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BUCKLING TESTS OF TWO INTEGRALLY STIFFENED CYLINDERS SUBJECTED TO BENDING

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16. Abstract

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SUMMARY

Results of buckling tests are reported for two cylinders subjected to bending loads. The cylinders were 1.96 m (77 in.) in diameter and integrally stiffened by rings and stringers of rectangular cross section. Both cylinders buckled in a general-instability mode with considerable margin against local-instability modes which involve buckling of either the skin or the stiffening elements. Buckling occurred at loads approaching the loads predicted by the use of a small-deflection buckling theory which included the effect of stiffener eccentricity; but the quality of the cylinders was not high and the significance of the correlation obtained is clouded by the fact that actual cylinder dimensions varied considerably from point to point on the cylinders.

INTRODUCTION

Designers of shell structures subject to failure by buckling must rely on test data for ascertaining correlation or "knockdown" factors for use with buckling theories of shell structures. Reference 1 summarized the data available to the designer for stiffened cylinders in bending and/or axial compression. The study (ref. 1) indicated that the pertinent data available to the designer were quite limited and that no data were available on ring-and-stringer-stiffened cylinders for which buckling by general instability preceded local buckling of the skin. The present investigation was undertaken to provide one set of such data.

Bending tests of two integrally stiffened cylinders are reported and discussed. Both cylinders buckled in a general-instability mode with considerable margin against local-instability modes which entail buckling of either the skin or the stiffening elements. All visible evidence of having been tested disappeared when load was removed from the cylinders after the initial buckling tests; hence, a second test of each cylinder was conducted after the cylinder had been rotated 180° in its test fixture so that the cylinder generator subjected to maximum tensile stress in the first test was the generator of maximum compressive stress in the second test.

SYMBOLS

The units for physical quantities in the present paper are given in the International System of Units (SI) with equivalent values given parenthetically in U.S. Customary Units. Measurements and calculations were made in U.S. Customary Units. Factors relating the two systems are given in reference 2; factors used in the present paper are given in the appendix.

b	distance between stringers
d	distance between rings
h	thickness of cylinder wall
l	test length of cylinder
M	bending moment on cylinder
M_{cr}	theoretical bending moment on cylinder at buckling
M_{ult}	bending moment on cylinder at buckling
P	compressive load on test coupon
R	radius of cylinder to middle surface of skin
r	radius of fillet
t	skin thickness
t_r	thickness of ring
t_s	thickness of stringer
ϵ	strain in cylinder wall
θ	angle defining distance along circumference of cylinder from generator of maximum compressive stress

TEST SPECIMENS

The test specimens consisted of two circular cylinders integrally stiffened on the inside surfaces of the cylinders by rings and stringers of rectangular cross section. The principal difference between the cylinders was in ring spacing with the cylinder having the greater ring spacing also having the heavier rings. General features of the test cylinders are shown in figures 1 and 2; other construction details are given in figure 3 and in table I.

The results of many micrometer measurements to determine detailed wall geometry of the test cylinders are given in table I. The values given for skin and stiffener thicknesses denote average thicknesses and the percent of deviation from these thicknesses. The symbols used in table I are defined in figure 4, except for cylinder test length l. Cylinder length l was taken as the distance over which wall material had been removed to form the rings and stringers. The wall on each end of this test length is fastened to heavy end fixtures. (See fig. 1.)

The variations in skin and stiffener thicknesses given in table I are seen to be considerable. The values given were derived from measurements taken from the central portion of the cylinders where rather complete measurements were taken. However, these values do not differ systematically nor significantly from numerous check measurements taken elsewhere on the cylinders. The fabrication processes, which are described subsequently, resulted in variations in critical dimensions of ±0.15 mm (±0.006 in.) or larger. Structures with much smaller variations are desired for buckling tests. Large variations cause problems in interpreting cylinder behavior from test data, and these problems are discussed in a subsequent section of this paper.

Figure 5 gives results of measurements made to determine geometric deviations of the test cylinders from the perfect cylindrical shape. The deformations were measured from an imaginary surface formed by the approximate circles established by the heavy machined fixture at each end of the test cylinders and the straight-line generators connecting the two circles. Deviations from perfect-cylinder geometry as great as 2.5 mm (0.10 in.) were measured. The deviations were determined with the use of a scanner, especially built for such purposes, which autographically plotted deviation as a function of axial location on the cylinder as the scanner was moved along a generator of the cylinder. Input to the two axes of the plotter was achieved with the use of a linear variable differential transformer-type displacement transducer and a 10-turn potentiometer. The imperfection measurements depicted in figure 5 were taken in the area of maximum compressive stress for each of the two tests of each cylinder. Individual measurements were taken at 7.6-cm (3-in.) intervals ($\theta = 4.44^{\circ}$) in both directions around the circumferences of the cylinders with the initial measurement at the generator of maximum compressive stress. Final measurements were taken at 29.2 cm $\left(11\frac{1}{2}\text{ in.}\right)$ from the generator of maximum compressive stress on lines adjacent to the weld lines.

Each cylinder was fabricated from 10 equal-width panels of 2219-T81 aluminum alloy; the panels run the entire length of the cylinder. The panels were milled from 1.27-cm $\left(\frac{1}{2}\text{-in.}\right)$ plate in a flat condition. The first operation consisted of removing approximately 1.5 mm (0.06 in.) of material from each surface of the plate. Then the pockets between stringers and rings were formed by mechanical milling to a depth, width, and length which left approximately 0.38 mm (0.015 in.) of material to be removed later by chemical milling. The panels were then rolled into cylindrical panels. The pockets formed by milling were filled with a low-melting-temperature alloy for the rolling operation in order to prevent ring buckling and the formation of flat surfaces between stringers. After the panels had been rolled to the desired cylindrical shape, the filler alloy was removed by melting and the 0.38 mm (0.015 in.) of excess material was removed by chemical milling. The panels were fabricated initially from 2219-T31 aluminum alloy and were aged to the T81 condition in the time interval between the forming and the chemical milling operations.

The panels were welded together along generators of the test cylinders with tungsten inert gas fusion welding. A cross section of a weld is shown in figure 3; this cross-sectional shape was used to provide a relatively large moment of inertia and light weldments in an attempt to prevent shrinkage and hence geometric imperfections resulting from the welding. The attempt was not entirely successful, however, as can be seen from figure 5 which indicates a general bowing of the cylinder wall at six of the eight weld lines shown. Imperfections at generators away from the weld lines were generally more randomly shaped.

TEST PROCEDURE

The test cylinders were loaded in bending through a loading frame with the use of a hydraulic testing machine. Two tests were made of each cylinder. For the second test the cylinder was rotated 180° in its fixture after the first test so that the cylinder generator subjected to maximum tensile stress in the first test was the generator of maximum compressive stress in the second test. During the first test, considerable strain-gage data were taken from the region of the cylinder subjected to maximum compressive stress, but only a limited amount of data was taken during the second test. The test setup shown in figure 6 is the same as that shown in figure 4 of reference 3. The present tests were conducted in the same loading frame and fixtures. The presence of stray loads in the test cylinders was minimized as far as practicable by employing rollers between moving surfaces and by counterbalancing fixtures near their centers of gravity. Rollers were used between the loading frame and the floor supports as well as between the loading frame and the testing machine. Use of the rollers allows the cylinders to shorten during application of load and helps to restrict the loads at the roller locations to normal loads. The rollers were case hardened, as were the surfaces against which they reacted.

Resistance-type wire strain gages were mounted on the stringers, rings, and skins of the test cylinders prior to testing and strains from the gages were recorded during each test with the use of the Langley central digital data recording facility. Data from the gages were used to determine the strain distribution in each cylinder and to help assess the reaction of the cylinder to applied loads.

After both tests were completed, the cylinders were cut into convenient pieces for measuring wall geometry. In addition, coupons 11.4 cm $\left(4\frac{1}{2}\text{ in.}\right)$ wide and 16.5 cm $\left(6\frac{1}{2}\text{ in.}\right)$ long were cut from the test cylinders near their neutral axes. The coupons were instrumented with strain gages and tested in axial compression in order to understand better the stiffness properties of the walls.

TEST RESULTS AND DISCUSSION

Results of measurements made to determine the circumferential distribution of longitudinal strain in the cylinder walls in the region of maximum compressive stress are given in figure 7; data from two longitudinal stations are given for the first test of each of the cylinders. The strains given are the result of correcting the measured strains, with the assumption of a linear strain distribution, to the centroid of the cross section of the wall. The correction was made in an attempt to obtain membrane strains which could be compared with calculated strains.

