or cells. Fig.11 shows the effect of speed on life based on laboratory tests and some confirmation from operational hovercraft.

Insufficient systematic testing of seal materials has been pursued at this point in time but a general trend is available to guide further developments. The current British hovercraft use neoprene-coated nylon fabrics at weights up to 140 oz/yd^2 (for the SR.N4) while the two 100 ton SES employ PVC-nitrile coated nylon fabric at 70 oz/yd^2 weights. Both materials have a tensile strength capability approaching 1000 lb/in. The large variety of material choices and characteristics such as weave form, bonding and coating adhesion, thickness and ply make it difficult to present a simple story for non-metallic seals. From laboratory tests it can be shown that all coated fabrics lose life, either by abrasive wear or delamination or other failure modes, at a logarithmic rate with speed. It is also shown that, up to a point, increasing the weight (usually by coating thickness) for a given fabric increases its life. Where that point is, has not been satisfactorily determined in a systematic manner for a series of materials and designs.

Further, correlation between full-scale operational data and laboratory tests has not been encouraging. Laboratory testing using small material samples flagellating under the action of impinging water jets, with the jet flowing at speeds up to 100 knots, is the main technique used. Other test methods include the addition of air cushion pressure action on the backside of the flagellating material sample in an attempt to simulate operational conditions. Correlation between the two test methods is inconsistent. For coated fabrics Fig.11 records life at the point when the coating has cracked to the fabric such that the crazed surface resembles the sidewall of an old tyre. This is usually a stable condition for several hours (on the test stand) before total failure occurs through delamination and abrasion. Curiously enough this agrees with the life recorded for operational craft when the entire material has 'worn' such that the fingers have to be replaced. Current British-practice utilises neoprene-coated nylon fabrics of differing weights. The Vosper-Thornycroft data record life of 200-800 hours for 40 oz/yd^2 material when operating up to 40 knots *over water* while data from the SR.N4 operations record life of 100-450 hours for 140 oz/yd^2 material when operating up to 50 knots *over water and sandy beaches.* Which was the dominant effect? Speed, material weight, size or sand abrasion? One feature common to all operational and laboratory data, however, is that wear increases dramatically with speed and the 80-100 knot speed goals for SES places a demand on the material supplier. The problem is compounded as the SES become larger because of the strength requirements of the basic fabric. The hoop tensions increase by the square of the scale factor (Froude scaling) such that today's materials of 1000 lb/in tensile strength become 5,000-9,000 lb/in for tomorrow's large high speed SES. New core fabrics are being explored and the tire building technology will greatly aid research into SES materials. Likely candidates are nylon ply, boron, steel cord and aromatic organic fibers (Reference 10). Early wear test results on some of these new materials are encouraging and new seal configurations are evolving to minimize strength reguirements for operational SES.

Propulsion and inlets

It was mentioned earlier that large-tonnage high-speed SES in the foreseeable future would require hydrodynamic propulsive means to become an economic and feasible ship. Both the SES-100A and the SES-100B were designed to optimise waterjet and waterscrew propulsion in the high-speed region of 70-80 knots. Both the waterjet and the supercavitating waterscrew had to contend with cavitation limits on the thrust characteristics but by good hydrodynamic design exhibited good propulsive efficiencies at high speed (in excess of 40% above 60 knots) and added considerably to the state of the art of high-speed propulsion systems. The SES-100A pod or ram inlet was the first 70-80 knot waterjet inlet and the SES-100B was the first semi-submerged high-powered supercavitating propeller.

For the large high-speed SES there are various contenders for propulsion including both air and water propulsion.

