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Report 1725

DAVID TAYLOR MODEL BASIN
WASHINGTON D.C.

DEPARTMENT OF THE NAVY

HYDROMECHANICS

AERODYNAMICS

STRUCTURAL MECHANICS

APPLIED MATHEMATICS

TESTS OF RING-STIFFENED AND SANDWICH COMPOSITE CYLINDERS UNDER EXTERNAL HYDROSTATIC PRESSURE

by

M.A. Krenzke
and
T.J. Kiernan

STRUCTURAL MECHANICS LABORATORY

RESEARCH AND DEVELOPMENT REPORT

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ABSTRACT

Four cylindrical models were tested under external hydrostatic pressure to determine the structural behavior of ring-stiffened and sandwich hulls of composite construction. Of particular interest in this series of tests were the effects of compartment length on collapse strength in the plastic general-instability mode and the relative strength-weight characteristics of ring-stiffened and sandwich cylindrical hulls. The test results demonstrated the importance of representing the actual prototype compartment length when testing a model which may collapse in the plastic general-instability mode. These tests also demonstrated that a semi-infinite sandwich hull normally will have less than a 10-percent strength advantage over an optimum semi-infinite ring-stiffened hull of the same material and weight. Based on these and earlier results, the strength-weight characteristics of composite semi-infinite cylindrical hulls of various combinations of materials are estimated.

INTRODUCTION

The development of underwater vehicles with positive buoyancy to operate at great depth is of particular interest to oceanographers, who desire to explore the oceans and their floors, and to naval strategists, who are studying the possible military advantages that such a vehicle may offer. Of obvious importance in the design of such a deep-sea vehicle is the achievement of a high strength, low density pressure hull. The basic approach to this structural problem must be through the use of hull materials with high strength-to-weight ratios.

Unfortunately, many of the hull materials which show favorable strength-weight characteristics cannot be fabricated satisfactorily with current procedures, particularly in the thicknesses required. Examples of these materials are superstrength steels, high strength aluminum and titanium alloys, and reinforced plastics. In an effort to find a method of construction which enables the use of these and other nonweldable materials as they become available, the Model Basin investigated a new concept in pressure hull design.[1] This concept, referred to herein as composite construction, involves the use of a thin jacket encasing rings of high

[1]References are listed on page 27.

strength material. In an operating underwater vehicle, the jacket would hold the strength components in place and provide watertight integrity, longitudinal strength to resist bending moments, and corrosive protection for the strength elements. A more detailed presentation of composite construction concepts is given in Reference 1.

Under sponsorship of the Bureau of Ships, the Model Basin is currently conducting a rather extensive structural model program to further investigate the use of composite construction for deep-depth pressure hulls with collapse depths between 5000 and 30,000 ft. These investigations include experimental studies of fabrication techniques, methods of penetrating and closing off the ends of composite cylinders, elastic behavior, static collapse strength, fatigue life, and dynamic characteristics. HY-220 steel, HY-60 to HY-80 aluminum alloys, HY-140 and HY-200 titanium alloys, and glass-reinforced plastics are being used as the strength elements. Materials used in the jackets include HY-80 and HY-100 steel, an HY-30 aluminum alloy, an HY-120 titanium alloy, and a fiberglass-reinforced plastic with a nominal yield strength of 35,000 psi.

This report describes the static tests of four models designed to investigate the structural behavior of ring-stiffened and sandwich composite cylinders to depths of 20,000 ft, particularly their relative strength-weight characteristics and the effect of compartment length on collapse strength in the plastic general-instability mode. Estimates of the strength-weight characteristics of composite semi-infinite cylindrical hulls of various combinations of materials are presented on the basis of the results of these and earlier tests.

DESCRIPTION OF MODELS

Four models of composite construction, designated DSRV-1, DSRV-1L, DSRV-4M, and DSRV-4L, were fabricated. Aluminum was selected as the basic hull material because of its ease of fabrication. Models DSRV-1 and DSRV-1L were ring-stiffened cylinders of machined 7079-T6 aluminum rings placed inside an HY-100 steel jacket. Models DSRV-4M and DSRV-4L were sandwich-type cylinders of machined 7079-T6 aluminum rings inside a 5086-H32 aluminum jacket. Sketches of the models are shown in Figure 1. Table 1 presents a summary of model geometries and material properties together with associated geometries and assumed material properties for an arbitrarily selected 10-ft-diameter prototype hull. Representative stress-strain curves for the basic hull material of each model are shown in Figure 2.

The effect of bulkhead spacing, or overall length, on collapse strength was of particular interest in this series of tests. Models DSRV-1 and DSRV-4M were approximately one diameter long and the radial deflection of the ends was restricted. Thus, they represented cylinders of finite bulkhead spacing. Models DSRV-1L and DSRV-4L were four diameters long and the ends were permitted to deflect radially. The bulkhead spacing of four diameters minimized the influence of compartment length on collapse pressure.[2] Thus, these models represented cylinders of semi-infinite length.

The typical bay geometry for each model was selected to provide a collapse pressure of 6667 psi, equivalent to a collapse depth of 15,000 ft based on a yield strength of 100,000 psi for HY-100 steel, 30,000 psi for 5086-H32 aluminum, and 62,000 psi for 7079-T6 aluminum. The total cross-sectional area of the material required for Models DSRV-1 and DSRV-1L was determined by arbitrarily setting the average circumferential stress in the frame and shell at collapse equal to 1.05 times the average yield strength of the section. The typical section of Models DSRV-1 and DSRV-1L had a ratio of weight of hull to weight of displacement of about 56 percent. The geometry of Models DSRV-4M and DSRV-4L was selected after Model DSRV-1 was tested and was influenced by the favorable results. Based on these favorable results of DSRV-1 and the anticipated advantages in structural efficiency of a sandwich hull as compared to a ring-stiffened hull, the average circumferential stress in the combined web, shell cross section of the sandwich hull models DSRV-4M and DSRV-4L was arbitrarily set equal to 1.15 times the average yield strength at the design collapse pressure of 6667 psi. Thus, the stress intensity at collapse was approximately equal to the two-dimensional Hencky-Von Mises yield stress developed in an unstiffened cylinder of equivalent weight and material. The typical section of Models DSRV-4M and DSRV-4L had a weight-to-displacement ratio of about 52 percent.

The typical bay geometries of the two ring-stiffened composite cylinders, Models DSRV-1 and DSRV-1L, were identical when scaled to the same diameter. Since the basic material, 7079-T6 aluminum, is a strain-hardening material, it was considered necessary to obtain uniform stress levels throughout the hull and to provide adequate margin against elastic instability[2,3] to enable utilization of the full strength of the material. Therefore, stiffeners were placed at relatively close intervals to minimize bending[4] and to provide very high elastic shell stability.[3] The size and shape of the stiffeners were such as to provide an elastic general-instability pressure for a semi-infinite cylinder of 2 1/2 times the design collapse pressure.[2] The thickness of the outer steel jacket was selected to provide sufficient strength to resist bending loads which might occur while surfaced.

The typical bay geometries of the sandwich composite cylinders DSRV-4M and DSRV-4L were also similar and were selected on the same stability considerations as used for the ring-stiffened cylinder. The inner and outer shell rings had a variable thickness designed to eliminate bending according to the theory of Short.[5]

The models were assembled in a manner feasible for large diameter hulls. The first step in the assembly was the welding of the jacket to an end ring and the slipping of the nonweldable rings in place. The nominal diametrical clearance between the outer jacket and the inner ring of each model corresponded to 5/32 in. for a 10-ft-diameter prototype hull. The jackets for the shorter models were formed from a single shell; the jackets for the larger models consisted of four shells rolled into cylinders and joined by circumferential welds. The longer models were assembled by first inserting the aluminum rings to within several inches of the free end of the first section of jacket. Then the second section of jacket was welded in place, the weld was ground smooth on the inner surface, and the inner rings were placed again within several inches of the free end. The entire models were assembled in this manner until all rings were in place. The final step in the assembly of each model was the joining of the second weldable end ring and the outer jacket by a single circumferential weld.

TEST PROCEDURE

Foil resistance strain gages were used to measure strains in the longitudinal and circumferential directions of each model. Gage location diagrams are shown in Figure 3.

Model DSRV-1 was subjected to a pressure of 3000 psi in the 37-in. tank at the Model Basin. Oil was used as a pressure medium to eliminate the need for waterproofing the strain gages. The model was tested to collapse in the 4-ft-diameter pressure tank located at the Naval Research Laboratory. Here water was used as a pressure medium; no strain measurements were made since facilities were not available to bring strain-gage leads out of the tank.

Models DSRV-1L, DSRV-4M, and DSRV-4L were tested in the TMB 17 1/2 in.-diameter, high pressure tank. Water was used as a pressure medium, and strains were recorded during each test.

Special attention was given to the rate at which pressure was applied to each model. Each pressure increment was held at least 5 min, and the final pressure increment did not exceed 2 percent of the observed collapse pressure.

