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DAVID TAYLOR MODEL BASIN

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FEASIBILITY OF A CONNECTED SPHERE PRESSURE HULL FOR THE RESCUE VEHICLE OF THE DEEP-SUBMERGENCE SYSTEMS PROJECT

by

Martin A. Krenzke

and

Gerald D. Ward

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STRUCTURAL MECHANICS LABORATORY

RESEARCH AND DEVELOPMENT REPORT

July 1965 Report 2007

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ABSTRACT

The feasibility of a connected-sphere pressure hull for possible use in the rescue vehicle of the Deep-Submergence Systems Project is studied through the design and tests of a simple, small-scale model. Of particular concern is the performance of the hull at the sphere-cylinder juncture. The reinforcement of this juncture in the model is accomplished by using a membrane design concept. The elastic behavior and collapse strength verify the validity of the design approach as well as the feasibility of the connected sphere configuration for pressure hull applications.

ADMINISTRATIVE INFORMATION

The work described in this report was conducted under Special Projects Office, Department of the Navy, Project Order 5-0003 of 3 December 1964.

INTRODUCTION

The Deep-Submergence Systems Project (DSSP) has been established under the direction of the Navy's Special Projects Office to develop improved systems related to location, identification, rescue from, and recovery of objects on the ocean floor. Two distinct vehicles, the rescue vehicle and the search vehicle, will be developed as part of these systems. Present plans are to have six rescue vehicles and four search vehicles operational by 1970.

As a result of early feasibility studies, the Preliminary Design Group of the Bureau of Ships proposed two basic configurations for the pressure hull of the rescue vehicle. One configuration consists of two 7-ft-diameter spheres connected by a short, small-diameter cylinder. The second configuration is a 7-ft-diameter, 7-ft-long cylinder with hemispherical end closures. Although the Navy has considerable experience in designing cylindrical hulls, the connected-sphere design has no precedent. Of particular concern is the performance of the hull at the sphere-cylinder juncture.

To establish the feasibility of the proposed connected-sphere configuration, the Taylor Model Basin designed and tested a simple, small-scale model of the projected prototype. Titanium was arbitrarily selected as the hull material. The analyses used to determine the basic hull scantlings have been reported in Reference 1.* The procedure used to design the sphere-cylinder juncture is presented in this report together with the results of the model test.

JUNCTURE ANALYSIS

The design of the sphere-cylinder juncture was accomplished through the use of a membrane design concept previously applied to cone-cylinder and cylinder-hemisphere junctures as well as to axisymmetric penetrations in spherical shells.[2,3] The underlying principle of this concept is that an effective reinforcement is one which results in zero rotation and a deflection which is equal to the membrane deflection of the critical structural element. This is achieved through an iteration process by successively determining the loads on the juncture assuming zero rotation and membrane deflection and then solving for the area, shape, and position of the reinforcement required to satisfy the initial assumptions.

The notation used in deriving the basic equations is presented in Figure 1. From the assumption of membrane stresses in the sphere, the following general equations evolve:

$$\theta_R = \theta_C = - \theta_S = 0 \qquad [1]$$

$$\Delta_R = \Delta_C = \Delta_S = \frac{R_{os} R_{ms} (1 - \nu) \sin\phi}{2 E h_s} \qquad [2]$$

where θ_R is the rotation of the reinforcement,

θ_S is the rotation of the sphere at its juncture with the reinforcement,

θ_C is the rotation of the cylinder at its juncture with the reinforcement,

Δ_R is the deflection of the reinforcement normal to the longitudinal axis,

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*References are listed on page 13.

Δ_C is the deflection of the cylinder at its juncture with the reinforcement normal to the longitudinal axis,

Δ_S is the deflection of the sphere normal to the longitudinal axis at its juncture with the reinforcement,

ν is Poisson's ratio, and

E is Young's modulus.

The deflection and rotation of the reinforcement may also be expressed as

$$\Delta_R = \frac{F_R \, (R_{CG})^2}{EA_R} \qquad [3]$$

$$\theta_R = \frac{(R_{CG})^2 \, M_R}{EI_R} = 0 \qquad [4]$$

where A_R is the area of the reinforcement,

I_R is the moment of inertia of the reinforcement,

F_R is the resultant force on the reinforcement, and

M_R is the resultant moment on the reinforcement.

