IMPLOSIVE DEVICES FOR LONG-RANGE UNDERWATER SIGNALLING

by

HENDRIKUS W. TINKELENBERG

B.S., The Massachusetts Institute of Technology
(1962)

SUBMITTED IN PARTIAL FULFILLMENT

OF THE REQUIREMENTS FOR THE

DEGREE OF MASTER OF

SCIENCE

at the

MASSACHUSETTS INSTITUTE OF

TECHNOLOGY

February, 1966

Signature of Author...
Department of Meteorology, September 20, 1965

Certified by...
Thesis Supervisors

Accepted by..
Chairman, Departmental Committee on Graduate Students

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ABSTRACT

This report deals with the work done to determine the feasibility of long-distance underwater signalling by means of implosions caused by pressure-activated rupture discs. The design and use of a laboratory instrument to study the behavior of rupture discs under pressure is described. The experiments performed with this instrument and their results are reported. A field experiment was conducted to determine the amount of energy obtained from implosion devices as well as the distance over which they can be heard. In the course of the experiments in the laboratory and in the field it was found that the pressure at which rupture occurs is determined by the depth of two mutually perpendicular grooves cut into one face of each disc, and that at a depth of 2500 feet (1100 psi) an implosive volume of 3 cubic feet is equivalent in potential energy to 1 lb. of TNT. It is also shown that the use of neutrally-buoyant floats equipped with implosive devices to study deep currents in the ocean is perfectly feasible.

Thesis Supervisors:	Henry M. Stommel	Delbar P. Keily
Title:	Professor of Meteorology	Associate Professor of Meteorology

ACKNOWLEDGEMENT

The author wishes to express his gratitude to Professor Henry M. Stommel and Professor Delbar P. Keily. The author also wishes to express his gratitude to the Office of Naval Research for the financial aid that made the project possible; to the Geophysical Field Station of Columbia University on St. David's, Bermuda, for their hospitality and the use of their facilities; to the Bureau of Commercial Fisheries in Honolulu, Hawaii, for their assistance and the hospitality enjoyed aboard the R/V Townsend Cromwell; to the personnel of the Pacific Missile Range whose splendid cooperation helped make the field experiment a success; and to the personnel of the M.I.T. Weather Radar Research Project whose craftmanship helped make the laboratory experiments successful. Finally, the author wishes to acknowledge his indebtedness to Mrs. Jane McNabb and Mrs. Jutta Thorne whose typing made this report a reality.

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I. INTRODUCTION

The neutrally-buoyant float has been an accepted oceanographic instrument for nearly a decade (Stommel, 1954). The first practical floats were designed and built by J. C. Swallow who used them quite successfully in May and June of 1955 (Swallow, 1955). Since then they have been used profitably by Swallow and others in various studies of the deep circulation in the ocean (Swallow, 1957; Swallow and Hamon, 1959; Swallow and Worthington, 1961).

As developed by Swallow, a neutrally-buoyant float consists of a long thin aluminum tube, closed at both ends, and of such a design that under the influence of hydrostatic pressure the float as a whole is less compressible than seawater. If such a float is adjusted to have a small amount of negative buoyancy at the sea surface, it will sink when set overboard. As it descends it will become compressed due to the hydrostatic pressure and its density will increase. Since seawater is more compressible than the float, the density of the water surrounding the sinking float will increase at a greater rate. Thus at some depth the float obtains a small amount of positive buoyancy with respect to its surroundings and will continue to drift at this depth. A float carries batteries and electronic gear inside, a transducer outside, and emits signals at fixed pre-determined intervals.

A ship at the surface can track the progress of the float by these signals.

With electronic signalling the useful range of even very large floats is of the order of miles or at best a few tens of miles. Thus a float is useful only as long as a ship is available to track it. Whereas in theory it seems quite possible to track a number of floats simultaneously with the ship, in practice only one float can be tracked successfully for any worthwhile length of time. It is obvious that to obtain a significant measure of the distribution in space and time of the deep currents at some location in the ocean using neutrally-buoyant floats, the cost of such data in ship-hours and man-hours expended becomes prohibitive. Since, however, the concept of neutrally-buoyant floats itself represents both a great advance in the mapping of deep currents and an economical tool, it became clear quite early to oceanographers that it was the method of tracking the floats that needed to be improved.

The discovery of the SOFAR channel* in the ocean and the important developments made in the use of underwater sound during the second World War showed several people in the field the way to an elegant tracking system. Differing only in details, they conceived of a shore-based network of SOFAR listening stations and a multitude of neutrally-buoyant floats released either by planes or

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* See Appendix A.

ships to drift at many different locations and depths in the ocean. At regular intervals each float would create a signal of sufficient strength to be recorded at the listening stations. If at least three stations received such a signal an accurate fix on the float's position could be made and its progress plotted. In these schemes the timing of the signals is of great importance. An accurate timing mechanism is necessary so that a float may be identified by its unique time of reporting and also to prevent signals from arriving simultaneously and masking each other.

The type of signal to be used and the means to activate it are important considerations. Several ideas have been proposed. Stommel (1954) envisioned a parent float which would release an explosive sinker and a compensating float to restore neutral buoyancy at regular intervals. This would require only one accurate timing mechanism. Another method would set adrift a cluster of neutrally-buoyant floats tied together by a neutrally-buoyant string. At regular intervals, one float would release itself, flood a previously air-filled compartment to sink, or release some ballast to ascend into the SOFAR channel, and generate a signal to indicate the position of the cluster. In this plan more than one clock might be necessary, but the extra payload represented by the compensating floats would be avoided.

For the development of the air-sea rescue network which was one of the first practical applications of SOFAR, the only satisfactory long-range signalling devices were high energy explosives such as TNT. Since it was clear to oceanographers that for ocean-wide signalling electronic methods would be far too inefficient, it is not surprising that the early proposals referring to an ocean-wide Swallow float tracking system take the use of explosives for granted. There is, however, a great difference, both with respect to engineering requirements and moral or political considerations, between a one-shot no-delay rescue signal and an array of many tens or hundreds of oceanographic survey floats equipped with high explosives set adrift in international waters for long periods of time.

When the author became interested in the tracking of deep currents by means of neutrally-buoyant floats in the summer of 1963, the existence of an ocean spanning network of shore-based SOFAR listening stations had become a reality as regards the North Pacific and North Atlantic oceans. The time had come to develop a neutrally-buoyant float with long-range signalling capabilities. Some effort was given to the design of a safe and reliable explosive float. This was abandoned, however, not only for reasons set forth in the previous paragraph, but also because handling and storage of the floats prior to their release represented a grave problem. Thus other sources of energy had to be found.

When a hollow body collapses under excessive hydrostatic pressure, energy is released as a consequence of the work done by the water. If, for example, the body is a sphere, the energy is equal to the product of the volume and the pressure at which collapse occurs. For buoyancy, all Swallow floats have a large hollow volume. To use the potential energy represented by this volume is an appealing idea and a harmless one. Since Swallow floats are designed to withstand nearly all pressures encountered in the ocean, a method had to be developed to allow a controlled implosion to occur at a specified pressure.

Another basic problem was to determine the amount of useful energy that could be obtained from an implosion. It is impossible to do this in a liquid-pressurized tank of any finite volume; as soon as the test vessel begins to collapse, the pressure in the tank decreases substantially. A 1947 unpublished report of the Woods Hole Oceanographic Institution estimated without experimental evidence that at 1800 psi a volume of one-half cubic foot would release an amount of energy approximately equivalent to that obtained from one pound of TNT. Even if this estimate were off by a factor of ten, it might still be better to use implosive floats rather than to make extensive use of explosive floats.

This report describes in some detail the work undertaken to develop and study controlled implosion devices. It also describes a field experiment conducted to determine the useful range of implosion devices for long-distance signalling.

II. INITIAL DEVELOPMENTS

During the summer of 1963 Allyn C. Vine of the Woods Hole Oceanographic Institution had in his possession several small signalling devices designed to implode at different pressures. These were obtained to test the usefulness of such devices in ship-to-bottom echo sounding. Their implosive volume was formed by a cylinder, 18 inches long with a diameter of 2.5 inches, or a volume of approximately 0.05 cubic feet.

Two such devices were made available to the author for a field test in cooperation with the Geophysical Field Station of Columbia University at St. David's, Bermuda. Both were set for 1880 psi which corresponds to the depth of the SOFAR Channel around Bermuda. The devices were placed on the R/V Crawford which sailed on 1 August 1963 from Woods Hole for Barbados, passing close by Bermuda. In addition to the devices, two small explosive charges were put on board equipped with 1800 psi pressure detonators. The charges were equivalent to 1/18 lbs of TNT and were needed to calibrate the signals from the implosion devices.

The first explosive charge and implosive device were each equipped with ballast and thrown overboard five minutes apart when the Crawford was 94 nautical miles south of Bermuda. Six hours later at a distance of 176 nautical miles the second pair was treated similarly. At the expected time of arrival for the first implosive signal a ship passed over the hydrophone raising the noise level sufficiently to mask any incoming signals. Fortunately, two explosive charges and the second implosive signal were received clearly. A comparison of the records obtained from the second explosive charge and the second implosive signal indicates that the implosive signal was equivalent in energy to 1/80 lbs of TNT. Considering the volume of the device, it was tentatively estimated that at 1880 psi a volume of 4 cubic feet is approximately equivalent in potential energy to 1 lb of TNT, which differs by a factor of eight from the 1947 estimate of the Woods Hole Oceanographic Institution (see page 5). The large distance over which this implosion signal had been heard encouraged the author to continue with the development of neutrally-buoyant implosive signalling devices.

The implosive devices used in the Bermuda field test were hollow cylinders, with a stiff disc at one end and a rupture disc at the other. These rupture discs are commercially available and consist of small metal plates, slightly concave in appearance, and prestressed in such a manner as to collapse under a known critical pressure. Their main purpose is to guard against damage from excessive pressures in industrial fluid and gas installations.

The results of the Bermuda test indicated that for ocean-wide signalling larger implosive volumes would be needed. It was also clear that larger rupture discs were required so that both the length and the diameter of the floats could be increased.

The largest inner diameter of tubing which has a wall thickness which can withstand the pressures at depth in the ocean, yet still is thin enough to be managable and inexpensive, is approximately six inches. Rupture discs are not commerically available in this size. The cost of developing them commercially proved prohibitive. It was decided to try and develop rupture discs within the scope of the project at the Massachusetts Institute of Technology.

It was assumed at once that the rupture discs should be made of metal to have both the strength to withstand great pressures and the ductility to deform and eventually rupture at critical pressures. In the design of metal rupture discs, two considerations are of prime importance. First, when rupture occurs, a disc should shatter rapidly and completely so as not to impede the forced flow of water into the tube. Second, some fairly straightforward method to control the pressure at which a disc will rupture had to be developed.

To assure proper collapse of the disc one surface may be scored with one or more grooves running from edge to edge through the center and dividing the disc into equal sections. The optimum number of grooves had to be determined experimentally and was found to be 2. A disc so scored will break into four sections which fall freely into the tube. One score leaves two half discs, each of which is too large to move freely. The behavior of discs with more than two scores was essentially identical to discs with two mutually perpendicular scores. Rupture would begin along one groove and slightly later start along a second groove. Once begun along a second groove, collapse into four sections would be rapid and complete.

That rupture will commence first along one groove and only later along a second groove is due to the fact that the metal used is mill-rolled. The least strength is perpendicular to the direction of rolling in the plane of the material. Thus the material below the grooves which lies most nearly perpendicular to the direction of rolling will have the least resistance to tearing under deformation caused by pressure. To counteract this undesirable behavior it was decided that all discs should be milled so that the two mutually perpendicular grooves would both intersect the direction of rolling at a 45° angle. The effect can still be observed, however, on any disc subjected to incomplete rupture due to decay in pressure. See Figure II-1.

Besides determining a rupture pattern for a disc, the grooves also cause the weakness which allows rupture to take place. Since it is obvious that the deeper the groove the greater this weakness, it seemed probable that some sort of relation between groove depth and rupture pressure existed. To examine this possibility further, an instrument was made to hold a disc and subject it to pressures up to 3000 psi in the hydraulic pressure tank of the Woods Hole Oceanographic Institution.

The choice of metal to be used for the discs is at least initially a question of taste. The author chose aluminum for two reasons. Since the tubes themselves were to be made of aluminum, the use of aluminum rupture discs would yield more possibilities for attaching the discs to the tubes; besides mechanical methods, welding could be considered as a means of fitting tubes with rupture discs. Also, since aluminum is a relatively weak material, thick plates would have to be used. If it was found that the rupture pressure of a disc were dependent on the ratio of the groove depth to the disc thickness, it would be easier to control this ratio with a thick aluminum plate than with a thinner steel plate.

The instrument used to discover if such a relationship does indeed exist is shown in Figure II-2. It consists of an aluminum tube which rests on a heavy aluminum bottom plate. The bottom plate also anchors six bolts which are used to tighten a steel collar against the top edge of the tube. The disc to be tested is placed with the grooves facing inward on top of the tube. The collar is placed over the disc. When the nuts are tightened on the bolts, the O-rings make the instrument watertight even at high pressures. The instrument is then placed in the pressure tank. When the tank pressure is increased the disc will deform until it ruptures since it is supported by only one atmosphere of pressure inside the tube.

A few initial tests were performed on discs of thickness 0.250

inch and 0.160 inch with groove depth ranging from 0.006 inch to 0.080 inch. The grooves were cut with a 60° V-cutter. The same tool has been used on all discs treated in this report. It was soon learned that the thicker discs were not suitable for our purpose. Even with very deep grooves they remained sufficiently strong to deform without ever rupturing. Thus, all results reported herein pertain to discs of thickness 0.160 inch.

The early tests in the Woods Hole pressure tank showed conclusively that rupture pressure does indeed vary with groove depth. Unfortunately, the accuracy of the results suffered greatly from two major faults of the combined system. The collar of the instrument shown in Figure II-2 does not maintain a continuous clamp on the disc as the pressure increases, for as the O-rings compress under pressure the tension on the collar provided initially by tightening the six nuts is relaxed. Thus, although a watertight seal was maintained, the disc could slide freely on the tube and in many tests with shallow grooves it deformed irregularly and was forced into the tubes without rupturing.

The pressure in the tank at the Institution is built up by means of a high-pressure small-displacement compressor which forces water into the tank. Since insufficient precautions were taken to prevent the cyclic hammering of the compressor from affecting the pressure in the tank, the pressure at all times fluctuated around its mean by from 50 to 200 lbs per square inch. This made it almost impossible to take measurements to the accuracy required.

Since both the tubes and the discs were made of aluminum, it was decided at the time to try welding as a means of attaching a disc to its tube. Two methods were tried.

In the first method, a disc of 6.625 inch diameter was placed symmetrically on a tube with an inner diameter of 6.00 inch and an outer diameter of 7.00 inch. The exposed top of the tube and the edge of the disc were then fused together by means of inert-gas arc welding. This type of bond did not prove strong enough to withstand the deformation of the disc under pressure. Also, since the disc was rather small, it could be forced into the tube at low pressures. See Figure II-3.

In the second method, a disc of diameter 7.750 inch was placed symmetrically on a similar tube. The two were fused together by welding completely around the common joint of the tube and the disc. This type of joint was strong enough to hold any disc completely at pressures up to 3000 psi. See Figure II-4.

The second method is probably superior to the first because the leverage exerted on the joint by the deforming disc is less using the second method. It was found that the behavior of the welded discs was the same as that of the clamped discs provided that care was taken to prevent the disc from heating up during welding.

Since at least one practical means of attaching the discs to the tubes for actual oceanographic work had now been assured, the design and use of an instrument to study the behavior of the discs in detail became of prime importance to the project. The work done to this end is described fully in the next three chapters.

Figure II-1. Effect of the direction of rolling on the rupture pattern when rupture pressure cannot be maintained.