TEST RESULTS

Models DSRV-1, DSRV-1L, DSRV-4M, and DSRV-4L withstood maximum pressures of 8000, 7350, 8450, and 7800 psi, respectively. Models DSRV-1 and DSRV-4M were tested to destruction. Models DSRV-1L and DSRV-4L were tested to pressure levels at which excessive creep was observed from the strain readings while the pressure load remained constant. When it was decided that the maximum attainable pressure had been reached, the pressure was dropped off before complete destruction of Models DSRV-1L and DSRV-4L occurred.* When these models were removed from the tank, a maximum out-of-roundness of about 3/8 in. was observed in each Model. Typical plots of pressure versus strain are presented in Figure 4.

DISCUSSION

All four models apparently collapsed in the plastic general-instability mode. Models DSRV-1L and DSRV-4L definitely would have failed by

*Models DSRV-1L and DSRV-4L will be used at a future date to investigate the resistance of composite aluminum hulls to dynamic loading.

plastic general instability. This is demonstrated both by the measured out-of-round shape after the test and by the bifurication of the recorded circumferential strains in the frames as shown in Figure 4. The circumferential strains recorded at pressures near the collapse pressure indicate that Model DSRV-4M also failed by plastic general instability. The damage, however, was too extensive to determine the mode of collapse by visual inspection. No strains were recorded near collapse on Model DSRV-1. Moreover, the model was damaged too extensively to determine the mode of failure after collapse occurred. However, the elastic shell instability, as calculated by theory for monolithic shells,[3] was about double the elastic general-instability collapse pressure.[2] Thus it appears that it also failed in the overall mode.

An analysis for determining the plastic general-instability collapse strength of a cylindrical hull is presented in Reference 1. The theoretical collapse pressures calculated using this analysis are compared in Table 2 with the observed collapsed pressures of these models and those of two composite titanium sandwich models[1] which also failed in the plastic general-instability mode during previous tests at the Model Basin. Excellent agreement between the theoretical and experimental collapse pressures was obtained for each model.

One method of evaluating the structural efficiency of a stiffened cylinder under external hydrostatic pressure is to compare its experimental and theoretical collapse pressures with the theoretical collapse pressure in the yield mode of an unstiffened cylinder of the same size, weight, and material. Table 3 presents structural efficiency factors* for this series of models. The theoretical collapse pressures of the equivalent unstiffened cylinders were obtained by applying the Hencky-Von Mises yield criterion[6] to the three-dimensional stresses at midplane as obtained by the Lame[6] solution of the stresses in a thick-walled cylinder. It is realized that the 0.2-percent offset method of obtaining yield strength, which is used in

*The structural efficiency factor is defined as the ratio of the collapse pressure of a stiffened cylinder to the theoretical pressure at which collapse occurs by yielding for an equivalent unstiffened cylinder.

all the strength computations in Table 3, is arbitrary and, therefore, factors of more than 1 are attainable.

Table 3 shows that a relatively high degree of structural efficiency was achieved in each model. These high structural efficiency factors, together with the high ratios of experimental to theoretical collapse strength indicated in Table 2, demonstrate that composite hulls are as efficient as monolithic machined hulls. Since residual stresses reduce the collapse strength of welded hulls,[7] a properly designed composite hull may be more efficient than a welded hull of similar material and geometry.

These tests demonstrate the effect of bulkhead spacing on the collapse strength of cylindrical hulls in the plastic general-instability mode. For example, the structural efficiency factor for the long ring-stiffened hull, Model DSRV-1L, was about 10 percent below that of the short ring-stiffened hull, Model DSRV-1. The efficiency of the long sandwich hull, Model DSRV-4L, was 9 percent below that of the short sandwich hull, Model DSRV-4M. The significance of this comparison is that it demonstrates the importance of representing the actual prototype compartment length when testing a model to determine the collapse strength of a hull which may collapse in the plastic general-instability mode. If it is not practical to test a section representing the full compartment length, it appears that the effect of overall length of cylindrical shells with closely spaced frames may be estimated using the plastic general-instability analysis outlined in Reference 1.

The strength-weight advantages of sandwich hulls over ring-stiffened hulls which collapse in the plastic general-instability mode may be estimated by comparing the structural efficiency factors of Models DSRV-1L and DSRV-4L. These models lend themselves to this type of comparison because:

1. Neither model was affected to any extent by residual stresses.

2. The shape of the stress-strain curves for each of the basic hull materials was very similar.

3. Each model failed in the plastic general-instability mode.

4. The ratio of elastic to inelastic collapse strength was of the same magnitude for both models.

5. Each model had favorable stress conditions since both were designed to minimize bending.

A comparison of the structural efficiency factors given in Table 3 for Models DSRV-1L and DSRV-4L shows that the sandwich hull was about 6 percent stronger than the ring-stiffened hull experimentally and only about 4 percent stronger theoretically.* This advantage of sandwich hulls over ring-stiffened hulls will vary with collapse depth for any given material, but it now appears that it is not likely to exceed 10 percent when comparing truly optimum designs of each type which collapse in the plastic general-instability mode.**

Previous estimates[8] made by the Model Basin, of up to a 20-percent advantage were not based on optimum design of both sandwich and ring-stiffened cylinders. In this previous comparison, geometrical configurations of the ring-stiffened cylinders were restricted and resulted in a less than optimum design. Thus the strength advantage of sandwich hulls over ring-stiffened hulls may be marginal at many depths and nonexistent for some materials at very shallow or very deep depths. The main advantage, therefore, which many sandwich hulls offer is the use of thinner plating. Offsetting this advantage are the inherent fabrication problems of sandwich construction.

The measured strains indicate that the strain distributions were very favorable in each model. Bending did not play an important role in the ring-stiffened models since the maximum measured strains occurred in the circumferential direction in the frames. The shell thickness of Models DSRV-4M and DSRV-4L varied between stiffeners in a manner which theoretically eliminates bending in a typical section.[5] Strain measurements on Models DSRV-4M and DSRV-4L indicate that some bending did occur, however, since the longitudinal strains on the inner surface of the inside

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*Similar results may be obtained by comparing the efficiency factors of the shorter ring-stiffened and sandwich hulls, Models DSRV-1 and DSRV-4M. However, this comparison favors the sandwich cylinder since the elastic general-instability collapse strength of Model DSRV-4M was considerably greater than that of Model DSRV-1.

**No greater advantage would be expected for hulls which fail between stiffeners; that is, in the inelastic shell buckle modes.

shell at midbay were only about one-half the calculated values. Although the measured strains indicated the presence of bending in the sandwich hulls, the maximum strains were located in the circumferential direction. Thus, it is unlikely that bending affected the strength of the sandwich hulls.

The estimated collapse depth versus ratio of cylindrical composite hull weight to displacement for steel, aluminum, titanium, and reinforced plastic composite hulls is presented in Figure 5. The relationship shown for composite aluminum hulls was obtained by drawing a straight line through the experimental points representing the results of Models DSRV-4L and the unpublished results of ring-stiffened Models DSRV-9A and DSRV-6A. Model DSRV-4L represented a semi-infinite hull, and Models DSRV-9A and DSRV-6A had finite deep frame, or bulkhead, spacings. The additional weight required for the deep frames in Models DSRV-9A and DSRV-6A was included when calculating their weight-to-displacement ratios. Thus, the curve for the composite aluminum hulls, as well as the other curves shown in Figure 5, represent strength-weight estimates for semi-infinite cylinders. The titanium curves are based on the test results of Model DSRV-3L and on the assumption that the efficiency of the titanium hulls is proportional to the efficiency of the aluminum hulls. The experimental points representing Model DSRV-3L as well as all other experimental points shown in Figure 5 have been obtained by linearly adjusting collapse depths to nominal yield strengths. Since the model yield strengths were higher than the nominal yield strength for each model, the experimental collapse depths have been adjusted conservatively. The composite steel and fiberglass curves were obtained in the same manner as the composite titanium curves. However, Models DSRV-10 and DSRV-16 have not been tested so their strength can only be estimated at this time. The collapse depth of Model DSRV-10, a ring-stiffened membrane cylinder with deep frames, was estimated to be the depth at which the average circumferential stress equals 1.05 times the weighted yield strength. The collapse of Model DSRV-16 a cylinder with closely spaced rectangular frames, was estimated to occur when the average circumferential stress in the fiberglass rings reaches 90,000 psi.

Curves similar to those presented in Figure 5 may be developed for many more feasible types of composite hulls. For example, the use of

HY-120 titanium rings encased in an HY-120 steel jacket would eliminate the need to weld any portion of a titanium hull. Another promising hull is composed of HY-200 or greater steel rings encased in HY-100 to HY-120 steel jackets. If proper design procedures are used, the strength-weight characteristics of these and other composite hulls should be similar to those of machined monolithic hulls of the same materials and may be estimated accordingly.

The use of composite construction offers several attractive advantages over that of conventional welded construction. The chief advantage is that it does not require welding of the basic hull material. A second advantage is that the jacket serves as a watertight envelope and protects the strength elements against corrosion. Since the strength elements, or inner rings, of composite hulls are machined, their strength is not affected by initial imperfections and residual stresses. In addition, machining of the rings permits the designer to use geometries and configurations which produce uniform stress patterns.[5,7] Thus, composite hulls are very efficient under hydrostatic loading and may, in some cases, offer an increase in structural efficiency over conventional welded construction.

