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REFERENCE NO. 66-16

STRENGTH OF THE ALVIN HULLS

Joseph B. Walsh
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Woods Hole, Massachusetts

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STRENGTH OF THE ALVIN HULLS

by

Joseph B. Walsh and James W. Mavor, Jr.

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TECHNICAL REPORT

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Approved for Distribution E. E. Hays

E. E. Hays, Chairman
Department of Applied Oceanography

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ABSTRACT

Results are presented of pressure tests, measurements and analyses of the strength of the three pressure hulls constructed for the deep submergence vehicle ALVIN. Comparison of stress distribution as measured in various tests and predicted theoretically is made. Failure of the hull can occur by buckling or by yielding over an appreciable fraction of shell thickness or by yielding at a stress concentration. A DTMB analysis predicts collapse of the three hulls No. 1, 2, and 3 at 7040, 7160, and 6720 psi respectively. No. 1 hull has been tested to 4400 psi. From strain measurements, isolated yielding at the inside surface of hull No. 2 (presently in ALVIN) will occur at a pressure of 5800 psi. However, yielding through the entire section would not occur until near the predicted collapse pressure. The maximum Mises equivalent stress at the test pressure of 3300 psi is 62,200 psi. The comparable material yield strength is 110,000 psi. At a stress concentration in way of the hole in the hull for the release mechanism, local stress is estimated from model data to be 93,000 psi at the test pressure of 3300 psi. A close fitting insert of yield strength 125,000 psi is used with hull No. 2. The strength of the plexiglas viewing ports, the electrical lead-throughs and the hull release mechanism are referenced but not discussed.

INTRODUCTION

This report presents a summary of the various pressure tests, measurements, and analyses that have been made as a part of the Deep Submergence Research Vehicle project at the Woods Hole Oceanographic Institution (WHOI) to satisfy us that the ALVIN hulls are structurally adequate. Several groups - David Taylor Model Basin (DTMB), Southwest Research Institute (SWRI), and Litton Systems, Applied Science Division (Litton) - in addition to WHOI have participated in the test program in major ways, and, in most cases, have published project reports covering their own phase of the testing. The present report is meant to be a convenient reference to these project reports; in addition, it includes some unpublished calculations and analyses carried on at WHOI, and a discussion of overall results.

Three hulls - designated Hull no. 1, no. 2, and no. 3 - were fabricated to the same ALVIN design (Fig. 1)and(Walsh and Rainnie, 1964). Although the hulls were meant to be identical, unavoidable variations in material and procedure resulted in differences between the final products, which, although small, cause appreciable differences in hull strength. A preliminary evaluation of the hulls suggested that hull no. 2 should be used in ALVIN, hull no. 1 should be used as a structural test model, and hull no. 3 should be held in reserve. Accordingly, hull no. 1 was subjected to extensive testing, whereas the structural testing of hull no. 2 was more of a confirmatory nature.

The "strength" of a pressure hull cannot be characterized by a single number because failure can occur in more than one way. The collapse pressure, that is, the pressure at which the shell buckles, is a key parameter and, perhaps, the most important. However, failure could be considered to have occurred if yielding over an appreciable fraction of the hull wall thickness should take place; such yielding could be disastrous if the corresponding plastic deformation should result in leakage or jamming of the hull release, for example. Finally, very high stresses over small volumes of material, such as might occur at stress concentrations, may also be dangerous if the material of which the hull is made has a low fatigue life or high notch sensitivity. Thus, to evaluate the strength of the ALVIN hull, three categories must be discussed - collapse pressure, yielding, and stress concentration. Separate sections are devoted to each of these subjects in this report.

COLLAPSE PRESSURE

Spherical shells stressed by external pressure fail by buckling, a type of instability in which a slight increase in pressure causes an enormous increase in the inward deflection of a portion of the shell. The collapse pressure is the pressure at which the first buckle appears at the weakest part of the hull, because one buckle is generally sufficient to cause failure. Thus, to predict the collapse depth of ALVIN, we must know where the weakest section of the hull is located and be able to calculate the buckling strength of that section.

Extensive experimental and theoretical work at DTMB has shown that buckling in apparently uniform spherical shells occurs in regions where there actually is a non-conformity in geometry or material properties. A review of the details is given by Kiernan (1964); but, qualitatively, we may say that local high radius of curvature, low thickness, or low Young's modulus, when they occur over a sufficiently large region, characterize a region of weakness. Accordingly, to predict collapse pressure, one must make a detailed survey of the hull configuration and determine the stress-strain curve for the material. The buckling strengths of the regions which appear weak are computed, and the lowest value determines the collapse strength of the hull.

Such a procedure was followed by DTMB to predict the collapse pressure of the three ALVIN hulls. An excellent report of their procedure and results was prepared by Kiernan (1964). DTMB made over 1200 measurements of radius on each hull and developed maps of hull contour. They measured the stress-strain characteristics of six speciments from each hull prepared from material cut from the hatch and window penetrations. Hahn and Clay supplied the thickness measurements. This data was then used by DTMB to find six regions of weakness for each hull. The weakest in each hull was chosen as the collapse pressure, with the results shown in Table 1.

TABLE 1

Collapse Pressure

Hull No.	Collapse Press. (psi)
1	7040
2	7160
3	6720

DTMB is confident that these values accurately predict the collapse pressures of the ALVIN hull. In the first place, good agreement has been obtained in the past between predicted and measured values of buckling pressures. Also, the analysis tends to be conservative (i.e., the actual buckling pressures should be somewhat greater than those predicted) for the range of shell parameters within which the ALVIN pressure hulls fall. These factors give confidence that the values in Table 1 are the buckling pressures which would be obtained in pressure tests of spherical shells with the same pertinent dimensions - diameter, thickness, out-of-roundness, etc. - as in the ALVIN hulls.

The ALVIN hulls differ from the spherical shells in the DTMB experiments in that the pressure hulls contain inserts in way of windows and hatches which are appreciably thicker than the pressure hull proper. In some cases, the areas in which buckling may be expected lie adjacent to the window inserts. Thus, the buckling pressures predicted by DTMB could be in error if appreciable stresses are induced by the presence of an insert.

The opinion of DTMB, based on strain gage data from a pressure test on the 0.287 scale aluminum model of ALVIN (Bynum and Dehart, 1963) was that the inserts did not cause an appreciable increase in stress; accordingly, no adjustment of predicted buckling pressures was necessary. However, the more comprehensive program of stress measurement later carried out on hulls no. 1 and 2 shows that stresses in some positions on the window insert weld are somewhat higher than those in the sphere proper (see Hull Stresses). Individual gages near the window inserts indicated stress as much as 10-15% greater than the value for gages far removed from discontinuities; thus, a somewhat lower buckling pressure than predicted might be anticipated for "flat spots" adjacent to window inserts. The predicted collapse pressures in Table 1 can still be considered reliable in spite of the insert stresses for several reasons:

1) The high stresses are local and are therefore not completely effective in reducing the buckling strength.
2) The worst flat spot in each hull, which was used to estimate its collapse pressure, did not occur near a window insert. Thus, although the insert stresses might influence a nearby flat spot adversely, the collapse pressure of the hull would not change unless that flat spot should become the critical one.
3) The theory, as mentioned above, tends to underestimate the true collapse depth.

The above discussion gives confidence in the collapse pressures estimated by DTMB. However, direct verification by a pressure test of one of the hulls is still desirable for a vehicle such as ALVIN, where safety of the occupants is of the greatest importance. A pressure test of hull no. 1 was performed at Southwest Research Institute in an attempt to measure collapse pressure, but the tank failed at 4400 psi before the hull collapsed. Strains were measured at pressures up to 4000 psi before the failure. Some of the readings were found to be in error because of malfunction of the automatic recording equipment. The other measurements showed no general departures from linearity, and stresses were well below yield at 4000 psi, as would be expected.

The sudden decrease in pressure when the tank lid blew off lifted the hatch and sheared the hatch hinge bolts. Some damage to the hatch O-ring groove occurred, but the hull was otherwise unharmed. Sphericity of the hull was checked with templates later and found to be satisfactory. Thus, hull no. 1 can still be used for comparison of predicted and actual collapse pressures. Such a comparison will allow one to assume with confidence that the collapse pressure of hull no. 2 is the predicted value.

HULL STRESSES

The stresses in the ALVIN hull, as determined from experimental and theoretical stress analyses, are presented and discussed in this section. A total of five experimental investigations, each consisting of several experiments, were carried out at SWRI. These will be grouped under the headings: .286 Model Tests, Hull no. 1 tests, and Hull no. 2 tests. Two theoretical studies of the hull stresses were also made. A preliminary theoretical analysis of stresses in the window inserts was carried out at Litton. A simple and approximate analysis was made at WHOI; this analysis is presented and compared with the experimental results.

0.286 Model Test

The first test was performed as a preliminary proof test of the window insert reinforcement design (stress around penetrations through the insert were also measured and are covered under STRESS CONCENTRATIONS). A summary of the report on this test prepared by Bynam and De Hart (1963) is presented below.

A 0.286 model of half of the ALVIN hull with one window located at the pole was machined from an available 7079-T6 aluminum hemisphere. The window insert configuration which was eventually used on ALVIN and one with the same shape but a smaller window diameter were tested. Strain gages to measure meridianal (with respect to the axis of the window) and hoop strains were placed on the inside and outside surfaces along two meridians separated by 90°. The model was subjected to external pressure in the SWRI tank, strains were recorded, and stresses were calculated from the strains. The results for only the present window configuration are shown in Figures 2 and 3. In the figures, the magnitude of the stresses at test depth (3300 psi or approximately 7500 ft. of sea water) as extrapolated from data at 1500 psi is plotted as a function of gage position. Distances in Figures 2 and 3 refer to full scale dimensions to faciliate comparison with data on hulls no. 1 and 2 presented later in this section.

The stresses in Figures 2 and 3 are evidently quite reliable, judging by the small difference in values at corresponding positions along the two different meridians. The normal stress in the uniform shell is the membrane stress σ_m, given by

$$\sigma_m = \frac{p\, r_0^2}{2\, r_m h}$$

where p = pressure = 3300 psi

r_o = outside radius = 41 in.

r_m = mean radius = 40.33 in.

h = thickness = 1.33 in.

Thus, $\sigma_m = 3300 \dfrac{(41)^2}{(2)\ (40.33)\ (1.33)}$

$= 51{,}500$ psi

Figures 2 and 3 show that stresses at points removed from the reinforced region closely approximate the theoretical value.

The error incurred by using an aluminum model to determine stresses in a steel structure should be negligible, in principle. The only characteristic of the prototype which was not accurately modeled was Poisson's ratio, which differed by about 10 per cent (0.3 for steel, 0.33 for aluminum). However, Poisson's ratio enters the calculations through the term $(1-\upsilon^2)$, so the 10 per cent difference resultes in an error of only one or two per cent. A more important lack of similitude arises from the impossibility of scaling the effect of welding stresses and distortion in the model. Measurement of stresses in a full scale model and in the prototype is desirable.

Hull No. 1 Tests

Hull No. 1 was subjected to three pressure tests during which strain gage data was taken. These will be referred to as the 2500 psi test, the 3300 psi test and the 4400 psi test. The 2500 psi test and the 3300 psi test were designed to obtain a detailed description of stresses in the hull; and therefore a large number of gages, over 350, were used. The 2500 psi test was undertaken prior to pressurizing to the 3300 psi test pressure to guarantee that no unforseen high stresses might be encountered. The stresses in both tests were found to be well below the yeild strength of the steel. The 4400 psi pressurization was supposed to be carried to collapse; however, the tank failed at 4400 psi, and data was taken only to 4000 psi. The results of these three tests are presented below.

2500 psi Test. The 2500 psi test data was not analyzed at SWRI because they felt that it was superseded by the 3300 psi data. However, the 2500 psi data was found to be of use when questions about the validity of certain 3300 psi data points arose, and therefore most of the 2500 spi strain data was reduced at WHOI. The stresses at hatch and window reinforcements are shown graphically in Figures

4, 5, 6, and 7. The values in the figures are linear extrapolations to the 3300 psi test depth of the 2500 psi data. The general location of the gages are indicated on the graphs; more exact description can be found in the SWRI report (Briggs and De Hart 1964).

3300 Psi Test. Stresses measured at the window inserts and at the hatch insert are plotted in Figures 8-11 and 14-17. The values plotted in these figures generally are the values published in the SWRI report. In some cases, however, the published values were found to be incorrect and were recalculated from the original data. Thus, occasionally points in the figures will be found to disagree with the SWRI values. Several pressurizing cycles were necessary in order to read all the gages. In one run, the pressure was raised only to 2700 psi, instead of 3300 psi. Data from this run was extrapolated to 3300 psi for plotting.

Stresses at gage positions on flat spots along the equatorial weld were found to be near the theoretical membrane stress, and consequently are not of further interest here.

Collapse Test. The main purpose of this test was to determine, if possible, the pressure at which yielding of the hull began from the pressure-strain curves. As mentioned previously, the pressure was not raised above 4400 psi, and strain gage data was taken only to 4000 psi. No yielding was expected to occur at this pressure, and indeed no nonlinearity in the pressure-strain curves, other than that which could be ascribed to experimental error, could be determined.

Hull No. 2

Two tests, called here the 1st run and the duplicate run, were made on Hull

No. 2. Although these tests were made on the same day, two separate pressurizing cycles were involved. The hull in the first run was pressurized to 3300 psi. In the duplicate run, the highest pressure which could be attained was 2970 psi. The data for the duplicate run has been linearly extrapolated to 3300 psi for comparison with the first run.

The results presented in the SWRI report on Hull No. 2 (Briggs and De Hart, 1964) will be found to differ from those given in the present report. At the time of publication, the WHOI group felt that the stresses in the SWRI report were incorrect because the code used for referring recorder readings to gage positions on the hull was not correct. At the time of the test, the code was copied down by the WHOI representative, J. W. Mavor, Jr.; the code in the SWRI report did not agree with WHOI's version. The question of whose code was correct was resolved to our satisfaction by recalculating all the data on Hull No. 2 by means of the WHOI code. The stresses determined from the WHOI code were found to agree with stresses measured on Hull No. 1 and on the 0.286 model, whereas the SWRI version did not. Therefore, the results of the WHOI calculations are presented here in Figures 12 through 19.

YIELD STRENGTH

The highest stresses recorded were from gages mounted in Hull No. 2. In the SWRI report on the Hull No. 1 stress analysis, a stress of 95,500 psi (at 3300 psi pressure) is reported; however, inspection of the original data shows that this value is unreliable because of a large shift in the zero calibration during the course of the run.

As mentioned previously, SWRI calculated stresses in Hull No. 2 using a different code than that used by WHOI. However, in no part of the hull were the stresses much greater than the membrane value over a significant area (this is valid using either code). Thus, premature yielding over an appreciable area of the shell, which would affect the DTMB analysis of collapse depth will not occur. This is shown by data (from WHOI code) tabulated below in Table 2 from three isolated gage positions in hull 2 at which stresses most nearly approached yield.

TABLE 2

Maximum Stresses in Hull No. 2 at 3300 psi.

Position	Location	σ_1, psi	σ_2, psi
134	Stbd. window weld, inside	63,700	52,600
334	Equator weld flat spot, inside	65,400	58,100
243	Port window flat spot, outside	64,900	49,100

The pressure at which yielding begins can be found by comparing the compressive yield stress with the "equivalent" stress from the Mises yield criterion computed for the three locations in Table 2. The equivalent stress $\bar{\sigma}$ under biaxial stressing, given by $\sqrt{\tfrac{1}{2}(\sigma_1^2 + \sigma_2^2)}$ is presented in Table 3.

TABLE 3

Equivalent Stress at 3300 psi from Mises Criterion

Position	$\bar{\sigma}$
134	59,000
334	62,200
243	58,700

The minimum yield stress at 0.2% offset for any specimen taken from the hemispheres was found to be 110,000 psi (Mavor, January 1965). The pressure at which yielding begins at the most highly stressed position (334) is

$$\sigma_y = \frac{110,000}{62,200} \; 3300$$

$$= 5800 \text{ psi}$$

Thus, yielding in the shell begins at an isolated point at a pressure greater than 5800 psi. General yielding through the section or over an appreciable area would not occur until an appreciably higher pressure were applied. For example, the stresses are lower than the membrane value on the inside; calculation shows that yielding through the entire section would not occur until the external pressure was very nearly equal to the collapse pressure. Therefore, premature yielding of the shell appears to be unlikely, and the DTMB collapse pressure estimates can be accepted as valid. In addition, we see that testing at a pressure of 3300 psi will result in no leakage or jamming due to large scale plastic flow because yielding does not occur until the pressure is in excess of 5800 psi.

