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LMSC-4-05-71-9 • MAY 1971

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AN EXPERIMENTAL STUDY OF THE STRENGTH AND STABILITY OF THIN MONOCOQUE SHELLS WITH REINFORCED AND UNREINFORCED RECTANGULAR CUTOUTS

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CONTRACT NAS 9-10372

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PALO ALTO, CALIFORNIA

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FOREWORD

The research described in this report was performed under Contract NAS9-10372 with the NASA/Manned Spacecraft Center, Houston, Texas, with Dr. F. J. Stebbins as Contract Monitor.

ABSTRACT

Axial compression tests were run on eleven thin-walled aluminum cylinders having rectangular cutouts. Various types of reinforcement were used around the cutouts, and some tests were run with no reinforcement. The test results are compared with the cylinder buckling loads prior to installation of the cutouts (obtained without damaging the cylinder by using a "buckle-capture" technique), and correlated with computer-predicted failure loads. The latter were based on the use of the STAGS computer program.

For thin cylinders such as these, the test and computer-based analysis shows that for small to moderate size cutouts, reinforcement of the cutout is of no benefit unless the cylinder is of extremely high (geometrical) quality. For cylinder quality and cutout size where reinforcement is beneficial, the relative merits of the various reinforcement configurations are discussed, and an empirical basis for design is proposed.

CONTENTS

Section 1

INTRODUCTION AND CONCLUSION

One of the critical problems in the structural design of launch vehicles and spacecraft is the determination of the required reinforcement around cutouts in the primary shell structure. Although aircraft have always had relatively large cutouts in the primary structure, the major design consideration for aircraft is fatigue, and thus operating stress levels are moderate to low. The simplified design rules for reinforcing a cutout (e.g., the reinforcement area should equal the area of the material removed by the cutout) have been adequate to prevent collapse of the fuselage under compressive loading. On spacecraft and launch vehicles, however, the operating stress is much higher, and aircraft design rules are not adequate.

To predict collapse loads for shells with cutouts requires a nonlinear analysis and has until very recently been clearly outside the state of the art in shell analysis. The large number of parameters makes it impossible to produce design charts by use of a purely empirical approach, and a theoretical analysis has been limited by very high computer costs. Consequently design of cutout reinforcement has been based on rules of thumb which generally are quite conservative due to the uncertainty involved. However, recent improvements in computer technology as well as in numerical analysis methods have brought the computer cost down to a level where it now appears feasible to establish design procedures with a more solid foundation.

The first nonlinear analysis of cylindrical shells with rectangular cutouts was presented in Ref. 1. At that time it was not economically feasible to analyze shells which were thin enough for collapse to occur in the elastic range. This essentially made meaningful comparisons impossible between test and theory for metal cylinders. Later improvements (Ref. 2 and 3) have not only extended the generality of the computer program but also improved its efficiency so that it now is possible to shed some light on the problem of the collapse of shells with cutouts through a combination of analytical and experimental investigations.

The STAGS (STructural Analysis of General Shells) computer program is an analytical means for predicting collapse of shells with cutouts. The development of this program has been sponsored by the Air Force Flight Dynamics Laboratory (AFFDL), the Air Force Space and Missile Systems Organization (SAMSO), and Lockheed's Independent Research Program. The experimental work in this report was designed from its inception to complement that analytical effort. The primary objective of the present program was to provide high quality experimental data from relatively simple configurations (circular cylinders having rectangular cutouts) for comparison with analytical predictions and for verification of the STAGS computer program. A second objective was to develop design guidelines for use in preliminary sizing of the reinforcement around cutouts in cylindrical shells as used in aerospace vehicles. For the range of parameters considered in this program, the experimental results have confirmed the theoretical predictions of the STAGS code and it is anticipated that a more extensive analytical parametric study will develop more detailed design curves for selecting reinforcement configurations for cutouts in stiffened cylinders.

As the work progressed it became increasingly apparent that computer analysis should precede the test work to aid in selecting the most suitable specimen dimensions. As a result, considerably more computer work was included in the preparation of these tests than was originally planned.

Eleven thin-walled aluminum cylinders with cutouts were tested in axial compression. Each cylinder was tested first without cutouts to establish a reference level for this cylinder. Due to the sensitivity of axially loaded cylinders to small initial imperfections, this step was necessary for a proper understanding of the test results. Damage to the specimen during these preliminary tests was avoided by use of a buckle limiting device, consisting of an electrically isolated mandrel mounted inside the cylinder. If the gap between the cylinder and mandrel is small enough, stresses in the buckled specimen will remain in the elastic range.

In view of the small size of this program and the complexity of the problem, all conclusions should be considered tentative. However, we can state the following conclusions with reasonable assurance:

1) For cylinders with an unreinforced cutout good agreement is obtained between test and theory. As reinforcement is added at the cutout edge, the analysis shows that the critical load becomes sensitive to initial imperfections in the shell (away from the cutouts). This behavior is not surprising as the unreinforced hole (included in the analysis) constitutes an imperfection which is well defined and dominates other imperfections.

2) For a given level of imperfection in the original cylinder there is a size of hole above which a test result can be expected to agree with the computed nonlinear collapse load for a perfect shell (including the cutout).

3) For smaller holes, the shell is imperfection sensitive and for such holes there is little benefit in the addition of reinforcement. For instance, if the original cylinder (without cutout) carried about 40% of the classical load, a cutout as large as 45° of the circumference might as well be left without reinforcement.

4) Regarding the type of reinforcement, moment of inertia is primarily needed to suppress bending of the cutout edge. A solid section with large area in relation to its moment of inertia is undesirable because it supplies less bending stiffness and tends to augment the stress concentration at its termination. This merely relocates the site where buckling will first occur.

5) A suitably proportioned longitudinal stiffener is more efficient than the frequently used rectangular frame. The circumferential reinforcement around the cutout seems to be of little value.

6) A method of analysis for cylinders with unreinforced as well as reinforced cutouts is proposed but additional verification should be obtained before it can be adopted as a design procedure.

7) To be a valuable extension of this work, any future tests should be on cylinders with a higher value of the quality parameter ϕ, and with reinforcement even lighter than the present type.

Section 2

TEST SPECIMENS AND PROCEDURES

2.1 Specimen Material and Geometry

The eleven cylinders tested were machined from 6061-T6 aluminum tube stock. This extruded tubing raw stock has an outer diameter of 12.75 inches and an inner diameter of 11.75 inches.

All cylinders were machined to the dimensions shown in Fig. 2.1, the thickness of the thin-walled portion being the only variable within the set of eleven specimens.

The thickened end rings are not the same at each end because a close fitting rigid mandrel had to be inserted from one end. The threaded holes into the end rings serve to attach the buckle capture device, and thus do not have to carry heavy loads. Nevertheless, thread inserts were incorporated into the thinner end ring to supply a more rigid and positive attachment point. The thread inserts were "Keenserts" (NAS 1394CAL).

The purpose of the end rings is to help distribute the load uniformly and to serve as an attachment ring for the buckle capture device.

The test cylinders were measured for wall thickness variation at 24 degree stations around the circumference and at 1.75 inch intervals longitudinally, starting one inch from one of the end rings. The results of these measurements are tabulated in Tables 2.2 through 2.12. A summary of thickness measurements is given in Table 2.1. This table lists the minimum and maximum thickness measured, and the average thickness, based on the seventy-five thickness measurements.

It should be emphasized that considerable care is required to obtain a plus/minus .001-inch variation in thickness on a diameter of twelve inches and when the thickness is only ten to fourteen thousandths. Procedures will be discussed in the next subsection.

2.2 Specimen Manufacture

The appropriate length of raw stock was first machined internally to a diameter of 12.115 $\pm$ .0005 inches. The inner contour of the thicker end ring was also machined in this step.

The aluminum cylinder and a thick-walled steel mandrel .008 inches larger in diameter (at room temperature) were then placed in a furnace and slowly brought to 200°F. At this temperature the aluminum cylinder could be placed on the steel mandrel. Upon cooling, the cylinder was ready for external machining, that is to say, shrunk fit onto the mandrel. Fig. 2.2 shows the steel mandrel and one of the aluminum cylinders after machining.

The machining of the outer surface was done in three successively "finer" passes leading to the desired thickness (.009 or .014 inches, nominally). The variations in thickness observed in the cylinders (Tables 2.1 through 2.12) are due to minor eccentricity of the lathe, tool wear, vibration and temperature effects. Considerable precautions were taken to minimize these effects.

The finished cylinder is removed from the mandrel by placing the unit in a furnace and reheating it to 200°F, at which temperature it slides right off.

2.3 Measurement of the Cylinders

The cylinder was measured at seventy-five locations equispaced in the circumferential and axial directions, as explained in Section 2.1. This was done with a sheet metal micrometer, as shown in Fig. 2.3. The micrometer has a six-inch deep throat and a spherical-tipped anvil. Although the micrometer reads to .0001-inch precision (with a vernier), minor misalignment of the micrometer's measuring axis makes it difficult to get readings which repeat to better than $\pm$ .0006. (The micrometer is usually intended for use on flat sheet for which it is easier to be sure that the micrometer is correctly aligned.) For this reason, readings were rounded off to the nearest thousandths of an inch.

The locations of measurement points were marked on the cylinder with the help of a template, and the values measured written at the locations with a soft wax pencil.

2.4 The Buckle Capture Technique

When a cylinder with a high R/t ratio buckles, a diamond pattern of buckles is formed with quite high bending stresses in certain regions. The purpose of the buckle capture technique is to limit the magnitude of the bending stresses in the buckles. This is achieved by the use of a close fitting mandrel placed inside the cylinder prior to axial loading, which limits the depth of the buckle amplitude. Precautions are taken to be sure that no axial load is carried by the mandrel. This is done by attaching it to the cylinder end ring at one end only. At the other end, lateral support is required, and this is provided by means of linear ball bushings which permit small cantilevered shafts attached to the cylinder loading plate to slide axially. The bushings are pressed-fit into an intermediate bracket which serves to electrically isolate the mandrel segment and makes it possible to adjust its radial position relative to the cylinder. This assembly is shown in Fig. 2.3, disassembled, and partially installed in a cylinder in Fig. 2.4.

Since contact with the mandrel would constitute lateral support for the cylinder membrane (allowing it to sustain a greater axial stress before buckling), an electrical sensing system is used to insure that the cylinder and mandrel are not in contact. Any such contact closes an electrical circuit which turns on a warning light.

The mandrel is built of three separate segments which can be positioned radially at both ends of the cylinder so that the gap between the cylinder and mandrel can be adjusted as required. For the .014-inch wall cylinders (which present the greatest problem since bending stresses are proportional to the wall thickness), it was found that a gap of six to ten thousandths is suitable. The gap is "set" using a seven-mil (.007 inches) shim, which is removed after the mandrel fasteners are tightened. If a smaller gap is used, the cylinder can come in contact with the mandrel before it buckles. The onset of buckling is unmistakable since the formation of buckles produces a sharp noise. The contact of the cylinder and mandrel due to too small a gap is caused by the gradual growth of imperfections under increasing load. These imperfections, which, as will be seen later, are the true measure of a cylinder's quality (from a load carrying standpoint), are minor deviations from the true cylindrical form - a slight "waviness" of the cylindrical surface, too small to be detected by the naked eye.

The stress-strain curve for 6061-T6 departs from true linearity (i.e., elasticity) at a surprisingly low level. Although the yield point is usually given as 35000 psi, some plastic behavior is apparent even at 20000 psi, which is, for most structural purposes, regarded as well within the elastic range of the material. The significance of this is that some small permanent set occurs on the first buckle, even with the mandrel set at the "optimum" gap. The first buckle thus introduces a new set of "low-level imperfections", so that the buckling load achieved after the cylinder is unloaded and re-loaded is lower than the buckling load achieved on the first loading cycle. But thereafter, the subsequent buckling load levels remain essentially at the same level. This is because no new level of imperfections is introduced on subsequent buckles. Once again, it should be emphasized that the new imperfections (introduced by the first buckling) are not visible and must therefore be a "waviness" of micro-inch amplitude. The very pronounced pattern visible after buckle (with a mandrel) is only of a few mils in amplitude (see Fig. 2.9). The eye is extremely sensitive to geometrical imperfections when they occur on polished surfaces.

The tests with cutouts are therefore not performed on "damaged" cylinders. The buckle capture technique merely alters the imperfection level slightly. Since the cylinders already vary considerably in imperfection level as they "arrive" at the first loading test, the purpose of the buckle capture tests is to establish at what point on this relative scale the cylinder is located. Some of the cylinders had a first buckling load which was lower than the second (or "repeatable") buckling load of other cylinders of the same or smaller thickness. The first buckling load of Cylinder #7 with a minimum thickness of 13 mils was 3075 lbs, whereas the second or "repeatable" buckling load of Cylinder #5 with a minimum thickness of 12 mils was 3970 lbs. Obviously, minimum thickness is not the only criterion of quality. It is difficult to explain this phenomenon. We suspect that the specimens may be susceptible to "damage", i.e., imperfection addition, in general handling. And yet, considerable care is taken in this respect, notably in avoiding touching the thin membrane portion after release from the fabrication mandrel and during measurement. The latter process is the most likely culprit, and unfortunately it cannot be deleted.

It has also been observed that the repeatable buckling load can be altered by repositioning the cylinder relative to the end loading plates. The reason for this variation in buckling load is obvious: the contacting faces of the loading plates and the cylinder also have their waviness and imperfections. If a high spot on the cylinder coincides with a high spot on the loading plate, the load transmitted in this region (in lbs per lineal inch) is bound to be higher than in regions where two low spots coincide. When a pair of high spots match up, and also coincide with a thin region of the cylinder, the "repeatable" buckling load will drop markedly. Changing the relative position of the end plates once more returns the buckling load to the previous higher level, confirming the diagnosis. The variety of ranges possible from one cylinder to another is, once again, a function of the imperfection level, but this time the imperfection level of the cylinder end planes. Note that for Cylinders #7 and #10 this range was only ±15 lbs, whereas for Cylinder #5 the range was ±150 lbs. In both cases the end plane tolerances on flatness were ± .0005 inches, and these were in fact checked while the cylinder was still on the lathe. But the smallness of these imperfections can be appreciated better when it is realized that ±150 lbs represents only ±4% of the buckling load in question.

The buckling loads obtained in twenty-five successive tests on each cylinder are listed in Tables 2.13 through 2.23. Four buckling loads are registered with the top and bottom plate set in the "zero-degree" position. The first of these (shown in parentheses), usually much higher than the rest, is that of the first loading cycle and should be disregarded. Three buckling loads are then determined with bottom plate in the zero position and the top plate set in the 90-degree, 180-degree and 270-degree positions. Then the top plate is held in the zero position and the bottom plate rotated to the 90, 180 and 270 positions. For each combination of positions, the buckling process is repeated three times. The repeatable buckling load reported in Table 3.1 is the mid-range value for these 24 tests (the first buckling load, or 25th test value, is disregarded in determining this mid-range).

The wide range of imperfection levels, even when thickness tolerance is closely held and the manufacturing process carefully controlled, makes it imperative that each thin-walled cylinder be rated by the buckle capture technique so that a good reference load exists for the subsequent tests with cutouts.

2.5 Installing the Cutouts and Reinforcement

Following tests with the buckle capture technique, two rectangular cutouts were made on the cylinder. In each case, these were centered at the cylinder midheight and 180 degrees apart on the circumference.

The cutouts were made by drilling 0.062-inch diameter holes at each corner of the proposed cutout, and then sawing along prescribed lines with a high-speed dental wheel. The wheel is driven by a hand-held Dremel motor. The cylinder is held in a felt-lined wood cradle, and the operator's hand is braced on a bar fastened to the cradle. Some cleaning up and deburring with a swiss file is necessary. Because of the high speed of the abrasive wheel, almost no tool pressure is required. The width of the cut is about 0.025 inches.

The size of the cutouts on all cylinders was 45 degrees of arc by three inches in the axial direction. One exception to this was Cylinder #1 which had cutouts with a 30-degree arc. This cylinder constituted an exploratory test. The arc was increased to 45 degrees on all subsequent cylinders because this makes the range between buckling with and without cutout wider, and because for small cutouts the stress concentrations fall in the plastic range.

Tables 3.1 and 3.2 summarize the cylinder test parameters and buckling loads. In these tables it is seen that four cylinders were tested without reinforcement on the cutouts.

All reinforcement of the cutouts consisted of angle sections. Fig. 2.6

shows the three basic types of reinforcement referenced in Tables 3.1 and 3.2.

These very thin angles were machined from bar stock. A "back-up" bar is needed when machining the last outstanding leg. Thickness tolerance was $\pm$ .001 inches. The figure also shows the tapered end details used in all reinforcing application except the type "P" reinforcement of Cylinder #7, and the location of holes used to attach the reinforcement to the cylinder (using 2-56 screws). The purpose of the screws was to provide good clamping during the bonding of the reinforcement to the cylinder. It is felt that the bonding is the primary fastener and that the screws could have been removed after they had served their clamping function during the bonding. The cement used was Hysol 0151 with a 24-hour room temperature cure.

All reinforcement was installed on the outside surface of the cylinder with the exception that Cylinder #10, which had the same reinforcement as Cylinder #9, but installed on the inside of the cylinder.

Figures 2.7 and 2.8 show how angle reinforcement (with the same cross section as type "A") was arranged as the "picture frame" around the cutout of Cylinder #7. This is called type "P" reinforcement in Table 3.1.

2.6 Method of Loading

In all tests, with or without cutouts, the cylinders were loaded by a screw-driven "SR-4, FGT" universal testing machine of 50,000 lb capacity. This machine has several loading ranges. The two ranges used were 2500 or 10,000 lbs full scale. The resolution of this machine is 0.2 percent of the "full scale" being used, and the accuracy is 0.5 percent of the "full scale" used, or the resolution figure, whichever is larger.

The load is applied to the cylinder through a two-inch thick aluminum end plate at each end of the cylinder. These square plates have their contacting surfaces machined to a flatness better than $\pm$ 0.005 inches.

The more usual arrangement in a cylinder compression test is to have one of the end plates resting directly on the platen of the machine and to have a spherical seat bearing between the other plate and the cross head. This method has been discarded as unsatisfactory because the spherical seat bearing is only free to rotate while the cylinder is at very low loads. At higher loads, the friction in the spherical seat is too great to permit rotation.

A better solution is to place one of the end loading plates (the lower one) on top of the cross-head, place the cylinder over this and the other end loading plate at the top of this stack, then pull down on the upper plate with a pull rod which passes through both loading plates, the cylinder and the cross-head, and is connected to the platen of the test machine. The latter is then driven downwards to load the cylinder. In addition to the rod's flexibility, a two-axis flexure is added to this tension train, providing assurance that the upper loading plate is completely free to rotate about any axis. With close tolerances on the rod and through-holes, concentricity of the loading axis with the cylinder axis is also easier to insure.

The loading rate, which is not critical in tests such as these, was approximately 400 lbs per minute. The loading was stopped at regular load intervals to permit scanning of the strain gages. During these stops, no unloading (or stress relaxation) was observed.

2.7 Strain Gages and Related Instrumentation

A total of 176 strain gages were used on the eleven cylinders tested. Of these, 30 were part of three-element rosettes. Twelve more were part of two-element "T-rosettes". The ten three-element rosettes and six two-element rosettes were all placed on Cylinder #2. The remaining 134 were 1/8-inch gage length W. T. Bean BAE-13-125BB-120 gages. Eastman 910 cement was used to bond the gages to the cylinder. In all cases (including rosettes), gages were arranged in back-to-back pairs so that bending stress (or strain) could be separated from membrane stress (or strain). The data tabulated are given in the form of membrane and bending stress (or strain) at a "station", which means "at a back-to-back pair of gages or rosettes".

Strain gage signals were recorded by means of a digital Data Acquisition System (DAS). The measuring element of the DAS is an integrating digital voltmeter which reads to microvolts. The DAS also includes a channel scanner, a printer (for test monitoring), and a tape punch. The punch tape is "read" and processed by a Tymshare computer which substracts the zero datum, applies the required scale factor and tabulates the data in any specified form. In the case of rosettes, the Mohr circle of stress equations are solved. The resolution of the system is ± 5 microstrain, and the accuracy is ± 1.25 percent (or better) of the value being read (or five microstrain, whichever is greater). Most of the inaccuracy stems from uncertainty in the gage factor (which is quoted to ± 1.0 percent accuracy), so that on a relative basis, the accuracy is probably even better than the 0.5 percent.

A shunt calibration is performed with a high precision resistor on a leg of the bridge whose resistance has been measured to 0.1 ohm accuracy. Line resistance errors are corrected and the bridge power supply voltage is held to within ± 0.1 percent.

In the case of rosette data where strain rather than stress is reported, the ± 1.25 percent accuracy (of the reading) still holds except that an additional absolute error may exist in that the elastic modulus is assumed to be 10.3×10^6 psi and Poisson's ratio to be 0.30. On a relative basis (i.e., comparing stresses at different load levels or at different stations on the same cylinder) the modulus and Poisson errors can be disregarded. Since all cylinders were cut from the same piece of tube, the variation of properties from cylinder to cylinder is very small and comparisons of stress from one cylinder to another therefore presents only a small error possibility.

For most cylinders, only strain measurements were reported. This is because with single element gages only the strain is known unless the stress at the point is uniaxial. For Cylinder #2, rosettes were used and the full stress condition is measured, so stresses can be given in the tables. Stresses were reported for the single element stations on this cylinder because they were used at points for which it was known (from the geometry and loading condition) that the stress was practically uniaxial. This last remark also applies for the ten single-element stations of Cylinder #8. In this last case it was known (from data on Cylinder #2) that although the stress was not uniaxial, the stress transverse to the gage element was so small that errors less than five percent would result if the stress was assumed to be uniaxial.

Cylinders #2 and #8 were heavily strain gaged because these had unreinforced cutouts, and the strain gages made it possible to study the growth of bending stresses preceding buckling.

Most of the reinforced cylinders were strain gaged at seven stations. The general goal here was to determine how much of a strain concentration the reinforcing was causing. Bending stresses (as roughly inferred from the bending strain tabulations) were not very large compared to those seen in cylinders having unreinforced cutouts. From the standpoint of comparisons with computer analyses, strain gages and the deformations they measured were more interesting and valuable in the unreinforced cylinders than in the reinforced cylinders.

Strain gage data are tabulated in Tables 3.3 through 3.13 and curves are plotted for Cylinder #2 in Figs. 3.1 through 3.3.

The location of strain gage stations on each test cylinder is given in Figs. 3.4 through 3.10.

TABLE 2.1

SUMMARY OF CYLINDER THICKNESS (MILS)

Cylinder Number	Minimum Thickness	Maximum Thickness	Average Thickness
1	14	16	14.76
2	14	15	14.68
3	12	14	12.81
4	12	16	14.64
5	12	14	13.27
6	12	15	13.67
7	13	15	13.73
8	9	11	9.72
9	8	11	9.50
10	9	11	9.53
11	9	11	9.53

TABLE 2.2

THICKNESS MAPPING FOR CYLINDER #1

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.014	.014	.014	.014	.014
	.014	.014	.014	.014	.014
48	.014	.014	.014	.015	.015
	.014	.014	.015	.015	.016
96	.015	.015	.015	.015	.016
	.015	.015	.015	.015	.016
144	.015	.015	.015	.015	.015
	.016	.016	.015	.015	.015
192	.016	.015	.015	.015	.015
	.016	.016	.015	.015	.015
240	.016	.015	.015	.015	.015
	.015	.015	.015	.015	.015
288	.015	.015	.014	.014	.014
	.015	.014	.014	.014	.014
336	.014	.014	.014	.014	.014

Cut out centers at 85° and 265°

TABLE 2.3

THICKNESS MAPPING FOR CYLINDER #2

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.014	.014	.014	.014	.014
	.014	.015	.014	.014	.014
48	.015	.015	.014	.014	.015
	.015	.014	.014	.015	.015
96	.015	.014	.014	.015	.015
	.015	.014	.015	.015	.015
144	.015	.014	.014	.015	.015
	.015	.015	.015	.015	.015
192	.015	.015	.015	.015	.015
	.015	.015	.015	.015	.015
240	.015	.015	.015	.015	.015
	.015	.015	.015	.015	.015
288	.015	.015	.015	.015	.015
	.014	.014	.014	.015	.015
336	.015	.014	.014	.014	.015

Cut out centers at 5° and 185°

TABLE 2.4

THICKNESS MAPPING FOR CYLINDER #3

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.013	.012	.012	.012	.012
	.013	.012	.012	.012	.012
48	.013	.012	.012	.012	.013
	.013	.013	.012	.013	.013
96	.013	.013	.013	.014	.014
	.013	.013	.013	.013	.014
144	.013	.013	.014	.013	.014
	.013	.014	.014	.013	.014
192	.013	.013	.013	.013	.014
	.013	.013	.013	.013	.013
240	.013	.013	.013	.013	.013
	.013	.013	.013	.013	.013
288	.013	.013	.013	.013	.013
	.013	.013	.012	.013	.013
336	.012	.012	.012	.013	.013

Cut out centers at 0° and 180°

TABLE 2.5

THICKNESS MAPPING FOR CYLINDER #4

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.015	.014	.014	.014	.012
	.015	.015	.014	.014	.012
48	.015	.015	.014	.014	.013
	.015	.015	.014	.014	.014
96	.016	.016	.015	.015	.014
	.016	.016	.015	.015	.015
144	.016	.015	.015	.015	.015
	.016	.015	.015	.015	.014
192	.016	.015	.015	.014	.014
	.016	.016	.014	.014	.013
240	.016	.015	.014	.014	.013
	.016	.016	.014	.014	.014
288	.016	.016	.015	.014	.014
	.016	.015	.014	.014	.014
336	.015	.015	.014	.014	.013

Cut out centers at 58° and 238°

TABLE 2.6

THICKNESS MAPPING FOR CYLINDER #5

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.013	.013	.012	.012	.013
	.013	.012	.012	.012	.013
48	.014	.013	.012	.012	.013
	.013	.013	.013	.013	.013
96	.014	.013	.013	.014	.013
	.014	.013	.013	.014	.014
144	.014	.013	.013	.014	.014
	.014	.013	.013	.014	.014
192	.014	.013	.013	.014	.014
	.014	.014	.013	.013	.014
240	.014	.013	.013	.013	.014
	.014	.014	.013	.014	.014
288	.014	.013	.013	.013	.014
	.014	.013	.013	.013	.014
336	.013	.013	.013	.013	.013

Cut out centers at 24^{o} and 204^{o}

TABLE 2.7

THICKNESS MAPPING FOR CYLINDER #6

(thicknesses in inches)

Degrees	-3.50	-1.75	0.00	+1.75	+3.50
	INCHES FROM CENTER				
0	.013	.013	.013	.013	.013
	.014	.013	.013	.013	.012
48	.014	.014	.013	.013	.013
	.014	.014	.013	.014	.014
96	.014	.014	.013	.014	.014
	.014	.014	.014	.014	.014
144	.015	.014	.014	.014	.014
	.014	.014	.014	.014	.014
192	.014	.014	.014	.014	.015
	.014	.014	.014	.014	.014
240	.014	.014	.014	.014	.014
	.014	.013	.013	.014	.014
288	.013	.014	.014	.014	.014
	.014	.013	.013	.013	.013
336	.013	.013	.013	.013	.013

Cut out centers at 0° and 180°

TABLE 2.8

THICKNESS MAPPING FOR CYLINDER #7

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.014	.013	.013	.013	.013
	.014	.013	.013	.013	.013
48	.014	.013	.013	.013	.013
	.014	.014	.014	.014	.014
96	.015	.014	.014	.014	.014
	.015	.014	.014	.014	.014
144	.015	.015	.014	.014	.014
	.015	.015	.015	.014	.014
192	.015	.014	.014	.014	.014
	.015	.014	.014	.014	.014
240	.014	.014	.014	.014	.014
	.013	.013	.013	.013	.013
288	.013	.013	.013	.013	.013
	.014	.014	.013	.013	.013
336	.014	.013	.013	.013	.013

Cut out centers at 0^{o} and 180^{o}

TABLE 2.9

THICKNESS MAPPING FOR CYLINDER #8

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.010	.009	.009	.009	.009
	.010	.009	.009	.009	.009
48	.010	.010	.009	.009	.009
	.010	.010	.009	.009	.009
96	.010	.009	.009	.009	.009
	.010	.009	.009	.009	.009
144	.010	.010	.009	.009	.009
	.011	.010	.010	.009	.009
192	.011	.010	.010	.010	.010
	.011	.011	.010	.010	.010
240	.011	.011	.010	.011	.010
	.011	.010	.010	.010	.010
288	.011	.010	.010	.009	.009
	.010	.010	.009	.009	.009
336	.010	.009	.009	.009	.009

Cut out centers at 36^{o} and 216^{o}

TABLE 2.10

THICKNESS MAPPING FOR CYLINDER #9

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.009	.009	.009	.009	.009
	.009	.009	.009	.009	.009
48	.009	.009	.009	.009	.009
	.009	.009	.009	.008	.008
96	.010	.009	.009	.009	.009
	.010	.010	.009	.009	.009
144	.010	.010	.010	.010	.011
	.010	.010	.010	.011	.011
192	.010	.010	.010	.010	.011
	.010	.010	.010	.010	.010
240	.010	.010	.010	.010	.010
	.010	.010	.010	.009	.010
288	.010	.010	.010	.009	.009
	.009	.009	.009	.009	.009
336	.009	.009	.009	.009	.009

Cut out centers at 36° and 216°

TABLE 2.11

THICKNESS MAPPING FOR CYLINDER #10

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.009	.009	.009	.009	.009
	.009	.009	.009	.009	.009
48	.010	.010	.010	.010	.010
	.010	.010	.010	.010	.010
96	.011	.011	.010	.010	.010
	.011	.011	.010	.010	.010
144	.011	.010	.010	.010	.010
	.010	.010	.010	.010	.010
192	.009	.009	.009	.010	.010
	.009	.009	.009	.009	.009
240	.009	.009	.009	.009	.009
	.010	.009	.009	.009	.009
288	.010	.009	.009	.009	.009
	.010	.009	.009	.009	.009
336	.009	.009	.009	.009	.009

Cut out centers at 144^{o} and 324^{o}

TABLE 2.12

THICKNESS MAPPING FOR CYLINDER #11

(thicknesses in inches)

	INCHES FROM CENTER				
Degrees	-3.50	-1.75	0.00	+1.75	+3.50
0	.009	.009	.009	.009	.009
	.009	.009	.009	.009	.009
48	.009	.009	.009	.009	.009
	.010	.010	.009	.009	.009
96	.011	.010	.010	.010	.010
	.011	.010	.010	.010	.010
144	.011	.010	.010	.010	.010
	.010	.010	.010	.010	.010
192	.010	.010	.010	.010	.010
	.010	.010	.010	.010	.010
240	.010	.009	.009	.009	.009
	.010	.009	.009	.009	.009
288	.010	.010	.009	.009	.009
	.010	.009	.009	.009	.009
336	.009	.009	.009	.009	.009

Cut out centers at 49° and 229°

TABLE 2.13

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #1

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(4450)	
4100	4030
4090	4020
4090	4020

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
3970	4040
3960	4040
3970	4040

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
3990	4040
3980	4040
3980	4040

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
4070	4030
4060	4030
4060	4030

Mid-range value 4030 lbs

Range $\pm$ 70 lbs

TABLE 2.14

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #2

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(4620)	
4560	4620
4540	4610
4550	4610

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
4610	4600
4600	4600
4600	4600

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
4590	4560
4590	4560
4580	4550

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
4620	4580
4610	4580
4610	4570

Mid-range value 4585 lbs

Range $\pm$ 35 lbs

TABLE 2.15

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #3

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg / Btm Pl 0 Deg	Btm Pl 0 Deg / Top Pl 0 Deg
(4500)	
4450	4170
4170	4170
4170	4160

Top Pl 0 Deg / Btm Pl 90 Deg	Btm Pl 0 Deg / Top Pl 90 Deg
4140	4250
4130	4230
4130	4230

Top Pl 0 Deg / Btm Pl 180 Deg	Btm Pl 0 Deg / Top Pl 180 Deg
4330	4180
4320	4170
4300	4170

Top Pl 0 Deg / Btm Pl 270 Deg	Btm Pl 0 Deg / Top Pl 270 Deg
4340	4120
4340	4110
4340	4110

Mid-range value 4280 lbs.