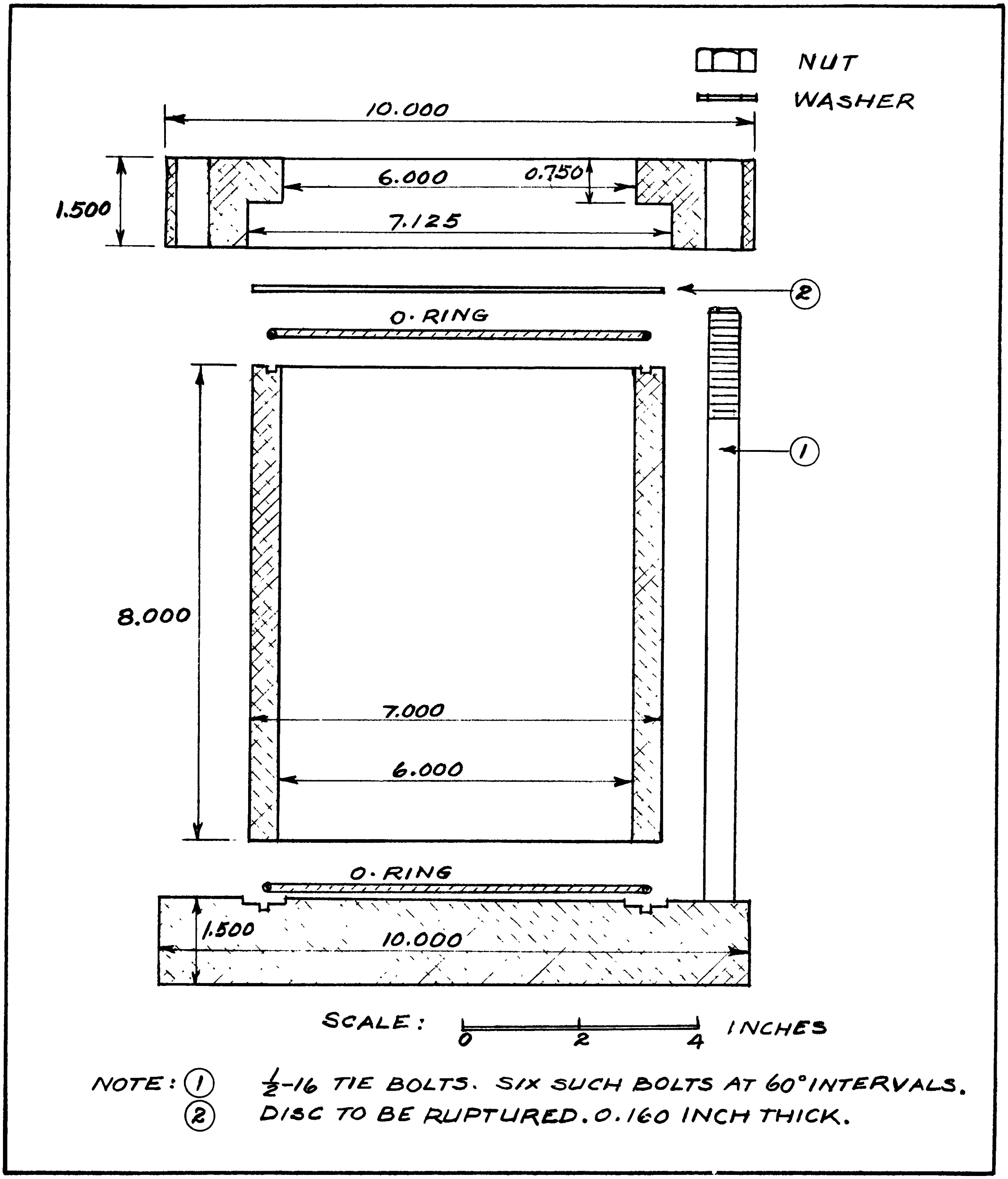

Figure II-2. Instrument to rupture discs in the pressure tank of the Woods Hole Oceanographic Institution.

Figure II-3. Unsatisfactory method of welding a disc to a tube.

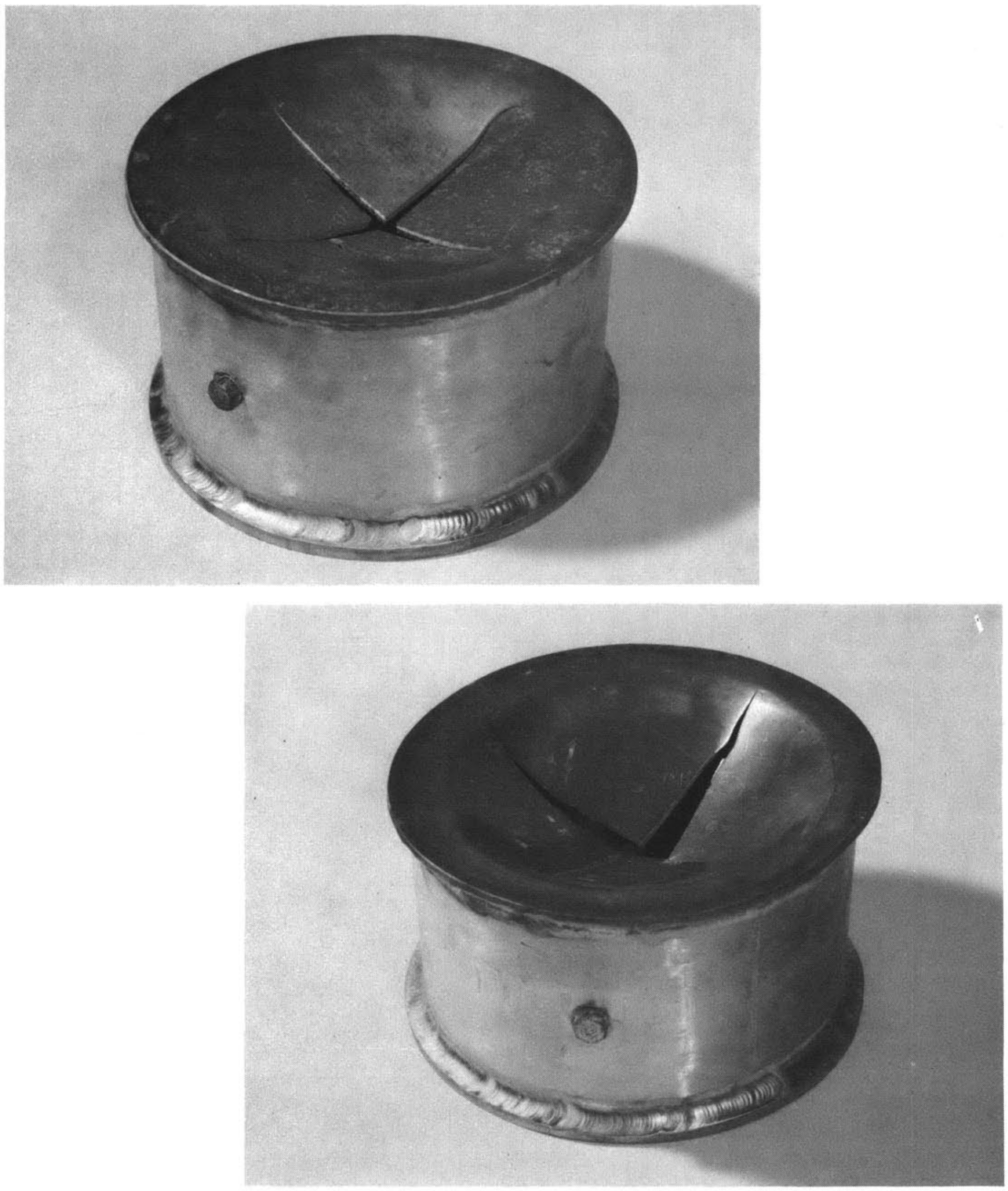

Figure II-4. Satisfactory method of welding a disc to a tube.

III. DESIGN OF A TEST INSTRUMENT

When it was realized that the experimental approach to the study of rupture discs had to be improved, the requirements were quite clear from the work that had just been completed. The experiments required a smoothly increasing pressure and a continuous positive clamping of any disc being tested. An additional requirement became of great importance, however. Since it was planned to use these discs on neutrally-buoyant floats, their deformation under pressure at all times prior to rupturing should be known accurately in order to predict the effect of this deformation on the relative buoyancy of the float. Thus some means of accurately measuring the shape of the disc had to be devised.

Some effort was dedicated to designing an improved pressure system for the pressure tank at the Woods Hole Oceanographic Institution. Since the only way a disc inside the tank could be monitored was by means of four electric leads into the tank, it seemed that this approach would certainly become a time consuming adventure in electronic telemetry. It was therefore thought advantageous to invert this approach; rather than to subject the disc to a high pressure environment and deform it into the tube, one could build up the pressure inside the tube and deform the disc outwards into a one atmosphere environment. With the latter approach, the instrument could sit on a laboratory bench and the disc could be monitored visually at all times, and its deformation measured by means of micrometer dial gauges. With this new approach a smooth pressure build-up can be assured quite simply by using a manual hydraulic pump such as is commercially available. Since the SOFAR Channel is nowhere deeper than 6600 feet, which corresponds to 3000 psi, all that is required is a pump capable of creating 5000 to 10000 psi. The problem of continuous clamping had yet to be solved, however, while a new problem was introduced by this new method.

It was seen that the instrument shown in Figure II-2 would be forced together at its seams when under pressure from the outside and so automatically tend to seal itself. If such an instrument is under pressure from the inside, however, it will be forced to come apart at its seams and thus be highly susceptible to leaking. After some thought, an engineering approach was evolved which, if successful, should solve both the leaking and the clamping problems.

Consider again Figure II-2. Under pressure from the inside the tube will expand radially. The bolts, subjected to stresses created by the outward pressure on the bottom, and on the disc-and-collar assembly, will stretch. This elongation would allow the disc to lift off the tube so that leaking would occur. It is known that if all expansions remain within the elastic limits of the materials used, such expansions are linear functions of the applied stresses. Imagine that the dimensions of the tube and bolts, and the number of bolts, are chosen so that the radial expansion of the tube and the lengthwise stretching of the bolts, are of the same order of magnitude. Let the outside of the bottom of the tube be beveled at some angle and rest in a groove in the bottom plate. Let the outside wall of the groove be beveled at the same angle. Any radial expansion of the tube will force the tube to ride up on this beleled surface and move upward relative to the bottom plate. If the angle is chosen correctly, this amount of upward motion will exactly compensate for the stretching of the bolts.

This action should guard against leaking provided, of course, that the two joints are properly fitted with O-rings. Furthermore, if the angle is a little steeper than it needs to be, continuous clamping action between the tube and the collar to hold the disc should be assured. The angle of the bevel cannot be made so steep as to be nearly parallel to the side of the tube, for in that case rather than riding up in the beveled groove of the bottom plate, the tube might bind against the steep side of the groove.*

The instrument and its associated devices are shown in Figures III-1 through III-9. Since the instrument is basically axisymmetric, Figure III-1 suffices to show in exploded view the assembly of the

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* See Appendix B for the calculations necessary to assure a sufficient safety factor in the overall design of the instrument, and to ascertain that the expansion of the tube and bolts are of the same order of magnitude.

several parts which are shown in full detail in Figures III-2 through III-6.

The bottom plate (see Figure III-2) is made of hot-rolled steel plate, two inches thick with a diameter of 12.125 inches. The hole in the center feeds hydraulic fluid into the chamber of the tube. The large hole, two inches from the center, holds a stopcock for draining the hydraulic fluid. The beveled groove three inches from the center accepts the tube. The fifteen holes at 4.625 inches from the center are tapped for 3/4-16 threaded tie rods which are used to hold the disc and collar firmly on the tube. The six 3/8-16 holes at 5.750 inches from the center hold six rods that support an aluminum table (see Figure III-5) 4 inches above the disc and collar. The table has holes in it through which probes may be lowered to rest on the disc. The deflection of the disc can then be measured by means of a micrometer dial gauge which moves freely on the aluminum table (see Figure III-7). The three 1/2-20 holes in the bottom of the plate accept three bolts by which the whole instrument is supported and leveled.

The tube (see Figure III-3) is extruded from hot-rolled steel. It was machined to be perfectly round and uniform from a piece 3-1/4 inch long, with a nominal inner diameter of 6 inches and a nominal outer diameter of 8 inches. The dimensions of the O-ring grooves are specified by the manufacturer.

The collar (see Figure III-4) is machined of a hot-rolled steel plate, 2 inches thick with a diameter of 11 inches. The great thickness is required to equalize the pressure of the 15 individual bolts on the disc and tube. The holes for the bolts are made large for a loose fit to facilitate placement of the collar. Note that the lower part of the collar which slides over the tube is of sufficient diameter to allow for radial expansion of the tube.

Figure III-6 shows in detail the rods and support rods required to complete the pressure vessel. Figure III-7 shows the instrument completely assembled and in use. The five probes used to measure the disc are all 6.000 inches* long. The probe used to monitor the upward travel of the collar relative to the table and bottom plate is 5.000 inches long. Figure III-8 shows the manual hydraulic pump and the pressure gauges used. The pump is a Black Hawk P-39 with a reservoir of 39 cubic inches and a pressure capability of 10000 psi. The 0 to 5000 psi Acco gauge is accurate within 25 psi which is also as close as it may be read with confidence. The gauge was calibrated to read 2000 when subjected to a static pressure of 2000 psi. The 0 to 200 psi Marsh gauge is accurate to 2 psi but cannot be read any better than 2-1/2 psi. It was calibrated to read 100 when subjected to a static pressure of 100 psi. The Marsh gauge was used to compensate for the unsatisfactory reading

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* Certain critical dimensions specified to three decimal places are assumed to be accurate within 0.001 inch.

of the Acco gauge below 150 psi. It is protected from pressures over 200 psi by a high pressure valve. Figure III-9 shows a top view of the instrument after removal of the aluminum table and collar.

After the several parts of the instrument had been machined they were assembled. Next the instrument was leveled in order to make the top of the tube perfectly horizontal. This was necessary so that the amount of air trapped between the disc and the hydraulic fluid in the tube would be minimized. The two beveled surfaces and the lower O-ring were lubricated with silicon grease (Dow Corning stopcock lubricant).

When the instrument was first assembled, an aluminum disc with a diameter of 7.000 inches and a thickness of 0.250 inch was used to test all components. For this test only the deflection in the middle of the disc was monitored. The pressure was slowly increased to 3000 psi. When this pressure had been reached, the center deflection was 0.749 inch. The instrument was left for half an hour to allow for creepage of the disc. At the end of this period, deflection was 0.752 inch and the pressure had dropped to 2775 psi. Again the pressure was brought to 3000 psi. No noticeable change in the center deflection was observed. After two hours, the disc had come to equilibrium, with a center deflection of 0.753 inch and a pressure slightly below 3000 psi. The pressure was again carefully increased to read exactly 3000 psi. Since absolutely no change in the center deflection was observed, the apparatus was left under pressure for 24 hours. During this period no discernable change in either deflection or pressure occurred.

This test was repeated twice more with similar discs. Since the same pattern of behavior occurred each time, it was clear that the instrument did not leak significantly. Examination of the edges of the discs also showed that the clamping action of the tube and collar was quite sufficient at all times during any test.

During the construction of the test instrument, the literature was searched to discover if any test similar to those about to be made had ever been conducted. The next chapter deals more fully with this aspect of the work.

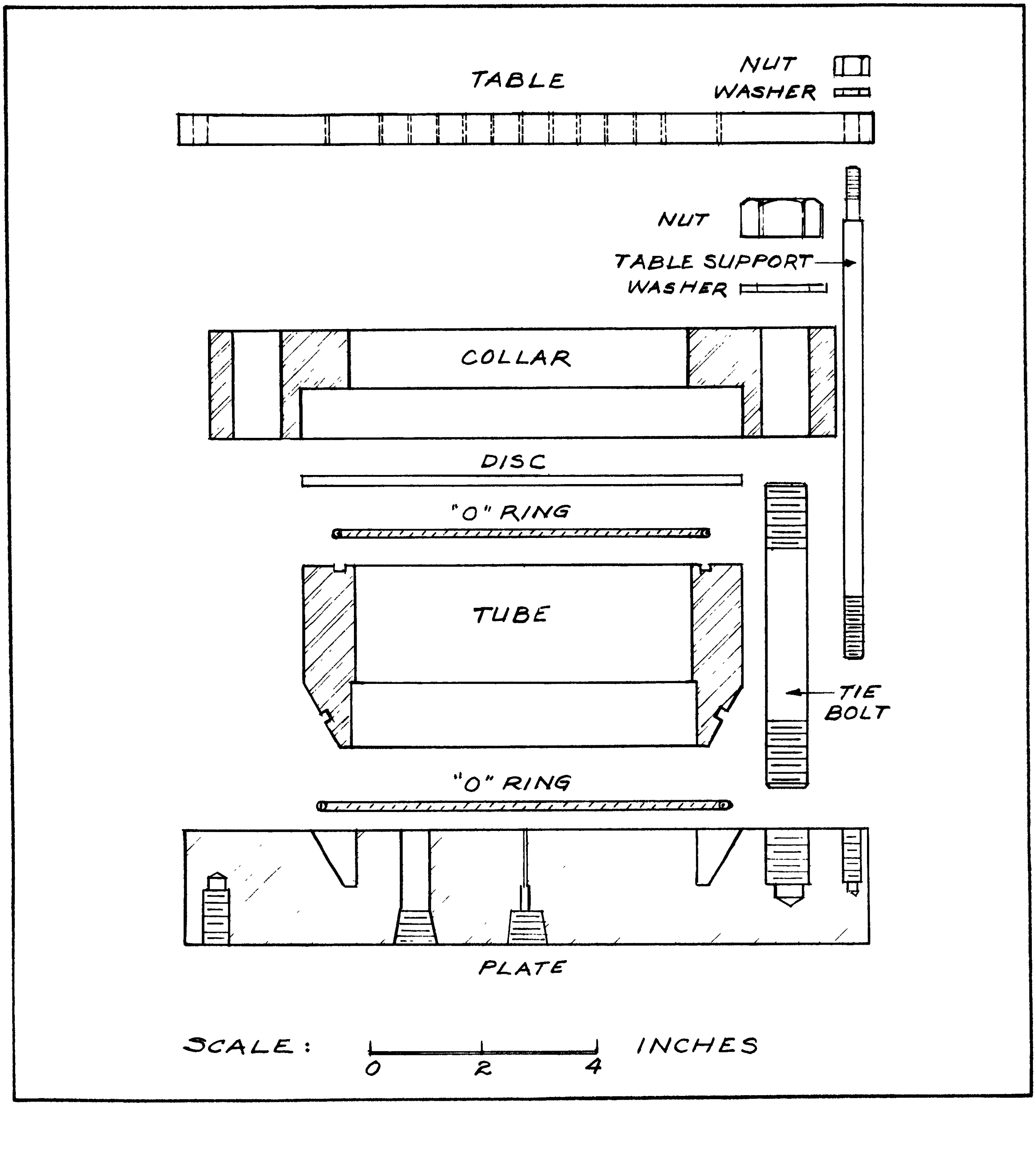


Figure III-1. Exploded view of the pressure vessel designed to deform and rupture discs with a diameter of 7-3/4 inches.

