CONSIDERATIONS ON THE STRUCTURAL DESIGN OF HIGH PERFORMANCE MARINE VEHICLES

BY

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and

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The opinions expressed herein are those of the authors, and do not necessarily reflect the official view of the U.S. Navy Department or of the Naval Services at large.

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ABSTRACT

Discussed are structural trials data from several planing hulls and one high length-to-beam ratio, surface effect ship. The collected data are used to modify several well-known methods for predicting hydrodynamic loadings and structural response of planing hulls, hydrofoils, air cushion vehicles, and surface effect ships. The merits and weaknesses of these and other less well-known methods are discussed. Simplified methods, developed by the authors, are presented for determining design-limit pressures for hydrofoils, air cushion vehicles, and surface effect ships which should produce consistent, yet not overly conservative, results.

SUMMARY

The data and methods presented in this paper are intended to allow designers to make more intelligent choices for hull loadings, depending on the requirement for craft service, the level of detail desired, and the amount of time and manpower available for the design. A designer of each vehicle type should find use for the simplified methods presented herein. Indeed, the major reason for preparing the paper was to synthesize the results of research into methods that could be easily and rapidly applied by the average designer; in such form, they should be of great utility in conceptual or feasibility design stages. Conversely, for the designer who desires more detail and can afford the time and cost, reference is made to the more general and the more complex methods such as those of Band et al.[1] and Jones and Allen[2]. The numerical examples included demonstrate the general utility of the methods.

NOMENCLATURE

A_D = design area, sq. ft.

A_R = reference area, sq. ft.

L = impact-load acting on the reference area, lb.

M = bending moment

N_z = impact-load factor

P = pressure, psi

$\bar{P}$ = average pressure over the reference area, psi

P_D = design limit pressure, psi

P_M = maximum pressure, psi

W_L = vehicle gross weight, lb.

d = full load draft, ft.

ℓ = length of an equivalent beam selected in stress analysis procedures

w = load per unit length on the equivalent beam

Δ = full load displacement, long tons

BACKGROUND

Since there has been no shortage in the past of papers dealing with high-performance surface vehicles, it would be understandable if someone should ask, "Why another"? The answer is that during the years that these papers have been written, our time has been spent conducting sea trials and acquiring experimental structural and motion-related data from various high-performance vehicles, to an extent that precluded writing papers of our own. However, the urging of our colleagues, who believed our data were of value to the marine design community, has resulted in this summary of our findings. It is not our intention to refute claims or design methods previously published. Rather, the paper shows to a great extent how accurate many of the "tried and true" methods really are; oftentimes, the scope of applications for them may be broader than the first authors believed.

To put the research accomplished in perspective, it first began in the late 1960's when serious consideration was given to construction of the 100-ton surface effect ship (SES) test craft. At the time, there appeared to be no all-inclusive design-oriented method capable of predicting hull impact loadings and pressures. As a consequence, we began to research the problem and eventually the results of our effort were presented in two reports[2,3] dealing with impact loadings. Since these reports were based primarily on prismatic hull forms, either planing or seaplane hulls, it seemed logical that comparison with similar data would be appropriate.

Fortunately, during this time the Navy was involved with the development, test, and evaluation of several planing hulls, and we had an opportunity to install instrumentation and to evaluate the structural performance of two large planing hulls during rough-water operations. This was the first significant increase in the data base since the work of Jasper[4]. A basis was thus established for evaluating the prediction method of Reference 2 as well as the other more popular design methods with regard to planing hulls. Also, during this time, we were involved in instrumentation and evaluation in rough-water operation of a high length-to-beam (L/B) ratio, surface effect ship test craft (a high L/B SES). This data source provided information for an evaluation of the Reference 2 method and of the various other popular design methods as applied to SES's. When these three data sources were added to the increasingly more available full-scale and model data gathered on other ship types, including catamarans and hydrofoils, it became obvious that a summary of the data was indeed needed.

LIMITATIONS OF DATA

It should be recognized that the data gathered in the course of our investigations were primarily in aid of the design of military vehicles, which by nature are required to perform under very strenuous conditions. Moreover, the structural designer is concerned with the most extreme of these conditions. Thus as shown in Figure 1 most of the data have been collected under conditions far more severe than those which would be encountered by ships in commercial service. Furthermore, most of the data that will be presented were gathered from smaller vehicles of 100 tons displacement or less. The primary emphasis was on local loadings, since these loadings govern a vehicle of this size the most. The greatest amount of and most detailed data were gathered about planing hulls, so a

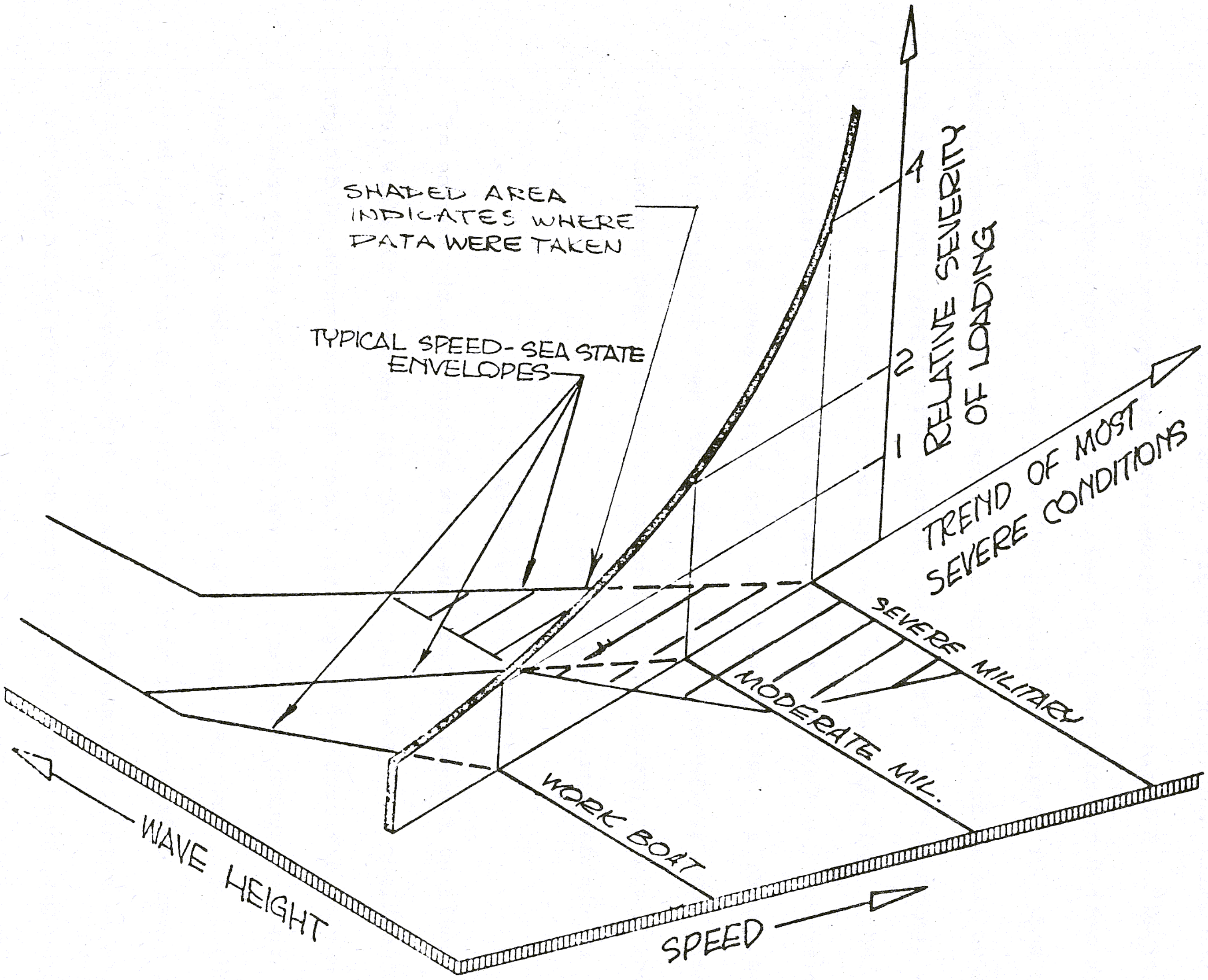

FIGURE 1- SPEED/WAVE HEIGHT OPERATIONAL INTERACTIONS

large portion of this paper will deal with planing-hull correlation. However, some of the data can be easily generalized; therefore, guidance will be provided for other types of vehicles, including hydrofoils, air cushion vehicles, and surface effect ships.

PLANING HULLS

There is certainly no dearth of methods available for use in the structural design of planing hulls; probably the best known is the venerated Heller and Jasper method[4]; however, also included may be Spencer[6], Silvia[7], Stiles[8], and Danahy[9]. All of these sources have contributed useful, design-oriented information to the planing-hull community. For the sake of brevity, however, the majority of comparisons and comments in this paper will be about the Heller and Jasper and the Spencer methods.