STRESS CONCENTRATION

A number of gages were placed near areas of anticipated stress concentration such as bolt holes and other penetrations. The stresses indicated by these gages were well nelow the membrane stress in all cases. There is some doubt whether the strain registered by the gage did indeed indicate the effect of the stress concentration at all. The gage could not be placed right at the edge of a penetration because of the finite size of the gage and, in some cases, because of interference with parts of the submarine itself. The stress field due to a stress concentration decays sharply with distance, so the gage probably indicated stress only slightly higher than the average level in that region. The average level of stress several diameters away from a penetration was generally low because penetrations were located in reinforced areas.

An attempt was made to extrapolate from values measured near the edge of a penetration to the value at the penetration itself. Analysis of the stress at the penetration for the hull release mechanism is given in the Appendix. The maximum stress in this region estimated from model test data was found to be about 93,000 psi at test depth. This value, although less than the yield stress at 0.2% offset of 110,000 psi was considered to be too high for such a critical location. Accordingly, a close fitting insert of 17-4 PH Steel having a yield strength of 160,000 psi was used. In hull No. 2 this was later replaced with K-Monel of yield strength 125,000 psi in a redesigned hull release.

An indication that stress concentrations are not severe is given by the condition of the tapped holes after the test to 3300 psi. No trouble in removing or replacing bolts was experienced, suggesting that local stresses were not high enough to cause yielding.

DISCUSSION

The evidence that has been presented above indicates that the strength of the ALVIN pressure hulls is more than adequate. The safety factors (safety factor is the ratio of the collapse pressure to operating pressure) for hulls 1, 2, and 3 are 2.6, 2.7, and 2.5 respectively. These values are greater than originally specified in the preliminary design and are very generous when compared with the factor of safety generally specified in military submarines or other vehicles similar to ALVIN. Thus, the indications are that the present design exceeds the structural requirements in all respects.

Attempts will inevitably be made to improve the present hull configuration if other vehicles of the same general specifications as ALVIN are ever considered. The results shown in this report suggest several areas where possible changes might be warranted. First, the nominal thickness of the hull in the present design might be decreased considerably. This would decrease the collapse pressure, but, as mentioned above, the present safety factor is rather generous. A considerable saving in hull weight could be derived in this way while still maintaining an adequate margin of safety. A standard has not yet been established. If it is necessary, of course, that the same hight fabrication standards which prevaile in the construction of the present hulls be maintained in order that out-of-roundness be at an acceptable low level in a new design with a thinner shell.

Another possibility to consider is reducing the diameter of the window inserts. Flat spots seemed to occur in the areas where the insert welds were in closest proximity. Increasing the separation between these welds probably would reduce the tendency for distortion in these regions. If the thickness of inserts is kept at the present value, decreasing the diameter of the insert has the effect of making the transition from hull to maximum reinforcement thickness somewhat sharper. Often increasing the severity of a transition has the effect of increasing the maximum stress at the discontinuity. However, the stresses at the discontinuities in the present design are close to the membrane value, and the possible increase in stress due to a sharper transition could be offset by a decrease in stresses due to welding distortion.

It would be very desirable to have available stress values predicted from elastic shell theory to compare with the experimental values in the regions around inserts. Predicting the stresses in the ALVIN hull requires analysis of spherical shells with non-uniform thickness. Litton attempted to solve this difficult problem, but their results were found to disagree with the experimental values. A simpler approach was tried by the WHOI group. The analysis, which is presented in the Appendix, produced results which agree in a very approximate way with the observed behavior: as shown in Figures 20 and 21, the theoretical results fall near the mean of the values for the inner and outer surfaces. Such results are not adequate for accurate prediction of maximum stresses, however, and the theoretical stress analysis of the pressure hull must be left an unanswered problem.

Finally, in discussing the strength of the ALVIN pressure hulls, we must consider the integrity of several components which are not a part of the hull proper. The plexiglass viewing parts were tested in a special jig to a pressure above the hull collapse depth without failing (Mavor, December 1965). In addition, the electrical lead-throughs and the hull release mechanism were pressure tested separately (Mavor and Sharp, August 1965) as well as in the unmanned dive to 7500 ft.

ACKNOWLEDGEMENTS

The contributions of several persons to the conduct of the tests and analysis reported herein are gratefully acknowledged. They include Harold E. Froehlich of Pragrammed and Remote Systems, Inc., formerly GMI and Litton; Dr. Robert C. De Hart, Edward M. Briggs, and Douglas Bynum of Southwest Research Institute; Martin A. Krenzke and Thomas J. Kiernan of the David Taylor Model Basin; and Larry Megow of Hahn & Clay Inc.

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APPENDIX

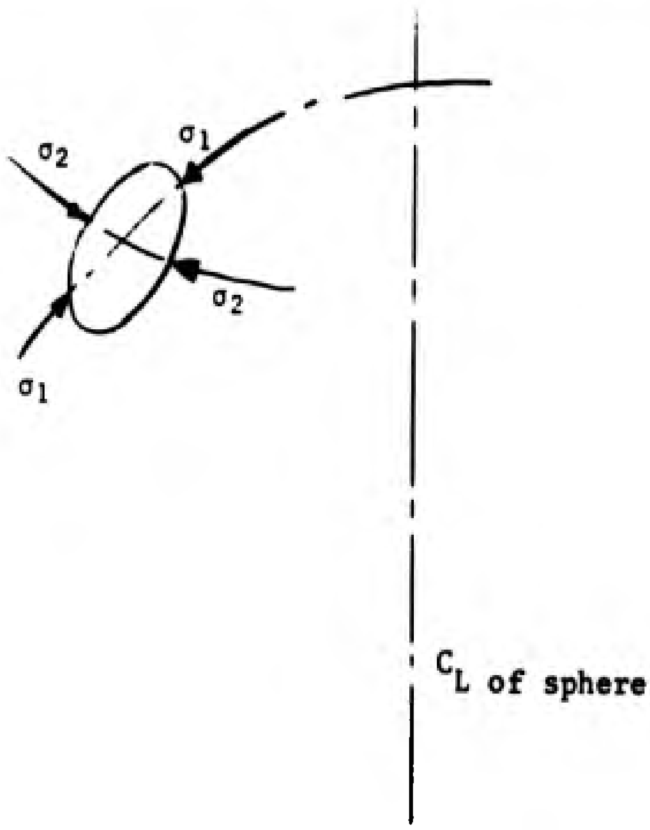
STRESS AT THE HULL RELEASE PENETRATION

Tests were run at SWRI on a 0.286 scale model of the port insert to measure stresses. Gages mounted at a hole simulating the hull release penetration did not measure the highest stress at the hole for two reasons - the gage had to be placed about 0.04" from the hole and only the circumferential strain (circumferential with respect to the penetration) was measured. The following analyses estimate the maximum stress actually existing at the hole by two methods.

A. Stress Concentration Factor

The first method makes use of the nominal stresses in the region of the hole and the stress concentration factor.

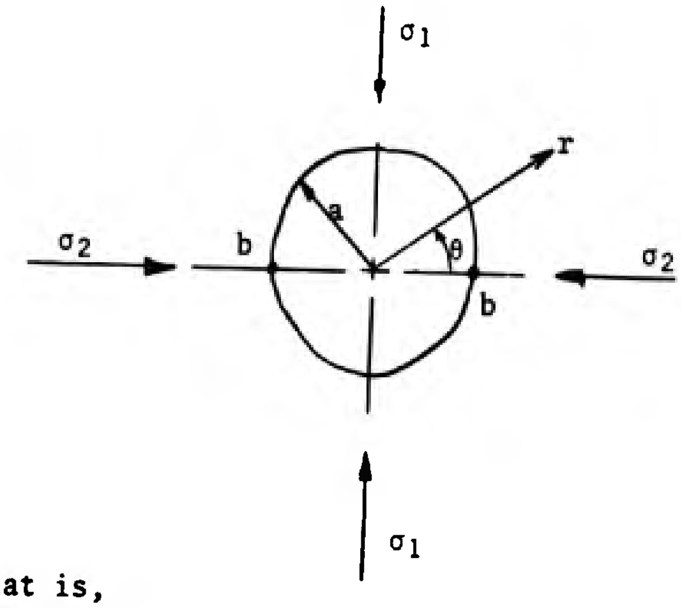
The nominal stresses are found from the SWRI report, Fig. 13, p. 33.



$\sigma_1 = 13.3p$

$\sigma_2 = 11.8p$

Theory shows that the highest stress at a small hole in a biaxially stressed plate occurs at points "b" (see below) and has the magnitude $3\sigma_1 - \sigma_2$ (for $|\sigma_1|>|\sigma_2|$)



that is,

$$(\sigma_\theta)_{max.} = 3\sigma_1 - \sigma_2 = 3(13.3p) - 11.8\ p$$
$$= 28.1\ p$$

B. The second method makes use of the measured value of the circumferential strain and the theoretical value of the radial strain to find the actual circumferential stress at the gage location. This value is then extrapolated to the edge of the hole using the theoretical stress distribution to find the maximum value.

The calculations were simplified by expressing the stress distributions in terms of a co-ordinate "Δ", the distance from the edge of the hole. Since Δ/α is small, the stress distributions could be expressed to an accuracy of greater than ± 5% by simple expressions:

$$\sigma_r = \frac{\Delta}{a}\left(3\sigma_1 - \sigma_2\right)$$

and

$$\left(\sigma_\theta\right)_{max} = \left(\sigma_\theta\right)_{a+d} + \frac{\Delta}{a}\left(7\sigma_1 - 5\sigma_2\right)$$

Thus, the radial stress at the gage is, for a=0.192" and Δ=0.04"

$$\sigma_r = \frac{0.04}{0.192}\,[3(13.3) - 11.8]p$$

$$= 6.0 \text{ p}$$

The circumferential strain at the gage is given by

$$E\varepsilon_\theta = \sigma_\theta - \mu\sigma_r.$$

The value of $E\varepsilon_\theta$ measured by SWRI is given by their value of σ_θ (see their report, p. 45, for gage 213).

$$\left(E\varepsilon_\theta\right)_{meas.} = 19.0 \text{ p}$$

therefore, for $\mu = 0.3$

$$\left(\sigma_\theta\right)_{a+d} = 19.0 \text{ p} + 0.3(6.0)\text{p}$$

$$= 20.8 \text{ p}$$

The maximum stress at the hole is thus

$$\left(\sigma_\theta\right)_{max} = 20.8 \text{ p} + \frac{0.4}{0.192}\,[7(13.3) - 5(11.8)]$$

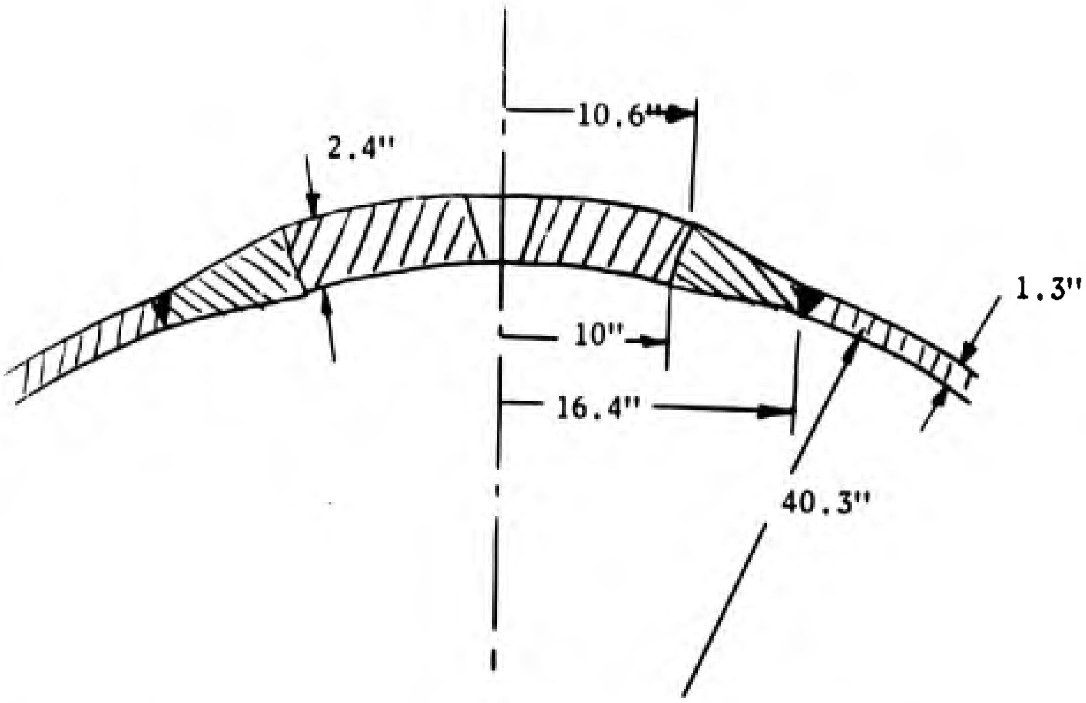
$$\underline{= 27.9 \text{ p}}$$

The two methods give values which agree very closely. The maximum stress at the test depth pressure of 3330 psi is

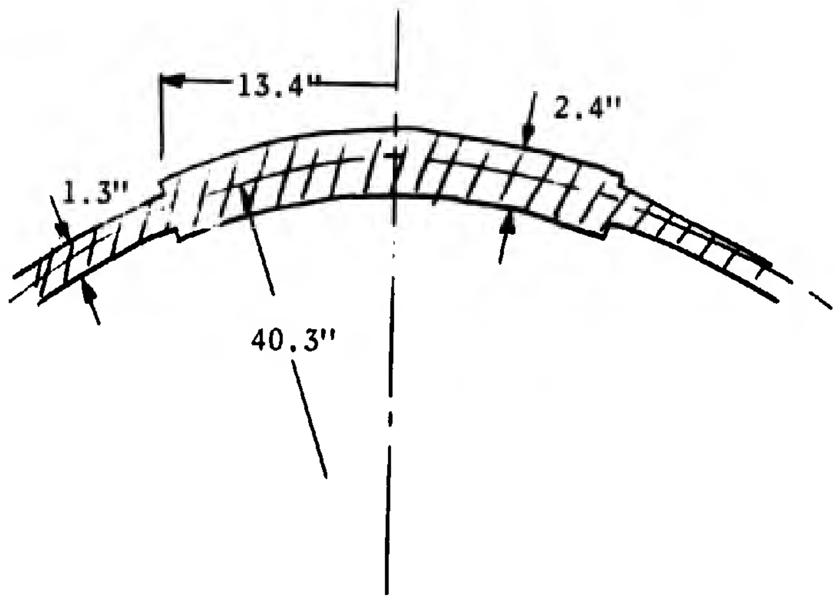
$$\sigma_{max} = (28.1)(3330)$$

$$= 93{,}500 \text{ psi.}$$

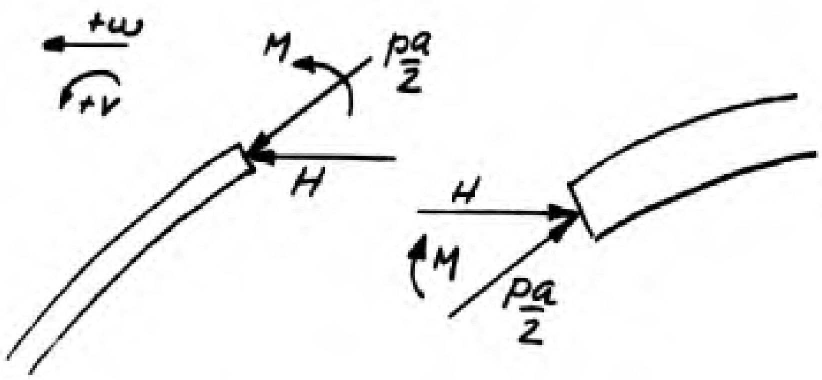
STRESSES IN HATCH



For the sake of simplifying the analysis, assume that the hatch and hatch insert form a solid shallow shell of constant thickness. Assume that the diameter of this shallow shall is the mean diameter of the insert. The effect of the port on the edge coefficients is ignored.



Calculate the discontinuity loads by dividing at the junction.



The edge coefficients for the sphere are found using the Esslinger approximation (see Ref. 1* for details of the following calculation).

$$4\rho^4 = 12(1-\ ^2)\ \frac{a^2}{h^2}$$

a = mean radius

h = thickness

= 0.3

$$= 12(0.91)\left(\frac{40.3}{1.3}\right)^2$$

$$= 10{,}520$$

$$2\rho^2 = 102.5$$

$$\rho\sqrt{2} = 10.13$$

$$\sin\phi = \frac{13.4}{40.3} \quad \text{where } \phi = \text{angle of opening}$$

$$= 0.332$$

$$\phi = 19.4^\circ$$

$$= 0.338 \text{ rad.}$$

$$\phi\rho\sqrt{2} = 3.42$$

*Appendix references are given at end of appendix.