The calculated values in figure 7 are based on the assumption of a linear strain distribution across the depth of the cylinder and on average measured values of wall geometry (table I). The "Poisson effect" of reinforcing rings and the extra wall material at weld lines were taken into account in the calculations. A comparison of measured and calculated values in figure 7 indicates that calculated strain is somewhat greater than measured strain. This result infers that the calculated bending stiffness of the test cylinders is too small. At a bending moment equal to $\frac{1}{2}M_{ult}$, the calculated stiffness is approximately 6 percent low for cylinder 1 and 4 percent low for cylinder 2. While these percentages are considerably smaller than the variations in wall geometry discussed previously, they are presumably a direct result of those anomalous variations. This reduction comes about because cylinder bending stiffness is dependent more on "average" wall geometry than on local geometry from point-to-point on the cylinder.

The strain distribution shown in figure 7 for cylinder 1 at $\frac{1}{2}M_{ult}$ indicates the presence of "soft spots" which may be associated with buckle-like deformations of the cylinder wall. This behavior is accentuated at higher loads in both cylinders and is more pronounced in the present tests than in previous similar tests (refs. 3 to 5) of cylinders of sandwich and corrugated-wall construction. The soft spots do not appear unduly to affect buckling load of the cylinders, as is discussed subsequently.

A more direct measure of the local behavior of the cylinder wall under compressive load is given in figure 8 where load-strain curves for coupons tested in uniform axial compression are compared with calculated load-strain curves. The coupons were cut from the walls of the tested cylinders near the neutral axes of the cylinders, a region which had remained unbuckled and essentially unstressed during the buckling tests of the cylinders. The coupons were loaded to strains considerably in excess of those experienced at the extreme compression fiber of the test cylinders during the cylinder buckling tests. Axial strains measured in the coupon tests are in reasonable agreement with those calculated from the geometry of the coupons, an indication that axial stiffness can be calculated with reasonable accuracy and that local buckling of the skin and stringers had little, if any, influence on the buckling behavior of the test cylinders.

Results of strain measurements taken along the generator of maximum compressive stress of the test cylinders are given in figure 9. Longitudinal strains in the stringers and skin as well as circumferential strain in the rings and skin are shown. The data generally indicate an increase in strain with a corresponding increase in load, with strains measured on one side of the cylinder wall increasing faster than the strains on the other side. This behavior indicates the growth of buckle-like deformations with load, generally a growth of the buckle-like shape indicated in figure 5 for generators of maximum compressive stress. Near ultimate load, growth of the initial buckle-like pattern is generally accelerated except at station 6 of both cylinders and possibly at station 5 of cylinder 1 where a departure from the initial pattern appears to be underway.

A comparison between test and calculated bending moments at buckling is given in table II. The calculated moments at buckling were obtained with the use of reference 6 from which buckling of cylinders in bending is calculated for a cylinder in a membrane state of stress that neglects prebuckling deformations and the discreteness of stiffening members. The agreement between test and calculated moments at buckling is good if the initial deviations in wall geometry from that of a perfect cylinder (fig. 5) are considered. In the worst case the cylinder buckled at 92 percent of the predicted buckling moment. Imperfections as large as those depicted in figure 5, and the subsequent growth of imperfections as mentioned in the discussions of figures 7 and 9, might be expected to produce substantially greater discrepancies between buckling loads obtained by calculations and tests. Evidently the types of deviation present were not very detrimental.

Buckling calculations were also made with the use of reference 7, which takes into account the effects of prebuckling deformations and of the discreteness of ring stiffeners for cylinders in uniform axial compression. These calculations were compared with calculations made by using reference 6 for cylinders in axial compression; the two calculations were found to give essentially the same buckling load. It can be inferred from these calculations that the effects of prebuckling deformations and of discreteness of stiffening rings would probably also be small for cylinders such as the test cylinders which were loaded in bending.

As a separate item of interest, the foregoing calculations indicated that the load intensity carried by the cylinders in bending was 12 percent greater than that carried by the cylinders in uniform axial compression. Hence the assumption, which is sometimes made, that the load intensity for buckling of cylinders in bending is the same as that for cylinders in uniform axial compression is not valid for stiffened cylinders of the proportions of the test cylinders.

Photographs of buckled cylinders are shown as figures 10 to 13. The buckle pattern is generally compatible with the calculated buckle pattern. Calculations predict buckling into four half waves in the axial direction with nearly square buckles.

The first buckling tests of the cylinders do not appear to have influenced appreciably the results of the second tests which were made after the cylinders were rotated 180° in their test fixtures. The buckling moment in the second test is lower for both cylinders than the buckling moment in the first test, but not appreciably lower, and the difference is probably associated with geometric variations rather than damage to the cylinders from the initial tests.

CONCLUDING REMARKS

Results of buckling tests are reported for two cylinders subjected to bending moments. The cylinders were 1.96 m (77 in.) in diameter and stiffened by rings and stringers of rectangular cross section. The cylinders buckled at loads approaching those calculated for cylinders in a membrane state of stress. However, the quality of the cylinders was not high and the significance of the result obtained is clouded somewhat by the fact that actual cylinder dimensions varied considerably from point to point on the cylinders.

Langley Research Center,
National Aeronautics and Space Administration,
Hampton, Va., April 29, 1971.

APPENDIX

CONVERSION OF U.S. CUSTOMARY UNITS TO SI UNITS

Conversion factors (ref. 2) for the units used in the present report are given in the following table:

Physical quantity	U.S. Customary Unit	Conversion factor (a)	SI Unit
Length	in.	0.0254	meters (m)
Force	lbf	4.448	newtons (N)
Moment	kip-in.	113.0	newton-meters (N-m)

[a]Multiply value given in U.S. Customary Units by conversion factor to obtain equivalent value in SI Units.

Prefixes to indicate multiples of units are as follows:

Prefix	Multiple
mega (M)	10^6
kilo (k)	10^3
centi (c)	10^{-2}
milli (m)	10^{-3}

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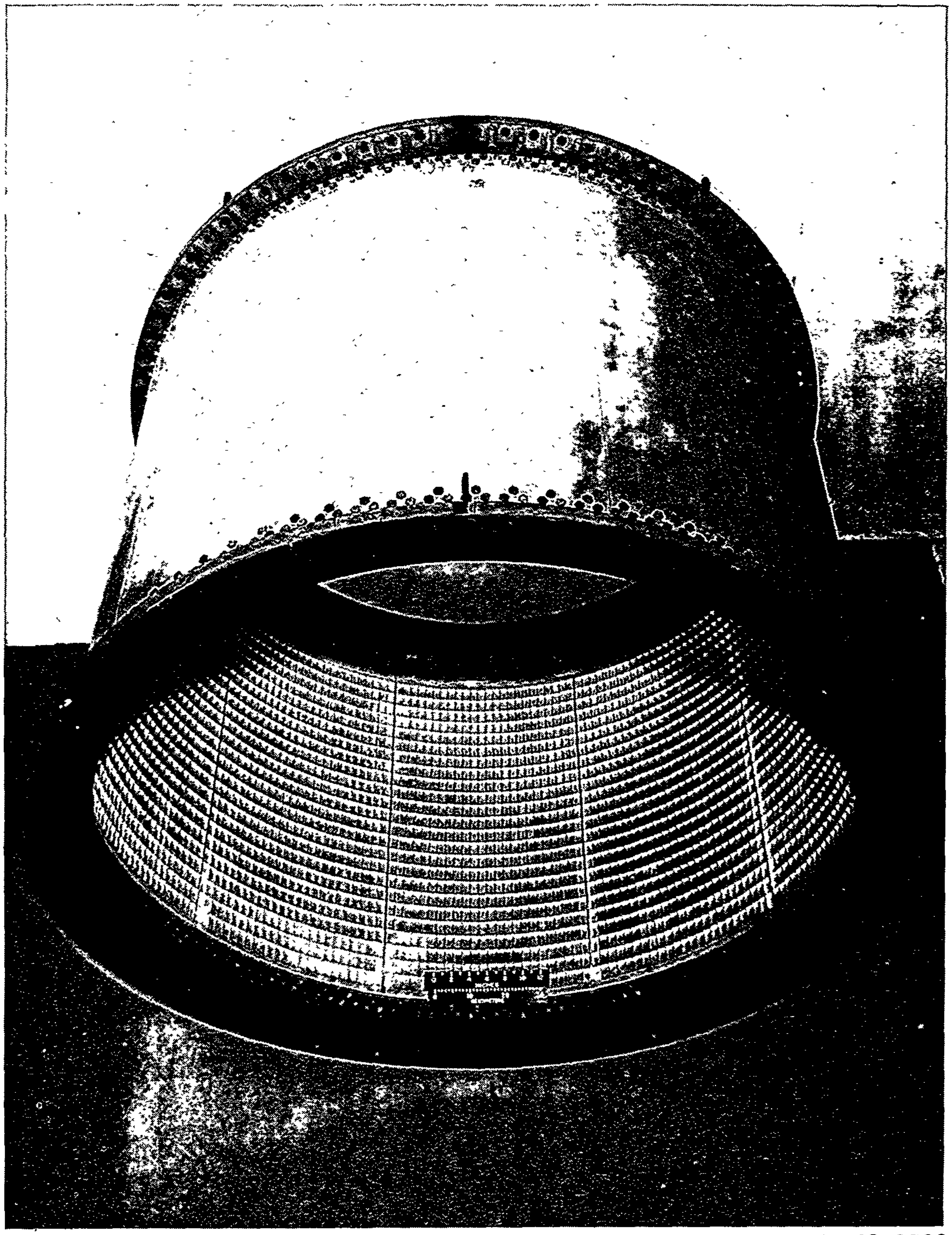
TABLE I.- DIMENSIONS OF TEST CYLINDERS