Inlet design for the waterjet continues to be developed with the emphasis on minimising the amount of appendage in the water. The characteristics to be considered other than broaching of the inlet due to ship motion are the cavitation inception on the internal and external surfaces. Fig.12 shows typical pressure distributions for a selected inlet velocity ratio (IVR), the ratio of inlet velocity to free stream velocity. This particular case shows cavitation on the external surface, marked A . Fig.13 shows the typical cavitation inception boundaries for both external and internal surfaces marked A and B respectively. The permissible range of operation, for non-cavitating flow, between the two boundaries is seen to reduce to a narrow corridor at the SES speeds contemplated. Ship motions in surge, wave orbital velocities and mechanical positioning of the inlet opening place stringent demands on the inlet at 80 knots. At the lower speeds a wider deployment of opening is required such that higher IVR are used. As speed is increased the inlet opening is adjusted accordingly to avoid cavitation with its associated deleterious effects on inlet structure and the waterjet pump.

Concluding remarks

This paper has selected a few of the key areas of specific interest to the large SES of 2000-4000 tons designed to operate at 80 knots plus. The two 100 ton SES have provided encouraging technological data but it is hoped that this paper has pointed out that much work remains to be done to make a large high speed SES a success in the next few years and further has defined several areas where investigations will be continued for SES development.

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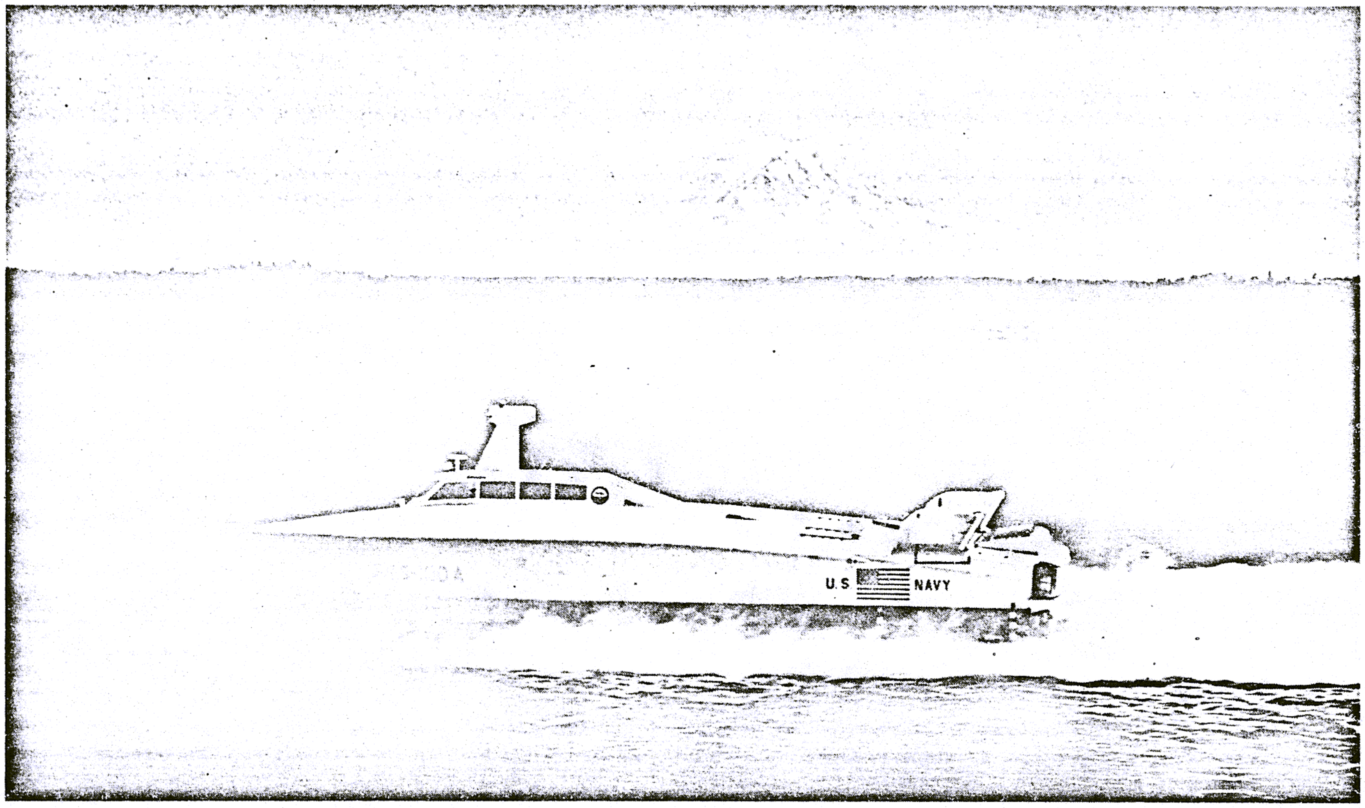

