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HYDROSTATIC TESTS OF A HIGH STRENGTH STEEL INTERNALLY STIFFENED HEMISPHERE

by

Frank M. Schwartz

STRUCTURAL MECHANICS LABORATORY
RESEARCH AND DEVELOPMENT REPORT

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Task 1960

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ABSTRACT

A stiffened steel hemisphere with a nominal yield strength of 150,000 psi was designed, fabricated, and tested to explore the structural efficiency of stiffened spherical shells. Test results show that the collapse pressure was approximately 30 percent greater than that predicted for an unstress-relieved, monocoque shell of equivalent weight with the same out of roundness. The collapse pressure approached that of a near-perfect, machined spherical shell. Thus, it appears that the detrimental effects of initial imperfections and residual stresses arising from fabrication processes for monocoque spherical shells may be at least partially overcome through use of properly designed stiffening systems. Based on the test results, it is estimated that an HY-150 stiffened steel spherical shell designed for a collapse depth of 10,000 ft would weigh 43 percent of its displacement.

ADMINISTRATIVE INFORMATION

The work described in this report was conducted under the sponsorship of the Naval Ship Systems Command, Project S-F013 02 03, Task 1960.

INTRODUCTION

Demands for more efficient end closure configurations for conventional submarines, the requirements of increased operating depths for hydrospace vehicles, and the needs of the aerospace industry have generated considerable interest in spherical shell structures in recent years. Interest at the David Taylor Model Basin has been directed toward establishing design criteria for spherical shells with hydrospace applications. To date, investigations have been primarily experimental and have resulted in rather reliable design procedures for unstiffened spherical shells.[1] These are based on experimental results of tests on both machined models and models manufactured according to feasible full-scale fabrication procedures. Thus, it is possible to predict collapse pressures of spheres with initial imperfections and residual stresses as well as to predict collapse of near-perfect specimens.

Relatively little experimental data exist for stiffened spherical shells. However, sufficient experimental work has been conducted to indicate that potential weight savings can be achieved if stiffeners are spaced at relatively close intervals and distributed in a particular array or grid-like pattern.[2,3] On the other hand, when the unsupported arc length between stiffeners is large, stiffening systems may be ineffective and may even weaken the shell.

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[1]References are listed on page 13.

This report describes the design considerations and the results of an exploratory test of a grid-stiffened, high strength steel hemisphere designed for a collapse depth of 10,000 ft.

DESCRIPTION OF THE MODEL

For the purposes of this exploratory study, the following characteristics and requirements were assumed:

Configuration – Internally stiffened spherical shell

Size – 10-ft OD

Design Collapse Depth – 10,000 ft

Material – Steel

Compressive Yield Strength – 150,000 psi
(0.2 percent offset)

The initial design criteria for this assumed prototype were established from unpublished experimental results on stiffened HY-100 (σ_y = 100,000) steel hemispheres tested at the Model Basin. These data indicated that for particular shell-frame parameters, a grid-stiffened HP-150 (σ_y = 150,000 psi) steel hemisphere with a margin of stability in the general instability mode of approximately 3.4 could be expected to collapse at approximately 85 percent of the yield pressure, P_y, of an equivalent thickness, near-perfect shell. (The margin of stability is defined as the ratio of the elastic general instability pressure to the yield pressure. The elastic general instability pressure P_{c-s} was determined from the work of Crawford and Schwartz[4] in the present case. The equivalent shell thickness was calculated by distributing the cross-sectional area of a single stiffener over a typical bay spacing.)

Stiffener dimensions and spacings were determined from one of the HY-100 steel hemispheres; tests of that model had indicated that the contribution of bending stresses to total stress levels was not excessive for area-of-frame to area-of-shell ratios of less than 0.2. The stiffener depth and width were determined by requiring that the elastic buckling stress calculated from plate theory (for the case in which the plate is loaded in uniaxial compression and where three sides are simply supported and the fourth, parallel to the direction of loading, is free) be twice the yield strength of the material. The resulting depth to width ratio of the stiffener was approximately 7. The HY-100 tests also indicated that the stiffening was effective in preventing local shell buckling at values of the geometric parameter θ (defined in Table 1) of approximately 1.4. Final dimensions, geometric parameters, and calculated pressures for the assumed prototype are shown in Table 1.

Modeling of the particular design presented in Table 1 was hampered by the availability of plate material in the thicknesses and yield strengths required. In addition, the cylindrical test adaptors which were available for the 4-ft tank test facility fixed the diameter of the model. Geometries, significant geometric parameters, and calculated pressures for the

TABLE 1

Design Parameters, Dimensions and Collapse Pressures for a High Strength Stiffened Hemisphere

	Symbol	Prototype	Model (as fabricated)
Radius to the middle surface of the shell, in.	R	60	15.155
Average shell thickness, in.	t_s	0.88	0.273
Arc length of typical shell element, in.	b_s	11	2.726
Frame thickness, in.	t_w	0.50	0.125
Frame height, in.	b_w	3.50	0.900
Cross-sectional area of shell element, sq in.	A_s		0.7442
Cross-sectional area of frame, sq in.	A_f		0.1125
Area of frame to area of shell ratio (stiffener area in one direction for typical shell element)	$\frac{A_f}{A_s}$	0.182	0.151
Equivalent shell thickness (in.) defined by: $\tilde{t} = t_s\left(1 + \frac{2b_w t_w}{b_s t_s}\right) = t_s\left(1 + 2\frac{A_f}{A_s}\right)$	t	1.192	0.357
Nondimensional geometric parameter defined by: $\theta = \frac{0.91\, b_s}{[Rt_s]^{1/2}}$ for Poisson's Ratio of 0.3	θ	1.4	1.24
Volume of frame to volume of shell ratio	$\frac{V_f}{V_s}$		0.300
Weight to displacement ratio	W/D	0.43	0.52
Yield strength, psi	σ_y	150,000	169,000
Elastic general instability buckling pressure, psi (see Reference 4)	P_{c-s}	17,600	19,800
Yield pressure for shell of equivalent thickness, psi	P_y	5,120	7,000
Ratio of elastic general instability pressure to the yield pressure	$\frac{P_{c-s}}{P_y}$	3.44	2.83
Calculated collapse depth, ft ($0.85\, P_y \times 2.25$)	$(C.D.)_c$	9,800	13,400
Experimental collapse depth, ft	$(C.D.)_E$		13,750