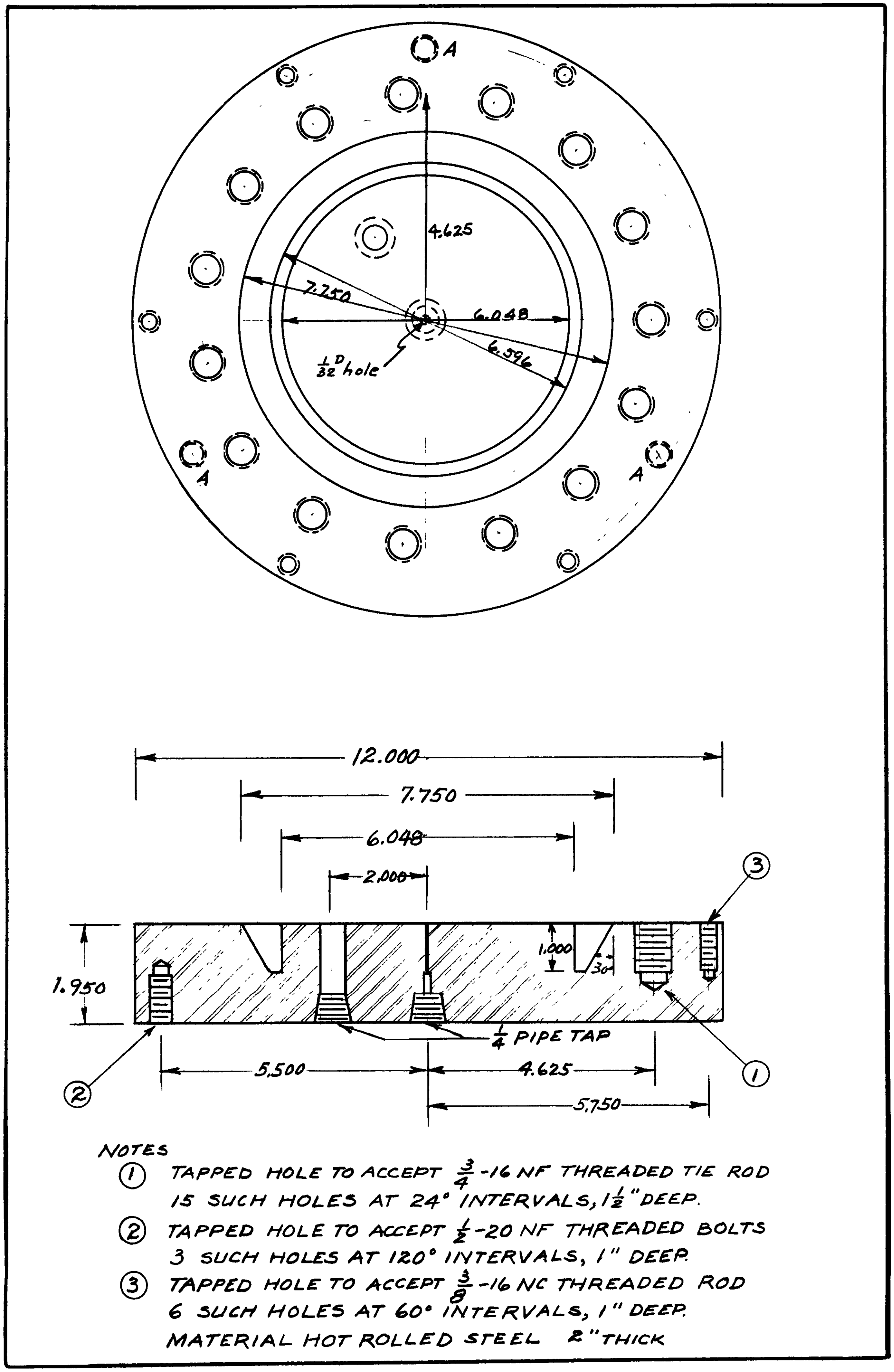


Figure III-2. Steel bottom plate.
Top: Top view.
Bottom: Cross section through the center.

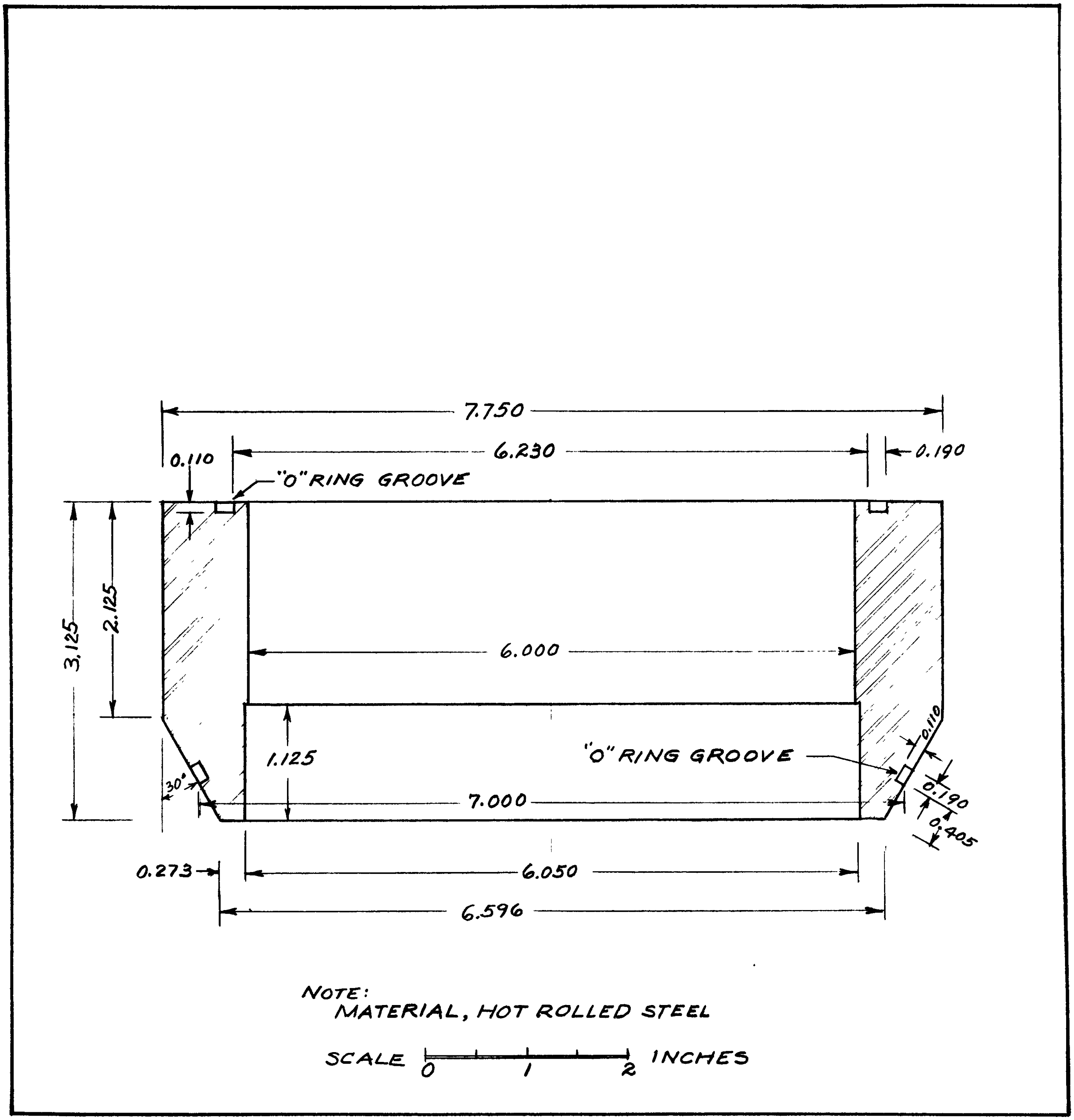


Figure III-3. Steel tube with bevelled seat.

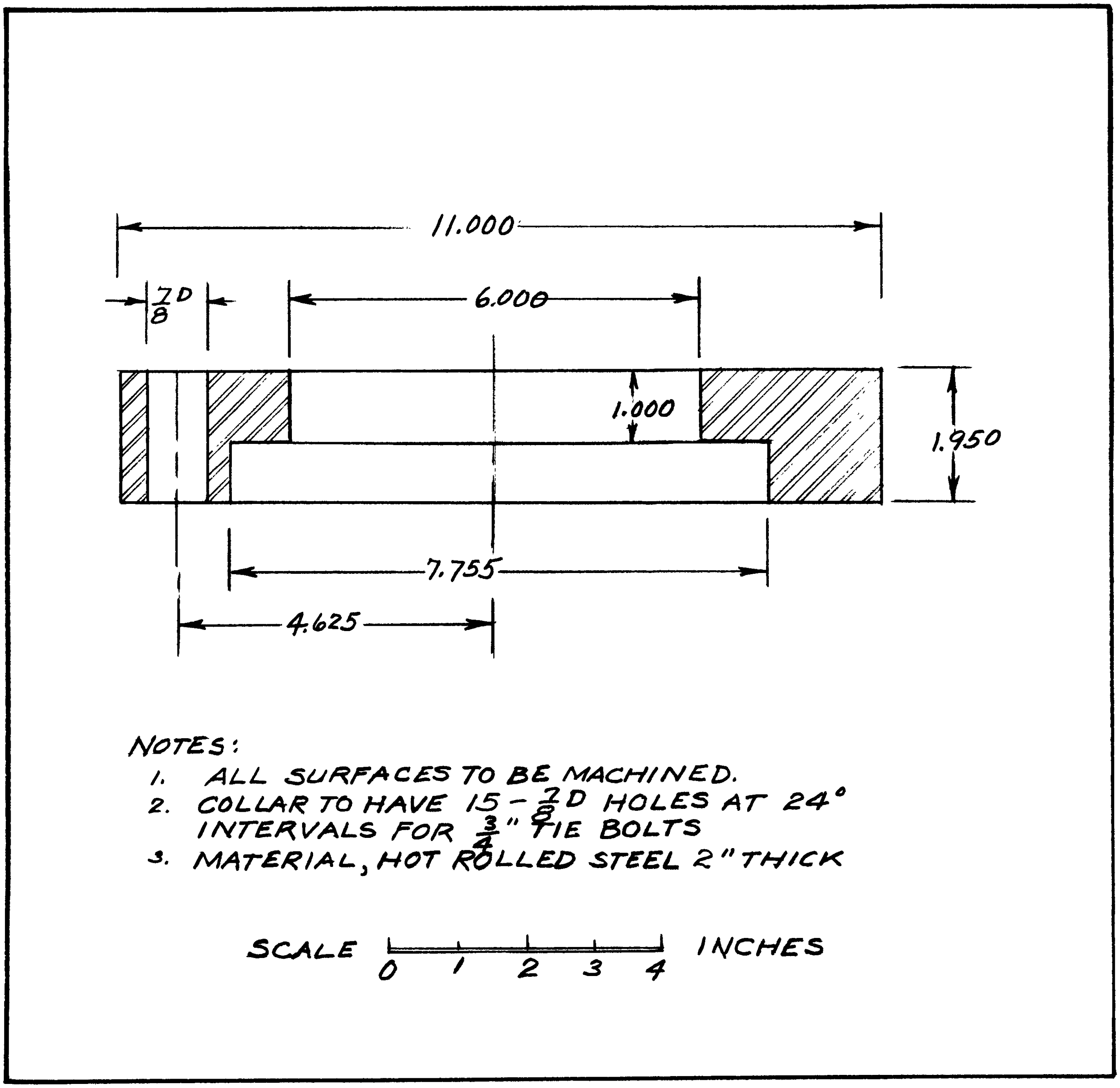


Figure III-4. Steel collar.

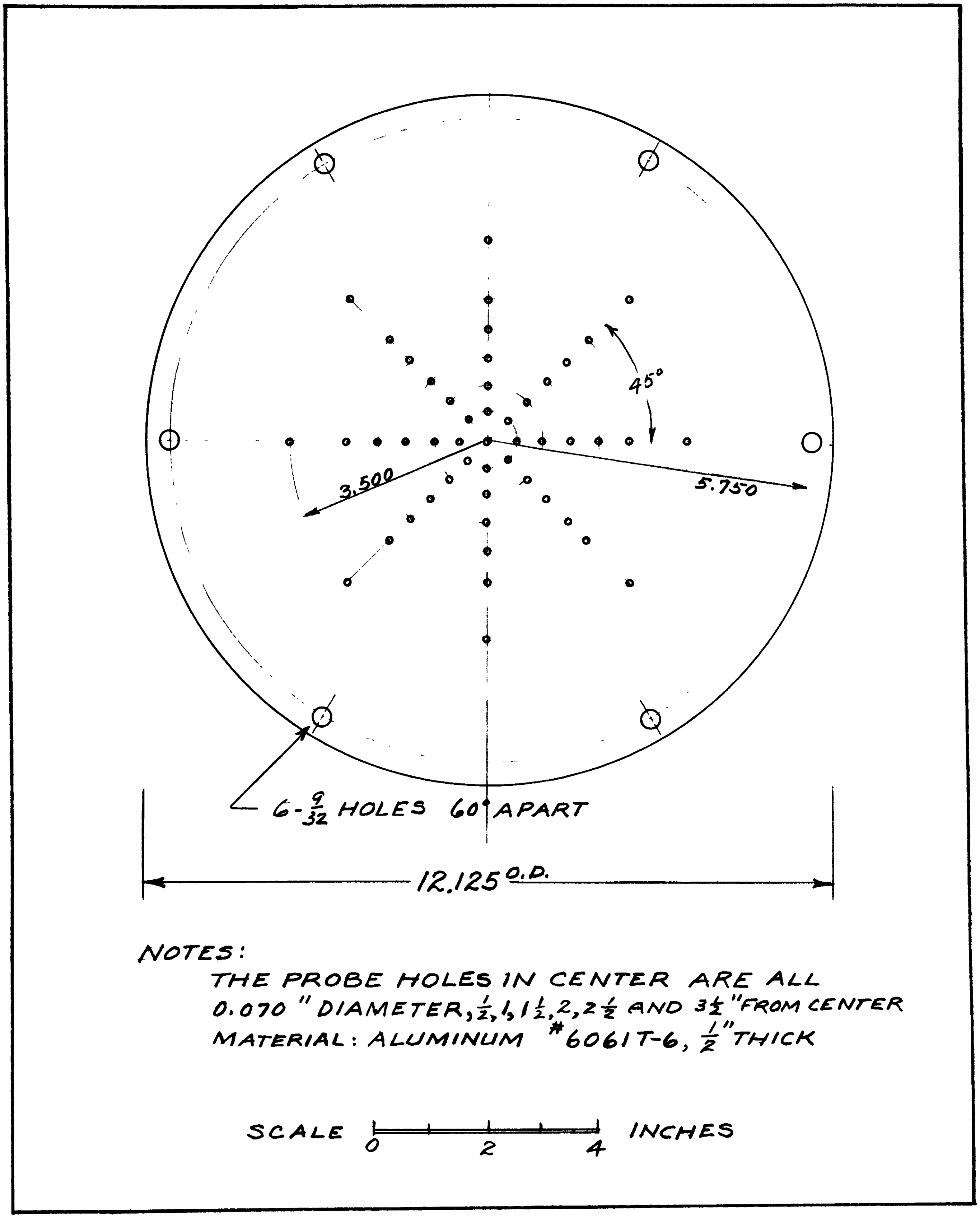


Figure III-5. Aluminum table, top view.