Dimension	Cylinder 1			Cylinder 2		
R	0.983 m	(38.71 in.)		0.983 m	(38.71 in.)	
l	1.22 m	(48.0 in.)		1.22 m	(48.0 in.)	
b	2.54 cm	(1.00 in.)		2.54 cm	(1.00 in.)	
d	5.08 cm	(2.00 in.)		15.2 cm	(6.00 in.)	
h	9.42 mm	(0.3707 in.)	±2%	8.99 mm	(0.3538 in.)	±2%
r	3.05 mm	(0.120 in.)		3.05 mm	(0.120 in.)	
t	1.16 mm	(0.0458 in.)	±10%	0.688 mm	(0.0271 in.)	±20%
t_s	1.41 mm	(0.0555 in.)	±15%	1.14 mm	(0.0447 in.)	±15%
t_r	1.38 mm	(0.0542 in.)	±15%	2.98 mm	(0.1173 in.)	±5%

TABLE II.- RESULTS OF BUCKLING TESTS

Cylinder	Test (a)	M_{ult}		M_{cr}		$\frac{M_{ult}}{M_{cr}}$
		MN-m	kip-in.	MN-m	kip-in.	
1	1	0.869	7690	0.921	8150	0.94
1	2	.868	7680	.921	8150	.94
2	1	.590	5220	.599	5300	.98
2	2	.554	4900	.599	5300	.92

[a]Test 2 was conducted after cylinder had been tested and then rotated 180° in test fixture so that tension side of cylinder in test 1 was compression side in test 2.



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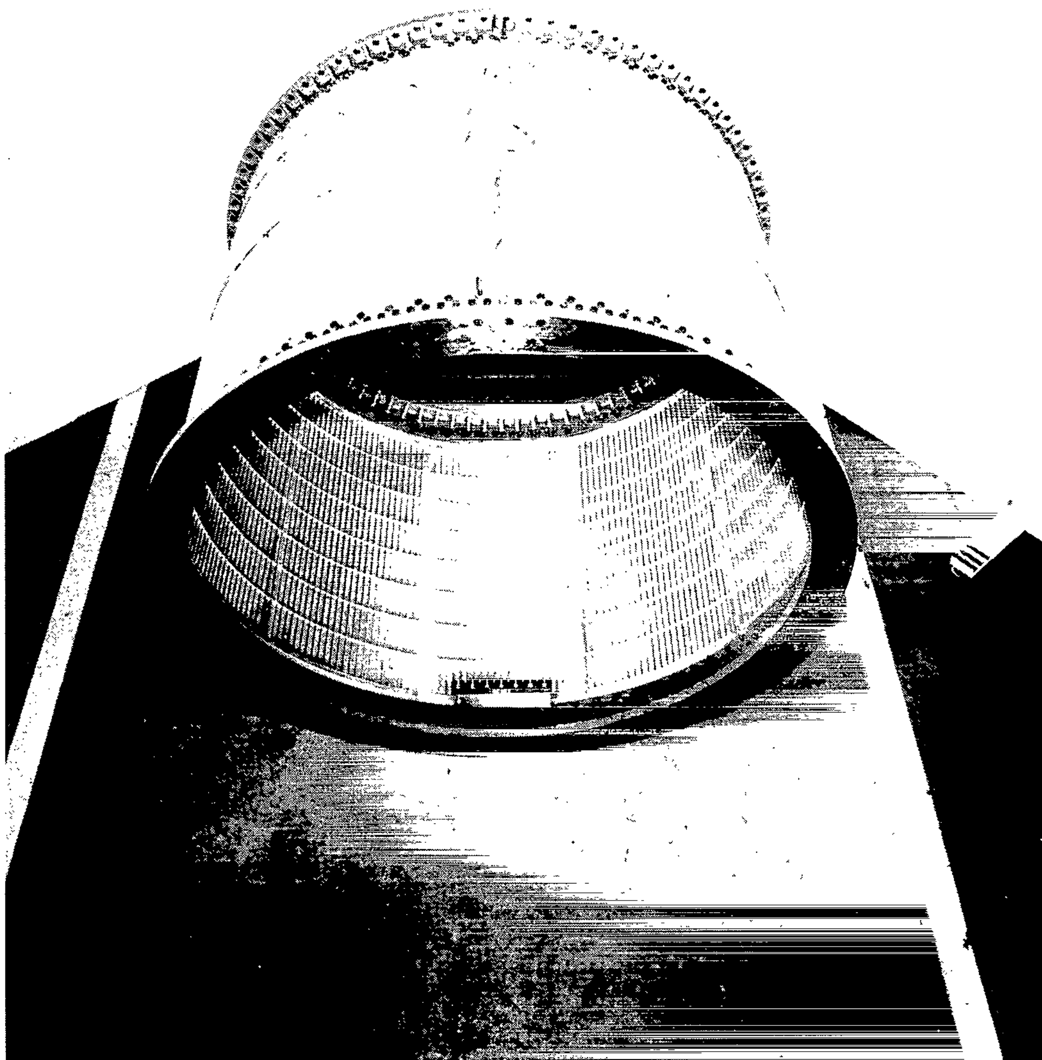
(a) Cylinder 1.

Figure 1.- General view of test cylinders.



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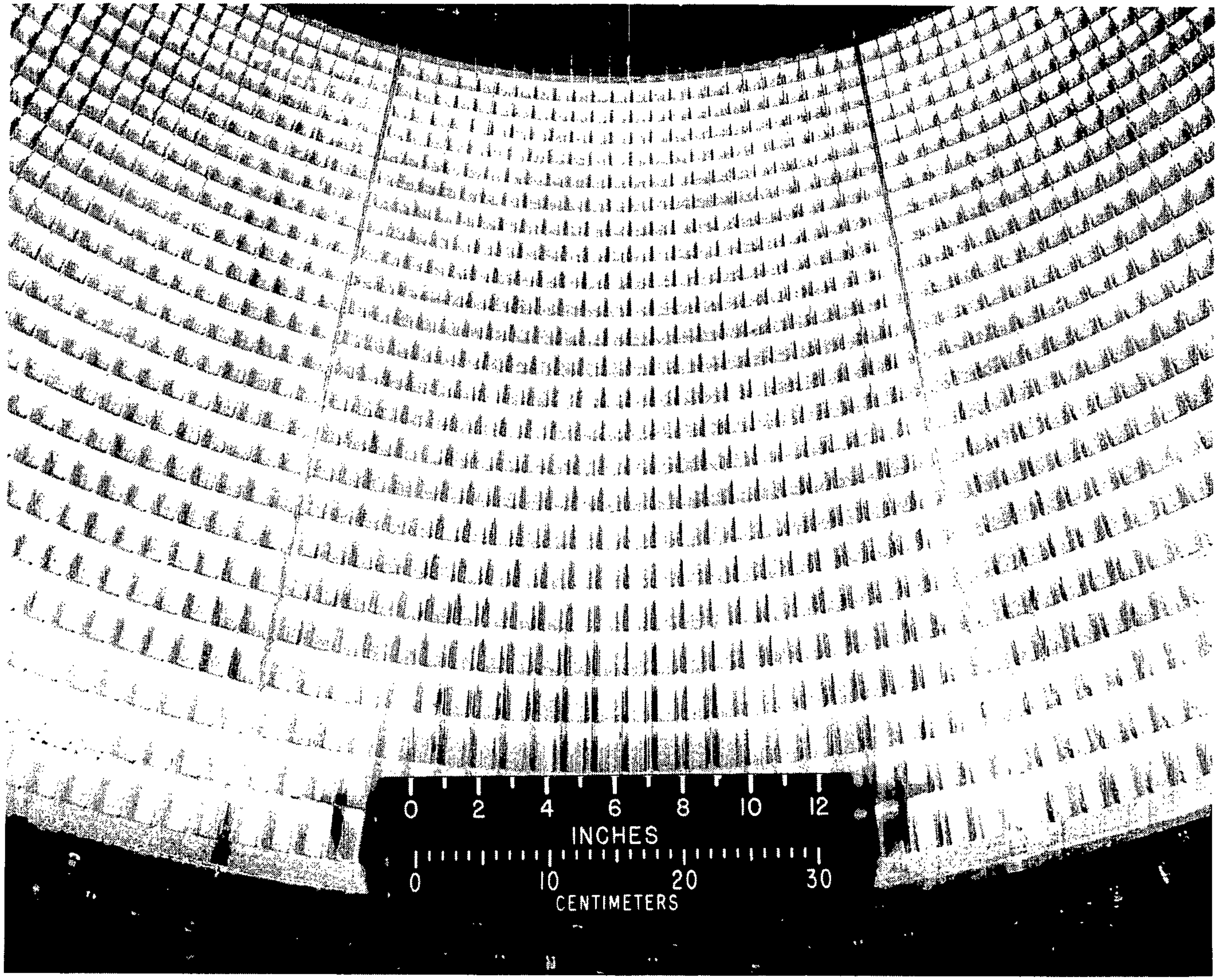
(b) Cylinder 2.