It is noted that b_s, θ, A_f/A_s, V_f/V_s, and $\tilde{t}$ refer to a typical shell element in the third circumferential tier from the boundary of the hemisphere.

model are also presented in Table 1. The plate available for fabrication of the shell elements was thicker than that specified by the initial design criteria. The substantial increase in shell thickness is reflected by the ratio of weight to displacement shown in Table 1 and by differences between various nondimensional parameters. In addition the average yield strength of the steel used in the manufacture of the model was 169,000 psi, with a maximum variation of less than 2 percent among numerous specimens taken to evaluate the strength properties of the plate material. (A typical stress-strain curve is presented in Figure 1.) It should be noted that the ratio of the elastic general instability pressure to the yield pressure was higher for the assumed prototype than for the modeled geometry. Thus, the increased thickness of the shell was offset by the increase in yield strength. The unpublished results for the stiffened HY-100 steel hemispheres indicate that the margin of stability for the assumed prototype geometry against local shell buckling is sufficiently large to preclude failure in this mode. Thus, it was assumed that the modeled geometry would also be sufficient to preclude failure in the local shell buckling mode. The test adaptor utilized for the model did not provide realistic end conditions for the hemisphere. Thus, additional stiffeners were provided on the external surface at the boundary of the hemisphere to preclude premature failure due to high bending stresses. A schematic drawing of the model is presented in Figure 2.

The stiffened hemisphere was manufactured from HP-150 steel according to feasible full-scale fabrication procedures. The skin of the model was obtained by welding together six formed, 60-deg spherical segments and a formed spherical cap. Frames were cut from plate material and rolled for welding installation normal to the shell inside surface. None of the material for the skin or frames was stress relieved following the forming operation or fabrication of the model.

Deviations from a spherical radius were measured at close intervals over the surface of the hemisphere. These data serve as the input for a computer program which determines the mean radius, the center of the hemisphere, and the corrected departures from sphericity. The departures from sphericity are plotted in the form of a contour map as presented in Figure 3. Flat spots may be identified from examination of the figure as described in Reference 5. (The area between the circles in the figure represents that portion of the surface of the sphere where external stiffeners were provided.)

TEST PROCEDURE

The model was instrumented with Budd wire-resistance strain gages. These were concentrated in an area with large departures from sphericity as determined from examination of Figure 3. Strain gage locations are shown in Figure 4.

The pressure tests were carried out in the 4-ft tank, using oil as the pressure medium. Four tests were run to minimize nonlinearity of the strain data. The model was hydrostatically loaded to collapse on the fourth run.