GENERAL DISCUSSION ON HULL BOTTOM-PRESSURE DISTRIBUTIONS

The analytical work which was reported in References 2 and 3 led to some distinctly new ideas with regard to impact-pressure distributions and how they might be measured. While it had been expected that average pressure decreases with increasing area, shown by the familiar Heller and Jasper method, results of the computer program described in Reference 2 indicated that quite large reductions could be expected over very small areas. This behavior was determined by using computer-predicted pressure distributions as shown in Figure 2 and integrating constant pressure contours over the hull bottom. The integration was performed to establish the relationship between impact pressure and load distributions and hull surface area. As expected, the maximum pressures acted over relatively small areas of the hull surface, thus constituting a small portion of

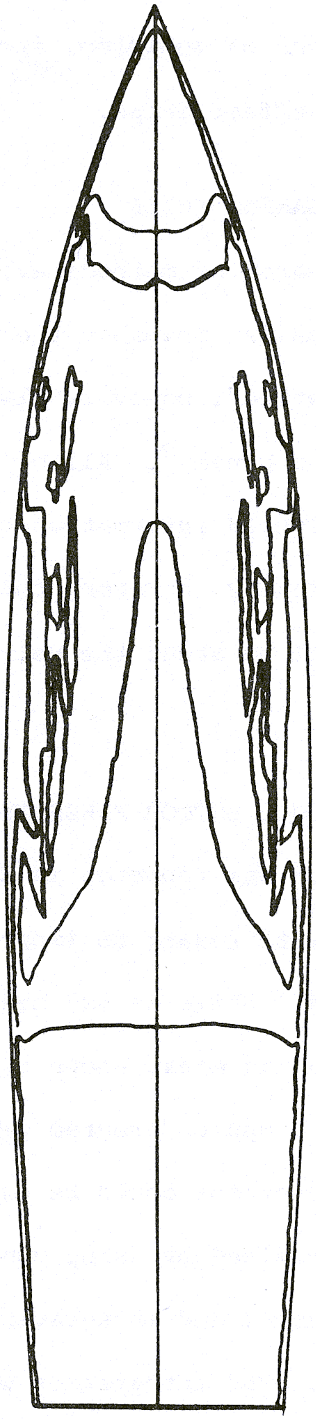

FIGURE 2 - TYPICAL CONTOURS OF PRESSURE ON HULL BOTTOM DURING IMPACT

the total impact load. Conversely, lower pressures were evident over a greater portion of the hull surface area and represented a higher percentage of the total load. A typical, pressure load versus area relationship is shown in Figure 3.

With the computer program now predicting very rapid pressure reduction, it was decided that an attempt would be made to use the two large planing hulls to determine the "average" distributed impact pressures acting over significant hull bottom areas as well as the maximum pressure for each impact occurrence. Thus, a few pressure transducers were located in close proximity to each other.Then, during the actual rough water trials, outputs of the transducers were electronically summed and averaged in various combinations to generate distributed pressure versus area information. Reference 2 describes the process in detail, and Figure 4 is included to demonstrate visually the pressure and load summing and averaging techniques. This method of collecting data was applied successfully to the two different planing hulls; thereby, gathering data for correlation with the two chosen design methods and with the computer program of Reference 2.

CORRELATION WITH DESIGN METHODS

Spencer[6] used Reference 2 to develop a generalized pressure-distribution function for crew-boat hull bottoms. However, no experimental data were available to Spencer for him to evaluate his form of the distribution function.

To correlate data with the Spencer method of predicting effective design pressures and his usage of the Reference 2 related pressure

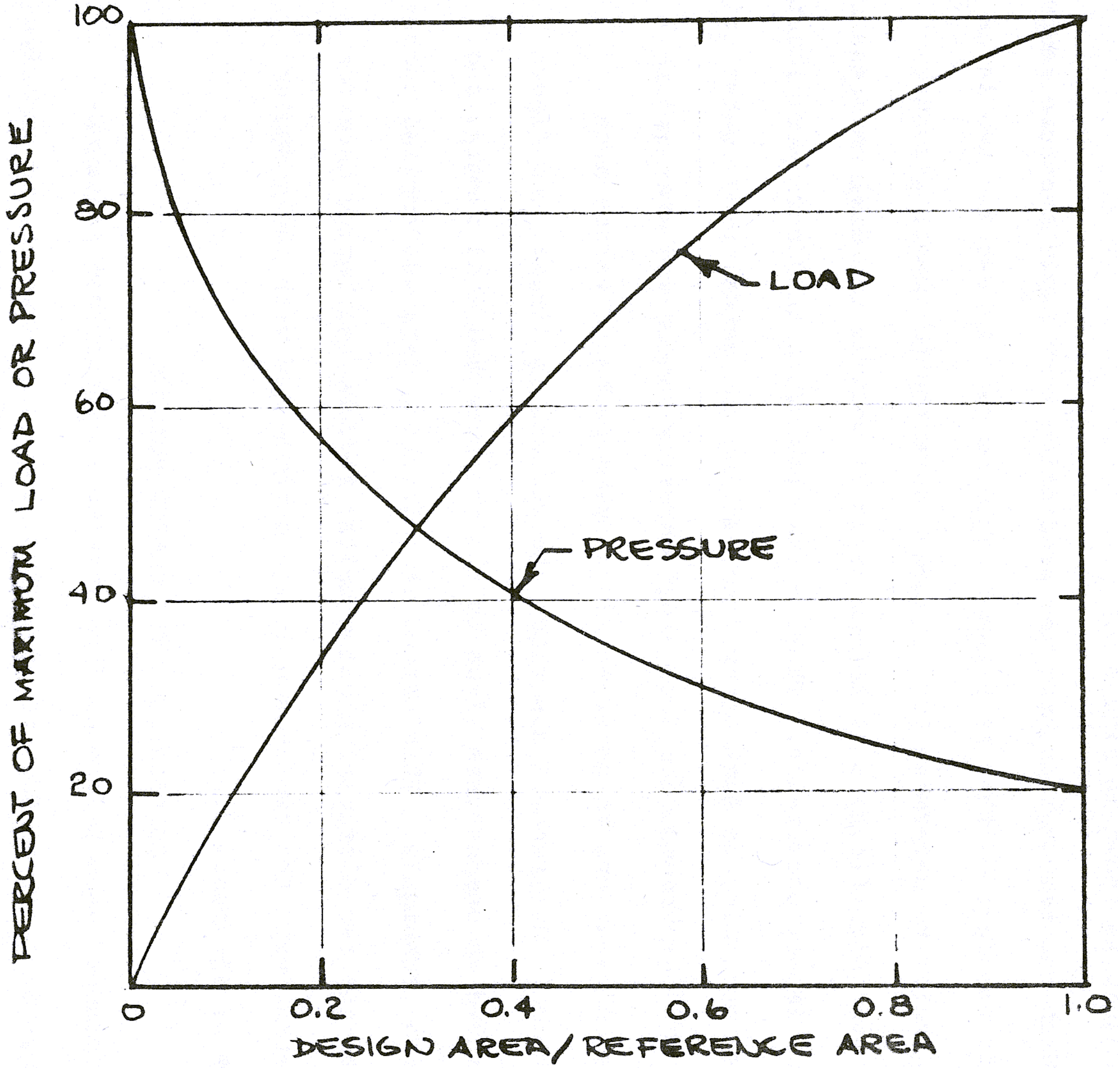

FIGURE 3 - SCHEMATIC OF PRESURE AND LOAD VERSUS AREA RELATIONSHIPS

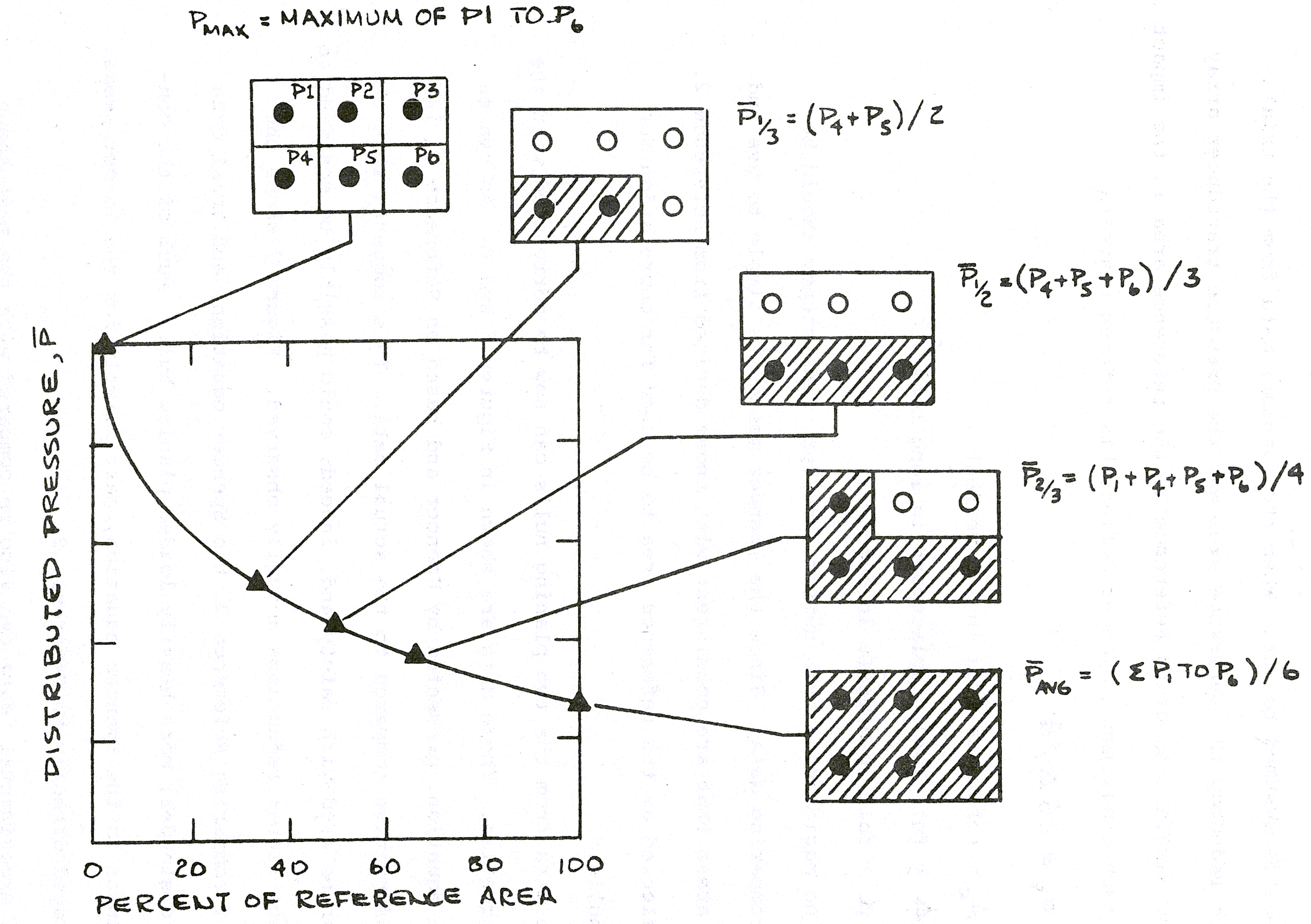

FIGURE 4 - GENERATION OF AVERAGE DISTRIBUTED PRESSURE FOR THE REFERENCE AREA FROM PRESSURE GAGE SUMMING AND AVERAGING COMBINATIONS