Figure 1.- Concluded.



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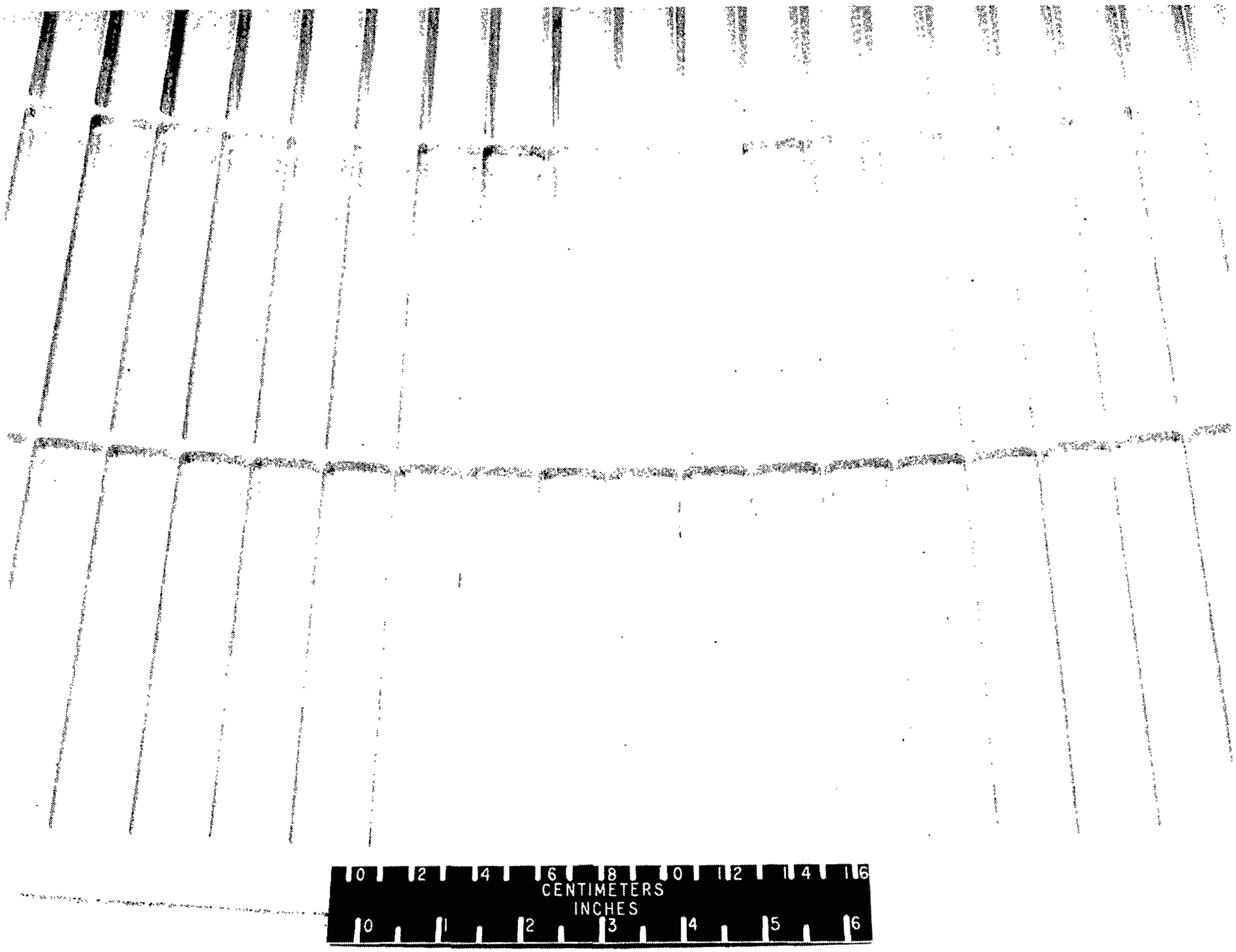
(a) Cylinder 1.

Figure 2.- Wall construction of test cylinders.



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(b) Cylinder 2.

Figure 2.- Concluded.

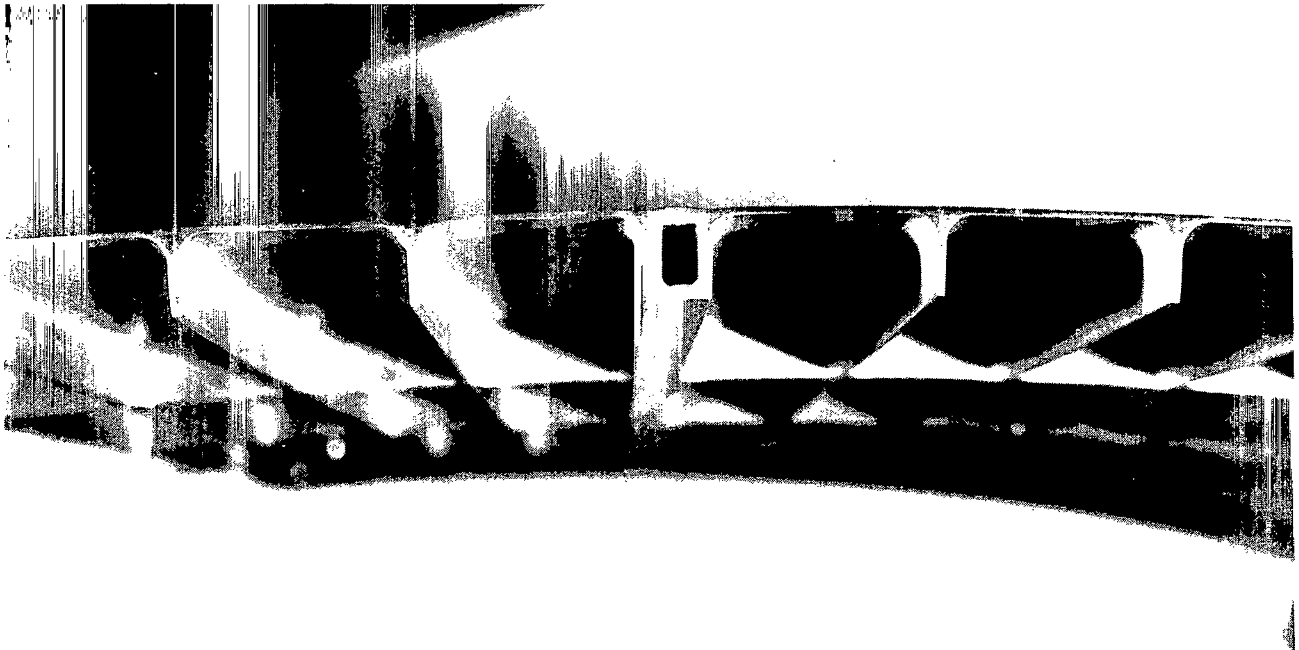

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Figure 3.- Longitudinal weld in wall of test cylinder. Stringer spacing equals 1 inch (2.54 cm).