From Ref. 1, p. 8 and Ref. 2, p. 252.

$$\psi_3(X) = \mathcal{R}e\ H_0'(3.42i) = 0.01292$$

$$(X) = \mathcal{I}m\ H_0'(3.42i) = 0.03518$$

$$(X) = \frac{d}{dX}\mathcal{R}e\ H_0'(3.42i) = 0.03600$$

$$(X) = \frac{d}{dX}\mathcal{I}m\ H\ (3.42i) = -0.02056$$

From Ref. 1,

$$X_e = (\psi_3\psi_4{}^1 - \psi_4\psi_3{}^1) + \frac{0.75}{\sqrt{2}\ \phi_0}(\psi_3{}^12 + \psi_4{}^12)$$

$$= [(0.01292)(-0.02056) - (0.03518)(-0.036)$$

$$+ \frac{0.75}{3.42}[(-0.036)^2 + (0.02056)^2]$$

$$= -2.655 \times 10^{-4} + 12.66 \times 10^{-4}$$

$$+ 0.2193(12.96 \times 10^{-4} + 4.21 \times 10^{-4})$$

$$= 10.005 \times 10^{-4} + 3.592 \times 10^{-4}$$

$$= 13.597 \times 10^{-4}$$

$$X_c = \frac{(\int\sqrt{2}\ \phi_0)^2\ (\psi_3\psi_3{}^1 + \psi_4\psi_4{}^1)}{X_e}$$

$$= \frac{(3.42)^2\ [(0.01292)(-0.036) + (0.03518)(-0.02056)]}{13.6 \times 10^{-4}}$$

$$= 11.70\ \frac{-11.88}{13.6}$$

$$= -10.22$$

$$X_d = \frac{(\mathcal{S}\sqrt{2}\ \phi_0)(\psi_3{}^{1\,2} + \psi_4{}^{1\,2})\ 2\sqrt{3(1-\nu^2)}^{1}}{X_e}$$

$$= \frac{(3.42)(17.17 \times 10^{-4})(3.31)}{13.6 \times 10^{-4}}$$

$$= \quad 14.3$$

$$W_c = \frac{(\mathcal{S}\sqrt{2}^1\ \phi_0)^3(\psi_3{}^2 + \psi_4)^2 - 2(\mathcal{S}\sqrt{2}^1\ \phi_0)^2(\psi_3{}^1\psi_4 - \psi_4{}^1\psi_3) + \frac{15}{16}(\mathcal{S}\sqrt{2}\ \phi_0)(\psi_3{}^{1\,2}+\psi_4{}^{1\,2})}{X_e\ 2\sqrt{3(1-\ {}^2)}}$$

$$= \frac{(3.42)^3[(0.01292)^2 + (0.03518)^2] + 2(3.42)^2(10.00)\times 10^{-4} + \frac{15}{16}(3.42)(17.17\times 10^{-4})}{(13.6 \times 10^{-4})(3.31)}$$

$$= \quad 18.9$$

The edge coefficients are (note change in signs)

$$\frac{EV_s}{H} = \frac{-X_c}{h\phi_0} = \frac{+10.22}{1.3(0.338)} = +23.2$$

$$\frac{EV_s}{M} = \frac{X_d}{h_s{}^2\phi_0} = \frac{14.3}{(1.3)^2(0.338)} = +\ 25.0$$

$$\frac{EW_s}{H} = \frac{WC}{\phi_0} = \frac{18.9}{0.338} = +55.9$$

$$\frac{EW_s}{M} = \frac{X_c}{h\phi_0} = \frac{+10.22}{1.3(0.338)} = +23.2$$

The edge coefficients for the shallow spherical shell insert are found from Ref. 3. The ordinates of Fig. 3 and Fig. 4 in Ref. 3 may be written as:

$$-\frac{Eh}{\alpha a H^1(1-\nu)} = -\frac{EW}{H^1}\left[\frac{h}{\phi_0 a(1-\nu)}\right] = \frac{-EW}{H^1}\ \frac{2.4}{(0.338)(40.3)(0.7)}$$

$$= -\ 0.256\ \frac{EW}{H^1}$$

$$\frac{24Eh\ (1+\nu)}{pa^2\ \alpha\mu^4} = \frac{E\omega}{p}\ \frac{24h(1+\nu)}{a^2\ \phi_0\mu^4} = \frac{24(2.4)(1.3)}{(40.3)^2(0.338)(37.6)}\ \frac{E\omega}{p}$$

$$= 0.00368\ \frac{E\delta}{p}$$

$$\frac{4(1+\nu)\ Eh^3\ ^2}{\mu^2\ \alpha a^2 m^4\ M} = \frac{E\omega}{M}\ \frac{a}{3(1-\nu)}\left(\frac{h}{b}\right)^3 = \frac{40.3}{3(0.7)}\ \left(\frac{2.4}{13.4}\right)^3\ \frac{E\omega}{M}$$

$$= 0.1105\ \frac{E\omega}{M}$$

$$\frac{4(1+\nu)\ Eh\ V}{\mu^4\rho H} = \frac{EV}{H^1}\ \ \frac{a}{3(1-\nu)}\left(\frac{h}{b}\right)^3 = 0.1105\ \frac{EV}{H^1}$$

$$-\ \frac{8(1+\nu)\ EhV}{\mu^2\alpha a\rho^2 p} = -\ \frac{EV}{p}\left[\frac{2}{3(1-\nu)}\ \left(\frac{h}{b}\right)^3\right] = -\ 0.00548\ \frac{EV}{p}$$

$$-\ \frac{(1+\ \ )\ Eh^3V}{\alpha am^4M} = -\ \frac{EV}{M}\left[\frac{h^3}{12(1-\nu)b}\right] = -\ 0.1228\ \frac{EV}{M}$$

The abscissa μ is defined as

$$\mu = \frac{b}{\sqrt{ah^1}}\ \ 12(1-\nu^2)^{\frac{1}{4}}$$

$$= \frac{13.4}{\sqrt{40.3(2.4)}}\ \ [12(0.91)]^{\frac{1}{4}}$$

$$= 2.48$$

The values of the ordinates in Fig. 3 and Fig. 4 are

$$-0.256\ \frac{E\omega}{H^1} = 0.63 \qquad \text{or} \qquad \frac{E\omega}{H^1} = -13.28$$

$$0.00368\ \frac{E\omega}{p} = 3.4 \qquad \text{or} \qquad \frac{E\omega}{p} = 171.2$$

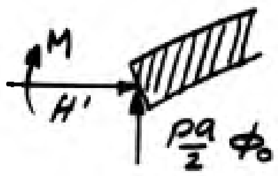
$$0.1105\,\frac{E\omega}{M} = 0.71 \qquad \text{or} \qquad \frac{E\omega}{M} = 6.42$$

$$0.1105\,\frac{EV}{H^{I}} = 0.71 \qquad \text{or} \qquad \frac{EV}{H^{I}} = 6.42$$

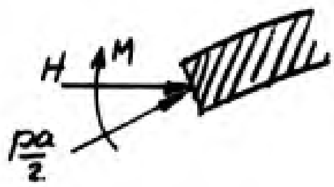
$$-0.00548\,\frac{EV}{p} = 0.71 \qquad \text{or} \qquad \frac{EV}{p} = -129.5$$

$$-0.1228\,\frac{EV}{M} = 0.77 \qquad \text{or} \qquad \frac{EV}{M} = -6.26$$

Note that a different set of edge loads are used in Ref. 3:



Ref. 3 notation



Notation of this report

It can be seen that the two notations are the same for

$$H^1 = H + \frac{pa}{2}\cos\phi_0$$

$$= H + 19.05\ p$$

The edge deflections of the hatch are thus

$$E\omega_H = -13.28\ (H + 19.05p) + 6.42\ M + 171.2\ p$$

$$EV_H = 6.42\ (H + 19.05p) - 6.26\ M - 129.5\ p$$

or

$$E\omega_H = -13.28\ H + 6.42M - 81.8\ p$$

$$EV_H = 6.42\ H - 6.26M - 7.3\ p$$

The deflections of the sphere are given by (see p. 4)

$$E\omega_s = 55.9\ H + 23.2\ M - p\ \frac{a^2(1-\nu)}{2h}\ \sin\phi_0$$

$$= 55.9\ H + 23.2\ M - p\ \frac{(40.3)^2(0.7)}{2(1.3)}\ (0.332)$$

$$= 55.9\ H + 23.2\ M - 145.5\ p$$

$$EV_s = 23.2\ H + 25.0\ M$$

Since $\omega_s = \omega_H$:

$$-13.28H + 6.42M - 81.8p = 55.9H + 23.2M - 145.5M$$

$$69.2H + 16.8M = 63.7\ p$$

Since $V_s = V_H$:

$$6.42H - 6.26M - 7.3p = 23.2H + 25.0M$$

$$16.8\ H + 31.3M = -7.3p$$

Solving these equations gives

$$H = -\frac{31.3}{16.8}M - \frac{7.3}{16.8}p$$

$$= -1.862M - 0.435p$$

$$69.2(-1.862M - 0.435p) + 16.8M = 63.7p$$

$$-129M - 30.1\ p + 16.8M = 63.7p$$

$$-112.2\ M = 93.8\ p$$

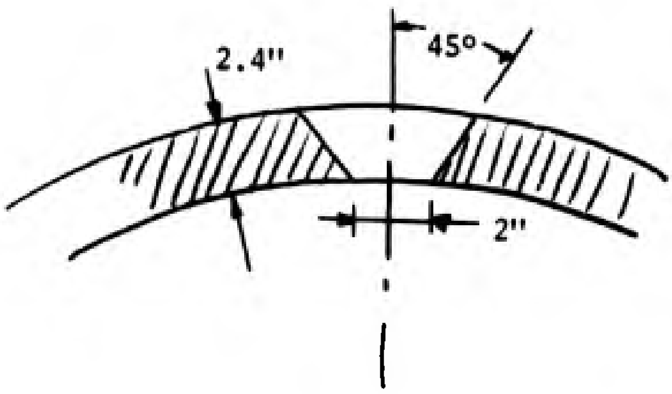
$$\underline{M = -0.84\ p}$$

$$H = -1.862\ M - 0.435\ p$$

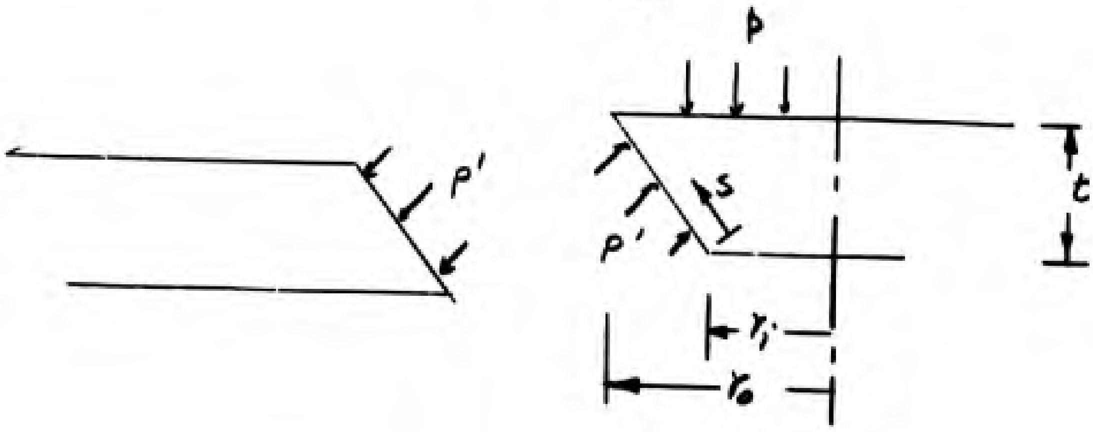
$$\underline{H = 1.14\ p}$$

These stress resultants are small compared with the membrane load of 20.2 p. The error in neglecting them entirely is less than the error in assuming a sharp discontinuity in shell thickness where actually a gradual transition occurs. The stress distribution through the hatch insert-hatch transition is thus predicted to be that found from simple membrane analysis.

The Hatch Port



Assume that the low-modulus plastic window conforms to the hole in the hatch such that it exerts a uniform pressure p' on the window seat. Also assume that the window will not exert an appreciable frictional shear stress on the seat.



The stress p^1 is found considering the equilibrium of the window.

$$2\int_0^{r_0} p\pi r dr = p^1 \int_0^{t\sqrt{2}} \left(\frac{1}{\sqrt{2}}\right) \left(r_i + \frac{s}{\sqrt{2}}\right) 2\pi ds$$

$$p\pi r_0^{\,2} = p^1 \left[\frac{2\pi}{\sqrt{2}} \left(r_i \; s + \frac{s^2}{2\sqrt{2}}\right)\right]_0^{t\sqrt{2}}$$

$$= \frac{2\pi}{\sqrt{2}} \left(r_i \; t\sqrt{2} + \frac{t^2}{\sqrt{2}}\right)$$

$$= 2\pi \; p^1 \; t \; r_M \quad \text{where} \quad r_M = \frac{r_0+r_i}{2}$$

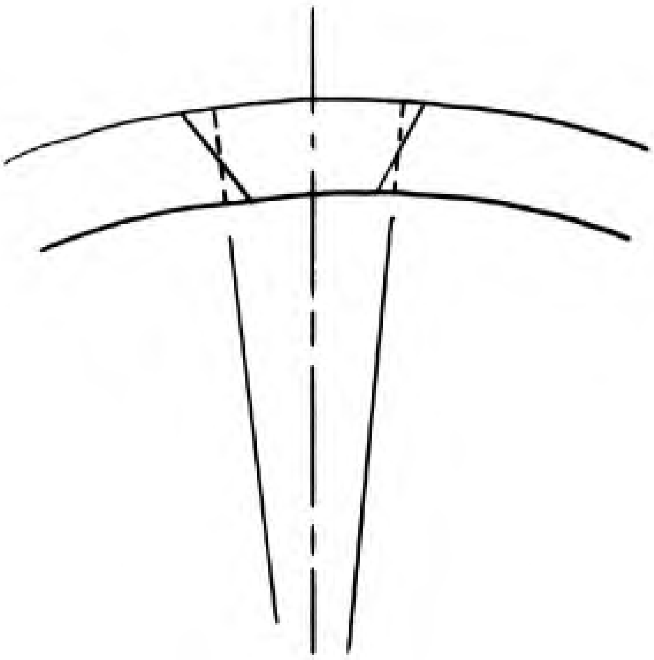
or

$$p^1 = p \, \frac{r_0^{\,2}}{2tr_M}$$

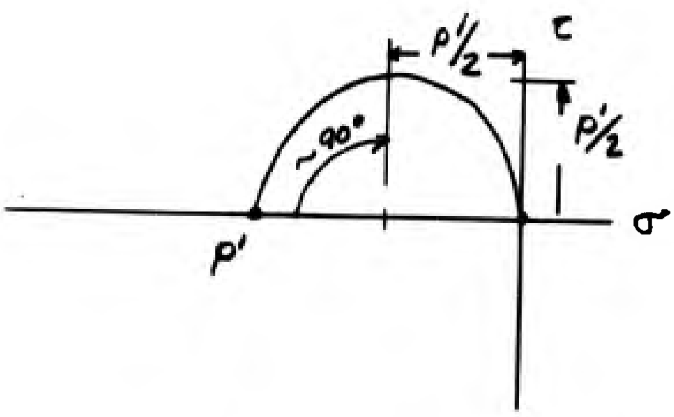
$$= p \frac{(3.4)^2}{2(2.4)(2.2)}$$

$$= 1.1 \; p$$

In order to analyse the effect of this stress in the material in the hatch cover away from the window, replace the port opening by an opening with the same mean diameter forming a cone with the center of the sphere as apex.



The stresses on this equivalent opening are found from Mohr's circle. The angle the new surface make with the vertical can be assumed to be zero. The stresses on the equivalent opening are:



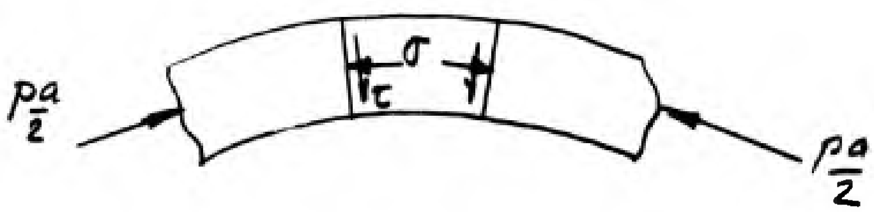
$$\tau \approx 0.5\ p^1$$

$$\sigma \approx 0.5\ p^1$$

or

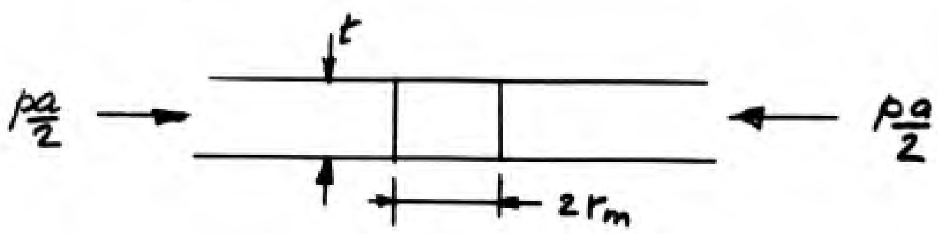
$$\tau \simeq 0.5 \text{ p}$$

$$\sigma \simeq 0.5 \text{ p}$$



The stress resultants $N_\phi = \sigma t = 1.2p$ and $Q_\phi = 1.2p$ are much smaller than the membrane load of about 20p; therefore, the stresses applied by the window may be assumed to be zero.