distribution function, there must be a reference area over which the total force is assumed to act. When the pressure data from the trials were first reduced, the reference area was the pressure transducer array area. For simulations using Reference 2, the reference area is the impact area for each condition. Spencer defines his reference area as:

$$A_R = 25\Delta/d$$

where A_R = reference area in square feet

Δ = full load displacement in long tons

d = full load draft in feet

There is no magic about this number; however, consistency should be used in comparing data. Since the Spencer method is simple to use and produces areas that are consistent with those derived from Reference 2, it was selected as the reference area to be used for correlation for planing hulls.

The data from the two planing hulls can now be correlated with the Spencer equation. These data are shown in Figures 5 and 6. As can be seen, the equation, presented by Spencer and based on Reference 2, is quite conservative compared to the actual data. This suggests that the method can be used with safety and, indeed, could possibly be modified to reflect the higher reductions actually observed. Figure 7 shows the results of comparing Reference 2, the Spencer equation, and trial data for a more slender, more heavily loaded planing hull. Much of the conservativeness in the Spencer equation, particularly for the larger areas, can be traced directly to Reference 2.

The experimental data can also be compared with the best known Heller and Jasper method for planing-hull design; see Reference 5. Notice

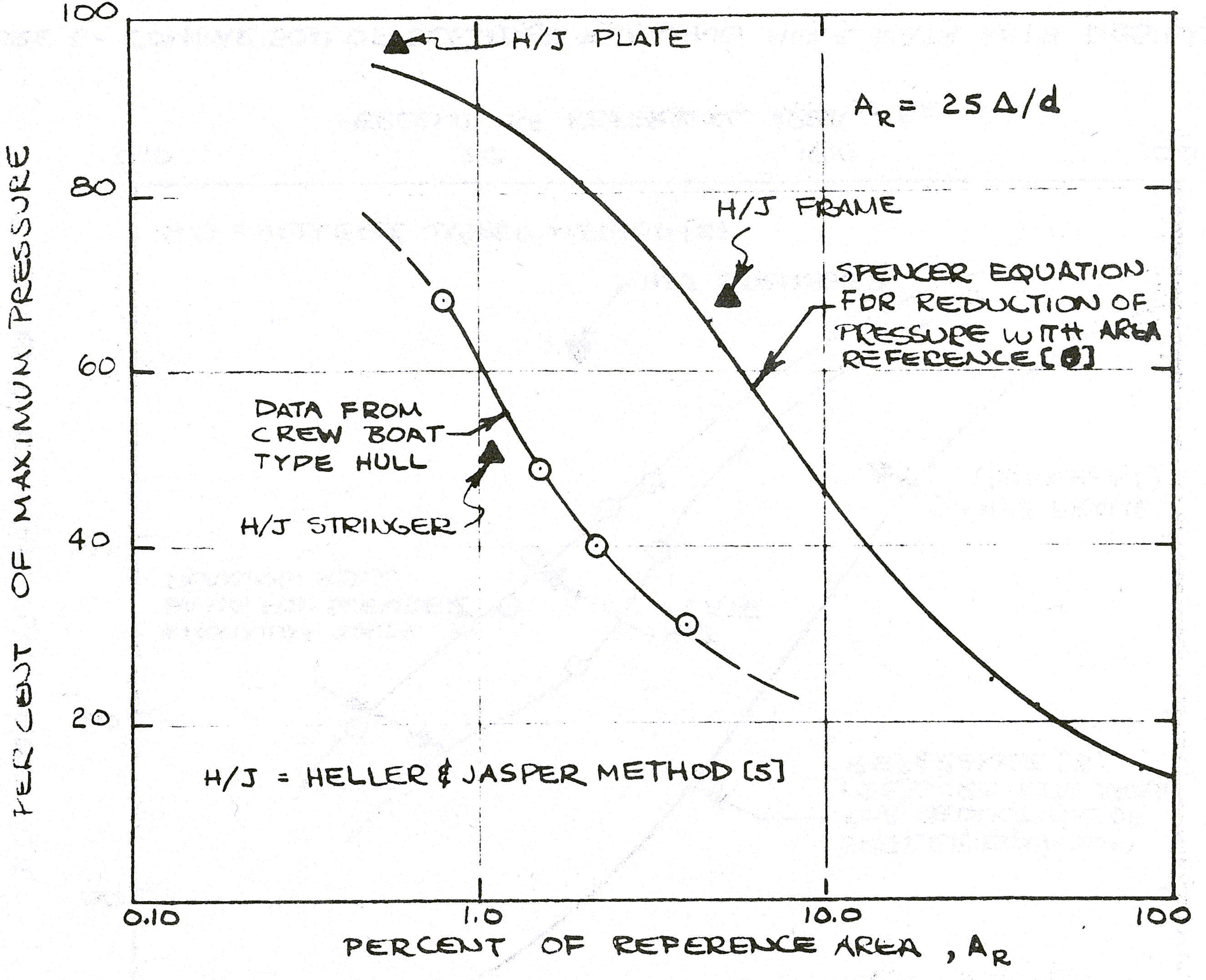

FIGURE 5- COMPARISON OF CREW BOAT DATA WITH DESIGN METHODS

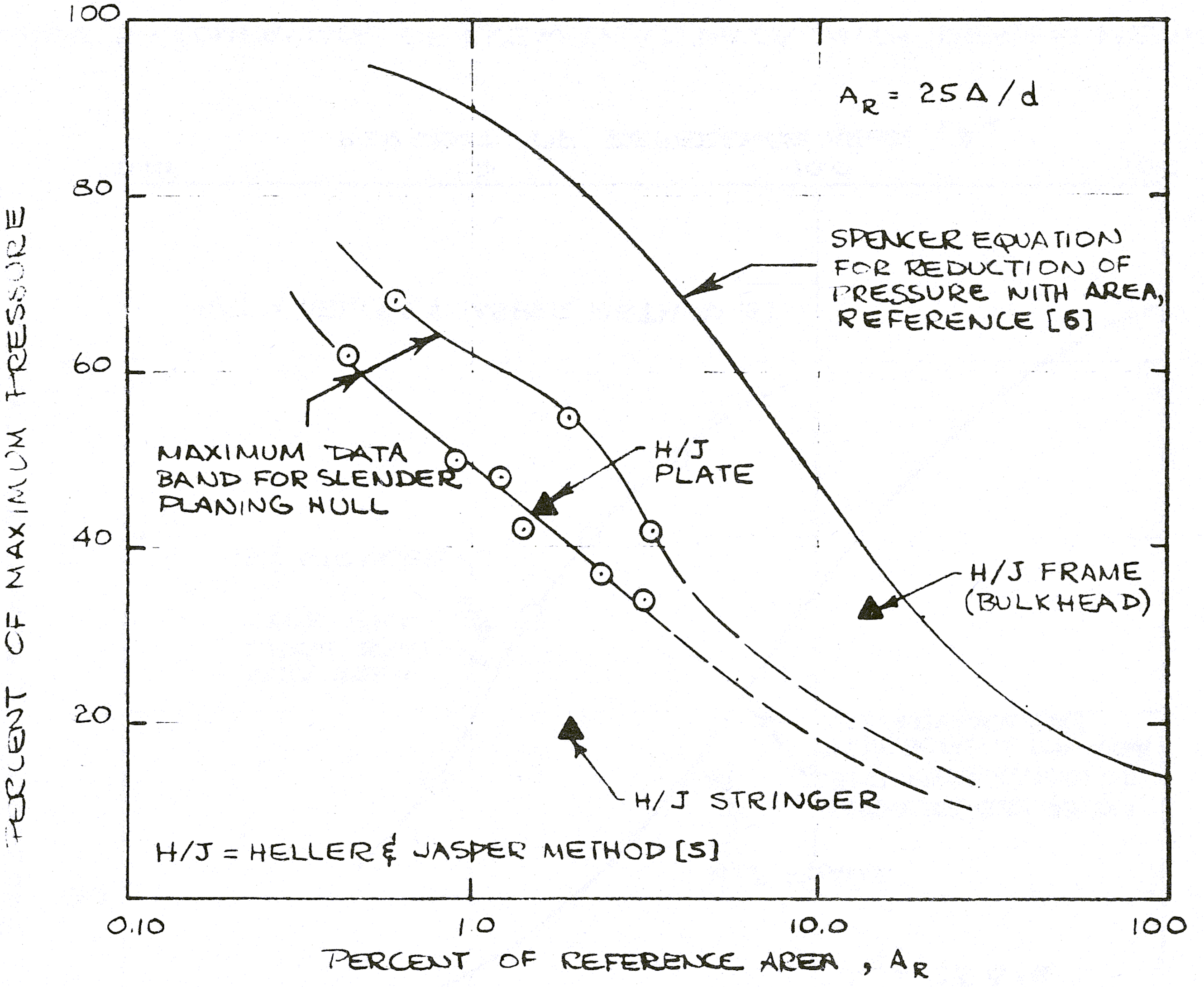

FIGURE 6 - COMPARISON OF SLENDER PLANING HULL DATA WITH DESIGN METHODS

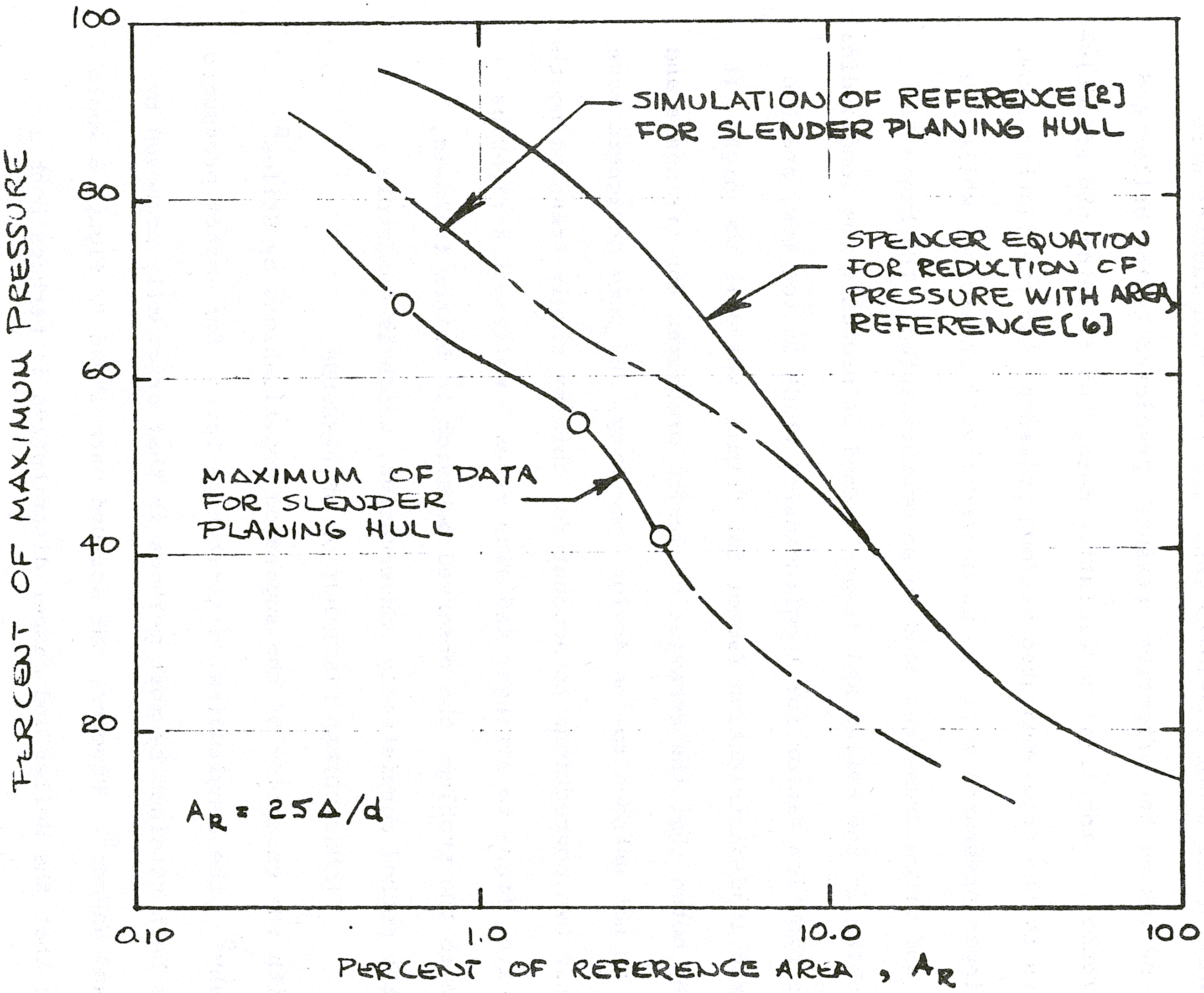

FIGURE 7 - COMPARISON OF MAXIMUM PLANING HULL DATA WITH OTHER DESIGN METHODS