As a result of Equations [1] and [2], M_s and $H_s = 0$. M_c and H_c may easily be found for the specified deflection and zero rotation by the method of influence coefficients described in Reference 4.

By combining Equations [2] and [3], A_R may be expressed as

$$A_R = \frac{2 \, F_R \, (R_{CG})^2 \, h_s}{R_{os} \, R_{ms} \, (1 - \nu) \sin\phi} \qquad [5]$$

By assuming the shape and size of the reinforcement, F_R can be estimated from equilibrium. The calculated area of reinforcement from Equation [5] may be compared with the assumed area. The correct area is determined by iteration.

The reinforcement must have the proper shape in addition to the correct area before Equations [1] and [2] are satisfied. After a fairly close approximation of the correct area is obtained through use of Equation [5], a check should be made on the rotation of the juncture. If the resultant moment M_R acting on the reinforcement is of significant value, a new shape of the reinforcement must be assumed; and the above procedure is repeated until the assumptions of zero rotation and membrane deflection are satisfied.

DESCRIPTION OF MODEL

Model RS-1 was machined from 6AL4V bar stock having a nominal yield strength of 126,000 psi and a Young's modulus of 18×10^6 psi. Scantlings of Model RS-1 together with those of the projected prototype are shown in Figure 2. A slight increase in shell thickness was provided at the meridional joints to ensure against premature failure at these locations. The sphere-cylinder junctures were designed according to the procedure described previously. A typical compressive stress-strain curve for the material used is presented in Figure 3. This representative curve was obtained by loading several uniaxial specimens at a stress rate of 250 psi/min beyond the proportional limit of the material.

The selection of the design operating depth and hull material was rather arbitrary but was based on the following considerations. The desired operating depth of the rescue vehicle is 6000 ft.* This depth is greater than required for the primary mission but is desirable for fulfilling a secondary mission of conducting oceanographic research. Tradeoff studies indicated that if an operating depth of 6000 ft and a weight-to-displacement ratio for the pressure hull of less than 0.4 are required, HY-110 titanium would be best suited for the hull material.

After the operating depth and hull material as well as the basic configuration had been decided, the shell thicknesses were determined by the procedures outlined in Reference 1. The shell thicknesses thus determined are a function of fabrication procedures and tolerances. The

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*A factor of safety of 1.5 is assumed.

geometry represented by Model RS-1 is for a stress-relieved hull having maximum initial departure from sphericity of about 0.1 in.[1]

TEST PROCEDURE AND RESULTS

Two separate hydrostatic pressure tests were conducted. First, a test was conducted with one hemispherical end of the model removed. The open end was placed against a flat closure plate which was equipped to take out strain-gage wires, and elastic strain data were recorded. Measured elastic strain sensitivities, the initial slopes of the pressure-strain plot, are presented in Figure 4.

The second test was conducted on the complete model, and it was tested to collapse. Pressure increments of 25 psi were applied every 2 min at pressure levels above 4000 psi until collapse occurred. This represented a loading rate which approximated that applied to the uniaxial specimens when obtaining representative stress-strain curves for the material. These loading procedures were used to minimize the effect of creep on the correlation of experimental and calculated pressure.

Model RS-1 collapsed at a pressure of 6175 psi. A photograph of the collapsed model is shown in Figure 5. It appears that collapse initiated about 45 deg from the crown area in one of the end hemispheres.

DISCUSSION AND CONCLUSIONS

The test results for Model RS-1 firmly establish the structural feasibility of the connected-sphere configuration for the pressure hull of the rescue vehicle as well as for other applications.

The strain sensitivities presented in Figure 4 indicate that negligible bending occurred in the spherical shell at its juncture with the reinforcement. This is consistent with the membrane design concept and once again verifies the validity of this approach.

The experimental collapse pressure can be accurately calculated by the Model Basin empirical analysis for near-perfect spherical shells.[5] A collapse pressure 3 percent below the experimental collapse pressure is obtained by use of this analysis. Since this same analysis has been used to consistently predict the collapse strength of near-perfect spheres and hemispheres with ideal boundaries, it can be concluded that the sphere-cylinder juncture did not affect the strength of Model RS-1. The apparent location of failure lends additional support to this conclusion.