Range $\pm$170 lbs

TABLE 2.16

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #4
(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(3920)	
3700	3760
3700	3760
3690	3760

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
3780	3770
3770	3770
3770	3770

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
3700	3770
3690	3770
3690	3770

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
3710	3770
3700	3760
3700	3760

Mid-range value 3735 lbs

Range $\pm$ 45 lbs

TABLE 2.17

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #5

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(4180)	
4120	4080
4120	4070
4120	4060

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
3840	4080
3840	4080
3840	4080

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
3830	4080
3830	4060
3820	4060

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
3900	3870
3910	3860
3900	3860

Mid-range value 3970 lbs

Range $\pm$150 lbs

TABLE 2.18

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #6

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(4110)	
3520	3500
3520	3500
3530	3500
Top Pl 0 Deg Btm Pl 90 Deg	**Btm Pl 0 Deg Top Pl 90 Deg**
3460	3400
3460	3400
3460	3400
Top Pl 0 Deg Btm Pl 180 Deg	**Btm Pl 0 Deg Top Pl 180 Deg**
3330	3350
3320	3340
3310	3340
Top Pl 0 Deg Btm Pl 270 Deg	**Btm Pl 0 Deg Top Pl 270 Deg**
3500	3300
3500	3290
3500	3290

Mid-range value 3360 lbs

Range $\pm$ 70 lbs

TABLE 2.19

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #7

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(3075)	
3056	3052
3056	3050
3056	3050

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
3070	3050
3070	3050
3070	3048

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
3048	3072
3048	3070
3048	3070

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
3040	3052
3040	3050
3040	3050

Mid-range value 3360 lbs

Range $\pm$ 70 lbs

TABLE 2.20

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #8
(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(1340)	
1300	1240
1295	1255
1300	1250

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
1305	1255
1305	1255
1310	1250

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
1285	1255
1290	1255
1280	1255

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
1285	1240
1285	1235
1280	1235

Mid-range value 1265 lbs

Range $\pm$ 35 lbs

TABLE 2.21

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #9

(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg / Btm Pl 0 Deg	Btm Pl 0 Deg / Top Pl 0 Deg
(1480)	
1455	1445
1450	1440
1450	1440

Top Pl 0 Deg / Btm Pl 90 Deg	Btm Pl 0 Deg / Top Pl 90 Deg
1450	1435
1445	1435
1445	1435

Top Pl 0 Deg / Btm Pl 180 Deg	Btm Pl 0 Deg / Top Pl 180 Deg
1415	1440
1415	1435
1415	1435

Top Pl 0 Deg / Btm Pl 270 Deg	Btm Pl 0 Deg / Top Pl 270 Deg
1455	1435
1450	1425
1450	1430

Mid-range value 1435 lbs

Range $\pm$ 20 lbs

TABLE 2.22

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #10
(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(1390)	
1385	1380
1390	1380
1390	1385

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
1360	1375
1360	1375
1360	1375

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
1370	1385
1370	1385
1370	1390

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
1365	1390
1360	1380
1360	1380

Mid-range value 1375 lbs

Range $\pm$ 15 lbs

TABLE 2.23

BUCKLING LOADS BEFORE CUTOUT FOR CYLINDER #11
(ALL VALUES ARE IN POUNDS)

Top Pl 0 Deg Btm Pl 0 Deg	Btm Pl 0 Deg Top Pl 0 Deg
(1540)	
1540	1525
1530	1525
1540	1525

Top Pl 0 Deg Btm Pl 90 Deg	Btm Pl 0 Deg Top Pl 90 Deg
1555	1555
1550	1555
1555	1555

Top Pl 0 Deg Btm Pl 180 Deg	Btm Pl 0 Deg Top Pl 180 Deg
1530	1575
1530	1575
1525	1575

Top Pl 0 Deg Btm Pl 270 Deg	Btm Pl 0 Deg Top Pl 270 Deg
1555	1590
1550	1585
1550	1590

Mid-range value 1555 lbs

Range $\pm$ 35 lbs

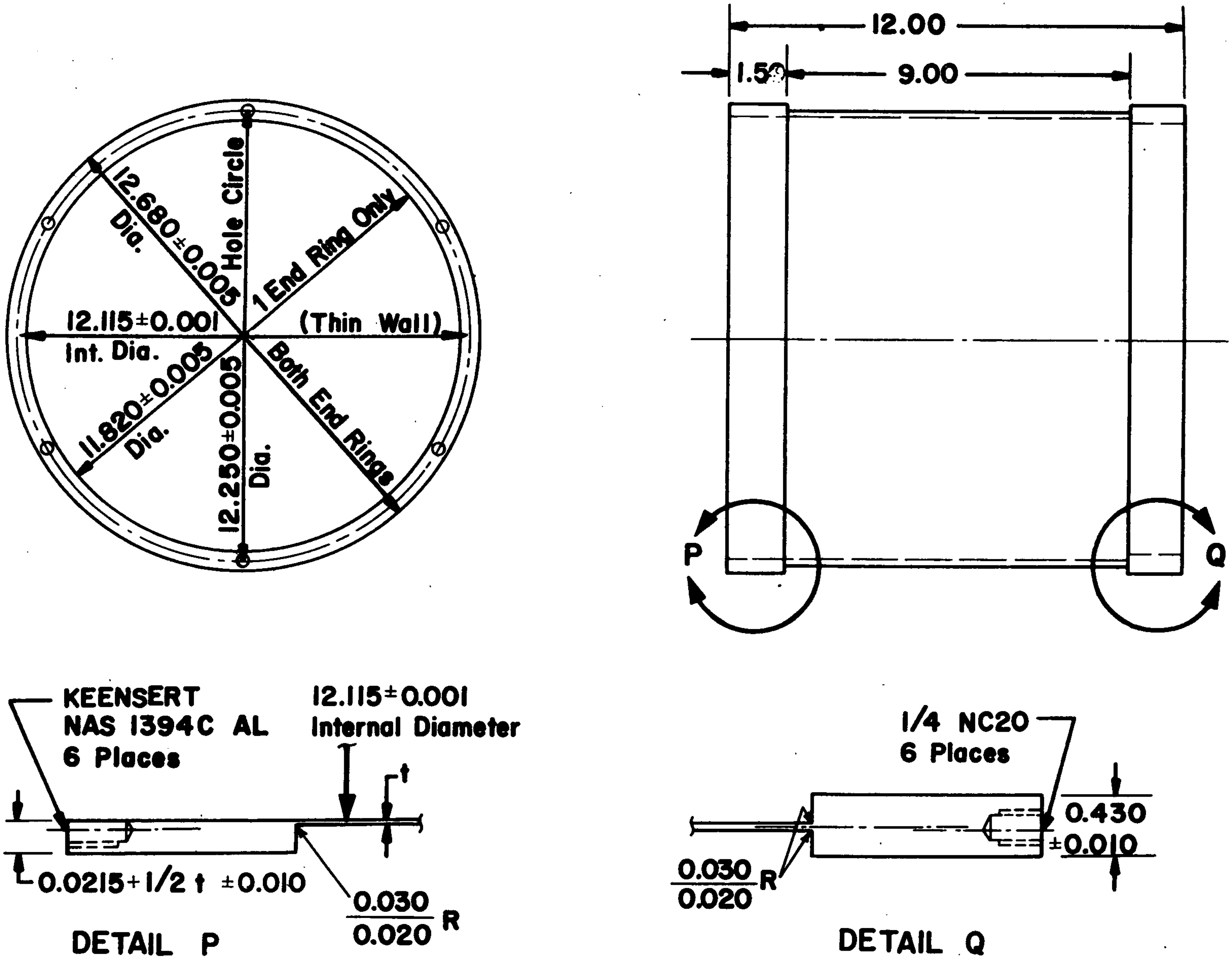


Fig. 2.1 Basic Dimensions of Test Cylinders

Fig. 2.2 Steel Fabrication Mandrel and Finished Cylinder

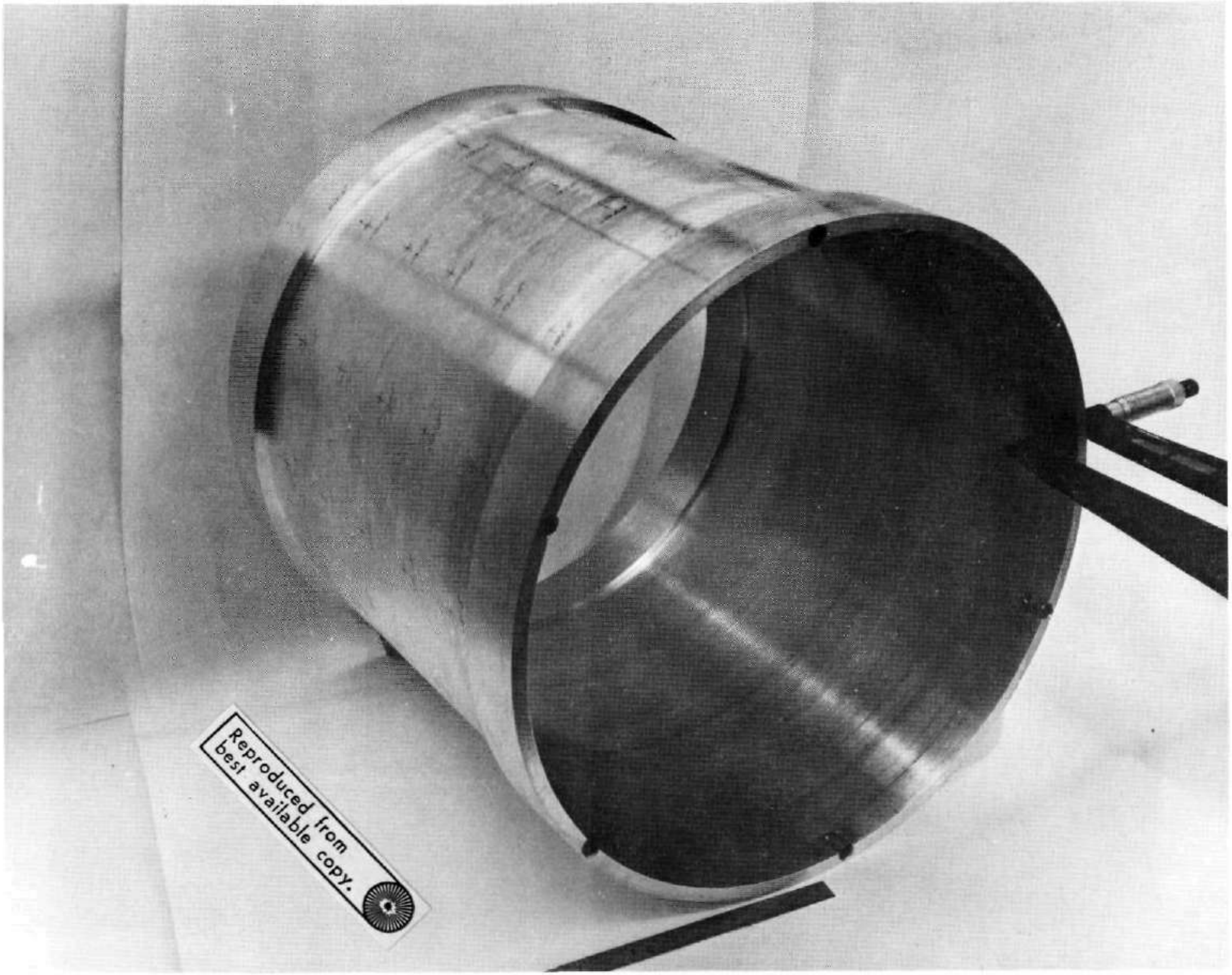


Fig. 2.3 Sheet Metal Micrometer Used in Thickness Mapping

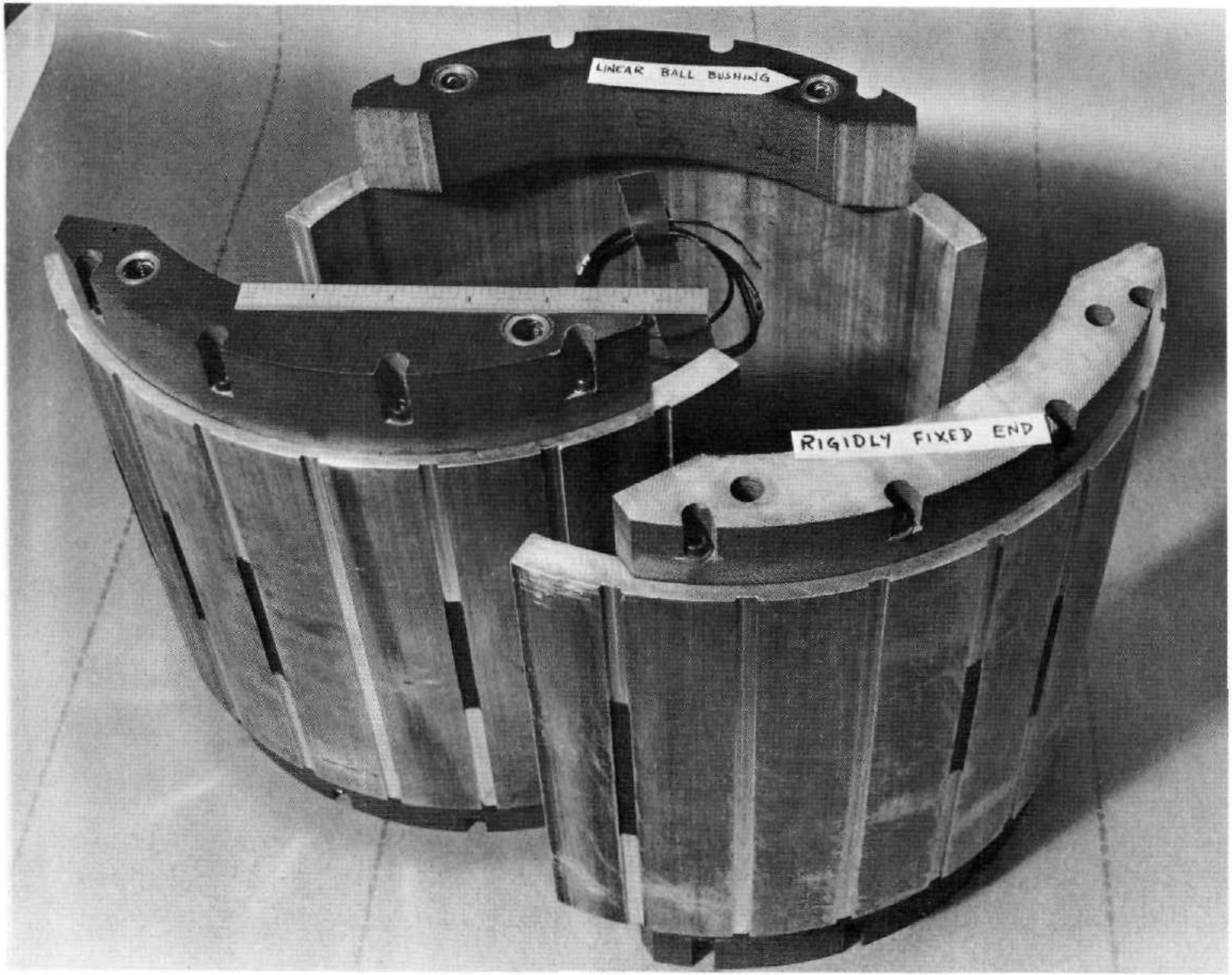


Fig. 2.4 "Buckle-Capture" Mandrel Segments Before Installation in Test Specimen

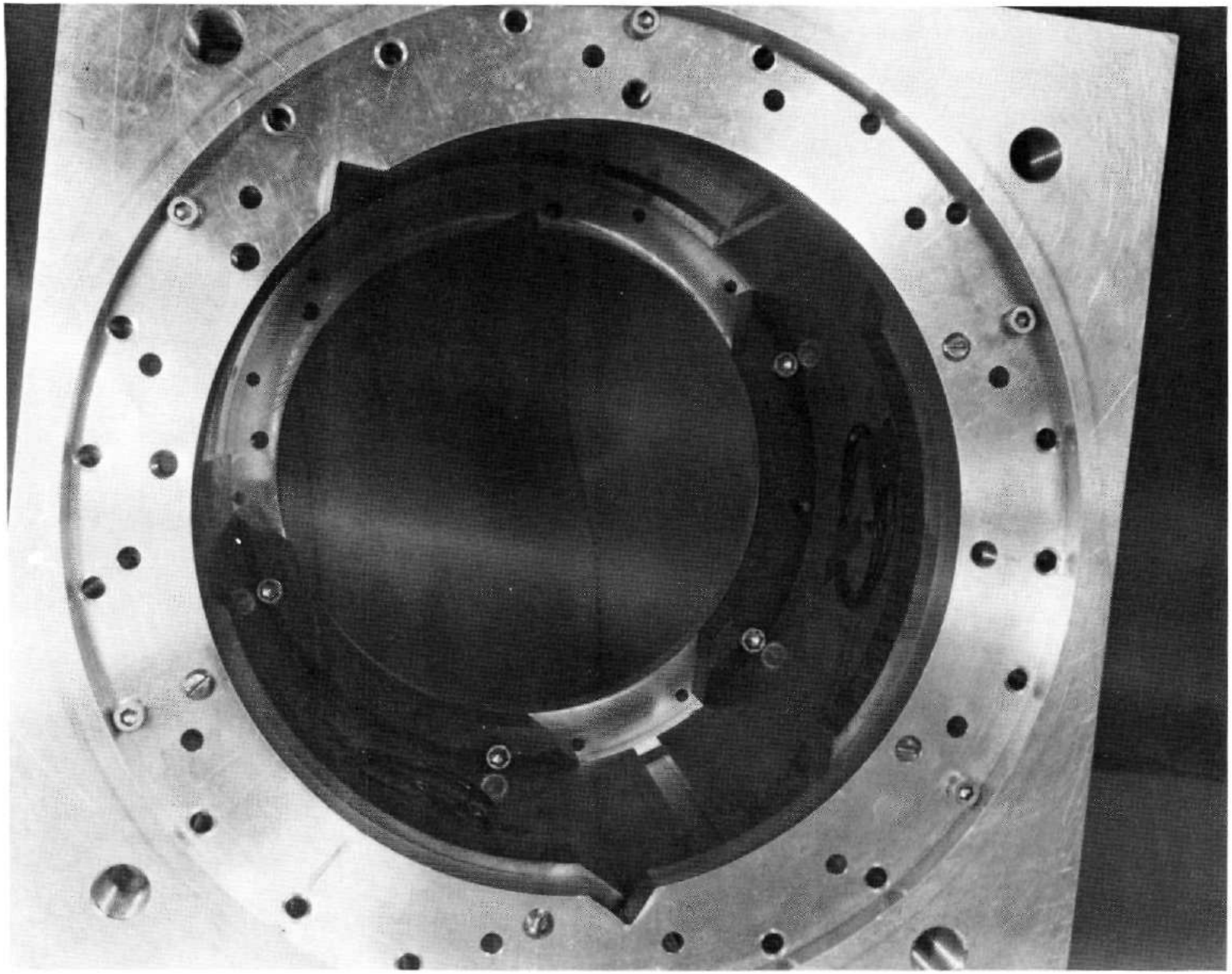
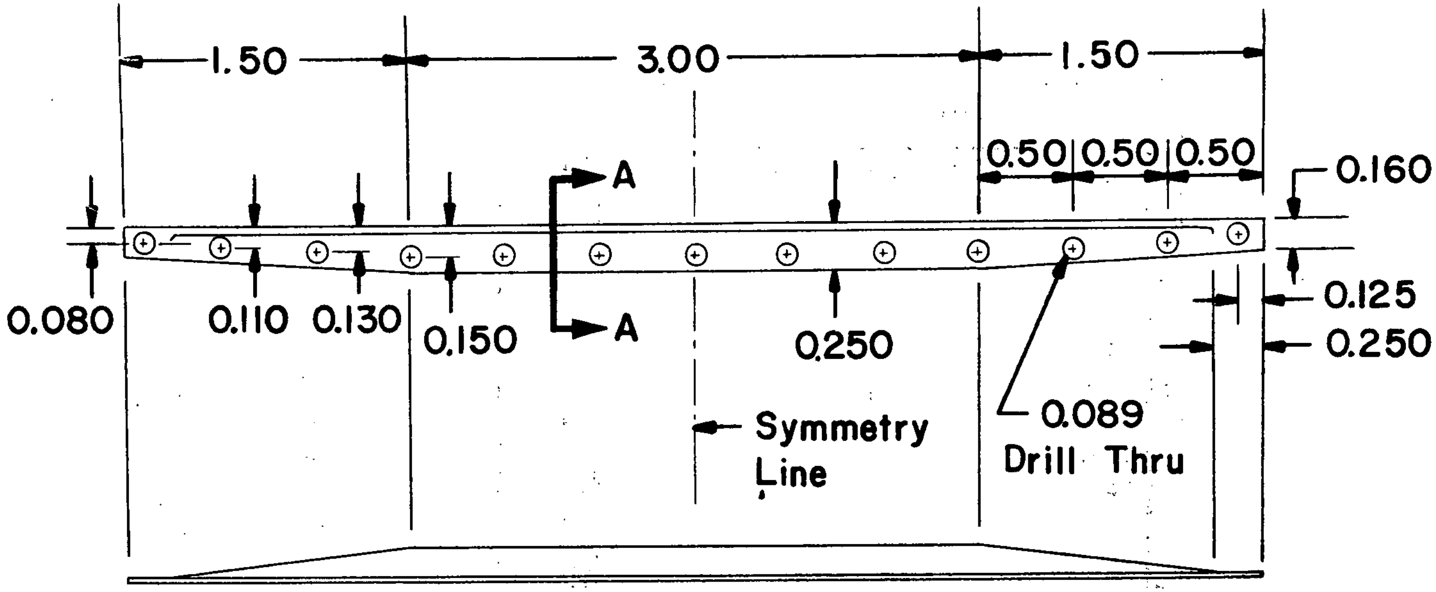
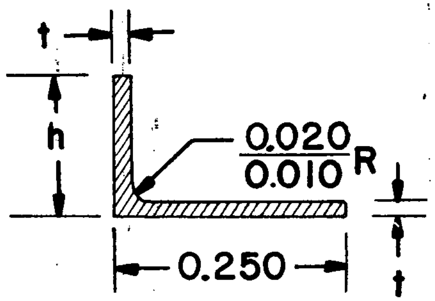


Fig. 2.5 "Buckle-Capture" Mandrel Segments Partially Installed in Test Specimen





SECTION AA

TYPE	t	h
A	0.020	0.150
B	0.010	0.150
C	0.010	0.080
P	0.020	0.150

Fig. 2.6 Geometry of Different Reinforcement Types

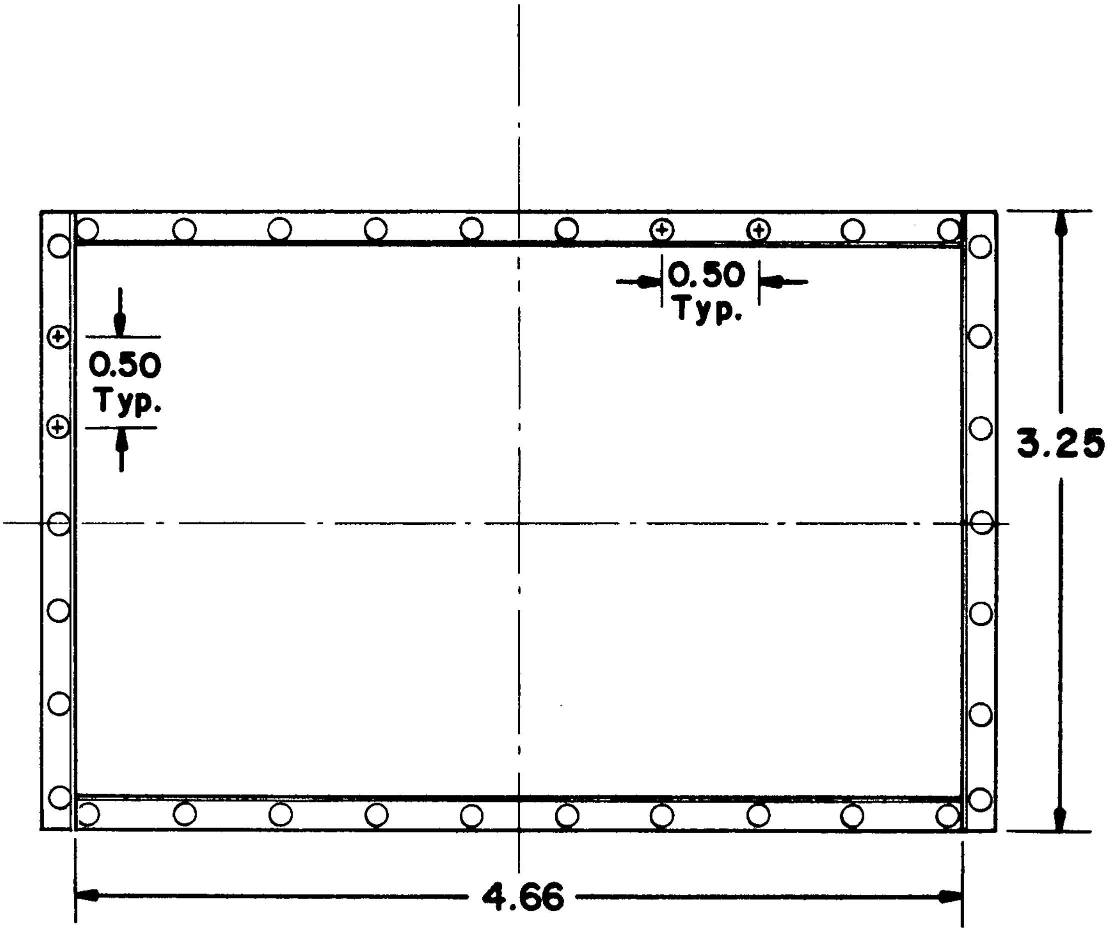


Fig. 2.7 Geometry of Type "P" Reinforcement

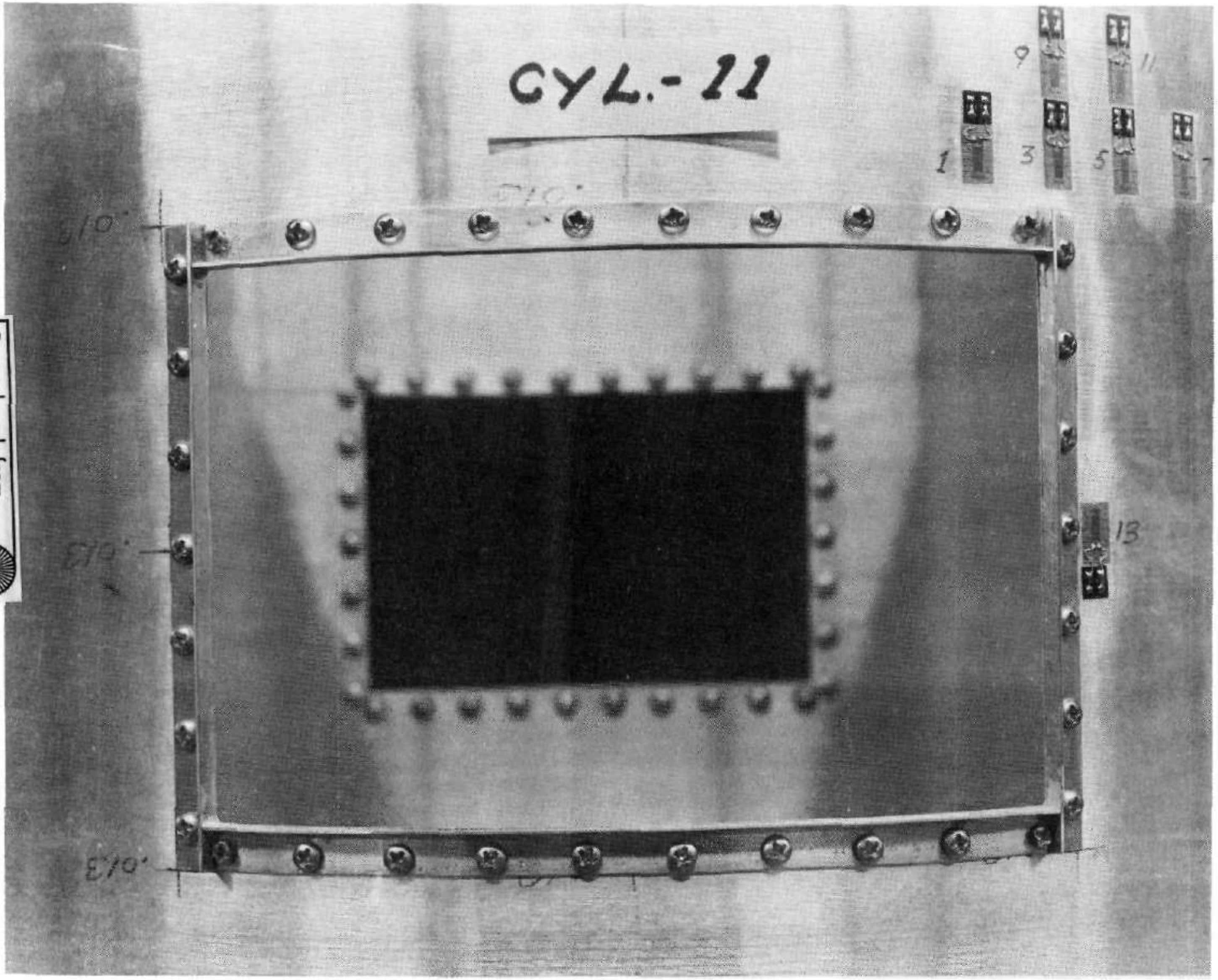


Fig. 2.8 Picture Frame Reinforcement on Cylinder #7

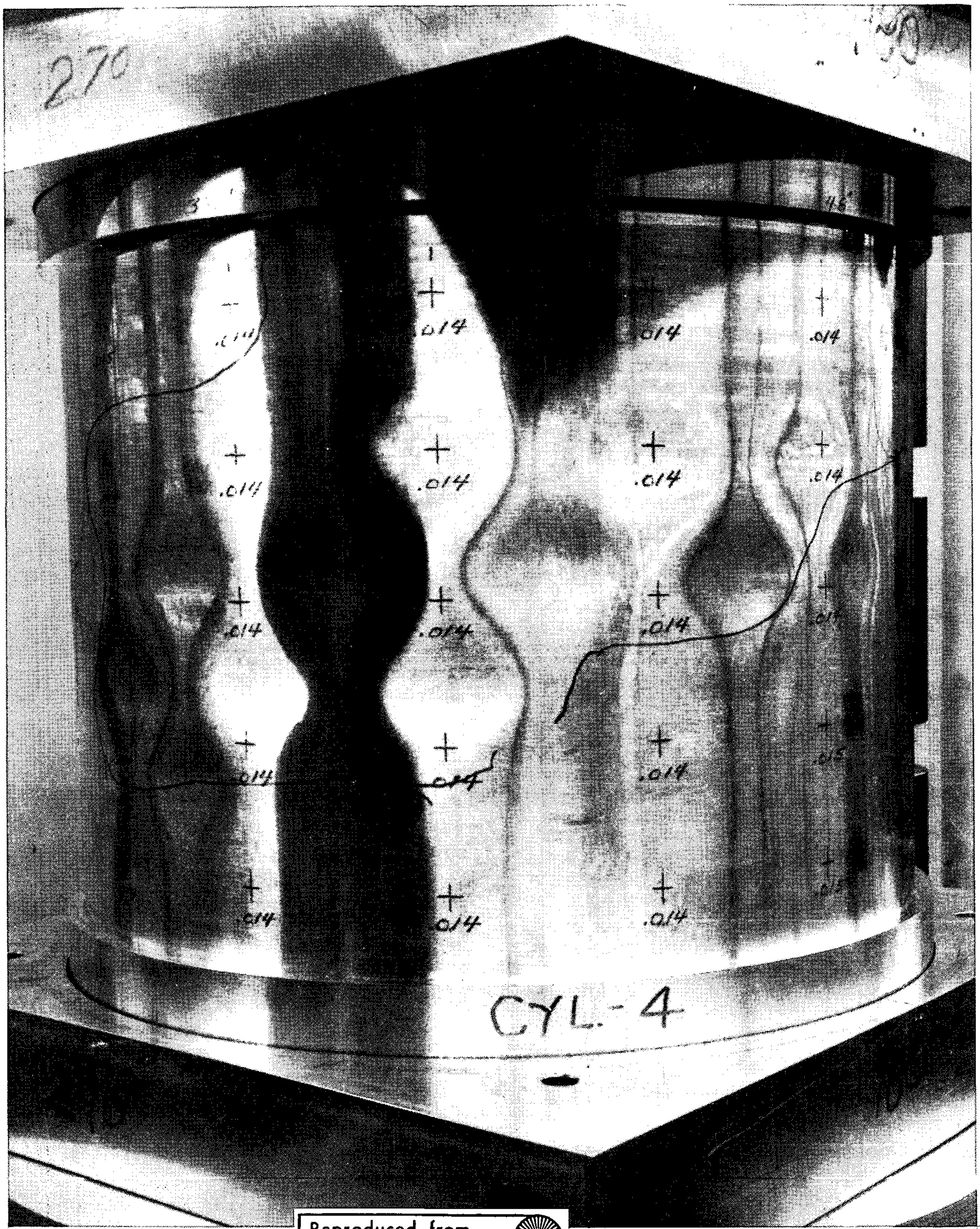


Fig. 2.9 | Buckle With Buckle With Buckle-Capture Mandrel In Place (Cylinder #1)

Section 3

TEST RESULTS AND EXPLANATORY COMMENTS

3.1 The Summary Tables

Tables 3.1 and 3.2 summarize all important test parameters and results for .014 and .009-inch thick cylinders, respectively (where these thicknesses are nominal rather than actual values).

Each summary table gives the following items:

The range of thicknesses measured on the cylinder, in mils. The first number is the minimum thickness, followed by a slash and the maximum thickness.

The average thickness in mils, based on the seventy-five measurements.

The classical buckling load (in lbs) based on the minimum thickness and equal to 0.6 E.t/R.

The first buckling load in lbs.

The first buckling load expressed as a percentage of the classical buckling load.

The "repeatable" buckling load (median value)

The range of the "repeatable" buckling load.

The "repeatable" buckling load expressed as a percentage of the classical buckling load.

The arc of the cutout, in degrees. (In every case, the height of the cutout was 3.00 inches.)

The type of reinforcement, if any. The various types are illustrated in Figs. 2.5 and 2.6.

The buckling load with the cutout, in lbs.

The number of strain gages used on that cylinder.

The "repeatable" buckling load expressed as a percent of classical, where the classical is based on the nominal thickness of 0.009 or 0.014 inches.

3.2 The Strain Gage Data Tables

Tables 3.3 through 3.13 are the strain gage data tabulations for the eleven cylinders. The reader is referred to subsection 2.7 for an extensive discussion of how this data was obtained and why some of the tables give the strains, and others the stress. Note also that a strain gage "station" means a set of back-to-back gages. The locations of the strain gage stations varies on each cylinder, and these locations are shown in Figs. 3.4 through 3.10.

Compressive strains (or stresses) are negative. A positive bending strain (or stress) means that the tension due to bending was on the outer face of the cylinder.

The solid lines in Figs. 3.1 through 3.3 represent the stress distribution in Cylinder #2, based on a computer run using the STAGS program. The points plotted are the actual stress measured on the cylinder by strain gages.

3.3 Photos of the Tested Cylinders

Fig. 3.11 and higher are photographs of the tested cylinders. The specimen numbers appearing on labels in the photographs should be disregarded, as they refer to a temporary numbering system used during the test program. The number appearing in the caption of the photograph is the pertinent number and agrees with the numbering in Tables 3.1 and 3.2.

TABLE 3.1

.014-INCH THICK CYLINDERS

Cylinder Number	1	2	3	4	5	6	7
Thickness range (mils)	14/16	14/15	12/14	12/16	12/14	12/15	13/15
Average thickness (mils)	14.76	14.68	12.81	14.64	13.27	13.67	13.73
Classical buckling load (lbs)	7389	7389	5430	5430	5430	5430	6370
First buckling load (lbs)	4450	4620	4500	3920	4180	4110	3075
FBL as percent of classical	60%	63%	83%	72%	77%	75%	48%
"Repeatable" buckling load (lbs)	4030	4585	4280	3735	3970	3360	3055
Range of "repeatable" load (lbs)	±70	±35	±170	±45	±150	±70	±15
RBL as percent of classical	55%	62%	79%	69%	73%	62%	41%
Arc of cutout (degrees)	30	45	45	45	45	45	45
Reinforcement type	None	None	None	A	A	B	P
Buckling load with cutout (lbs)	2740	2540	2050	3190	2850	2560	2600
Number of strain gages used	6	48	16	6	14	14	16
RBL as percent of nominal "t" classical buckling**	55%	62%	58%	50%	54%	46%	41%

**For cylinders in this table with a nominal thickness of .014 inches, the classical load is 7389 lbs.

TABLE 3.2

.009-INCH THICK CYLINDERS

Cylinder Number	8	9	10	11
Thickness range (mils)	9/11	8/11	9/11	9/11
Average thickness (mils	9.72	9.50	9.53	9.53
Classical buckling load (lbs)	3054	2413	3054	3054
First buckling load (lbs)	1340	1480	1390	1590
FBL as percent of classical	44%	61%	46%	52%
"Repeatable" buckling load (lbs)	1265	1435	1375	1555
Range of "repeatable" load (lbs)	±35	±20	±15	±35
RBL as percent of classical	41%	47%	45%	51%
Arc of cutout (degrees)	45	45	45	45
Reinforcement type	None	B	B*	C
Buckling load with cutout (lbs)	807	1275	1030	1055
Number of strain gages used	20	14	14	14
RBL as percent of nominal "t" classical buckling**			45%	

*Reinforcement on inside of cylinder

**See Table 3.1

TABLE 3.3

CYLINDER #1 .014 WALL, 30-DEGREE CUTOUT, NO REINFORCEMENT

S T A T I O N 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
200	-29	-14
394	-65	-30
595	-93	-60
796	-96	-107
1190	-69	-212
1385	-50	-266
1590	-27	-318
1797	2	-367
1997	32	-415
2187	63	-459
2397	102	-505
2627	146	-549
1217	-752	63

S T A T I O N 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
200	-58	16
394	-118	48
595	-181	139
796	-219	318
1190	-235	819
1385	-234	1111
1590	-227	1402
1797	-216	1687
1997	-206	1970
2187	-195	2247
2397	-183	2538
2627	-170	2844
1217	49	-3043

TABLE 3.3 - Concluded

CYLINDER #1 .014 WALL, 30-DEGREE CUTOUT, NO REINFORCEMENT

STATION 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
200	-42	-2
394	-84	-4
595	-130	-8
796	-183	-15
1190	-295	-32
1385	-352	-37
1590	-409	-38
1797	-462	-37
1997	-513	-32
2187	-561	-23
2397	-609	-9
2627	-652	18
1217	39	-3032

TABLE 3.4

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 1

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-508	105	29	6
402	-1018	99	135	-99
601	-1542	187	107	-150
797	-2041	227	165	-190
1004	-2625	278	197	-278
1199	-3076	366	243	-329
1400	-3587	434	348	-360
1598	-4171	542	380	-431
1802	-4726	621	421	-511
2002	-5254	746	470	-525
2113	-5568	811	453	-480
2217	-5886	896	513	-454
2312	-6218	1007	440	-87
2421	-6700	984	296	267
2526	-7233	785	241	466
263	-16954	4119	-15025	3572

STATION 2

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-380	0	-62	0
402	-791	-48	-55	11
601	-1267	-105	-94	-6
797	-1772	-116	-142	43
1004	-2240	-125	-152	14
1199	-2710	-136	-86	63
1400	-3175	-221	-173	37
1598	-3672	-224	-193	114
1802	-4216	-252	-200	105
2002	-4675	-309	-182	88
2113	-4990	-301	-199	117
2217	-5248	-349	-199	128
2312	-5639	-394	-212	63
2421	-6002	-468	-217	-11
2526	-6441	-578	-219	-121
263	221	-4704	993	-14543

TABLE 3.4 - Continued

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 3

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-306	74	12	74
402	-499	153	57	-6
601	-680	198	55	59
797	-845	210	109	11
1004	-1004	295	10	36
1199	-1120	315	53	17
1400	-1256	371	115	33
1598	-1386	411	25	-6
1802	-1454	439	56	2
2002	-1551	513	79	76
2113	-1551	496	79	19
2217	-1627	533	82	56
2312	-1454	496	56	19
2421	-1304	476	127	39
2526	-1111	476	81	39
263	1885	-337	175	-178

STATION 4

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-442	193	74	-46
402	-919	323	35	45
601	-1398	473	73	115
797	-1877	703	111	107
1004	-2401	865	83	129
1199	-2929	1063	132	188
1400	-3433	1262	85	247
1598	-3981	1497	153	343
1802	-4565	1735	185	363
2002	-5129	1953	198	402
2113	-5438	2087	286	416
2217	-5761	2217	241	506
2312	-6099	2379	321	529
2421	-6396	2484	361	534
2526	-6694	2628	401	500
263	-7122	2353	351	444

TABLE 3.4 - Continued

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 5

	AXIAL DIRECTION		CIRCUMF. DIRECTION	
AXIAL LOAD POUNDS	MEMBRANE STRESS PSI	BENDING STRESS PSI	MEMBRANE STRESS PSI	BENDING STRESS PSI
195	-315	179	-17	79
402	-734	349	-39	130
601	-1194	485	-21	67
797	-1667	675	-136	98
1004	-2152	819	-203	64
1199	-2574	1009	-149	95
1400	-3042	1188	-160	174
1598	-3499	1361	-218	148
1802	-3986	1559	-209	207
2002	-4443	1732	-268	181
2113	-4664	1817	-231	207
2217	-4922	1962	-230	173
2312	-5151	2055	-221	226
2421	-5390	2180	-241	212
2526	!-5659	2296	-192	169
263	3847	77	90	-261