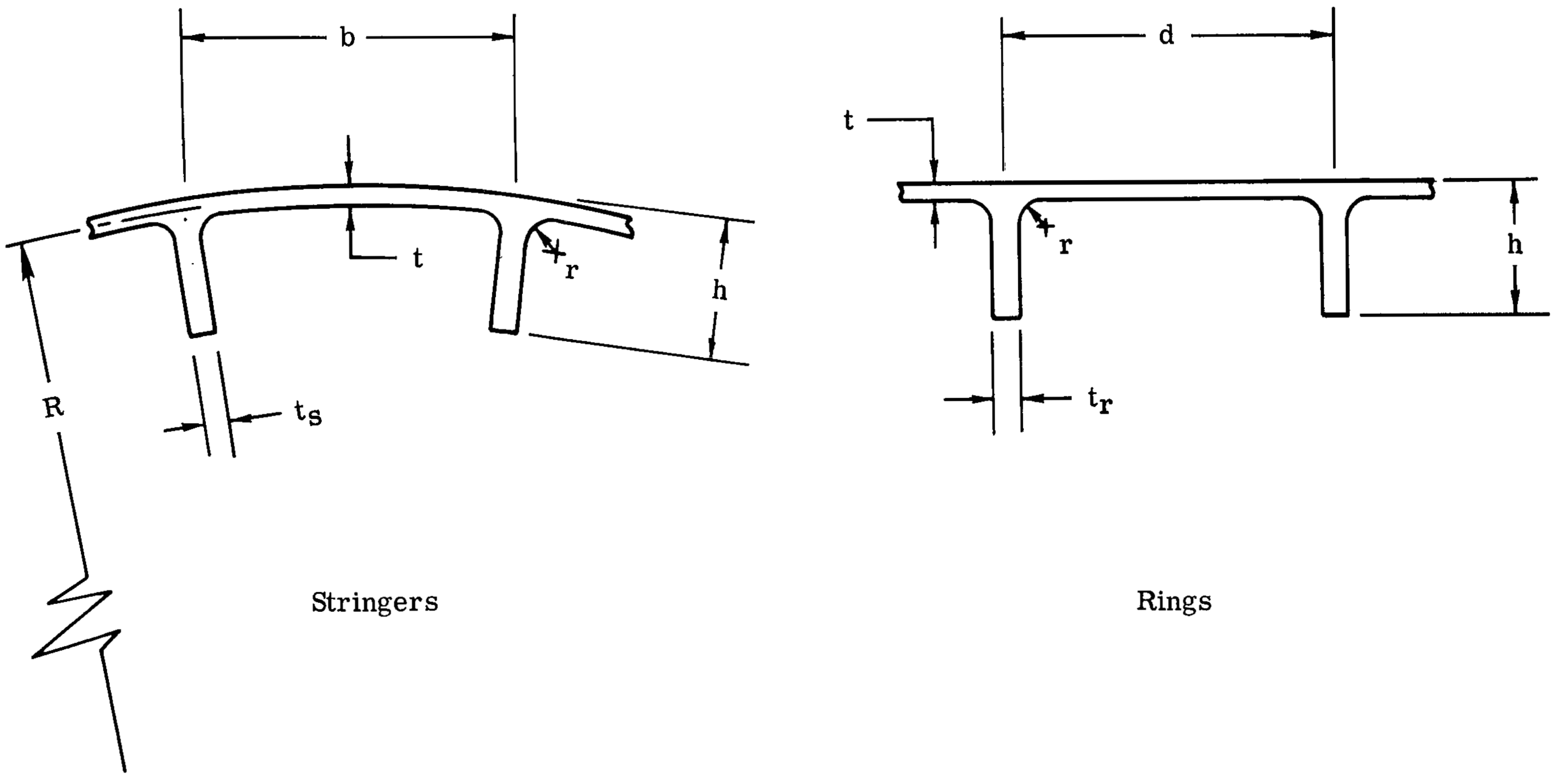
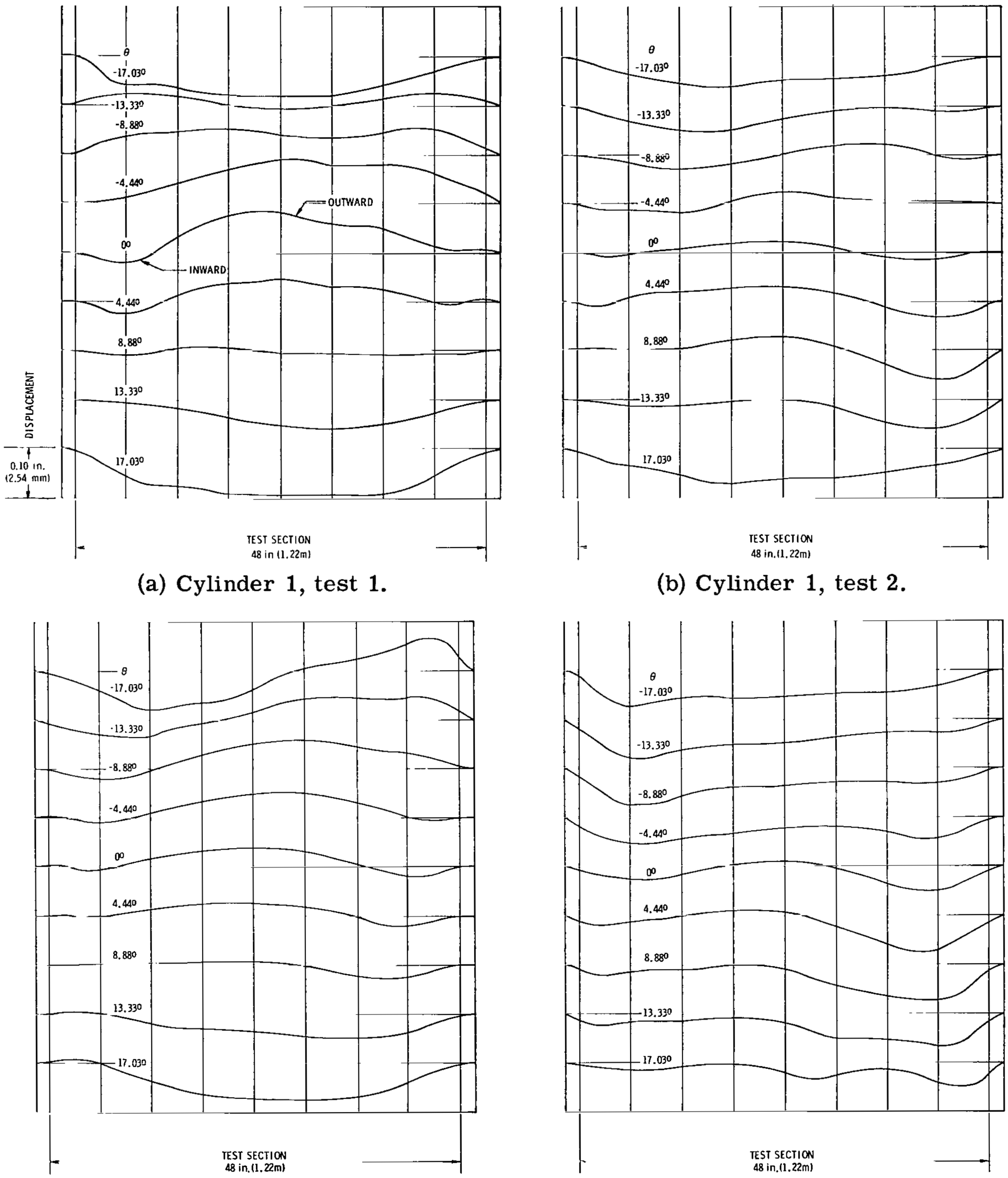


Figure 4.- Nomenclature used in definition of wall geometry of test cylinders.



(a) Cylinder 1, test 1.

(b) Cylinder 1, test 2.

(c) Cylinder 2, test 1.

(d) Cylinder 2, test 2.

Figure 5.- Scanner plots showing initial imperfections of test cylinders in area of generator of maximum compressive stress. Plots at ±17.03° taken adjacent to longitudinal weld lines.