Fig. 1 SES-100A Cruising below Mt. Rainier, Washington

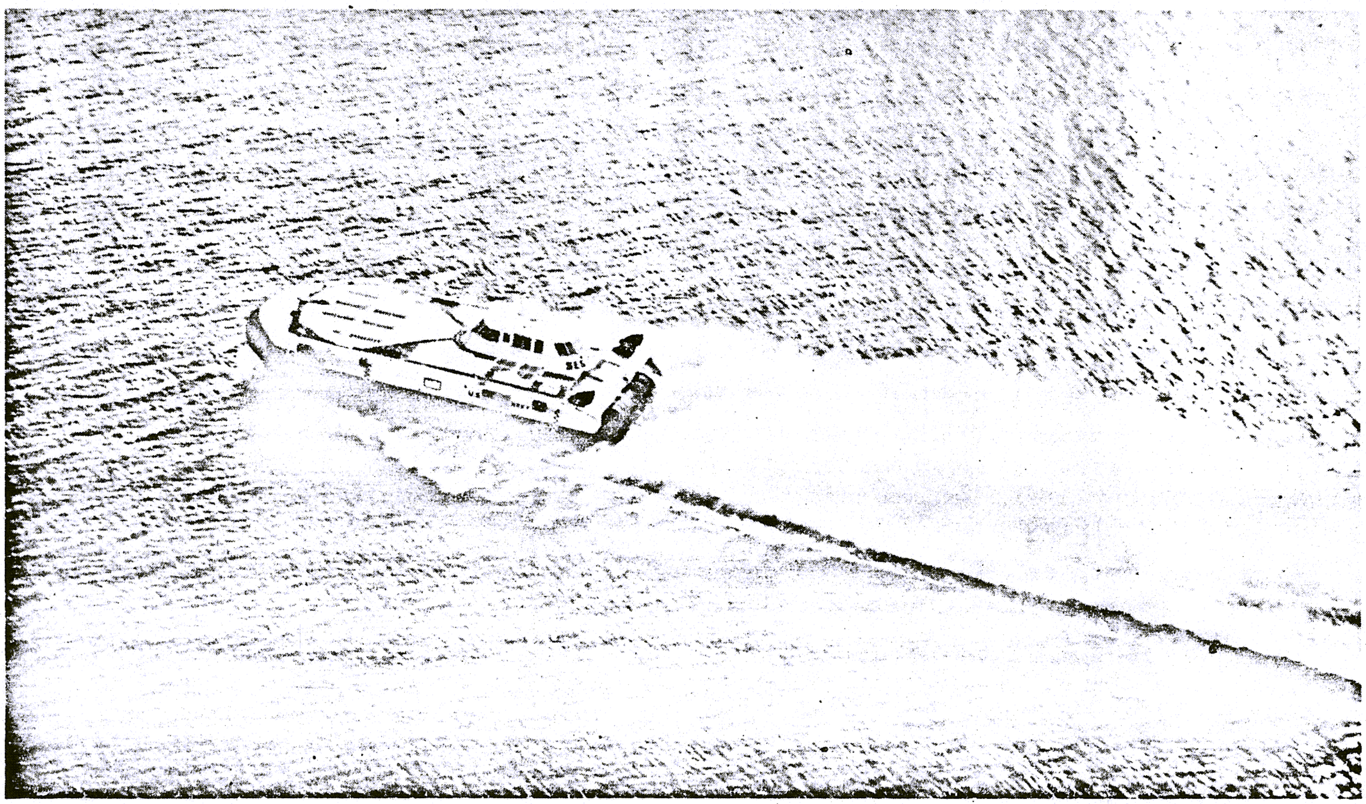

Fig. 2 SES-100B Cruising by New Orleans, Louisiana

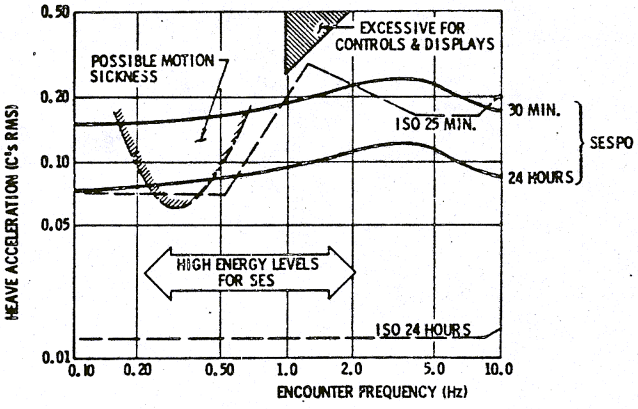