STATION 6

	AXIAL DIRECTION		CIRCUMF. DIRECTION	
AXIAL LOAD POUNDS	MEMBRANE STRESS PSI	BENDING STRESS PSI	MEMBRANE STRESS PSI	BENDING STRESS PSI
195	-1100	996	0	0
402	-1991	2410	0	0
601	-2830	4402	0	0
797	-3432	6838	0	0
1004	-3904	9563	0	0
1199	-4244	12209	0	0
1400	-4480	14803	0	0
1598	-4664	17449	0	0
1802	-4821	20069	0	0
2002	-4847	22453	0	0
2113	-4847	23606	0	0
2217	-4768	24418	0	0
2312	-3773	20750	0	0
2421	-2908	17790	0	0
2526	-2148	15772	0	0
263	4140	-19021	0	0

TABLE 3.4 - Continued

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 7

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-707	655	0	0
402	-1179	1598	0	0
601	-1546	2908	0	0
797	-1782	4506	0	0
1004	-1939	6288	0	0
1199	-2096	8070	0	0
1400	-2306	9799	0	0
1598	-2463	11633	0	0
1802	-2594	13493	0	0
2002	-2699	15222	0	0
2113	-2725	16139	0	0
2217	-2599	16847	0	0
2312	-2358	15091	0	0
2421	-2227	13703	0	0
2526	-2279	12864	0	0
263	-1965	-15537	0	0

STATION 8

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-550	393	0	0
402	-917	917	0	0
601	-1205	1624	0	0
797	-1493	2541	0	0
1004	-1729	3563	0	0
1199	-2017	4637	0	0
1400	-2279	5738	0	0
1598	-2568	6917	0	0
1802	-2777	8174	0	0
2002	-2987	9484	0	0
2113	-3039	10166	0	0
2217	-3065	10821	0	0
2312	-2489	10349	0	0
2421	-2279	10139	0	0
2526	-2306	10323	0	0
263	-2044	-12943	0	0

TABLE 3.4 - Continued

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 9

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-354	48	23	-11
402	-812	-8	40	-28
601	-1254	48	114	-11
797	-1798	20	106	-20
1004	-2304	11	135	-48
1199	-2754	20	180	-20
1400	-3270	3	181	-76
1598	-3756	31	190	-68
1802	-4275	51	267	-88
2002	-4733	51	284	-88
2113	-4974	43	341	-116
2217	-5296	26	296	-173
2312	-5565	6	344	-153
2421	-5843	17	364	-201
2526	-6120	45	384	-193
263	-8404	33540	-531	72982

STATION 10

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	57	0	17	0
402	93	-20	54	20
601	105	8	6	28
797	76	-20	-3	20
1004	76	-20	-3	20
1199	122	-48	62	11
1400	133	-76	14	3
1598	142	-68	42	31
1802	142	-68	42	31
2002	142	-68	42	31
2113	170	-96	51	23
2217	161	-88	23	51
2312	170	-96	51	23
2421	207	-116	88	43
2526	178	-144	79	34
263	190	99	31	-99

TABLE 3.4 - Concluded

CYLINDER #2 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 11

AXIAL LOAD POUNDS	AXIAL DIRECTION MEMBRANE STRESS PSI	AXIAL DIRECTION BENDING STRESS PSI	CIRCUMF. DIRECTION MEMBRANE STRESS PSI	CIRCUMF. DIRECTION BENDING STRESS PSI
195	-314	65	-17	45
402	-829	178	-16	79
601	-1353	340	-44	102
797	-1848	416	-64	99
1004	-2335	597	-55	101
1199	-2805	739	11	144
1400	-3272	900	-	166
1598	-3787	1070		217
1802	-4246	1223	17	211
2002	-4733	1421	26	271
2113	-5030	1526	66	276
2217	-5279	1639	95	310
2312	-5593	1743	78	316
2421	-5879	1820	70	313
2526	-6205	1953	101	327
263	-9152	2618	69	655

TABLE 3.5

CYLINDER #3 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

S T A T I O N 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
198	-78	8
395	-178	43
601	-333	268
791	-430	640
995	-480	1005
1192	-508	1338
1392	-525	1655
1590	-535	1950
1686	-533	2093
1787	-533	2243
1886	-530	2385
1984	-518	2533
2093	-90	1050
2072	-105	990
970	480	-2640

S T A T I O N 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
198	-50	0
396	-98	3
601	-158	3
791	-203	8
995	-253	13
1192	-308	18
1392	-363	18
1590	-413	23
1686	-435	25
1787	-463	28
1886	-493	33
1984	-513	43
2093	-670	-60
2072	-645	-40
970	-108	-4188

TABLE 3.5 - Continued

CYLINDER #3 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

S T A T I O N 3

=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
198	-48	-3
396	-100	0
601	-153	-3
791	-200	-5
995	-255	-5
1192	-308	-3
1392	-360	-5
1590	-413	-8
1686	-440	-5
1787	-470	-5
1886	-498	-8
1984	-528	-8
2093	-623	-23
2072	-608	-23
970	-145	-3035

S T A T I O N 4

=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
198	-40	0
396	-85	0
601	-135	0
791	-180	0
995	-230	0
1192	-280	-5
1392	-330	-5
1590	-383	-8
1686	-408	-8
1787	-438	-8
1886	-463	-8
1984	-490	-10
2093	-535	-10
2072	-525	-10
970	-1365	550

TABLE 3.5 - Continued

CYLINDER #3 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 5

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
198	-23	8
396	-43	18
601	-65	20
791	-75	25
995	-80	30
1192	-93	38
1392	-100	40
1590	-105	45
1686	-110	45
1787	-108	48
1886	-113	48
1984	-110	50
2093	-15	40
2072	-25	40
970	5	-35

STATION 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
198	-63	-3
396	-128	-8
601	-208	-3
791	-300	25
995	-443	228
1192	-545	620
1392	-585	970
1590	-605	1275
1686	-610	1410
1787	-613	1553
1886	-615	1685
1984	-615	1815
2093	-608	2143
2072	-610	2110
970	188	-2893

TABLE 3.5 - Concluded

CYLINDER #3 .014 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
198	-65	0
396	-145	5
601	-243	23
791	-360	130
995	-503	518
1192	-583	938
1392	-628	1323
1590	-655	1665
1686	-663	1823
1787	-670	1990
1886	-673	2138
1984	-680	2285
2093	-678	2313
2072	-673	2288
970	200	-25

STATION 8

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
198	-95	30
396	-228	158
601	-348	478
791	-400	760
995	-438	1048
1192	-463	1318
1392	-480	1580
1590	-495	1830
1686	-500	1945
1787	-503	2073
1886	-505	2185
1984	-508	2293
2093	-500	2310
2072	-505	2305
970	170	-35

TABLE 3.6

CYLINDER #4 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

STATION 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
200	-28	-3
400	-68	-3
580	-113	-3
800	-158	-3
1000	-200	0
1180	-240	0
1410	-288	+3
1610	-328	+3
1800	-365	+10
2000	-405	+10
2180	-445	+15
2390	-488	+18
2590	-525	+25
2790	-568	+33
3000	-600	+45
3190	-640	+55

STATION 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
200	-28	-8
400	-75	0
580	-113	+3
800	-160	+5
1000	-205	+10
1180	-245	+15
1410	-290	+20
1610	-335	+30
1800	-368	+33
2000	-410	+40
2180	-448	+48
2390	-480	+55
2590	-520	+65
2790	-555	+75
3000	-585	+85
3190	-613	+103

TABLE 3.6 - Concluded

CYLINDER #4 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

S T A T I O N 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
200	-40	+5
400	-78	+13
580	-128	+3
800	-173	+3
1000	-220	0
1180	-263	-3
1410	-310	-10
1610	-358	-13
1800	-395	-10
2000	-440	-15
2180	-485	-15
2390	-530	-20
2590	-578	-23
2790	-633	-33
3000	-670	-30
3190	-710	-40

TABLE 3.7

CYLINDER #5 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

STATION 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-48	8
415	-103	13
602	-163	23
822	-218	33
1010	-278	43
1189	-328	58
1413	-385	80
1611	-440	110
1807	-488	143
2019	-538	198
2110	-563	233
2205	-588	273
2303	-603	323
2416	-618	383
2522	-625	470
2607	-623	583
2698	-578	743
2808	-483	928
1039	-18	-118

STATION 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-33	-8
415	-73	-3
602	-118	-13
822	-165	-15
1010	-210	-20
1189	-255	-25
1413	-303	-33
1611	-353	-38
1807	-403	-48
2019	-450	-55
2110	-475	-55
2205	-498	-58
2303	-523	-63
2416	-548	-63
2522	-570	-65
2607	-595	-65
2698	-615	-60
2808	-645	-60
1039	-668	-48

TABLE 3.7 - Continued

CYLINDER #5 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

STATION 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-53	-3
415	-95	5
602	-153	8
822	-203	13
1010	-255	15
1189	-305	20
1413	-358	28
1611	-403	33
1807	-453	43
2019	-493	58
2110	-523	63
2205	-543	68
2303	-568	78
2416	-588	88
2522	-605	100
2607	-623	108
2698	-633	113
2808	-608	93
1039	-203	473

STATION 4

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-58	-3
415	-108	-8
602	-158	-8
822	-213	-8
1010	-263	-13
1189	-313	-13
1413	-363	-13
1611	-410	-10
1807	-460	-5
2019	-503	3
2110	-528	8
2205	-550	15
2303	-575	25
2416	-595	30
2522	-620	45
2607	-643	63
2698	-663	88
2808	-695	160
1039	-48	1138

TABLE 3.7 - Continued

CYLINDER #5 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

S T A T I O N 5

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-50	5
415	-93	8
602	-148	13
822	-195	20
1010	-243	23
1189	-288	28
1413	-330	35
1611	-373	43
1807	-413	53
2019	-453	68
2110	-475	75
2205	-493	78
2303	-510	85
2416	-525	95
2522	-540	100
2607	-550	110
2698	-545	120
2808	-488	138
1039	-1023	1658

S T A T I O N 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-45	0
415	-85	0
602	-133	3
822	-180	5
1010	-225	5
1189	-270	5
1413	-315	10
1611	-355	15
1807	-398	13
2019	-435	20
2110	-458	23
2205	-478	28
2303	-498	28
2416	-513	33
2522	-533	33
2607	-550	40
2698	-565	45
2808	-585	55
1039	620	615

TABLE 3.7 - Concluded

CYLINDER #5 .014 WALL, 45-DEGREE CUTOUT, TYPE A REINFORCEMENT

STATION 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
212	-55	0
415	-100	0
602	-150	0
822	-198	3
1010	-253	3
1189	-298	3
1413	-348	3
1611	-390	0
1807	-435	0
2019	-483	3
2110	-508	3
2205	-533	3
2303	-560	0
2416	-580	0
2522	-605	0
2607	-630	0
2698	-655	-5
2808	-690	-5
1039	293	723

TABLE 3.8

CYLINDER #6 .014 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT

STATION 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-38	3
507	-83	8
754	-125	10
1003	-168	18
1254	-215	20
1502	-260	25
1753	-303	33
2009	-353	43
2202	-383	48
2410	-420	55
1159	113	3

STATION 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-28	3
507	-63	8
754	-100	10
1003	-140	15
1254	-183	23
1502	-220	25
1753	-268	33
2009	-303	38
2202	-340	40
2410	-373	48
1159	218	-3

TABLE 3.8 - Continued

CYLINDER #6 .014 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT

S T A T I O N 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-40	0
507	-85	0
754	-130	5
1003	-170	5
1254	-215	10
1502	-258	13
1753	-300	20
2009	-343	23
2202	-378	28
2410	-413	33
1159	-15	5

S T A T I O N 4

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-60	5
507	-118	3
754	-178	8
1003	-230	10
1254	-290	10
1502	-343	18
1753	-395	25
2009	-445	30
2202	-490	35
2410	-528	48
1159	13	-3

TABLE 3.8 - Continued

CYLINDER #6 .014 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT

S T A T I O N 5

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-50	-5
507	-105	0
754	-153	3
1003	-203	3
1254	-258	8
1502	-305	10
1753	-353	13
2009	-400	20
2202	-440	25
2410	-475	30
1159	-8	-8

S T A T I O N 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
258	-55	0
507	-108	3
754	-170	0
1003	-225	5
1254	-283	8
1502	-338	13
1753	-390	20
2009	-445	30
2202	-483	38
2410	-520	45
1159	125	-20

TABLE 3.8 - Concluded

CYLINDER #6 .014 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT

S T A T I O N 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
258	-50	0
507	-100	5
754	-155	10
1003	-208	13
1254	-263	13
1502	-318	18
1753	-373	23
2009	-430	25
2202	-478	28
2410	-528	33
1159	-928	1333

TABLE 3.9

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

STATION 1

================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
203	-15	-10
407	-28	-23
617	-40	-35
808	-58	-48
918	-60	-55
1005	-68	-63
1108	-78	-68
1209	-83	-78
1310	-88	-83
1415	-93	-88
1513	-103	-98
1611	-108	-103
1716	-113	-113
1811	-118	-123
1912	-120	-135
2025	-128	-143
2119	-130	-155
2212	-138	-163
2325	-145	-175
2424	-148	-183
2616	-155	-205
1225	-260	695

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

STATION 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
203	-65	-15
407	-135	-30
617	-205	-50
808	-270	-65
918	-300	-75
1005	-328	-83
1108	-360	-95
1209	-393	-108
1310	-423	-118
1415	-455	-130
1513	-480	-145
1611	-510	-155
1716	-538	-173
1811	-565	-190
1912	-593	-208
2025	-623	-223
2119	-648	-248
2212	-670	-265
2325	-700	-290
2424	-723	-313
2616	-770	-370
1225	-555	1545

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

S T A T I O N 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
203	-58	3
407	-118	8
617	-180	15
808	-235	20
918	-263	23
1005	-285	30
1108	-315	35
1209	-348	38
1310	-373	38
1415	-398	43
1513	-430	50
1611	-453	53
1716	-483	58
1811	-510	60
1912	-535	65
2025	-568	73
2119	-590	75
2212	-620	80
2325	-650	85
2424	-678	93
2516	-733	98
1225	8	1123

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

STATION 4

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
203	-55	10
407	-110	20
617	-168	33
808	-223	43
918	-248	48
1005	-275	55
1108	-303	58
1209	-328	68
1310	-353	73
1415	-383	83
1513	-413	88
1611	-438	98
1716	-465	105
1811	-490	110
1912	-518	123
2025	-548	133
2119	-573	138
2212	-600	145
2325	-630	155
2424	-655	170
2616	-713	193
1225	-845	1940

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

S T A T I O N 5

=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O	S T R A I N
203	-60	-5
407	-118	-8
617	-178	-13
808	-235	-15
918	-265	-15
1005	-290	-15
1108	-323	-18
1209	-350	-15
1310	-378	-18
1415	-405	-20
1513	-435	-20
1611	-463	-23
1716	-490	-20
1811	-515	-25
1912	-545	-20
2025	-570	-20
2119	-598	-18
2212	-623	-18
2325	-655	-15
2424	-680	-15
2616	-730	-10
1225	-285	-855

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

S T A T I O N 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O	S T R A I N
203	-58	-8
407	-128	-18
617	-193	-28
808	-255	-35
918	-290	-40
1005	-318	-48
1108	-350	-50
1209	-383	-58
1310	-418	-63
1415	-448	-68
1513	-480	-75
1611	-513	-83
1716	-545	-90
1811	-575	-95
1912	-610	-105
2025	-645	-115
2119	-675	-120
2212	-703	-128
2325	-743	-138
2424	-773	-148
2616	-840	-165
1225	75	-1235

TABLE 3.9 - Continued

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

STATION 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
203	-70	0
407	-133	8
617	-193	8
808	-248	13
918	-280	15
1005	-308	18
1108	-340	20
1209	-368	23
1310	-395	25
1415	-420	30
1513	-455	30
1611	-480	35
1716	-505	40
1811	-535	45
1912	-560	50
2025	-588	53
2119	-615	55
2212	-645	60
2325	-675	65
2424	-700	70
2616	-753	88
1225	-360	395

TABLE 3.9 - Concluded

CYLINDER #7 .014 WALL, 45-DEGREE CUTOUT, TYPE P REINFORCEMENT

S T A T I O N 8

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
203	-43	3
407	-93	8
617	-140	15
808	-188	18
918	-213	23
1005	-233	28
1108	-258	28
1209	-285	30
1310	-305	35
1415	-330	40
1513	-358	43
1611	-383	48
1716	-408	53
1811	-433	58
1912	-458	63
2025	-485	70
2119	-510	75
2212	-535	80
2325	-568	88
2424	-593	98
2616	-645	115
1225	-265	1215

TABLE 3.10

CYLINDER #8 .009 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 1

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-472	367
102	-734	681
157	-1022	1127
203	-1336	1703
258	-1598	2489
308	-1834	3354
356	-1965	4218
405	-2096	5135
455	-2201	6131
507	-2279	7205
553	-2332	8096
603	-2332	9144
652	-2358	10218
709	-2332	12759
755	-2227	14489

STATION 2

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-210	262
102	-393	498
157	-550	812
203	-655	1231
258	-760	1755
308	-812	2332
256	-812	2961
405	-838	3563
455	-838	4244
507	-865	4952
553	-891	5554
603	-838	6288
652	-865	6995
709	-838	8856
755	-707	10192

TABLE 3.10 - Continued

CYLINDER #8   .009 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 3

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-183	183
102	-288	288
157	-419	472
203	-524	681
258	-629	943
308	-707	1231
356	-786	1572
405	-838	1939
455	-943	2306
507	-1048	2725
553	-1127	3065
603	-1179	3485
652	-1231	3956
709	-1362	5188
755	-1310	6131

STATION 4

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-183	79
102	-341	131
157	-524	210
203	-681	262
258	-838	419
308	-1022	550
356	-1153	681
405	-1310	786
455	-1493	969
507	-1651	1179
553	-1808	1336
603	-1965	1546
652	-2096	1782
709	-2384	2437
755	-2410	3039

TABLE 3.10 - Continued

CYLINDER #8 .009 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 5

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-576	734
102	-917	1389
157	-1284	2227
203	-1546	3118
258	-1886	4140
308	-2122	5161
356	-2306	6131
405	-2489	7100
455	-2672	8174
507	-2856	9353
553	-2987	10323
603	-3118	11554
652	-3196	13310
709	-2803	17213
755	-2856	17161

STATION 6

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-210	524
102	-288	969
157	-419	1572
203	-472	2148
258	-524	2882
308	-576	3563
356	-629	4244
405	-707	4952
455	-760	5685
507	-838	6498
553	-865	7205
603	-838	8122
652	-786	9380
709	-891	13572
755	-1022	13650

TABLE 3.10 - Continued

CYLINDER #8 .009 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 7

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-131	288
102	-157	524
157	-262	891
203	-367	1258
258	-419	1677
308	-524	2096
356	-603	2489
405	-655	2961
455	-734	3406
507	-838	3982
553	-838	4454
603	-838	5135
652	-681	6078
709	-288	11030
755	-393	11083

STATION 8

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-157	105
102	-288	236
157	-367	367
203	-550	550
258	-734	681
308	-917	917
356	-1100	1100
405	-1205	1362
455	-1336	1598
507	-1493	1913
553	-1572	2148
603	-1651	2541
652	-1546	3170
709	-550	8620
755	-524	8698

TABLE 3.10 - Concluded

CYLINDER #8 .009 WALL, 45-DEGREE CUTOUT, NO REINFORCEMENT

STATION 9

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-183	79
102	-367	52
157	-576	52
203	-786	52
258	-996	52
308	-1284	79
356	-1467	105
405	-1651	131
455	-1860	183
507	-2070	236
553	-2253	262
603	-2410	367
652	-2515	524
709	-183	7519
755	183	7781

STATION 10

LOAD POUNDS	AVERAGE STRESS PSI	BENDING STRESS PSI
58	-210	-52
102	-419	-52
157	-681	-105
203	-891	-105
258	-1127	-131
308	-1362	-210
356	-1624	-210
405	-1834	-262
455	-2096	-314
507	-2306	-367
553	-2541	-445
603	-2777	-524
652	-3039	-681
709	-812	5738
755	262	7441

TABLE 3.11

CYLINDER #9 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (EXTERNAL)

S T A T I O N 1

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
-102	-38	-3
-203	-68	-3
-299	-105	0
-398	-133	3
-500	-163	3
-601	-195	5
-697	-225	10
-789	-250	15
-895	-285	25
-1003	-318	33
-1093	-343	48
-1191	-373	58
-508	-88	968

S T A T I O N 2

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
-102	-35	-10
-203	-83	-13
-299	-125	-20
-398	-163	-28
-500	-198	-33
-601	-233	-38
-697	-268	-43
-789	-300	-45
-895	-338	-53
-1003	-375	-55
-1093	-408	-53
-1191	-445	-40
-508	143	653

TABLE 3.11 - Continued

CYLINDER #9 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (EXTERNAL)

STATION 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-102	-23	-3
-203	-58	-8
-299	-85	-10
-398	-113	-18
-500	-145	-20
-601	-173	-23
-697	-203	-28
-789	-228	-33
-895	-258	-38
-1003	-288	-43
-1093	-310	-50
-1191	-338	-53
-508	40	985

STATION 4

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-102	-33	-8
-203	-70	-20
-299	-103	-28
-398	-135	-35
-500	-163	-43
-601	-200	-55
-697	-230	-65
-789	-260	-75
-895	-298	-93
-1003	-333	-103
-1093	-363	-118
-1191	-393	-133
-508	-485	-155

TABLE 3.11 - Continued

CYLINDER #9 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (EXTERNAL)

STATION 5

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-102	-33	-3
-203	-65	0
-299	-100	0
-398	-135	0
-500	-165	0
-601	-195	0
-697	-228	3
-789	-250	5
-895	-280	5
-1003	-310	5
-1093	-335	10
-1191	-360	15
-508	-35	780

STATION 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-102	-35	0
-203	-70	0
-299	-108	3
-398	-138	3
-500	-170	5
-601	-198	8
-697	-230	10
-789	-253	8
-895	-285	10
-1003	-315	15
-1093	-343	18
-1191	-368	18
-508	-1003	-373

TABLE 3.11 - Concluded

CYLINDER #9 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (EXTERNAL)

STATION 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-102	-28	-3
-203	-60	0
-299	-90	5
-398	-120	5
-500	-155	10
-601	-193	18
-697	-230	20
-789	-258	28
-895	-295	35
-1003	-333	48
-1093	-368	58
-1191	-398	68
-508	50	-1400

TABLE 3.12

CYLINDER #10 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (INTERNAL)

S T A T I O N 1
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-100	-38	3
-208	-73	3
-305	-110	0
-403	-145	0
-508	-175	0
-600	-208	3
-700	-248	3
-808	-280	5
-905	-310	5
-1005	-343	8

S T A T I O N 2
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-100	-33	3
-208	-80	5
-305	-118	13
-403	-158	18
-508	-193	23
-600	-228	28
-700	-258	33
-808	-298	38
-905	-330	45
-1005	-365	55

TABLE 3.12 - Continued

CYLINDER #10 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (INTERNAL)

S T A T I O N 3

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-100	-20	-5
-208	-55	-5
-305	-88	-8
-403	-115	-10
-508	-148	-13
-600	-175	-10
-700	-205	-15
-808	-235	-15
-905	-260	-15
-1005	-288	-18

S T A T I O N 4

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-100	-33	-3
-208	-70	-5
-305	-108	-8
-403	-140	-10
-508	-173	-13
-600	-200	-15
-700	-233	-18
-808	-265	-25
-905	-295	-30
-1005	-323	-38

TABLE 3.12 - Continued

CYLINDER #10 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (INTERNAL)

S T A T I O N 5

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
-100	-40	0
-208	-80	5
-305	-123	8
-403	-163	13
-508	-205	15
-600	-245	20
-700	-290	25
-808	-333	28
-905	-378	33
-1005	-418	38

S T A T I O N 6

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
-100	-43	3
-208	-90	5
-305	-130	10
-403	-175	15
-508	-213	18
-600	-250	20
-700	-293	23
-808	-335	25
-905	-370	30
-1005	-413	33

TABLE 3.12 - Concluded

CYLINDER #10 .009 WALL, 45-DEGREE CUTOUT, TYPE B REINFORCEMENT (INTERNAL)

S T A T I O N 7

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
-100	-40	15
-208	-78	23
-305	-113	33
-403	-143	43
-508	-178	58
-600	-208	68
-700	-240	80
-808	-273	98
-905	-303	113
-1005	-335	135

TABLE 3.13

CYLINDER #11 .009 WALL, 45-DEGREE CUTOUT, TYPE C REINFORCEMENT

S T A T I O N 1
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
103	-30	-5
203	-70	-10
304	-110	-15
401	-145	-20
500	-185	-25
549	-203	-33
600	-218	-33
651	-240	-35
702	-260	-35
752	-275	-35
804	-295	-40
856	-313	-38
906	-328	-38
959	-345	-35
401	-360	-30
422	-360	-20
187	163	1013

S T A T I O N 2
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
103	-23	-3
203	-55	0
304	-90	0
401	-120	0
500	-153	3
549	-170	0
600	-180	0
651	-198	3
702	-210	5
752	-228	8
804	-245	5
856	-258	8
906	-268	8
959	-283	13
401	-295	15
422	-300	15
187	-210	665

TABLE 3.13 - Continued

CYLINDER #11 .009 WALL, 45-DEGREE CUTOUT, TYPE C REINFORCEMENT

S T A T I O N 3
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
103	-30	10
203	-65	20
304	-98	33
401	-130	50
500	-165	65
549	-180	75
600	-193	88
651	-210	95
702	-225	110
752	-240	130
804	-255	150
856	-273	173
906	-283	198
959	-293	238
401	-298	288
422	-283	398
187	150	690

S T A T I O N 4
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
103	-38	3
203	-70	5
304	-100	10
401	-128	13
500	-155	15
549	-168	18
600	-180	20
651	-193	28
702	-208	33
752	-215	35
804	-228	43
856	-233	48
906	-243	58
959	-250	70
401	-253	83
422	-248	103
187	-175	410

TABLE 3.13 - Continued

CYLINDER #11 .009 WALL, 45-DEGREE CUTOUT, TYPE C REINFORCEMENT

STATION 5
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
103	-33	-8
203	-68	-8
304	-100	-10
401	-133	-8
500	-158	-8
549	-173	-8
600	-185	-5
651	-200	-5
702	-210	-5
752	-220	0
804	-228	8
856	-235	15
906	-238	18
959	-243	28
401	-240	35
422	-233	43
187	-1148	1318

STATION 6
=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	MICROSTRAIN	
103	-33	8
203	-65	20
304	-95	30
401	-123	43
500	-153	53
549	-163	63
600	-175	65
651	-185	70
702	-195	80
752	-1520	1405
804	-213	93
856	-215	100
906	-220	105
959	-225	115
401	-228	123
422	-223	128
187	513	253

TABLE 3.13 - Concluded

CYLINDER #11 .009 WALL, 45-DEGREE CUTOUT, TYPE C REINFORCEMENT

S T A T I O N 7

=================

LOAD	AVERAGE STRAIN	BENDING STRAIN
POUNDS	M I C R O S T R A I N	
103	-35	0
203	-68	-3
304	-105	-5
401	-135	-5
500	-170	-5
549	-188	-8
600	-203	-3
651	-220	-10
702	-235	-10
752	-255	-5
804	-270	-10
856	-288	-8
906	-303	-8
959	-318	-8
401	-333	-8
422	-343	-8
187	-188	1963

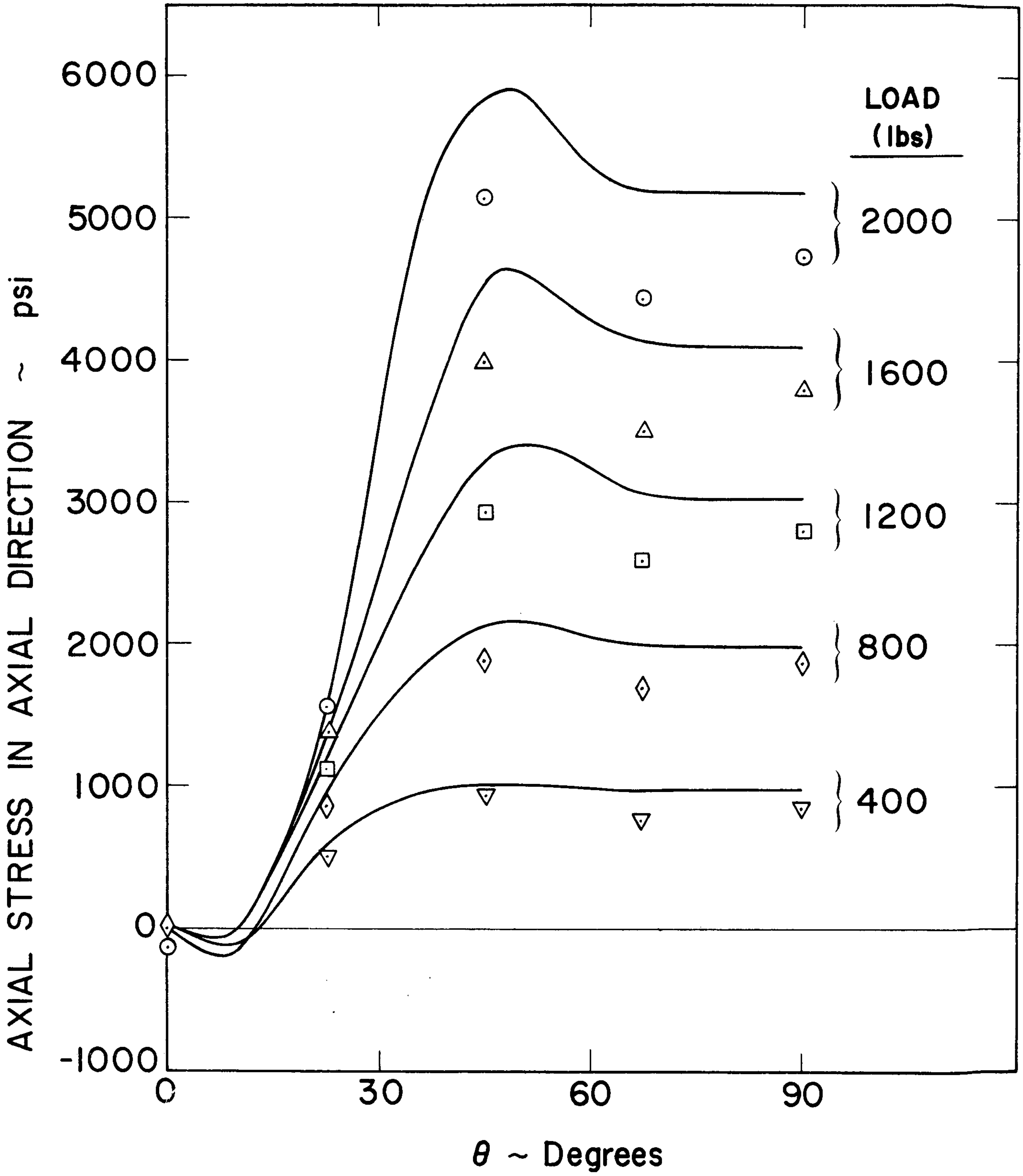


Fig. 3.1 Axial Stress 0.30 Inches From End Ring (Cylinder #2)

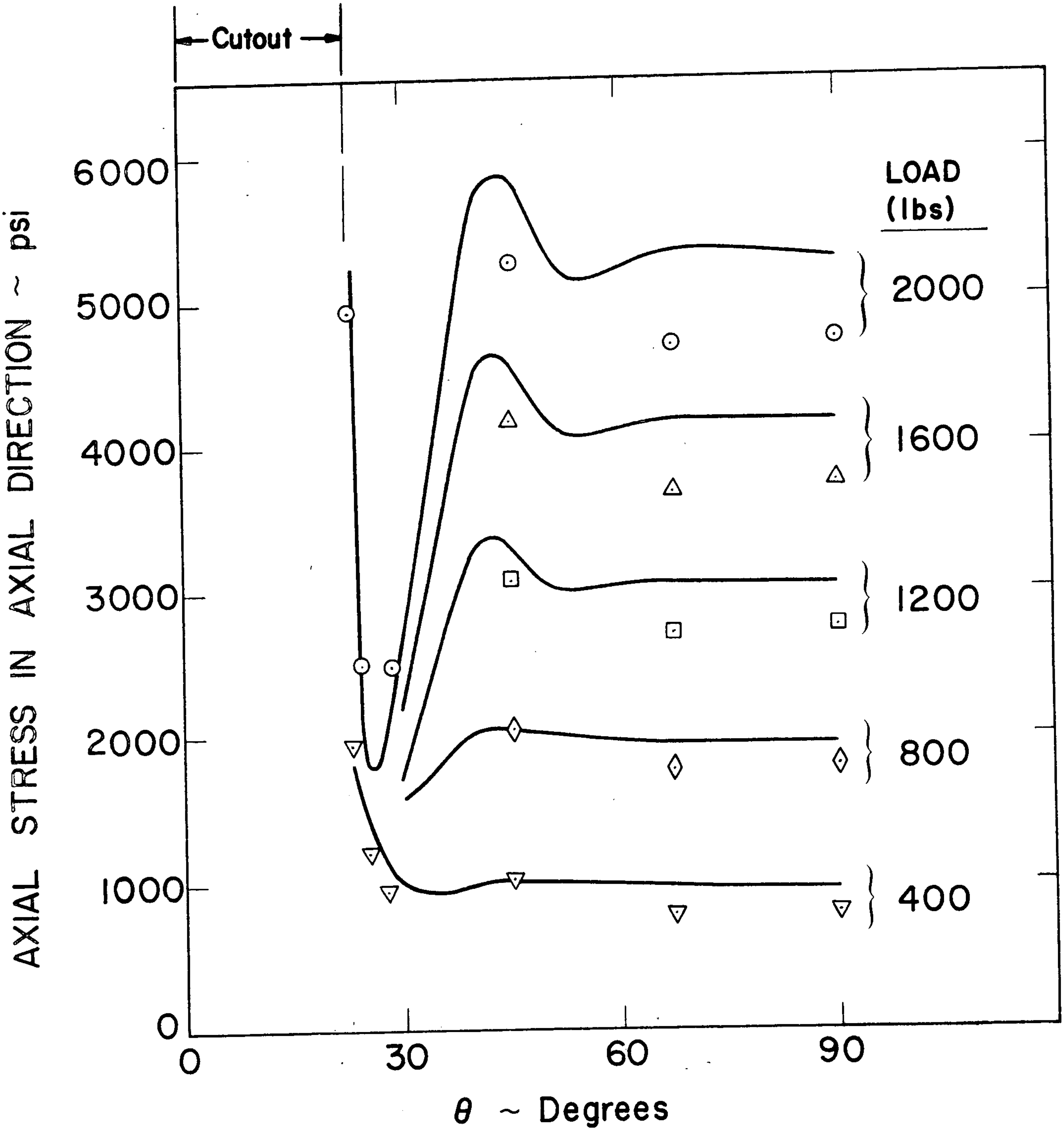


Fig. 3.2 Axial Stress at Cylinder #2 Midheight

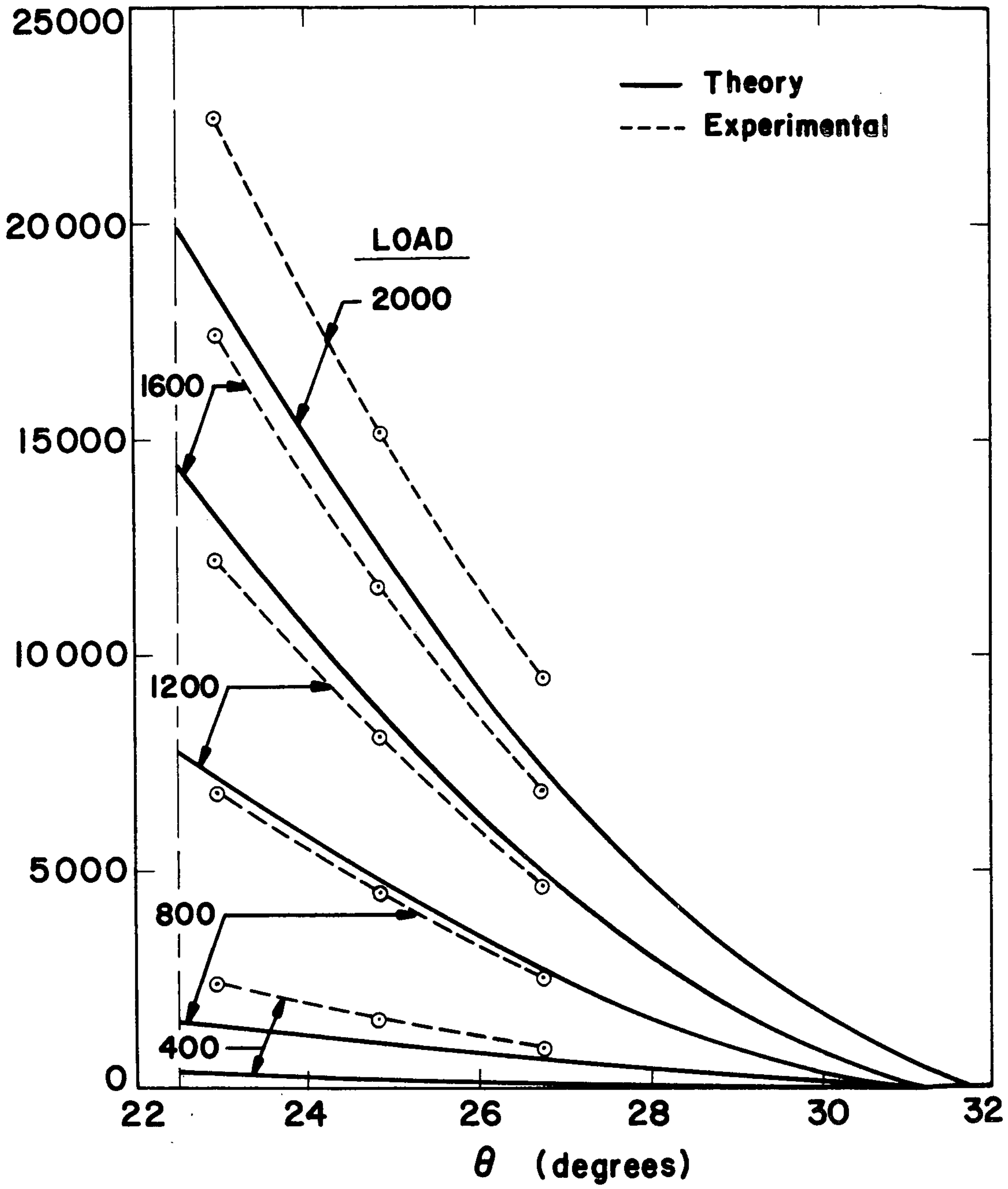


Fig. 3.3 Bending Stress Near Edge of Cutout (Cylinder #2 Midheight)