Although the test of Model RS-1 establishes the feasibility of this configuration, there are several significant areas in which a test of this nature does not provide sufficient information. First, the model was too small to permit detailed evaluation of stresses, particularly in the reinforcement and its connection with the sphere. In addition, the complete model was machined and thus did not reflect the presence of bending, which would be introduced if mismatch were present in a welded connection. Larger models fabricated according to full-scale procedures are required to adequately project these effects to prototype performance.

The collapse pressure of Model RS-1 is not indicative of the strength of a prototype hull. This is rather obvious since (1) the yield strength of the material used in the model was considerably greater than the assumed yield strength of the prototype (110,000 psi), (2) the model was free of most of the structural details which are necessary in the prototype, and (3) the model possessed near-perfect geometry, whereas a prototype would most likely have initial departures from sphericity as well as possible residual stresses. With the possible exception of Item 2, each of the above factors contributed to the additional strength of the model when compared to that of the projected prototype. These factors are discussed and evaluated in detail in Reference 1. The effect of such structural details as hatches, viewing ports, and rescue skirts (Item 2) on collapse strength could also be significant if proper attention is not given to them in the design process.

Finally, it is worth noting that the same analysis used to design the sphere-cylinder juncture of Model RS-1 may be used to design the juncture of nested spheres which do not have connecting cylinders. The performance of nested spheres designed according to this procedure should be similar to that of Model RS-1. However, large reinforcing rings and associated weight penalties will result for relatively large openings.[1]

ACKNOWLEDGMENTS

The continuing cooperation of Messrs. H. Bernstein of the Special Projects Office, W. Ferrara of the Bureau of Ships, and F. Dunham of the Model Basin is acknowledged. The design calculations for the sphere-cylinder juncture were ably performed by Messrs. R. Jones and J. Nott.