Fig. 3 Human tolerance to low frequency vibration

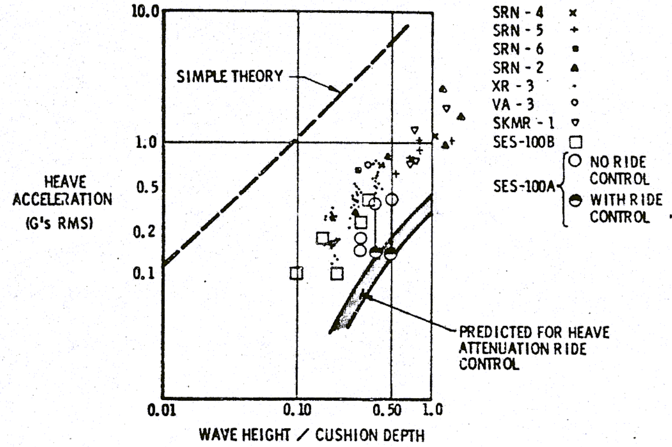

Fig. 4 Ride quality of today's SES

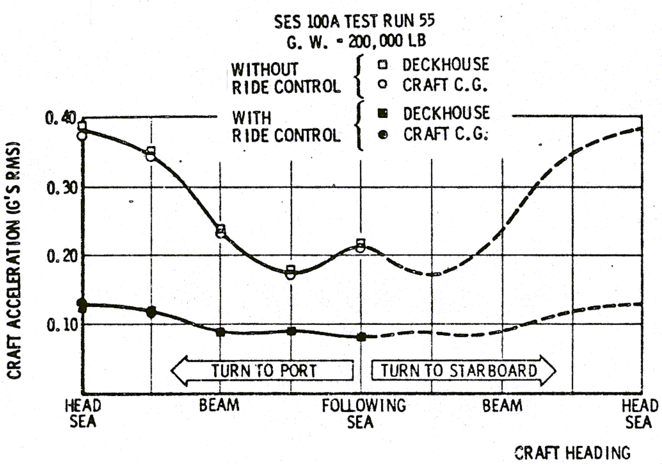