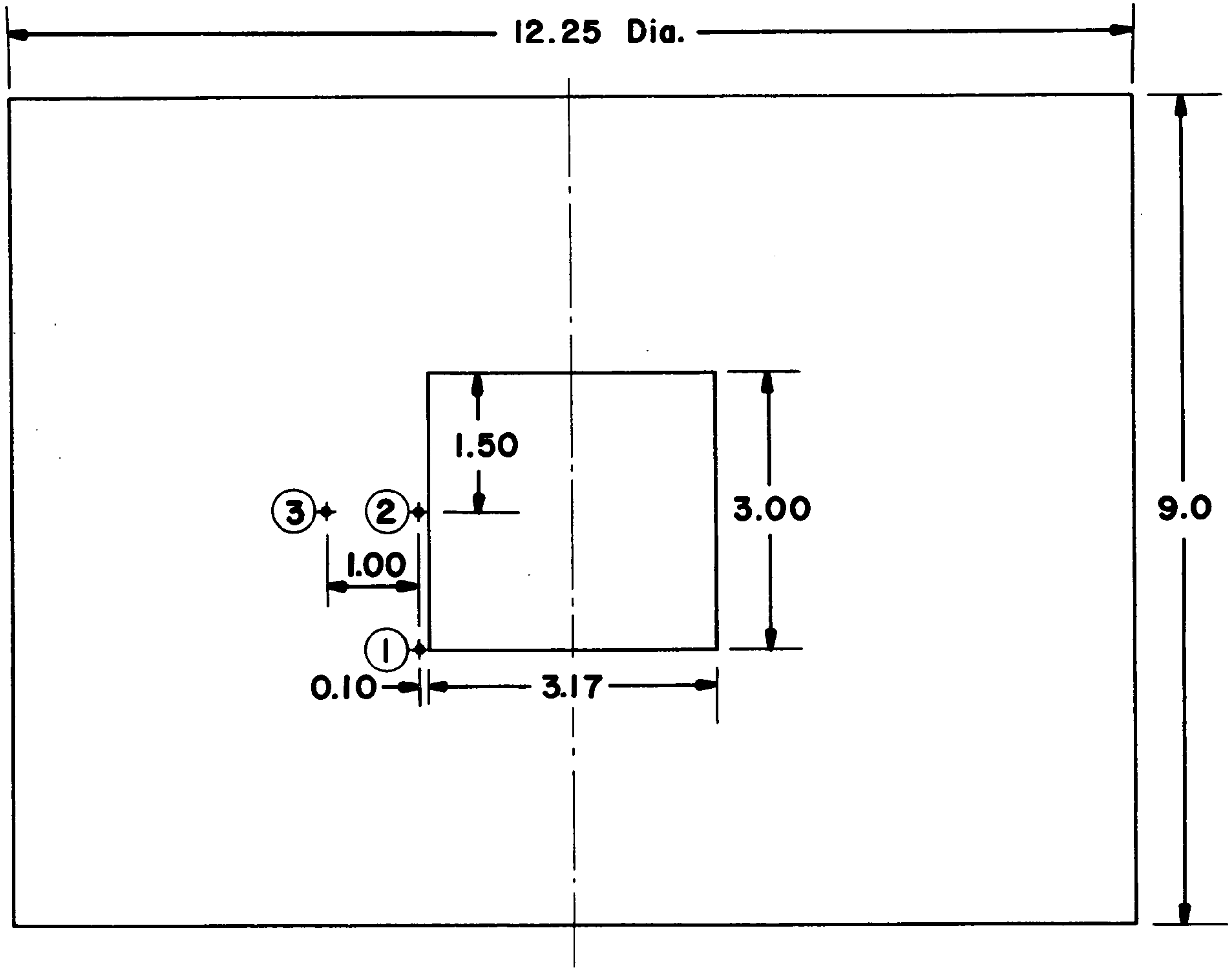


Fig. 3.4 Location of Strain Gage Stations for Cylinder #1

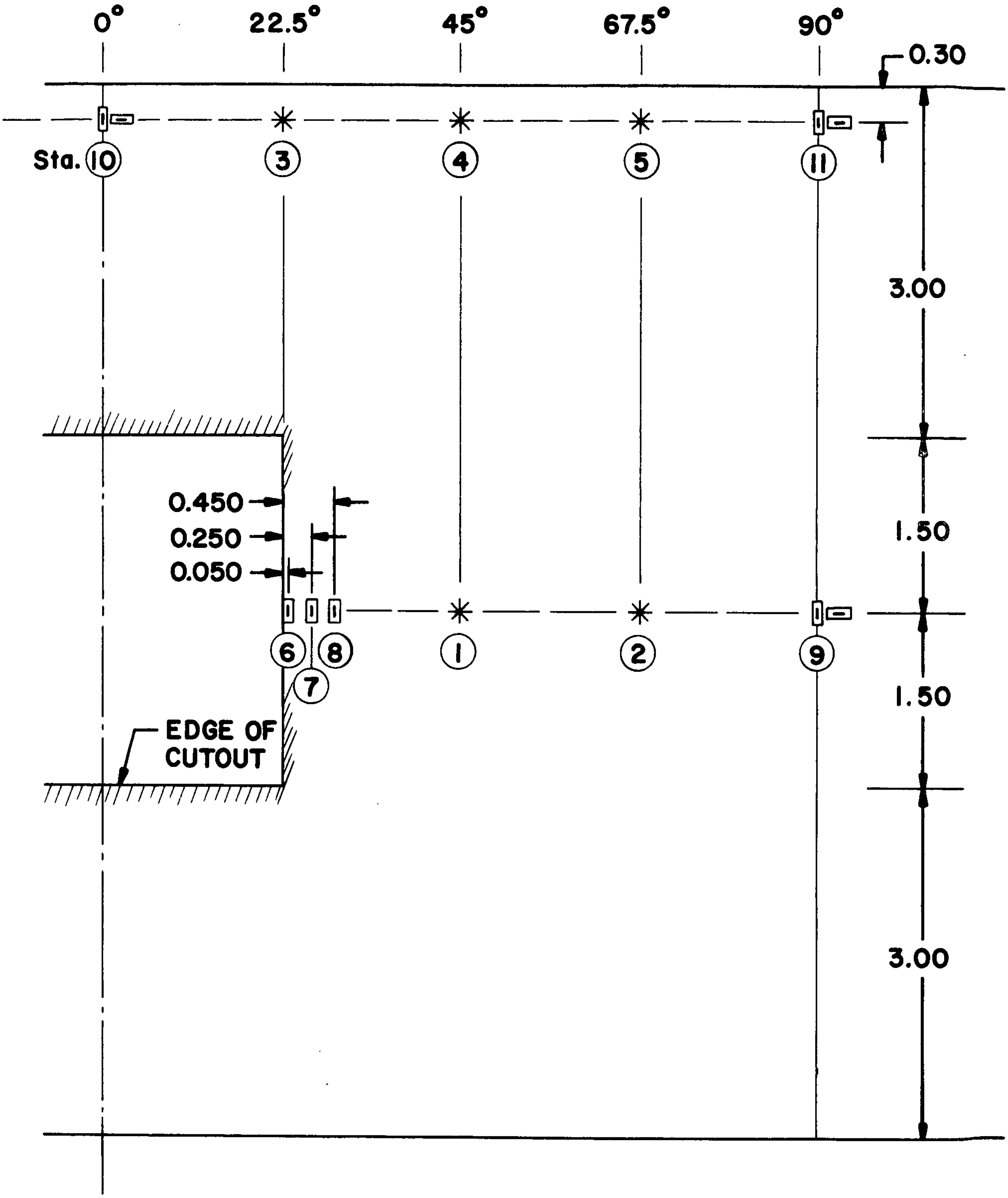


Fig. 3.5 Location of Strain Gage Stations for Cylinder #2

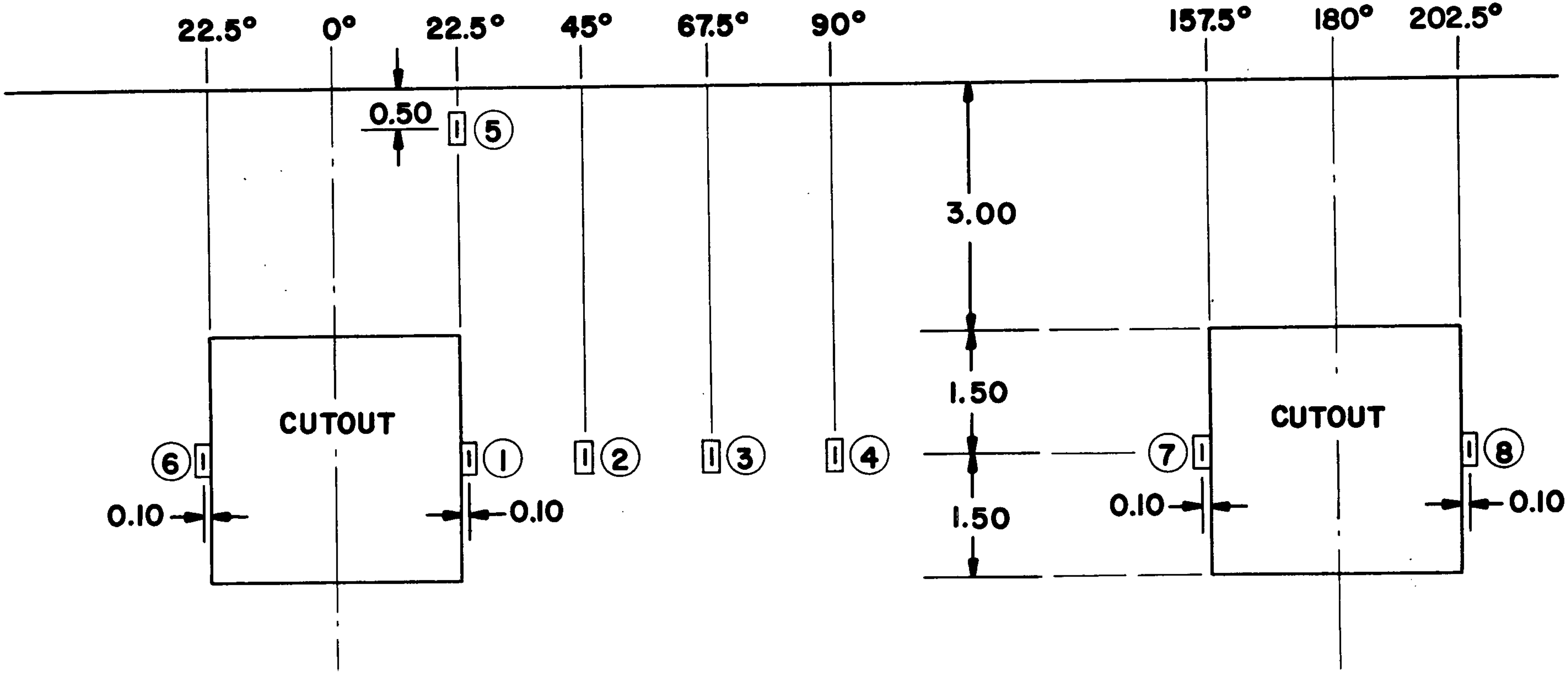


Fig. 3.6 Location of Strain Gage Stations for Cylinder #3

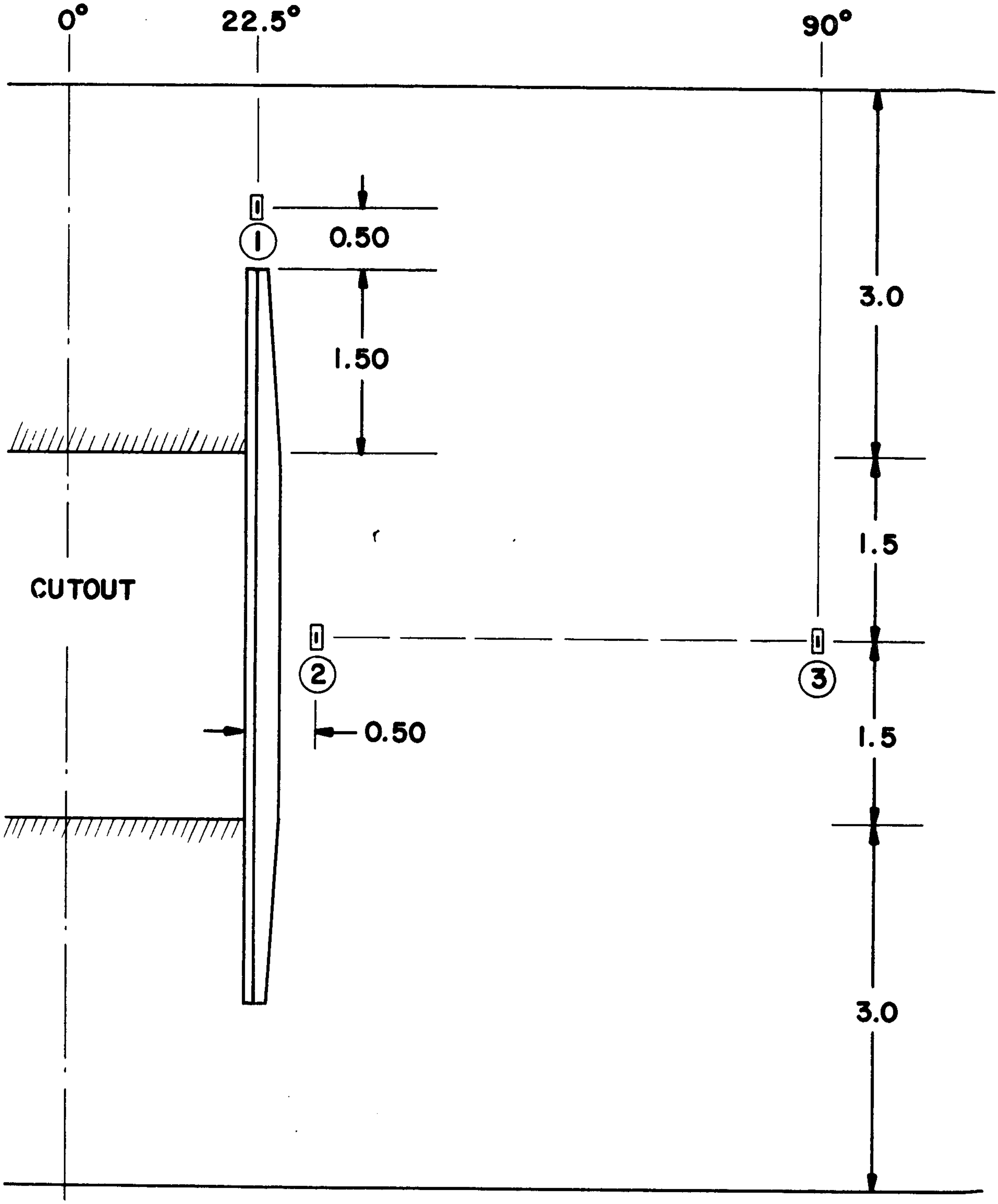


Fig. 3.7 Location of Strain Gage Stations for Cylinder #4

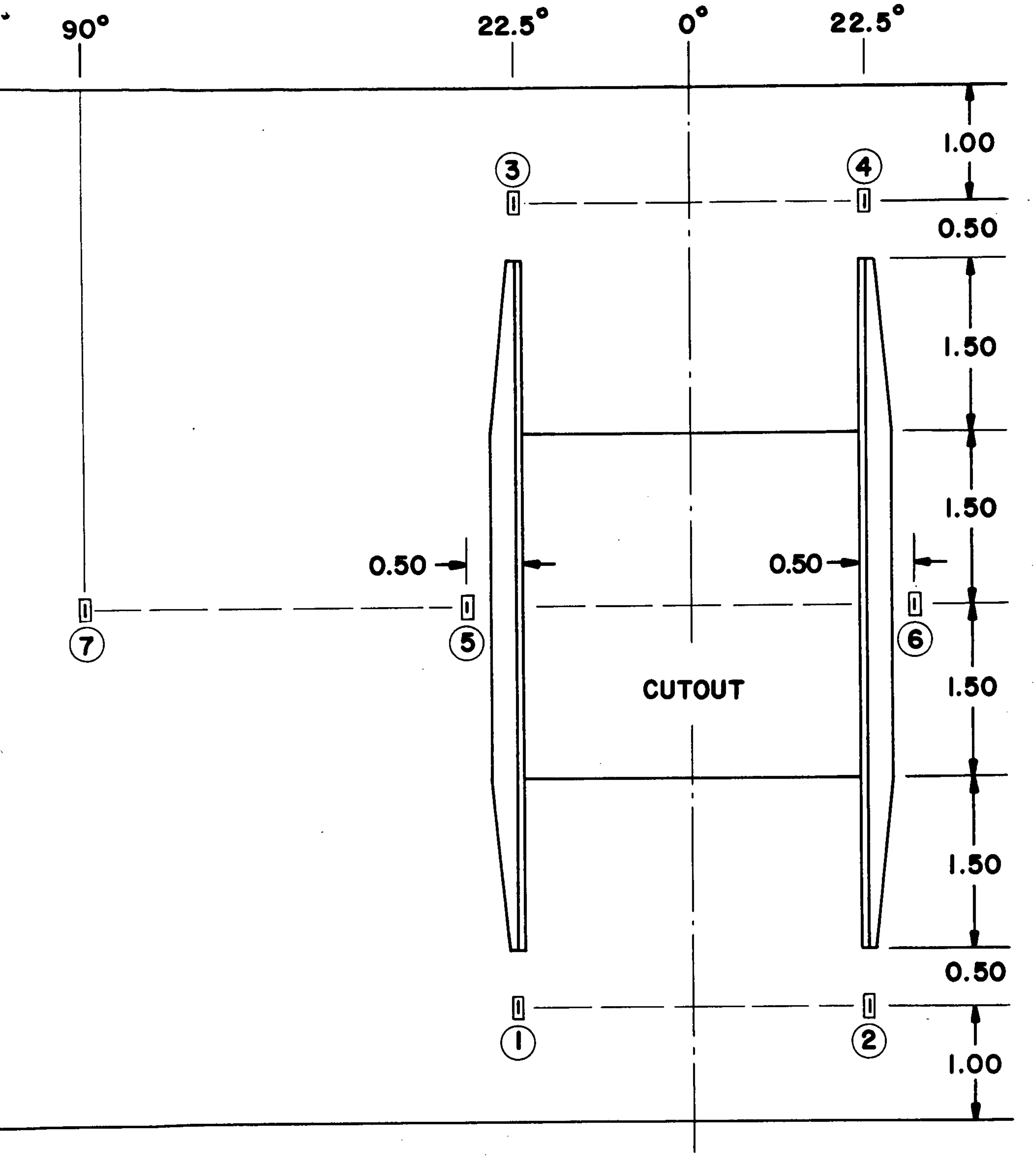


Fig. 3.8 Location of Strain Gage Stations for Cylinders #5, 6, 9, 10 and 11

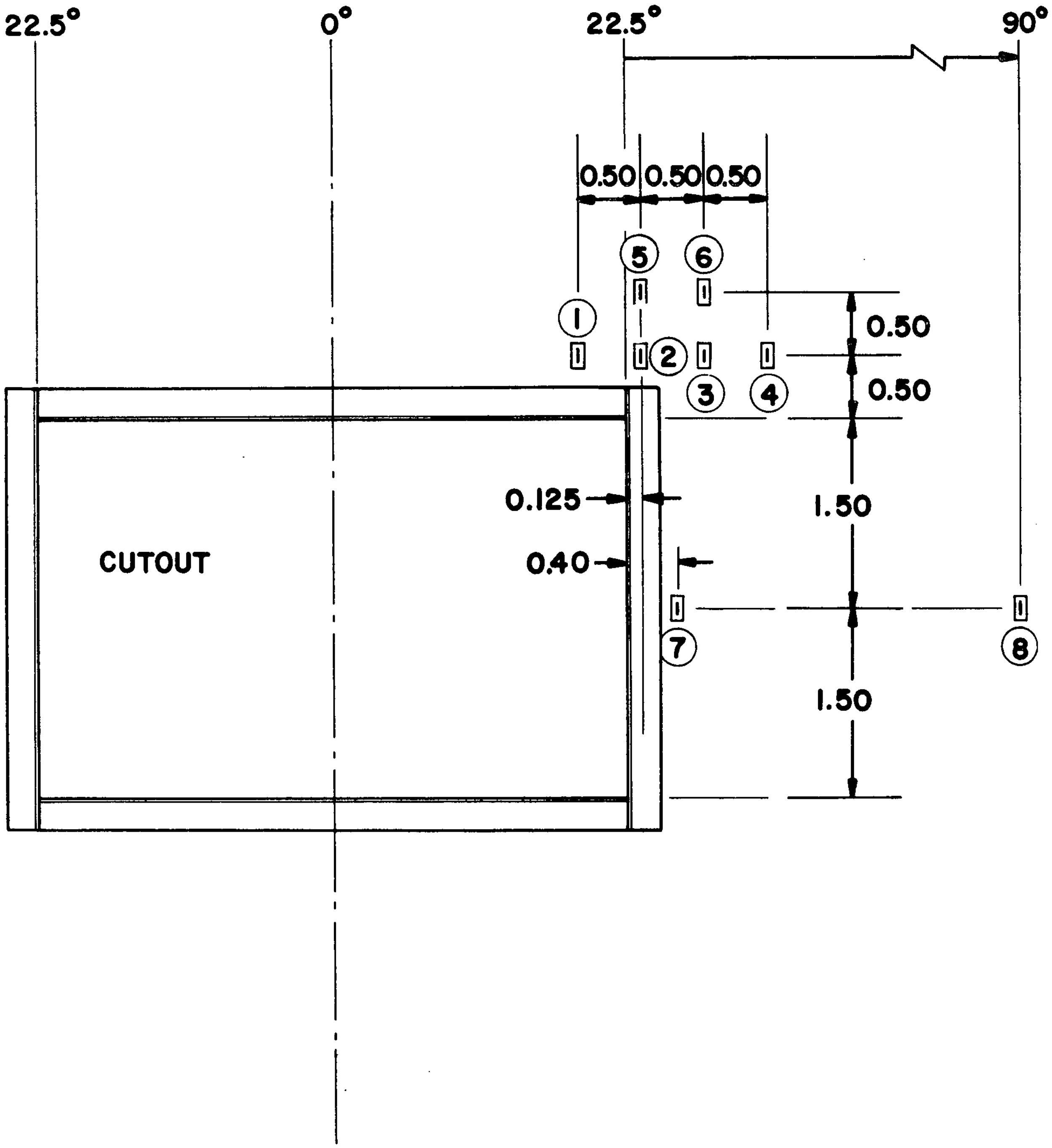


Fig. 3.9 Location of Strain Gage Stations for Cylinder #7

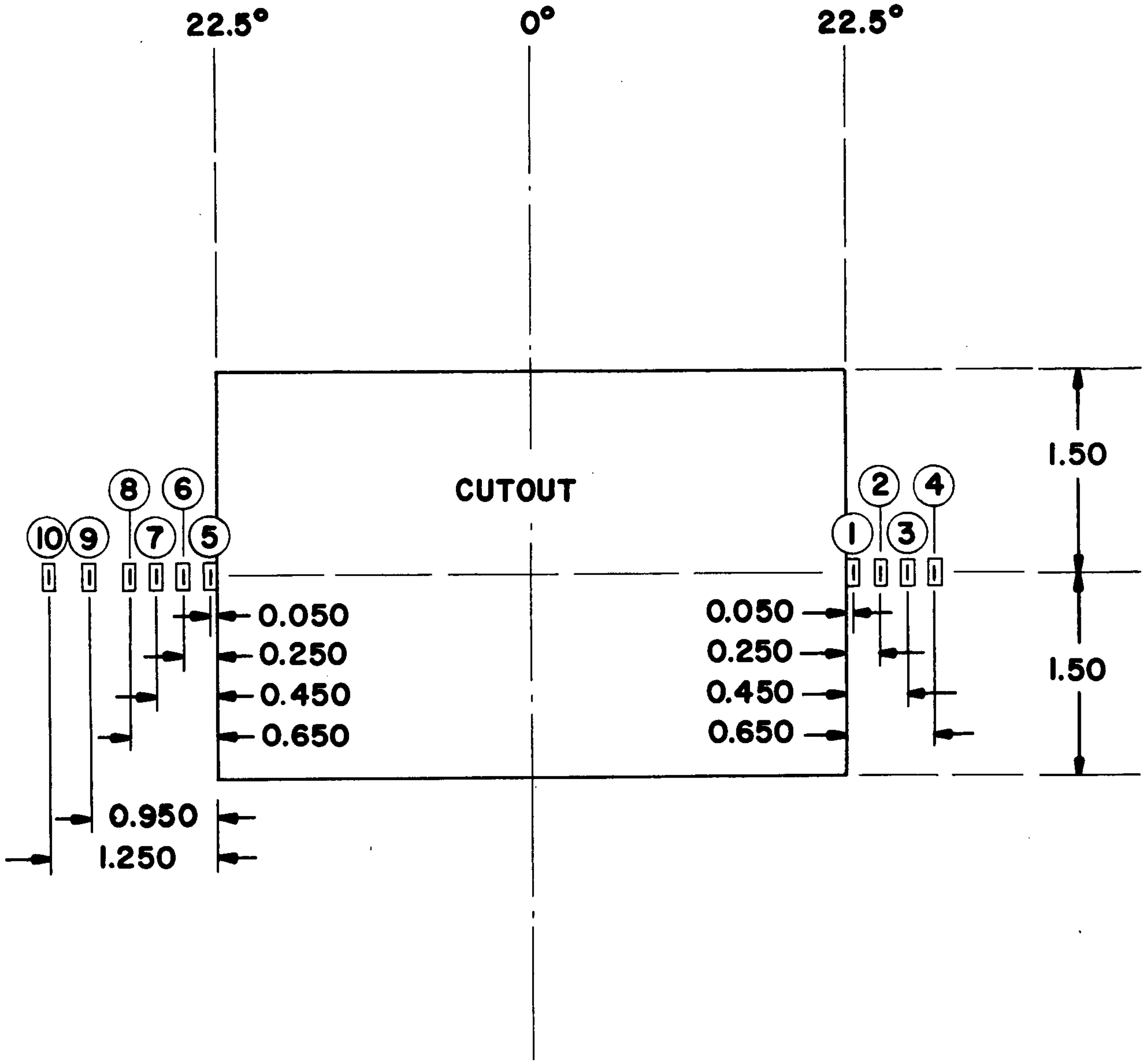


Fig. 3.10 Location of Strain Gage Stations for Cylinder #8