on Figures 5 and 6 that three points are plotted to indicate the reduction factors that the Heller and Jasper method would predict for the planing hulls under consideration. The reduction factors were derived by taking the effective pressures predicted by the Heller and Jasper method, using trial acceleration data, and dividing the effective pressures by the trial-measured maximum pressure. As the figures show, the stringer pressures would be unconservatively predicted, while the plating and frame pressures tend to be overpredicted. The basis for this problem in the Heller and Jasper method is probably the longitudinal load-distribution factor for longitudinals, which is too low, and the transverse load-distribution factor for frames, which is too high. It should be noted that the behavior of design pressures for stringers and framing is not unknown to the design community, and some designers have made their own corrections to account for this by simply transposing the values, i.e. frame to stringer and vice versa. While not a rigorous solution to the problem, the observed behavior is better followed, making the method consistently conservative, which is desirable.

LONGITUDINAL DISTRIBUTION OF PRESSURE

With the exception of the suggested modifications by Stiles[8] and Danahy[9], the longitudinal distribution factor for impact pressures that has been retained by most methods is that originally proposed by Heller and Jasper[5]. However, the accrued data shown in Figure 8 would indicate that the Heller and Jasper distribution is perhaps both a conservative and an unconservative approach for much of the bottom area. The authors would not agree with References 8 and 9 which increase the distribution factor aft of amidships, since the data suggest that it can

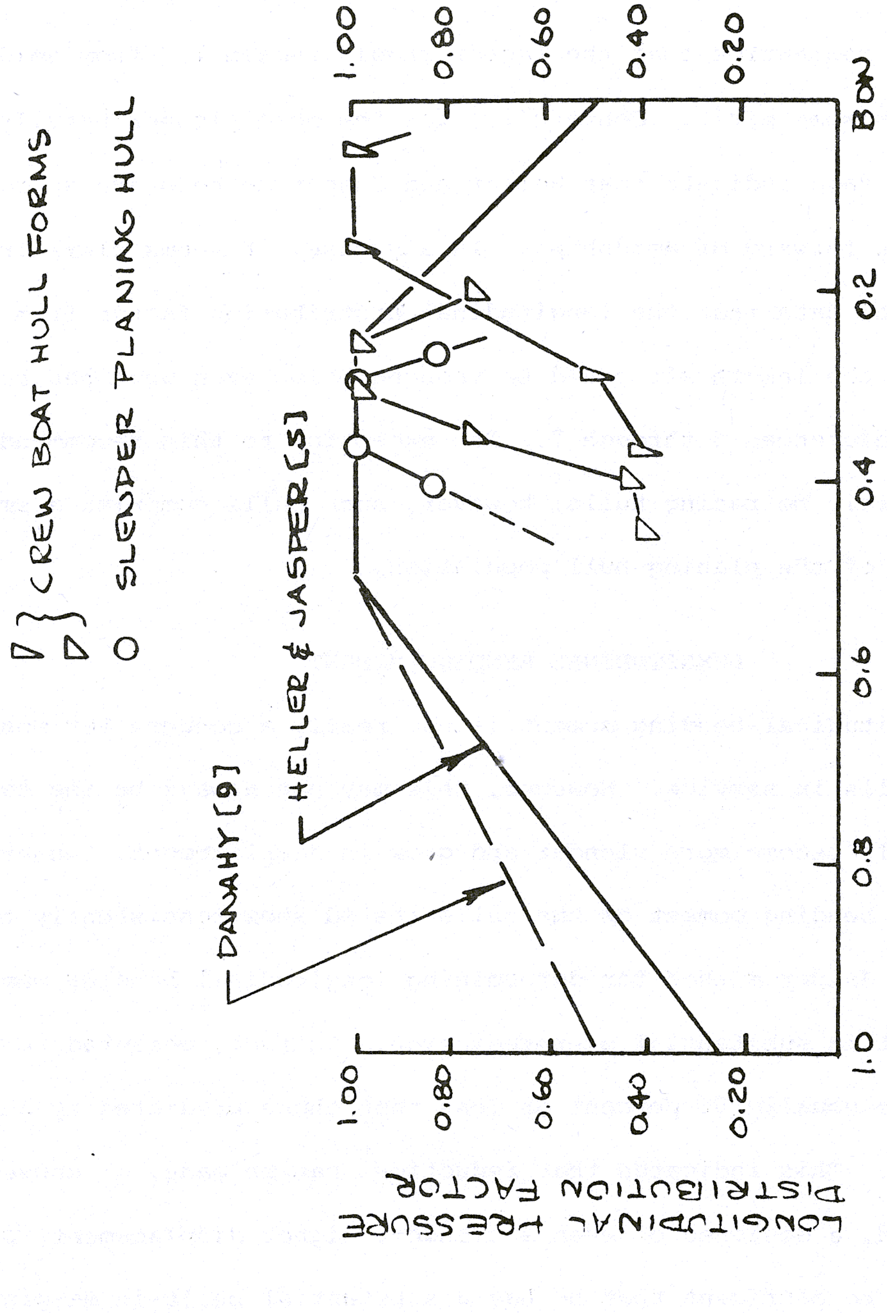

FIGURE 8-LONGITUDINAL PRESSURE DISTRIBUTION FACTORS FOR THREE PLANING HULLS

be further reduced. For some hull forms, though, it appears that the Stiles contention that the factor should remain 1.0 from amidships forward has some merit. Conversely, for the more slender heavily loaded hulls, the data indicate that Heller and Jasper methods are appropriate in the area forward of amidships. In any case, it seems clear from the experimental data that the longitudinal distribution factor from 40 percent of the length aft could be reduced below even what has been proposed by References 5 through 7. The exception to this recommendation could possibly be racing hulls; however, such hulls comprise a small percentage of the planing-hull population.