Fig. 5 SES-100A vertical accelerations vs. craft heading

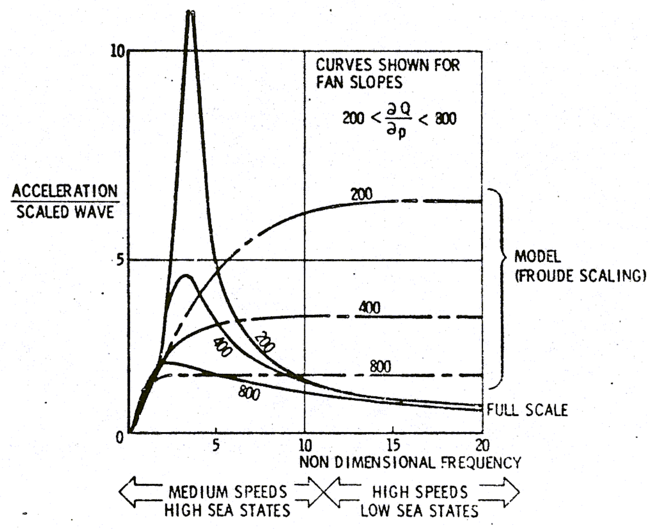

Fig. 6 Cushion compressibility effect

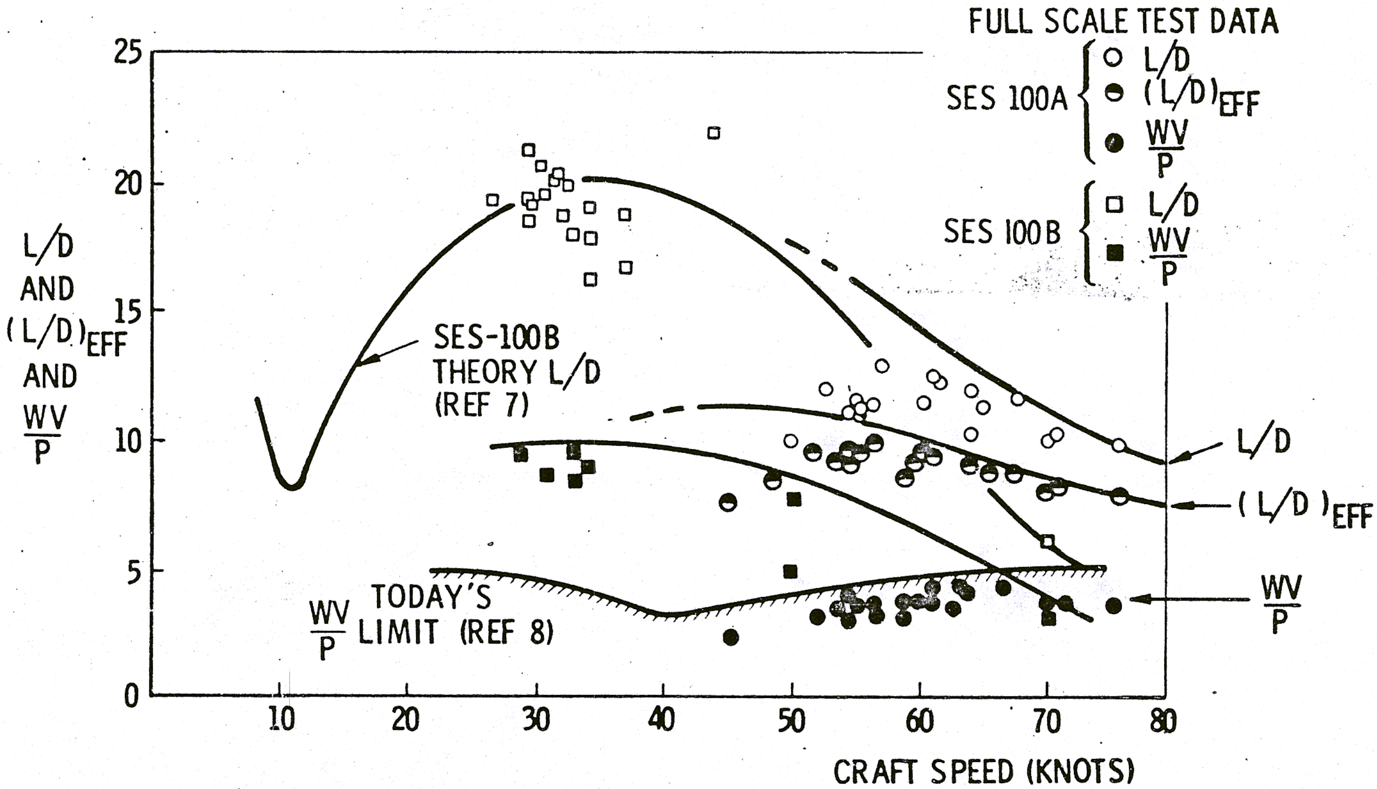