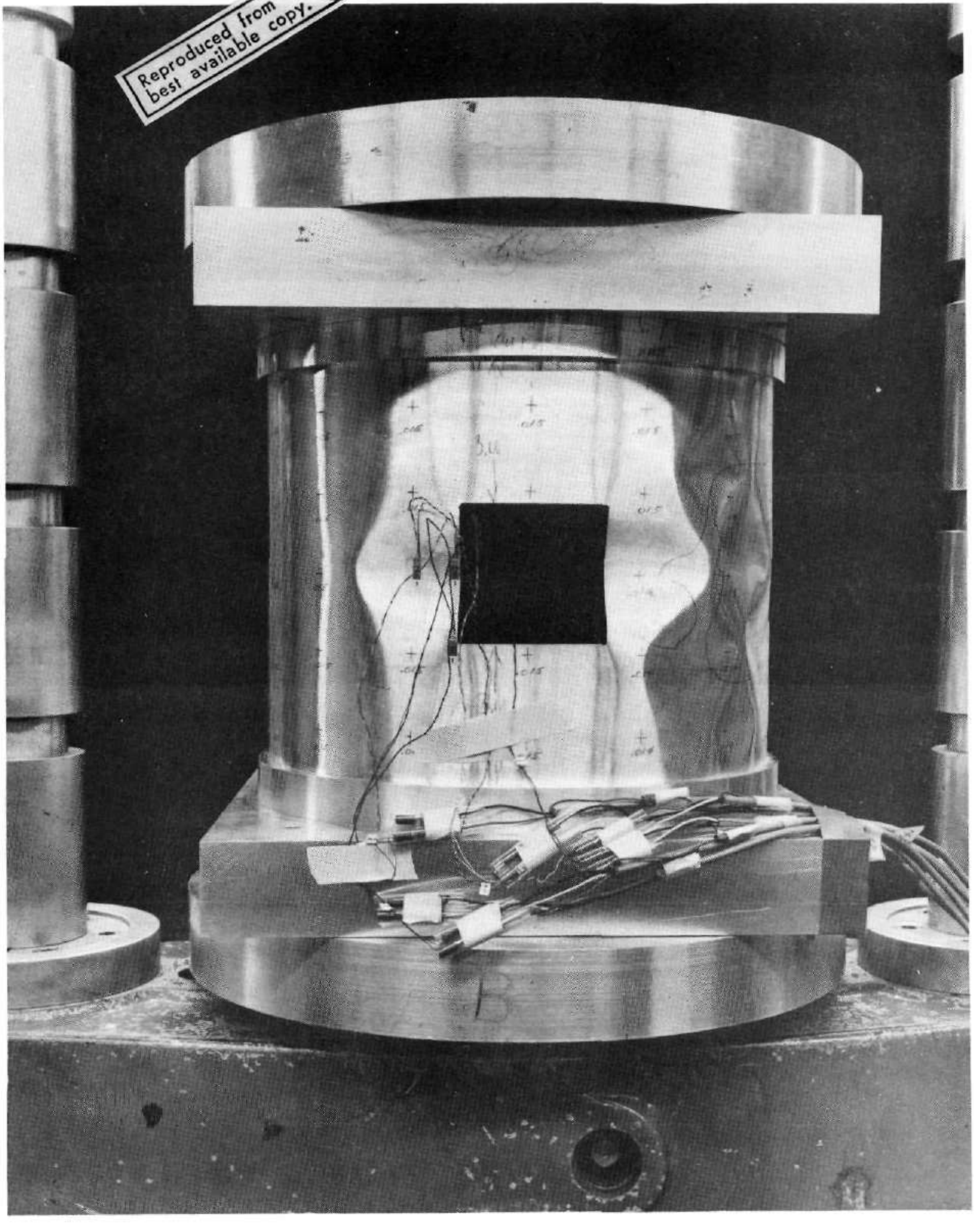


Fig. 3.11 Cylinder #1 After Buckling, General View

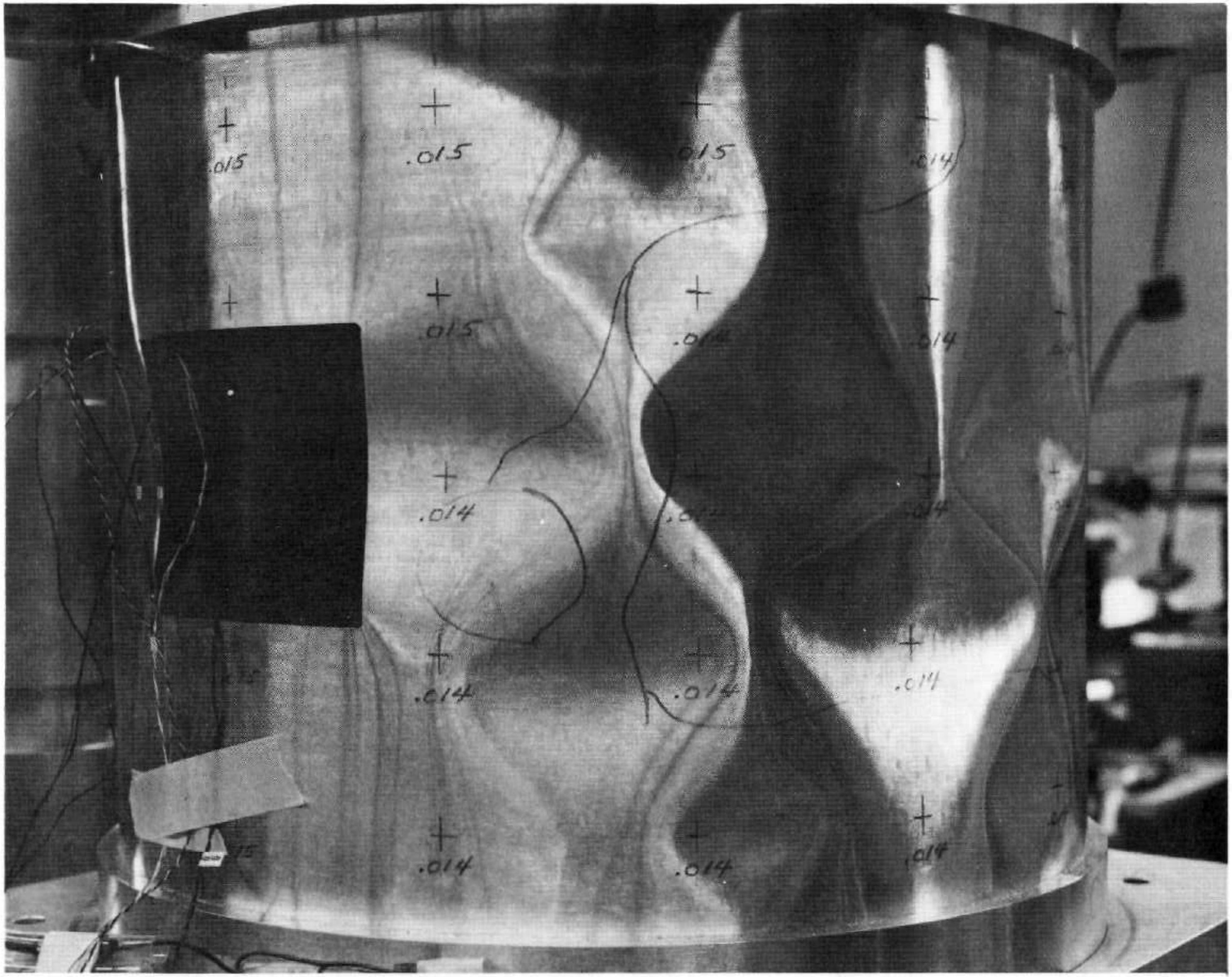


Fig. 3.12 Cylinder #1 After Buckling, General View

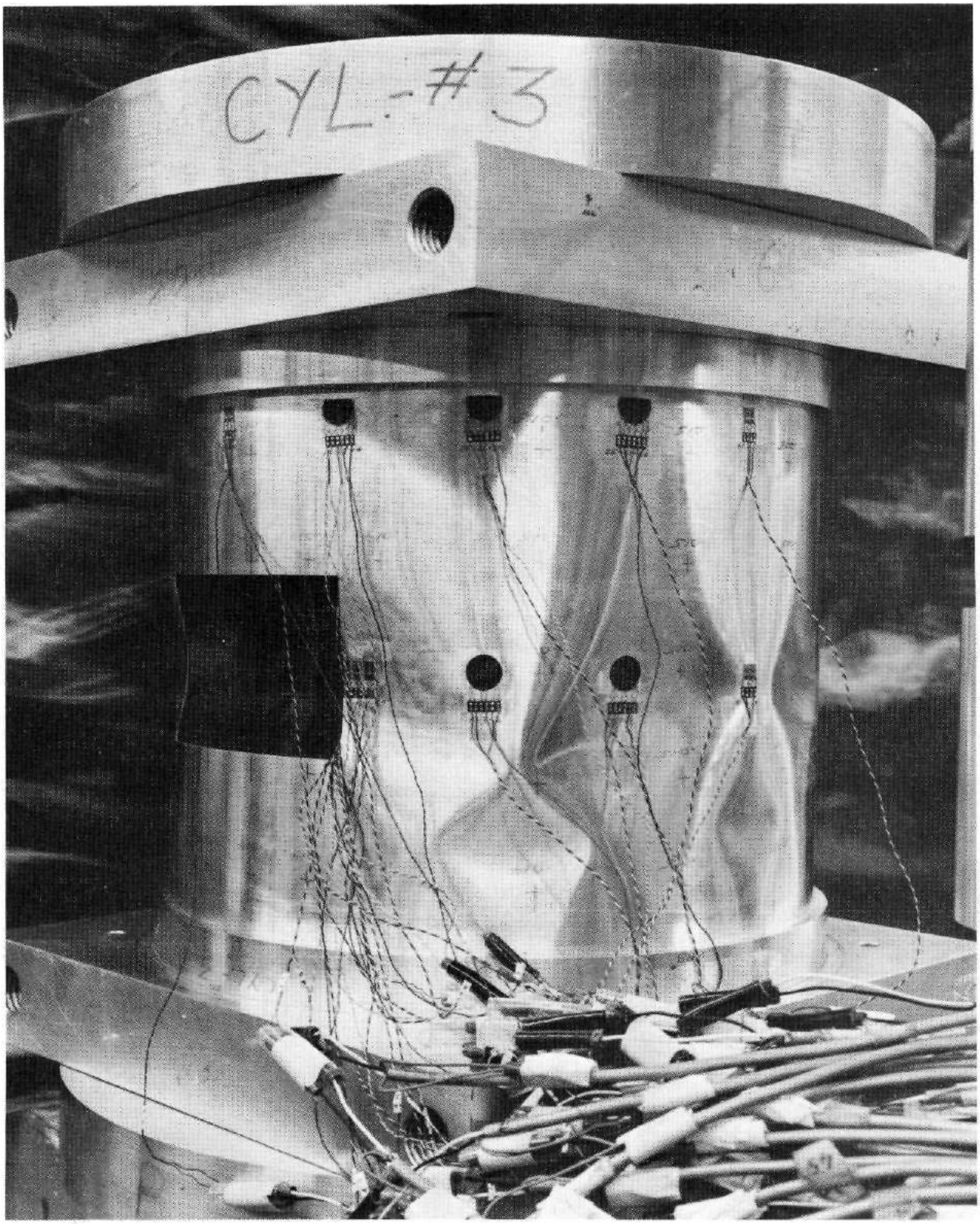


Fig. 3.13 Cylinder #2 After Buckling, General View

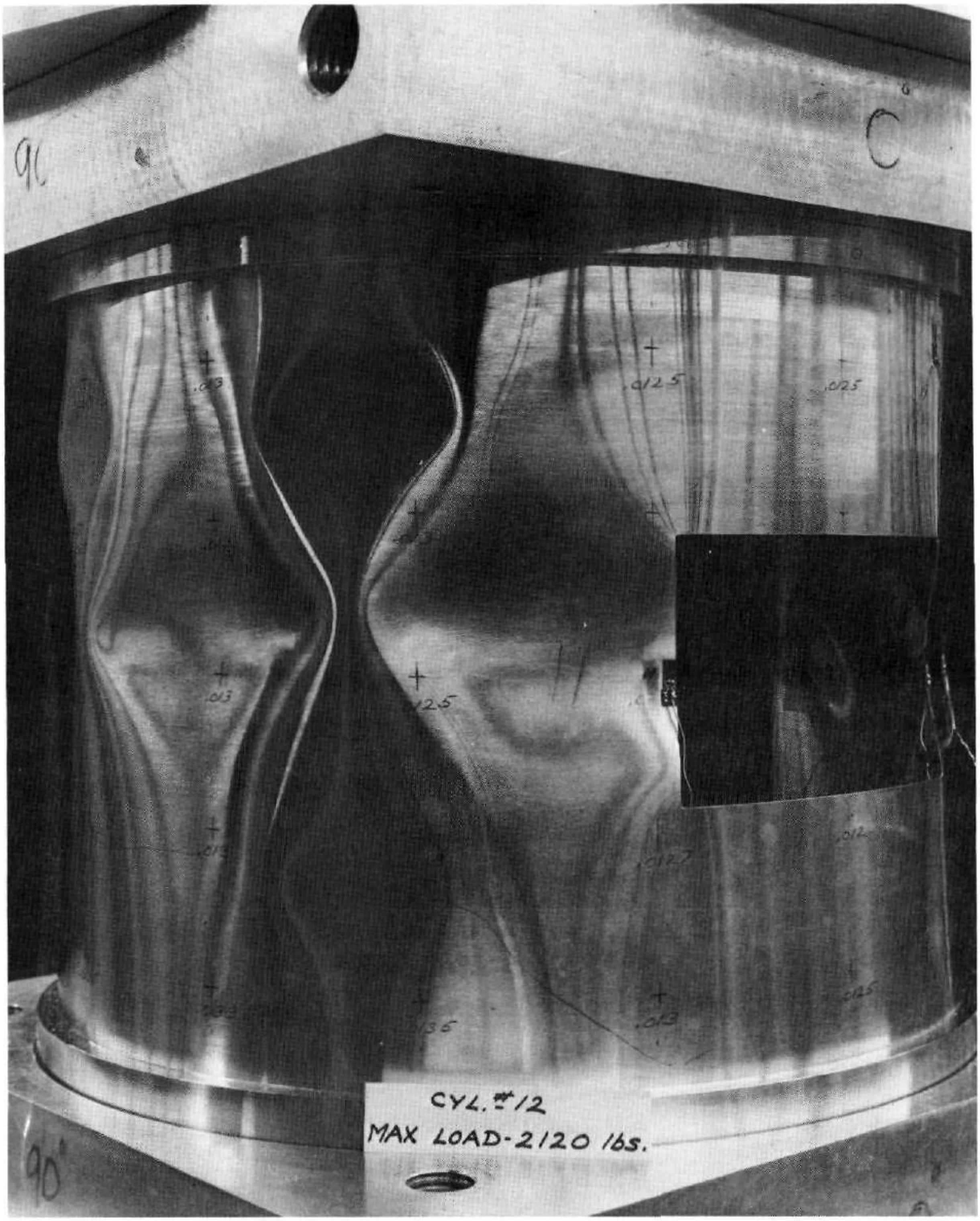


Fig. 3.14 Cylinder #3 After Buckling, General View, West Side

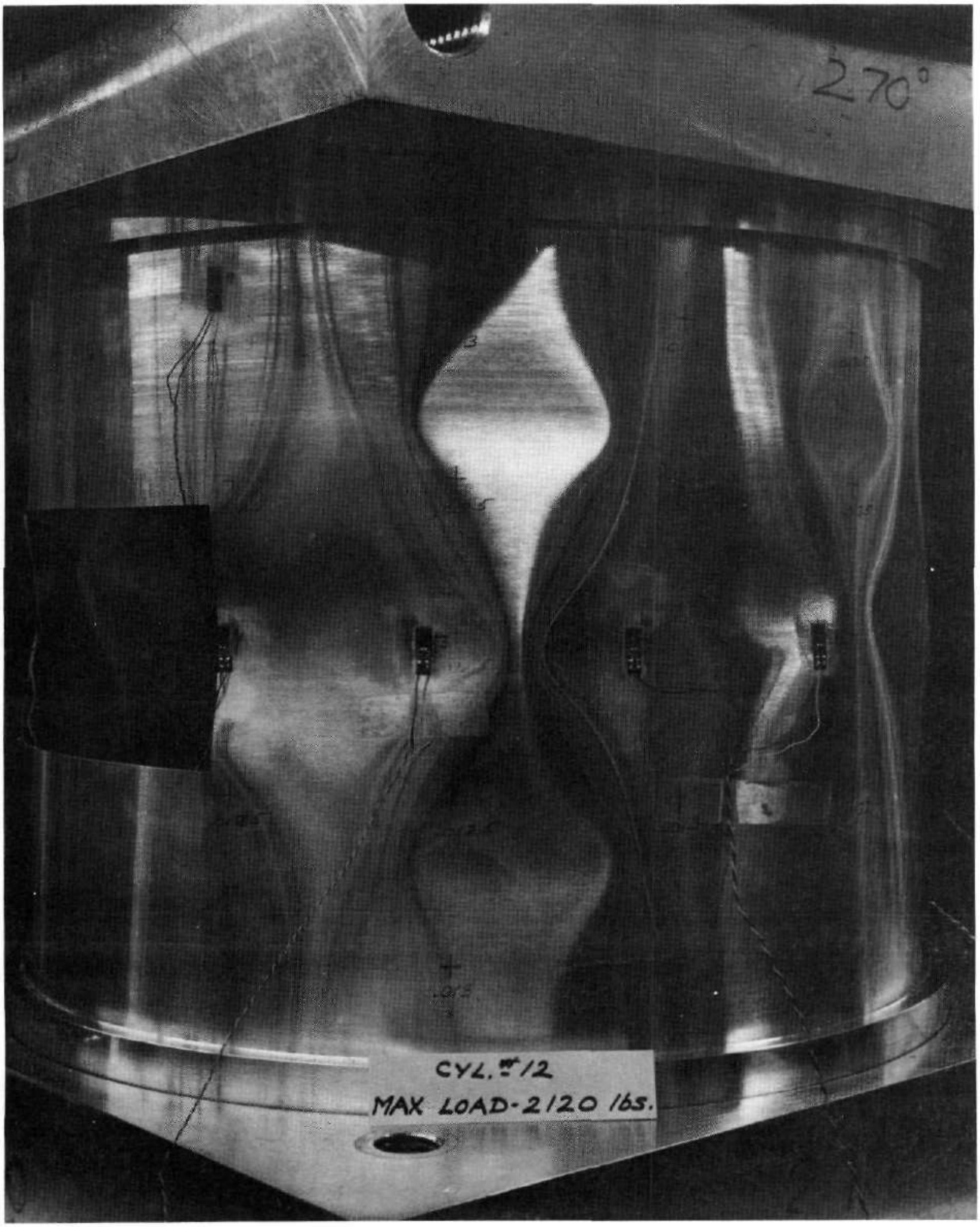


Fig. 3.15 Cylinder #3 After Buckling, General View, East Side

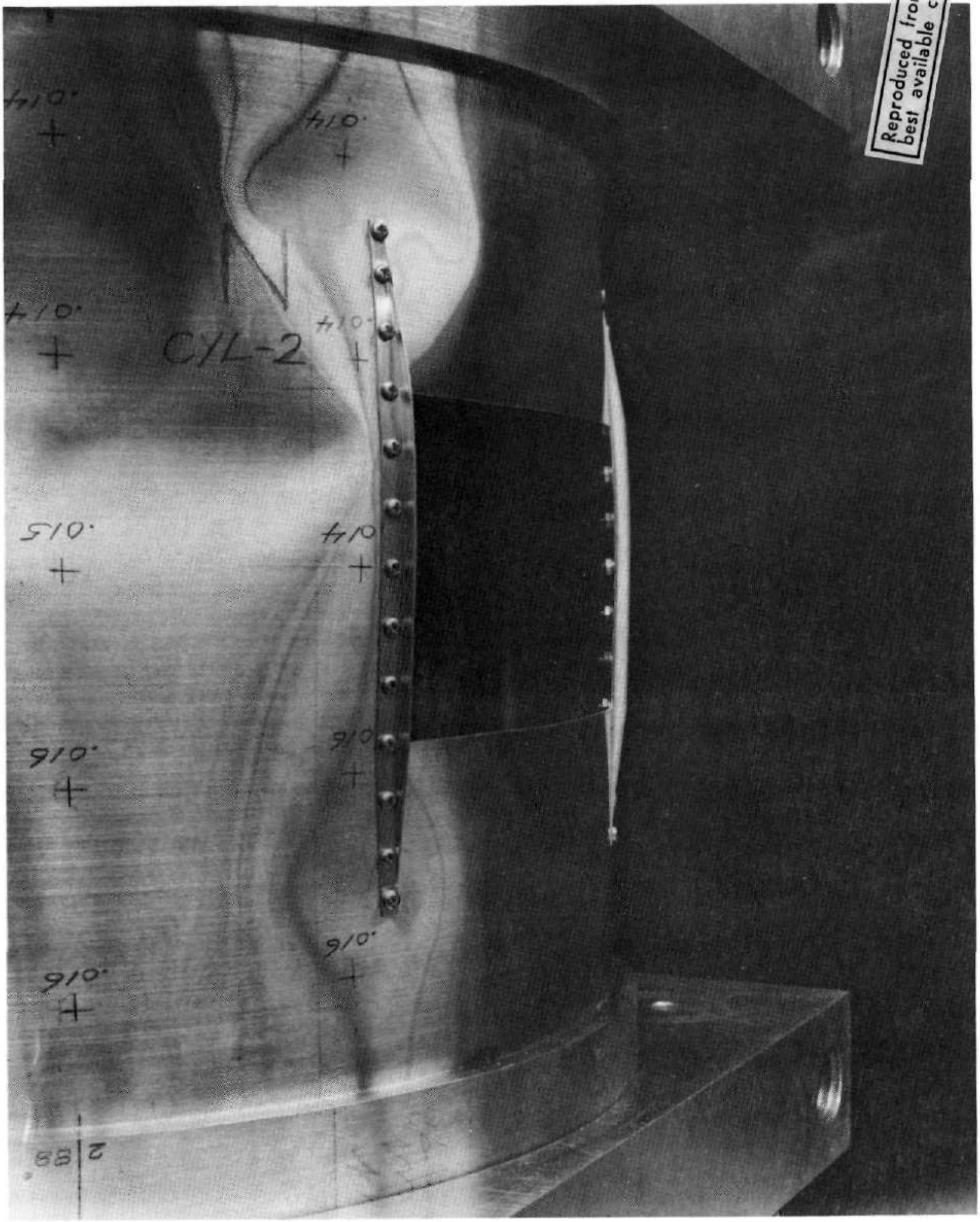
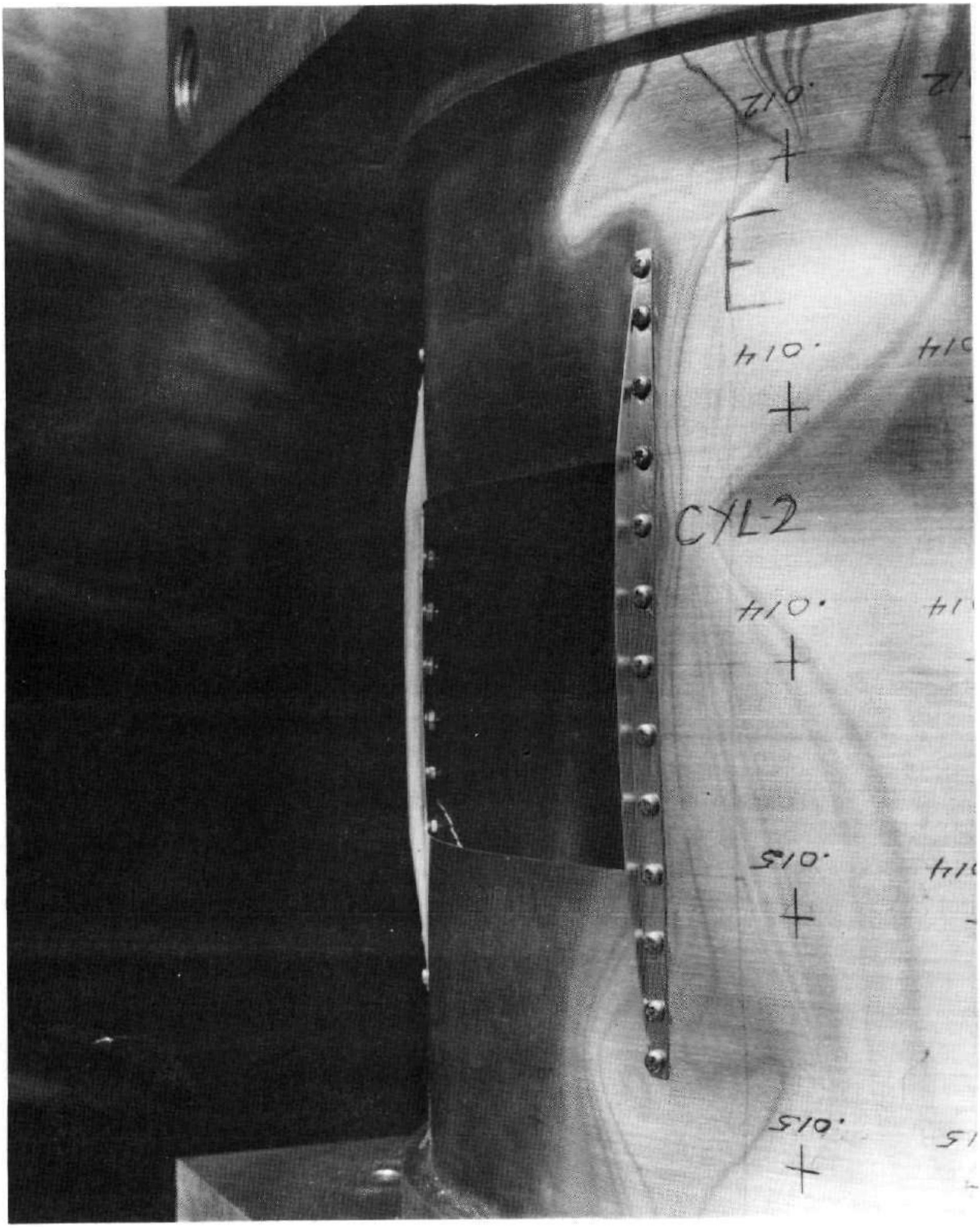


Fig. 3.16 Cylinder #4 After Buckling, Detail View From North Side


.012
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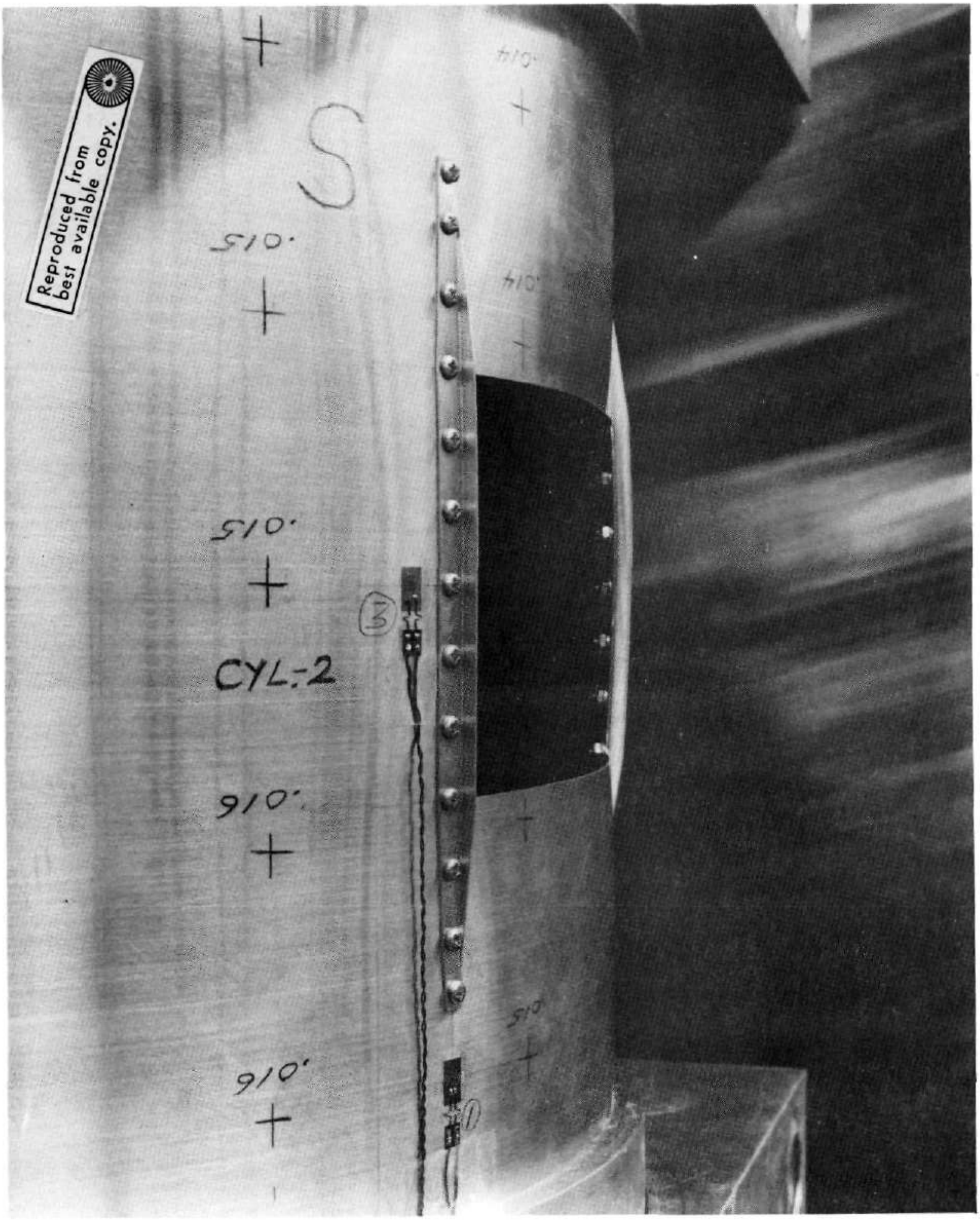