The use of composite construction also has disadvantages. Whereas machining the strength elements may offer additional static strength, it will, in many cases, increase the cost of construction. Composite hulls are probably weaker than welded hulls of similar material under explosive loading, particularly on the surface or at shallow depths. Special machinery foundations are required if the ring elements are nonweldable. It may be more difficult to remove large machinery from a composite hull than from conventional welded hulls.

Since composite construction has both advantages and disadvantages, the merits must be evaluated for each application. For example, it will likely compare unfavorably with welded construction at shallow depths inasmuch as highly satisfactory performance may be obtained with relatively thin sections of conventional materials such as HY-80 steel. On the other hand, it may have the advantage at moderately deep depths for larger diameter hulls over conventional weldable materials since the plate thickness involved might make the cost of welding the entire hull prohibitive. However, the true advantage of composite construction is realized in a hull designed for extreme depths since it enables the use of high strength nonweldable materials as the basic hull material.

CONCLUSIONS

1. It is possible to accurately calculate the effect of compartment length on the experimental collapse pressure of these models using the analysis presented in Reference 1 for the plastic general-instability strength of stiffened cylindrical shells.

2. The present tests demonstrate the importance of representing the actual prototype compartment length when testing a model which may collapse in the plastic general-instability mode.

3. It now appears that a sandwich hull will normally have less than a 10-percent strength advantage over an optimum ring-stiffened hull of the same material and weight.

4. A high degree of structural efficiency was obtained in each model. Since residual stresses reduce the collapse strength of welded hulls, a properly designed composite hull may be slightly more efficient than a welded hull of similar material and geometry.

5. The strength-weight characteristics of composite semi-infinite cylindrical hulls of various combinations of materials may be estimated on the basis of these and earlier tests; see Figure 5.

6. Composite construction permits the use of existing nonweldable materials with high strength-to-weight ratios in the design of pressure hulls for deep-depth application.

Figure 1 – Models DSRV-1, DSRV-1L, DSRV-4M, and DSRV-4L

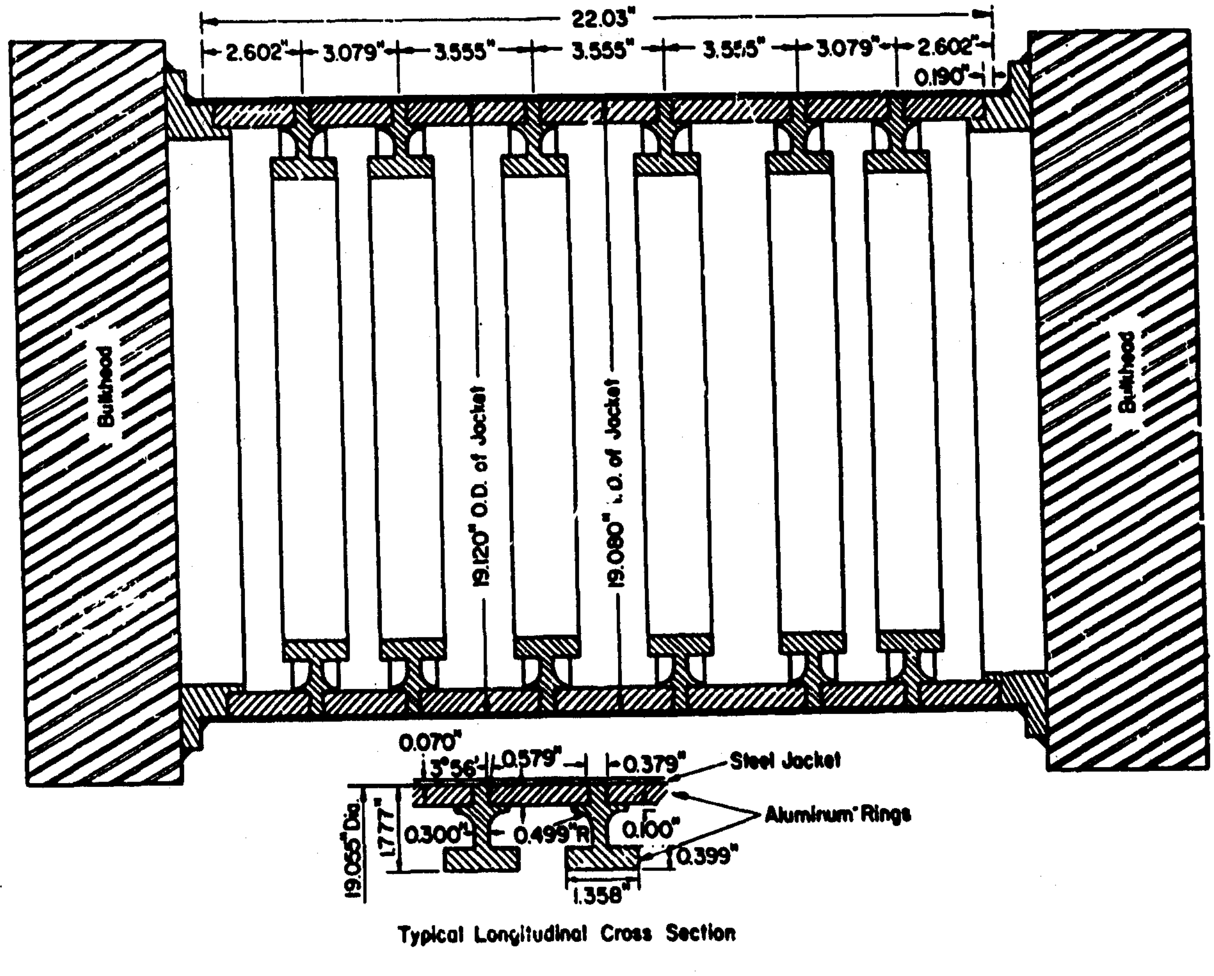
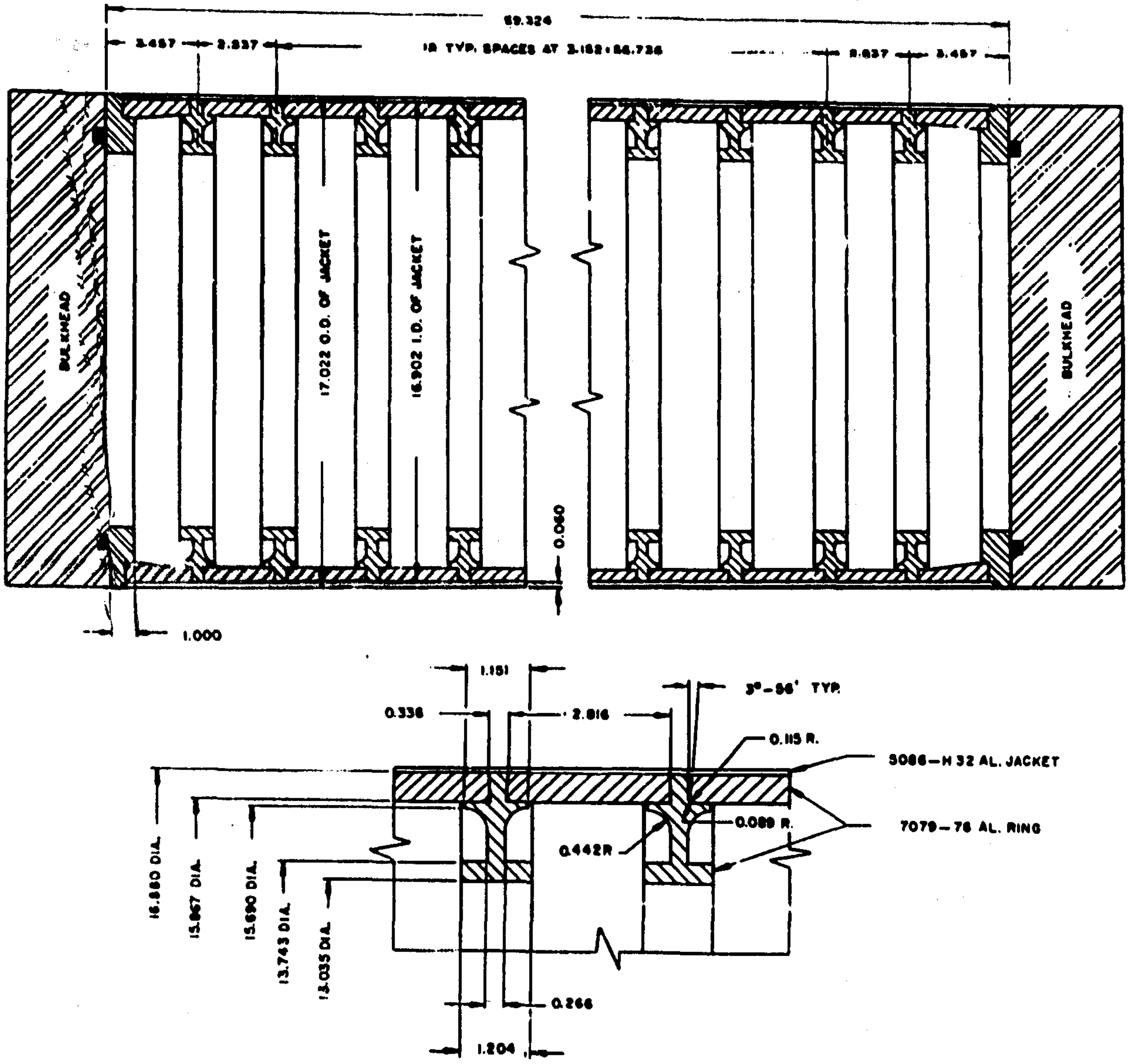
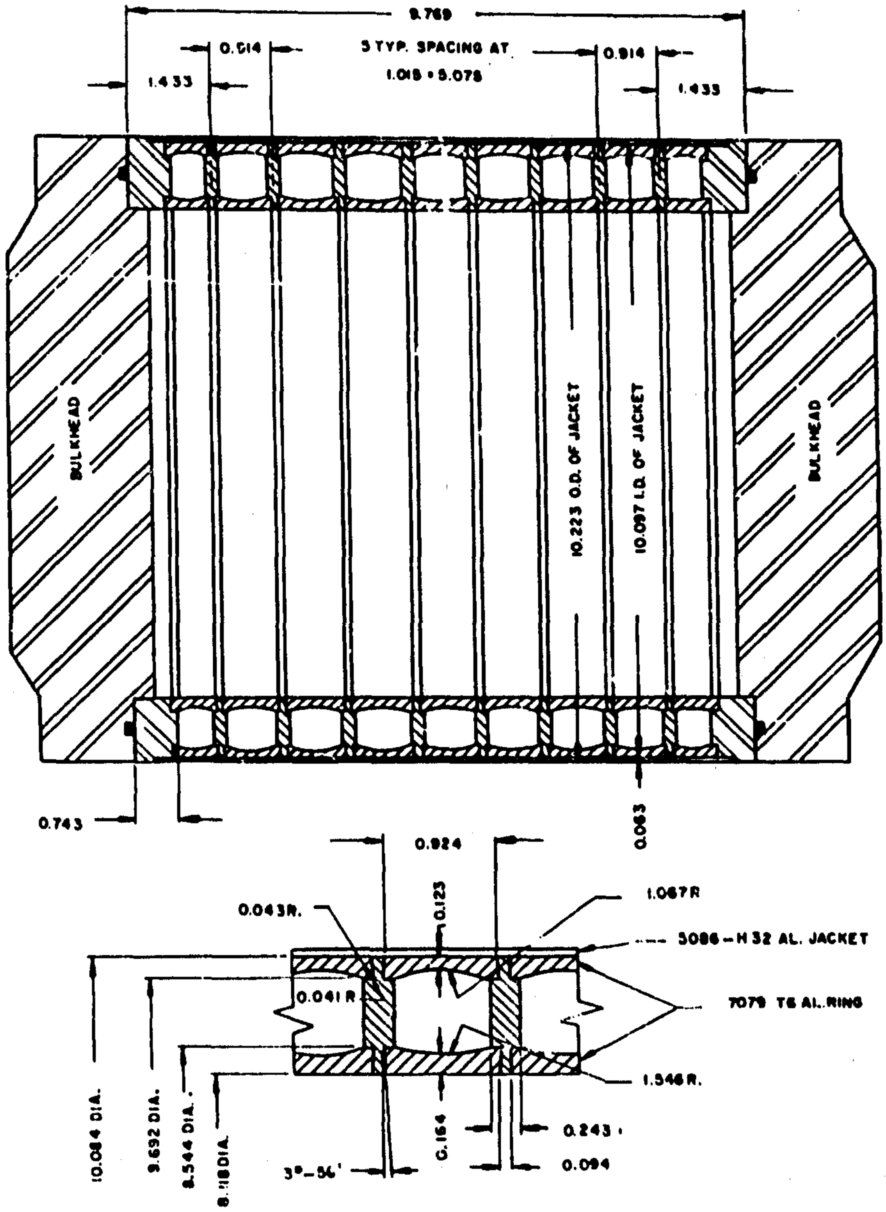