Fig. 7 Performance efficiency of SES

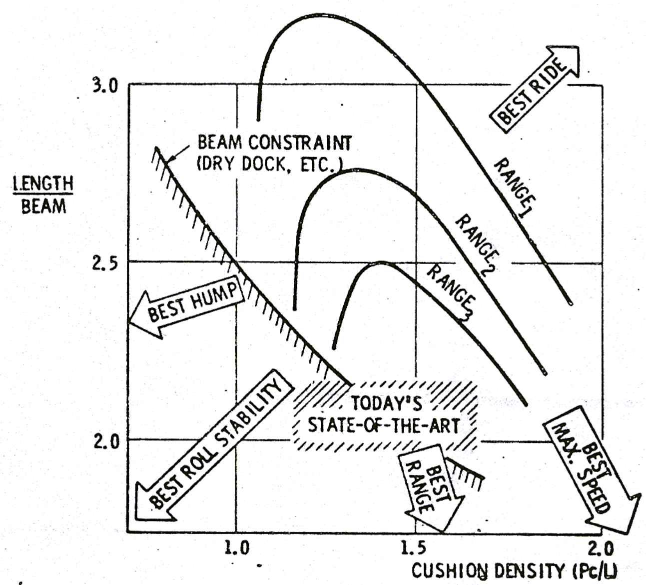

Fig. 8 SES sizing chart

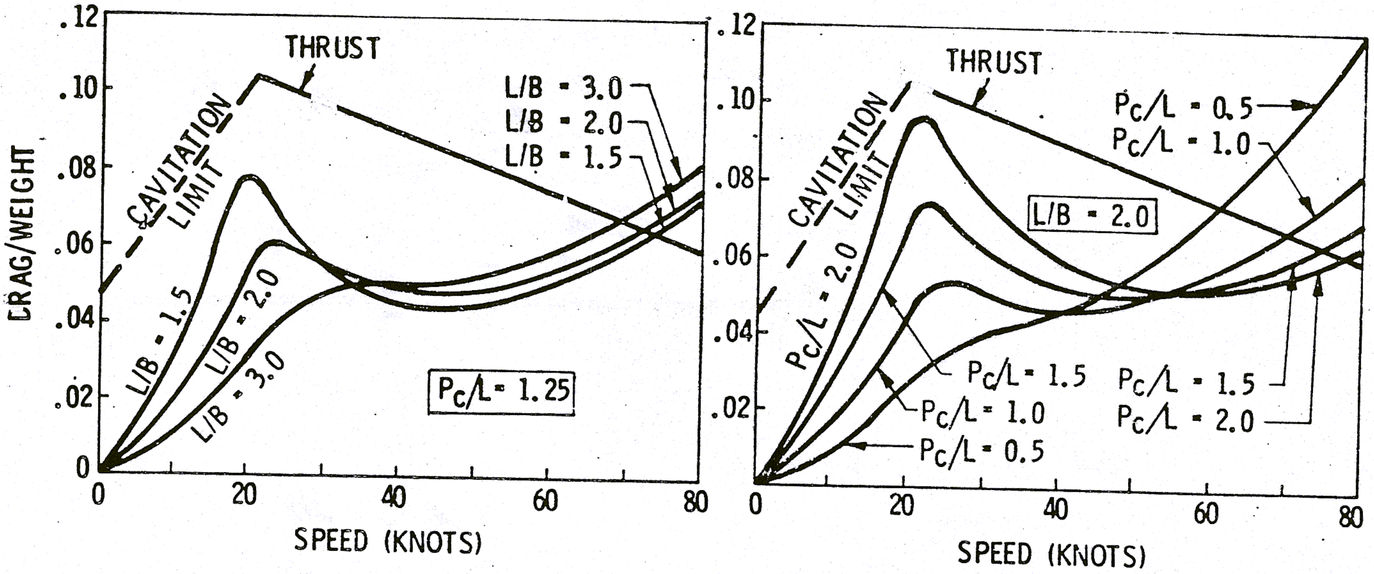

Fig. 9 Performance effect of L/B and Pc/L