Fig. 3.18 Cylinder #4 After Buckling, Detail View From South Side

3

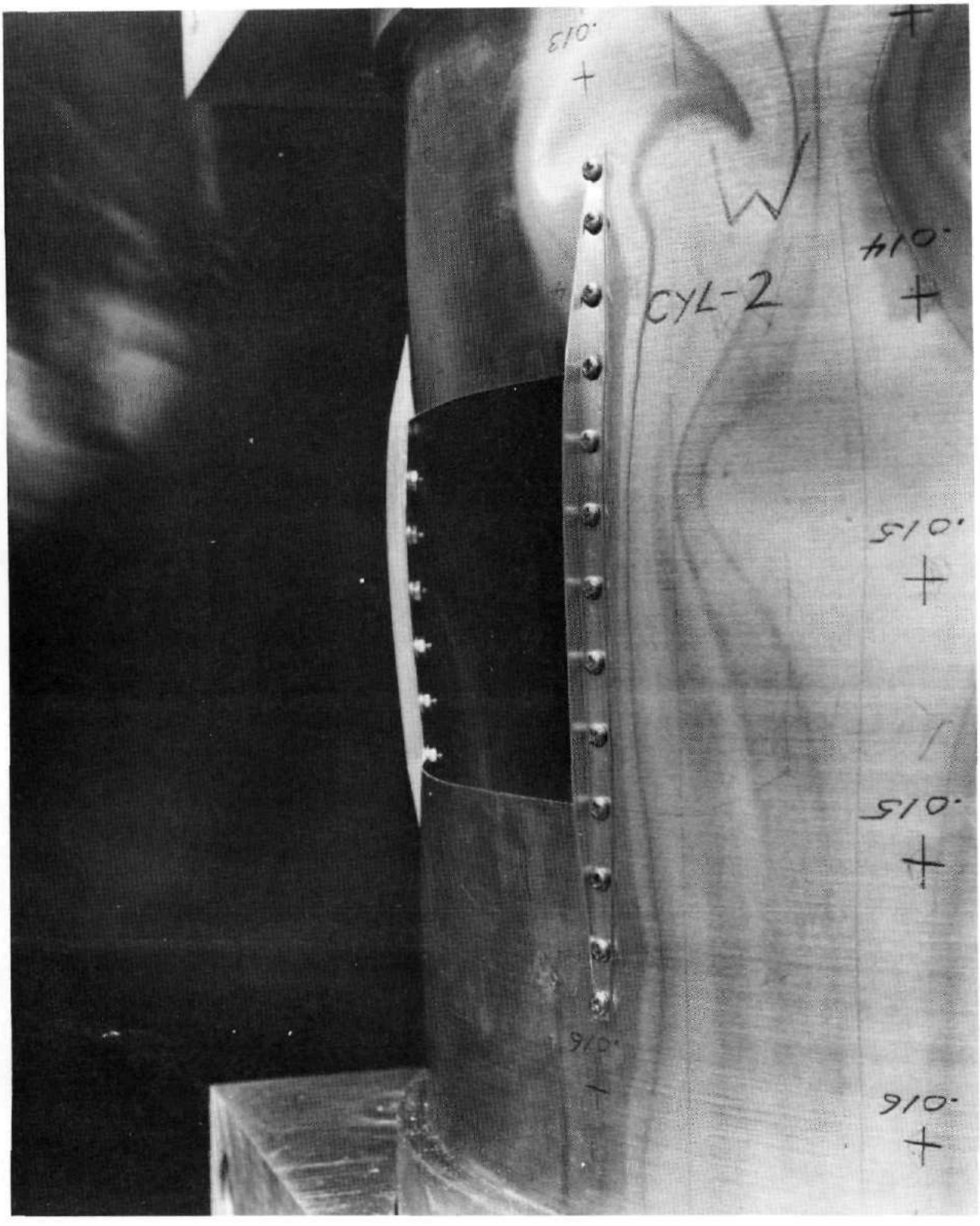

Fig. 3.19 Cylinder #4 After Buckling, Detail View From West Side

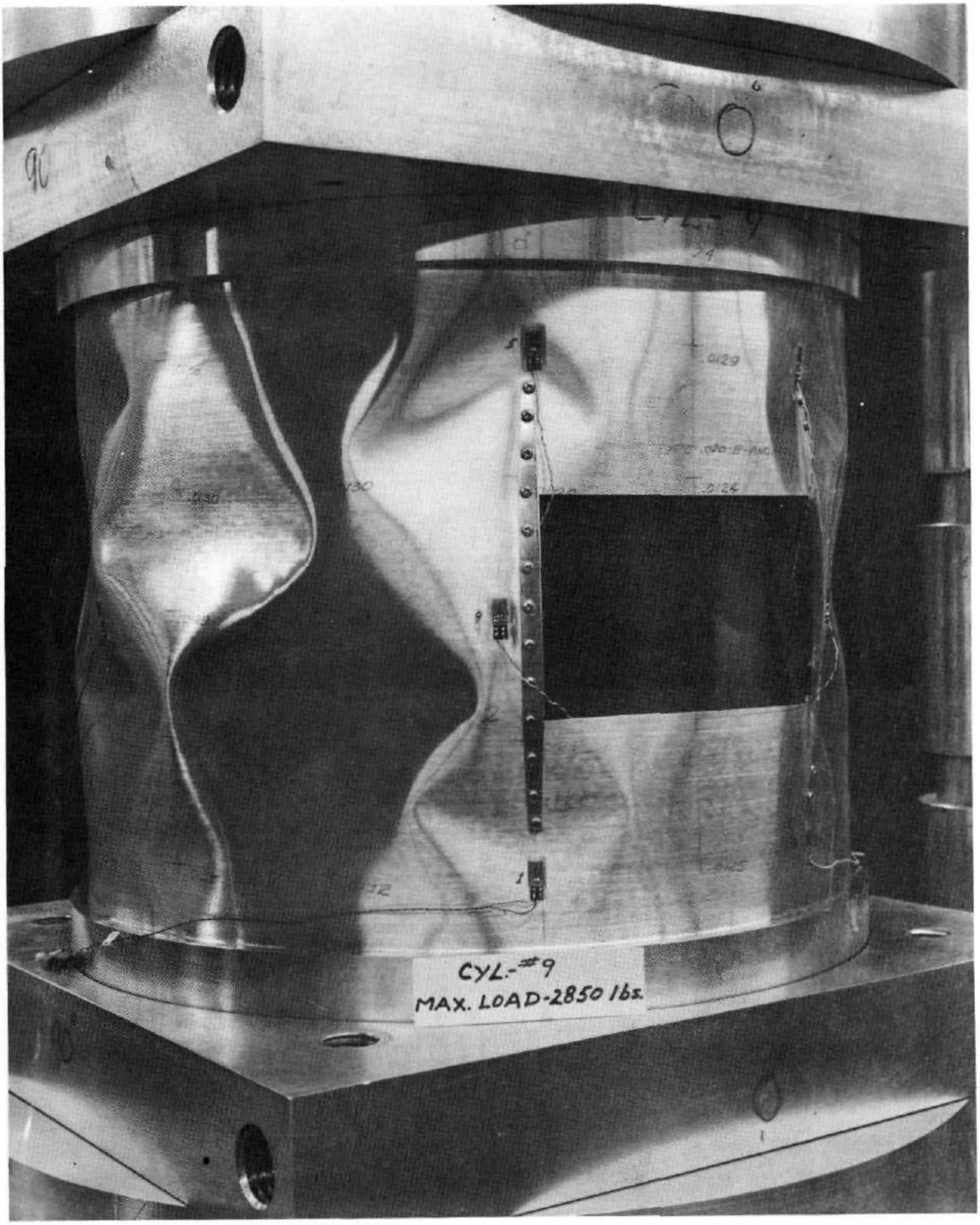

Fig. 3.20 Cylinder #5 After Buckling, General View

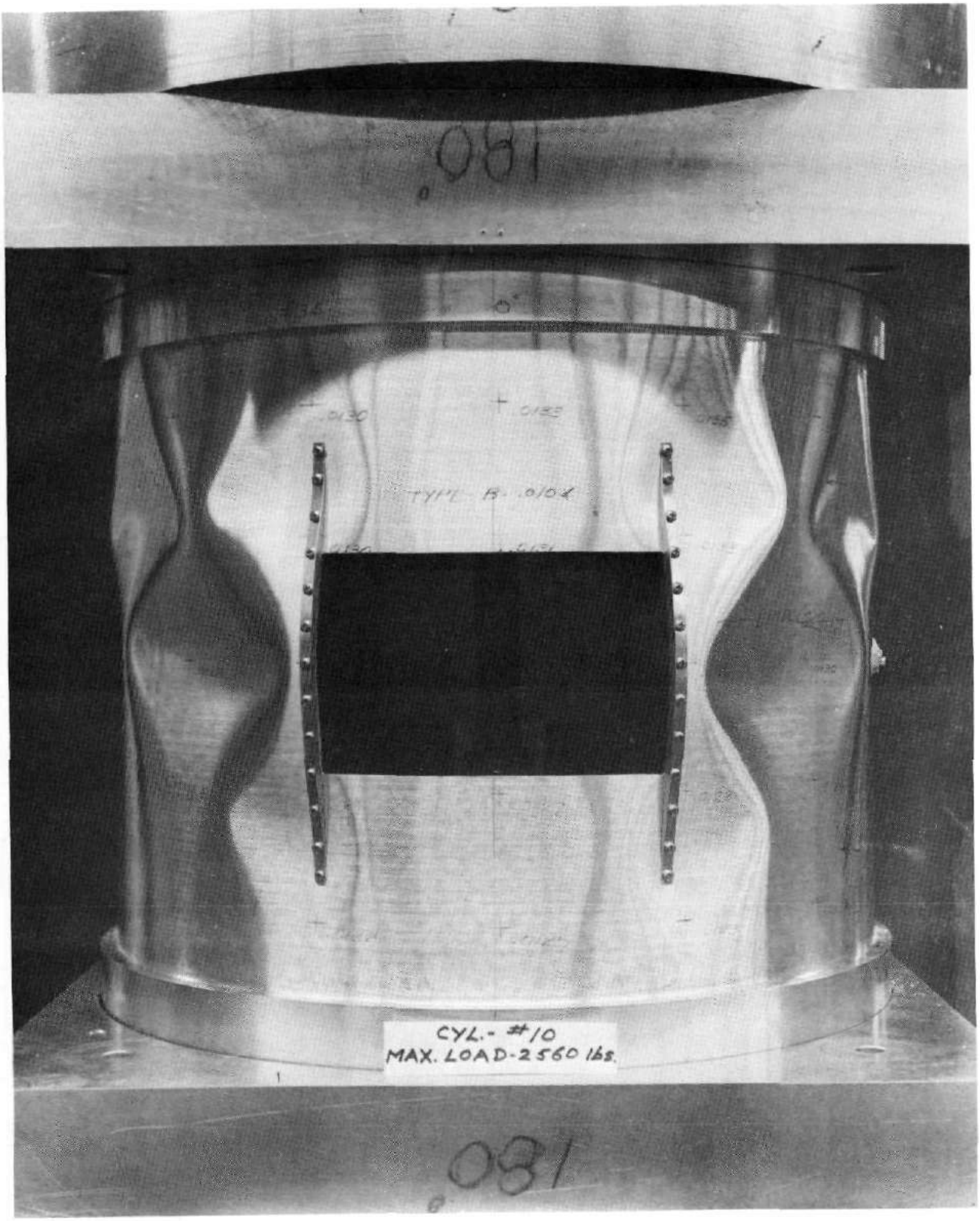

Fig. 3.21 Cylinder #6 After Buckling, General View

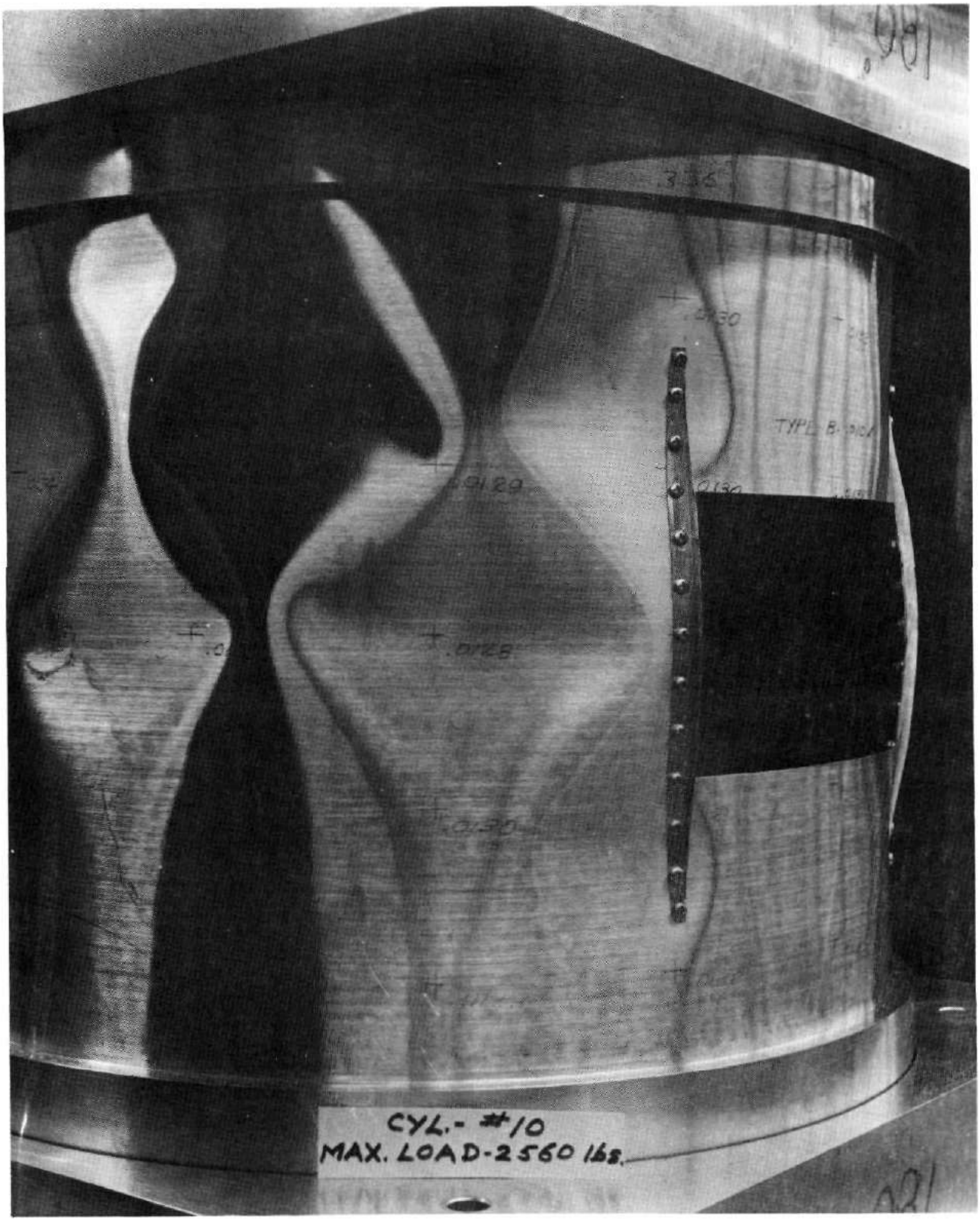

Fig. 3.22 Cylinder #6 After Buckling, Detail View From East Side

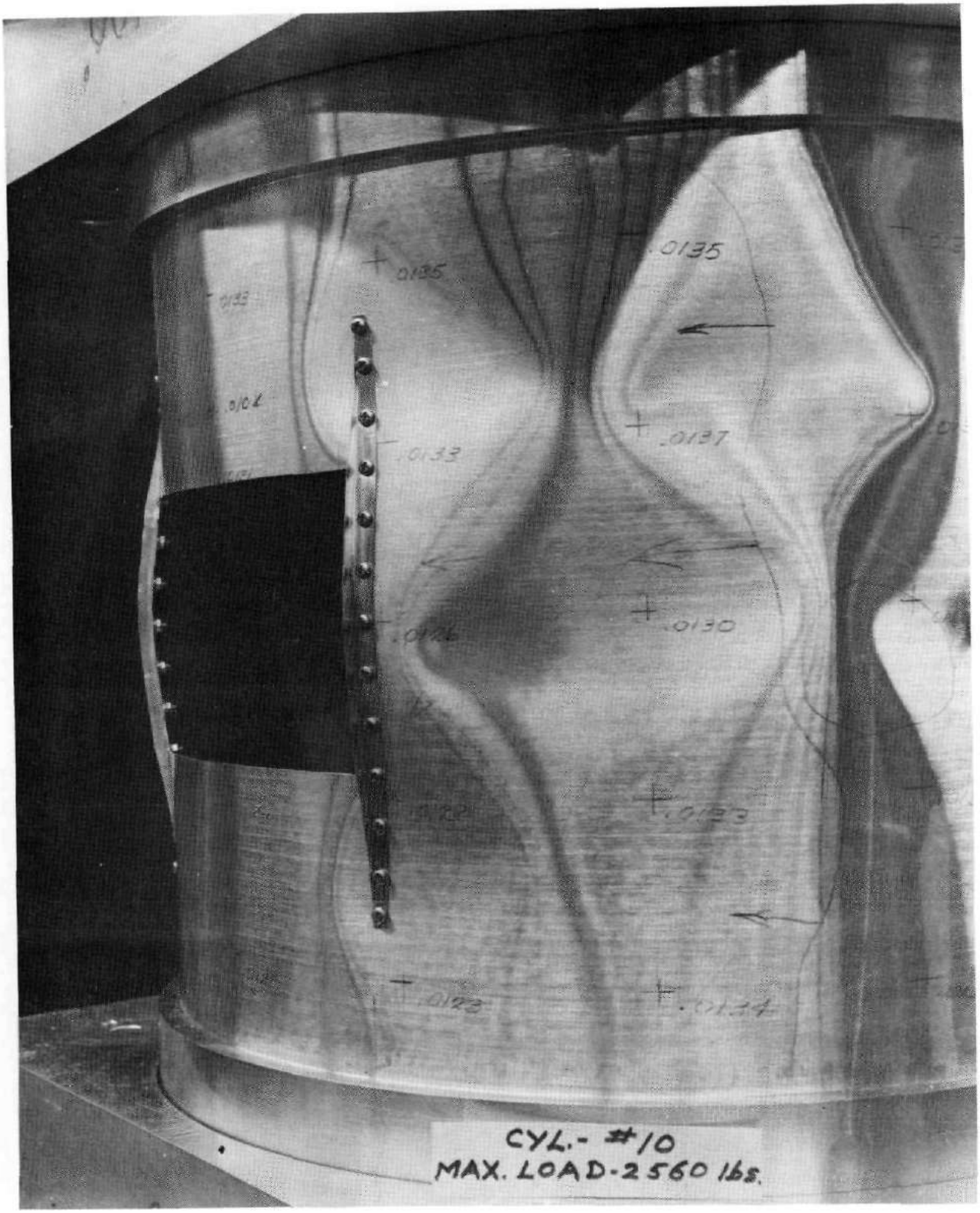

Fig. 3.23 Cylinder #6 After Buckling, Detail View From West Side

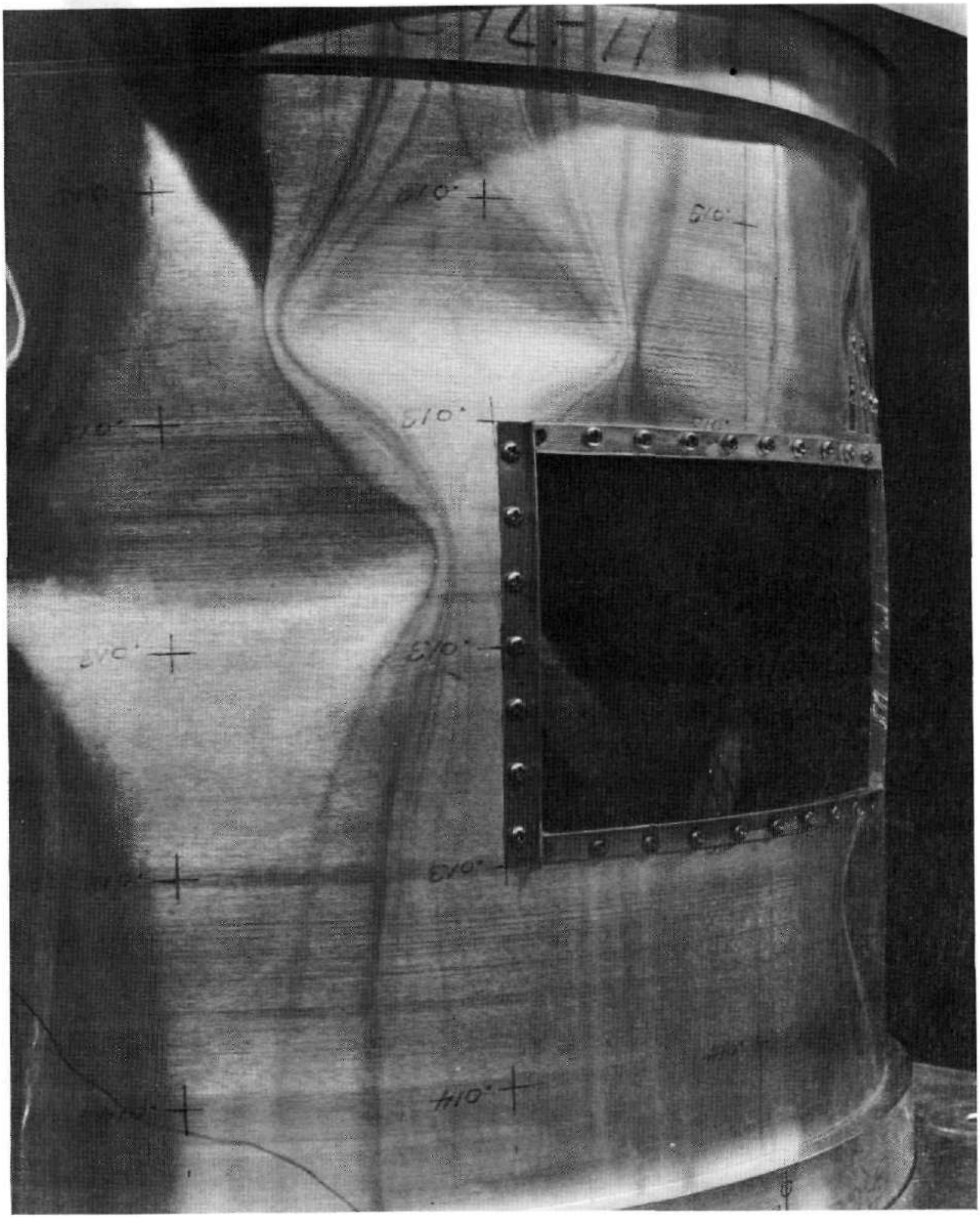

Fig. 3.24 Cylinder #7 After Buckling, Detail View From East Side

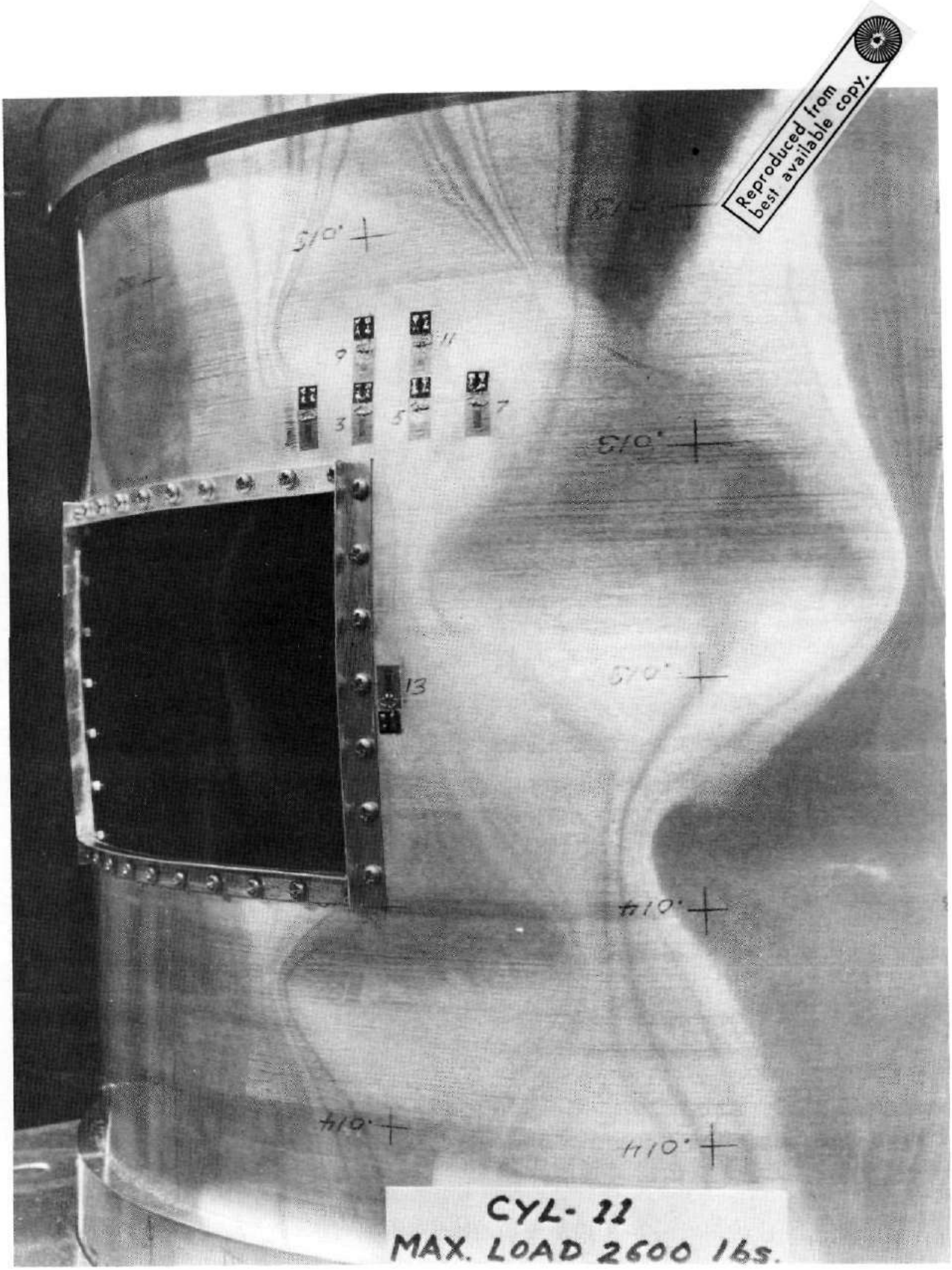

Fig. 3.25 Cylinder #7 After Buckling, Detail View From West Side

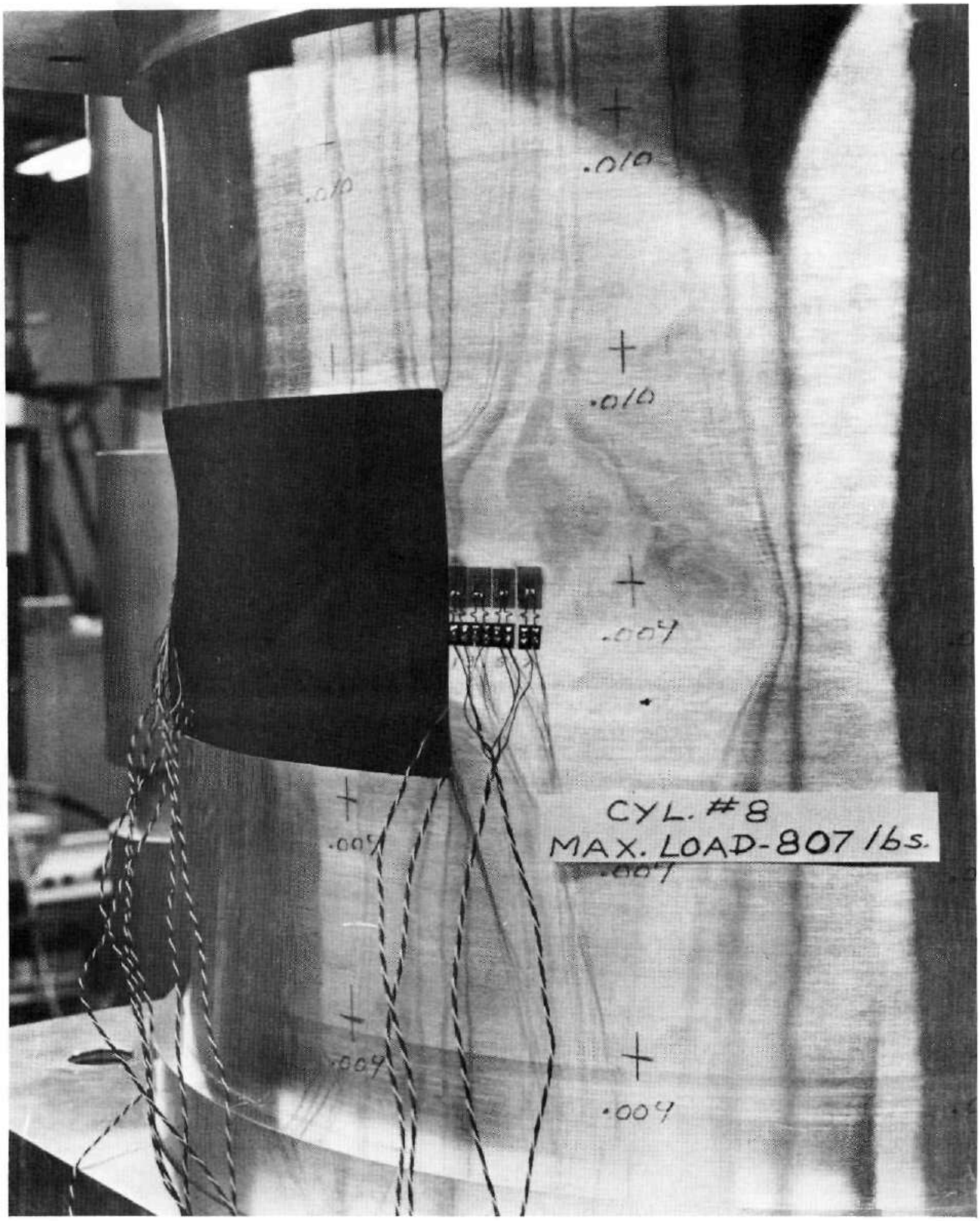

Fig. 3.26 Cylinder #8 After Buckling, Detail View, Cutout With Gages

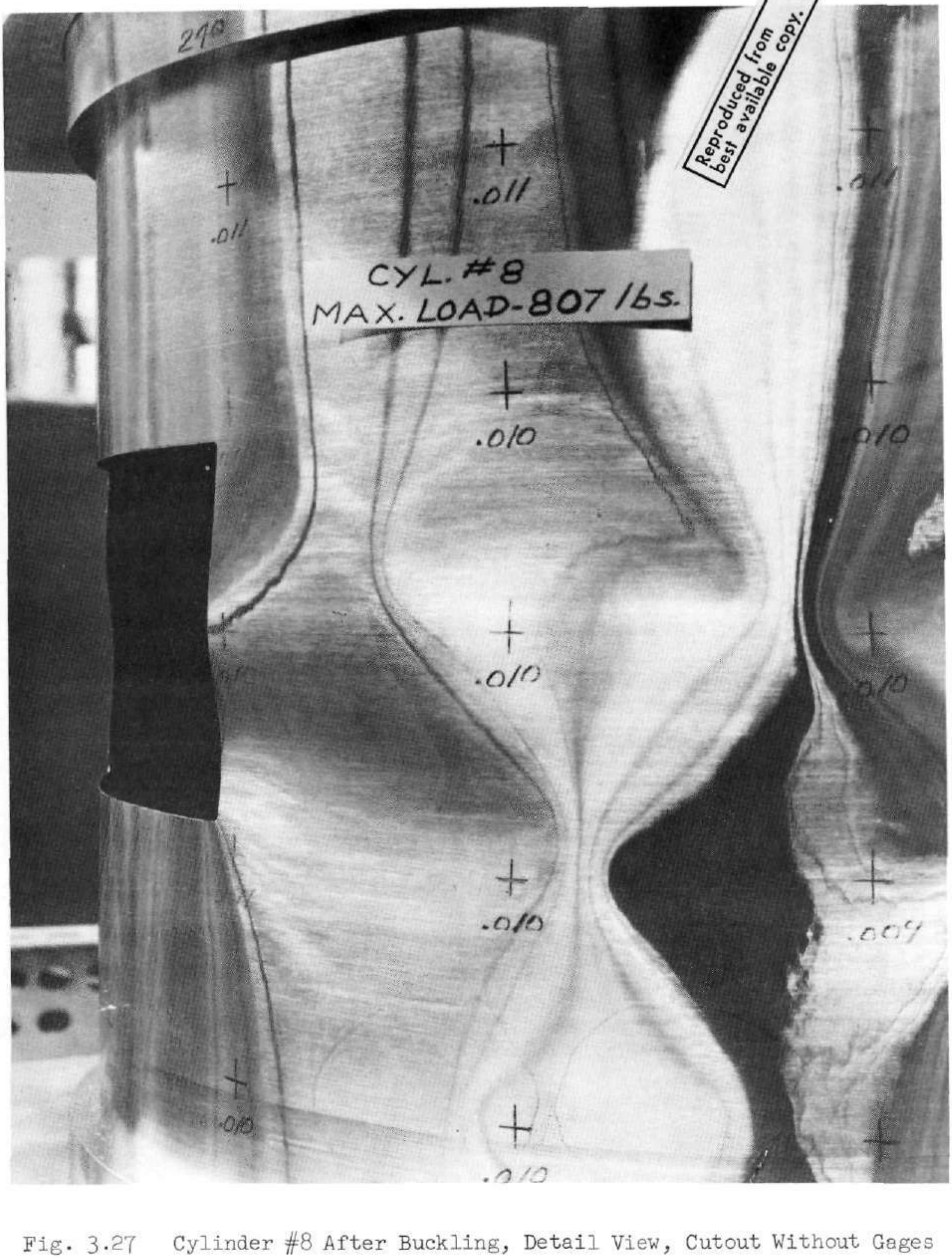

Fig. 3.27 Cylinder #8 After Buckling, Detail View, Cutout Without Gages

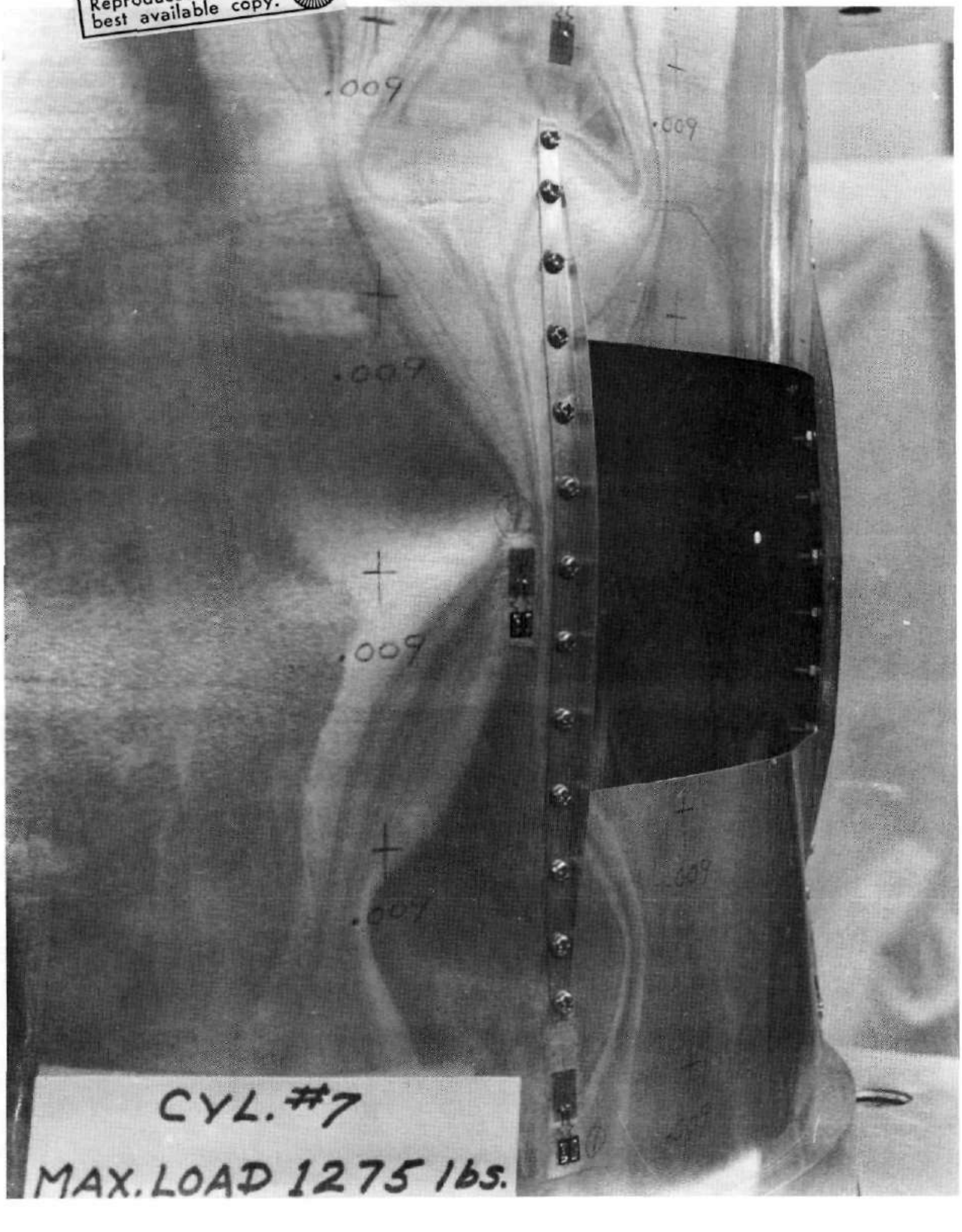

Fig. 3.28 Cylinder #9 After Buckling, Detail View

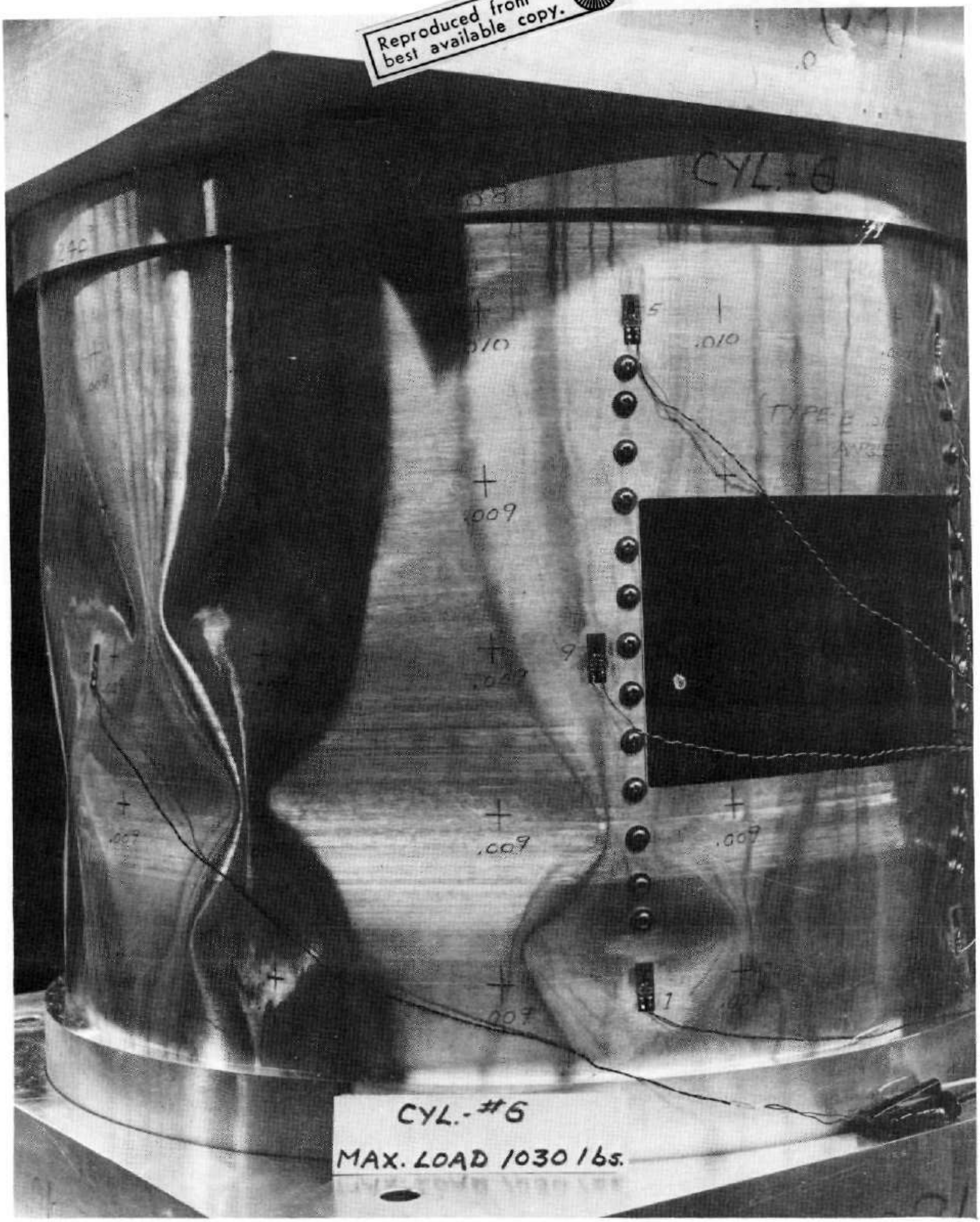

Fig. 3.29 Cylinder #10 After Buckling, General View

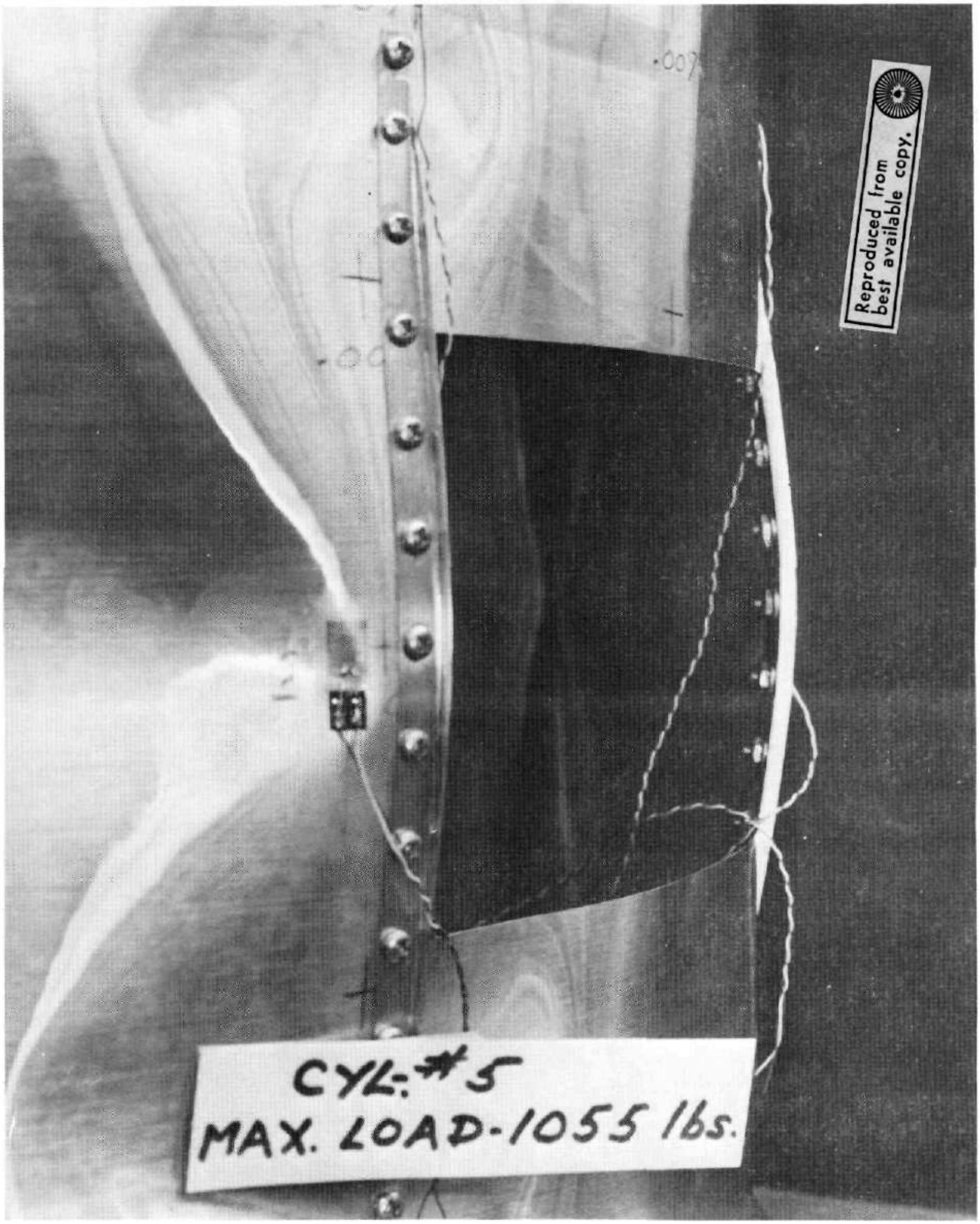

Fig. 3.30 Cylinder #11 After Buckling, General View

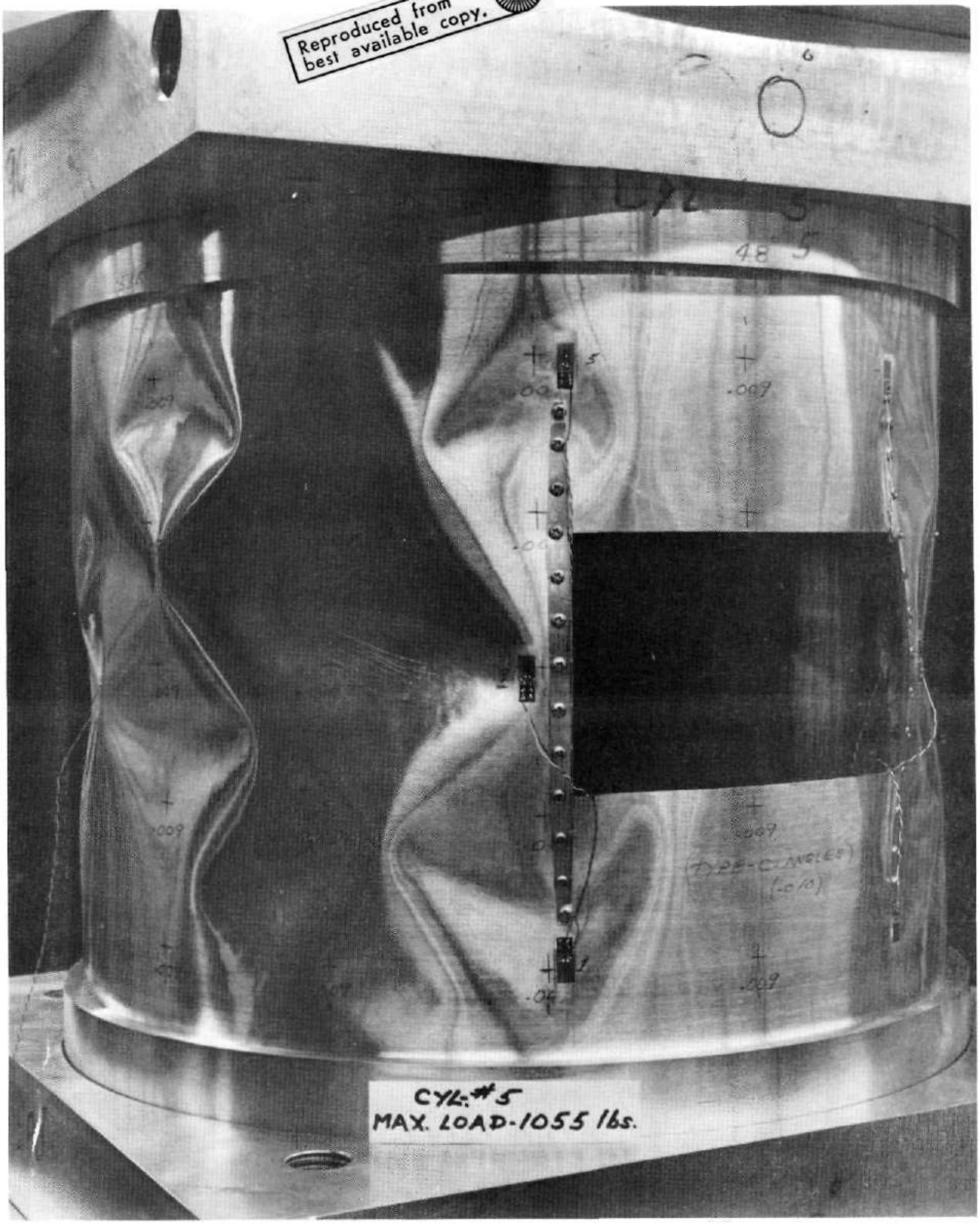

Fig. 3.31 Cylinder #11 After Buckling, Detail View

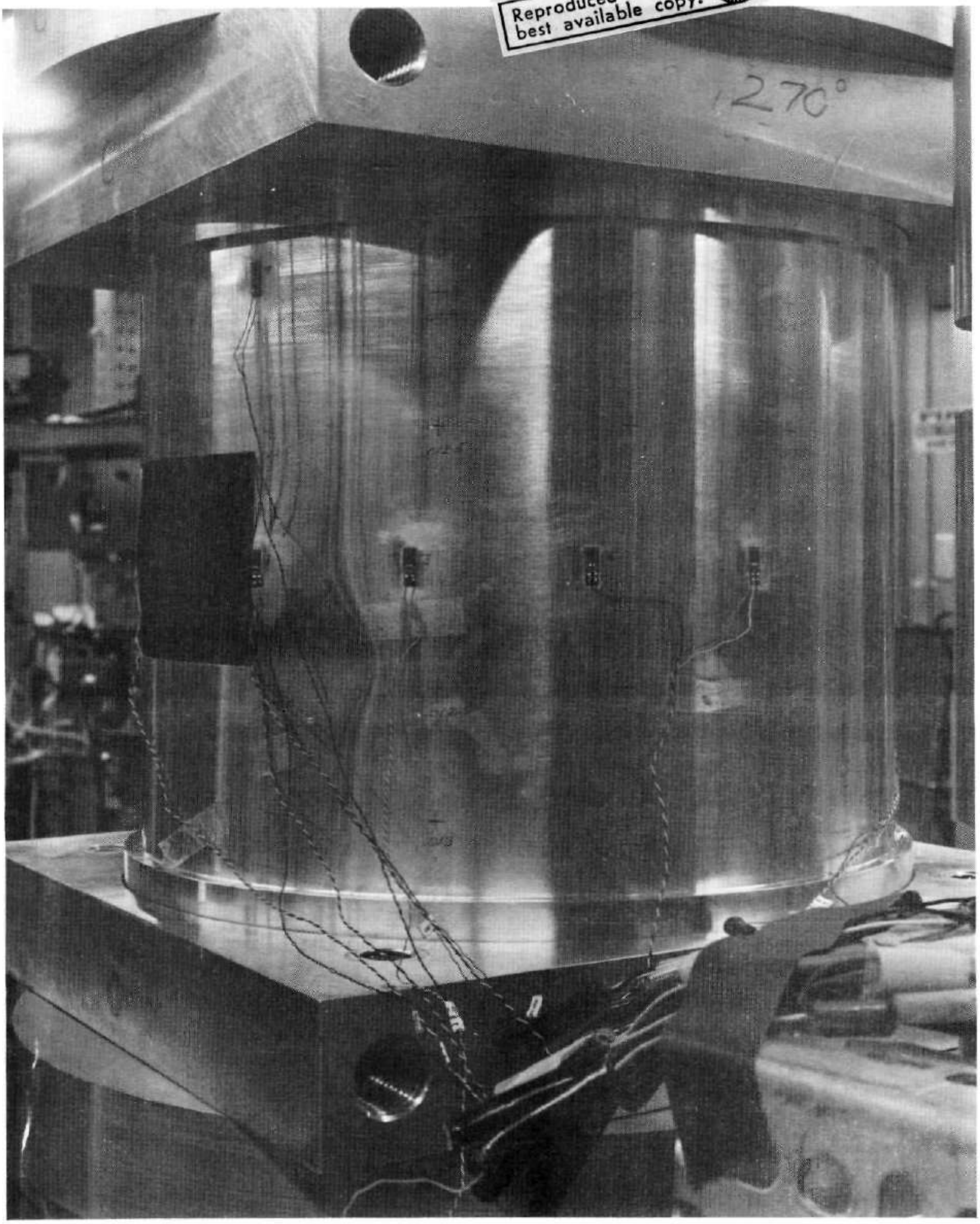

Fig. 3.32 First Buckle at 2050 lbs, Cylinder #3. Cylinder went on to carry 2170 lbs.