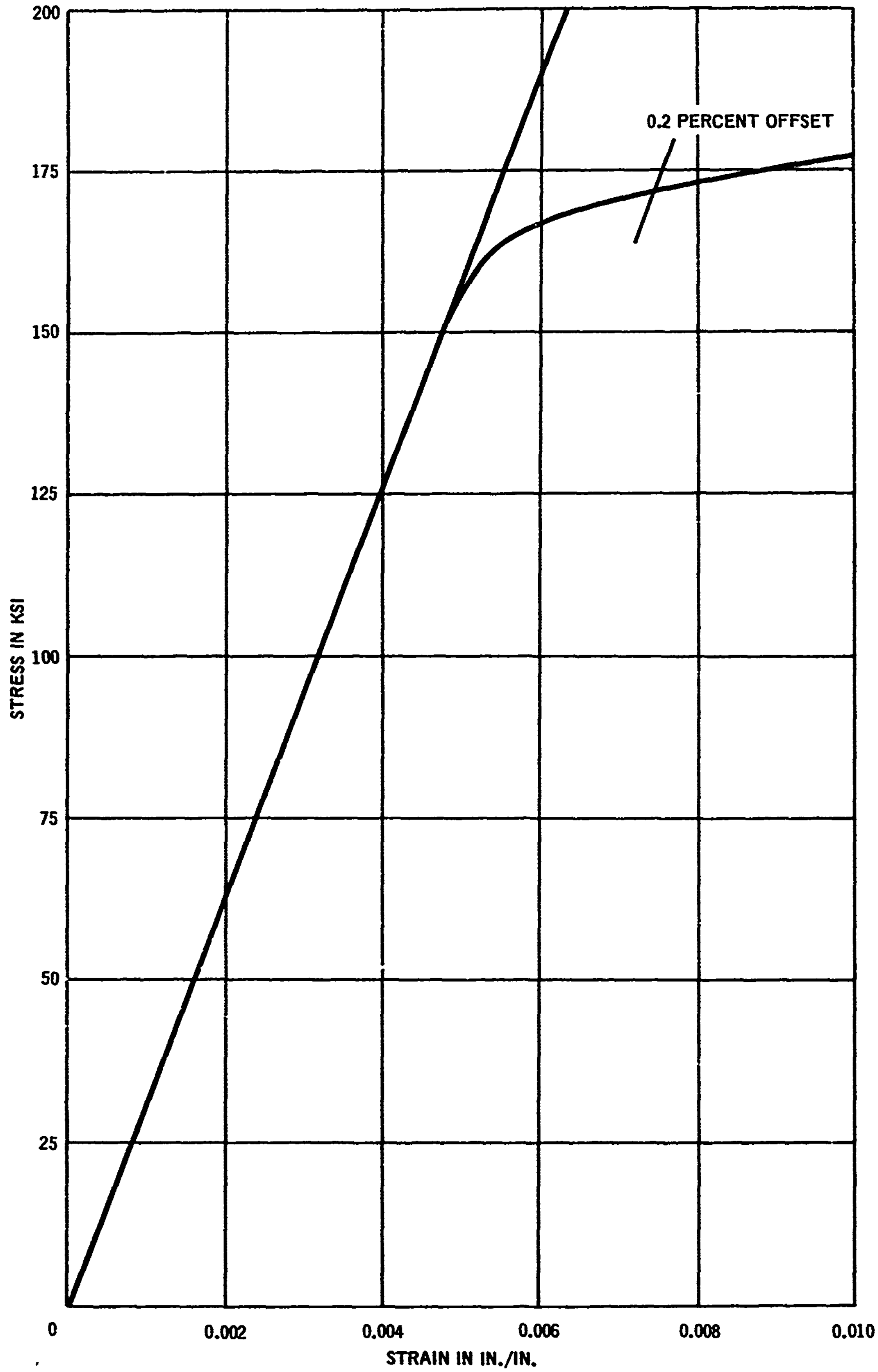


Figure 1 – Typical Stress-Strain Curve for Steel of Model 83

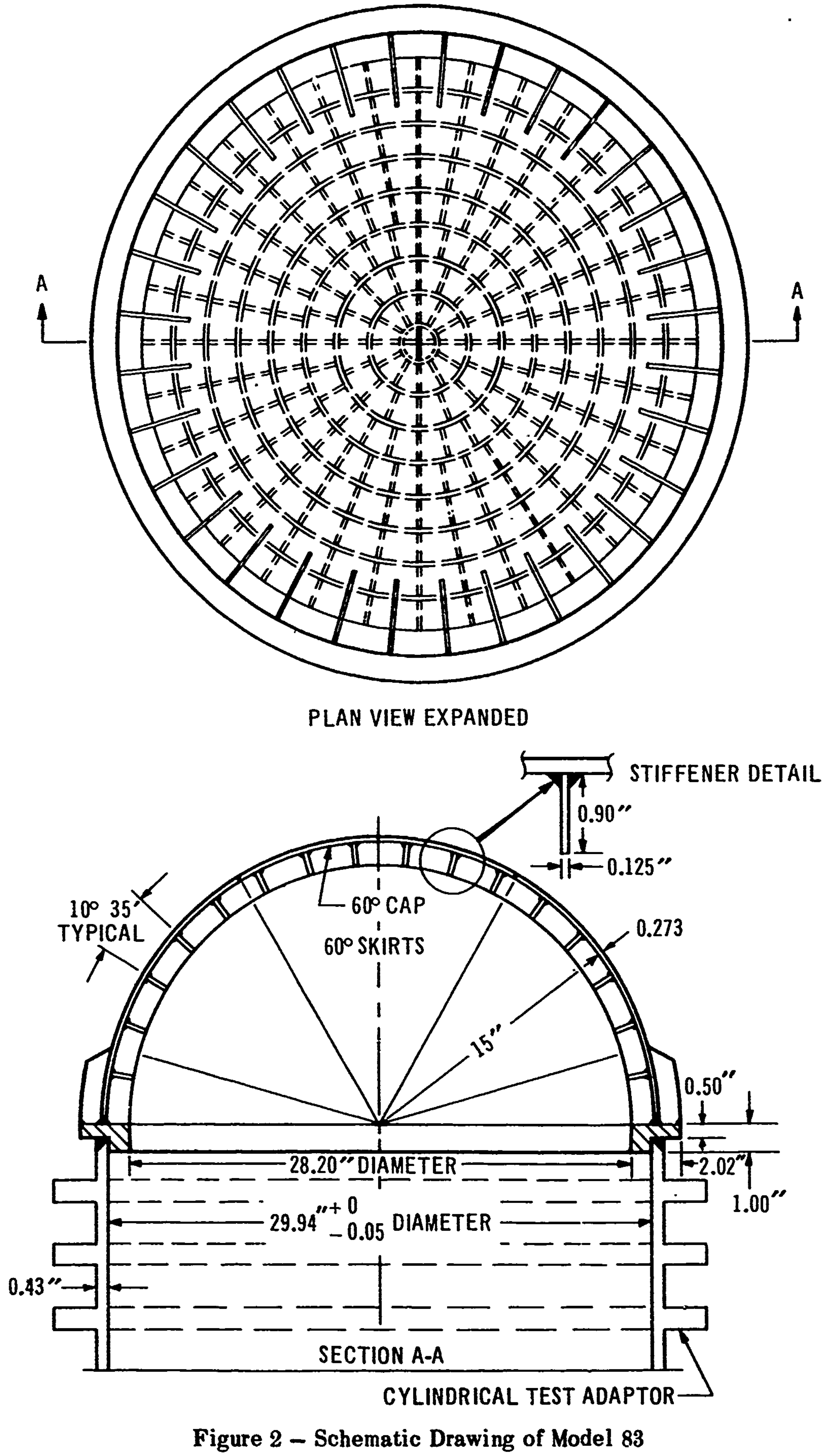