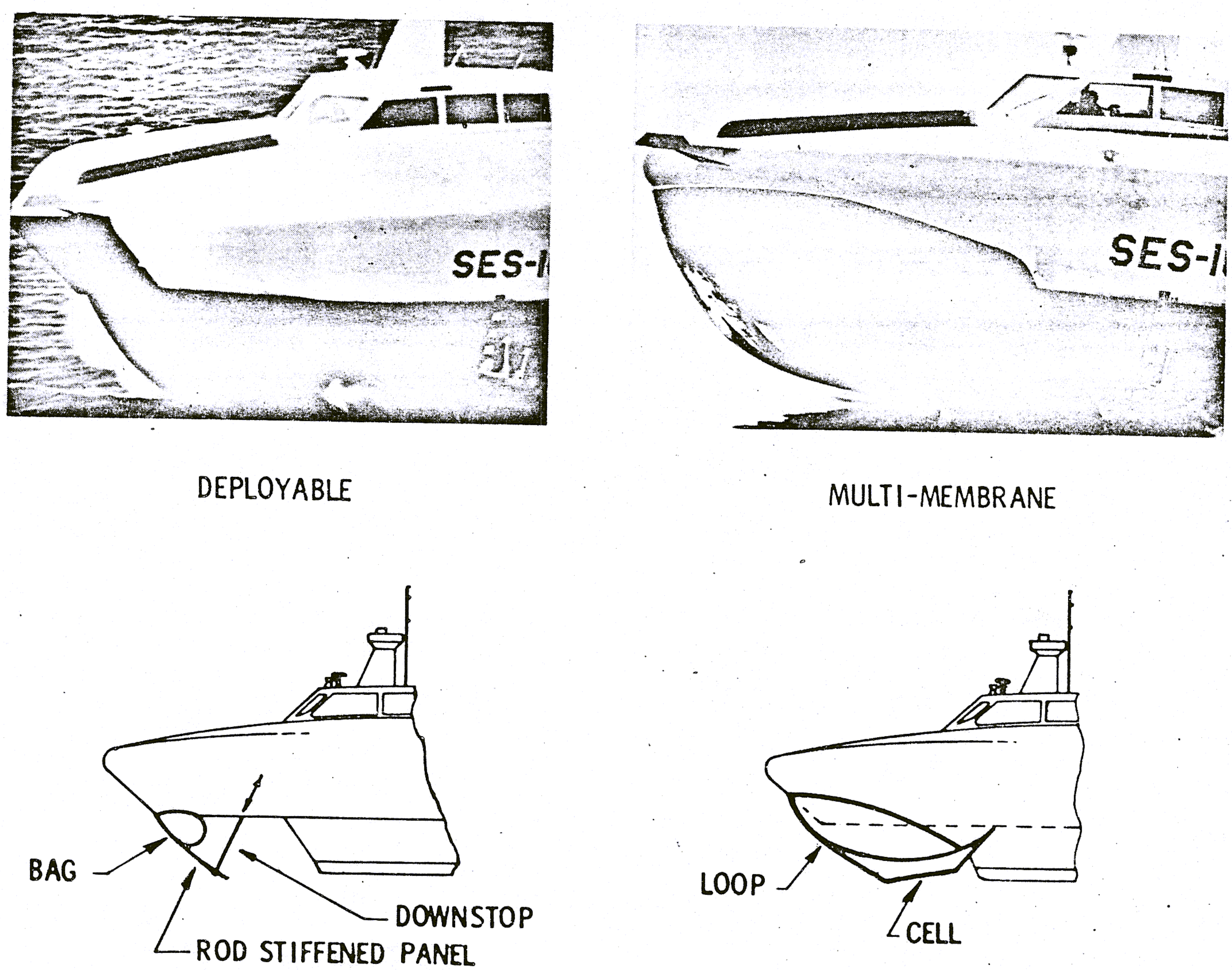

Fig. 10 SES-100A bow seals

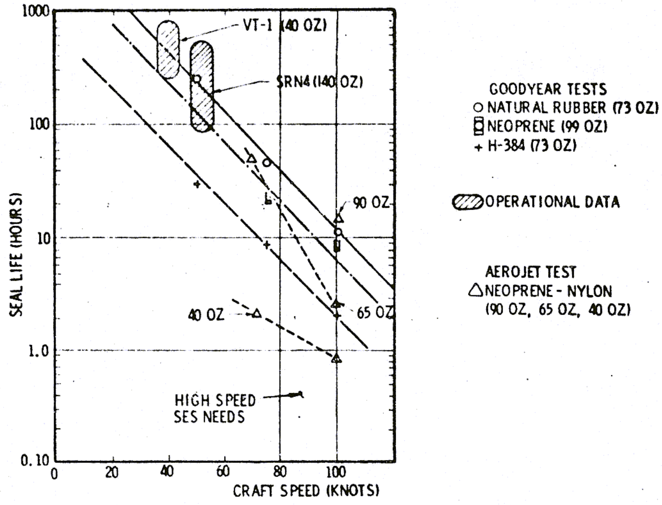

Fig. 11 Comparison of laboratory and operational seal life