Section 4

THEORETICAL RESULTS

Computer analysis was used in connection with this program for two different purposes. Pretest analysis is needed in order that the test specimen will be proportioned to give as much information as possible. Post-test analysis is needed for the enhancement of the understanding of the results obtained from the experiments. To a large degree the same computer runs could be used for both of these purposes and thus separate discussion of pre- and post-test analysis will not be undertaken. The theoretical results will be presented here and their influence on the choice of cutout geometry will be discussed. Correlation of experimental and theoretical results and a discussion of their significance will be presented in Section 5.

The computer program used in the analysis is STAGS, a program for the nonlinear analysis of shells of general shape. STAGS is based on the energy principle in combination with finite difference approximations. A detailed description of the program is given in Ref. 4.

For a thinner cylinder the finite difference grid must be finer and thus the computer time goes up. It appears that the price of the analysis is approximately inversely proportional to the square of the thickness. It is desirable then that the cylinders used in the program be as thick as possible, short of causing problems with inelastic deformations.

The first attempt at analysis was made for a shell with

$$R = 6.06$$
$$t = 0.020$$
$$\text{Cutout: } 30^{\circ} \times 3 \text{ in.}$$

It was found that for such a cylinder, stresses around the cutout would reach the proportionality limit of the material at about half the elastic collapse load. A second attempt was therefore made with a thinner-walled cylinder; i.e., t = .014. The critical load for this cylinder with a 30° cutout was 2650 lbs/in and examination of the stresses indicated that collapse would occur in the elastic range. However, the difference between the buckling load for a cylinder without cutout and one with unreinforced cutout was too narrow to permit a successful study of the efficiency of cutout reinforcement . Therefore, cylinders with 0.014-inch thickness and wider cutouts were also analyzed. The critical load for a 45-degree cutout was found to be 2250 lbs and with a 60-degree cutout it was 1900 lbs. The lateral displacements at the edge of the cutout for these three shells are shown as a function of applied load in Fig. 4.1. The displacement pattern for the cylinder with a 45-degree cutout is shown here in Fig. 4.2 and the distribution of stresses in the same cylinder is discussed in Section 5. Although the results of Ref. 1 provided some guidance, two attempts had to be made before a suitable finite difference grid was established. The grid which finally was chosen is shown in Fig. 4.3. The original plan called for test of two series of shells differing from one another only in shell thickness. As no shells can be thicker than .014-inch, the two nominal thicknesses chosen were t=.014 (R/t=430) and t=.009 (R/t=675).

The first attempt at analysis of shells with reinforced cutouts was made with a shell thickness of .014-inch and a 60-degree by 3-inch cutout. The type of reinforcement chosen was used in the analysis of Ref. 1. A solid rectangular stiffener was attached like a picture frame around the cutout. The computed critical load as a function of the thickness of the reinforcing frame is shown in Fig. 4.4. It is clear that this type of reinforcement is very inefficient for this shell. If the reinforcement is light, the cylinder buckles at the midlength of the cutout edge, and at a load only slightly above the load carried by a cylinder with unreinforced cutout. As the thickness of the reinforcing frame is increased the buckle shifts its location to a region above the corner of the cutout and, still, the increase in buckling strength remains slight. This is because the added area causes a stress concentration at the place where the reinforcement ends. The reason that the solid frame could be used to advantage for the cylinder in Ref. 1 is that that cylinder is so much thicker.

Clearly the reinforcing stiffener at the cutout edge should have bending stiffness but its area should be as small as possible. A thin angle section stiffener therefore appeared superior to one with the solid rectangular section. Also one might conjecture that for the case of axial compression the stiffening along the curved edges of the cutout may be of little value and that it may be better to sacrifice this part of the frame and instead extend the stiffeners in the axial direction. Linear analysis was used in a preliminary study which established the stiffeners selected as suitable (Figs. 2.5 and 2.6). It was also concluded that little would be gained by using a 60-degree cutout rather than one with a 45-degree arc and that the latter would be more representative of practical design. The 45-degree cutout was therefore adopted as the standard for all tests.

Computer results for the collapse load were obtained for three cylinders with 45-degree cutouts and 0.014-inch thickness. Two were of the type with axial stiffeners only; one with a stiffener thickness of 0.010-inches and one with a thickness of 0.020-inches. The third reinforcing configuration had a picture frame reinforcement (Fig. 2.7) with an angle of 0.020-inch thickness. These reinforcement configurations were then used in the test program. The higher stresses which can be reached in the shell with reinforced cutout makes it necessary to use a finer finite difference grid. The grid selected for analysis of these cylinders has 22 axial and 25 circumferential coordinate lines as shown in Fig. 4.5. For the cylinders with stringer reinforced cutouts, the maximum displacement shifts away from the cutout edge to a point about 4 degrees of arc from the edge as the load increases. For the two cylinders with axial reinforcement only, the displacements at this point are shown as a function of the axial load in Fig. 4.6. In Fig. 4.7 an attempt has been made to show how the critical load varies with the thickness of the reinforcement. The data points available are too few to indicate more than the trend. It seems clear, however, that the arrangement with only axial stiffeners is definitely superior.

Additional theoretical results were obtained for somewhat thicker cylinders, $R/t = 200$, with the same cutout. The effect of the size of a stringer reinforcement (Type A) was studied and the results are shown in Fig. 4.9. The grid used in this analysis contained 15 axial and 21 circumferential stations. From these results it can be concluded that the effect of an unreinforced cutout is somewhat more severe for thinner cylinders. With $R/t = 200$ the cutout reduces the critical load to 41.8% while for a shell with $R/t = 430$ the corresponding value is 30.5%.

Figure 4.10 shows a comparison of reinforcement efficiency for the two cylinders. The thinner cylinder responds quicker to small reinforcements than the thicker cylinder, but the thicker cylinder is somewhat more efficient. However, in both cases we can obtain little more than half the buckling load of the complete cylinder.

Of the thinner cylinders ($R/t = 675$) only one was analyzed as the computer time is very high for such shells. The reinforcement chosen for the analysis was type "C" (see Fig. 2.6) with an angle stiffener which has an outstanding leg with a reduced height of 0.080 inches. For reasonable accuracy in the results, it is necessary to use a very fine grid but the chosen grid with 28 axial and 33 circumferential stations appears to be satisfactory. This stiffener is so weak that the maximum displacement still occurs at the cutout edge. This displacement as a function of load is shown in Fig. 4.8.

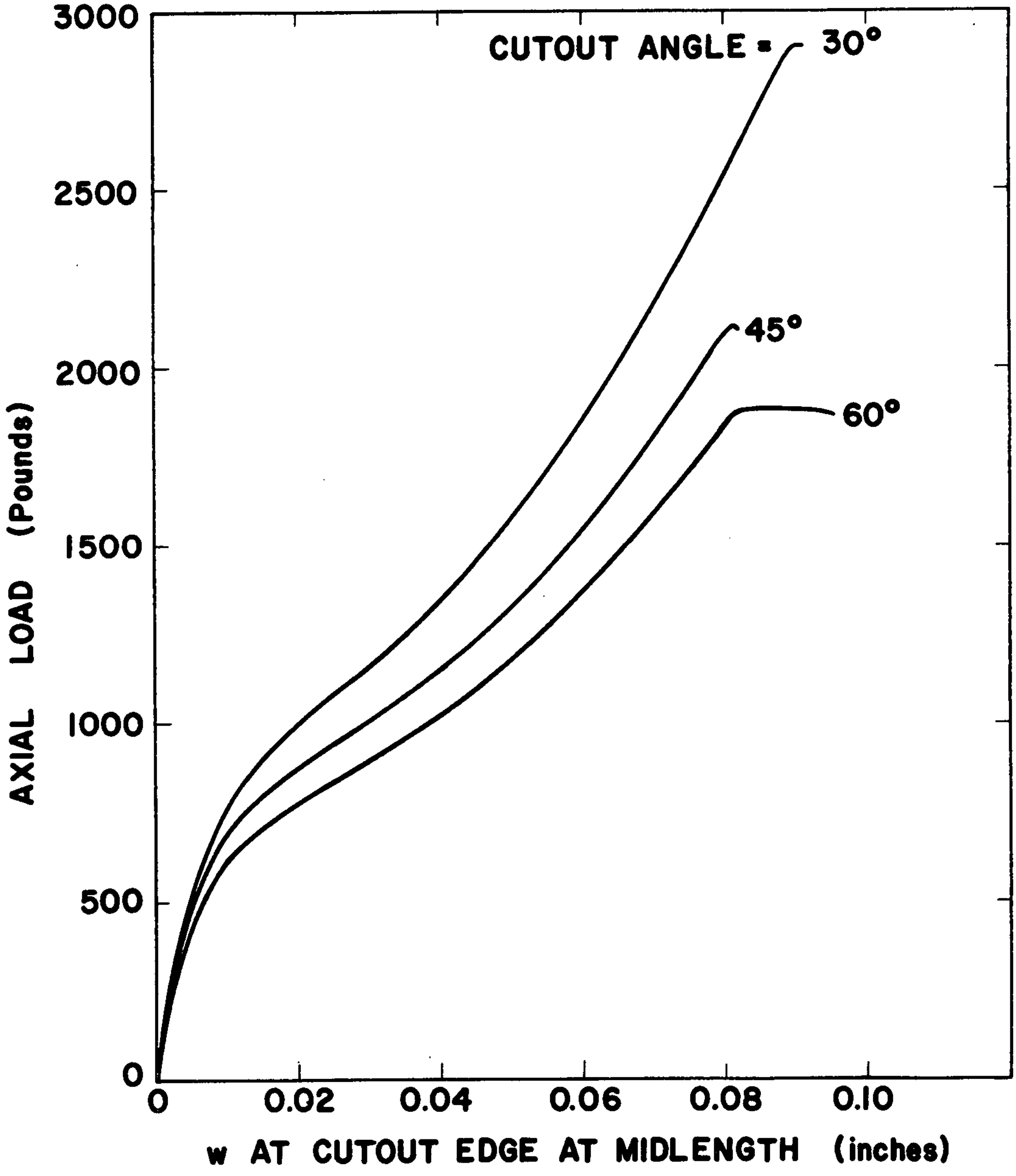

Fig. 4.1 Load-Displacement Curves for Cylinders with Unreinforced Cutouts ($t = 0.014$)

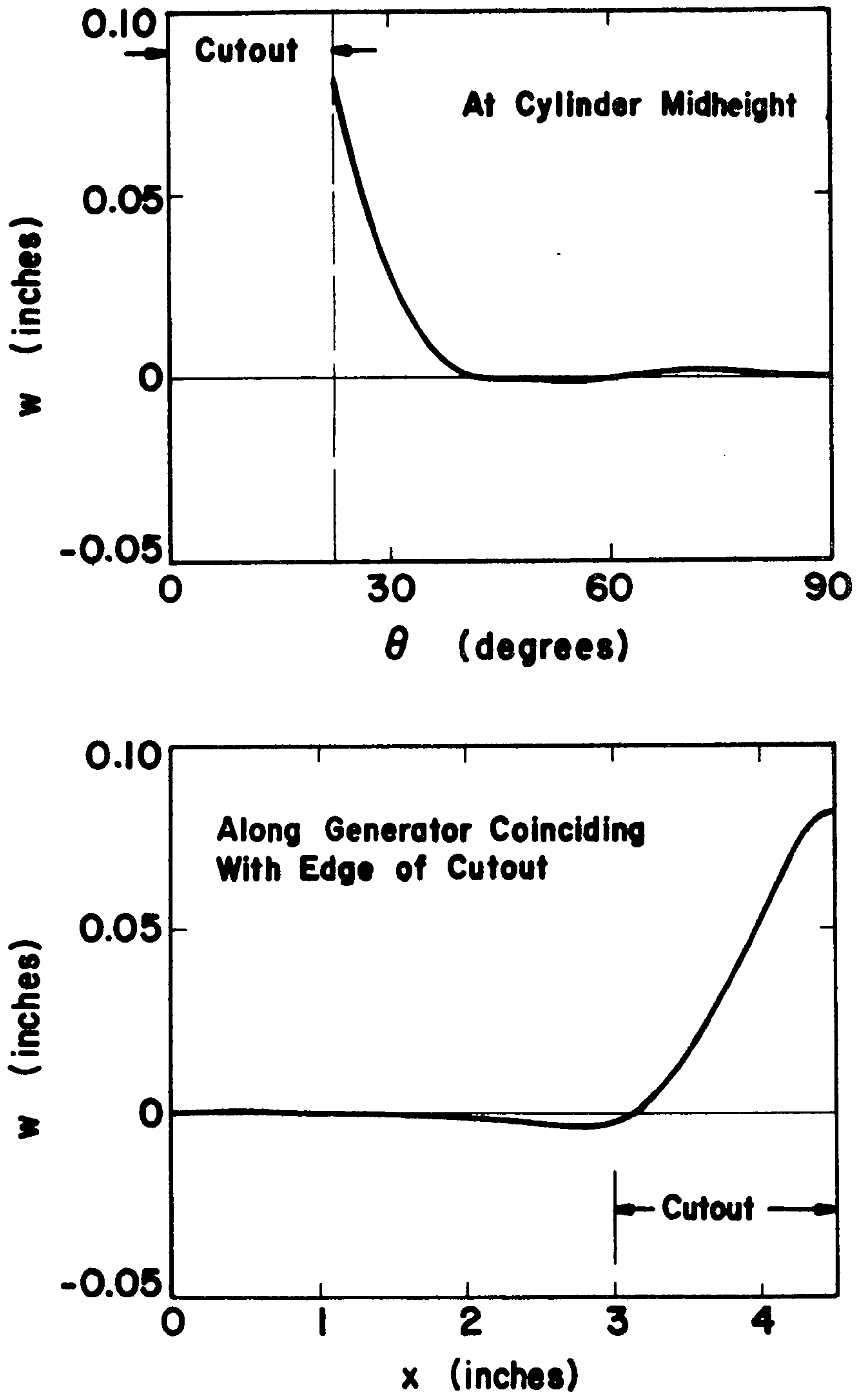

Fig. 4.2 Displacement Patterns with 45-Degree Unreinforced Cutout (t = 0.014)

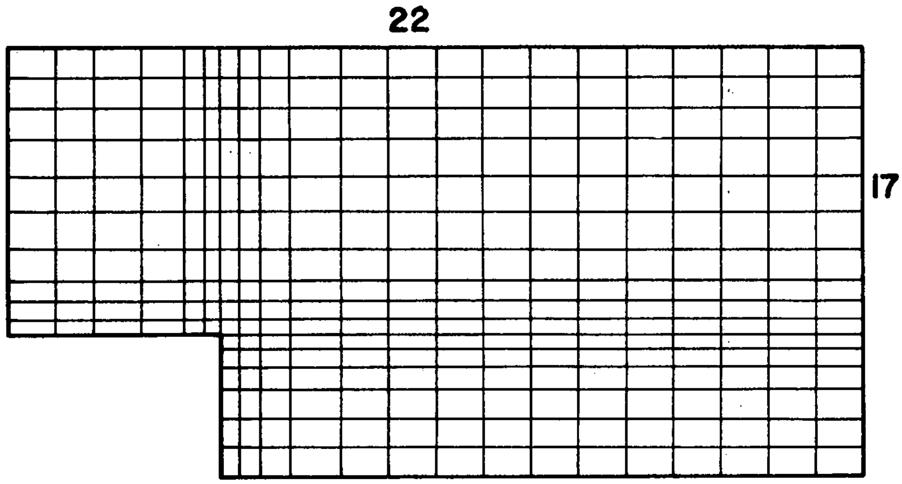

Fig. 4.3 Finite Difference Grid for Cylinder with 45-Degree Cutout

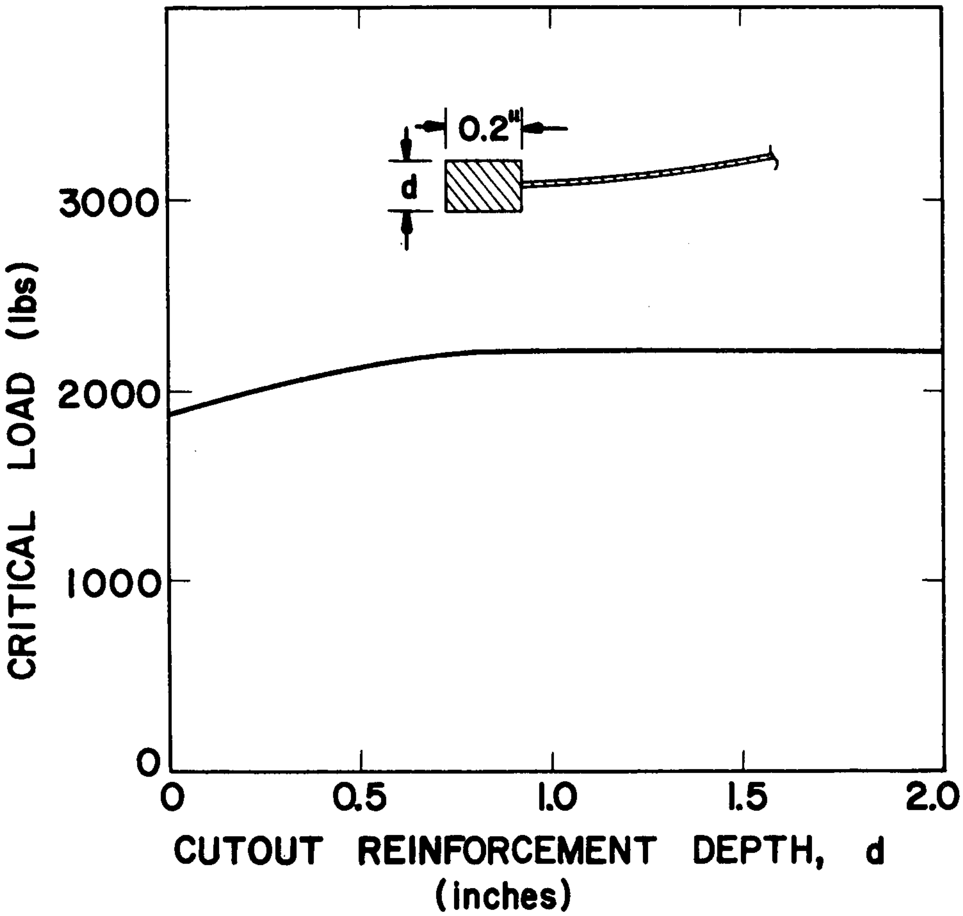

Fig. 4.4 Effect of Solid Reinforcement for 60-Degree Cutout (t = 0.014)

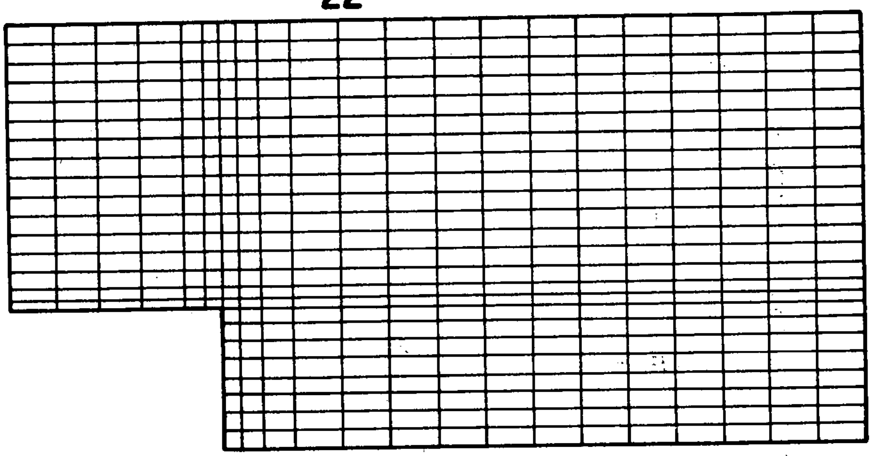

Fig. 4.5 Finite Difference Grid for Cylinder with 45-Degree Reinforced Cutout

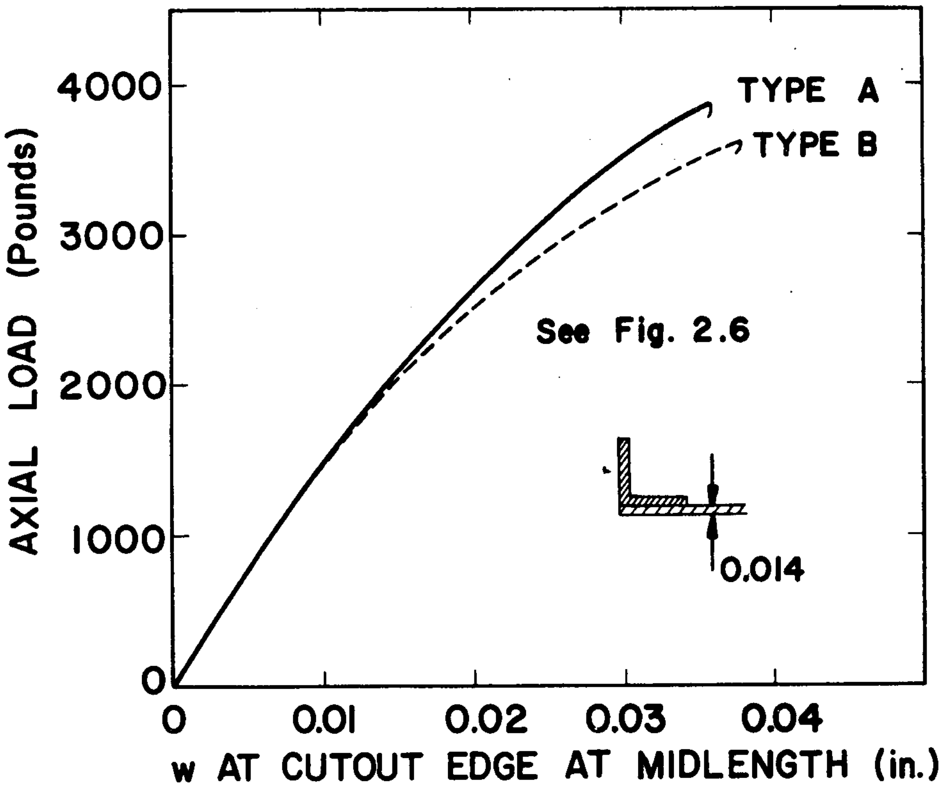

Fig. 4.6 Load-Displacement Curves for Cylinders with Stringer Reinforced Cutouts

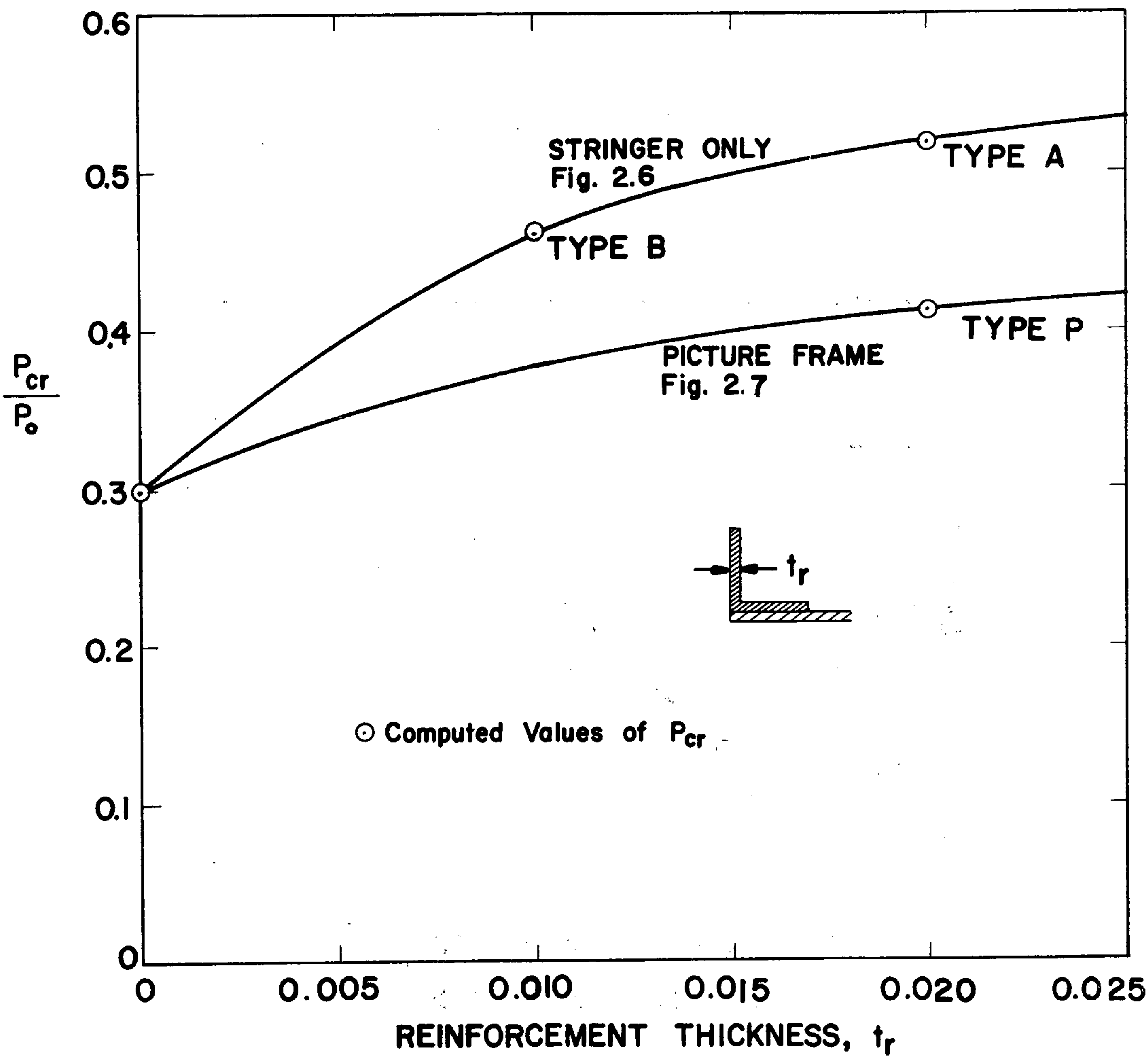

Fig. 4.7 Effect of Reinforcements Around 45-Degree Cutout (t = 0.014)

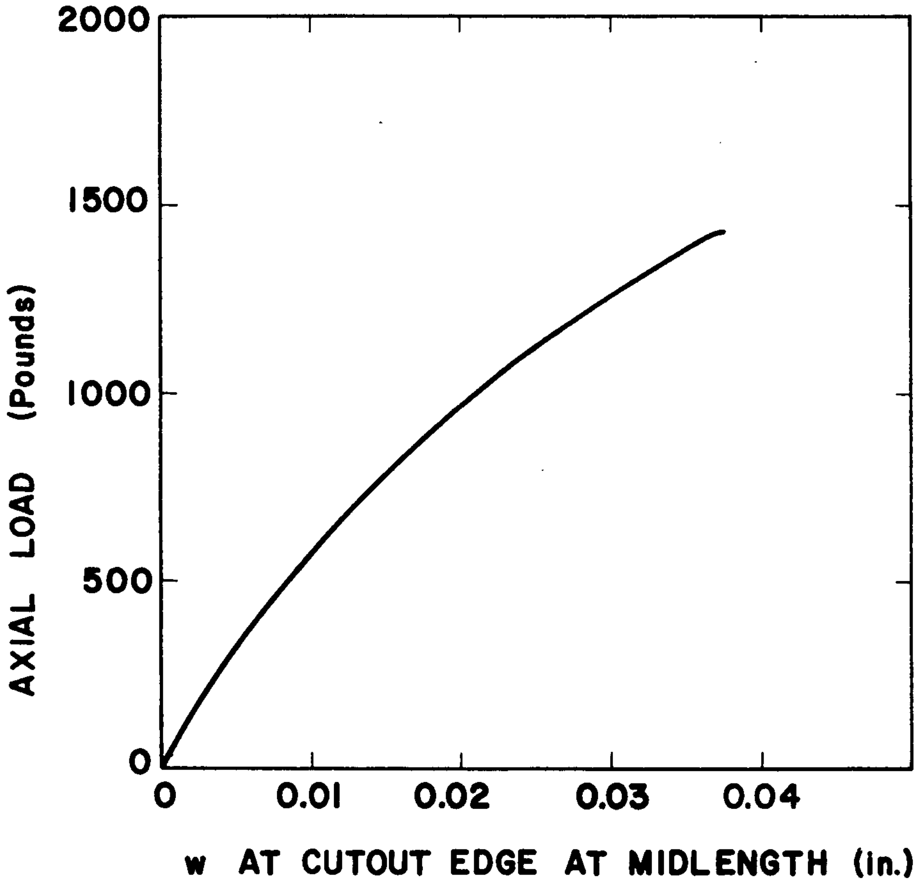

Fig. 4.8 Load-Displacement Curve for Cylinder #10 (R/t = 675)

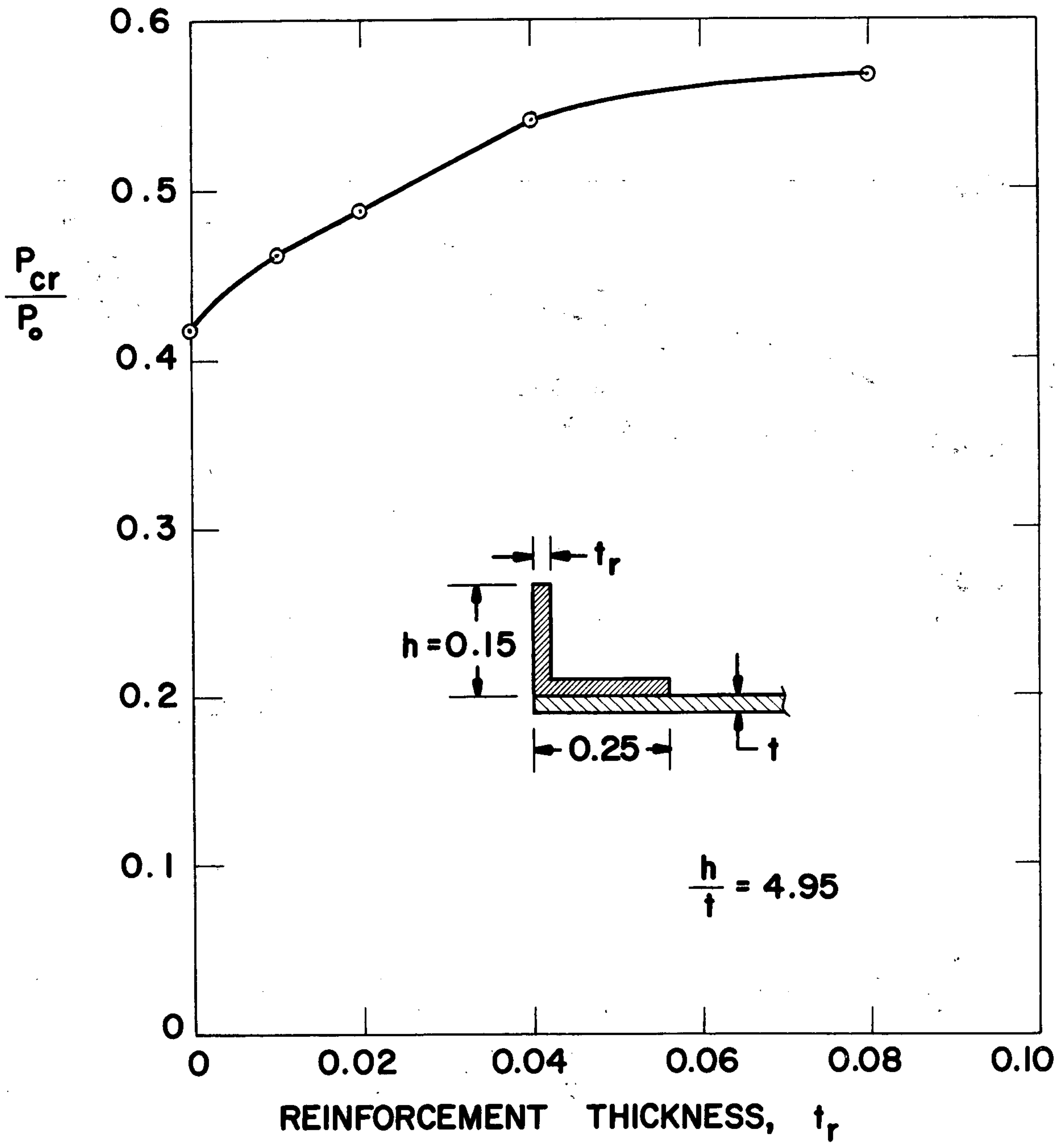

Fig. 4.9 Effect of Reinforcement Thickness Around 45-Degree Cutout (R/t = 200, t = .0303)

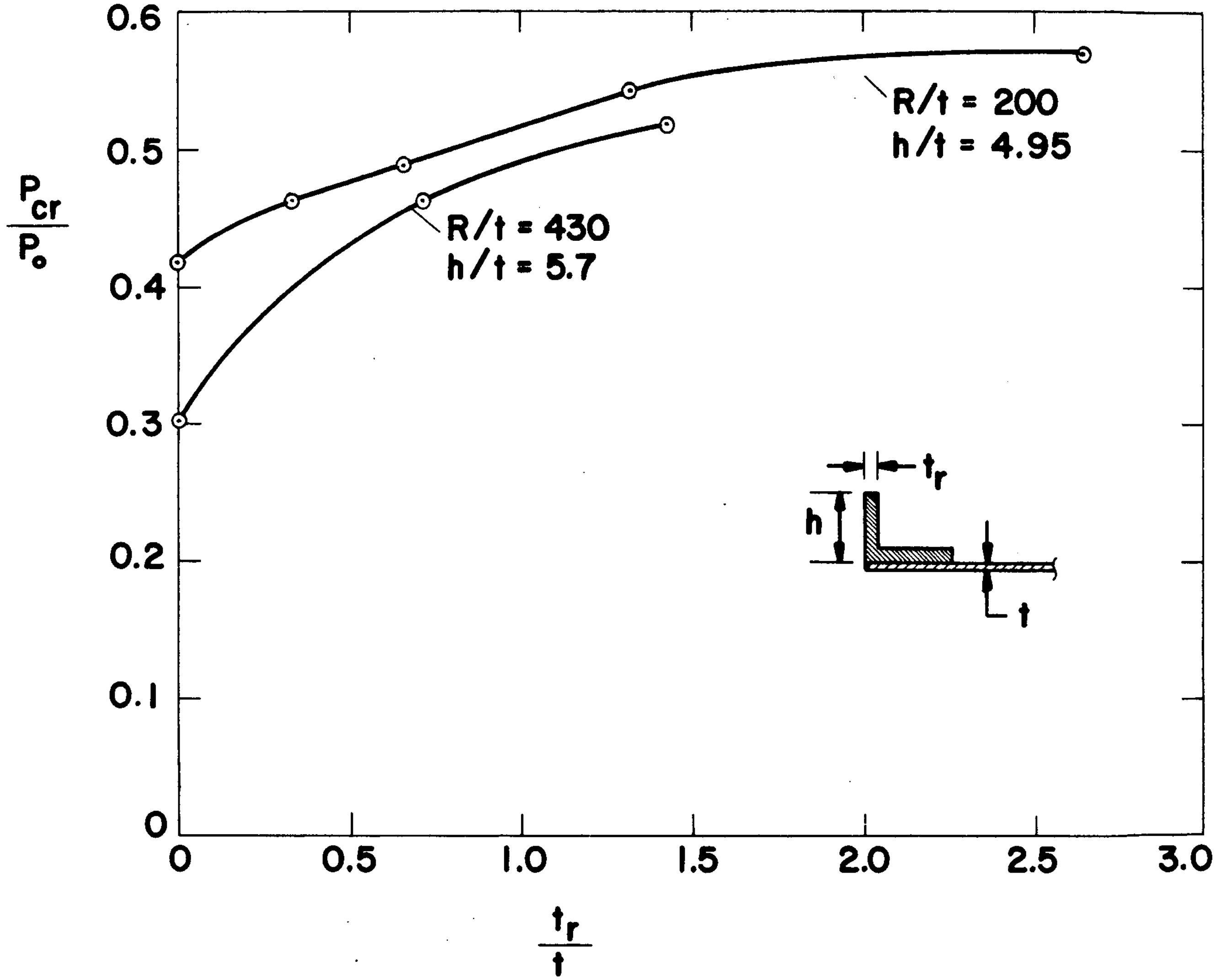

Fig. 4.10 Comparison of Reinforcement Efficiency for Cylinders with R/t = 200 and R/t = 430

Section 5

CORRELATION

The extensive strain measurements for Cylinder #2 (with thickness 0.014 inches and unreinforced cutouts) offers a good opportunity to compare theoretical and experimental results and thus verify the validity of the computer program. The solid lines in Figs. 3.1, 3.2 and 3.3 represent computed stresses. The points are the stress values determined by use of the strain gages.