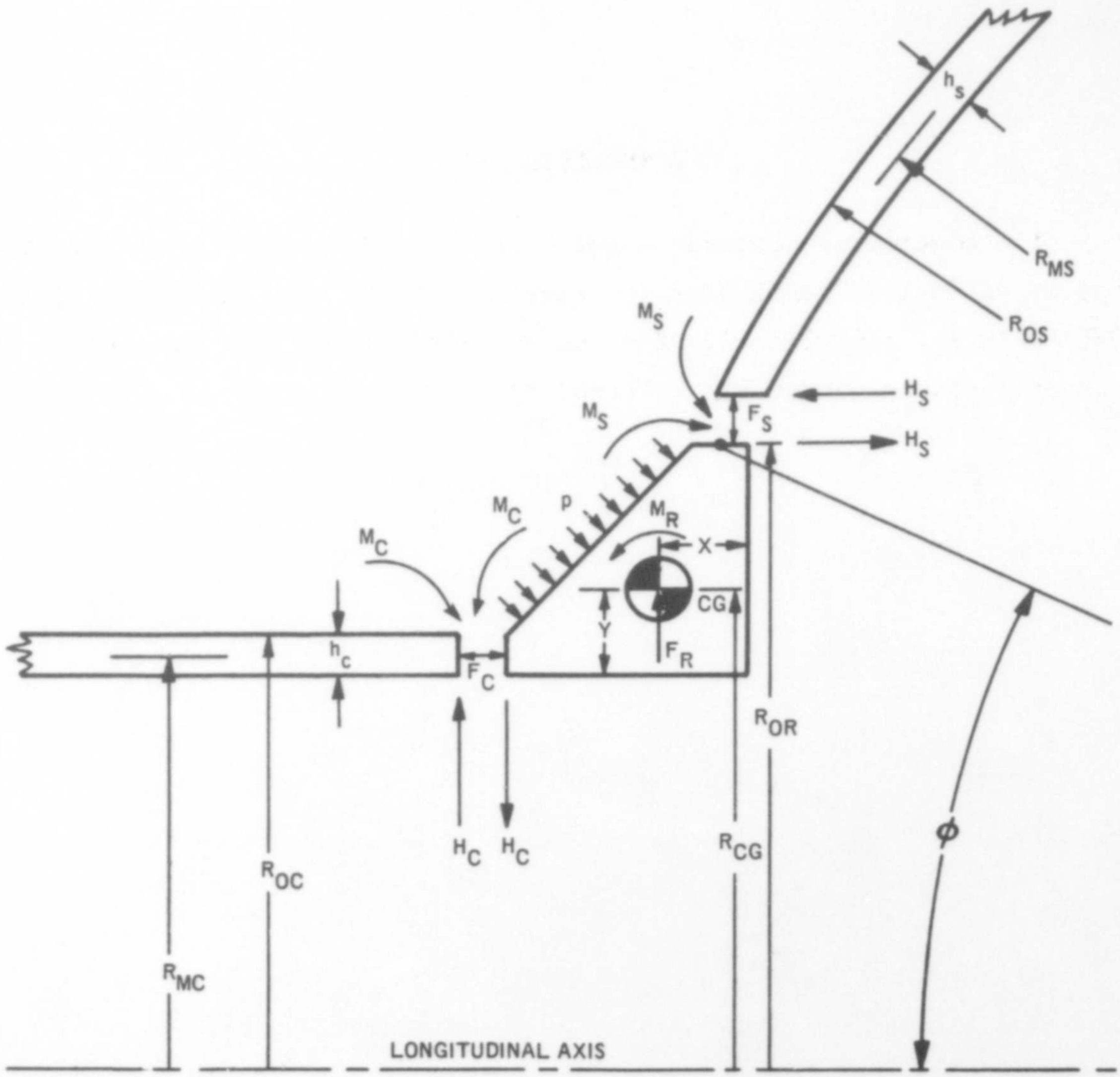

Figure 1 – Notation and Calculated Distribution of Loads for Sphere-Cylinder Juncture