Figure 1a – Model DSRV-1



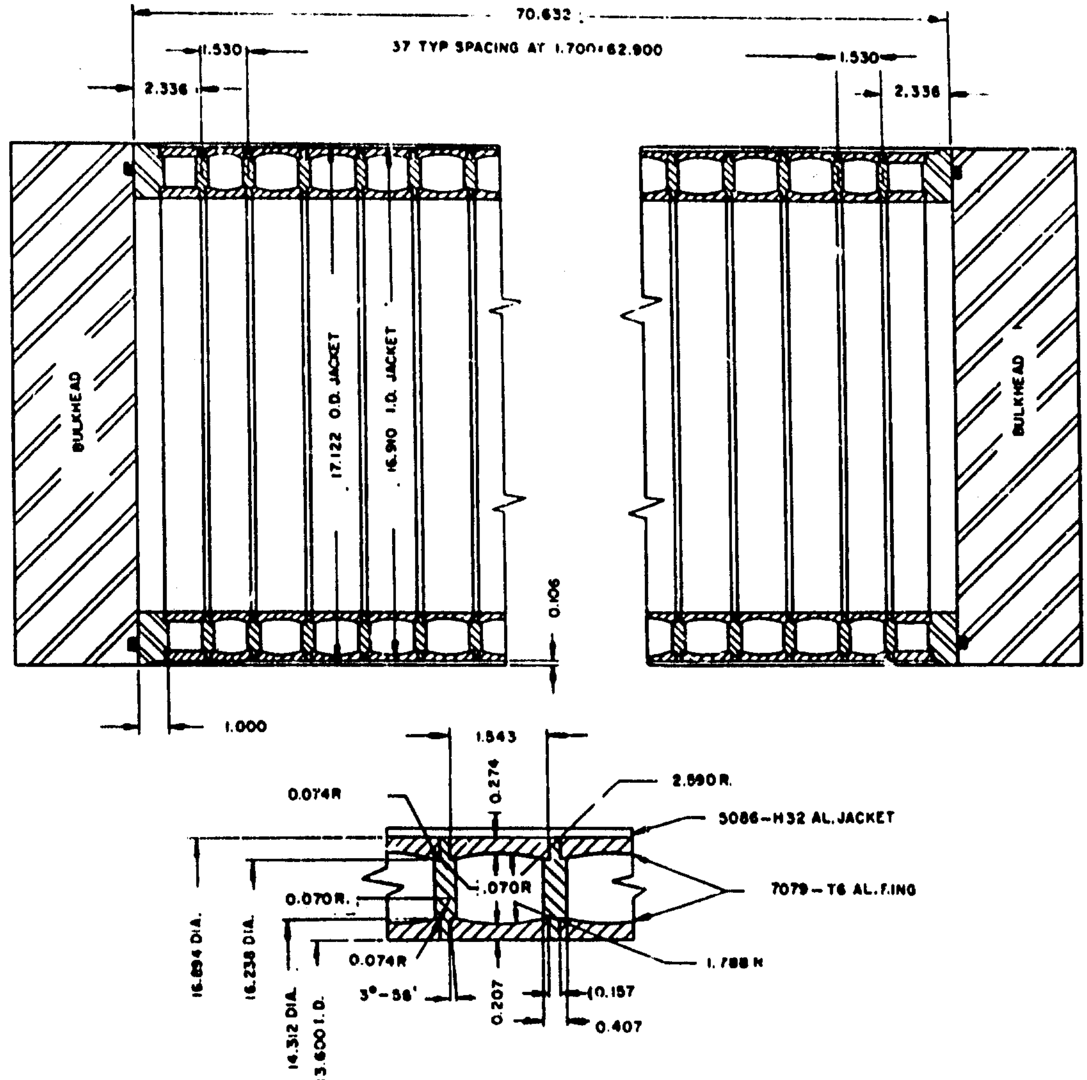
TYPICAL LONGITUDINAL CROSS SECTION

Figure 1b – Model DSRV-1L



TYPICAL LONGITUDINAL CROSS SECTION

Figure 1c — Model DSRV-4M



TYPICAL LONGITUDINAL CROSS SECTION

Figure 1d – Model DSRV-4L

Figure 2 -- Typical Stress-Strain Curves for 7079-T6 Aluminum Alloy Used in Models