Figure 2 – Schematic Drawing of Model 83

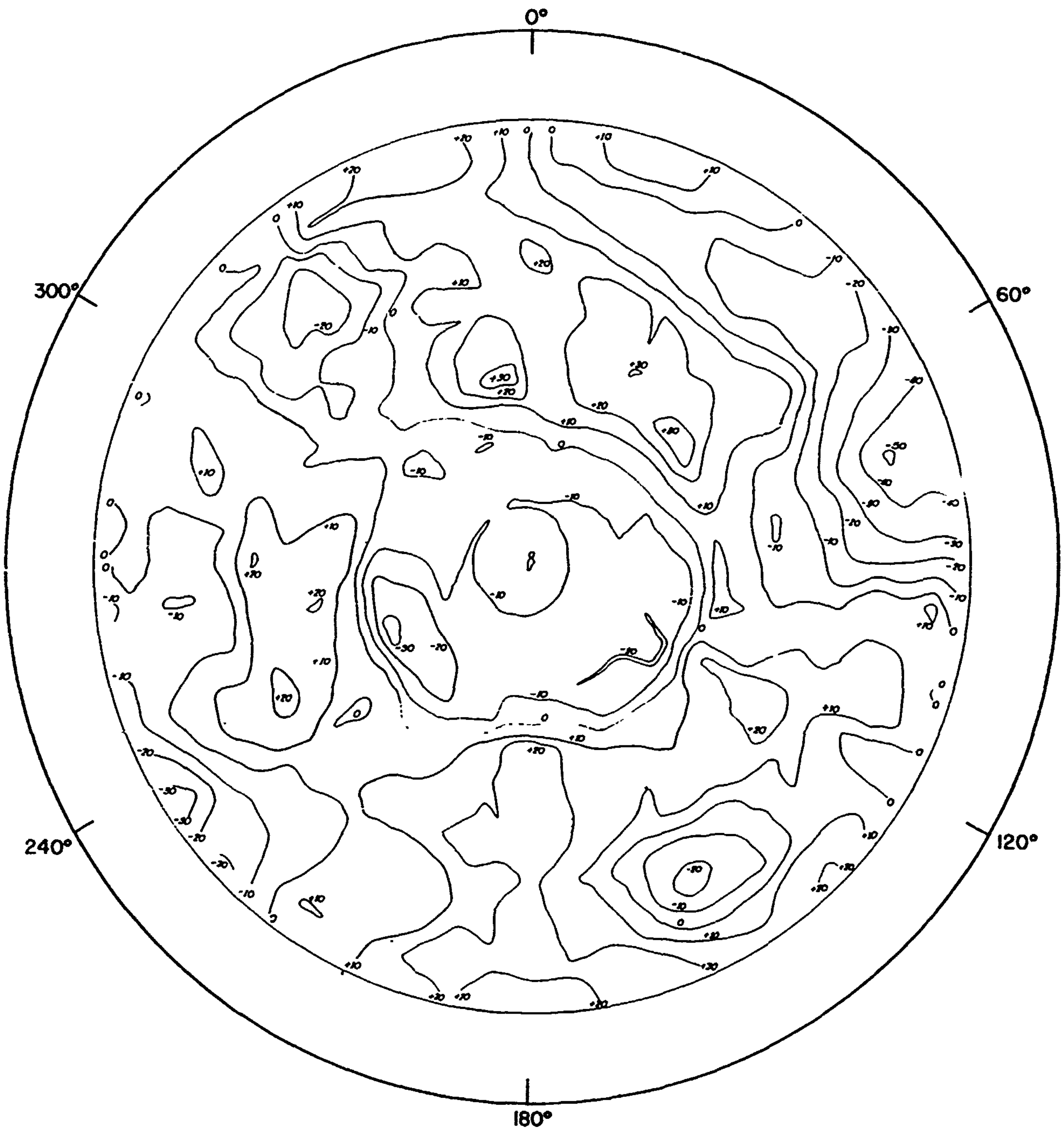
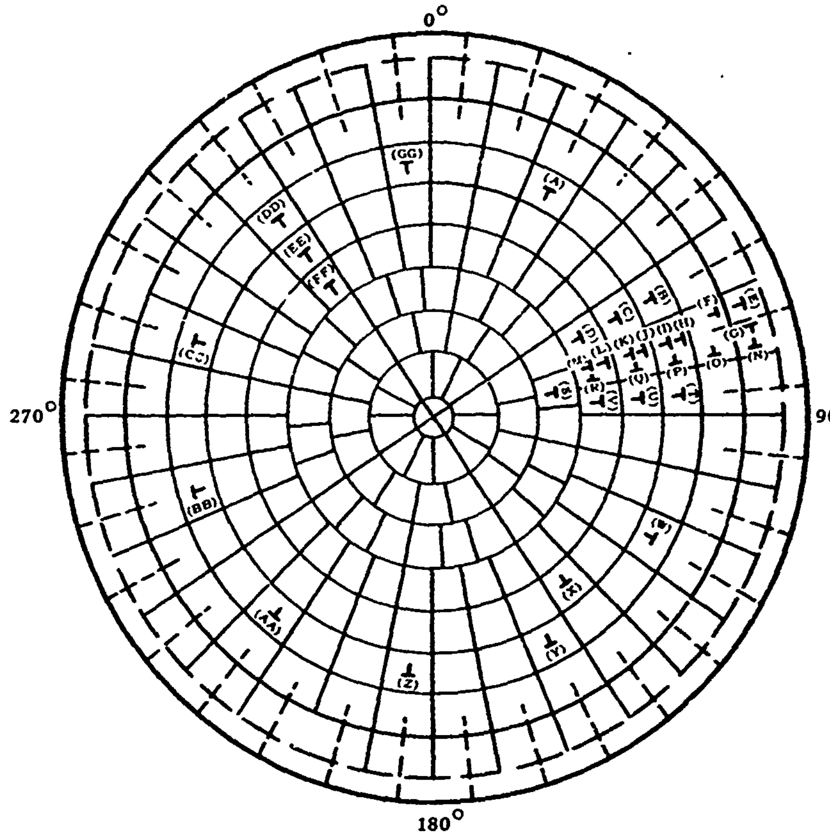