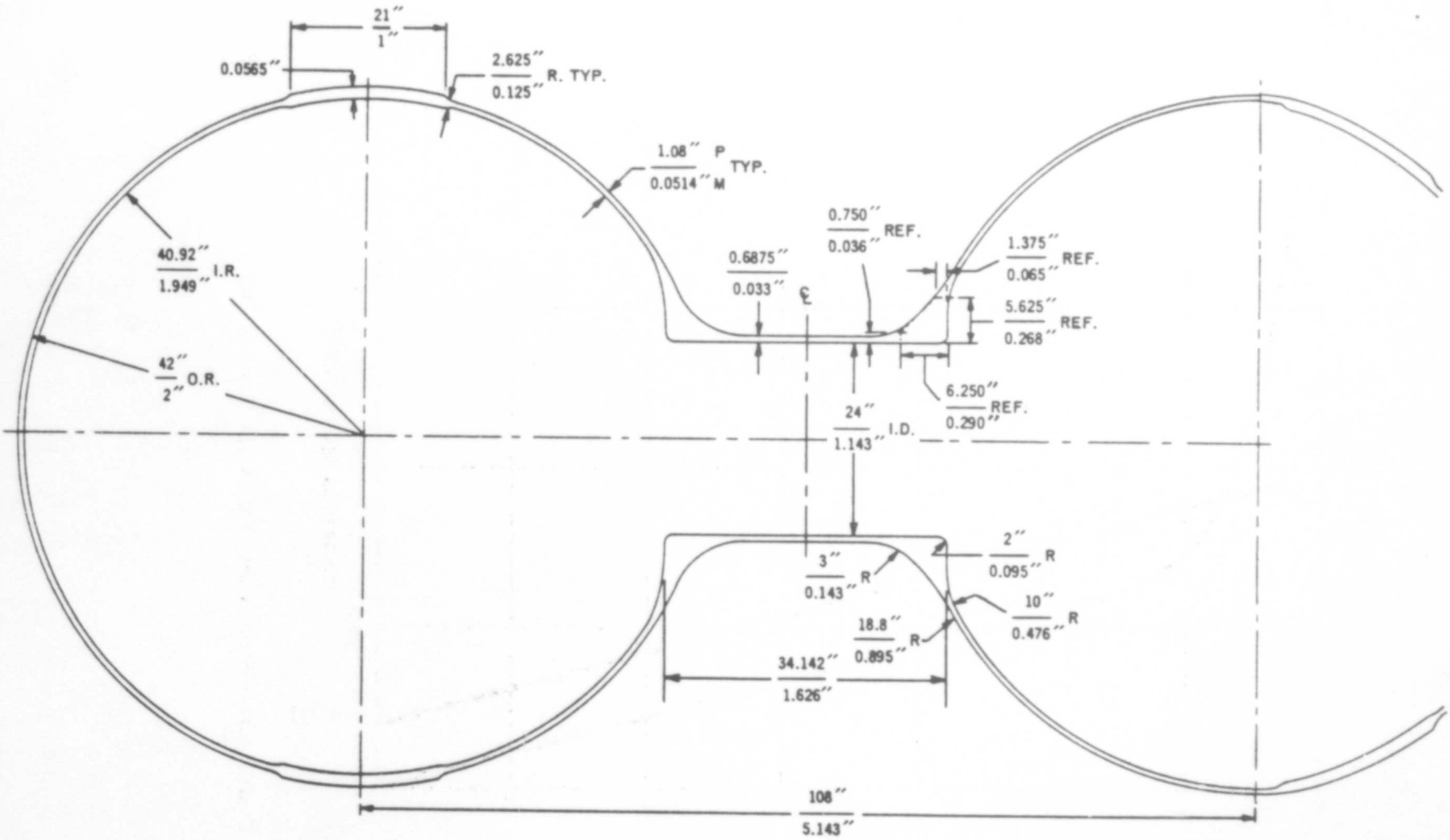

Figure 2 – Model RS-1

Dimensions given are nominal values. Model RS-1 was machined within ±0.002 in. of these values.

P indicates projected prototype dimensions

M indicates model dimensions

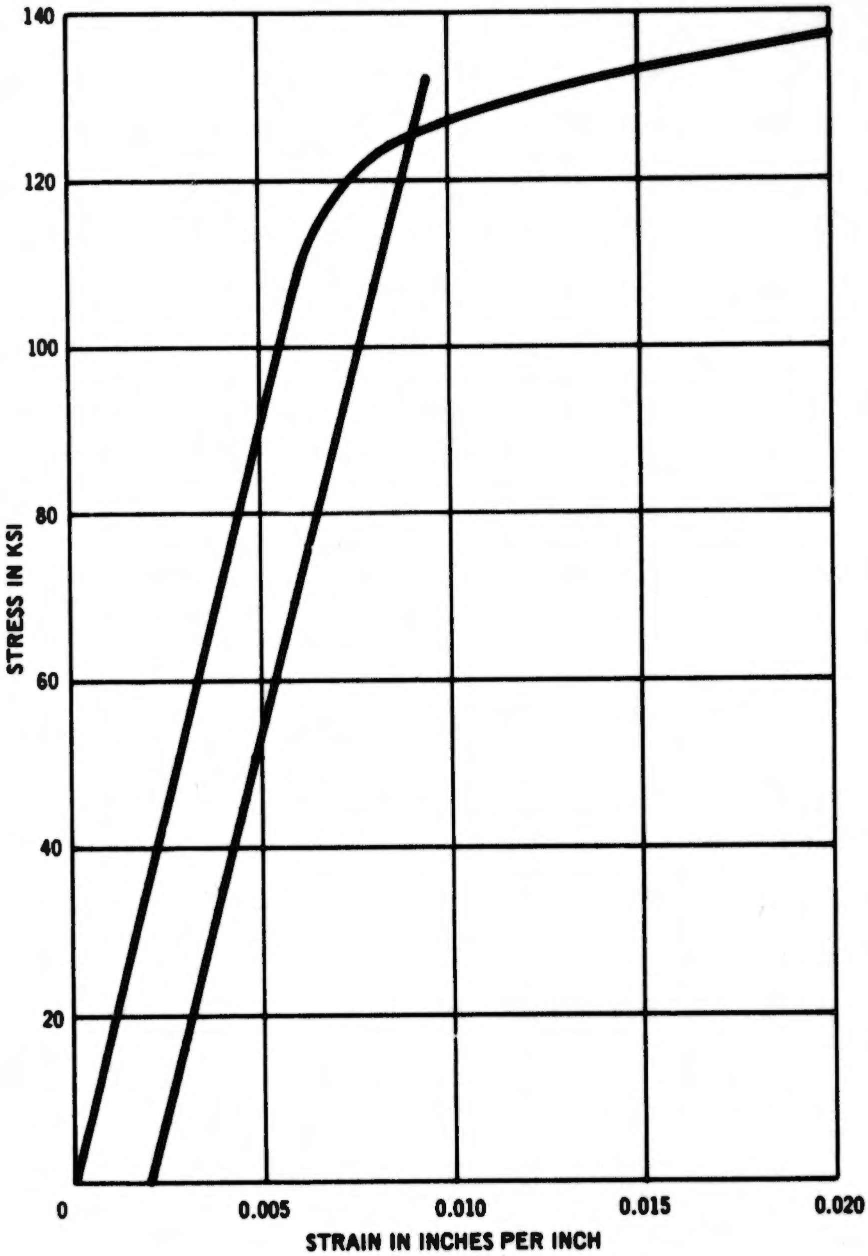

Figure 3 – Typical Stress-Strain Curve for 6AL4V Titanium Alloy Used in Model RS-1