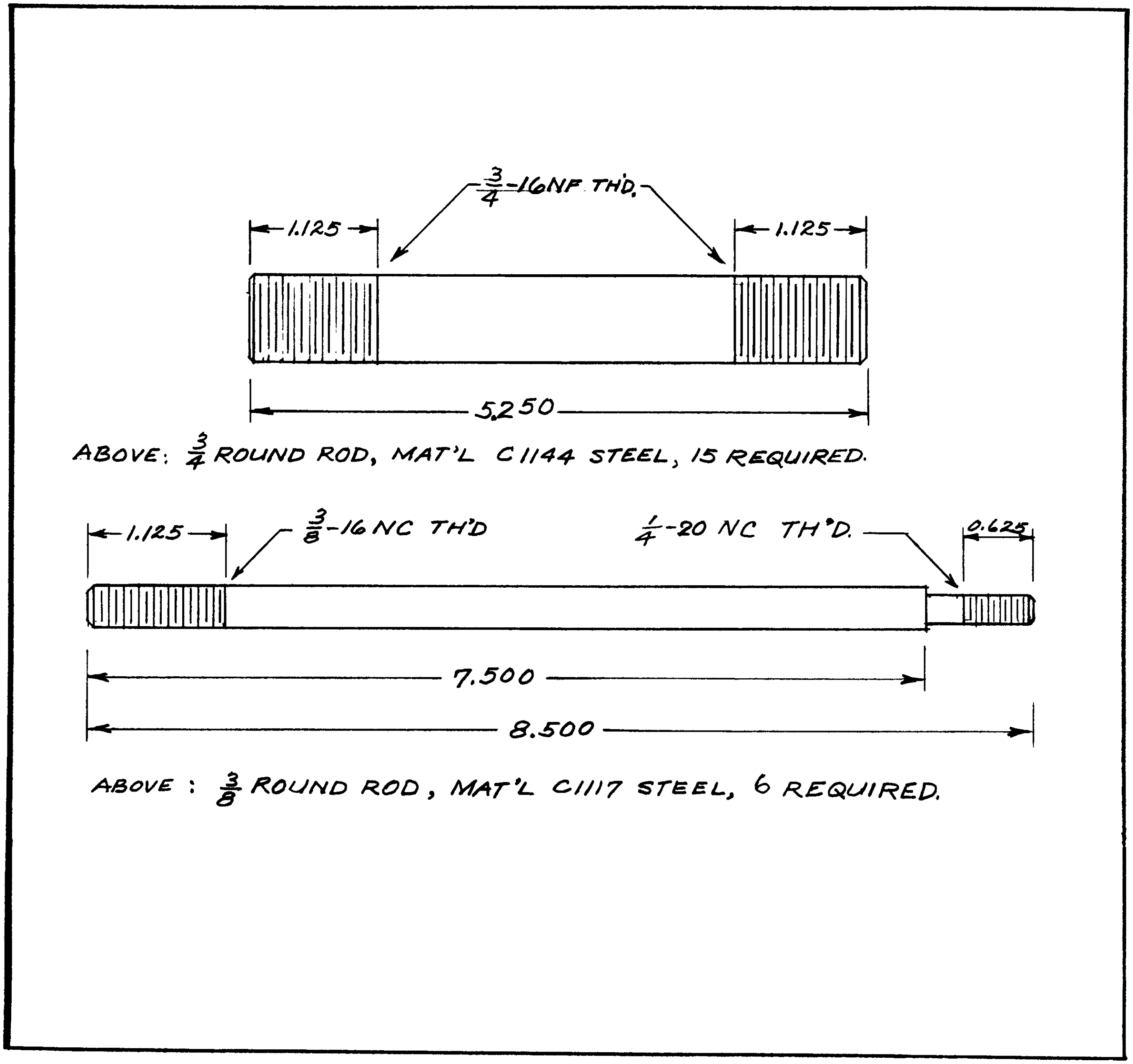


Figure III-6. Top: Collar tie bolts.
Bottom: Support rods for aluminum table.

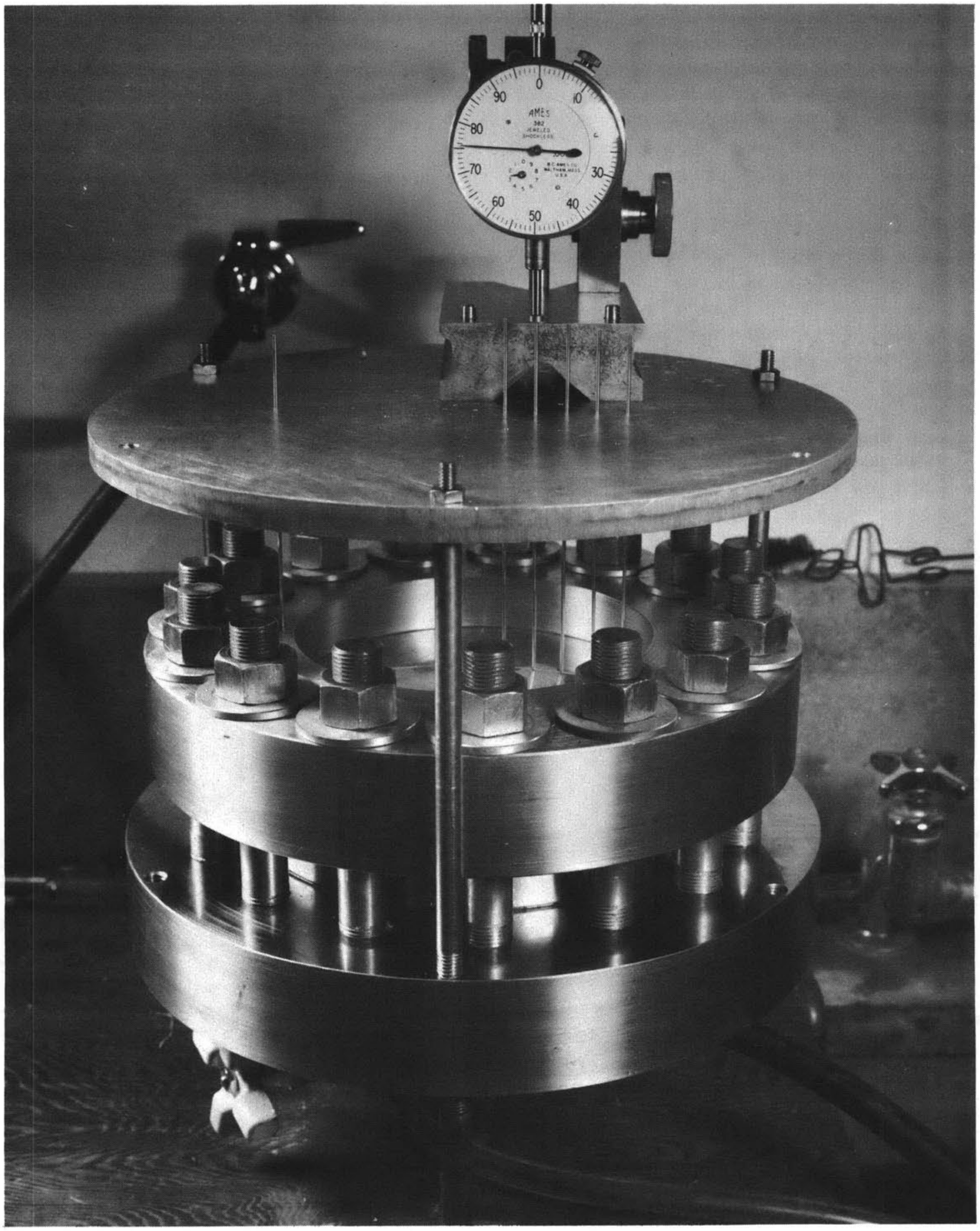


Figure III-7. The pressure vessel completely assembled
and in use.

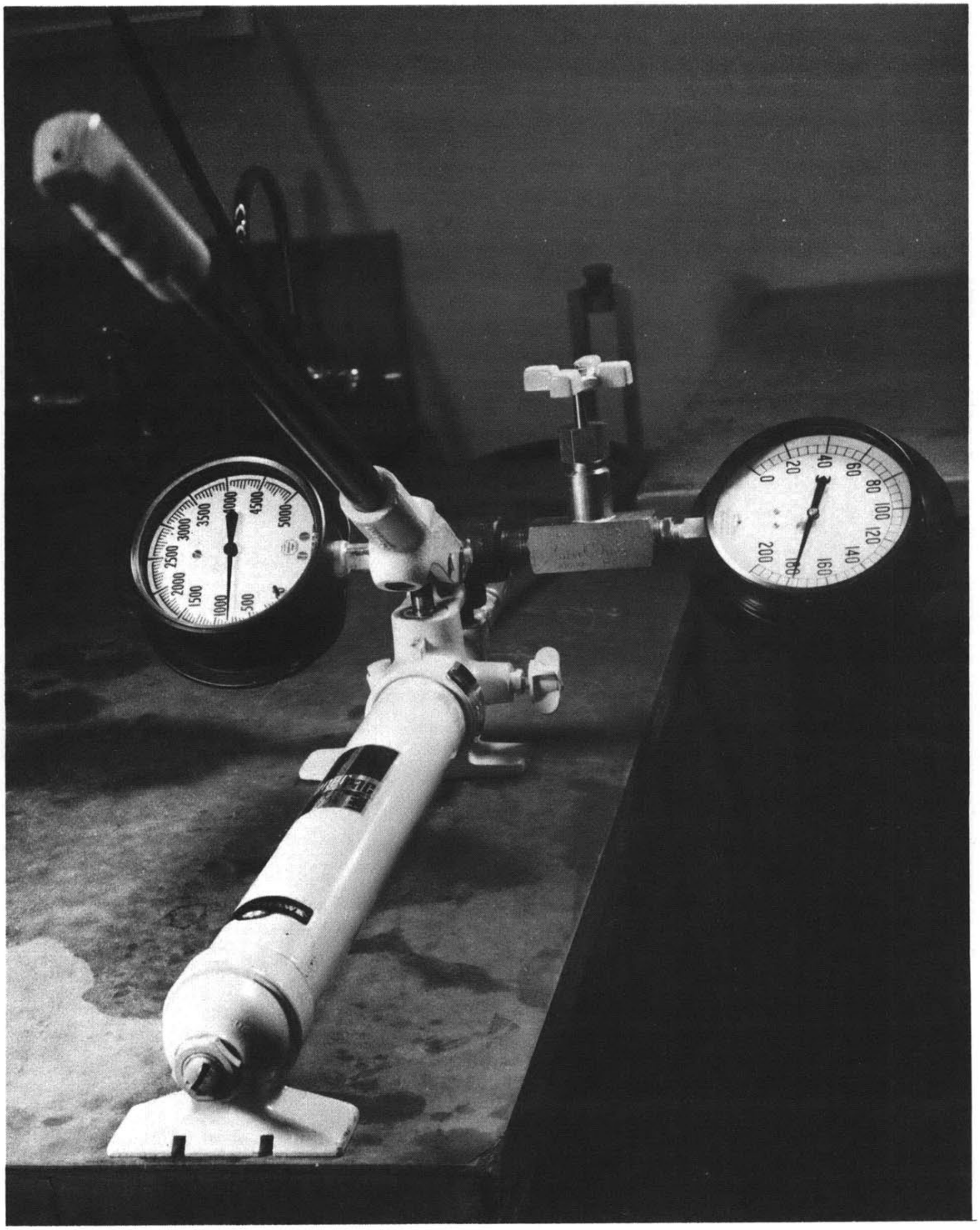


Figure III-8. The manual hydraulic pump (Black Hawk P39).
The Acco gage is on the left, the Marsh gage is on the right.

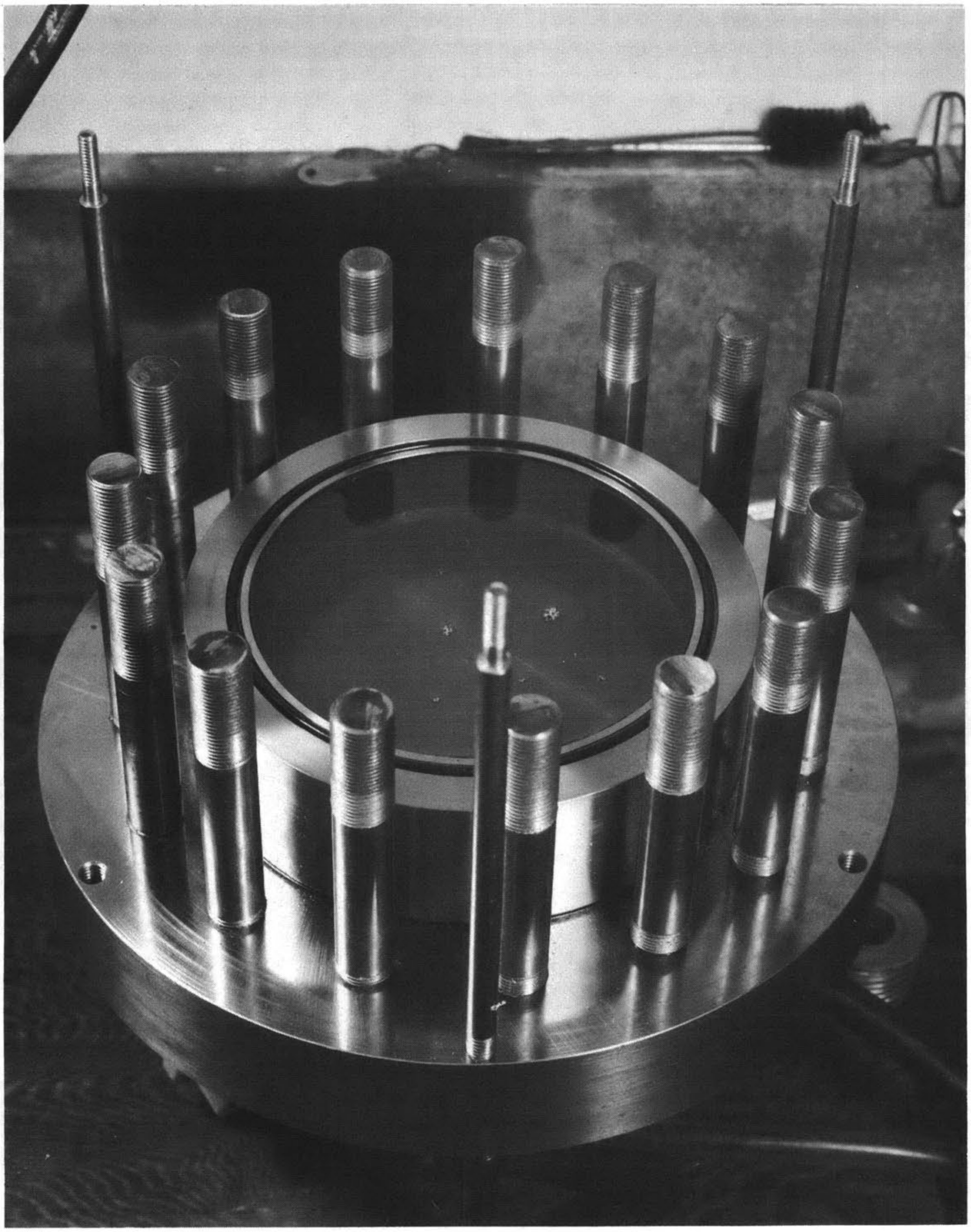


Figure III-9. View of the pressure vessel after removal of the aluminum table and the steel collar.

IV. THEORETICAL CONSIDERATIONS

The purely theoretical investigation of the deformation of metals under stress is still mainly a study in mathematical modeling and has only the basic concepts in common with the engineering approach to the problem. Thus while the work of such mathematicians as Sir A. E. H. Love (1944) is of great interest, the more pertinent information for the purposes of this report is found in the experimental studies conducted on the behavior of metal plates under pressure. All such engineering studies are concerned with the deformation of plates to an extent well short of creating permanent undesirable after effects. Thus all information that is found in the literature concerns behavior within the limit of elasticity of the metal.

Since the purpose of this project is to study the behavior of a specific material well beyond its elastic limit, most previous information has little bearing on this work. If one also keeps in mind that for this project each plate is deliberately weakened be scoring it, one realizes that this investigation is quite far removed from all other deformation studies.