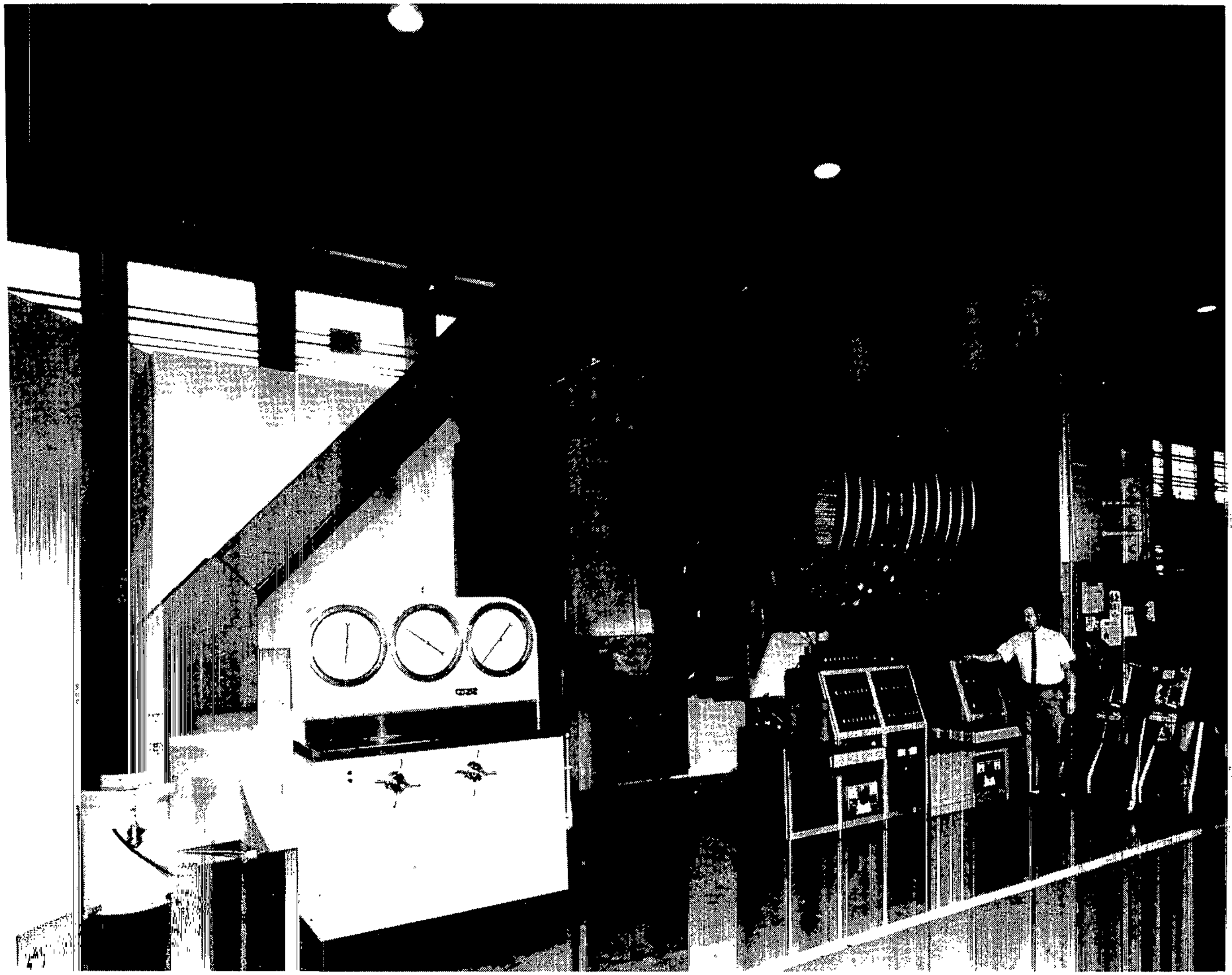


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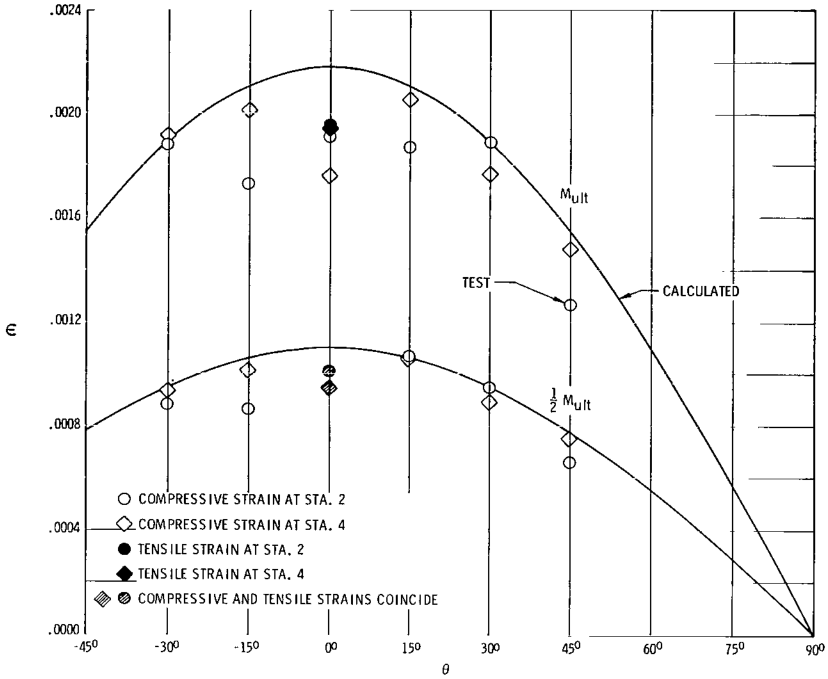
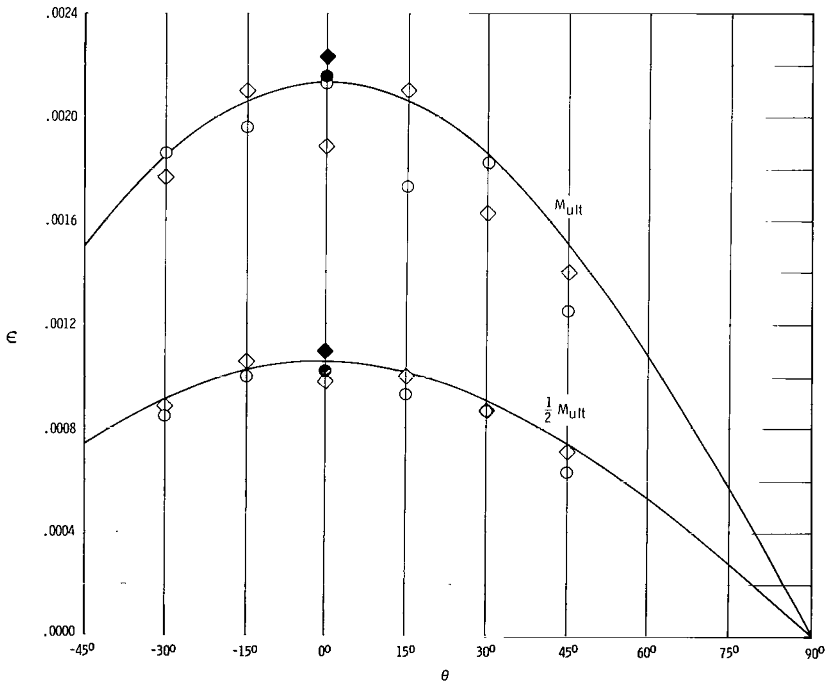
Figure 6.- Test setup.

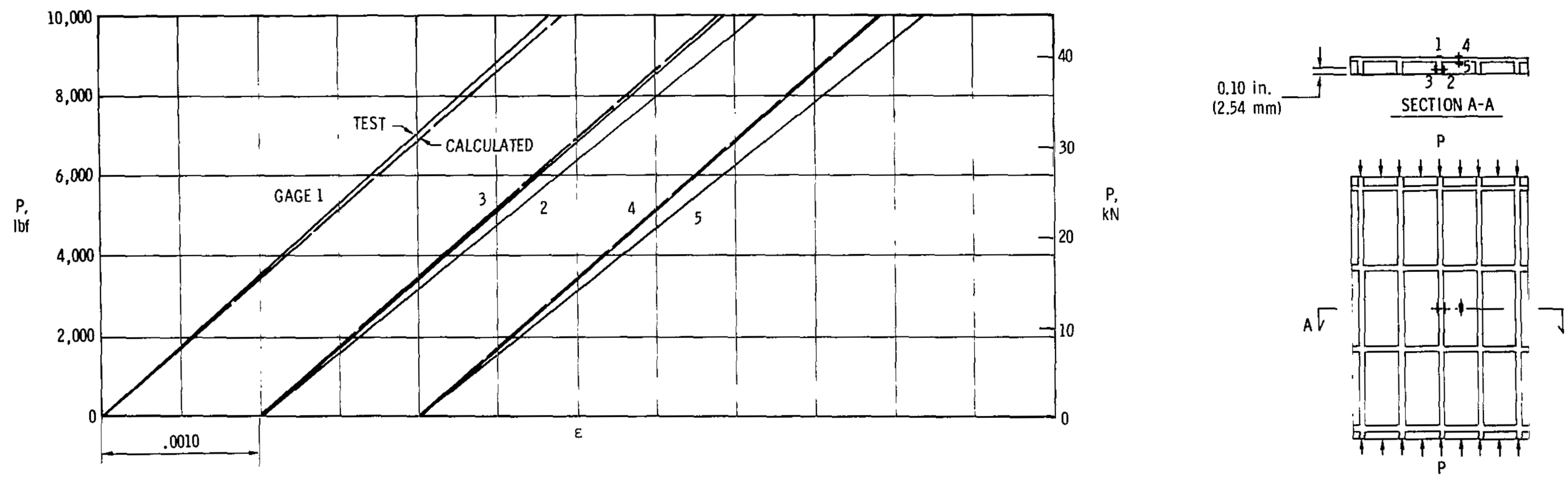


(a) Cylinder 1, test 1.