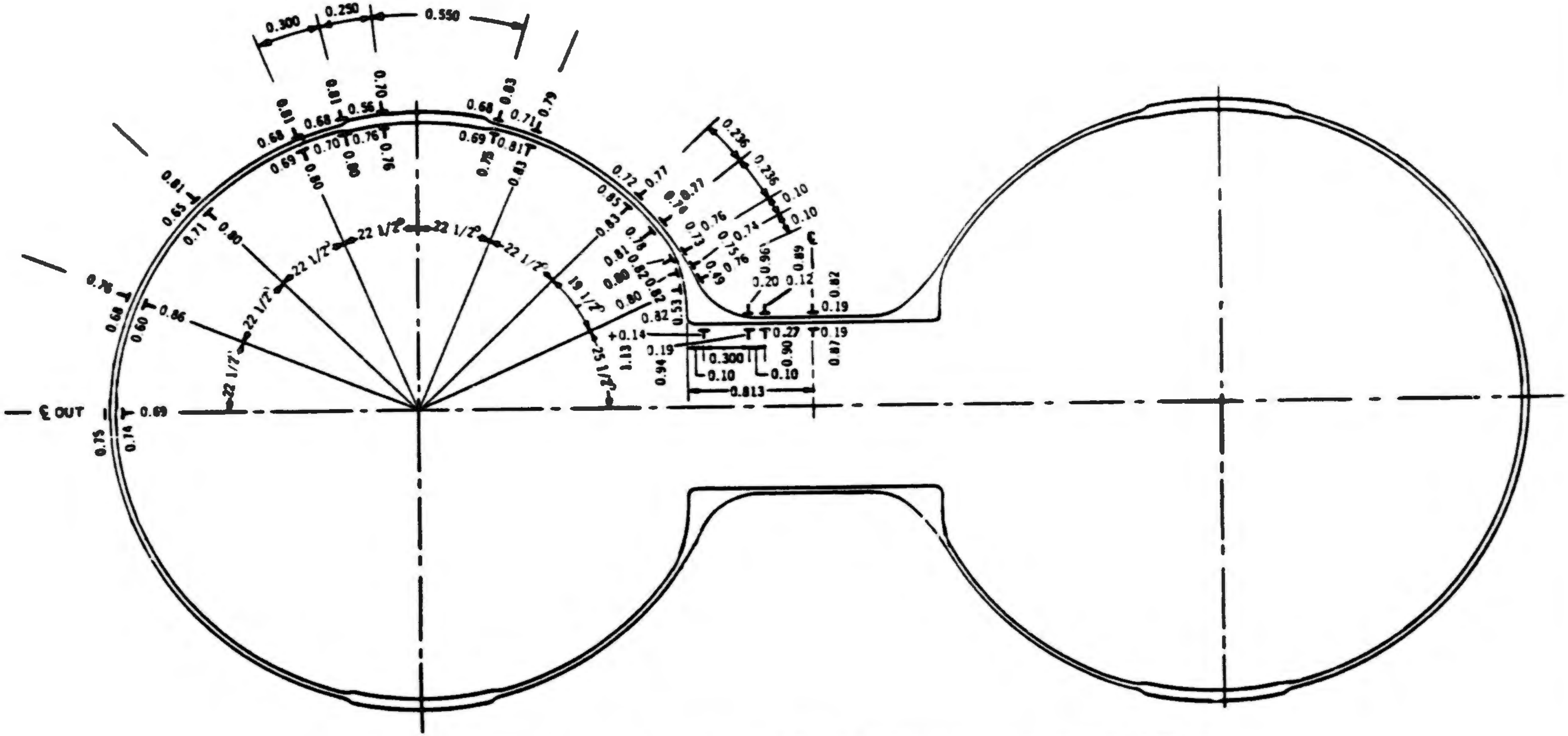

Figure 4 – Strain-Gage Locations and Strain Sensitivities for Model RS-1

*Asterisk indicates average of two gages located at 0 and 90 deg. In no case is the average more than 0.03 μ-in./in. from maximum or minimum sensitivities. All strain sensitivities are compressive except for one associated with + sign.