Figure 3.1, which shows axial membrane stress 0.30 inches from the end ring, indicates very good agreement between test and theory at all load levels. The agreement deteriorates somewhat as we move away from the cutout. The reason for this appears to be that the theoretical results are for a cylinder with a constant 0.014-inch thickness while the thickness of the actual test cylinder tended to increase to 0.015 or 0.016 inches. In Fig. 3.2, which shows the axial membrane stress at the cylinder midlength, the trend is about the same. At the edge of the cutout the agreement between experimental and theoretical stresses is exceptionally good. Away from the cutout the measured stresses tend to be somewhat lower than computed stresses because the thickness in this area is above nominal.

Bending stresses are generally so small that the dominating influence on these are the small imperfections in the shape of the test cylinder. Only at the edge of the cutout are these stresses big enough to make a comparison between test and theory meaningful. The axial direction bending stresses at the cylinder midlength and close to the cutout edge are shown in Fig. 3.3. Here the agreement is seen to be relatively poor for small load levels, where the influence of imperfections is dominant, but improves with increasing load.

Figs. 5.1 through 5.4 show a comparison between theory and measured membrane strains for Cylinders #4, #6, #7 and #11, respectively. On several of these cylinders some of the gage stations were placed symmetrically around the cutout, so that as many as four experimental records are available for one given location. The term "Location A" is thus used to indicate position relative to the symmetrical axes of the cutout or cylinder. For Locations "A" and "C", the data from the various stations on all the cylinders scatter about evenly above and below the theoretical curve, and agreement is thus generally good. At Location "B", the agreement is not as good, but it should be pointed out that there is a very steep stress gradient in this region (see Fig. 3.2), so that the placement of the gage is very critical, or conversely, measurements have a high probability of being "off" because of minor gage misplacement. Taking this into consideration, it is felt that agreement between test and theoretically predicted membrane strains is very good for the four reinforced cylinders covered.

For Cylinders #2 and #3, a reversal occurred in the trend of the bending moment at the cutout edge before the ultimate load was reached. For Cylinder #3, a small buckle which formed at the lower corner of one of the cutouts, was observed just above the load at which the bending moment reversal occurred. A photo of this buckle is shown in Fig. 3.32. The experimental value of the bending strain at one of the cutout edges on Cylinder #2 is shown as a function of the applied load in Fig. 5.5. Fig. 5.6 shows the same graph for Cylinder #3 at all four meridional cutout edges. We feel that the point of the bending stress reversal is the proper load level to compare with the theoretical collapse loads. For Cylinder #2 the theoretical load is then 2250 lbs. and the experimental load is 2200 lbs. Cylinder #3 is somewhat thinner; in the neighborhood of the cutout the thickness was 0.13 inches. If the collapse load is assumed to be proportional to the square of the thickness, the thickness corresponding to the test failure load of 2000 lbs., is 0.0132 inches, which agrees well with the measured thickness. For Cylinder #1 with a 30-degree cutout no stress reversal was observed before collapse. The critical load of 2740 lbs. compares well with the computed load of 2900 lbs. (The thickness varies in the neighborhood of a cutout between 0.014 and 0.015 inches.) In Fig. 5.7 the critical load is plotted as a function of the width of the cutout. In addition to the analytical results for 30, 45 and 60-degree cutouts, we know of course the critical loads for 0-degree and 180-degree cutouts. Due to the limited number of points the curve is rather uncertain, particularly for cutouts between 0 degrees and 30 degrees.

It is seen that in cylinders with reinforced as well as unreinforced cutouts theory and experiment agree very well on the stress distribution. In addition, for cylinders with unreinforced cutouts the theory predicts quite accurately the collapse load. In the case of cylinders with reinforced cutouts, it is evident that a reinforced cutout constitutes less of an imperfection than was generally found in these cylinders, so that a knock-down factor based on the imperfection level has to be applied to the computer based nonlinear analysis. This agreement between test and theory is encouraging and is one of the most important conclusions of the program. It indicates that it would be possible to make extensive studies of the efficiency of cutout reinforcement designs primarily on an analytical basis.

It is useful to note that we obtain a reasonably good approximation to the effective axial stress level by dividing the total load by the cross-sectional area of the cylinder which remains after the cutout is introduced. One should be cautioned that this remark, as well as the following observations, apply only to the situation in which the load is applied by constant end shortening. This accurately represents the test conditions, and is applicable to many practical problems as well (e.g., collapse of a section of a launch vehicle contained between two large bulkheads). However, cylinders to which a uniform axial edge load is applied will behave quite differently (the interior stress distribution is highly nonuniform and the collapse load will be lower than for the same shell with constant end shortening); such cases have not been studied extensively and are beyond the scope of the present effort.

The maximum stress σ_{cr} which the cylinder can sustain (under constant end shortening), even if the cutout is adequately reinforced, is the critical stress for a complete cylinder. In view of the sensitivity of axially loaded cylinders to geometrical imperfections, a cylinder without a cutout has a critical axial stress of

$$\sigma_{cr} = \phi \, \sigma_o \tag{5.1}$$

where ϕ is a knock-down factor tied to a probability level depending on the quality of the cylinder, and σ_o is the classical buckling stress for a perfect cylinder without a cutout, i.e.

$$\sigma_o = 0.6E\ t/R \tag{5.2}$$

Thus the maximum load the cylinder can sustain is the critical stress times the net area (assuming two equal unreinforced cutouts 180 degrees apart)

$$P_u = \frac{180 - \alpha}{180} \phi P_o = \psi P_o \tag{5.3}$$

where

$$\alpha = \text{angular arc of cutout}$$

$$P_o = 2\pi Rt\ \sigma_o$$

The validity of this approximation was established by both the theoretical and experimental work of this program. This method appears to be valid for the case of cylinders with reinforced cutouts if the area of the reinforcement is added to that of the remaining cylinder. If there is only one cutout the average stress may be somewhat lower, but tentatively it is recommended here that the same equation be applied to cylinders with one cutout.

If the reinforcement around the cutout is inadequate or nonexistent, the shell may collapse at a load significantly less than the upper bound P_u given by Eq. (5.3). This collapse load P_{NL} must be determined by a nonlinear analysis. The critical load P_{CR} for the shell is then the smaller of the two loads P_{NL} and P_u.

For a given value of the quality parameter ϕ there is a maximum size of a cutout that can be left unreinforced without reduction of the critical load. This relationship is shown in Fig. 5.8, and is based on computer runs for 30, 45 and 60-degree cutouts. It is also based on the fact, already stated, that some imperfections in the complete cylinder lower the buckling load more than do some cutouts. It is stressed here that because the investigation was not extensive enough, these are only tentative suggestions, and that there is a lot of scatter in the test data for cylinders with low values of the quality parameter ϕ . For instance, if $\phi = 0.41$, it means that only one percent of the cylinders tested will have a critical load less than $0.41\ P_o$.

If a 30-degree unreinforced cutout is made in such a cylinder, the test results will be concentrated around $\psi P_o = \frac{150}{180} \times 0.41 P_o = .34 P_o$. Introduction of reinforcement will not change this lower bound, or the 99% probability limit, but the average of several such tests may be considerably above $0.34 P_o$.

As the value of ϕ was determined for all test specimens before any cutouts were introduced, it is possible to obtain a preliminary evaluation of this method by application to all cases for which theoretical as well as experimental results are available. Such an evaluation is made in Table 5.1. Since it is difficult to take the variable thickness into account and since many of the computer runs were made before the cylinders were manufactured, all calculations here are based on nominal values of the thickness. In view of the thickness variation in any given shell, this approximation is not inappropriate. However, more analysis and additional experiments are needed before this method could be considered an established design procedure. As might be expected, the nonlinear analysis value provides the critical load for all shells with unstiffened cutouts (#1, #2, and #3). However, in spite of the very light stiffening used in some case, P_u is critical in all specimens with reinforced cutouts. Any future work should therefore be on cylinders that have even lighter cutout reinforcement and a higher value of the quality parameter ϕ.

TABLE 5.1

CORRELATION BETWEEN TEST AND THEORY

Specimen Number	ϕ	ψ	P_u	P_{NL}	P_{CR}	P_{EXP}
1	.545	.455	3360	2900	2900	2740
2	.620	.465	3440	2250	2250	2250*
3	.578	.435	3210	2250	2250	2000*
4	.503	.375	2780	3700	2780	3190
5	.538	.403	2980	3700	2980	2850
6	.455	.34	2500	3500	2500	2560
7	.413	.31	2290	3100	2290	2600
10	.45	.338	1030	1400	1030	1030

Key - ϕ, ψ and P_u see Eqs. 5.1 and 5.3

P_{NL} theoretical buckling load from nonlinear analysis of perfect shell with cutout

P_{CR} predicted buckling load (minimum of P_u and P_{TH})

P_{EXP} experimental buckling load

*Load at which bending strain reversed; this is somewhat lower than total collapse load shown in Table 3.1.

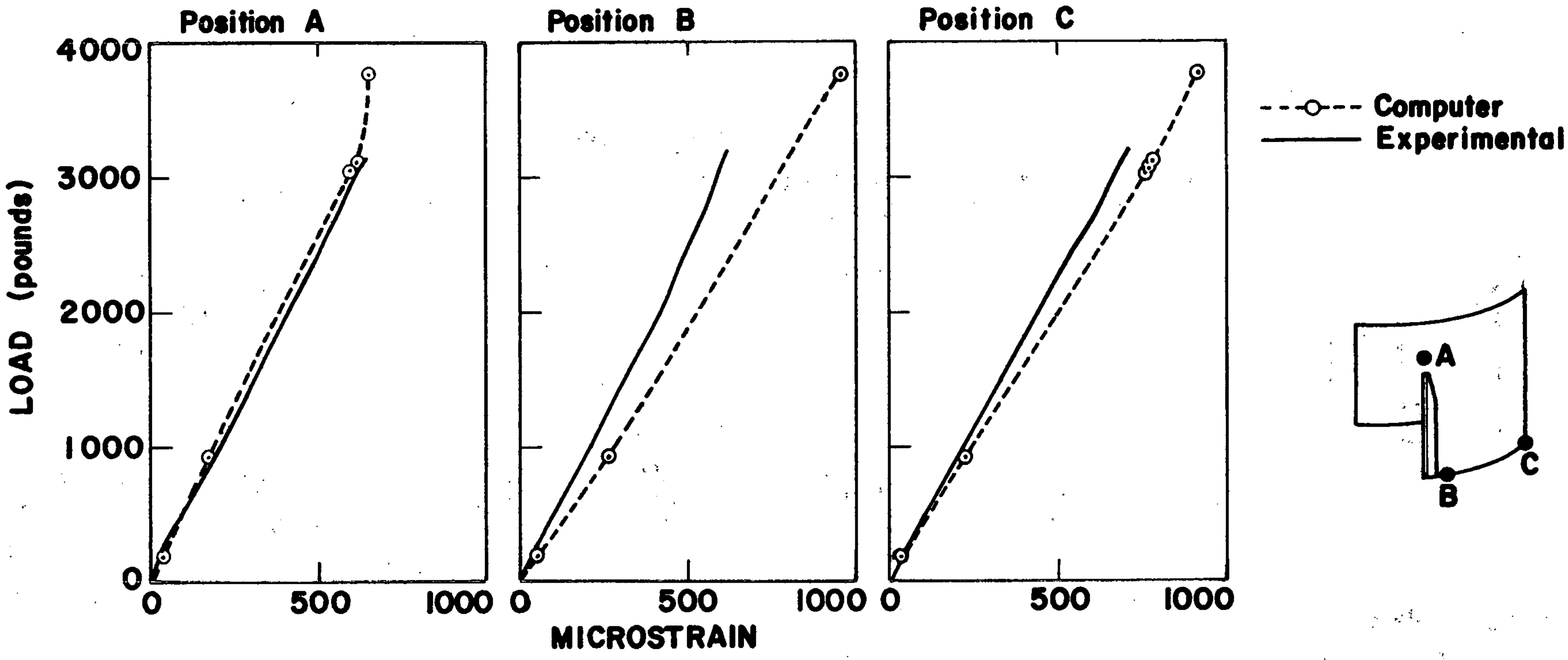

Fig. 5.1 Measured and Computed Membrane Strains in Cylinder #4

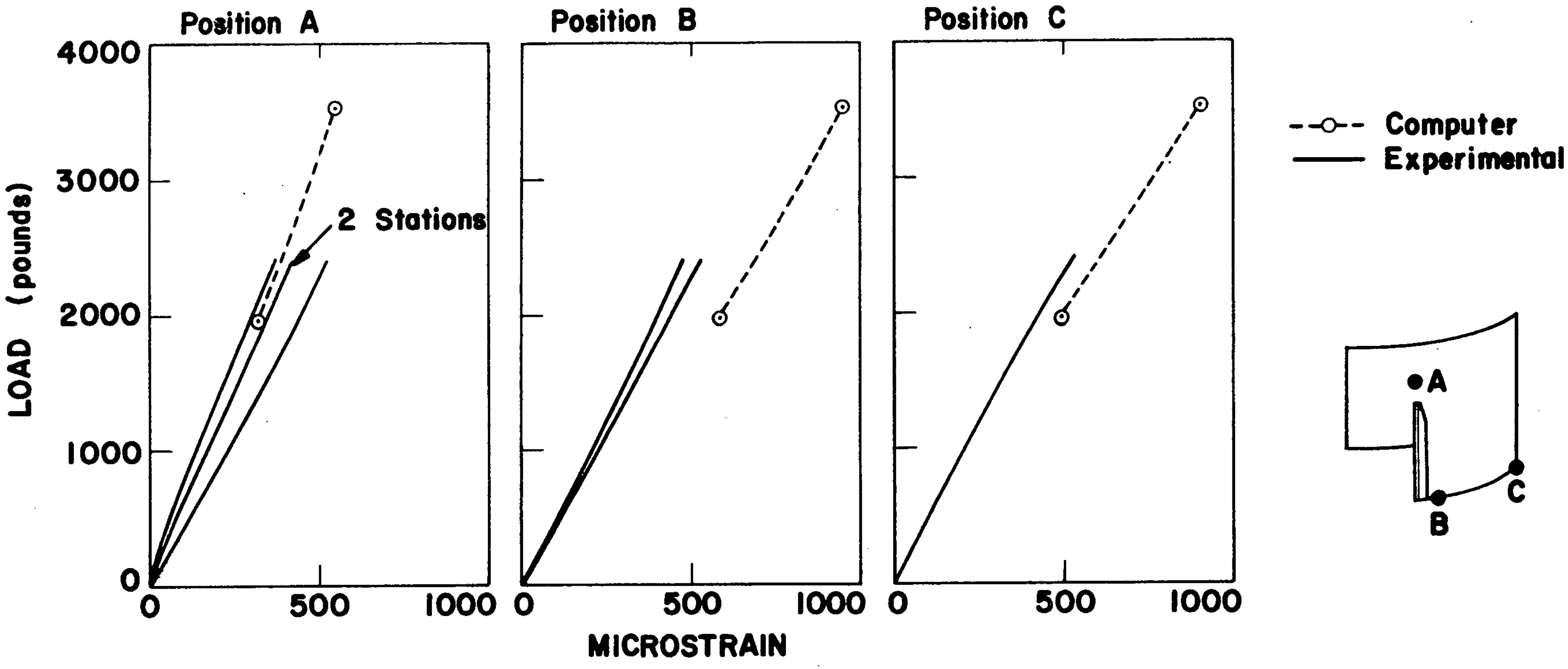

Fig. 5.2 Measured and Computed Membrane Strains in Cylinder #6

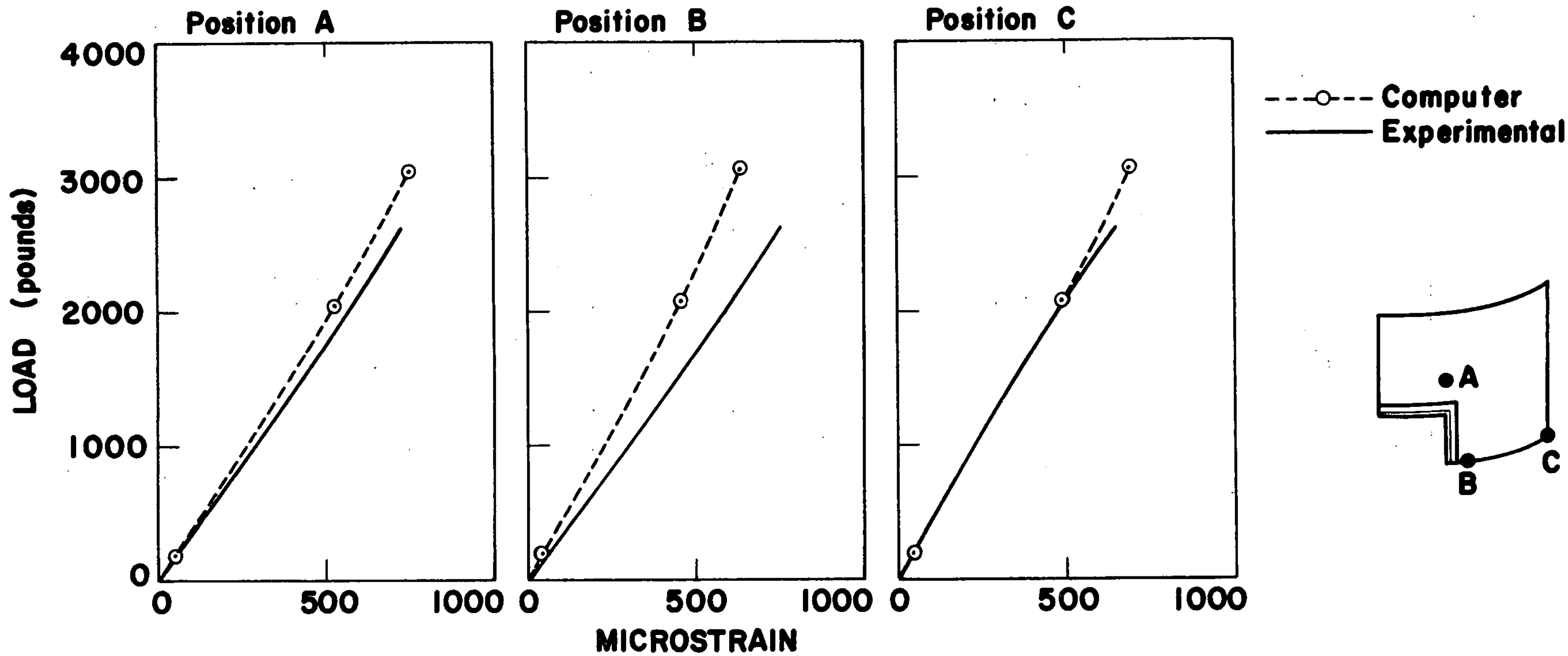

Fig. 5.3 Measured and Computed Membrane Strains in Cylinder #7

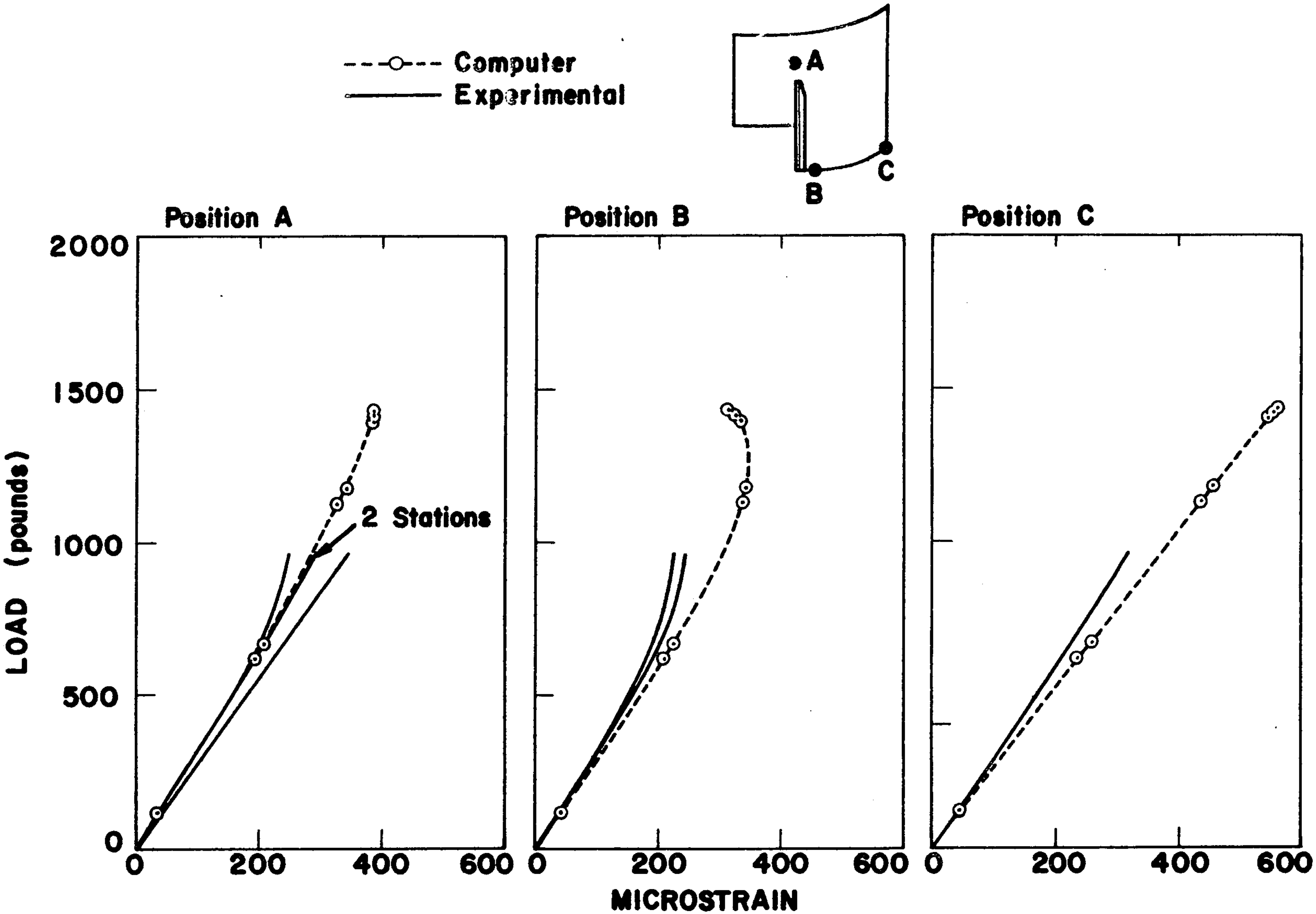

Fig. 5.4 Measured and Computed Membrane Strains in Cylinder #11

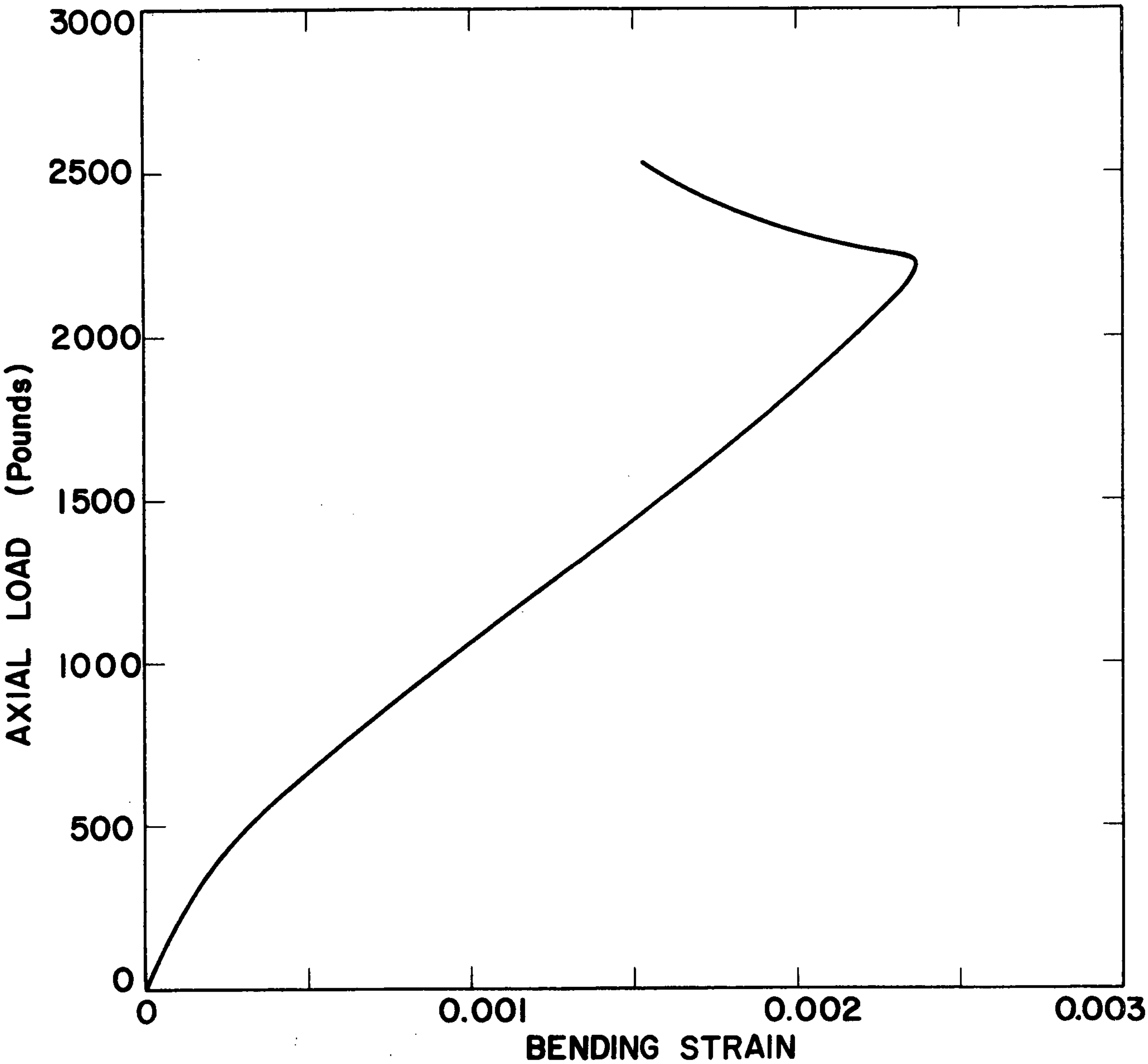

Fig. 5.5 Bending Strain Reversal at Edge of Cutout and Cylinder Midheight
(Test Data for Cylinder #2)

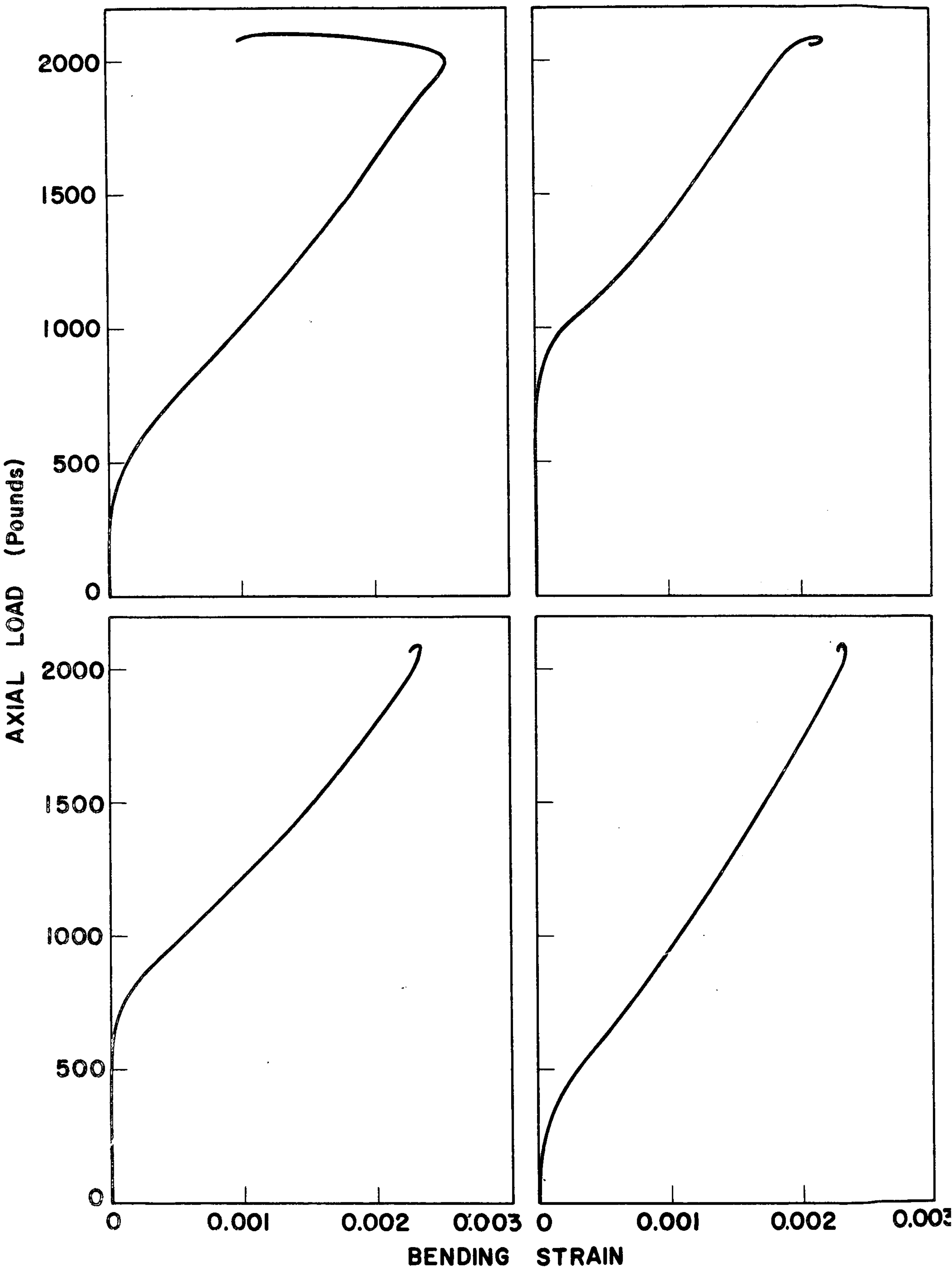

Fig. 5.6 Bending Strain Reversal at Cutout Midheight (Two Location at Each of Two Cutouts, Cylinder #3)

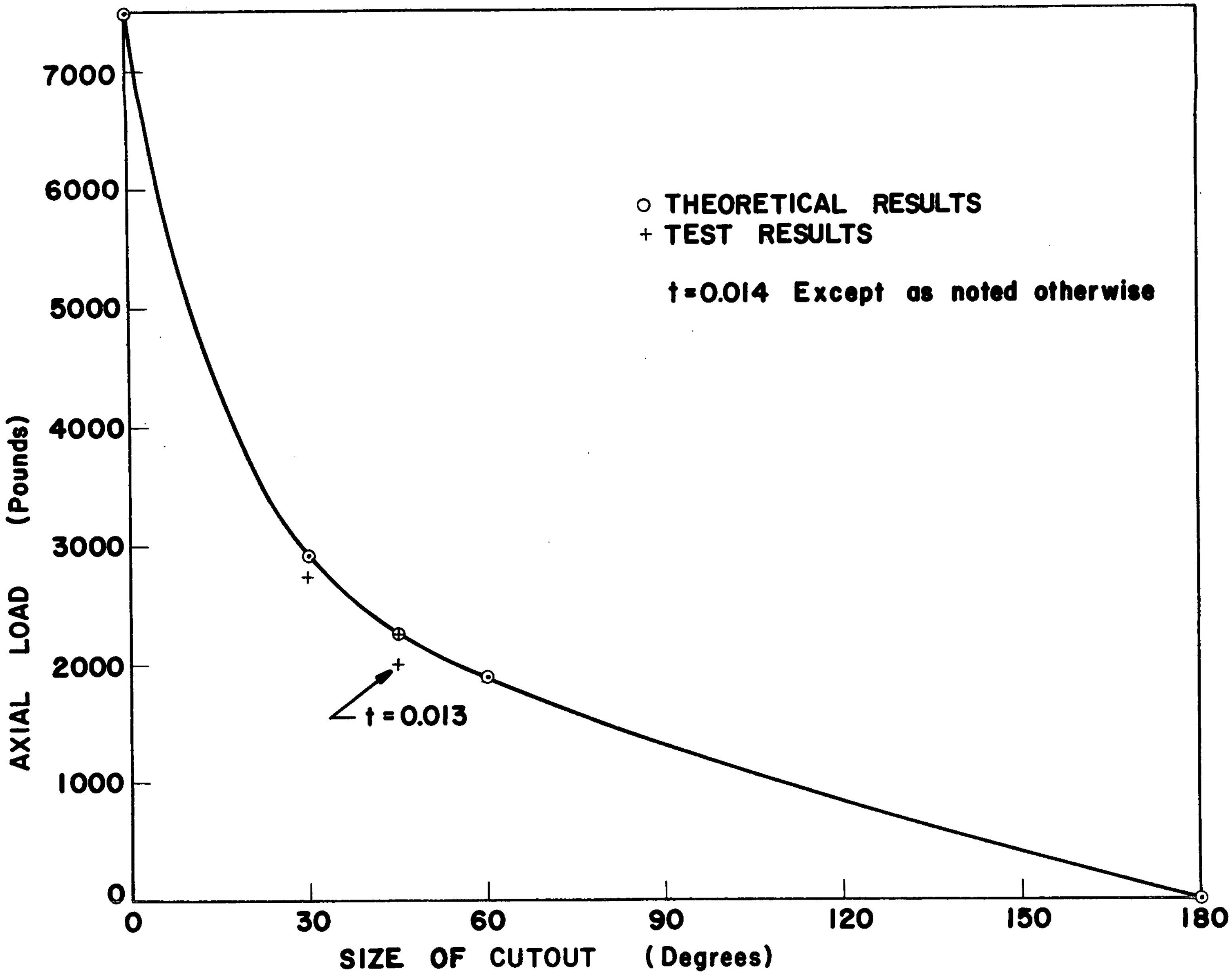

Fig. 5.7 Critical Load vs. Cutout Angle

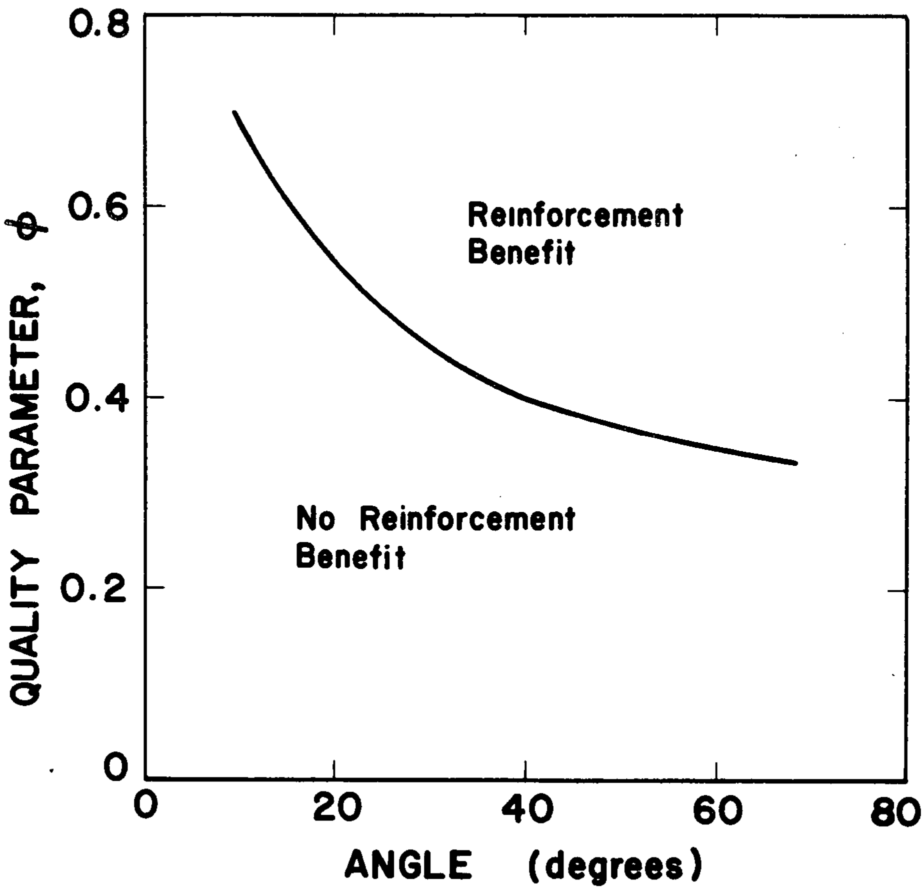

Fig. 5.8 Reinforcement Benefit as a Function of Cutout Arc and Cylinder Quality Parameter

Section 6

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