LONGITUDINAL BENDING MOMENT

Longitudinal bending moment is not really a concern for most planing hulls in service. However, this may not always be the case, should hulls become more slender and grow in displacement. Observations of midship bending moment on the hulls tested show consistently that the Heller and Jasper method for determining longitudinal bending moment will result in substantial overprediction. In fact, measured bending moments are usually 50 percent or less than those predicted by Heller and Jasper. This indicates that reductions can be made, or conversely, if retained, a designer of even a slender, higher displacement, planing hull would be confident that he has a substantial built-in margin against hull structural dynamic excitation or "whipping." *

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*Provided, of course, that the accelerations selected for design are appropriate. This will be discussed later in the paper.

BOUNDARY CONDITIONS FOR STRESS ANALYSIS

The common approach in designing local planing-hull structure is to use simple beam theory with fixed end conditions. Use of fixed end conditions is based on the assumption that the pressure acts uniformly over an area large enough for the forces on either side of the boundary to be substantially the same, thereby inducing fixity at the boundary. This assumption allows the maximum bending moment to be described as;

$$M = \omega \ell^2/12$$

where M = resulting moment

ω = load per unit length

ℓ = length of the equivalent beam

The pressure distributions noted during the trials are hardly uniform, even over very small areas. As a result it is not surprising that the stresses measured during the trials indicate less than fixed boundary conditions. In fact, a multiplicity of loading conditions can exist, which causes the end conditions to range from essentially pin ended to nearly fixed. However, for all the data acquired, the maximum value of bending moment has not exceeded the fixed-fixed value, although the maximum magnitude of this moment is as likely to occur at midspan as at the ends. It appears then, that current practice is a practical approach to the problem. However, it should be borne in mind that the fixed-fixed end condition does not adequately describe the physical circumstances, and an attempt to design a structure based only on this premise--distributing section modulus in a stiffener based on the fixed-fixed bending moment distribution-could lead to failures.

SELECTING IMPACT-LOAD FACTORS FOR PLANING HULLS

As most designers are aware, the Heller and Jasper method requires acceleration assumptions to derive a design pressure. Spencer includes acceleration in his formulas for crew boats by using the work of Fridsma[10]. Recently, Savitsky and Brown[11] have presented an emperical method for determining accelerations, and this method has been substantiated to a great extent by Blount and Hankley.[12]

However, the designer still must make a choice, which is whether to design to expected loadings for the power capabilities of the craft or to design to a possible lesser limit imposed by crew (or passenger) habitability. The planing-hull trials conducted to date have been terminated when the average of the one-tenth highest accelerations due to the impacts reached levels between 1.5 and 2.0 g's at the center of gravity, which incidentally, agrees with the statistical premises of Savitsky and Brown. For military planing hulls of less than 100 tons displacement, the authors would recommend a maximum value of 3.0 g's be used to determine hull bottom pressures. For other craft, it is left to the discretion of the designer, and, in the absence of model tests, the method of Savitsky and Brown for determining accelerations would be recommended.

HYDROFOILS

There are probably as many methods for determining hull bottom pressures for hydrofoils as there are types of hydrofoils, with the work of Jensen[13] probably the most used. Other methods have been summarized by Chuang.[14] Typically, from use of the various methods, pressures derived differ significantly for ships of essentially the same design speed and displacement. Though few pressure data have been gathered for hydrofoils, the data accumulated during the planing-hull trials allows some generalizations to be made. As an example, consider the design-limit longitudinal pressure distributions for hydrofoils; see Figure 9. Fully submerged foil, autopilot-controlled hydrofoils, operating in rough water, platform under almost all circumstances. Under these conditions, most hull impacts will occur during cresting or on slams following a broach of the forward foil. For both of these cases, the impacts should occur on the midbody and forward portions of the hull. An impact at a trim angle high enough to cause the stern to strike first would seem unlikely. For these reasons the distribution assumed for PLAINVIEW (AGEH-1) and PEGASUS (PHM-1) aft of amidships would not seem appropriate. Conversely, since most impacts occur in the forward half of the ship, the distribution factor for TUCUMCARI (PGH-2) and PEGASUS (PHM-1) would seem to be low. The design factors of HIGH POINT (PCH-1) and FLAGSTAFF (PGH-1) would seem the most appropriate for general use, would compare favorably with those of planing hulls (Figure 8), and are more conservative than Danahy.[9]

HULL BOTTOM DESIGN LIMIT PRESSURES FOR HYDROFOILS

As far as hull bottom, design-limit pressures are concerned, the differences are at least as striking as for the longitudinal distribution factors. Table 1 gives examples of the maximum design limit for hull bottom

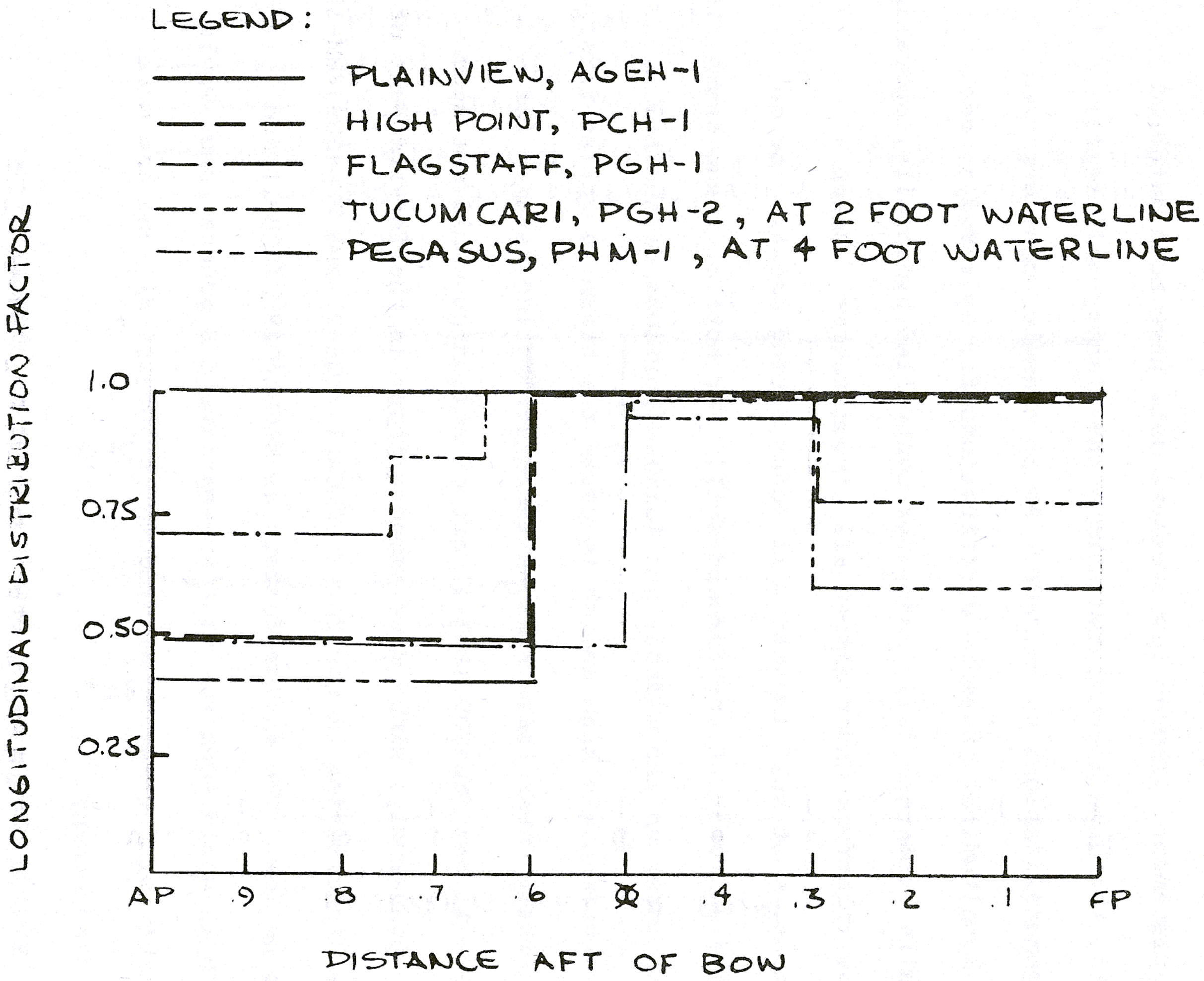

FIGURE 9 - DESIGN LONGITUDINAL PRESSURE DISTRIBUTION FACTORS FOR FIVE HYDROFOILS

HYDROFOIL	DISPLACEMENT, Δ, TONS	FULL LOAD DRAFT, d FEET	REFERENCE AREA, A_R FEET^2	DESIGN LIMIT PRESSURE* PSI	MAXIMUM PRESSURE P_m, PSI	AVERAGE PRESSURE $\bar{P}$, PSI	IMPACT LOAD $A_R \times \bar{P}$, TONS	IMPACT LOAD FACTOR, N_z
PLAINVIEW AGEH-1	304	6.25	1216	45	56	7.8	613	2.0
HIGH POINT PCH-1, MOD 1	121	3.92	774	30	47	6.6	328	2.7
FLAGSTAFF PGH-1	67	4.25	394	26	42	5.9	149	2.2
TUCUMCARI PGH-2	58	4.4	330	73	122	17.1	362	6.2

* FROM REFERENCE [15]

TABLE 1 - DEVELOPMENT OF LOAD FACTORS FOR HYDROFOILS BASED ON DESIGN PRESSURES

pressures for different hydrofoils. Note the vast difference in the design limit pressure for FLAGSTAFF and TUCUMCARI which are of similar displacement and have the same operational requirements.