Wire-resistance strain gages with a gage length of 0.031 in. were used on the spherical portion of model. The same type gage with a gage length of 0.063 in. was used on the cylindrical portion and around juncture. All dimensions are in inches.

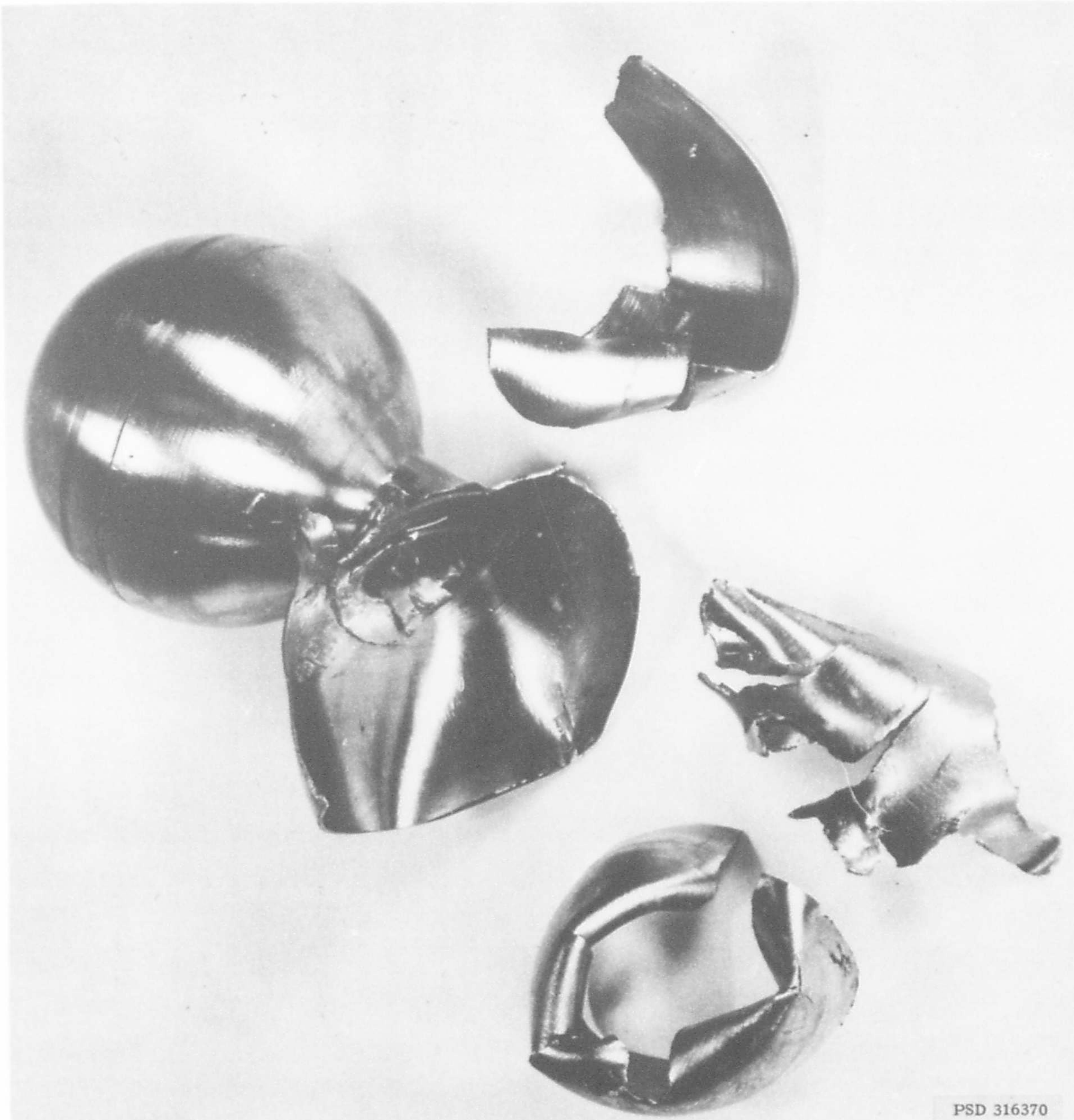

Figure 5 – Model RS-1 after Collapse

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