The two important parameters which distinguish this series of experiments from all others performed on circular discs of similar dimension are the thickness of the disc (h), and the ratio of the deflection at the center of the disc (w_o) to the thickness of the disc (w_o/h). Most experiments and formulas are created to study the behavior of discs cut from materials usually employed for construction and engineering purposes. While the thickness of these metals usually ranges from .005 inch to .030 inch, this study deals with discs cut from aluminum sheets that are .160 inch thick. Whereas, for most metals that are deformed within the limits of elasticity, the ratio w_o/h is less than 1, in the experiments presented here this ratio is consistently greater than 1 for the values of pressure of interest. Indeed, for most discs this ratio becomes as great as 5 or 6. The significance of the ratio w_o/h is due to the fact that we are dealing with rigidly clamped circular plates.

When a rigidly clamped circular plate is subjected to a moderate normal pressure distributed evenly over the plate, resulting in a deformation such that $w_o/h < 1$, the deflections observed in the plate are caused by local bending moments. That is, each individual section of the plate behaves as if it were supported freely at its ends and deformed by a deflecting force applied locally near its geometrical center. Under these conditions the plate is known as a Kirchhoff plate (Nadai, 1925). Assuming a plate of uniform cross section, its profile will be symmetrical under uniform pressure and may be described by relating the deflection w at a distance r from the center of a disc of radius a to the deflection w_o at the center:

$$\frac{w}{w_o} = \left[1 - \left(\frac{r}{a}\right)^2\right]^2, \quad \frac{w_o}{h} < 1 \qquad (1)$$

The behavior of a Kirchhoff plate is characterized by the absence of significant longitudinal stresses in the disc, and a zero slope at the clamped edge of the disc. For from (1)

$$\frac{dw}{dr} = 2w_o\left[1 - \left(\frac{r}{a}\right)^2\right]\left(-\frac{2r}{a^2}\right)$$

which vanishes at $r = a$.

As the normal, evenly distributed pressure ceases to be moderate and becomes great or excessive, the resulting deformation is characterized by $w_o/h > 1$. Now the longitudinal stresses created in the plate, due to the stretching of the material as it deforms between clamped edges, become of far greater importance than local bending moments. For thin plates, the yielding caused by stretching will quickly spread over the entire plate, which will then tend to go into a spherical equilibrium surface like a membrane under constant tension. The shape of the deflection profile is given by

$$\frac{w}{w_o} = 1 - \frac{r^2}{a^2} \qquad (2)$$

In this study, however, we are dealing with relatively thick plates. Since greater strains occur when a thick plate is bent sharply than when a thin plate is bent sharply, the yielding in a thick plate will tend to be localized at the edge of a plate, due to the sharp change in slope at the clamped edge. The shape of the deflection profile will approach that of a plate with freely supported edges (Nadai, 1925):

$$\frac{w}{w_o} = 1 - 1.245 \frac{r^2}{a^2} + 0.245 \frac{r^4}{a^4} \tag{3}$$

The relation between center deflection and pressure for an elastic plate of medium thickness with rigidly clamped edges has been investigated by approximate methods by a number of authors. Nadai (1925) derived the relation

$$\frac{w_o}{h} + 0.583 \left(\frac{w_o}{h}\right)^3 = \frac{3}{16} \frac{p}{E} \left(\frac{a}{h}\right)^4 (1 - u^2) \tag{4}$$

by solving the differential equation for a plate with large deflection ($w_o/h > 1$) subjected to a nearly uniform pressure distribution. In this and the following equations the symbols have the following meanings:

w_o = center deflection (inches)

h = thickness of disc (inches)

p = pressure (psi)

E = Young's modulus (psi)

u = Poisson's ratio

a = radius of the disc (inches)

Timoshenko (1928) derived

$$\frac{w_o}{h} + 0.488 \left(\frac{w_o}{h}\right)^3 = \frac{3}{16} \frac{p}{E} \left(\frac{a}{h}\right)^4 (1 - u^2) \qquad (5)$$

on the basis of an assumed radial displacement combined with energy considerations. Federhofer (1936) obtained

$$\frac{w_o}{h} + \frac{(19 - 9u)(1 + u)}{40} \left(\frac{w_o}{h}\right)^3 = \frac{3}{16} \frac{p}{E} \left(\frac{a}{h}\right)^4 (1 - u^2) \qquad (6)$$

from the differential equation of the problem together with a suitable assumption for the radial distribution of membrane tension.

The material used in the experiments to be described was tempered aluminum of type 24 ST 3. For this material, Young's modulus (E) is 10.5×10^6 psi and Poisson's ratio (u) is 0.33. With these numbers the value of the coefficient of the term $(w_o/h)^3$ in equation (6) is 0.533, which is midway between the corresponding values in equations (4) and (5).

While the three equations predict the same central deflection of a disc as a function of pressure, differing only for values of (w_o/h) significantly greater than 1, it was decided to plot the curve yielded by Nadai's equation (4) since in his work he set out specifically to study plates with center deflection greater than the thickness of the plate. Using equation (3), the deflections at a distance of 1 inch and at a distance of 2 inches from the center are also calculated. It is of interest to compare these curves with those obtained by measuring the behavior of a thick disc without grooves which is subjected to normal pressures in the instrument described in Chapter III. See Figures V-1a and V-1b.

The reader can see from the above figures that the theoretically predicted behavior and the actual behavior of the disc as it responds to increasing pressure are quite similar. It is obvious, however, that the magnitude of the deflections predicted by the formula are too small by a factor of 3. This is without doubt due to the fact that the principles behind the equation used for the comparison are violated in a number of ways. First, the disc was stressed far beyond its limit of elasticity causing the ratio w_o/h to be unusually large. Secondly, the disc was far thicker than any that are usually considered. In the case of the grooved discs, the grooves represent a third and no doubt the most flagrant violation of the assumptions on which the theoretical formulas are based.

V. EXPERIMENTS AND RESULTS

A. Preliminaries

Before going into a detailed discussion of the experiments and their results, an explanation of the technical aspects of obtaining data with the instrument described in Chapter III is in order. When the bottom plate has been assembled and the tube set into it, the instrument must be made perfectly level. Then hydraulic fluid can be poured into the tube until it reaches the top edge of the tube uniformly around the circumference. This minimizes the volume of air that is trapped between the fluid and the disc to be tested. Although the amount of air confined under the disc is of no concern regarding the behavior of the disc under pressure, the amount of potential energy stored in the compressed air and released upon rupture of the disc increases in direct proportion to the amount of air trapped. Since the release of this energy manifests itself through loud explosive noises and the forcible ejection of a large amount of hydraulic fluid through the rupture, it is desirable to minimize the amount of air confined under the disc.

Another reason for leveling the top of the tube before each experiment is to avoid the systematic error that would be introduced into the readings of the dial gauge if the aluminum gauge table and the disc were not parallel to each other. By making both the top of the tube and the gauge table level, the disc and the table will be parallel with a deviation of less than 0.001 inch over the width of a disc.

For this set of experiments the aluminum table was equipped with probe holes so that the deflection of a disc could be measured at its center and at distances of 1/2, 1, 1-1/2, 2 and 2-1/2 inches from the center at angular intervals of 45° about its circumference (see Figure III-5). If it were found during the experiments that the behavior of some or all discs was so irregular that more probe holes were necessary, they could be added as required. Since it is of interest to measure the deflection along a grooved radius, and also along a radius between grooves, the discs have to be placed on the tube so that the two mutually perpendicular grooves fall precisely below two mutually perpendicular lines of probe holes.

If all holes in the table corresponding to locations on the disc that needed to be monitored were equipped with probes, it would be impossible to set the dial gauge and its support upon the table to take the measurements. Therefore only five probes were used and as soon as the center deflection for a given pressure had been measured each radius of five probe holes was measured in turn. The probes were moved as necessary.

To be sure that the same location is monitored each time, each disc must be marked to show the location for the probes at the correct angular intervals and distances from the center before it is placed on the tube. By doing this a probe can be removed and placed repeatedly, deviating from the required location by no more than 1/64th inch in any direction.

When the disc is marked it is placed on the tube with the grooves facing out. This is done to simulate the attachment of a disc to a neutrally buoyant float where the grooves face into the float to protect the grooved surface from the environment. The collar is then positioned over the bolts and lowered onto the disc. Each bolt is then fitted with a flat washer and a nut. To take up the initial slack provided by the two O-rings, all nuts are tightened sequentially to 25 ft-lbs of torque. To prevent an increase of pressure due to the compression caused by tightening the nuts, the stop cock in the bottom plate is opened to allow hydraulic fluid to escape while the nuts are being tightened. The aluminum table is then placed on its six supporting rods. It is leveled by means of any three rods located at 120° intervals. When level, the table is secured by tightening the nuts on these three rods. The other three rods are then adjusted to provide additional support for the table and also secured by tightening their nuts.

When analyzing the data obtained from the experiments, it is assumed that all discs are initially perfectly flat. In practice this is never true and it is therefore necessary to measure the profile of the disc before pressure is applied to it. When the pressure in the system increases, the tube and collar move up relative to the table. This motion would introduce an error in the readings if it were not monitored. The convention established for all experiments was to adjust the dial gauge to read "zero" at zero pressure when placed on the probe that monitors the collar. This probe has a length of 5.000 inches and is left in place throughout the experiment. Since the difference in height between the top of the collar and the upper surface of the disc is 1.000 inch, and the probes used to measure the disc are 6.000 inches long, the initial profiles usually show a deviation from flatness ranging between plus and minus 0.010 inch. Thus any measurement at a given location for a given pressure has to be corrected by subtracting from it the reading on the collar at that pressure and the initial deviation from flatness for that location.

When a piece of metal is subjected to a deforming stress of constant magnitude the rate of deformation will decrease as the piece approaches the state of equilibrium appropriate to the applied stress. This phenomenon is known as creeping. Due to the creeping of metals there is a distinct difference in method, if not results, between subjecting a disc to a continuously and rapidly increasing pressure and subjecting such a disc to a discretely and slowly increasing pressure. It is this latter method that had to be employed in the experiment due to the nature of the instrument. Fortunately this latter method is a good representation of the pressure function to which a disc is exposed when attached to a float sinking in the ocean.

When a float adjusted to be neutrally buoyant at some depth is set overboard it may take as long as eight hours to reach its floating depth. Thus although the pressure increases continuously, it does so very slowly allowing enough time for creeping. While it would be ideal to subject the discs to a slowly increasing continuous pressure function, a slowly increasing discrete pressure function should yield satisfactory results. With this in mind, the pressure on any disc in the instrument has to be increased slowly, and before any measurements at a given pressure can be made, sufficient time has to be allowed for creeping to be completed.

B. The Experimental Program

As was pointed out previously, the purpose of the experiments was to find whether any consistent and predictable relation exists between groove depth and rupture pressure, and to study the deformation and behavior of the discs before rupturing. For this purpose, about 60 discs were cut, each disc having a diameter of 7-3/4 inch and a nominal thickness of 0.160 inch. The true thickness varied locally in each disc between 0.158 and 0.162 inch. The discs were divided into groups of 3 to 6 discs each. All discs in a given group were scored with two mutually perpendicular grooves of a given depth, each 6.00 inches long, and bisecting each other in the center of the disc. The depth of the grooves ranged from 0.012 to 0.050 inch. The choice of these depths is based on the preliminary experiments described in Chapter II. The tool used was again a standard 60° V-cutter.

Theoretically, a flat disc with uniform thickness throughout will deform with axial symmetry when subjected to normal pressures. This was observed to be true when testing the instrument and when measuring the deformation of the disc without grooves, but there is no reason to assume a priori that a disc with grooves will also deform with axial symmetry. Thus it will be of interest to observe the differences in the deflections measured along a groove and between grooves. Furthermore, since the material used is rolled aluminum of type 24ST3 and has a definite grain running the length of the rolled sheet as was discussed in Chapter II, it will also be interesting to see whether or not the grain causes any marked differences between the measurements obtained on the different parts of the disc.

During the preliminary experiments at Woods Hole it was observed that actual rupture is a very fast process. Since rupture occurs when the longitudinal stress created in the material by deformation becomes too severe to be withstood by the thin portion of the disc under the groove, the process of rupture probably consists of two phases. First, as tearing due to excessive stress commences, there will be a fairly rapid additional deforming of the disc. This will continue for only a short time, however, for the next phase, failure will follow immediately. Since this is a continuous and rapid process, it is doubtful that one will be able to observe it with the instrument of Chapter III. High speed photographic techniques would be better for observing actual rupture.

As can be seen in Figure V-1, the deformation of a disc, even without grooves, is quite considerable. If such a disc were placed on a neutrally-buoyant float its contribution to the change in the volume of the float as it sinks might be so large as to destroy the neutral buoyancy of the float. It was observed from the experiment performed on the disc without grooves that, after the disc had been subjected to a pressure of 2400 psi with a center deflection of 0.865 inch and then removed from the instrument, it retained its shape with a center deflection of 0.747 inch. The question was raised whether a grooved disc of known rupture pressure that had been preformed at a pressure just below its rupture pressure would, when placed in the instrument again, rupture at a pressure and with a center deflection equivalent to those of a similar disc that had not been preformed. If this were the case it would be a simple matter to preform discs for oceanographic purposes and thus greatly decrease the effect of the volume change of the disc on the buoyancy of the float.

With the consideration in mind the experimental program was divided into two distinct parts. The first concerns those discs that were deformed and ruptured directly, while the second part concerns those discs that were preformed before rupturing. The pertinent data regarding the first set are collected in Table V-1, while the data for the preformed discs are collected in Table V-2. In both tables the numbers in the first column are merely means for identifying the discs, while the second column shows the groove depth for each disc.

In Table V-1 the third column shows the pressure at which each disc ruptured. Note that the rupture pressure is always given in whole multiples of 25 psi, this being the limit of the accuracy of the pressure gauge. The fourth column shows the average rupture pressure for a set of discs with equal groove depth. The high and low deviation limits are also shown. The fifth column shows the maximum percentage deviation of the rupture pressures for a given group of discs to facilitate comparison between groups of discs. The sixth column shows for each disc the average rate of deflection of the center in thousandths of an inch per psi of supplied normal pressure. This ratio is a measure of the relative amount of longitudinal strain to which a disc is subjected before rupturing. The next column shows the average of this ratio for each group of discs. The eighth column shows the average center deflection of each group of discs at rupture. The last column identifies those individual discs whose behavior before rupture is shown graphically.

In Table V-2 the third and fourth columns show respectively the pressure at which a disc was preformed and the length of the forming time. The fifth and sixth columns show the deflection at the center during forming and after removal from the instrument. Thus the data in the sixth column are a measure of the permanent set of each disc. The seventh column shows the amount of time that elapsed before the preformed disc was placed in the instrument to be ruptured. The next two columns show respectively the rupture pressure for an individual disc and the average rupture pressure for a group of preformed discs. Again, the high and low deviations are shown. The tenth column shows the percentage deviation for each group of discs. The last column identifies those individual discs whose behavior after forming and prior to rupturing is shown graphically.

Examination of Table V-1 and V-2 shows that there exists a definite inverse relationship between the groove depth and rupture pressure for both types of discs. This inverse relationship is shown graphically in Figure V-2. The horizontal axis shows the inverse of groove depth in thousandths of an inch multiplied by 160 to yield quantities of order unity on that axis.

Both sets of points seem to form a straight line indicating that the inverse relationship is exact. Straight lines have been drawn through both sets of points to indicate this. Note that both lines intersect the horizontal axis at or near the point 1. This is encouraging, for one would expect that a disc 0.160 inch thick with grooves that are 0.160 inch deep would indeed rupture at 0 psi.

The two straight lines have nearly the same slope, as can be seen by examining Figure V-2. Provided that the deviations of rupture pressure from the straight line curves and the variation in rupture pressure for similar discs can be explained, it would appear that a preformed disc ruptures at the same pressure as a similar disc that has not been preformed.

Since more points are available for the directly ruptured discs, the curve passing through those points is assumed to be the more exact of the two. Its equation is

$$P = 216\left(\frac{160}{D} - 1\right)$$

where P is the rupture pressure in psi and D is the groove depth in thousandths of an inch. Then,

$$\frac{dP}{dD} = - \frac{34560}{D^2}$$

Thus for a nominal groove depth of 0.050 inch a variation in the depth of the groove of 0.001 inch will cause a variation in rupture pressure of 14 psi. When the uncertainty of the pressure gauge is added to this, the minimum expected variation in rupture pressure for discs with grooves that are 0.050 inches deep is 39 psi. The same reasoning yields an expected variation in rupture pressure for discs with grooves that are 0.012 inch deep of 265 psi.