Figure 3 – Out of Roundness Contours – Model 83

Contours are plotted at intervals of 10 mils. Minus contours indicate inward deviations, i.e., –10 indicates that the distance from the center of the sphere is 15.145 in. (where the radius to the middle surface of the shell is 15.155 in.). The area enclosed by the outside circle in the figure represents a hemisphere unfolded into a flat surface whose radial scale remains constant.

PLAN VIEW EXPANDED
INTERIOR OF MODEL



CIRCUMFERENTIAL GAGES

Location	Outside		Inside	
	Gage Number	Strain Sensitivity	Gage Number	Strain Sensitivity
A	164	0.61		
B	100	0.63	200	0.57
C	102	0.60	202	0.64
D	104	0.55	204	Out
E	108	0.58	208	0.52
F	110	0.65	210	0.63
G	112	0.53	212	Out
H	114	0.69	214	0.62
I	116	0.72	216	0.64
J	118	0.67	218	0.62
K	120	0.63	220	0.68
L	122	0.60	222	Out
M	124	Out	224	0.61
N	126	0.49	226	Out
O	128	0.61	228	0.82
P	130	0.68	230	0.72
Q	132	0.55	232	0.61
R	134	0.72	234	Out
S	106	0.53	206	0.60
T	136	0.58	236	0.57
U	138	0.57	238	0.65
V	140	0.55	240	0.55
W	160	0.62		
X	142	0.59	242	0.56
Y	154	0.66		
Z	152	0.55		
AA	150	0.60		
BB	158	0.64		
CC	156	0.56		
DD	148	0.63	248	0.56
EE	146	0.58	246	Out
FF	144	0.52	244	0.54
GG	162	0.50		

MERIDIONAL GAGES

Location	Outside		Inside	
	Gage Number	Strain Sensitivity	Gage Number	Strain Sensitivity
A	155	0.62		
B	101	0.57	201	Out
C	103	Out	203	0.55
D	105	0.56	205	Out
E	109	0.56	209	0.45
F	111	0.55	211	0.54
G	113	0.48	213	Out
H	115	0.61	215	Out
I	117	0.68	217	0.54
J	119	Out	219	0.50
K	121	0.61	221	0.59
L	123	Out	223	0.66
M	125	0.62	225	0.44
N	127	0.50	227	Out
O	129	0.61	229	0.44
P	131	Out	231	0.55
Q	133	0.57	233	0.71
R	135	0.61	235	Out
S	107	0.56	207	Out
T	137	0.60	237	0.53
U	139	Out	239	0.57
V	141	0.40	241	0.50
W	161	0.66		
X	143	0.72	243	Out
Y	155	0.77		
Z	153	0.62		
AA	151	0.60		
BB	159	0.58		
CC	157	Out		
DD	149	0.74	249	0.62
EE	147	0.69	247	Out
FF	145	0.63	245	0.60
GG	163	0.64		

Figure 4 – Strain Gage Layout Diagram and Strain Gage Sensitivities for Model 83

RESULTS AND DISCUSSION

Strain sensitivities for each gage are presented in the strain gage layout diagram of Figure 4. Typical pressure versus strain plots are presented in Figure 5.