A method of comparison as to the degree of conservativeness of these pressures, can be derived by using modification of planing-hull methods. For instance, if we assume the Spencer method for deriving a reference impact area and select a curve for pressure reduction that runs through the experimental data shown in Figures 5 and 6, we can predict maximum pressures that will be consistent with the design limit pressures; see Figure 10. The maximum pressure P_M is found by entering the curve at the smallest design area on the hull bottom- the area of a plate between stiffeners -and assuming that the design-limit pressure applies to that area. The maximum pressure is found by dividing the design-limit pressure P_D by the reduction factor at the design area. Furthermore, we can determine the total load over the reference area as;

$$L = \bar{P} \times A_R$$

where $\bar{P}$ is the average pressure over the reference area.
When the load thus determined is divided by the ship displacement the result is an effective impact-load factor N_z . This can be roughly compared to the maximum g's measured on hydrofoils in rough water operation.

DESIGN EXAMPLE

A numerical example is included for FLAGSTAFF. Required FLAGSTAFF characteristics:

Displacement , Δ = 67 tons

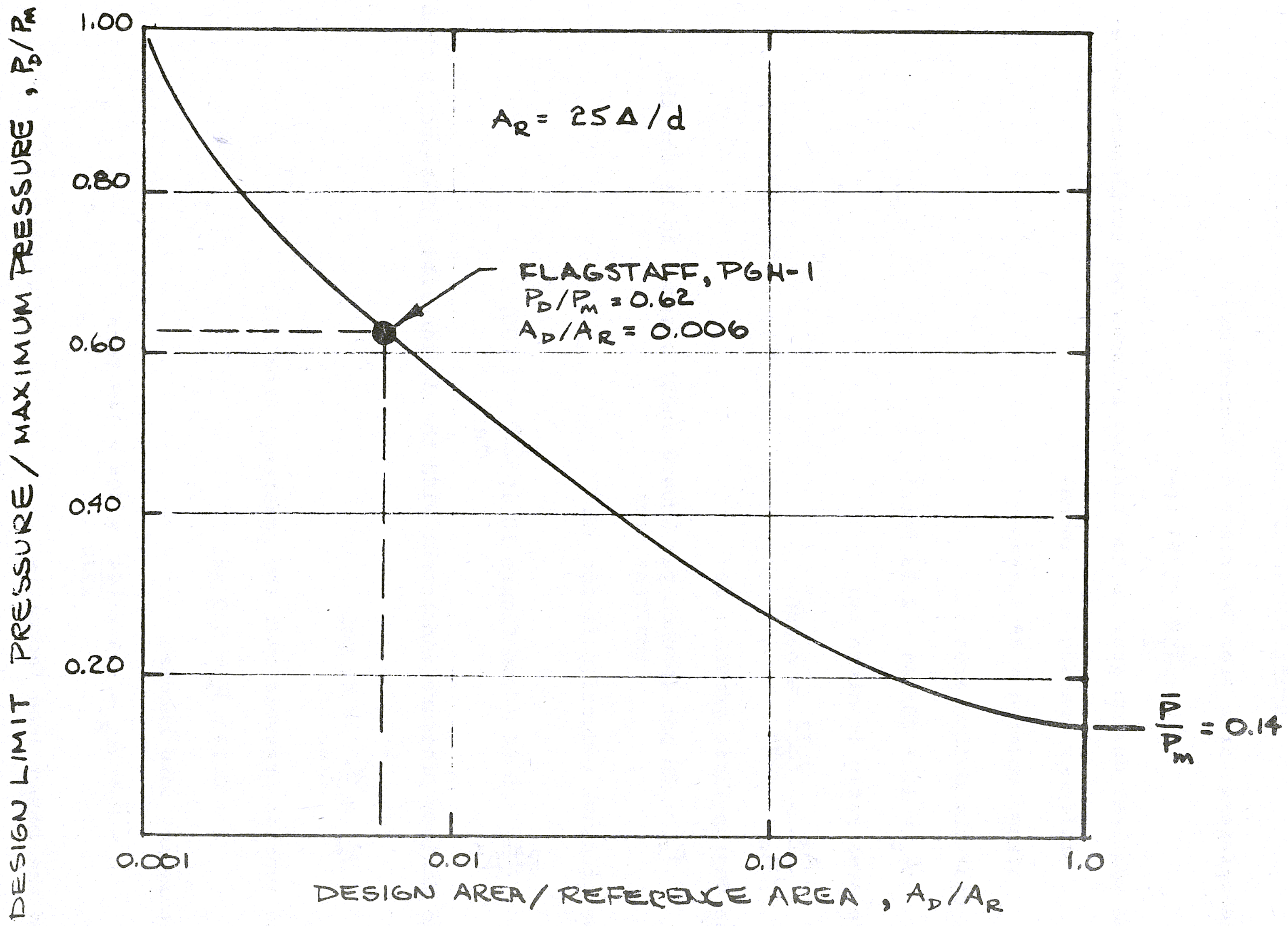

FIGURE 10 – HYDROFOIL EXAMPLE USING SYNTHESIZED PRESSURE/AREA CURVE

Draft (d) = 4.25 feet

the reference area from Reference 6 is given by:

$$A_R = \frac{25\Delta}{d} = 394 \text{ feet}^2$$

the smallest design area is the plating between two stiffeners where:

stiffener spacing = 12 inches

frame spacing = 28 inches

so that the design area is:

$$A_D = 12 * 28/144 = 2.33 \text{ feet}^2$$

the area ratio is given by:

$$\frac{A_D}{A_R} = \frac{2.33}{394} = 0.006$$

the design-limit pressure is:

P_D = 26 psi (pounds per square inch) assumed to be for the smallest area

the pressure reduction factor is:

$$\frac{P_D}{P_M} = 0.62 \text{ (from Figure 10 at } \frac{A_D}{A_R} = 0.006\text{)}$$

the maximum pressure consistent with the design-limit pressure is then:

$$P_M = \frac{26}{0.62} = 42 \text{ psi}$$

the average pressure over the reference area is:

$$\overline{P} = 0.14 \times P_M = 5.9 \text{ psi}$$

the impact load becomes

$$L = \overline{P} * A_R = 5.9 \times \frac{144}{2240} \times 394 = 149 \text{ Tons}$$

and the impact load factor is:

$$N_Z = \frac{L}{\Delta} = \frac{149}{67} = 2.2$$

As mentioned previously the maximum g's measured occurred during slams following a broach of the forward foil and were approximately 1.5 g's. Cresting produces much lower g levels, about 0.5 g's. Since all the impact factors derived from the previously described procedure are greater than 2.0 (Table 1) it would appear that the limit-design pressures are conservative by a factor of at least 1.33; those of TUCUMCARI, overly conservative by a factor of 4.0.

The reverse of the described process offers a simple approach to determine hull-bottom, design-limit pressures for hydrofoils, depending only on an assumed load factor. The approach is then similar to that of Heller and Jasper; only, the impact-load factor is substituted for the center of gravity heave acceleration. However, a good estimate of the design-limit pressures can likely be made by assuming N_z to be equal to the expected maximum heave acceleration. This approach should be useful for preliminary design purposes.* If more detailed analysis is desired, for particular impact conditions- such as a rolled impact or impacts at high trim angles -then the authors would recommend use of Reference 2.

AIR CUSHION VEHICLES AND SURFACE EFFECT SHIPS

As opposed to planing hulls and hydrofoils, there are relatively few verified methods for predicting impact pressures on ACV's and SES's, and only one generally recognized method. This is contained in the British Hovercraft Safety Requirements (BHSR)[16] published by the British Civil Aviation Authority. This method is also contained in a textbook (Reference

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*Hydrostatic pressure was not considered in the calculations of the impact-load factors of Table 1. If desired, the hydrostatic pressure can be added to the design limit pressure, as is the practice in the planing hull methods

17) and in a report by Mantle[18], which are more readily available to the designer. Other methods have been suggested, including Band et al.[1], Giannotti and Fuller[19], Giannotti[20], and of course, Reference 2. The BHSR method is simple to use; unfortunately, it seems to give good results for the wrong reasons. There have also been local structural failures on British commercial craft which were designed from these formulas. These craft were subsequently repaired, and newer craft were assigned higher limit pressures; however, the BHSR criteria remain unchanged.