Whereas the accuracy of the milling machine used to cut the grooves can be expected to be within 0.001 inch, the author feels that the variations in groove depth causing the observed scatter in rupture pressures is caused mainly by the local variations in the thickness of the discs. For when the V-cutter is used to cut a groove it is started at a given location on the disc where the thickness is assumed to be 0.160 inch. Due to the local variations in thickness, the average thickness of the disc can easily vary by 0.001 inch over the length of the groove. When this error is added to the inaccuracy of the milling machine the actual depth of the groove can easily vary by more than 0.001 inch yielding appreciable variations in rupture pressure.

Thus it is clear that the observed scatter in rupture pressures is to be expected. It may now be stated that both for directly ruptured discs and for preformed discs, the rupture pressure P in psi is a function of groove depth D in thousandths of an inch as given by the following relation:

$$P = 216\left(\frac{160}{D} - 1\right)$$

If the reader examines Table V-2 he will note that in preformed discs many different combinations of forming time and aging time were used. This was done to see whether some consistent relation exists between forming time and/or aging time and rupture pressure. This is not the case, as was to be expected. For once a disc has been placed under pressure and creeping is completed, or once it has been removed from the instrument, it is in static equilibrium with its environment so that no changes in its character should appear.

Figure V-3 shows that the average rate of deformation of a disc is proportional to the depth of the grooves cut in the disc. The relation is not linear, which is to be expected. As the groove depth approaches the thickness of the disc, it is only logical that the rate of deformation should increase rapidly, approaching infinity as the thickness of the disc below the groove approaches zero. Similarly, for very shallow grooves the relative thickness and strength of the disc changes much slower than the relative groove depth, yielding a slowly changing rate of deformation as a function of groove depth for shallow grooves. The main purpose of

Figure V-3 is to aid in calculating the center deflection of a disc at rupture. This is done my multiplying the average rupture pressure for discs of a given groove depth by the average rate of deformation of those discs as obtained from Figure V-3.

The data in Tables V-1 and V-2 were obtained from detailed measurements on approximately 60 discs. The data pertaining to some of these discs are shown graphically in Figures V-4 through V-11. For all discs readings were taken along both grooved diameters, and along both diameters at 45° to the grooves. After the first few discs were deformed, it became evident that all discs, independent of the grooves or the depth of the grooves, would deform with axial symmetry, exactly like a disc without grooves. Although the magnitude of the deflections did vary with groove depth, the manner of deforming did not differ from disc to disc. For a given group of discs with equal groove depth it was found that the magnitude of the observed deflections never varied by an amount greater than the inherent inaccuracy of the system. With this in mind, Figure V-4 through V-11 are drawn to represent a cross-section of the groove depths tested in the experiments. Each figure, although showing the data for a particular disc, may be regarded as representative of all discs of the same groove depth.

Each individual figure consists of four graphs. Each set of two graphs, marked (a) and (b), allows the reader to compare the behavior of a disc along a groove and between grooves. The two directions to be graphed were picked at random. Figure (a) shows the deflection as a function of pressure at the center of the disc, and at distances of one and two inches from the center. Inspection of a few graphs will show three distinct regimes of behavior. For pressures below 40 psi, the deflections are linear and elastic. As the limit of elasticity is approached, between 20 psi and 40 psi, the characteristic curve becomes non-linear. Once the limit of elasticity has been exceeded the deflections are again linear functions of pressure but with a smaller slope. The most interesting deviation from linearity is shown in Figure V-5a. This seems to be an example of a disc originally more ductile than the others being work-hardened and finally becoming stiffer than the other discs. Its average rate of deformation and rupture pressure were not affected.

The profiles of the discs at different pressures are shown in figures (b). Note the remarkable symmetry in all cases and the similarity of the profiles along a groove and between grooves. It is of interest to note that, judging from their profiles, all discs seem to behave like membranes under uniform tension rather than thick plates rigidly clamped at the edges.

Figures V-9 through V-12 pertain to pre-formed discs. The main difference is, of course, that the deflections at 0 psi are quite substantial already, and that the profile at 0 psi is not flat. Note in figures (a) that the elastic regime is absent. This is to be expected since the discs were formed at pressures well beyond those at which elasticity may be observed. Note also that near rupture the deflections of the preformed discs are of the same magnitude as those of the directly ruptured discs.

Once it has been ascertained that all discs deform with axial symmetry, the center deflection at rupture is the only quantity needed to calculate to a good approximation the change in volume contributed to a float by a deforming disc. The reader is referred to Appendix C for an explanation of the method used, and to Figure V-12 for the curves showing the change in volume as a function of rupture pressure or depth for preformed discs and for directly ruptured discs. Although the points plotted indicate straight line curves, both curves have been drawn through the origin for reasons of continuity. It is interesting to note that for directly ruptured discs the change in volume ranges from 3.36 to 13.58 cubic inches, while for preformed discs that rupture at depths of interest to oceanographers, the final change in volume lies between 2.54 and 3.04 cubic inches. It is obvious that for oceanographic purposes the use of preformed discs would be preferable and it remains to be shown in the next chapter that their use is also practical.

C. Uncertainties in the Measurements

The largest inaccuracy in the measured deflections is caused by the 25 psi uncertainty that is inherent in the readings of the Acco pressure gauge. Since the average deformation at the center of the discs ranges from 0.0003 to 0.0005 inch per psi, the uncertainty in a measured deflection ranges from 0.0075 to 0.0125 inch, depending on the groove depth of the particular disc. As the measurements are taken at increasing distances from the center, this uncertainty becomes smaller, ranging from 0.0065 to 0.0109 inch at 1 inch from the center and from 0.0041 to 0.0068 inch at 2 inches from the center.

A second source of significant inaccuracy is introduced by the practice of replacing the probes before each measurement. As was explained before, each disc is marked to indicate the proper location of the pointed probes, and when placed carefully no probe will deviate more than 1/64 inch in any direction from its required location. The worst case occurs during the maximum deformation of a disc, for then the average slope is a maximum. Any error in the horizontal placing of the probe along a curve of maximum slope will result in a maximum uncertainty in the reading of the dial gauge. In these experiments the greatest observed center deflection was 0.875 inch at 2400 psi. The average slope is then 0.292 causing a maximum possible uncertainty of 0.005 inch. In actual tests, repeated readings during this worst conditions showed that no two readings for the same location ever differed by more than 0.003 inch. At lower pressures and smaller deflections this error becomes significantly smaller.

Other sources of inaccuracy are present in the instrument but their contribution to the total inaccuracy is consistently less than 0.001 inch. The dial gauge is accurate within 0.0005 inch. Uncertainties due to uneven travel of the tube with respect to the bottom plate were measured and found to be less than 0.001 inch. Uncertainties caused by variations in the actual lengths of the probes are also less than 0.001 inch.

Whereas in the worst case, a maximum uncertainty of 0.016 inch in the measurements may be expected, the average maximum uncertainty will range from 0.005 to 0.008 inch. Since the average deflection is approximately 0.300 inch it follows that the instrument operates with an average uncertainty of 2 percent.

A source of uncertainties whose magnitudes are difficult to estimate is the variation in thickness and composition of individual discs. Since it is known that the thickness varies randomly by 2 percent, one may assume that the uncertainty contributed by variations in the material is at least 1 percent. Thus, bearing in mind that the instrument operates with a 2 percent uncertainty it is clear that the curves in Figures V-4 through V-11 deviate from true curves for perfect discs by as much as 5 percent.

Table V-1. Data pertaining to directly ruptured discs.
(continued on page 58)

Disc No.	Groove Depth (inch)	Rupture Pressure (psi)	Average (psi)	Percentage Deviation	Rate of Deform (.001 inch/psi)	Average	Max.Center Deflection (inch)	See Fig.
49								
49	.050	475			.515			V-4
50	.050	475	463^{+12}_{-13}	2.8	.513	.510	.236	
51	.050	450			.513			
52	.050	450			.498			
1	.040	550			.458			
2	.040	625	583^{+42}_{-33}	7.2	.458	.458	.252	
3	.040	575			.458			
17	.035	775			.438			
18	.035	775	800^{+75}_{-25}	9.4	.438	.437	.350	
19	.035	875			.435			
20	.035	775			.438			
4	.030	850			.428			V-5
5	.030	825	842^{+8}_{-17}	2.2	.432	.431	.364	
6	.020	850			.432			
31	.028	1225			.408			
32	.028	1200	1175^{+50}_{-125}	10.6	.410	.408	.480	
33	.028	1225			.407			
34	.028	1050			.406			
21	.026	1150			.390			
22	.026	1175	1206^{+69}_{-56}	5.7	.390	.388	.468	
23	.026	1225			.388			
24	.026	1275			.386			

Disc No.	Groove Depth (inch)	Rupture Pressure (psi)	Average (psi)	Percentage Deviation	Rate of Deform (.001 inch/psi	Average	Max.Center Deflection (inch)	See Fig.
26	.022	1300			.373			V-5
27	.022	1350	1333^{+17}_{-33}	2.5	.370	.371	.495	
28	.022	1350			.370			
7	.020	1375			.373			
8	.020	1450	1392^{+58}_{-42}	4.2	.372	.376	.524	V-6
9	.020	1350			.382			
35	.018	2100			.374			
36	.018	1825	1875^{+225}_{-175}	12.0	.378	.376	.705	
37	.018	1875			.378			
38	.018	1700			.375			
10	.016	2250			.3444			V-7
11	.016	1850	2056^{+194}_{-206}	10.0	.364	.354	.727	
15	.016	1975			.358			
16	.016	2150			.350			
39	.014	2125			.358			
40	.014	2225	2169^{+56}_{-44}	2.6	.370	.370	.803	
41	.014	2125			.371			
42	.014	2200			.381			
53	.012	2625			.362			
54	.012	2700	2633^{+67}_{-58}	2.5	.345	.353	.930	
56	.012	2575			.363			V-8

Table V-2. Data pertaining to pre-formed discs.

Disc No.	Groove Depth (inch)	Forming Pressure (psi)	Forming Time	Center Deflection (inch) During	After	Aging Time	Rupture Pressure (psi)	Average (psi)	Percentage Deviation	See Fig.
12	.030	750	1 min	.325	.193	20 hr	875	875^{+25}_{-25}	2.9	V-9
13	.030	750	30 min	.322	.194	42 hr	875			
14	.030	750	16 hr	.326	.199	25 hr	875			
29	.022	1200	30 min	.466	.347	75 days	1525	1450^{+75}_{-75}	5.2	
30	.022	1200	1 min	.465	.345	49 days	1375			V-10
45	.018	1600	72 hr	.598	.496	25 days	1675	1875^{+200}_{-200}	10.7	V-11
46	.018	1600	20 min	.627	.532	5 min	2075			
47	.014	2000	30 min	.743	.644	4 hr	2425	2438^{+12}_{-13}	0.5	
48	.014	2000	19 hr	.743	.657	22 hr	2450			
55	0.12	2400	117 hr	.879	.805	20 min	2625	2775^{+150}_{-150}	5.4	
53	.012	2400	3 hr	.884	.803	48 hr	2925			

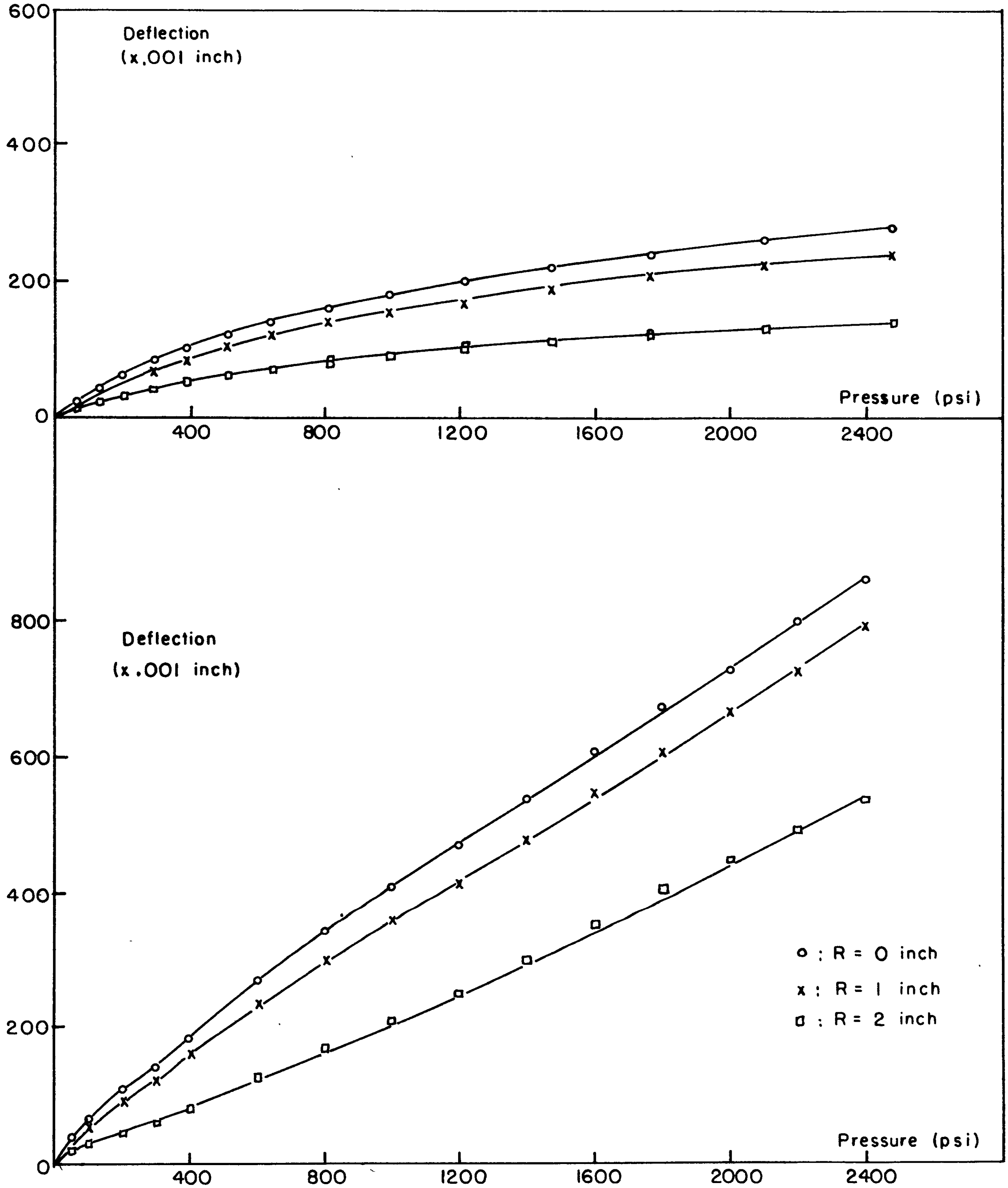


Figure V-1a. Deflection vs. Pressure at several distances from the center for a disc without grooves.
Top: Theoretical. Bottom: Experimental (disc no. 57).
No rupture.

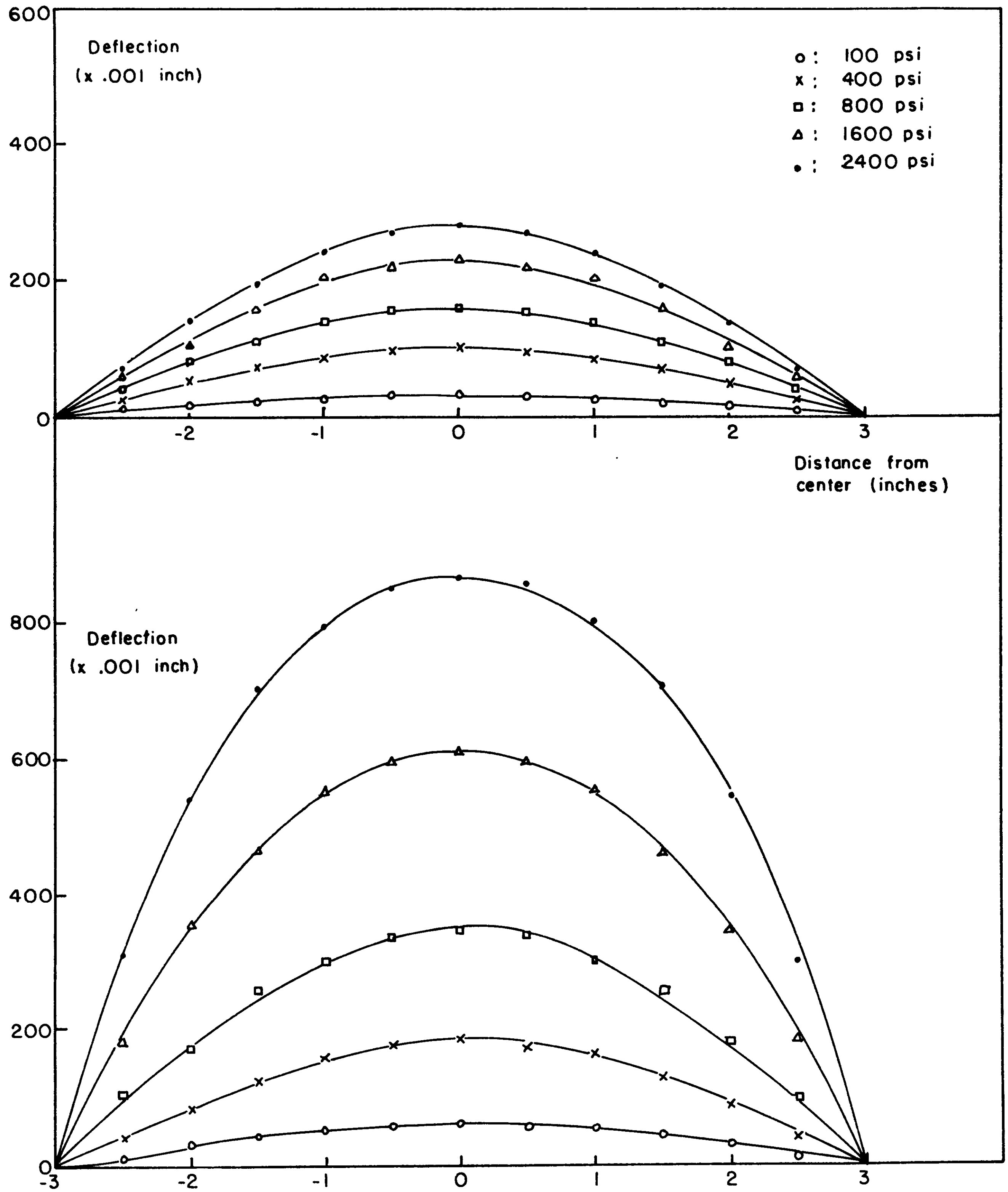


Figure V-1b. Profiles through center of a disc without grooves.
Top: Theoretical. Bottom: Experimental (disc no. 57).
No rupture.