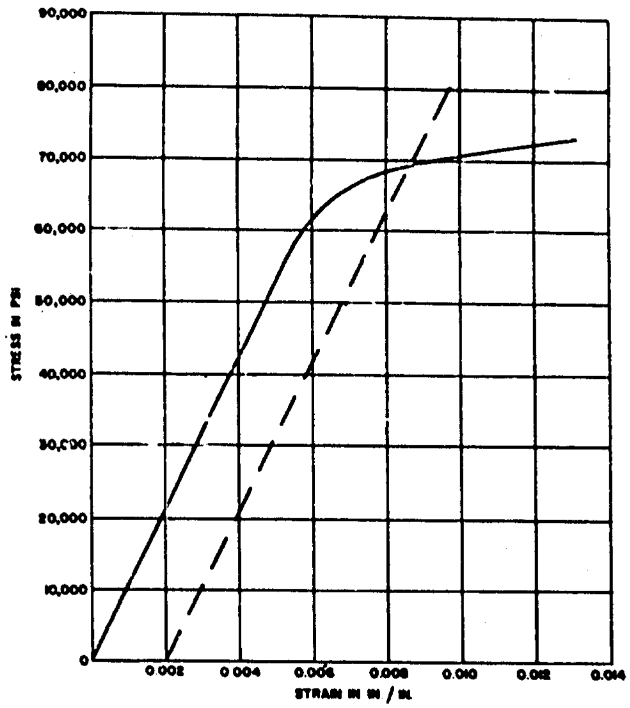


Figure 2a — Model DSRV-1

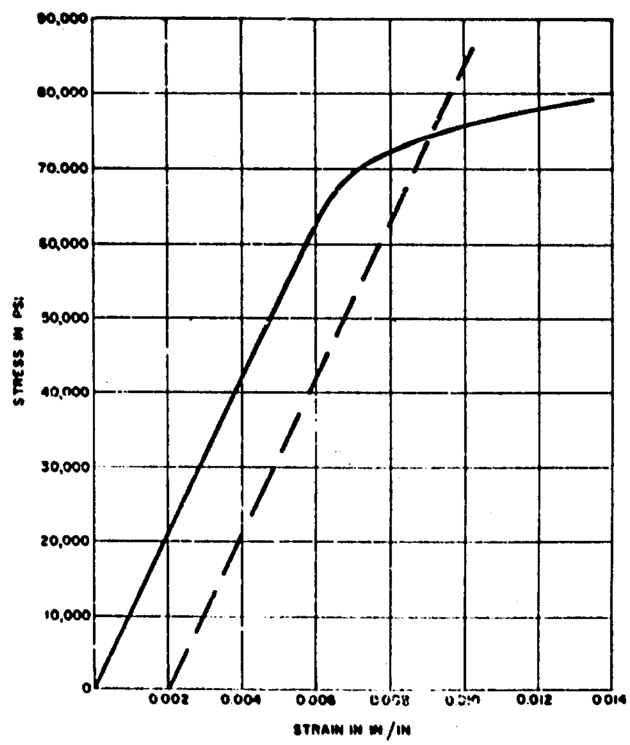


Figure 2b — Model DSRV-1L

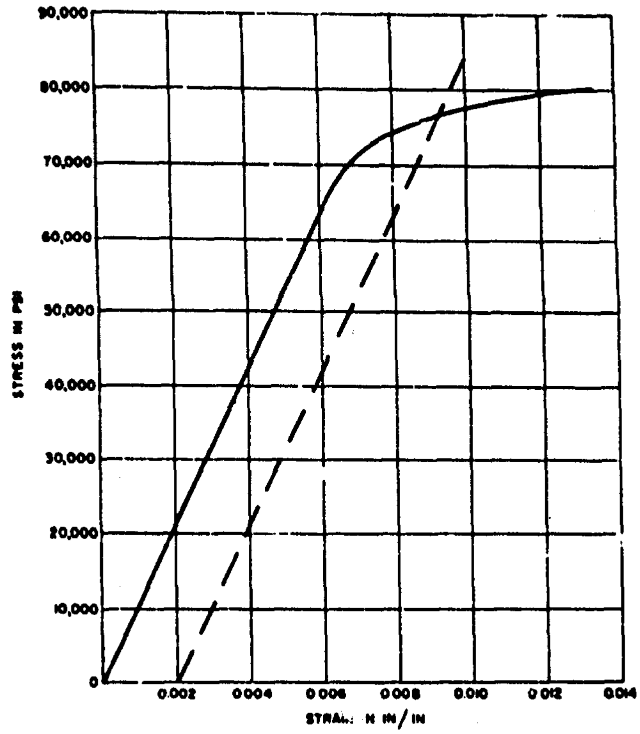


Figure 2c — Model DSRV-4M

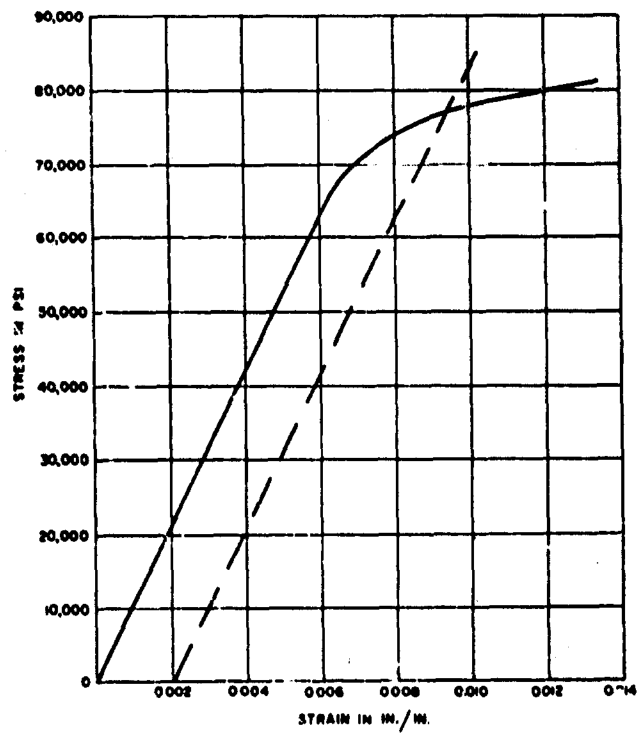


Figure 2d — Model DSRV-4L

Figure 3 – Location of Strain Gages

Numbers in the 100 series indicate exterior gages; in the 200 series, interior gages. Gages that measured circumferential strain are indicated by even numbers.