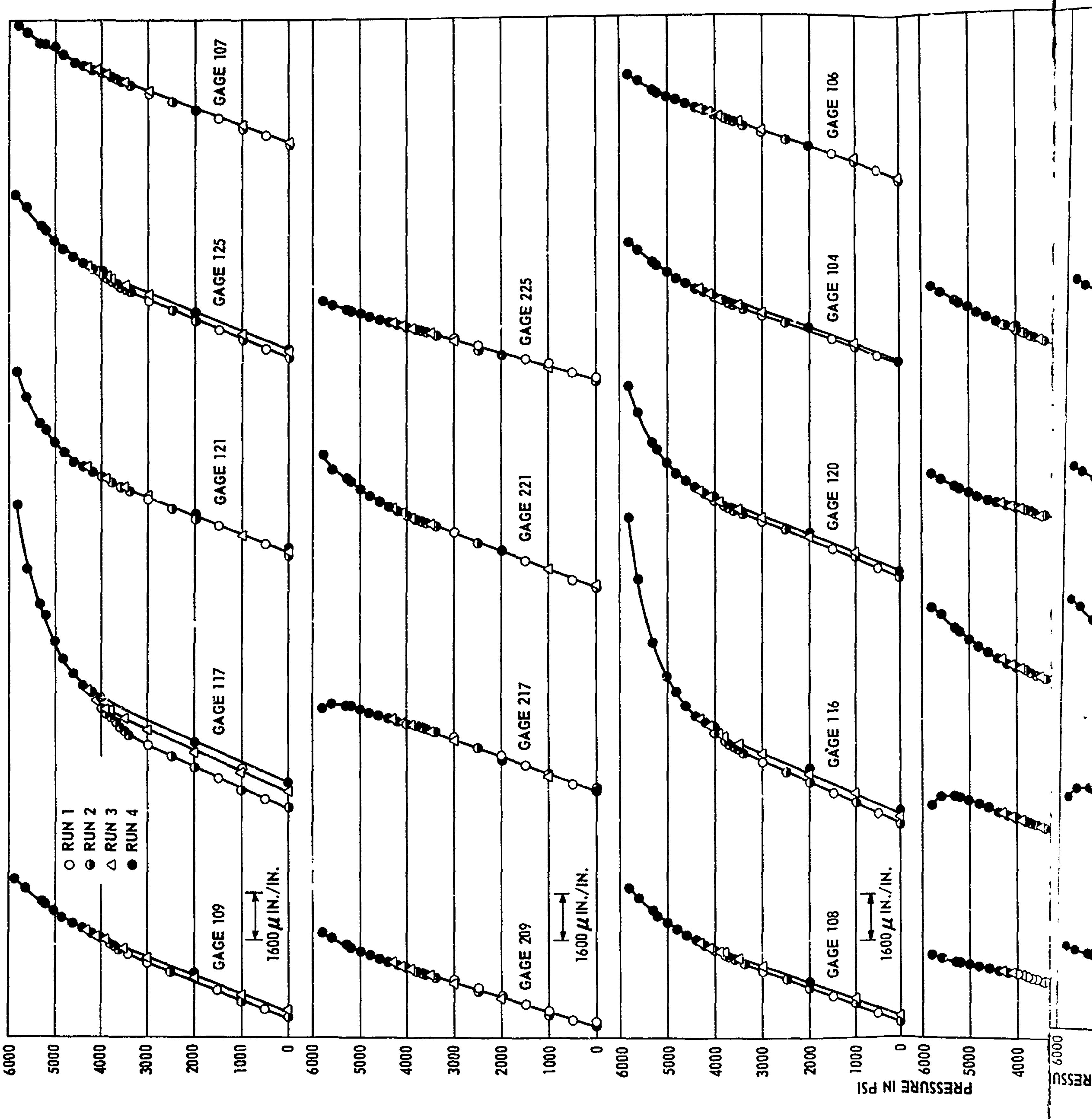
Model 83 collapsed at 6150 psi. Photographs of the collapsed model are shown in Figure 6.

Comparison of the collapse pressure of Model 83 with the initial design criteria showed that the model failed at 88 percent of the yield pressure P_y. Thus, the experimental result compared favorably with the initial design criteria even though the margin of stability was reduced from 3.4 to 2.8 because of properties and dimensions of available material. Thus, the estimate of the collapse depth for the assumed prototype should be conservative provided the margin of stability against local shell failure is sufficient. As mentioned previously, the test results of the HY-100 steel hemispheres indicate that this margin is sufficient.

The strain gage sensitivities presented in Figure 3 and the typical pressure versus strain plots shown in Figure 5 indicate the relatively minor effect of bending on total stress levels both adjacent to and away from the hemisphere boundary during that portion of loading where the relationship between pressure and strain remained linear–roughly 70 percent of the collapse pressure. There were only two cases in this load range where the total stress was greater than 12 percent of the membrane stress at locations where strain was measured. The largest contributions of bending to total stress levels were measured at Locations M and O (see Figure 3). In the first case, measurements were probably affected by discontinuity of the meridional stiffener in the adjacent bay; in the second case, they were probably affected by the presence of the external boundary stiffener. In both instances, the stress due to bending amounted to 17 percent of the membrane stress at that location.

Thus, it is reasonable to assume that for the geometry of this model ($\theta = 1.24$, $A_f/A_s = 0.151$), the effect of bending stress would be of little consequence in considerations of fatigue life. It is also probable that modest increase of the area-of-frame to area-of-shell ratio would increase the structural efficiency of the model design without significantly affecting bending stress levels. In addition, it is apparent that boundary effects did not materially affect the strength of the model.

To evaluate the test results reported herein, it is most meaningful to compare the strength of this stiffened hemisphere with that predicted for an unstiffened hemisphere of equivalent weight, i.e., equivalent shell thickness. To determine an equivalent shell thickness in this case, the stiffener volume can be considered, in effect, to be uniformly distributed over the surface area of a typical shell element. In this manner, the equivalent shell thickness is conveniently determined from



GAGE 109
GAGE 117
GAGE 121
GAGE 125
GAGE 107
GAGE 209
GAGE 217
GAGE 221
GAGE 225
GAGE 108
GAGE 116
GAGE 120
GAGE 104
GAGE 106
○ RUN 1
◐ RUN 2
△ RUN 3
● RUN 4
1600 μIN./IN.
PRESSURE IN PSI
6000
5000
4000
3000
2000
1000
0