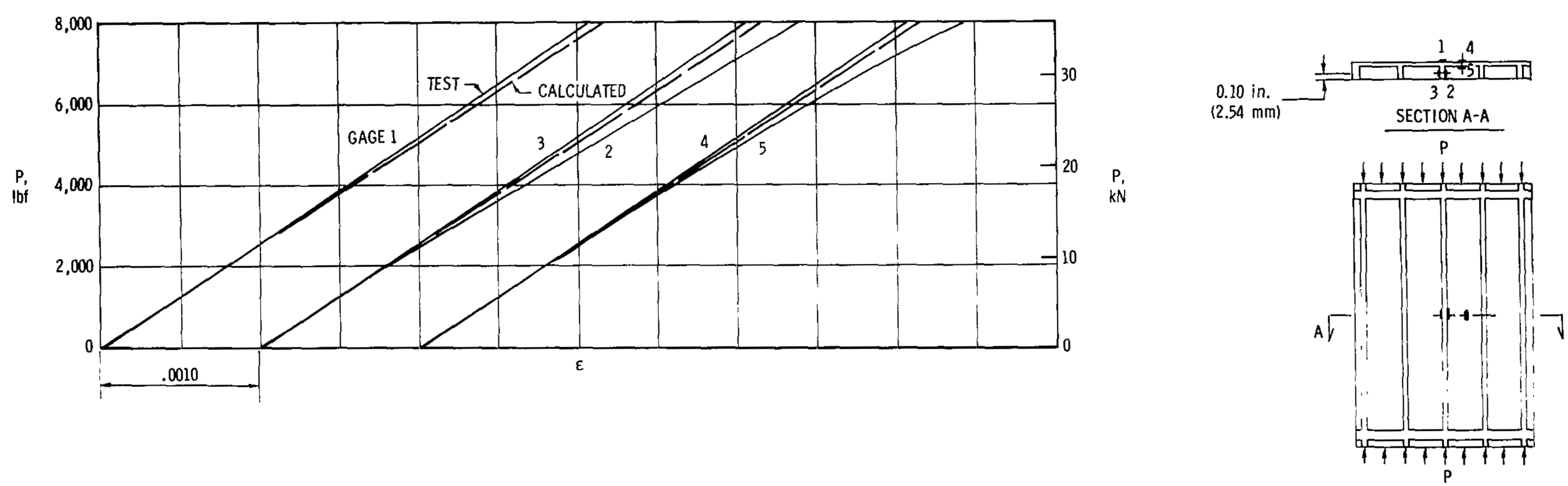


(b) Cylinder 2, test 1.

Figure 7.- Comparison between measured and calculated strain distribution for test cylinders. Station locations defined in figure 9.



(a) Cylinder 1.



(b) Cylinder 2.

Figure 8.- Comparison between measured and calculated strains in coupons cut from cylinder walls.

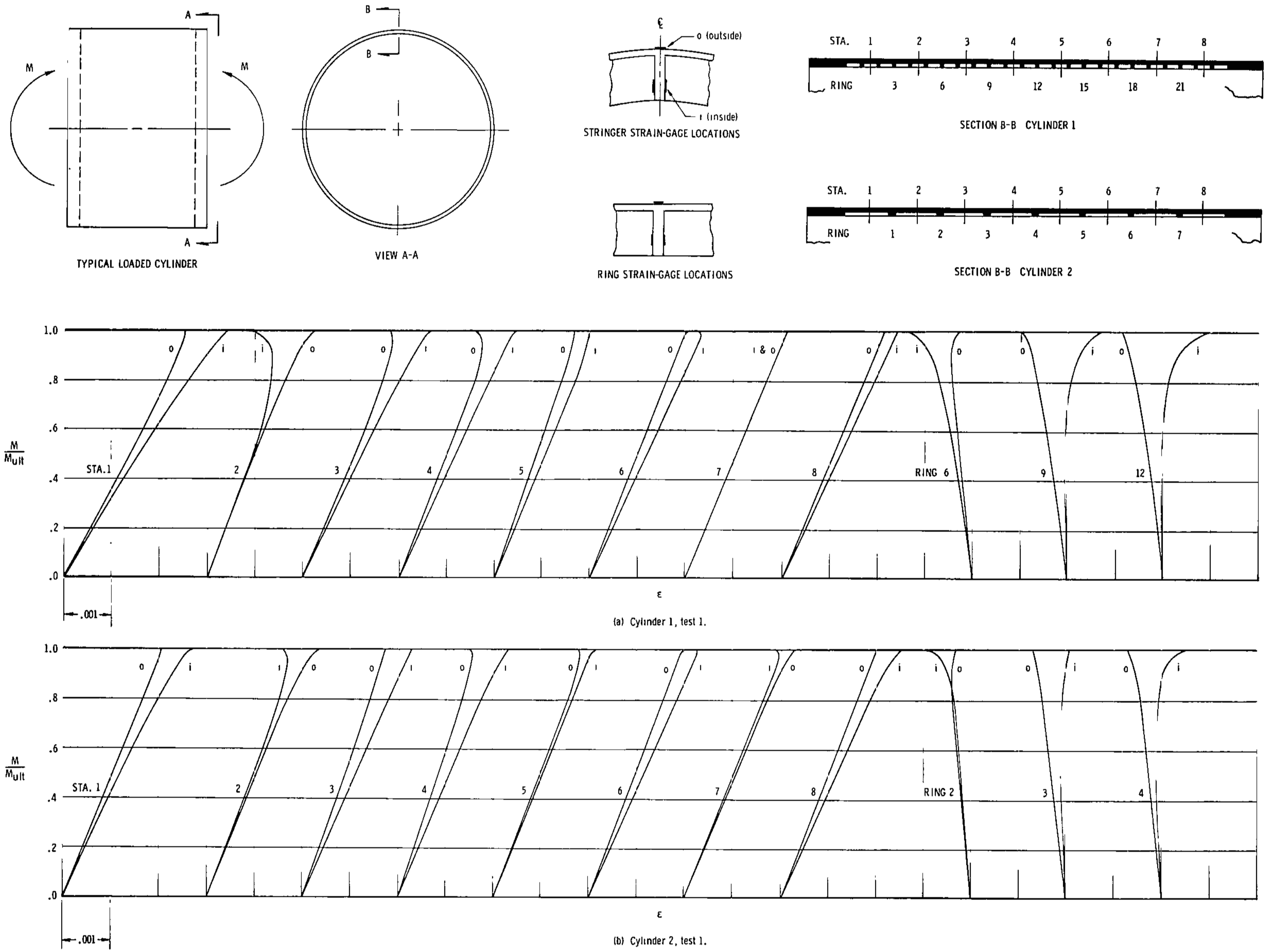


Figure 9.- Strain-gage data obtained along generator of maximum compressive stress. Curves labeled i (inside) show average strain from two gages.

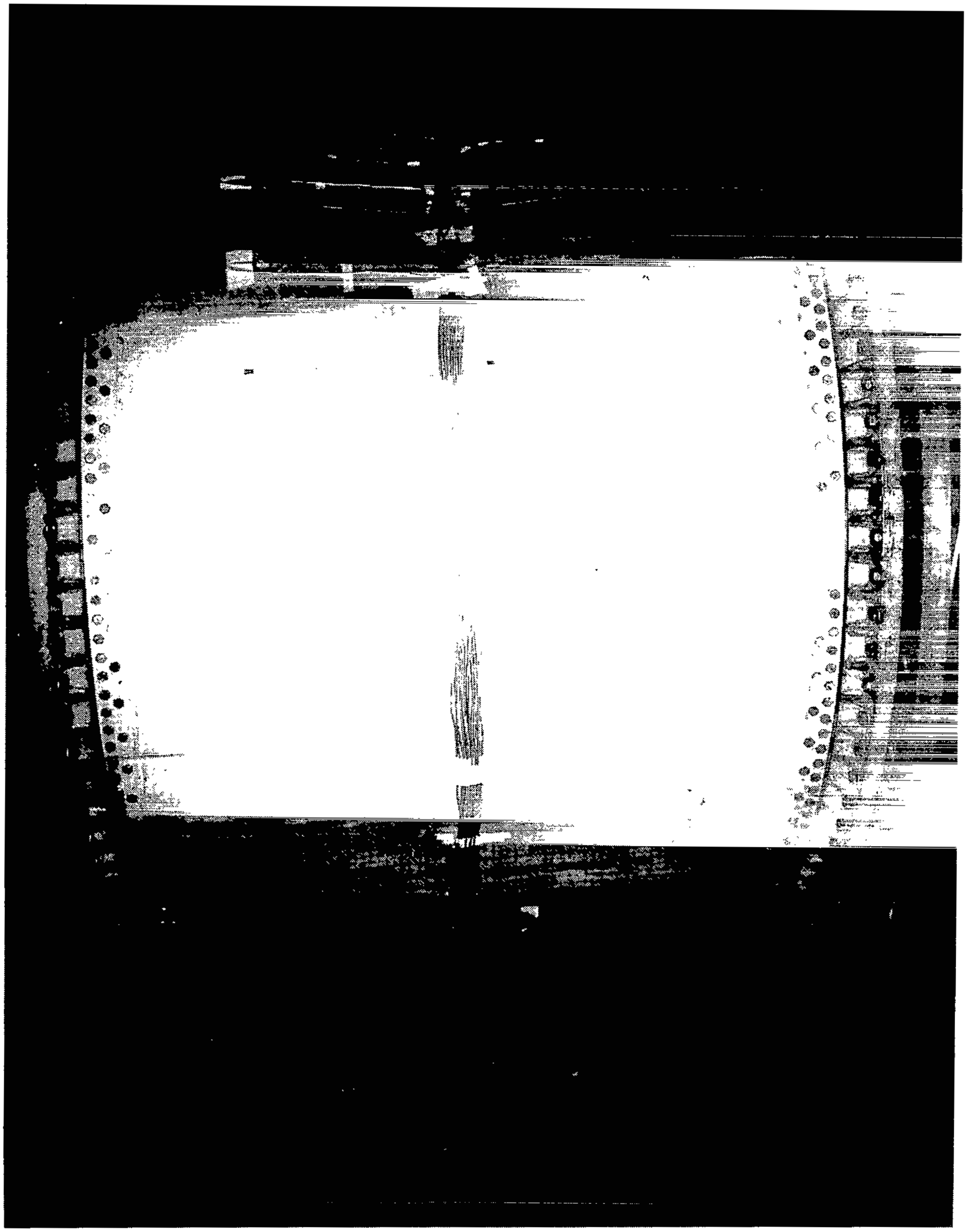


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Figure 10.- Buckling deformations obtained in cylinder 1, test 1.