The port can be treated as an unstiffened opening in a shallow plate under external pressure and a membrane condition at the edge. Since the hatch is only slightly dished, an approximation to the stresses at the window can be found by considering the hatch to be a flat plate under action of the edge load.



The hoop stress at the hole for a plate under equal biaxial stress is twice the applied stress. Thus the hoop stress at the hole is given by $\frac{pa}{t}$.

or

$$\sigma_{port} = \frac{40.3}{2.4} p = 16.8 p$$

This maximum hoop stress at the hole diminishes rapidly with distance from the hole. At about one or two hole radii away from the edge the hoop stress has diminished to the applied stress. Since the effect of approximating the port by a cone with the sphere center as apex would be felt one or two diameters from the hole, the stress calculated above should only be used as an indication of stresses actually existing at the hatch port.

Appendix References

1. Galletly, G. D., "Analysis of discontinuity stresses adjacent to a central circular opening in a spherical shell." DTMB Report 870, May, 1956.

2. Timoshenko, S., and Woinowsky-Krieger, S., Theory of plates and shells, McGraw-Hill, New York, 1959.

3. Williams, H. E., "Influence coefficients of shallow spherical shells." Jet Propulsion Lab. Tech. Rept. No. 32-51, February, 1961.

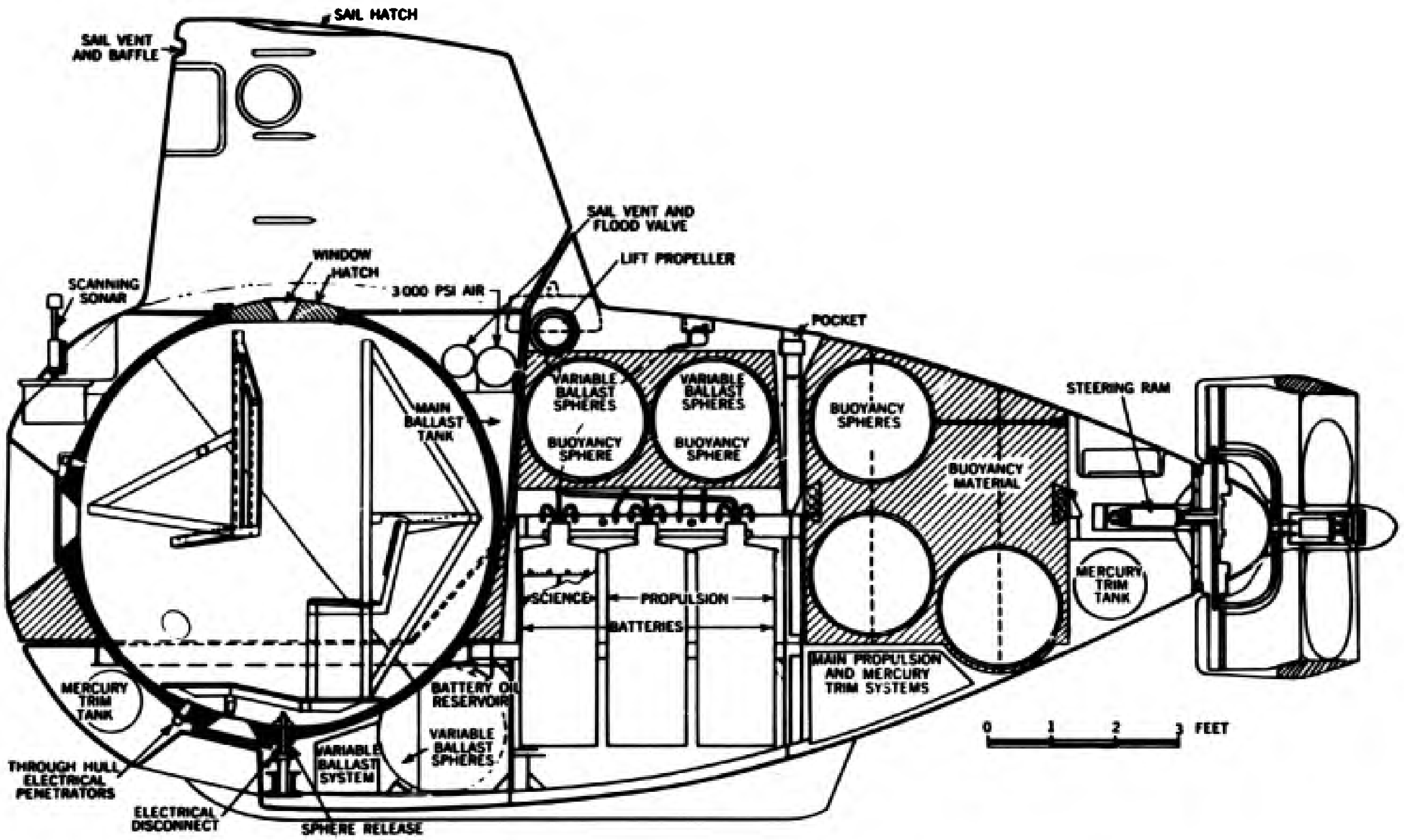


Figure 1. ALVIN inboard profile

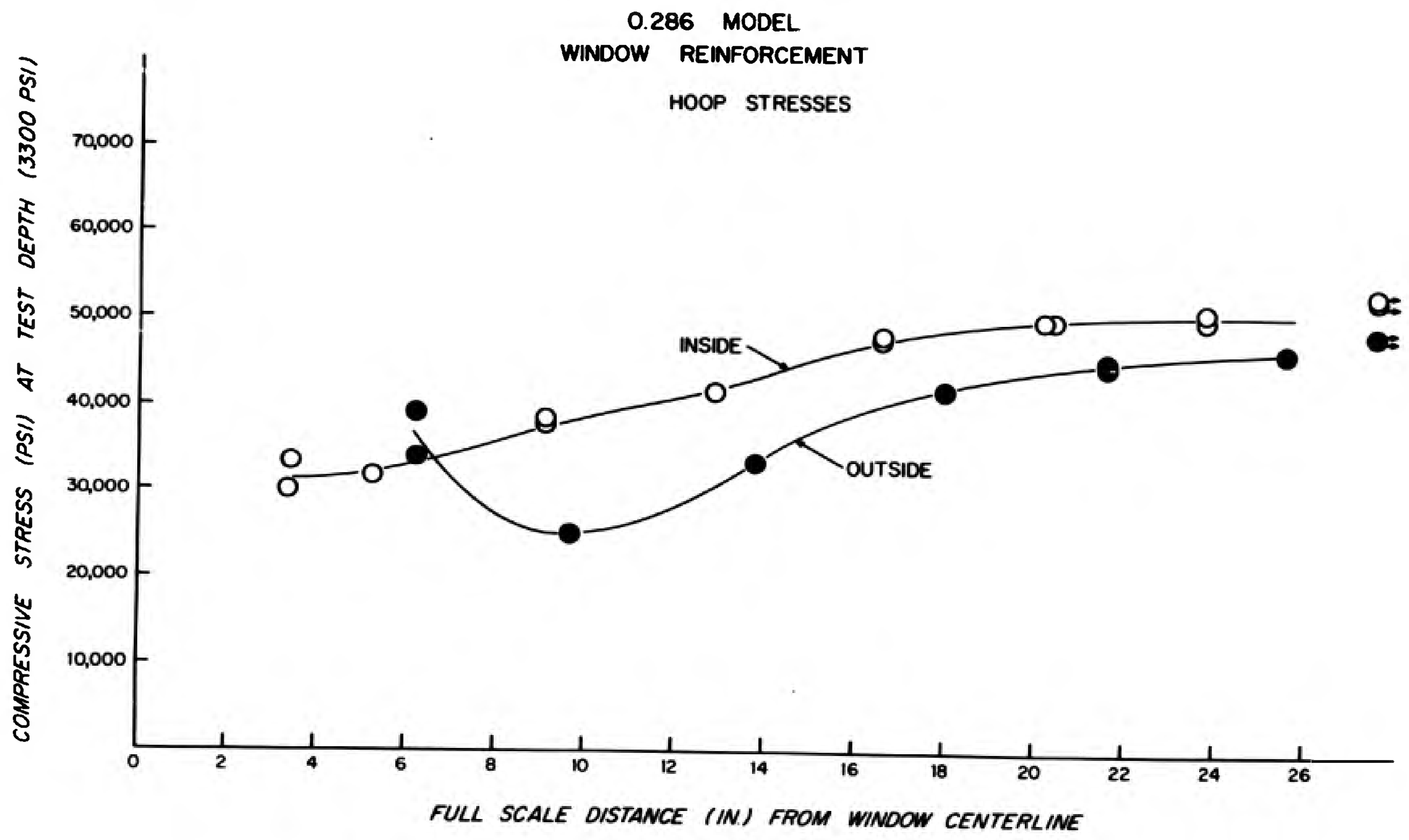


Figure 2. 0.286 model window reinforcement, hoop stresses.

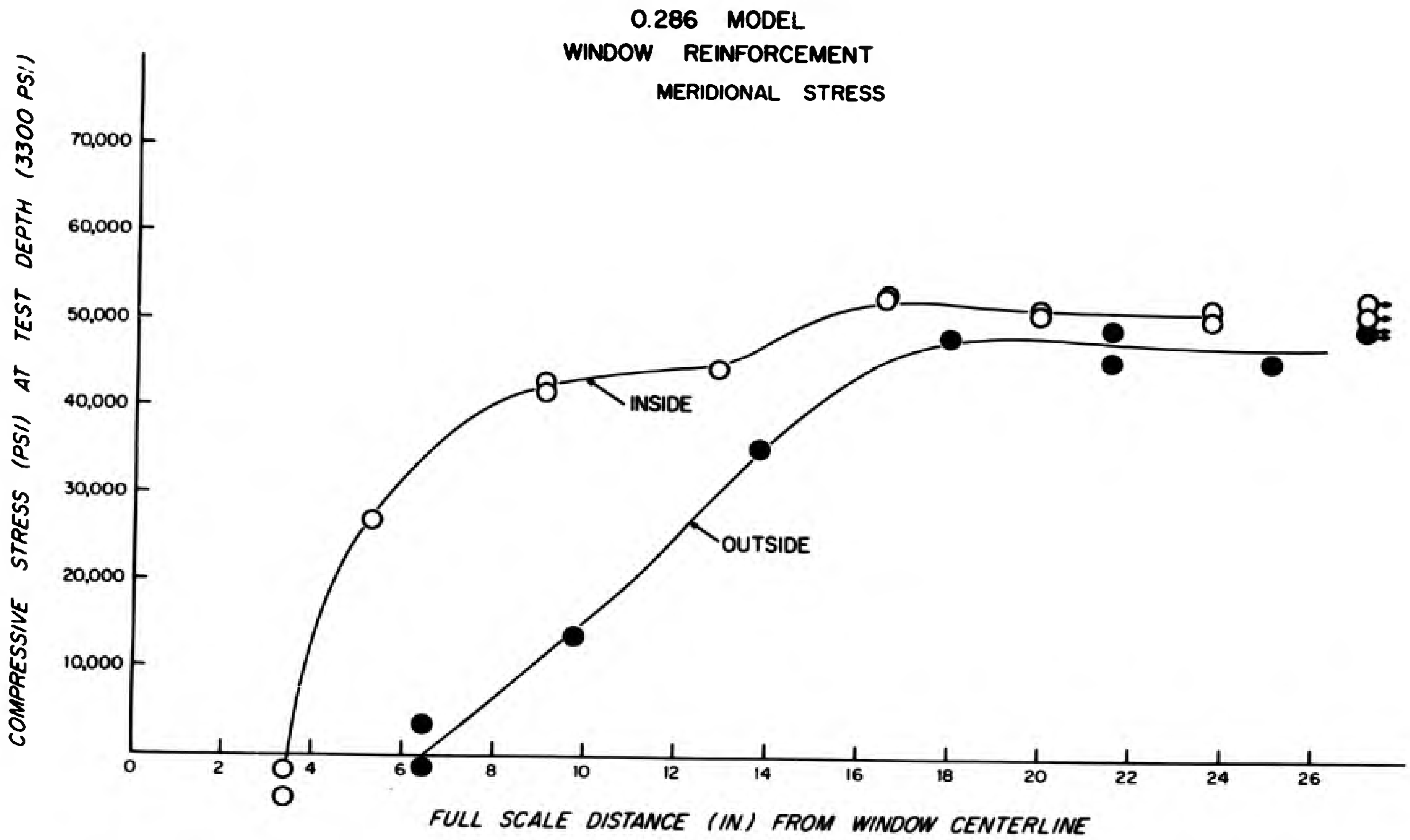


Figure 3. 0.286 model window reinforcement, meridional stress.

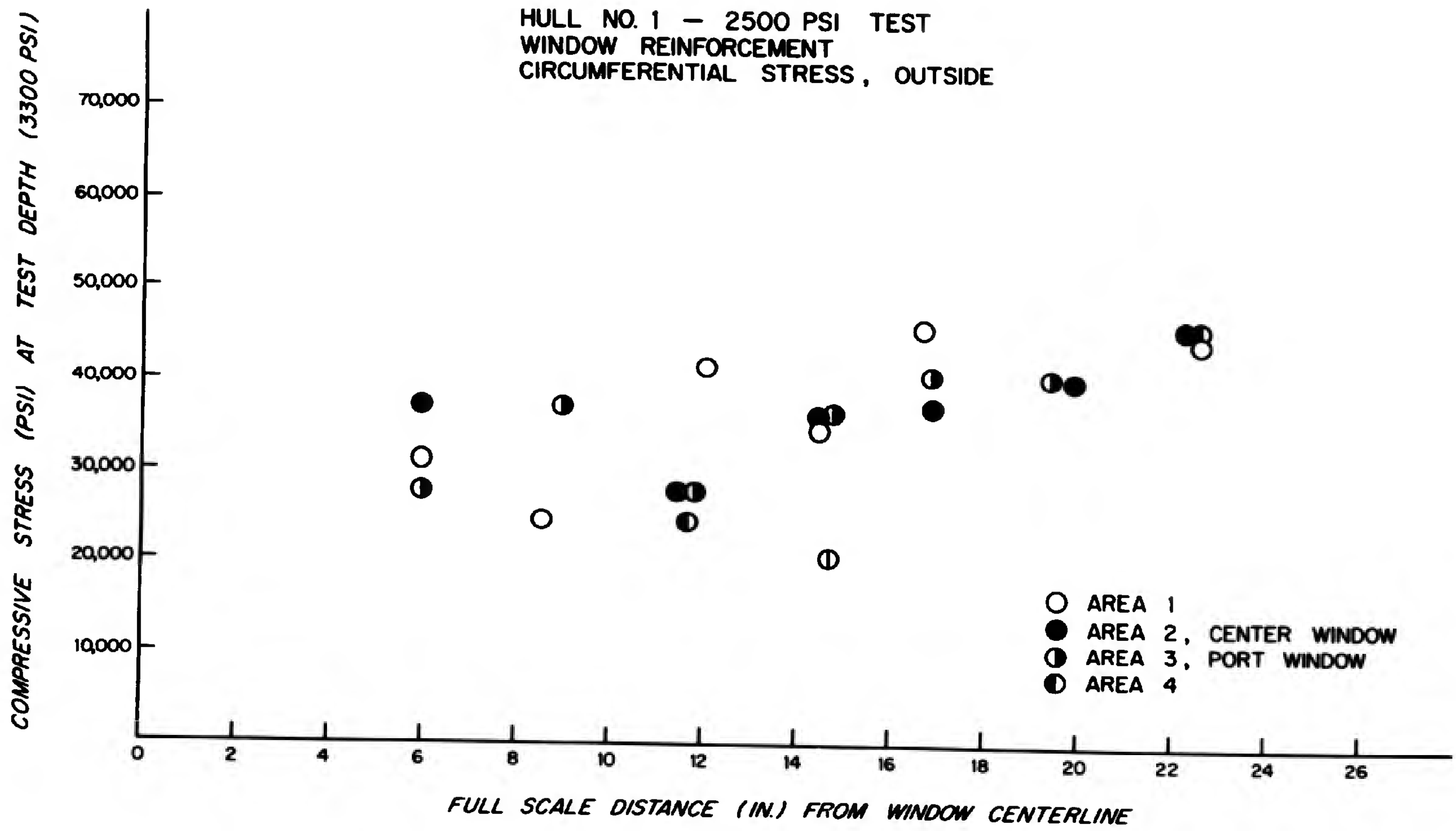


Figure 4. Hull No. 1 - 2500 psi test window reinforcement circumferential stress, outside.