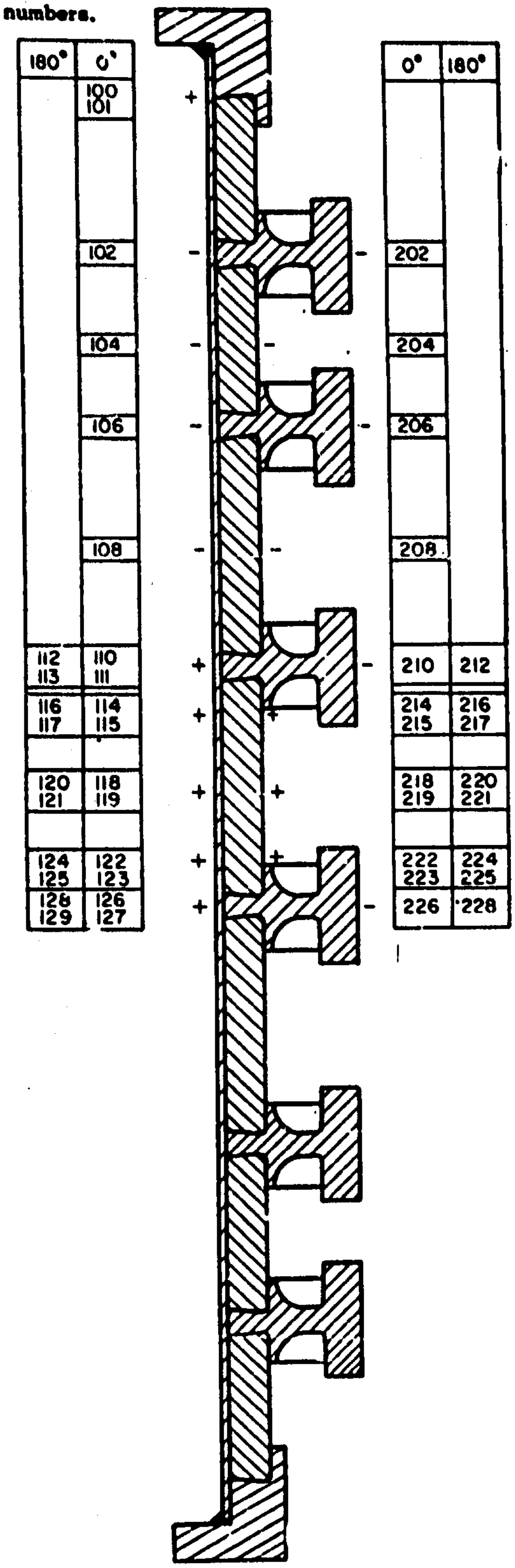


Figure 3a – Model DSRV-1

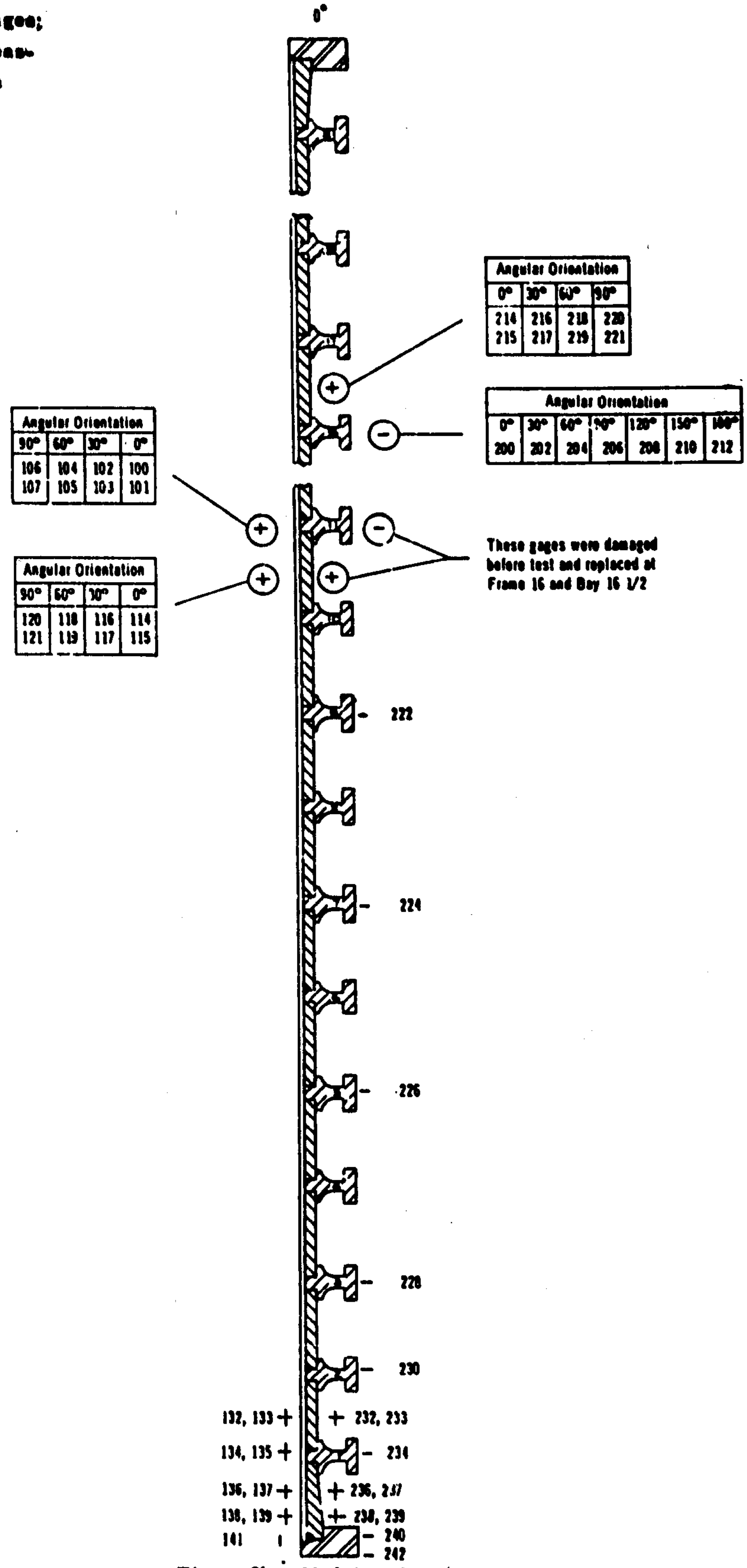
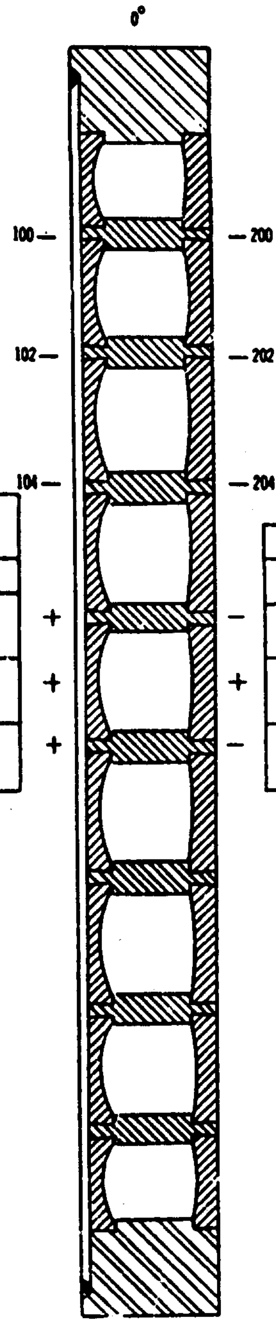


Figure 3b – Model DSRV-1L

ANGULAR ORIENTATION		
120°	60°	0°
118 119	112 113	106 107
124 125	122 123	120 121
130 131	128 129	126 127

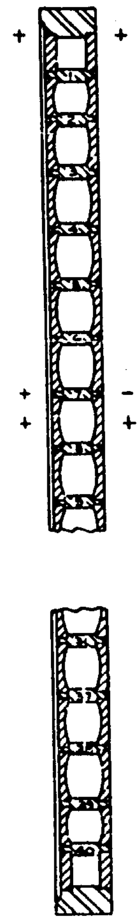


ANGULAR ORIENTATION						
0°	20°	40°	60°	80°	100°	120°
206	208	210	212	21[illegible]	216	218
220 221			222 223			224 225
226			228			230