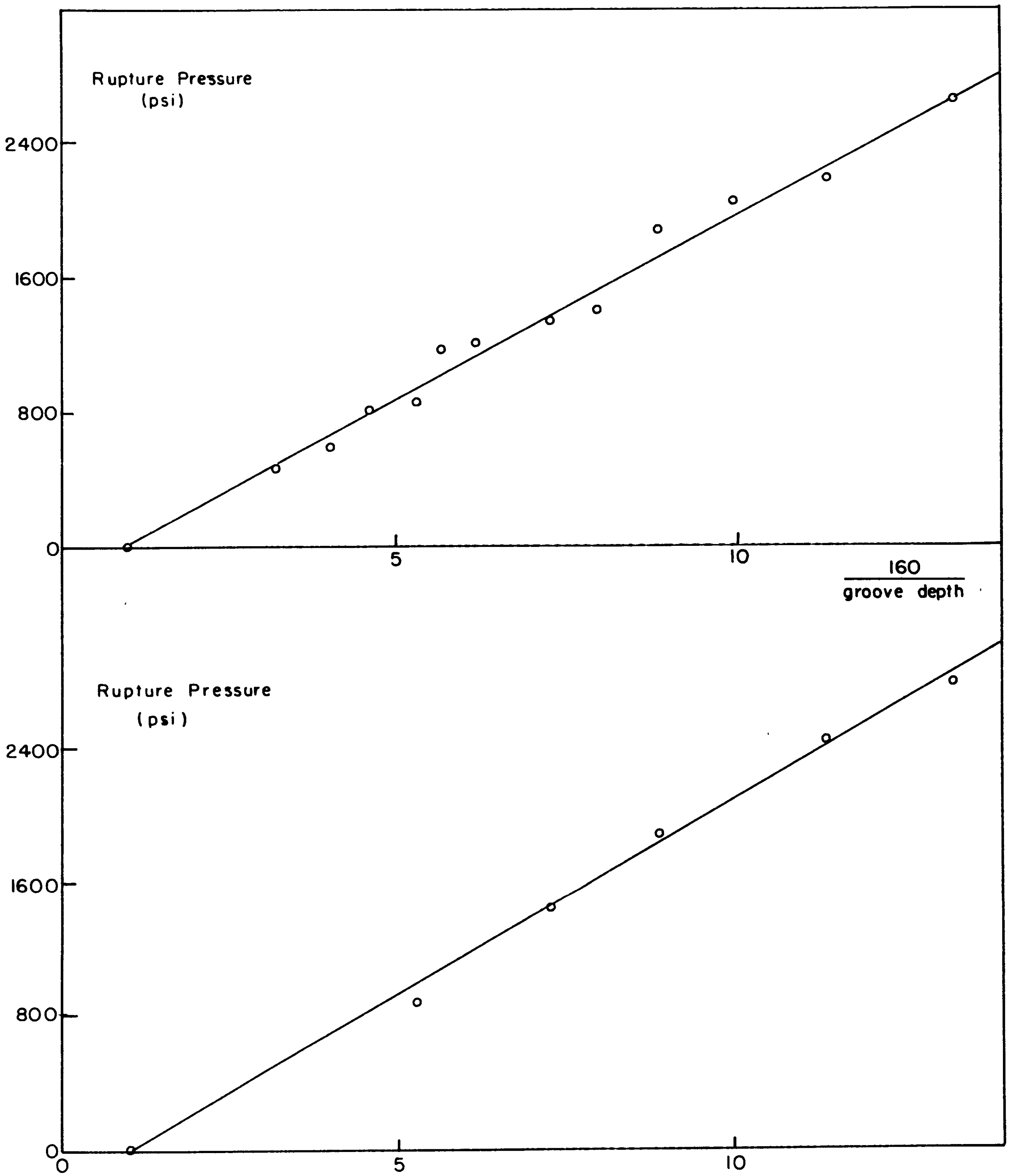


Figure V-2. Relation between rupture pressure and groove depth in thousandths of an inch.
Top: For discs directly ruptured.
Bottom: For pre-formed discs.

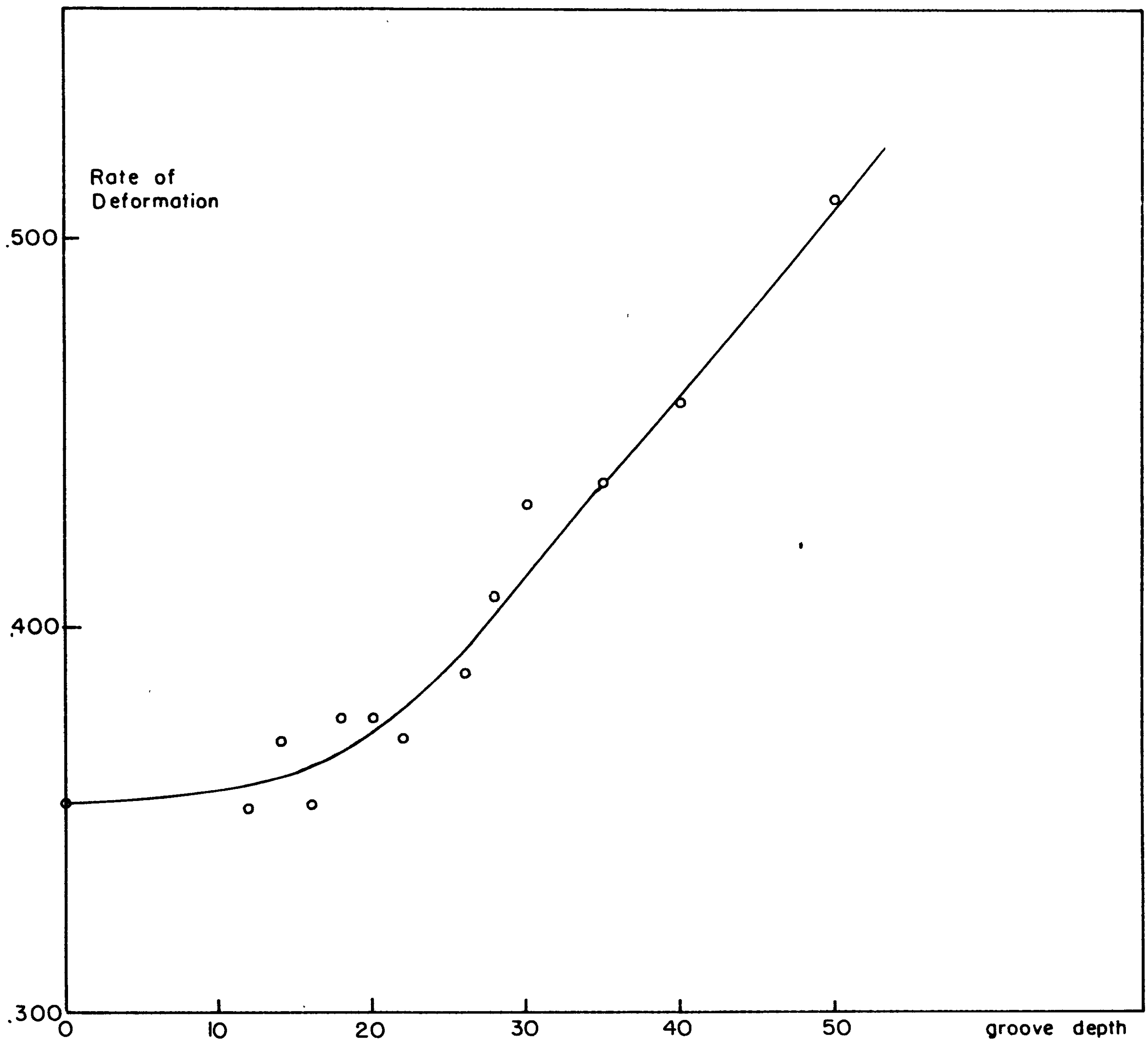


Figure V-3. Relation between average rate of deformation in thousandths of an inch per psi of applied normal pressure and groove depth in thousandths of an inch.
Data pertains to directly ruptured discs only.

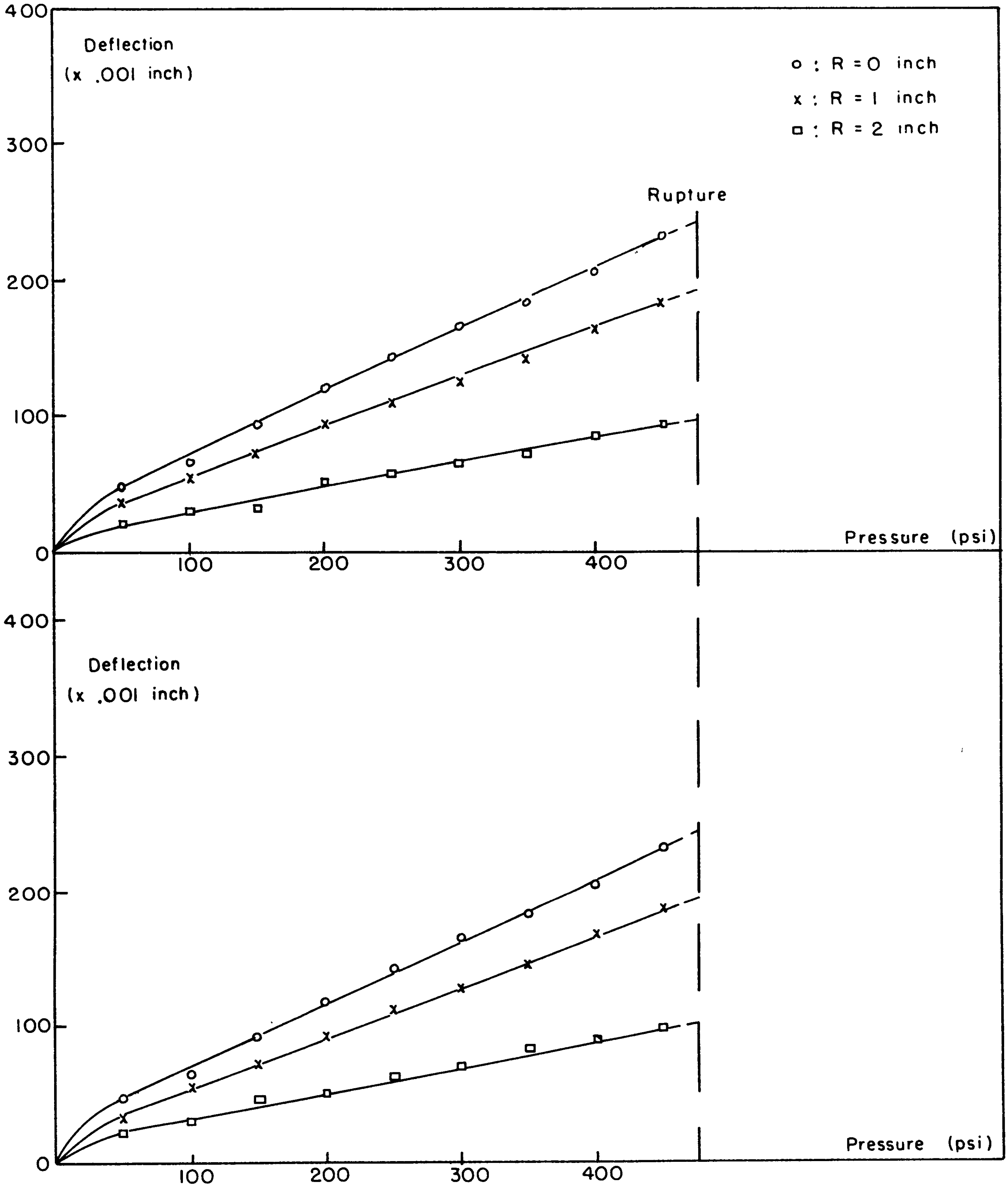


Figure V-4a. Deflection vs. Pressure at several distances from the center (disc no. 49).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .050 inch.
Average Rupture Pressure (discs no. 49, 50, 51, 52): 463 psi.

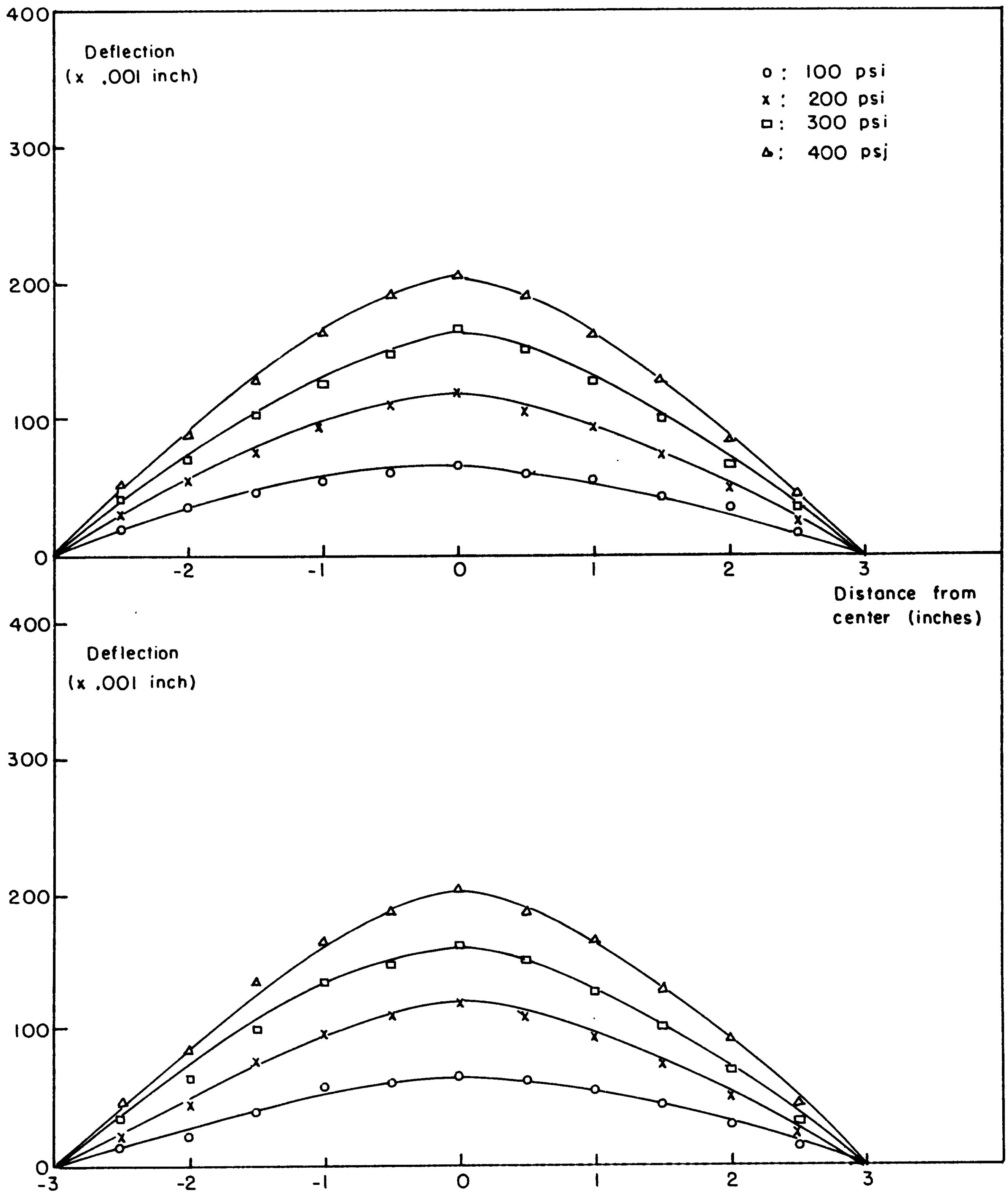


Figure V-4b. Profiles through center of disc (disc no. 49).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .050 inch.
Average Rupture Pressure (discs no. 49, 50, 51, 52): 463 psi.