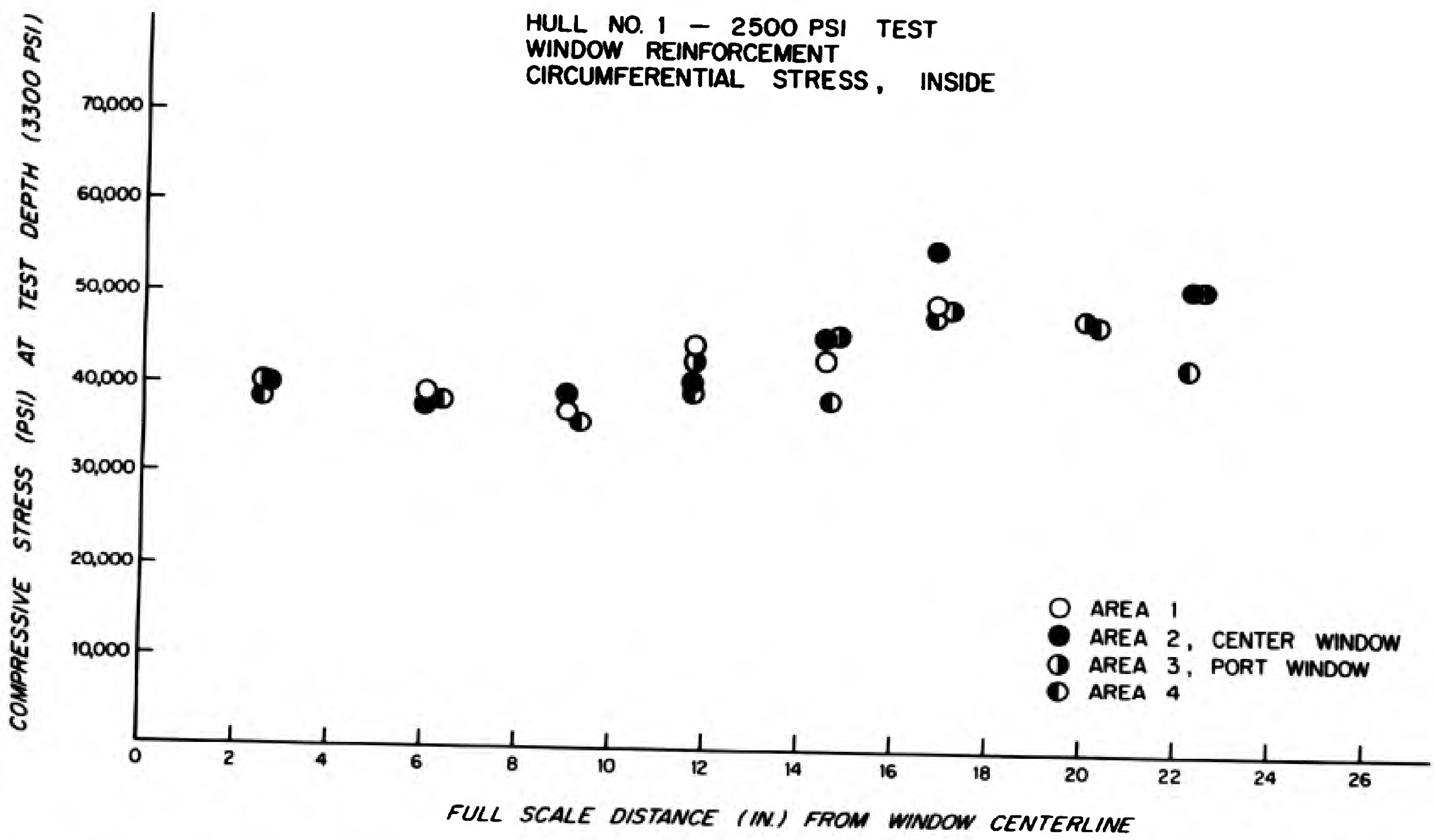


Figure 5. Hull No. 1 - 2500 psi test window reinforcement circumferential stress, inside.

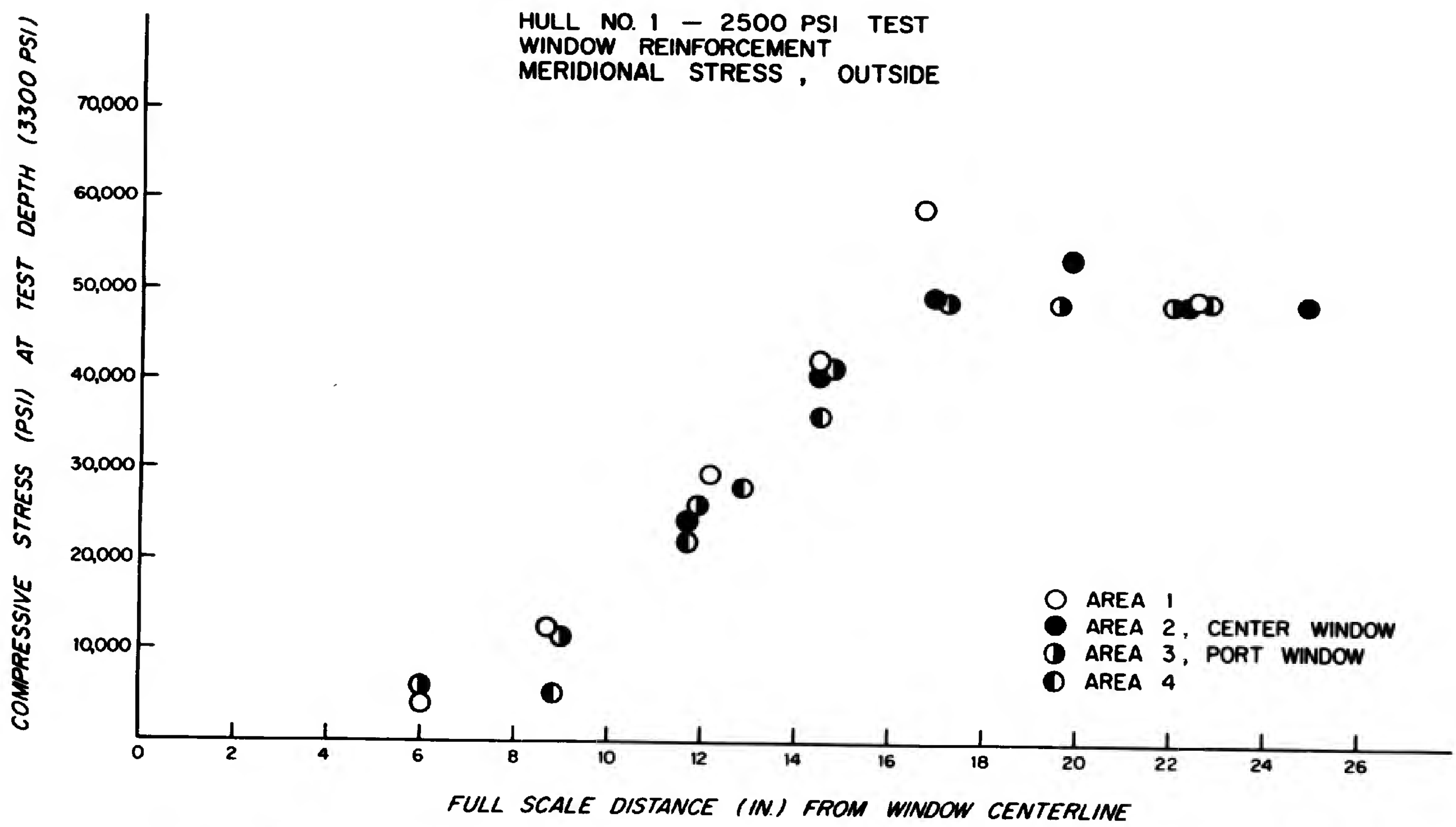


Figure 6. Hull No. 1 - 2500 psi test window reinforcement meridional stress, outside.

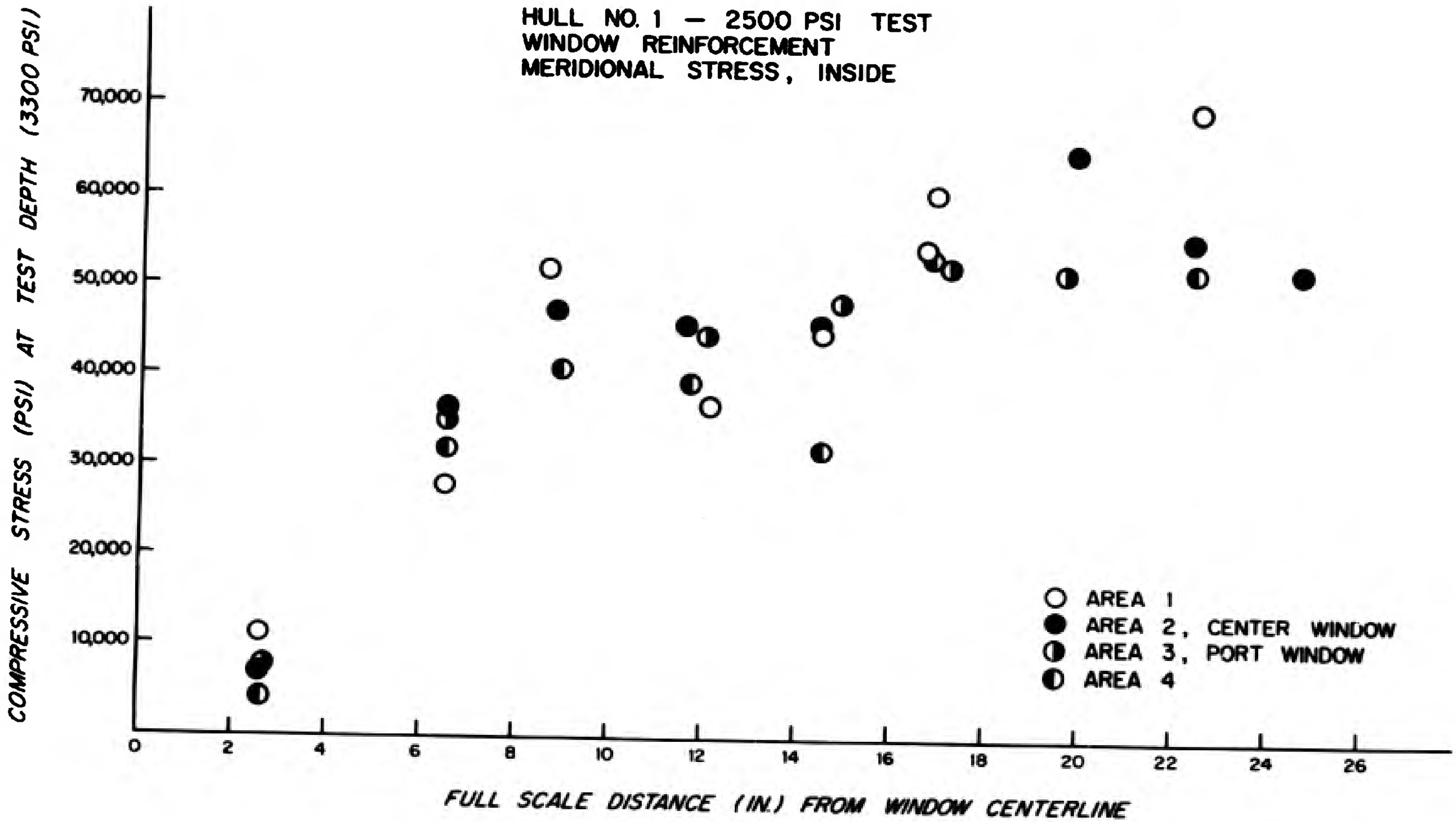


Figure 7. Hull No. 1 - 2500 psi test window reinforcement meridional stress, inside.

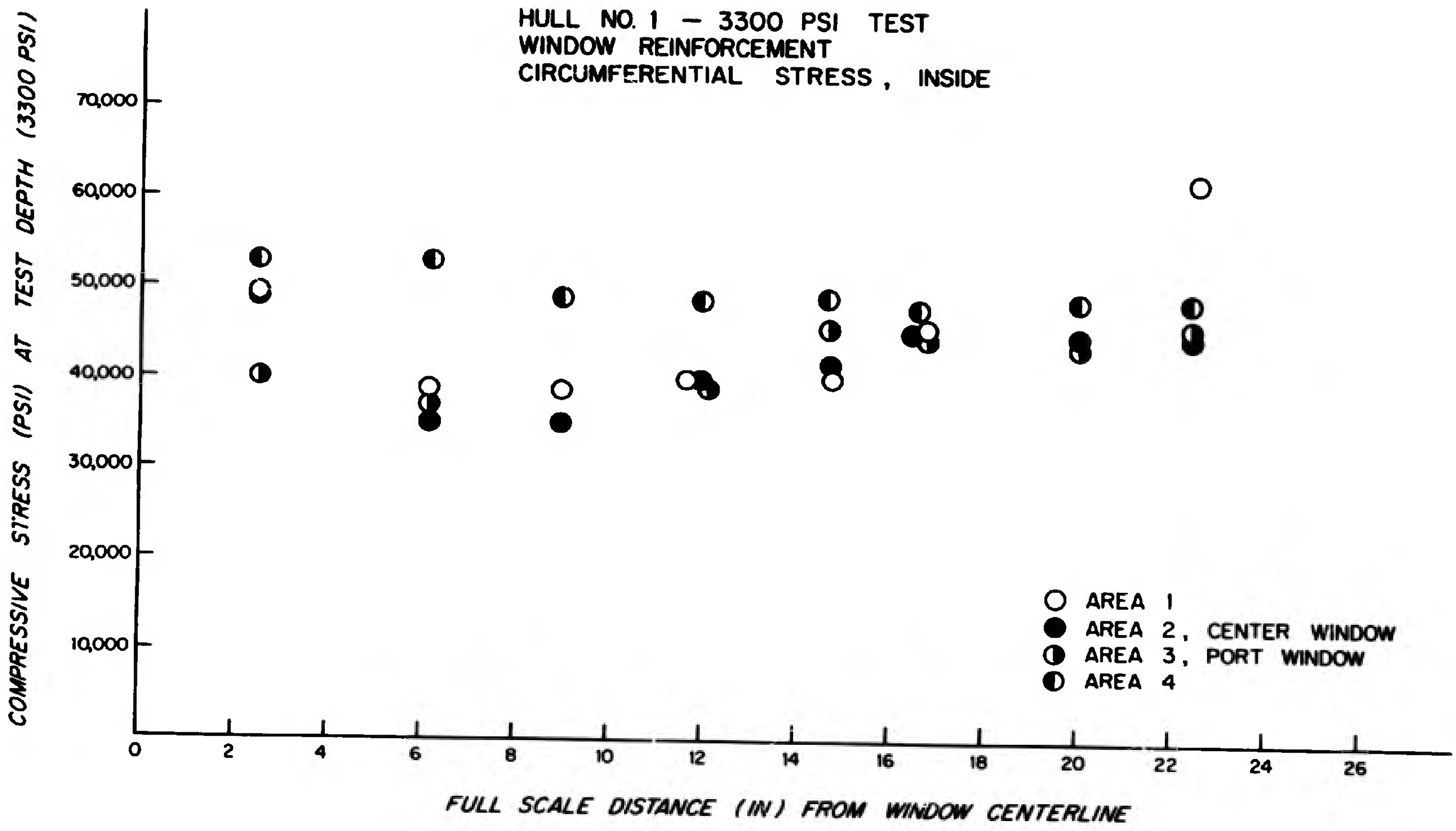


Figure 8. Hull No. 1 - 3300 psi test window reinforcement circumferential stress, inside.