Mantle[18] suggests that no matter what method is used, the design-limit pressures for craft designed in the United States usually follow the trend of 1 psi/knot of design speed in rough water. However, it is known that the maximum pressures do not necessarily occur at the maximum speed but can occur at lesser speeds, depending on craft-motion characteristics. For larger ACV's or SES's, this may well be when they are off-cushion, making speeds only to maintain headway. The problem that exists is, how to derive a simple yet consistent method for deriving design-limit pressures. We suggest that the answer lies in data gathered on a high L/B SES test craft, combined with a few assumptions from the BHSR method.

The high L/B SES was instrumented, calibrated, and tested in rough water with the express purpose of generating information about local pressure and load design for both on- and off-cushion, impact-induced loads. The test craft provided a source for determining the average distributed impact pressures acting over significant areas of the wet deck. Pressure and load data were derived by loading selected forward portions of the hull structure and then measuring the structural response, thereby making the entire forward portion of the wet deck a calibrated load sensor. The data gathered during the trials provide the basis for modifying the BHSR method and evaluating other design methods.

CORRELATION WITH DESIGN METHODS

In evaluating the design methods, the problem to be solved is similar to that required in the hydrofoil case; that is, a reference area is needed to which a pressure reduction curve can be applied. The BHSR method allows one to calculate N_Z , P_M , and $\bar{P}$, assumed to be;

$$\bar{P} = 0.44\, P_M$$

Intuitively, one would think that if these formulas were consistent, one could derive an impact area simply performing the following operation:

$$A_R = \frac{N_Z \times W_L}{\bar{P}}$$

where A_R = desired reference area

N_Z = impact-load factor

W_L = gross weight of the craft in pounds

$\bar{P}$ = average pressure

When this calculation was made for several ACV's, the results were indeed consistent, and areas could be derived in this manner. Thus, a BHSR reference area may be calculated for the high L/B SES test craft, and the BHSR criteria can be plotted in the proper relationship to SES data.

The 1 psi/knot contention of Reference[18] can be plotted, using the pressure area relationship developed by Buckley and reported in Reference 21.

For this method the reference area is developed as:

$$A_R = N_Z \times W_L / 0.5 P_M$$

where N_Z and W_L are as previously defined, and P_M = 1.0 psi per knot of design speed in rough water.

As in the hydrofoil case, the N_z selected was based on the maximum vertical g load experienced at the center of gravity of the SES test craft.

The results of these operations are shown in Figure 11. Note that the reduction factors of References 16 and 21 are conservative when compared to the actual data. This is one example of why BHSR criteria give the right answer for the wrong reason, i.e., the maximum pressures derived are too low; however, the distribution factor is very conservative. Therefore, they tend to cancel, and thus produce reasonable design-limit pressures. Also, BHSR criteria cannot derive pressures for intermediate design areas. The pressure reduction predicted by Reference 21 also is too conservative, and combined with an assumption of 1 psi/knot for maximum pressures, will generally yield conservative design pressures for present day operational ACV's. It does allow, however, for pressures to be derived for intermediate-sized design areas. It should be noted that the craft designed by this method have not as yet experienced a significant portion of their design load factors; so a strict evaluation of this method is not possible at this time.

BAG OR SEAL LOAD ALLEVIATION

There has been conjecture as to how much an inflated bag or seal of an ACV or SES alleviates pressure on the structure behind the seal.

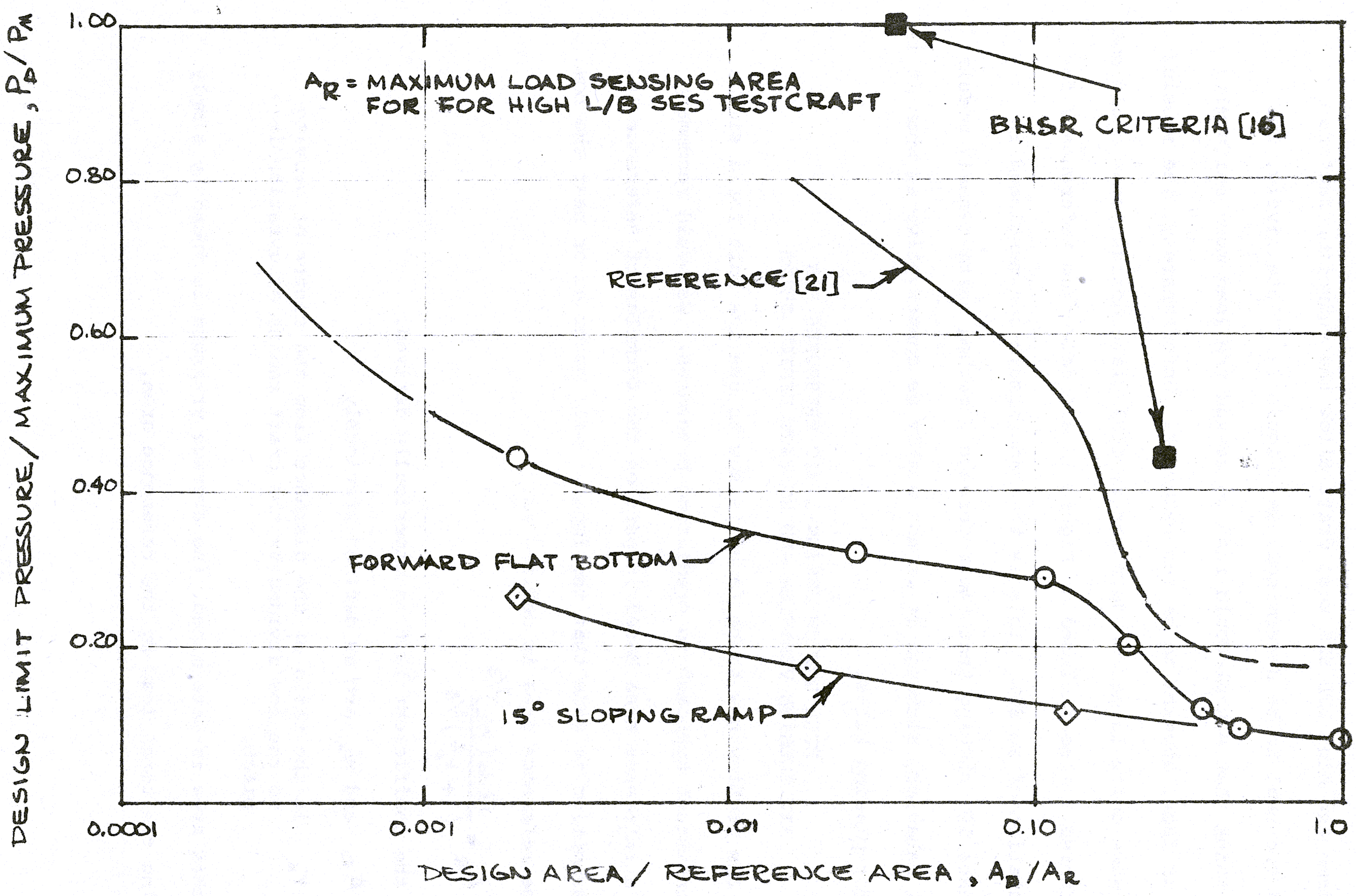

FIGURE 11 - COMPARISON OF HIGH L/B SES DATA WITH SEVERAL DESIGN METHODS

Data from the high L/B SES test craft allow the reduction factors behind the seal to be determined; see Figure 12. Intuitively, one would think that a seal would tend to spread the load more uniformly; this, in fact, appears to be the behavior. Unfortunately, the sensing areas were quite large, so data for typical plate design areas could not be gathered. Also included on Figure 12 are data from Reference 2 for a simulation of an SES. This is the same simulation mentioned in Reference 19, showing that the Reference 2 method quite closely models actual behavior, although it is not nearly as conservative as when it is used for planing hulls.

HULL BOTTOM DESIGN-LIMIT PRESSURES FOR AIR CUSHION VEHICLES AND SURFACE EFFECT SHIPS

The problem now to be faced is how to use the data for a simple yet consistent approach to predicting pressures. We would recommend using a reference area based in part on the formulas of Reference 16. The assumption is made that the impacts will occur at or near the bow, and the reference area is defined as:

$$A_R = \frac{12.6\,(W_L)^{2/3}}{(1 + r_x^2)^{2/3}}$$

where the coefficient 12.6 is emperically derived,

A_R and W_L are as defined previously

r_x is the ratio of the distance from the center of gravity to the bow divided by the craft radius of gyration in pitch.

Once this area is determined, the average pressure is found by simply dividing the total load by the reference area.