Figure 3c — Model DSRV-4M

ANGULAR ORIENTATION	
90°	0°
102	100
103	101

ANGULAR ORIENTATION	
0°	90°
200	202
201	203



ANGULAR ORIENTATION			
90°	60°	30°	0°
110 111	108 109	106 107	104 105
124 125	122 123	120 121	118 119

ANGULAR ORIENTATION						
0°	30°	60°	90°	120°	150°	180°
204	206	208	210	212	214	216
218 219	220 221	222 223	224 225			

Figure 3d — Model DSRV-4L

Figure 4 – Typical Plots of Pressure versus Strain

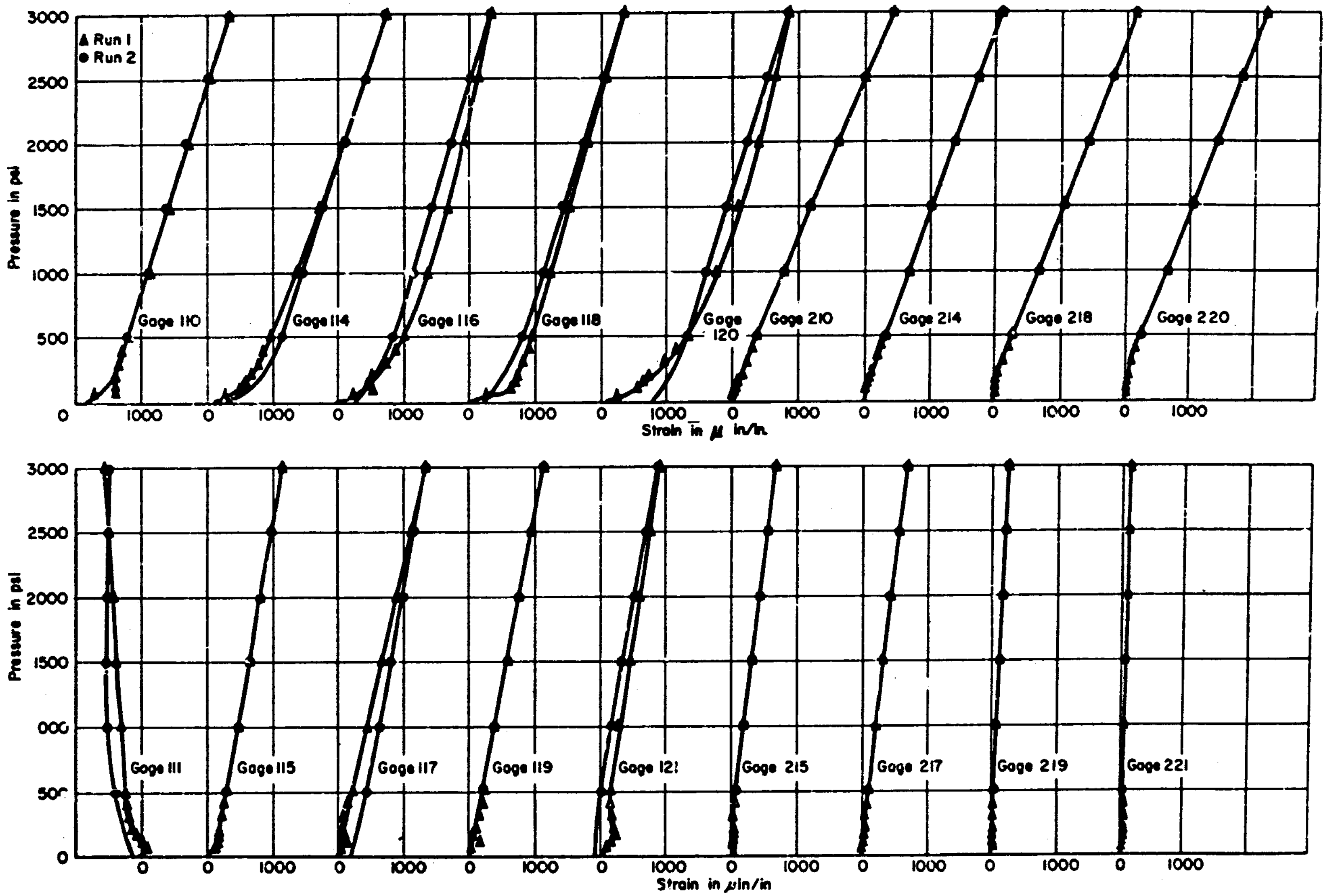


Figure 4a – Model DSRV-1

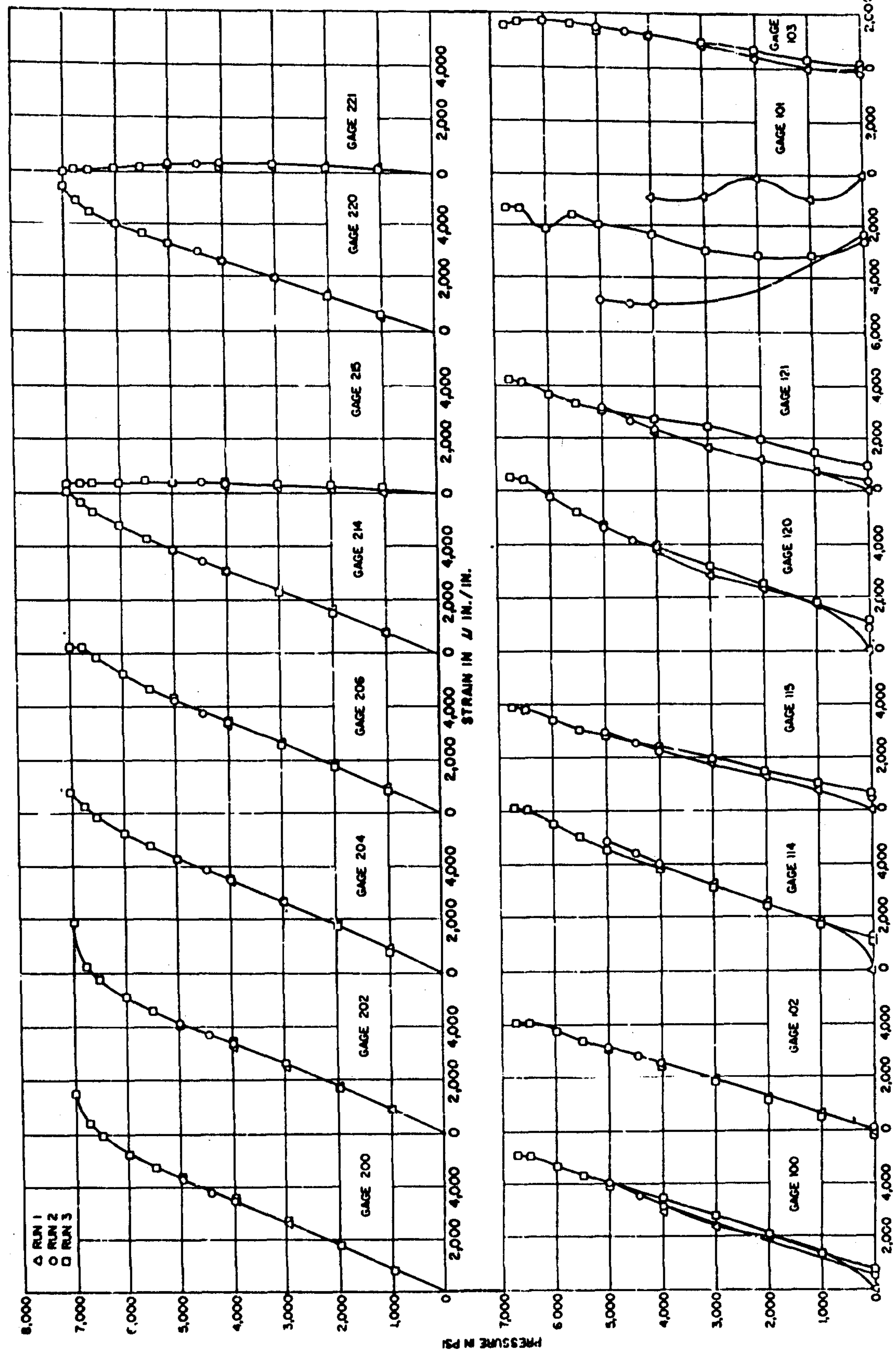


Figure 4b — Model DSRV-1L

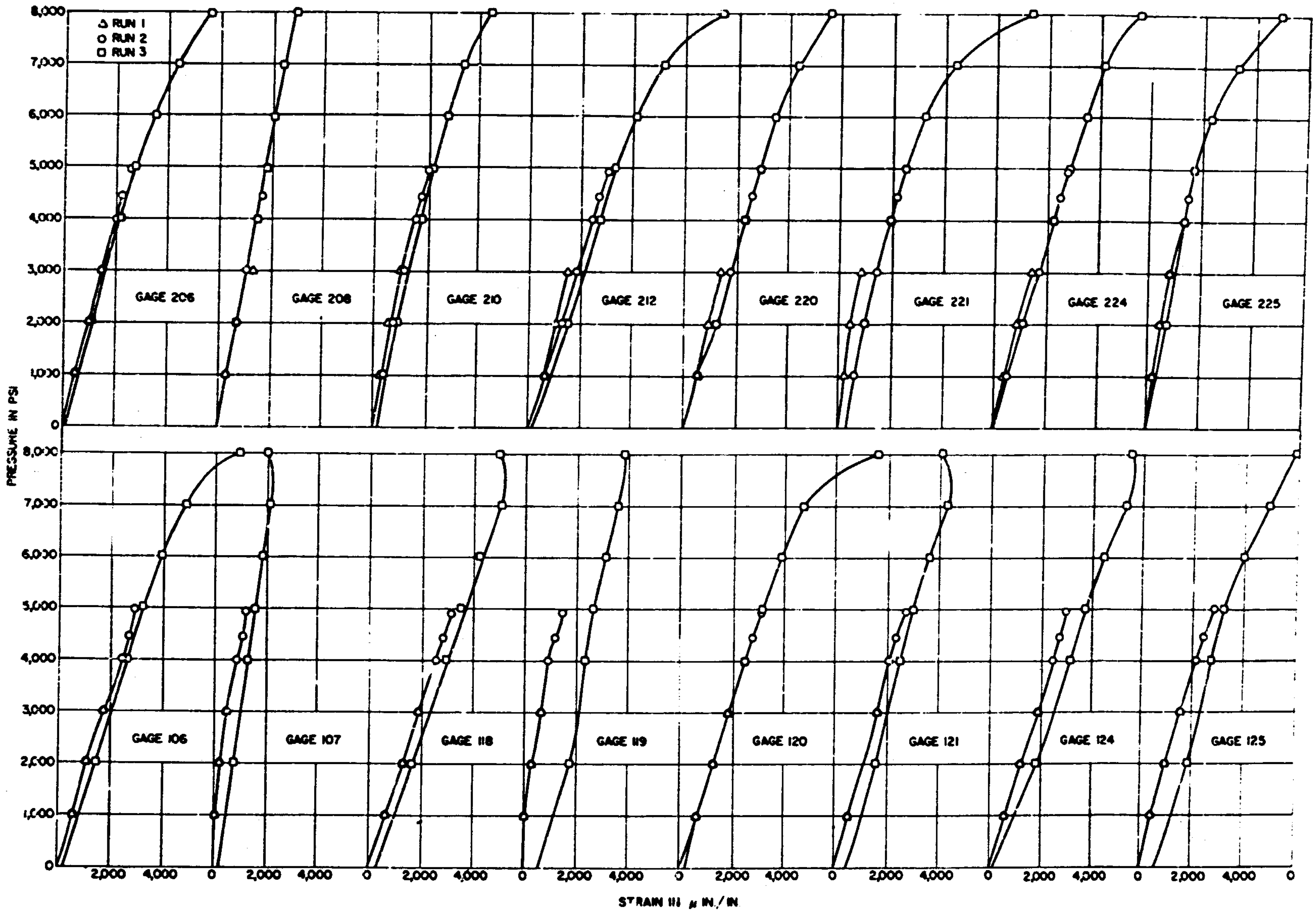


Figure 4c — Model DSRV-4M

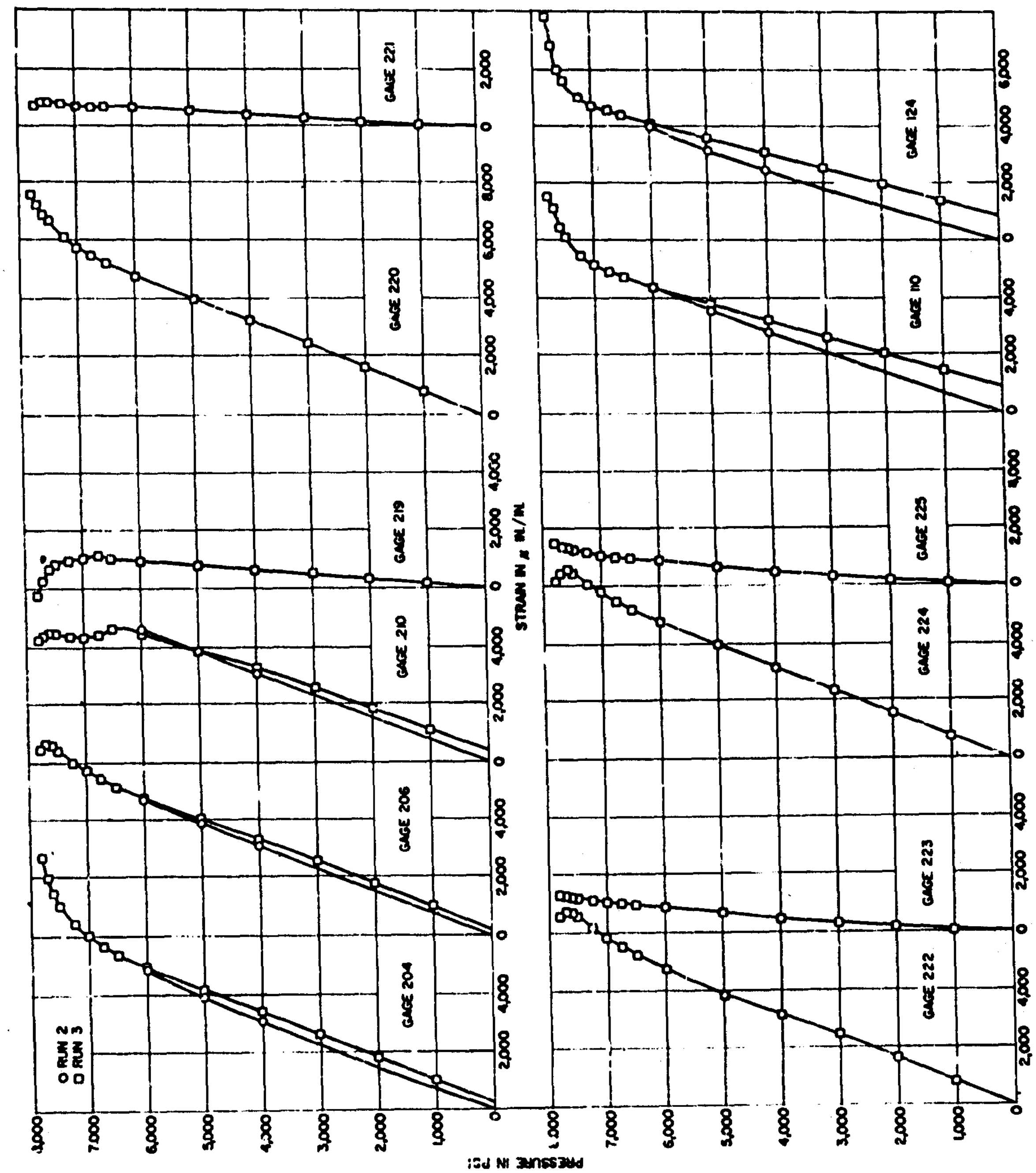


Figure 4d – Model DSRV-4L

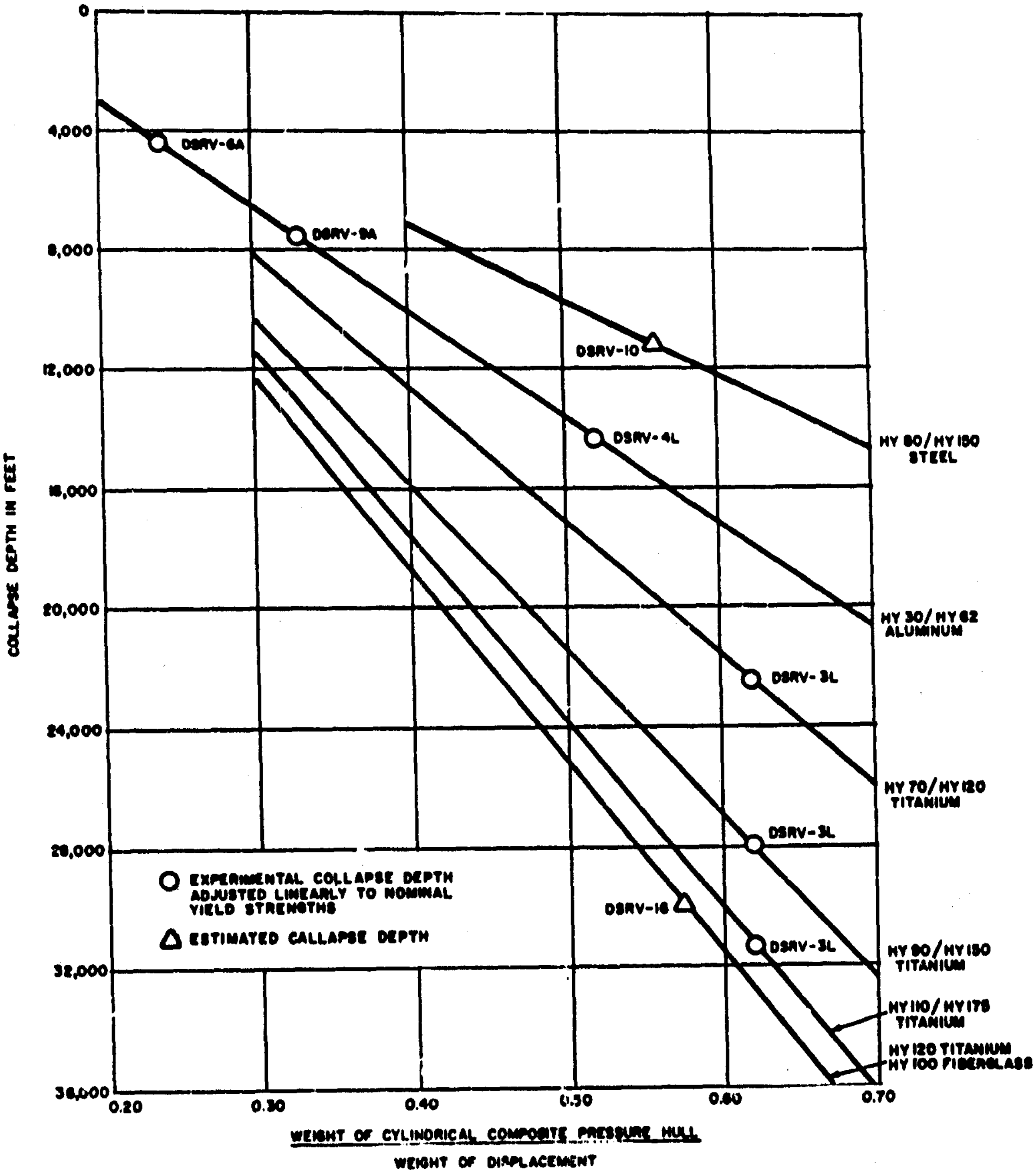


Figure 5 - Estimated Collapse Depth versus Ratio of Hull Weight to Displacement for Semi-Infinite Cylindrical Composite Hulls

TABLE 1

Dimensions and Material Properties of Models and Corresponding Prototypes

A 10-ft-diameter prototype hull was arbitrarily selected.

Property	Ring-Stiffened Cylinders			Sandwich-Type Cylinders		
	Model DSRV-1	Model DSRV-1L	Prototype	Model DSRV-4M	Model DSRV-4L	Prototype
Outside Diameter, in.	19.175	17.000	120	10.21	17.136	120
Jacket Thickness, in.	0.060	0.060	1/8	0.063	0.106	1/4
Average Outer Shell Ring Thickness, in.	0.579	0.507	1 5/8	0.149	0.248	1 1/4
Average Inner Shell Ring Thickness, in.	---	---	---	0.181	0.301	2 1/8
Typical Web Spacing, in.	1.555	1.152	22 1/4	1.015	1.700	12
Web Thickness, in.	0.300	0.266	1 7/8	0.243	0.407	2 7/8
Flange Width, in.	1.358	1.204	8 1/2	---	---	---