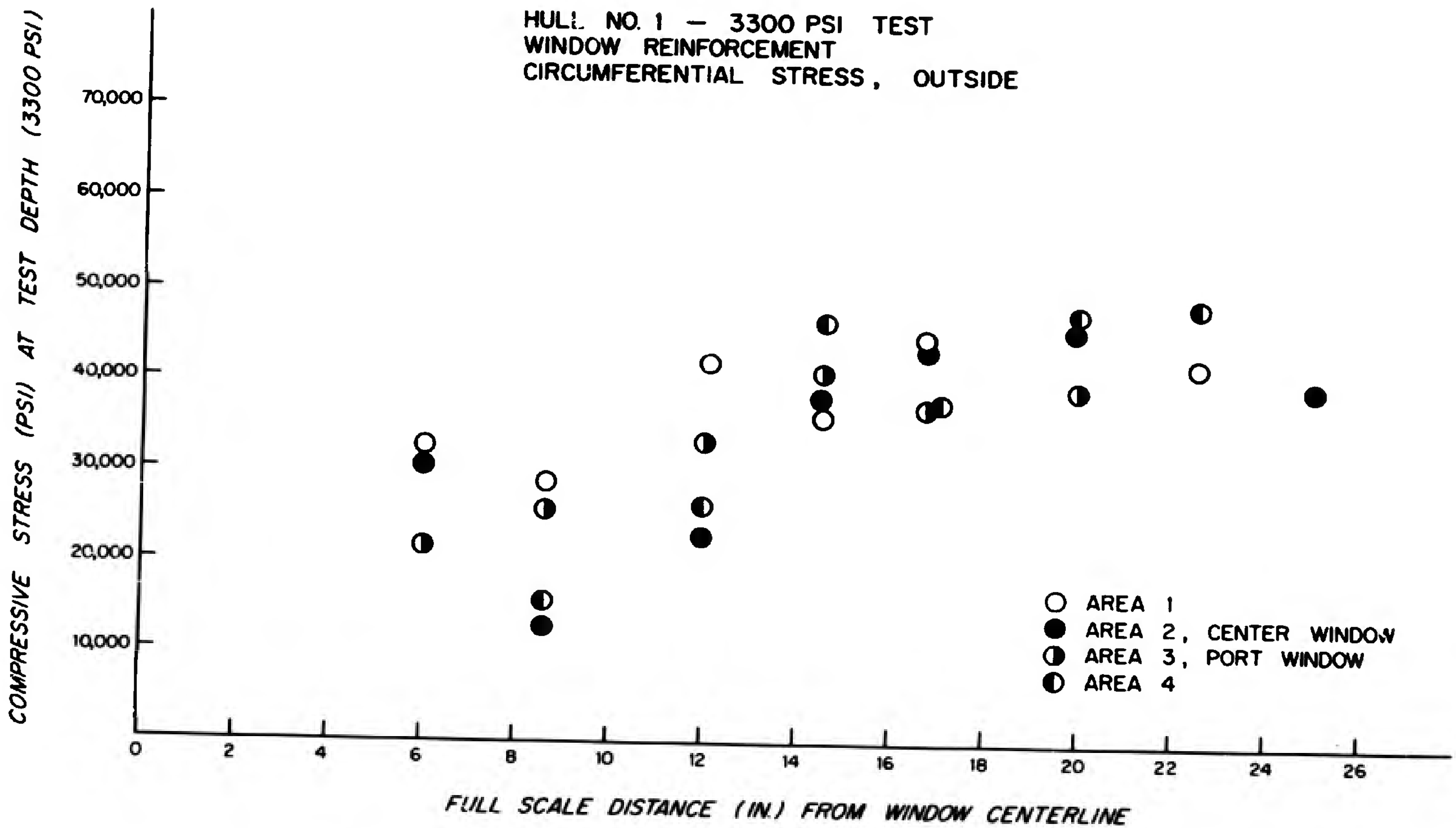


Figure 9. Hull No. 1 - 3300 psi test window reinforcement circumferential stress, outside.

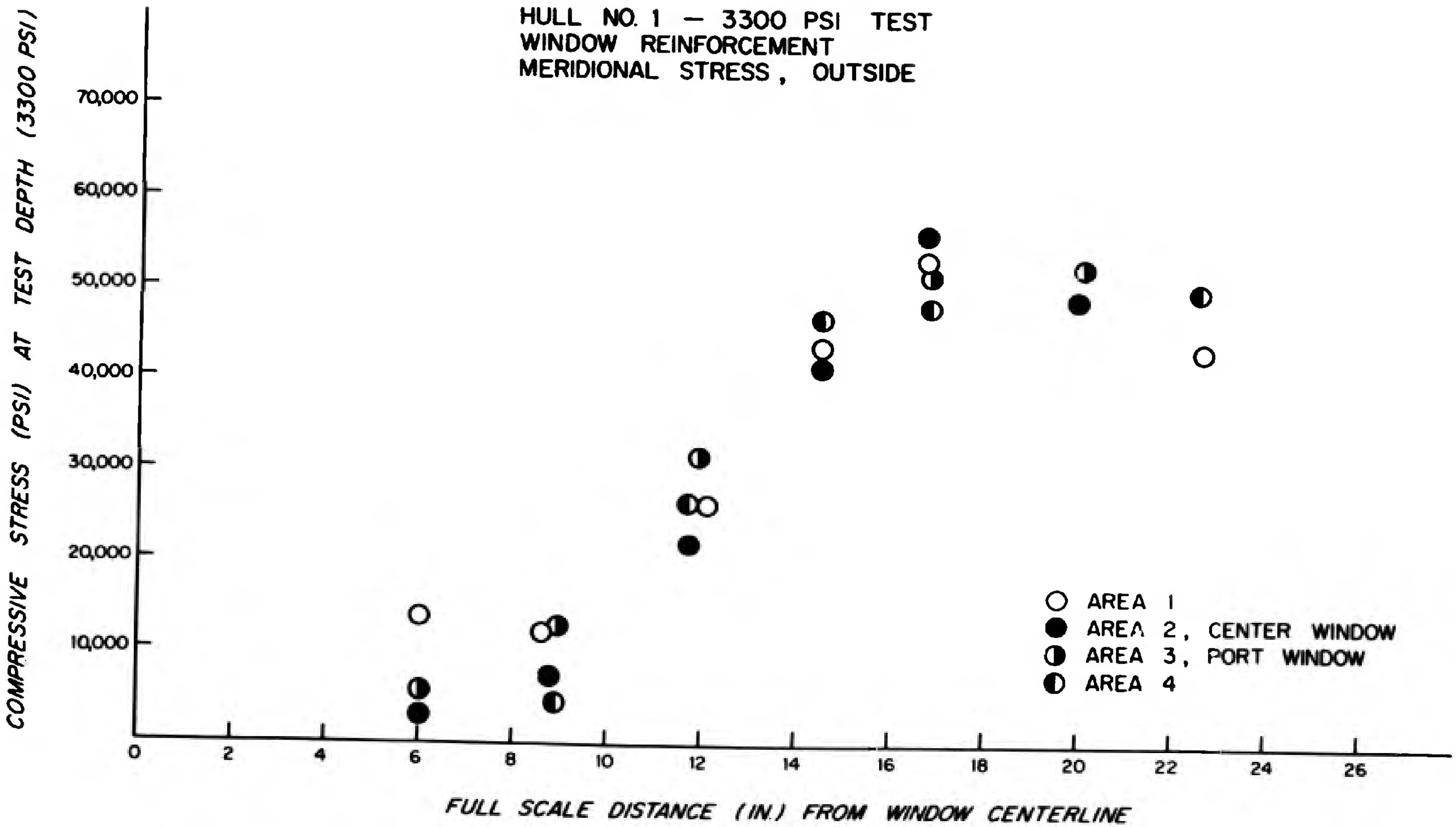


Figure 10. Hull No. 1 - 3300 psi test window reinforcement meridional stress, outside.

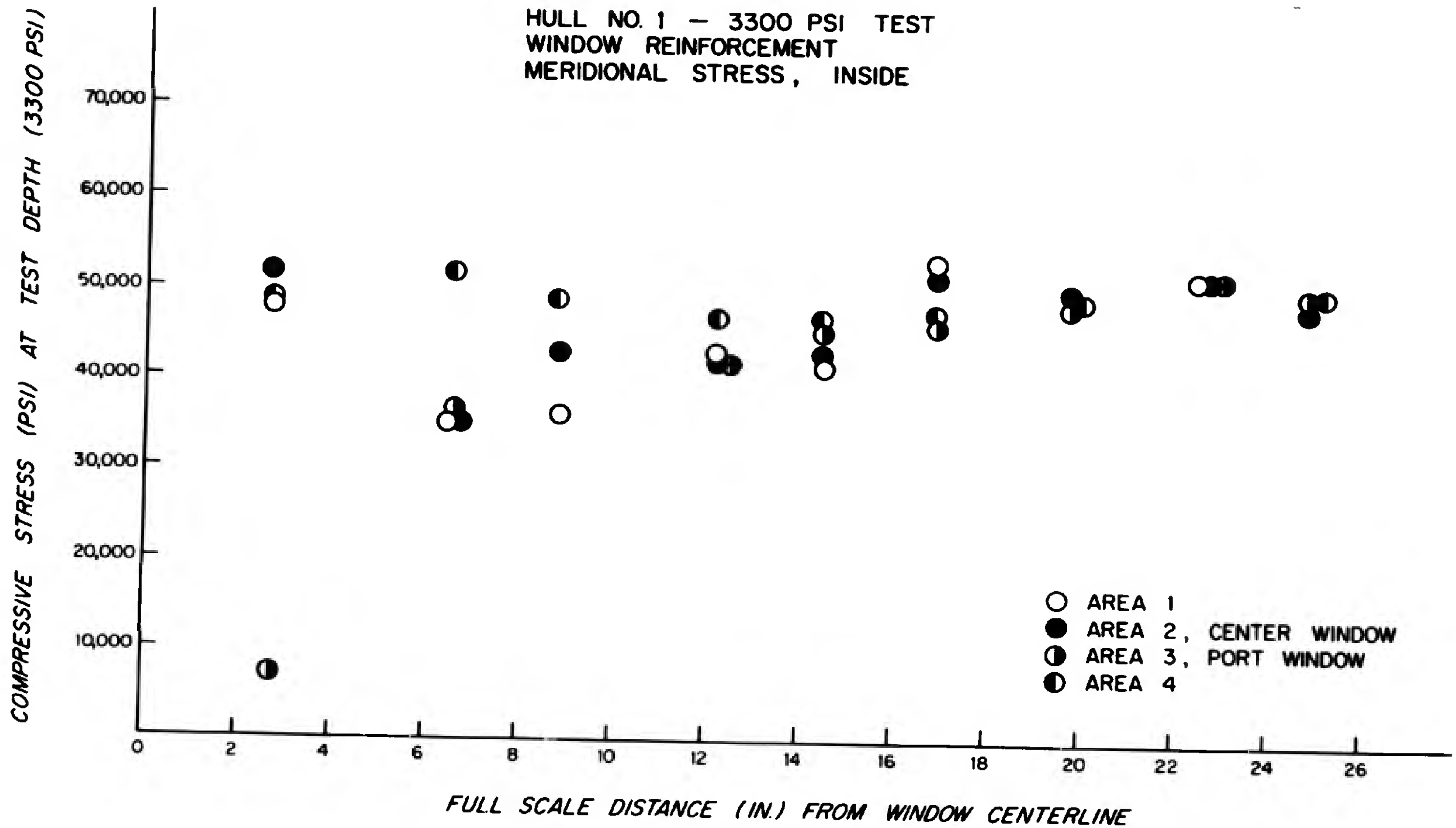


Figure 11. Hull No. 1 - 3300 psi test window reinforcement meridional stress, inside.

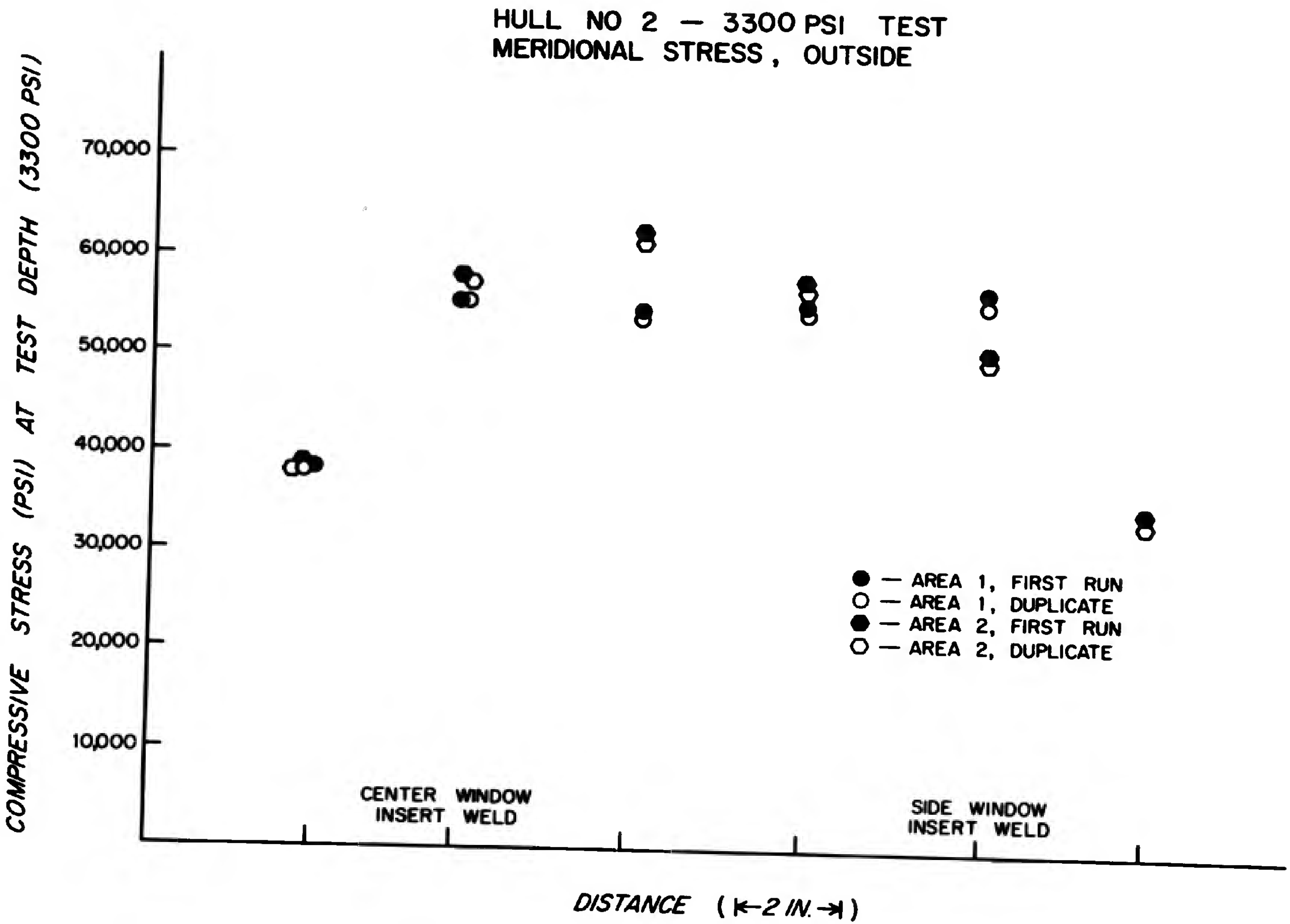


Figure 12. Hull No. 2 - 3300 psi test meridional stress, outside.

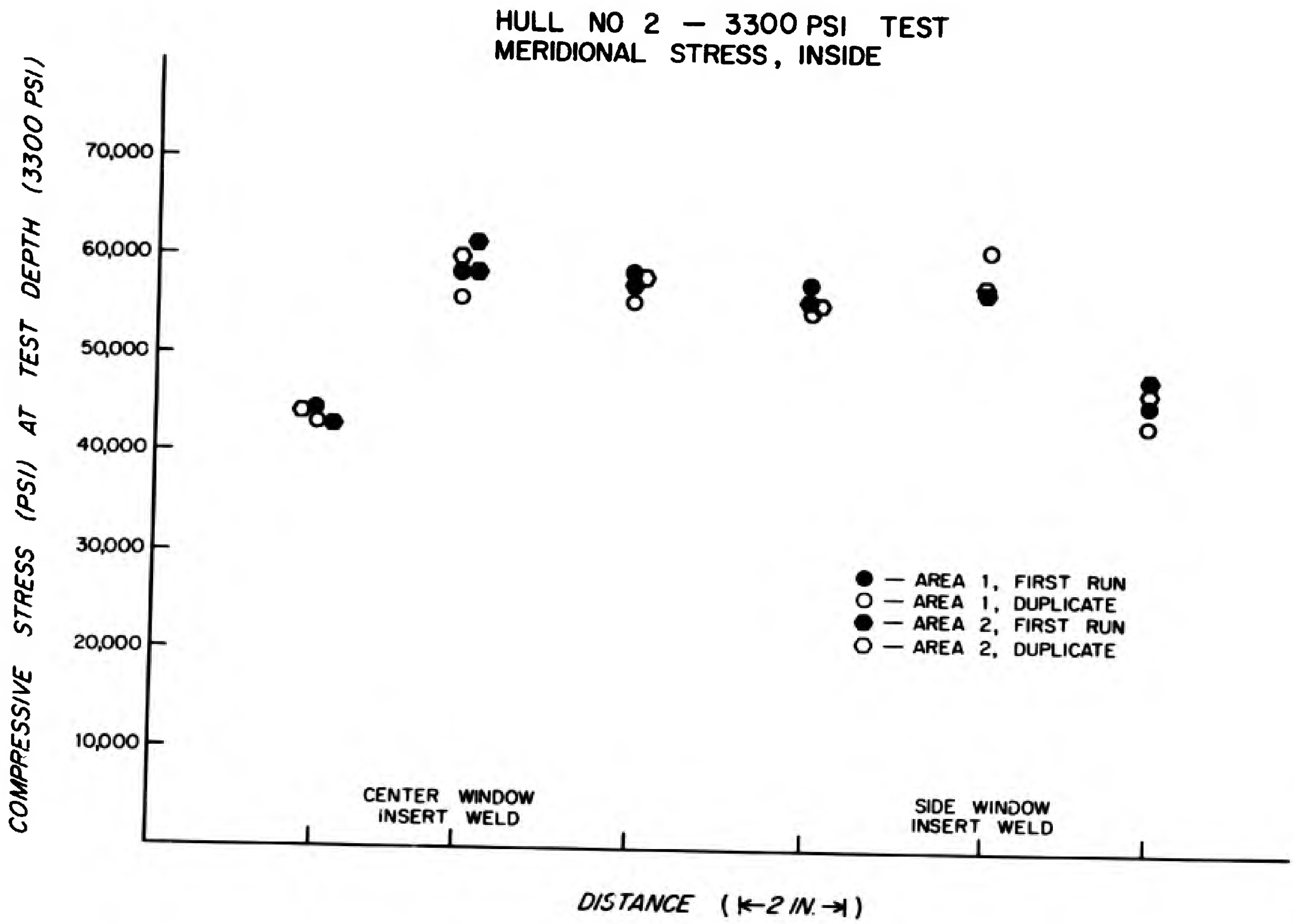


Figure 13. Hull No. 2 - 3300 psi test meridional stress, inside.