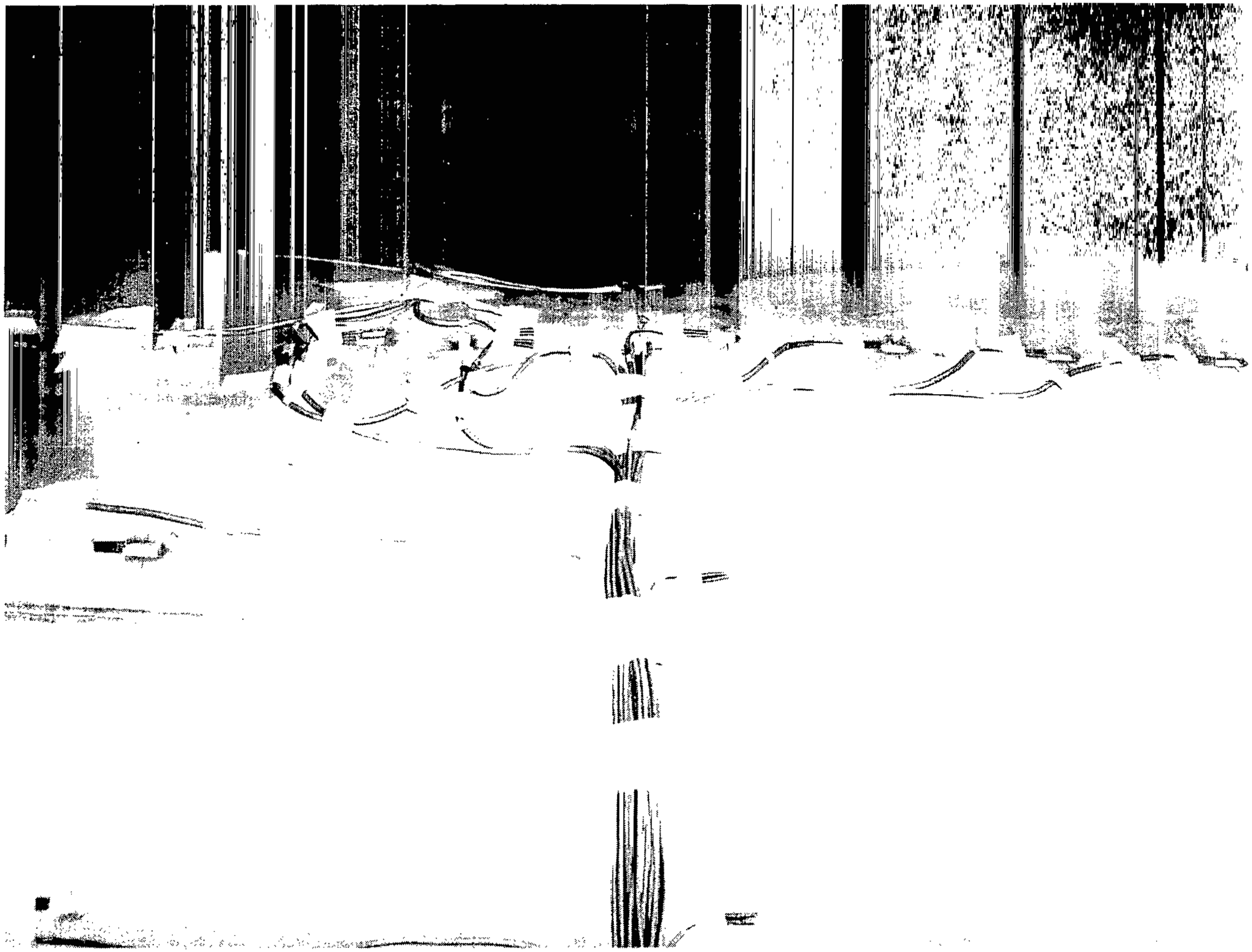

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Figure 11.- Closeup of buckled cylinder. Cylinder 1, test 1.

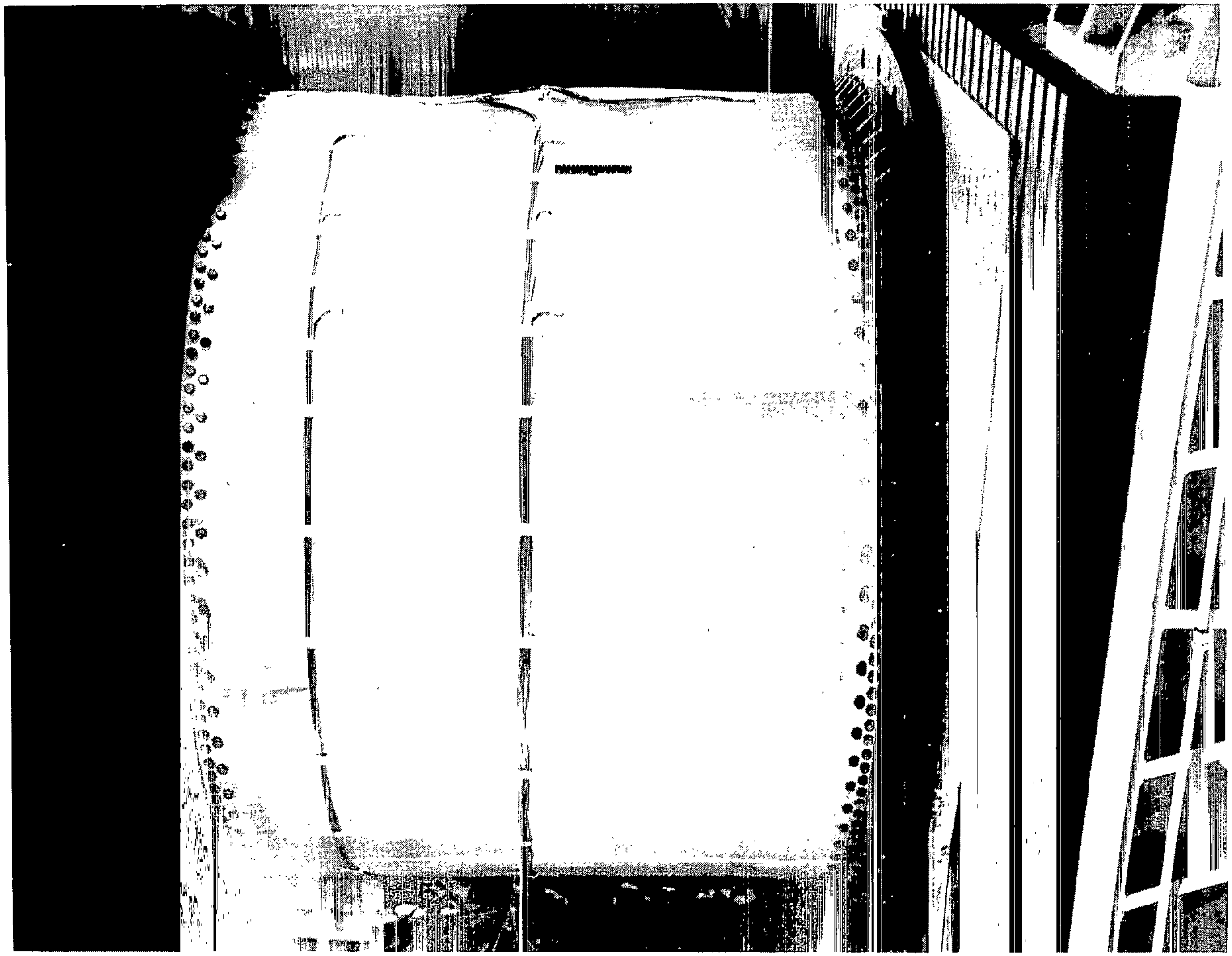

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Figure 12.- Buckling deformations obtained in cylinder 2, test 1.

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Figure 13.- Closeup of buckled cylinder. Cylinder 2, test 2.

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