Flange Thickness, in.	0.399	0.354	2 1/2	---	---	---
Minimum inside Diameter, in.	14.703	13.035	92	8.118	13.600	96
Weight of Pressure Hull / Weight of Displacement	0.56	0.56	0.56	0.52	0.52	0.52
Compressive Yield Strength of Jacket, psi	97,000	87,500	100,000	30,000	30,000	30,000
Compressive Yield Strength of 7079-T6 Aluminum Rings, psi	70,000	74,500	62,000	76,490	77,000	62,000
Young's Modulus of 7079-T6 Aluminum Rings, psi × 10^6	---	---	10.5	---	---	10.5

TABLE 2

Ratio of Theoretical Plastic General-Instability Collapse Pressure to Experimental Collapse Pressure

Model	Bulkhead Spacing, L_b / Diameter, D	Experimental Collapse Pressure (p_{exp}) psi	Theoretical Elastic General-Instability Collapse Pressure (p_{cr}) psi	Theoretical Plastic General-Instability Collapse Pressure (p_{st}) psi	$\frac{p_{st}}{p_{exp}}$
DSRV-3*	1	14,250	74,000	13,800	0.97
DSRV-3L*	3	15,750	30,000	15,300	0.97
DSRV-1	1	8000	35,000	7850	0.98
DSRV-1L	4	7350	17,000	7700	1.05
DSRV-4M	1	8450	55,000	8600	1.02
DSRV-4L	4	7800	19,000	7900	1.01

*Data from Reference 1.

TABLE 3

Structural Efficiency Factors

Model	Experimental Collapse Pressure (p_{exp}) psi	Theoretical Plastic General-Instability Collapse Pressure (p_{st}) psi	Yield Pressure of Equivalent Cylinder (p_{eq}) psi	Structural Efficiency Factor: Experimental p_{exp}/p_{eq}	Structural Efficiency Factor: Theoretical p_{st}/p_{eq}
DSRV-1	8000	7850	8435	0.95	0.93
DSRV-1L	7350	7700	8850	0.83	0.87
DSRV-4M	8450	8600	8750	0.97	0.98
DSRV-4L	7800	7900	8820	0.88	0.90

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