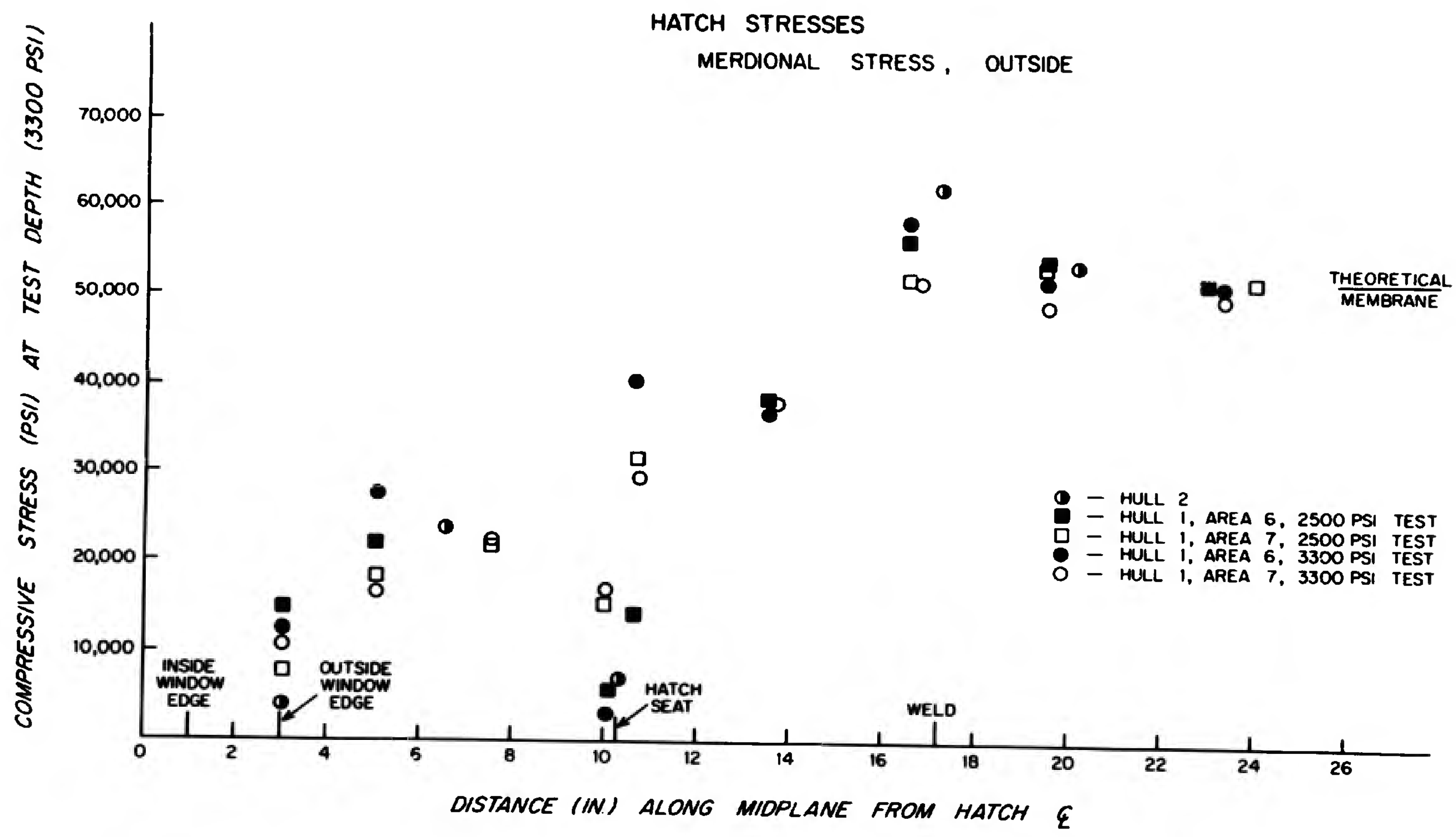


Figure 14. Hatch stresses merdional stress, outside.

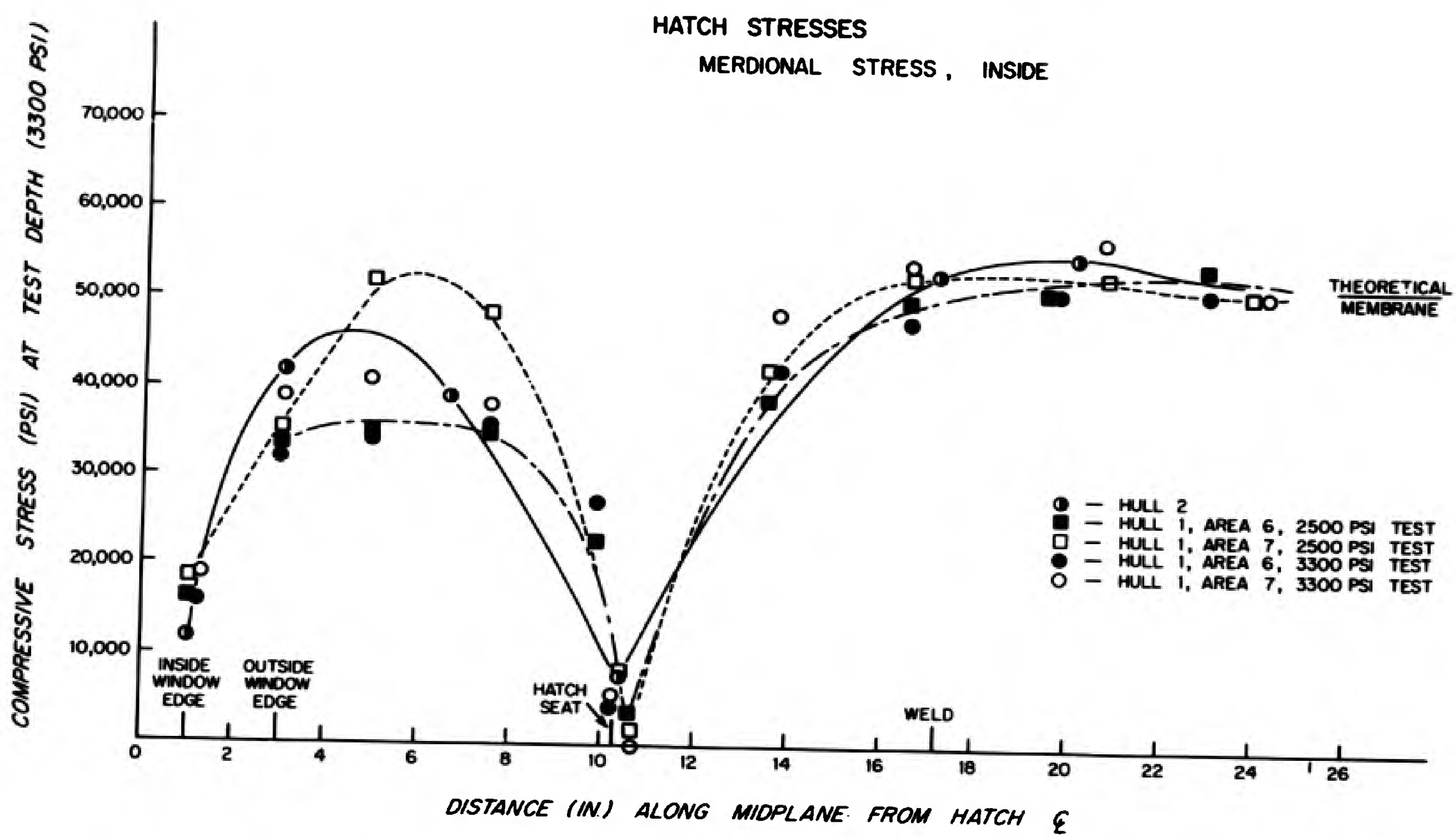


Figure 15. Hatch stresses merdional stress, inside.

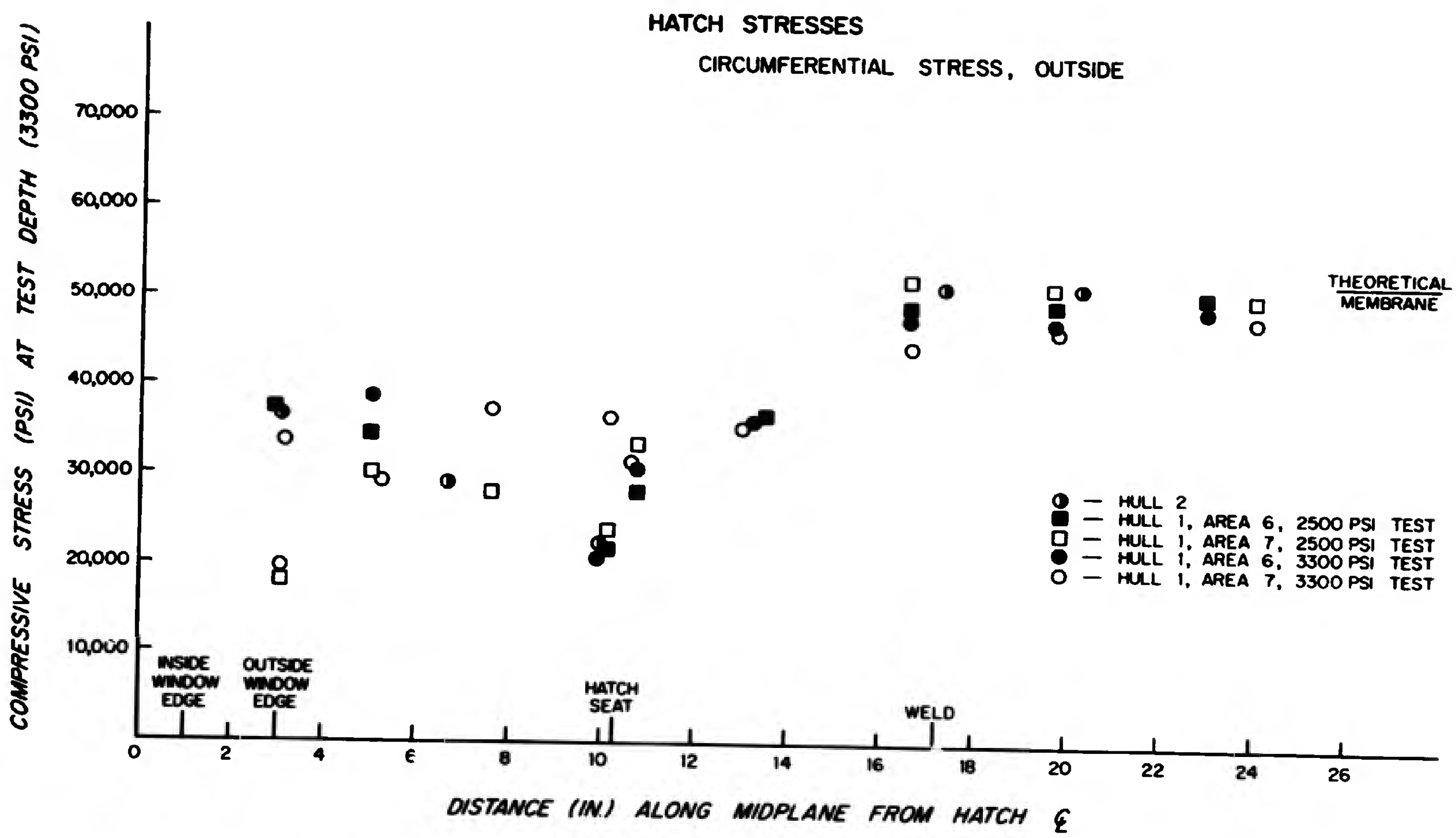


Figure 16. Hatch stresses circumferential stress, outside.

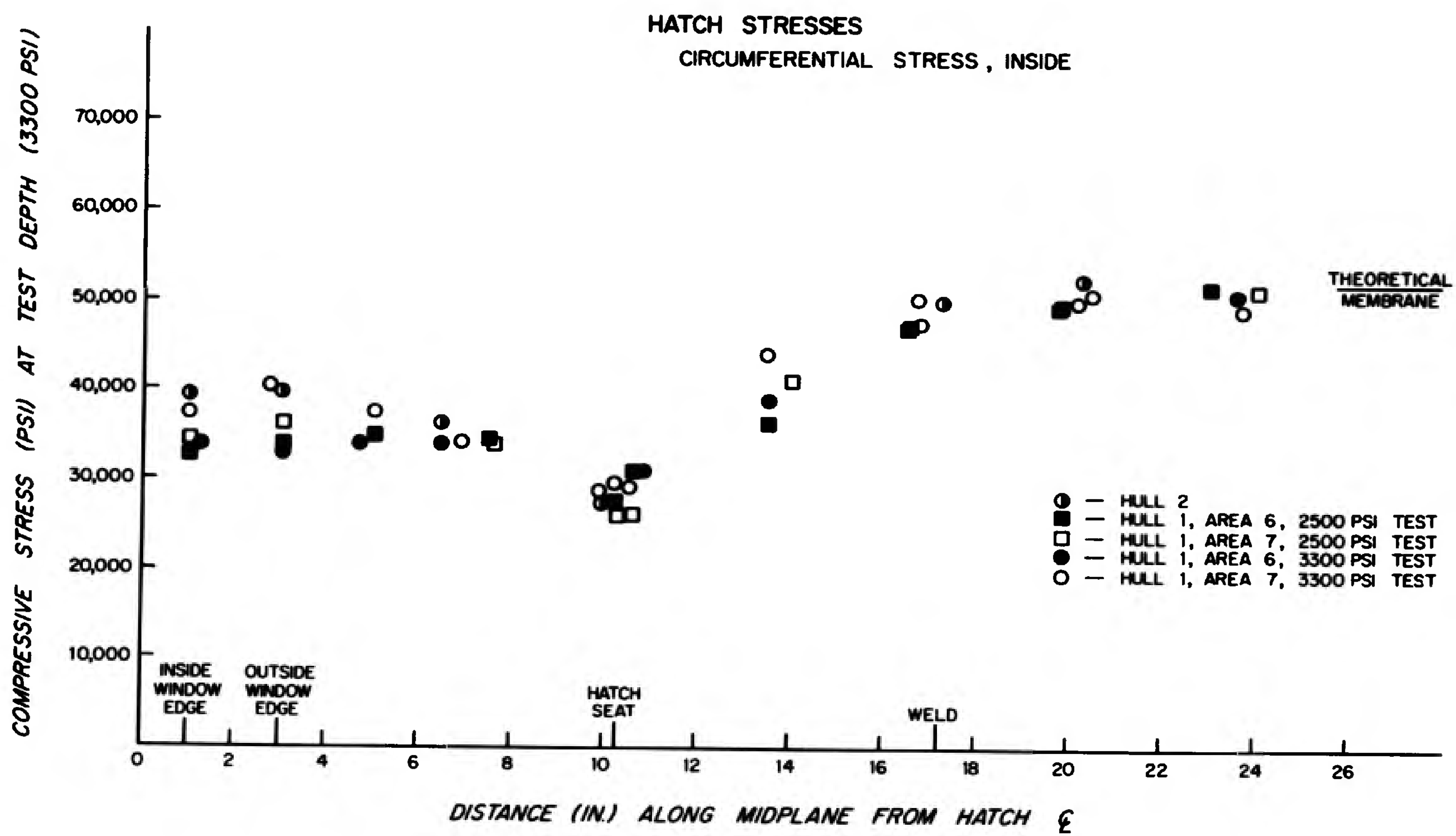


Figure 17. Hatch stresses circumferential stress, inside.