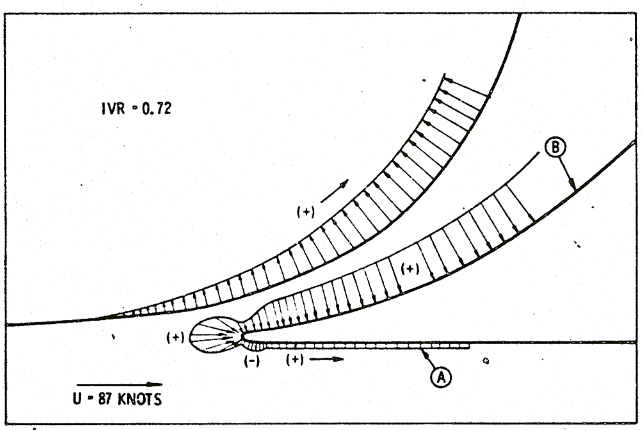

Fig. 12 Cavitation inception on flush inlet

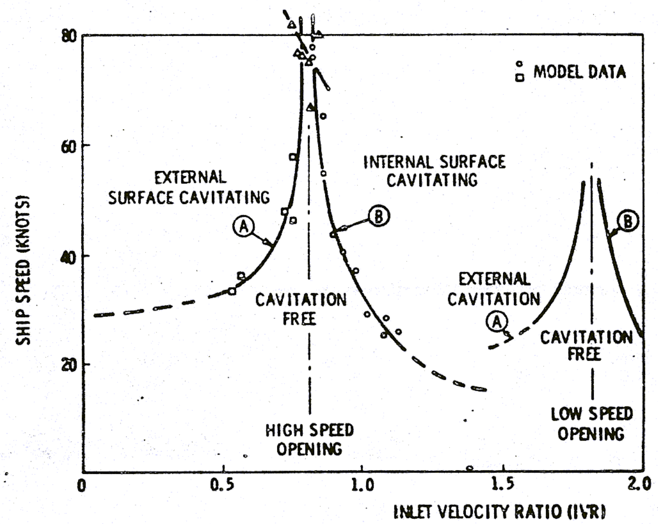

Fig. 13 Cavitation boundaries for flush inlet