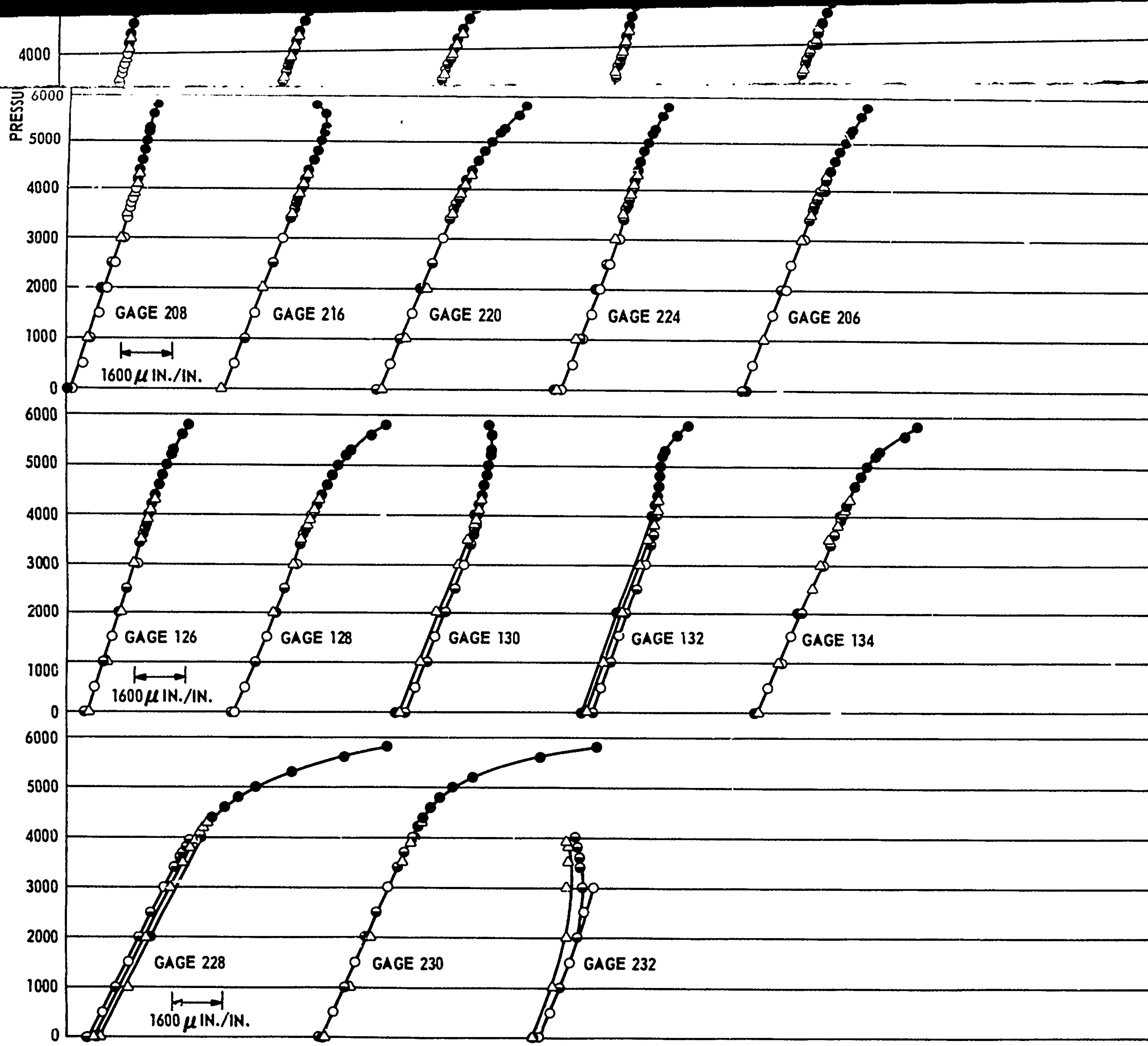


Figure 5 – Typical Pressure versus Strain Plots for Model 83

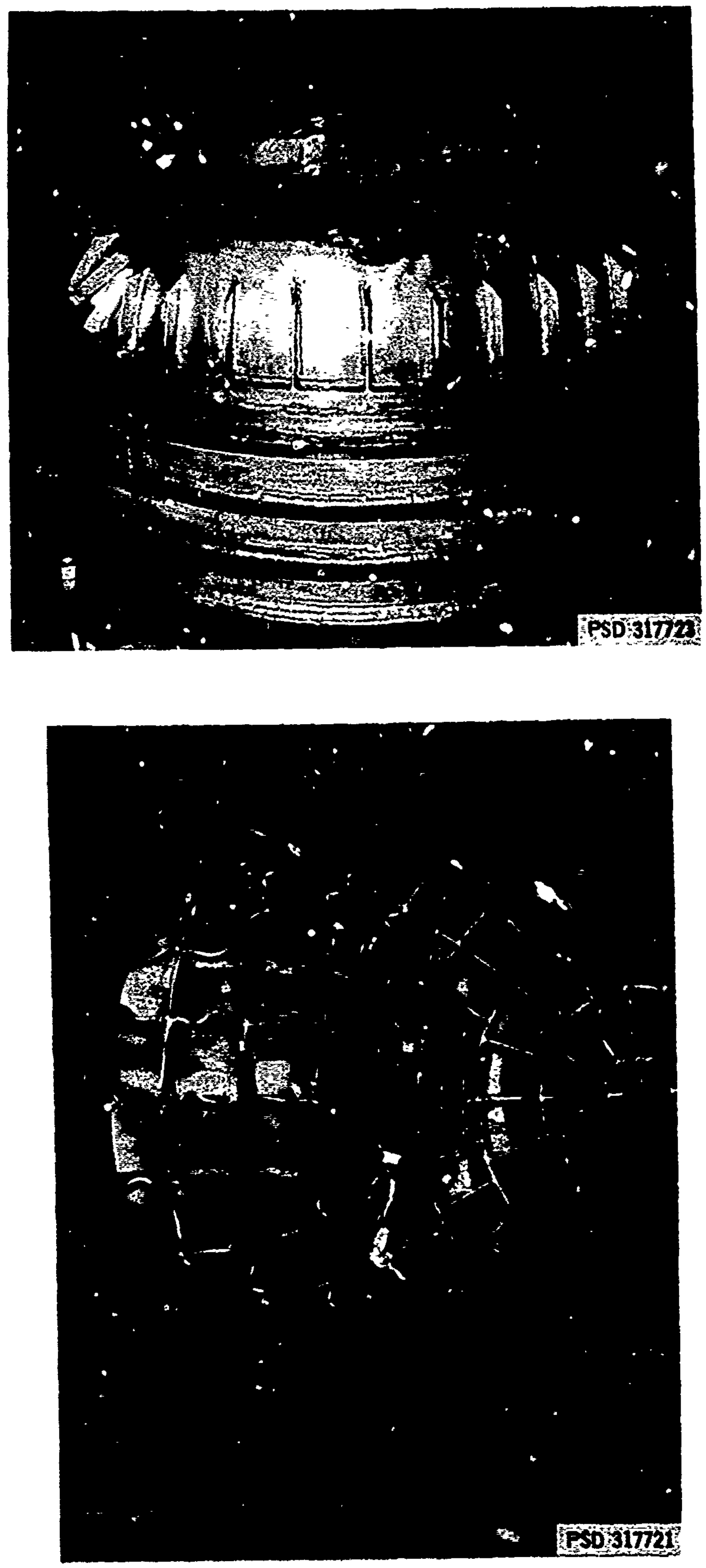


Figure 6 – Model 83 after Collapse

$$\tau = t_s \left(1 + 2\ \frac{b_w t_w}{b_s t_s}\right) = t_s \left(1 + 2\ \frac{A_f}{A_s}\right)$$

Elastic and inelastic collapse pressures for an equivalent thickness hemisphere with and without geometrical imperfections may be predicted from the analysis in Reference 6. As described therein, the collapse strength of shells with initial imperfections depends primarily on the local radius R_1 as determined from the deviations from a nominal radius and the out-of-roundness Δ over a critical arc length. Use of this local radius together with reduction factors derived from Model Basin test results on unstiffened spherical shells constructed according to full-scale fabrication procedures allows for prediction of collapse pressures for both a stress-relieved and an unstress-relieved sphere of equivalent thickness.

With the aid of the departures from sphericity plotted in Figure 3, the ratio of the local radius R_1 to the nominal radius R was found to be 1.13 for a shell of equivalent thickness (determined from the geometry of Model 83). This value of R_1/R is in good agreement with those values determined for other HY-150 steel, 30-in. diameter, unstiffened hemispheres tested at the Model Basin.

Comparison of the strength of Model 83 in accordance with the procedures outlined above showed that the experimental collapse pressure of the stiffened hemisphere was 1.34 times greater than the predicted failure of a fabricated unstress-relieved monocoque shell of equivalent weight.* Thus, it is apparent that a considerable saving in weight can be achieved through efficient stiffening of spherical shells. It is important to note also that the collapse pressure of the stiffened hemisphere was only 14 percent less than would be expected for a machined (near-perfect) stress-free, unstiffened shell of the same weight. This is particularly significant considering that the structural efficiency of the model could probably be improved by increasing the frame area and decreasing the thickness of the shell.

CONCLUSIONS

The following conclusions are drawn from the test results for the stiffened hemisphere reported herein:

1. Collapse pressures approaching those of near-perfect machined spherical shells may be obtained for particular stiffener-shell parameters.

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*This comparison is based on typical bay geometry. It may not be practical nor desirable from local shell stress considerations to achieve a constant ratio of stiffener area to shell area over the entire surface. However, with an internal stiffening configuration, the effect on efficiency should be small since in the present model, for example, the W/D ratio for the complete sphere was only 1 percent greater than that calculated using typical bay geometry.