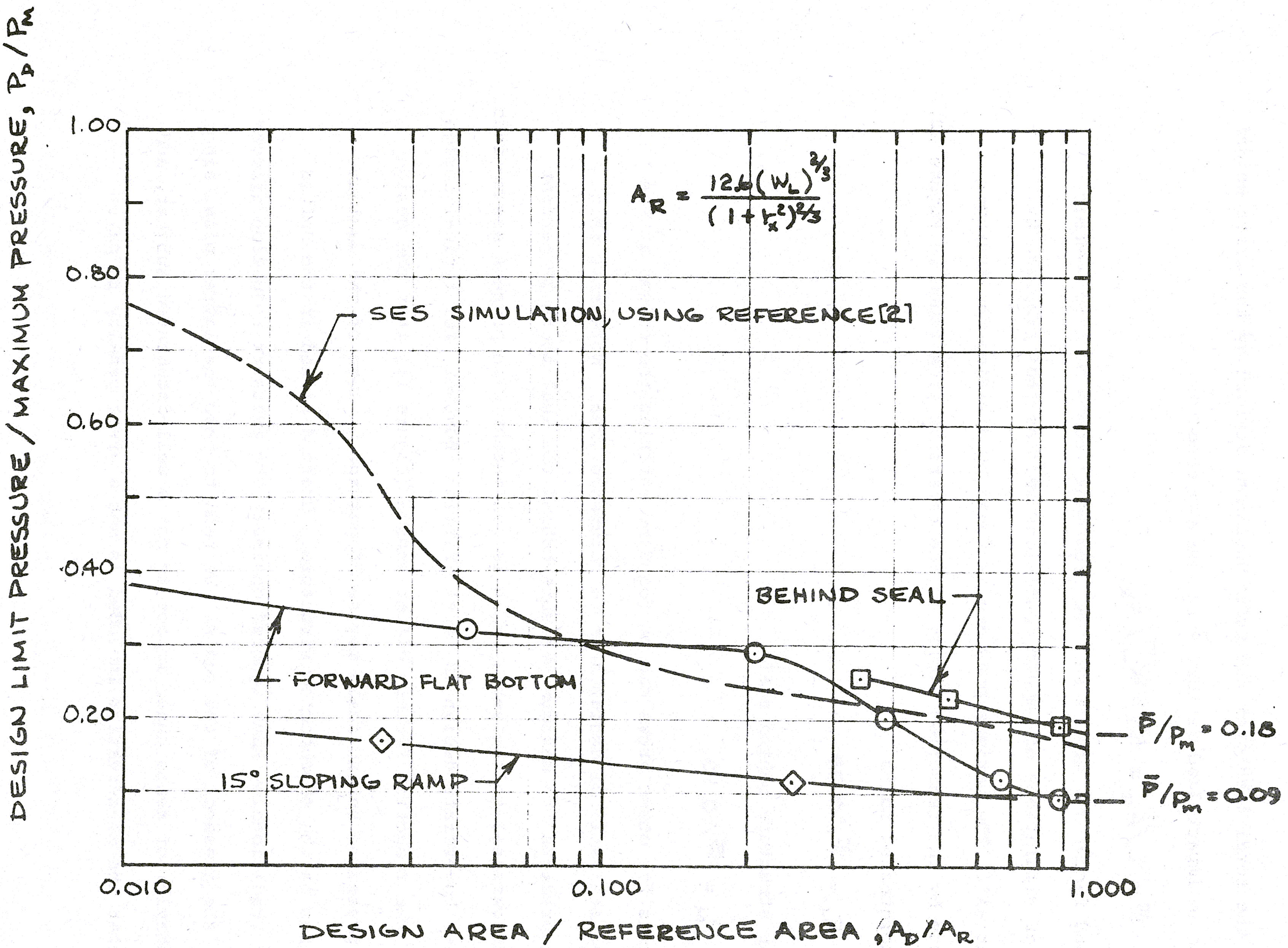

FIGURE 12 - COMPARISON OF SES SIMULATION WITH HIGH L/B SES DATA

Since the total load over the reference area divided by the craft weight yields an impact-load factor, $\bar{P}$ can be expressed as:

$$\bar{P} = L/A_R = N_Z \times W_L / A_R$$

The load factor N_Z can be derived from model tests or from dynamic computer simulations of impact conditions such as that proposed by Band et al[1]. The maximum pressure is found by dividing the average pressure by the reduction factor at the reference area found in Figure 12. For unprotected hulls

$$P_M = \bar{P}/0.09$$

and for structure behind seals

$$P_M = \bar{P}/0.18$$

Pressures for other areas can be found by entering the curves at the appropriate ratio of design area to reference area. Note that this method predicts the maximum pressure behind the seal to be 50 percent of that external to the seal. This is consistent with present assumptions used for hovercraft design in Great Britain. However, for typical design areas, the reduction factors differ enough (Figure 11) that the design limit pressures behind seals may be 70 percent or more of the maximum pressures on the unprotected structure. Craft designed in the United States traditionally have ignored entirely the pressure reduction effects due to the presence of the seals. We tend to take the latter view, since the behavior of seals under impact is not a well-documented subject, and feel that the conservatism included is not a great penalty to pay.

DESIGN EXAMPLE

The Amphibious Assault Landing Craft (AALC) JEFF(A) will be used as an example.

Required JEFF(A) characteristics:

$W_L = 340{,}000$ Pounds

$r_x = 2.31$

$N_Z = 1.1$ (Load factor that will produce 3.0 g's at pilot seat)

The reference area is found to be:

$$A_R = \frac{12.6\,(340{,}000)^{2/3}}{[1+(2.31)^2]^{2/3}}$$

$$= 17{,}926 \text{ IN}^2$$

The average pressure then becomes:

$$\bar{P} = \frac{(1.1)(340{,}000)}{17926} = 20.9 \text{ PSI}$$

The maximum pressure is:

$$P_M = \bar{P}/0.09 = 232 \text{ PSI}$$

The smallest design area (A_D) is 360 in^2

$$A_D/A_R = \frac{360}{17926} = 0.020$$

The pressure reduction factor is:

$\frac{P_D}{P_M} = 0.35$ (taken conservatively from Figure 12 at $A_D/A_R = 0.020$

The design limit pressure then becomes:

$$P_D = 0.35 \times 232 = 81 \text{ PSI}$$

Limit-design pressures derived by this approach for several ACV's are shown in Table 2.

LONGITUDINAL DISTRIBUTION OF PRESSURES

One final comment about the longitudinal distribution of pressures. Admittedly, the pressures predicted for the bow area are high. However, the remainder of the craft will see much lower pressures as can be seen in Figure 13. These results appear to be quite consistent with BHSR criteria when one considers that the SES test craft had a length-to-beam ratio three times that of the craft used to generate BHSR criteria. The only exception would be perhaps the factor of 1.0 should be carried a bit further aft and perhaps the pressures for the majority of the hull relative to the bow could be also lowered more than present practice suggests.

AIR CUSHION VEHICLE	MAXIMUM DESIGN LIMIT PRESSURE REF [18]	MAXIMUM DESIGN LIMIT PRESSURE*
AALC JEFF(A)	62	81
AALC JEFF(B)	50	71
SR-N4	22.5	30
SR-N5	18	21
BH-7	22.5	20

* AS PREDICTED BY SIMPLIFIED METHOD

TABLE 2 - COMPARISON OF DESIGN LIMIT PRESSURES FOR ACV'S WITH THOSE PREDICTED BY METHOD

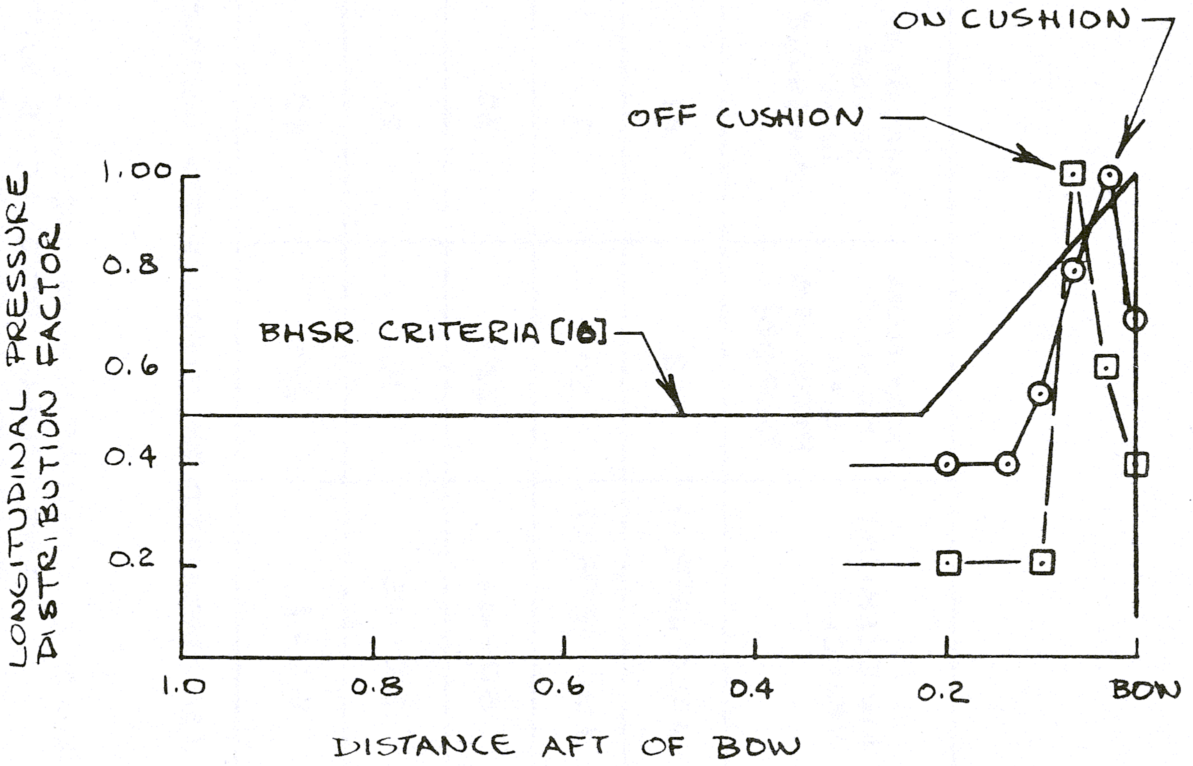

FIGURE 13 - LONGITUDINAL PRESSURE DISTRIBUTION FACTORS DERIVED FROM HIGH L/B SES DATA

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