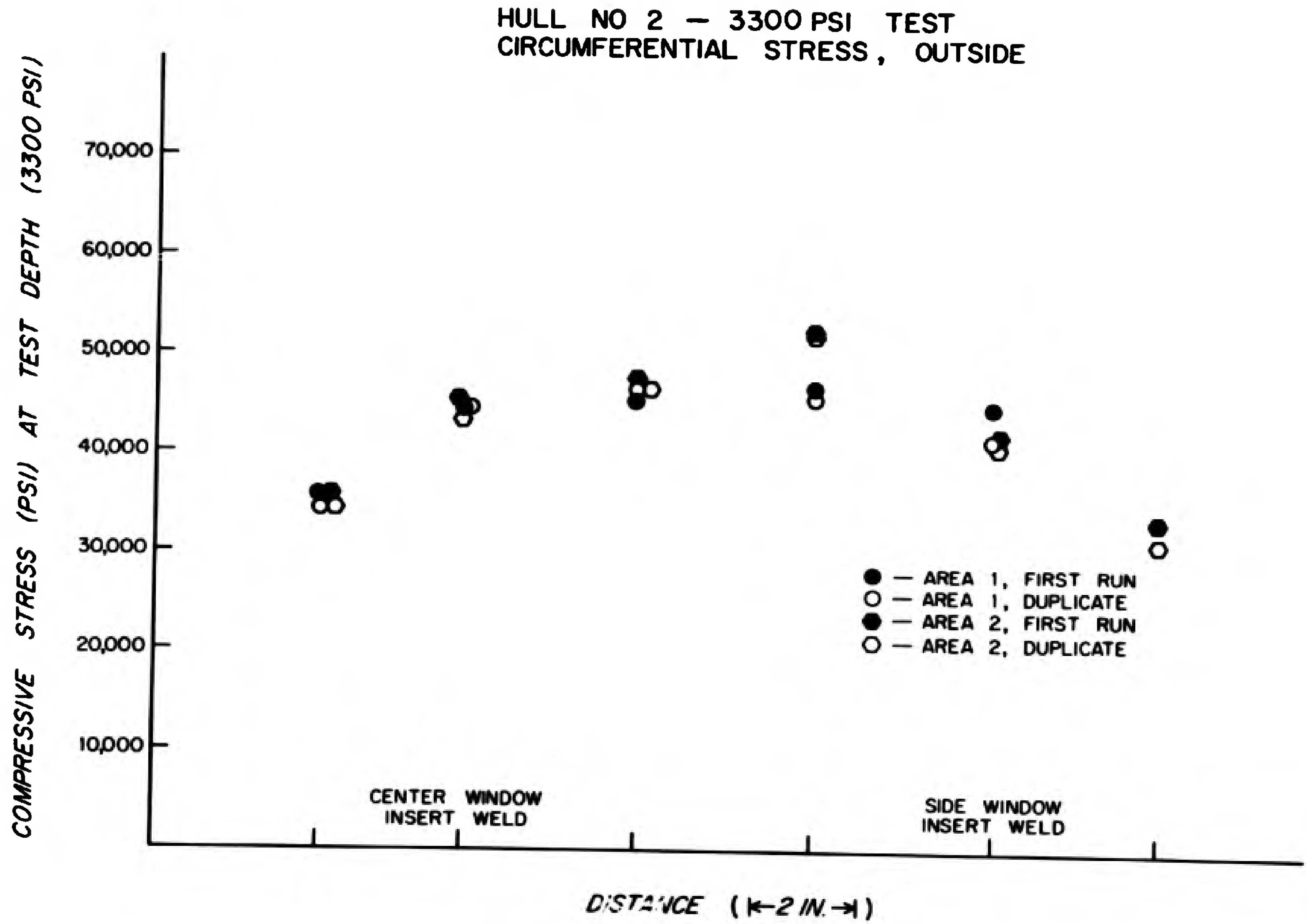


Figure 18. Hull No. 2 - 3300 psi test circumferential stress, outside.

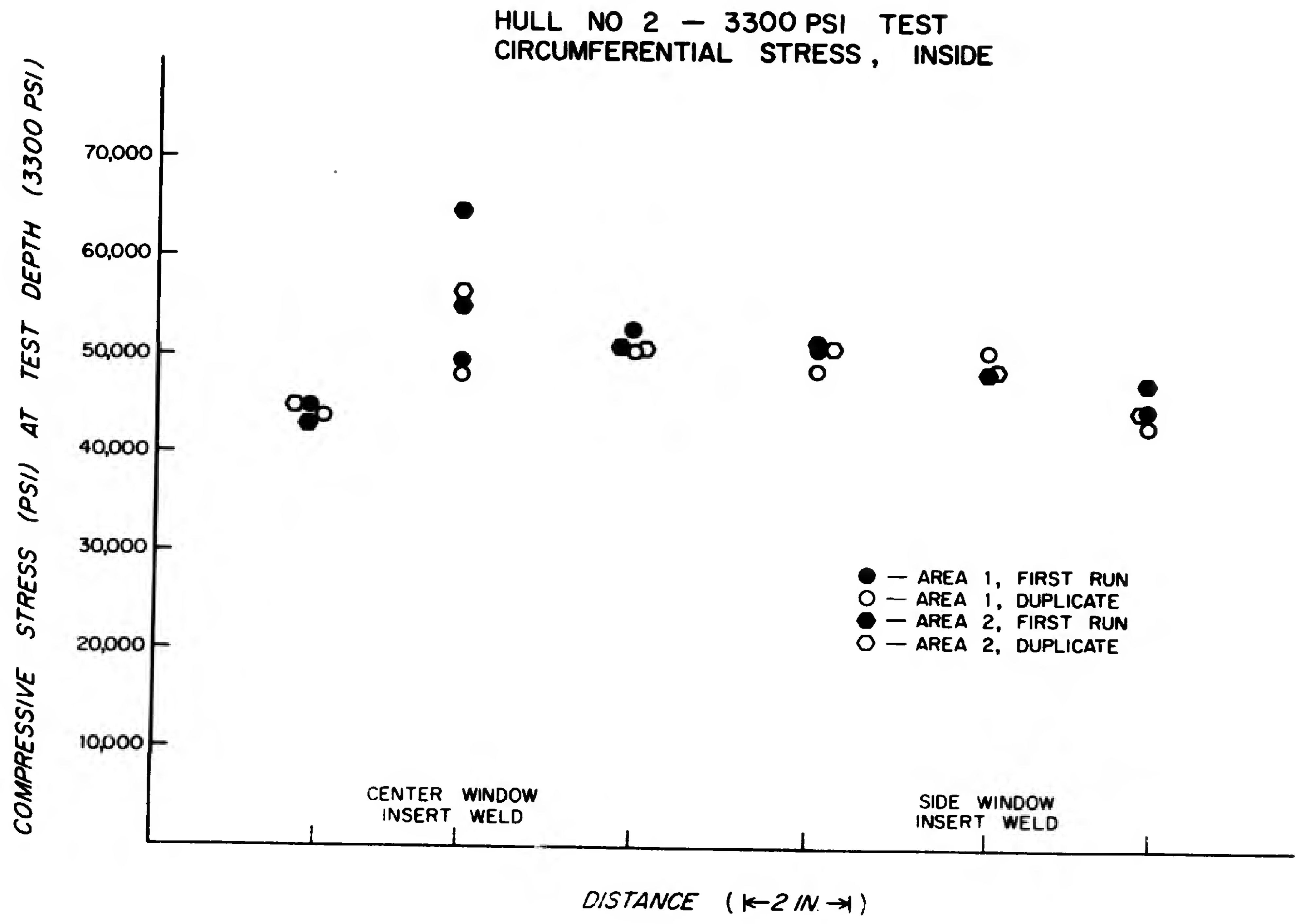


Figure 19. Hull No. 2 - 3300 psi test circumferential stress, inside.

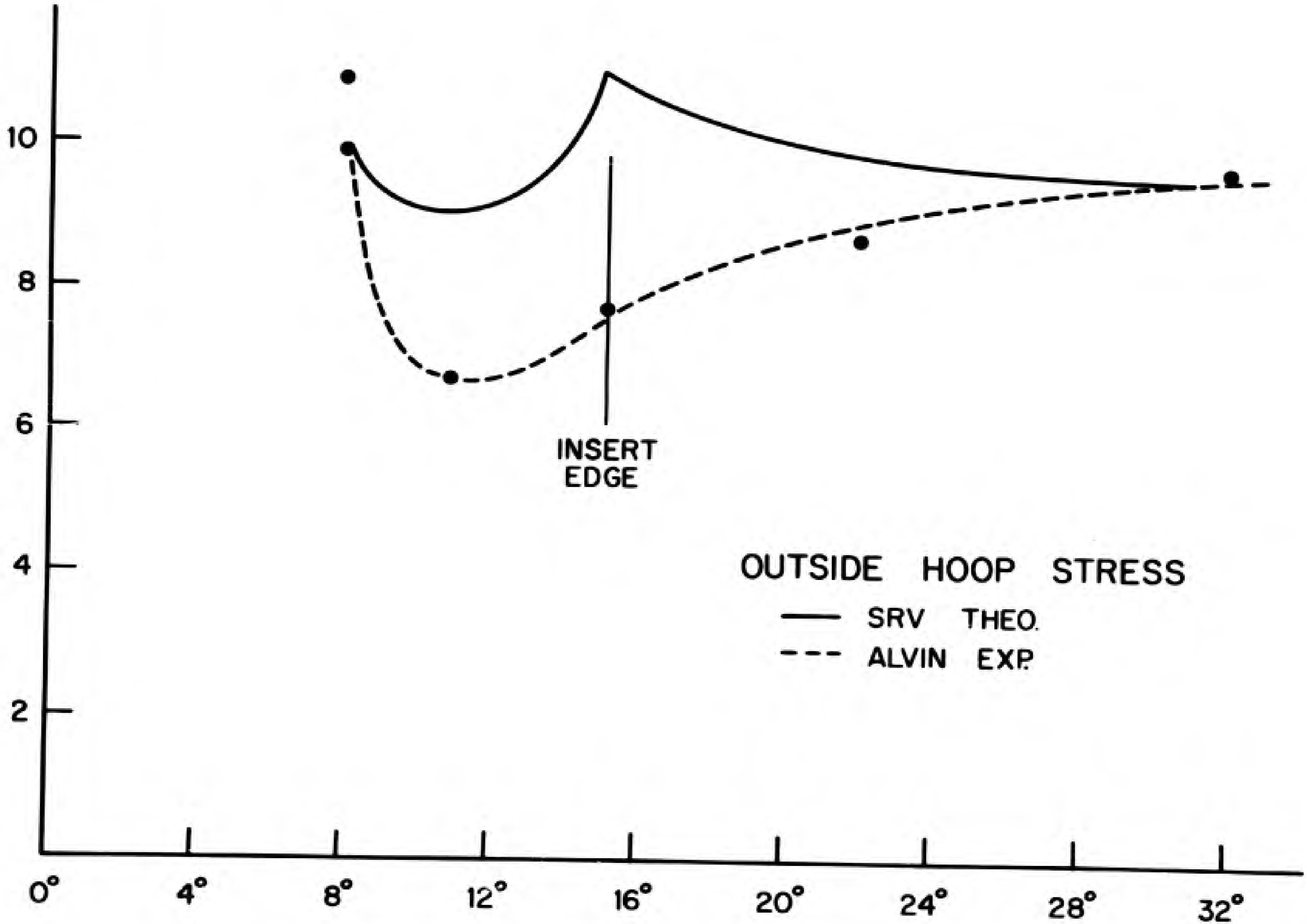


Figure 20. Window reinforcement comparison of experiment and theory.

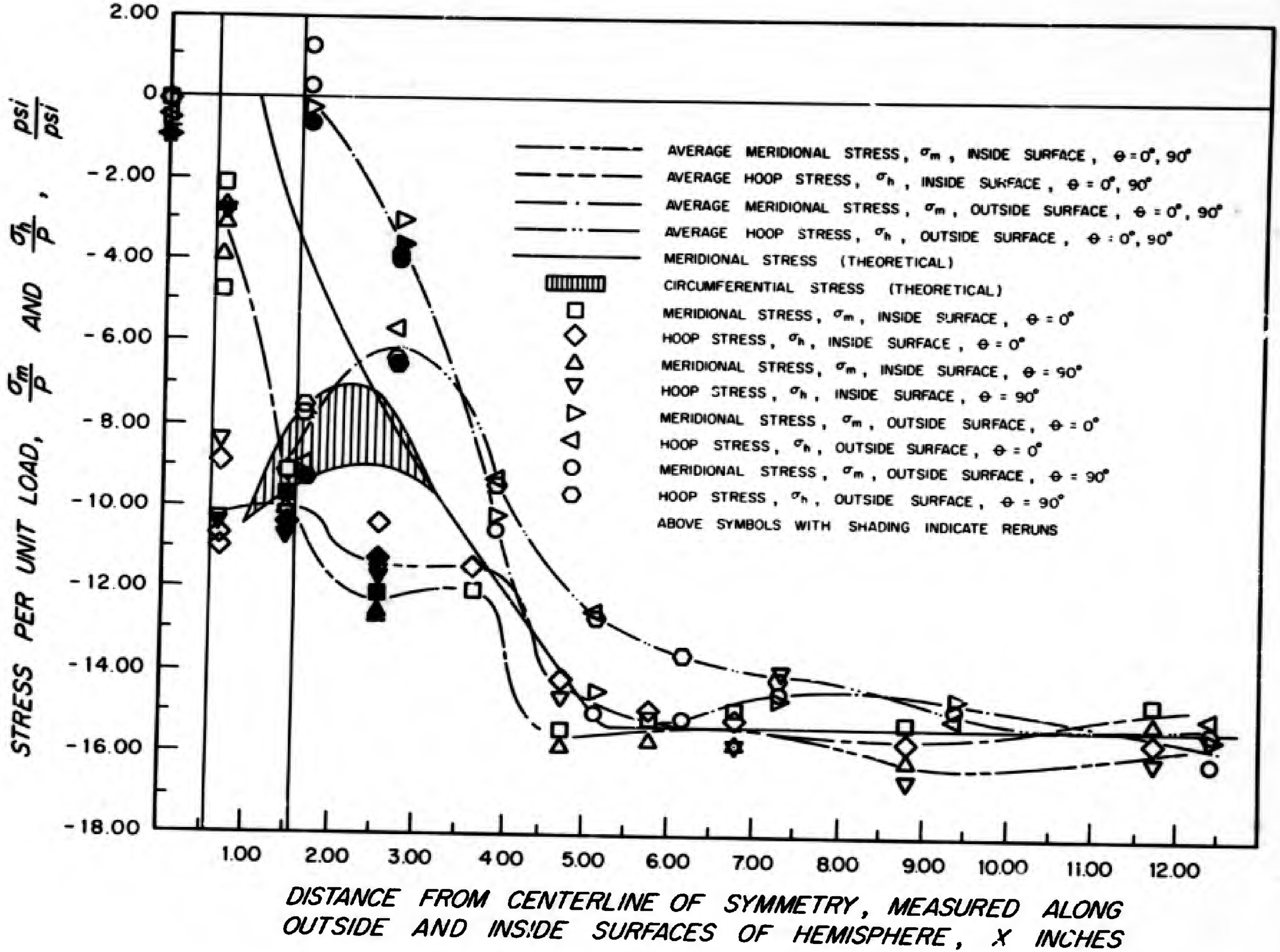


Figure 21. 0.286 scale model test data compared with theoretical.

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13. ABSTRACT

Results are presented of pressure tests, measurements and analyses of the strength of the three pressure hulls constructed for the deep submergence vehicle ALVIN. Comparison of stress distribution as measured in various tests and predicted theoretically is made. Failure of the hull can occur by buckling or by yielding over an appreciable fraction of shell thickness or by yielding at a stress concentration. A DTMB analysis predicts collapse of the three hulls No. 1, 2, and 3 at 7040, 7160, and 6720 psi respectively. No. 1 hull has been tested to 4400 psi. From strain measurements, isolated yielding at the inside surface of hull No. 2 (presently in ALVIN) will occur at a pressure of 5800 psi. However, yielding through the entire section would not occur until near the predicted collapse pressure. The maximum Mises equivalent stress at the test pressure of 3300 psi is 62,200 psi. The comparable material yield strength is 110,000 psi. At a stress concentration in way of the hole in the hull for the release mechanism, local stress is estimated from model data to be 93,000 psi at the test pressure of 3300 psi. A close fitting insert of yield strength 125,000 psi is used with hull No. 2. The strength of the plexiglas viewing ports, the electrical lead-throughs and the hull release mechanism are referenced but not discussed.

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