2. The detrimental effects of initial imperfections and residual stresses arising from fabrication processes for monocoque shells may be at least partially overcome through use of properly designed stiffening systems.

3. The contribution of bending stress to total stress levels is small for the geometry of Model 83. Thus it is reasonable to assume that the effects of bending stress could be of little consequence in consideration of fatigue life for stiffened spherical shells with low values of A_f/A_s and the geometry parameter θ.

4. It is estimated that a stiffened spherical shell fabricated from HY-150 steel and designed for a collapse depth of 10,000 ft would weigh approximately 43 percent of its displacement.

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13. ABSTRACT

A stiffened steel hemisphere with a nominal yield strength of 150,000 psi was designed, fabricated, and tested to explore the structural efficiency of stiffened spherical shells. Test results show that the collapse pressure was approximately 30 percent greater than that predicted for an unstress-relieved, monocoque shell of equivalent weight with the same out of roundness. The collapse pressure approached that of a near-perfect, machined spherical shell. Thus, it appears that the detrimental effects of initial imperfections and residual stresses arising from fabrication processes for monocoque spherical shells may be at least partially overcome through use of properly designed stiffening systems. Based on the test results, it is estimated that an HY-150 stiffened steel spherical shell designed for a collapse depth of 10,000 ft would weigh 43 percent of its displacement.

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HYDROSTATIC TESTS OF A HIGH STRENGTH STEEL INTERNALLY STIFFENED HEMISPHERE, by Frank M. Schwartz. Jan 1967. ii, 16p. illus., graphs, diags., tabs., refs.
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A stiffened steel hemisphere with a nominal yield strength of 150,000 psi was designed, fabricated, and tested to explore the structural efficiency of stiffened spherical shells. Test results show that the collapse pressure was approximately 30 percent greater than that predicted for an unstress-relieved, monocoque shell of equivalent weight with the same out of roundness. The collapse pressure approached that of a near-perfect, machined spherical shell. Thus, it appears that the detrimental effects of initial imperfections and residual stresses arising from fabrication processes for monocoque spherical shells may be at least

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