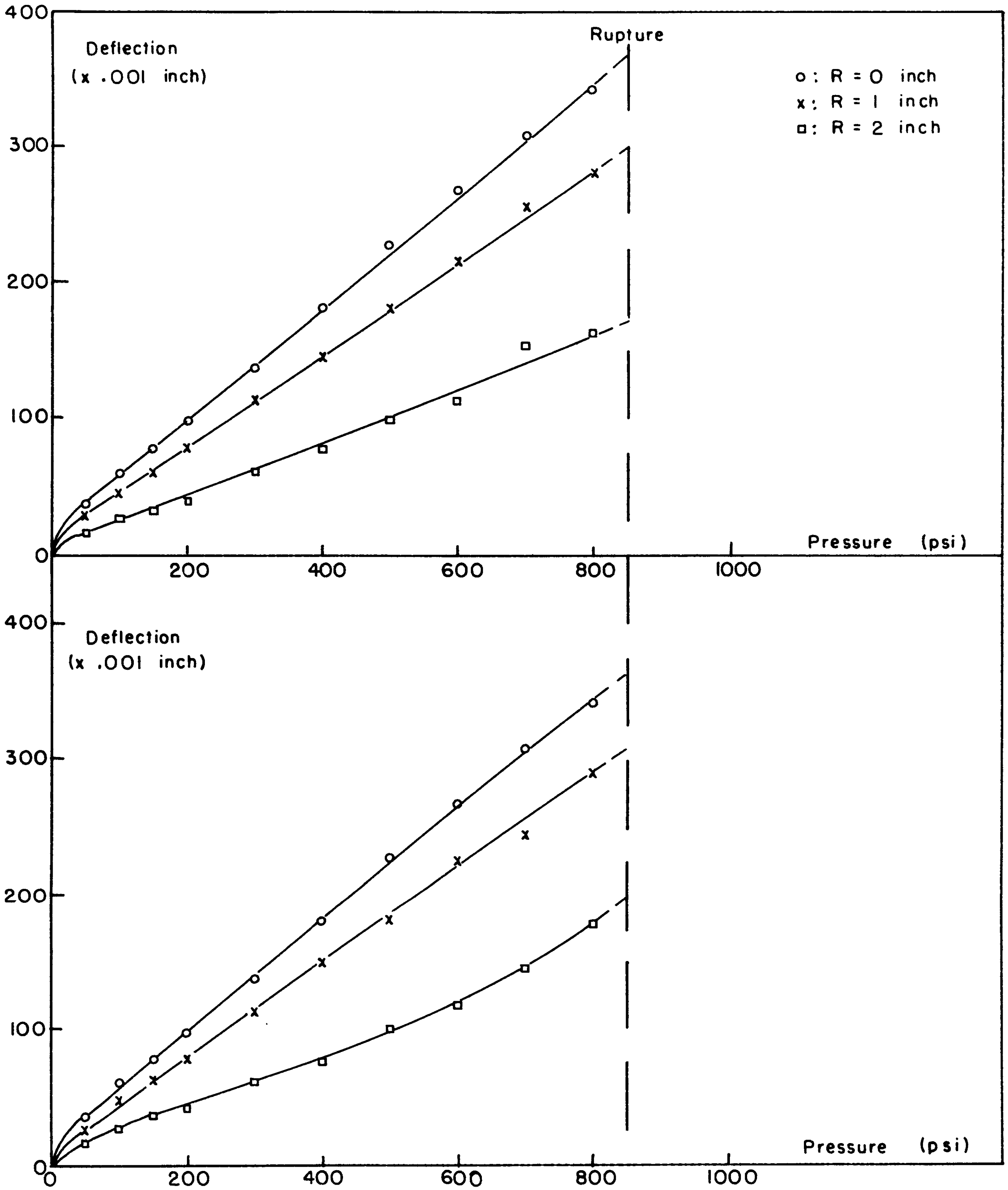


Figure V-5a. Deflection vs. Pressure at several distances from the center (disc no. 4).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .030 inch.
Average Rupture Pressure (discs no. 4, 5, 6): 842 psi.

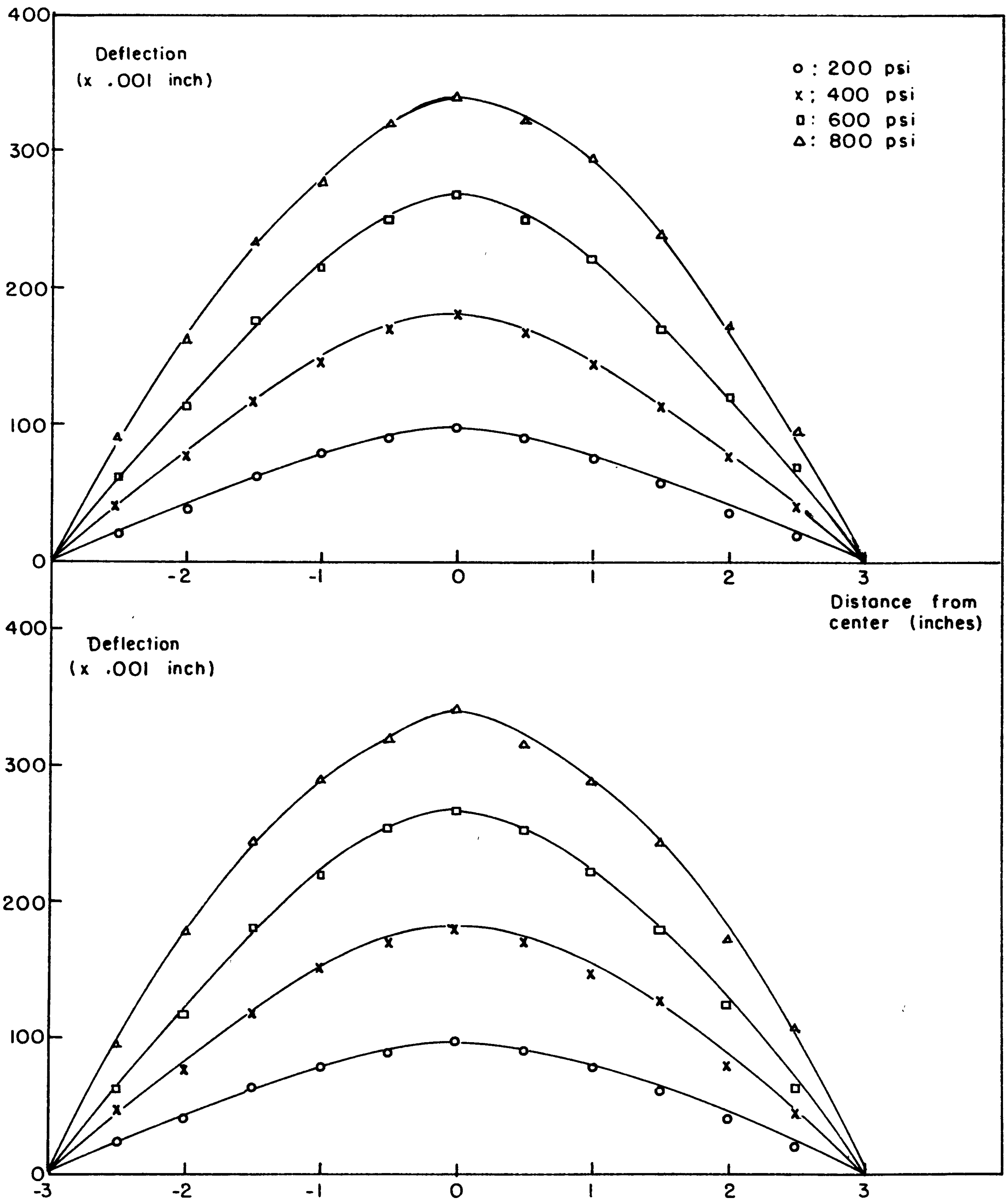


Figure V-5b. Profiles through center of disc (disc no. 4).
Top: Between the grooves. Bottom: Along the grooves.
Groove depth: .030 inch.
Average Rupture Pressure (discs no. 4, 5, 6): 842 psi.

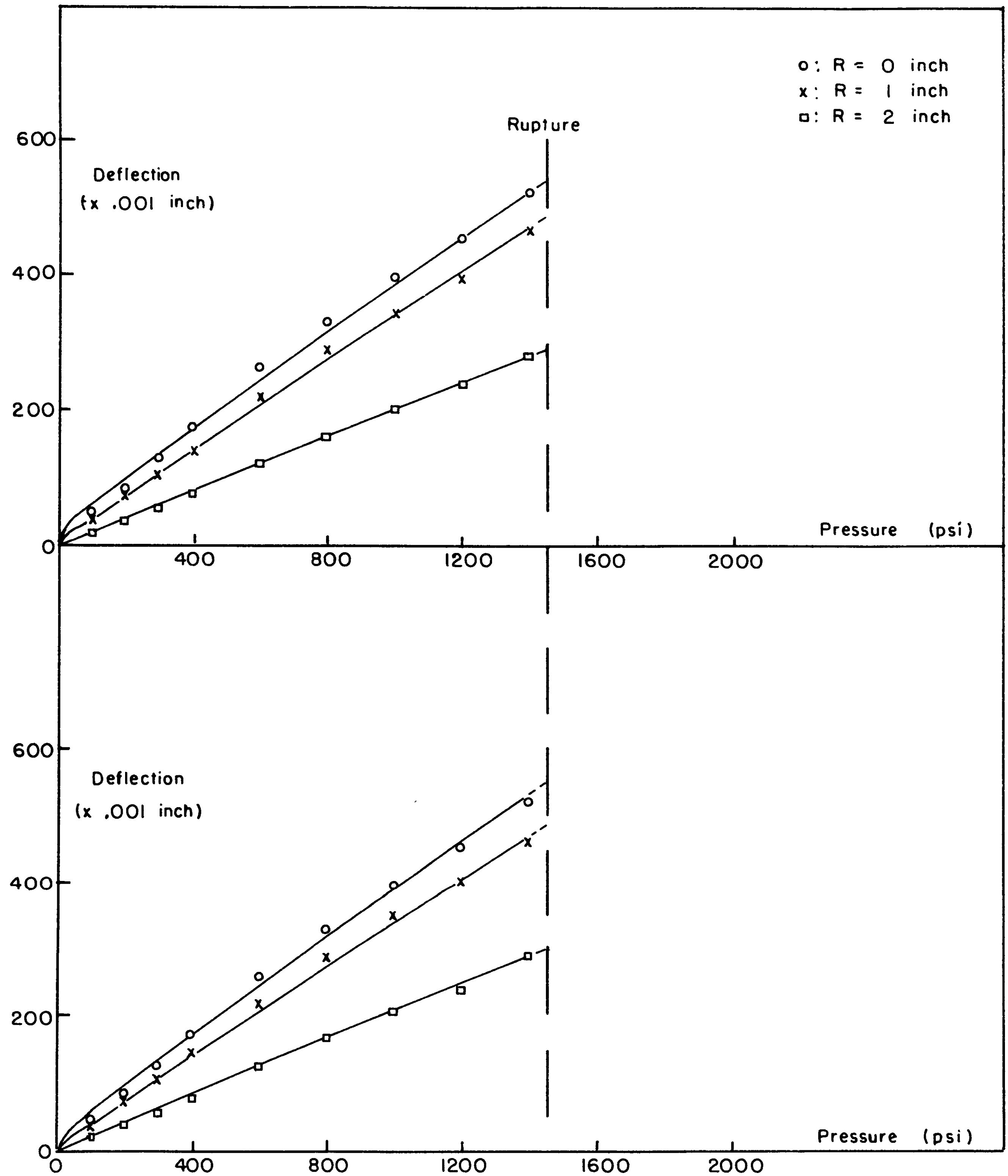


Figure V-6a. Deflection vs. Pressure at several distances from the center (disc no. 8).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .020 inch.
Average Rupture Pressure (discs no. 7, 8, 9): 1392 psi.

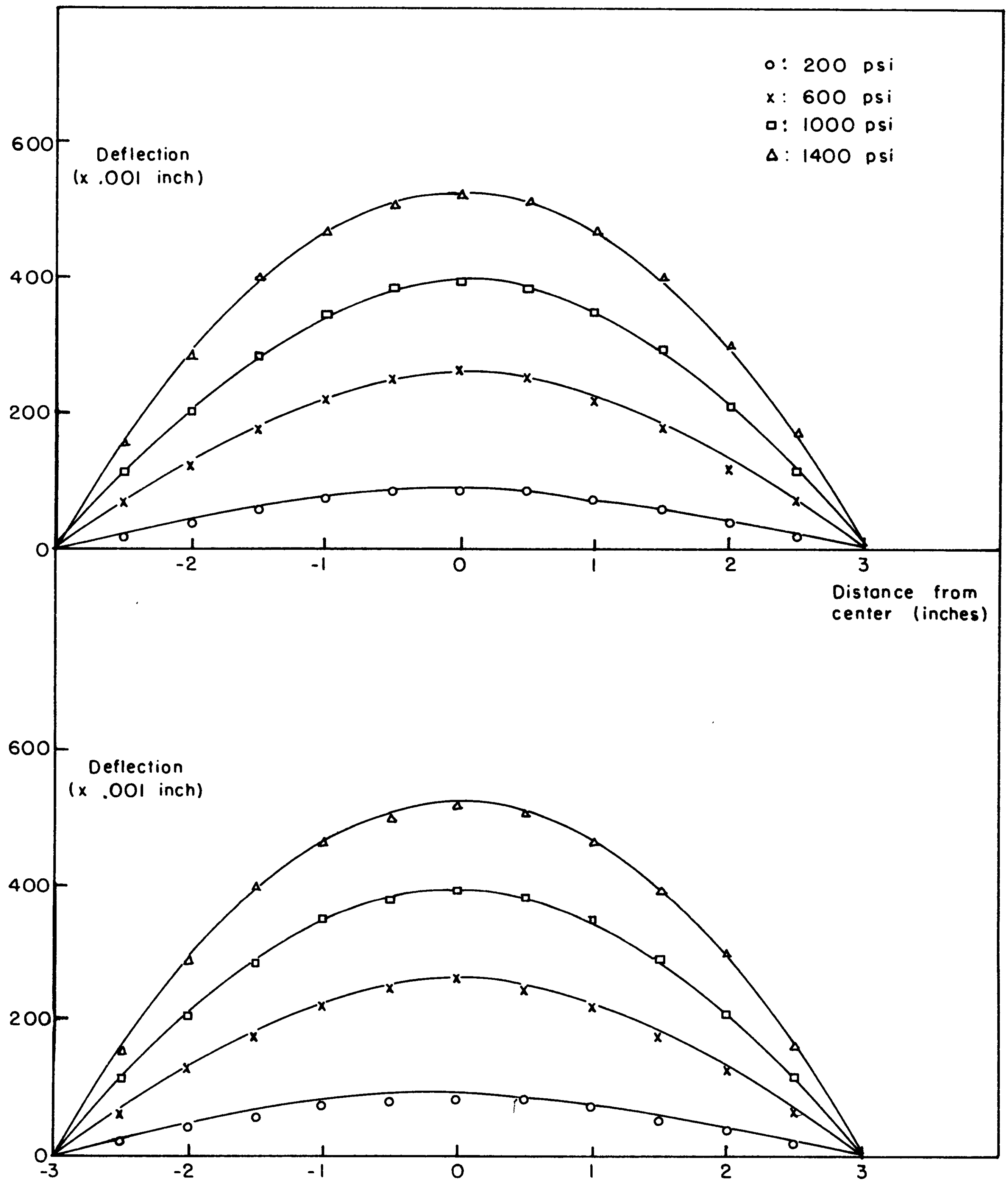


Figure V-6b. Profiles through center of disc (disc no. 8).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .020 inch.
Average Rupture Pressure (discs no. 7, 8, 9): 1392 psi.

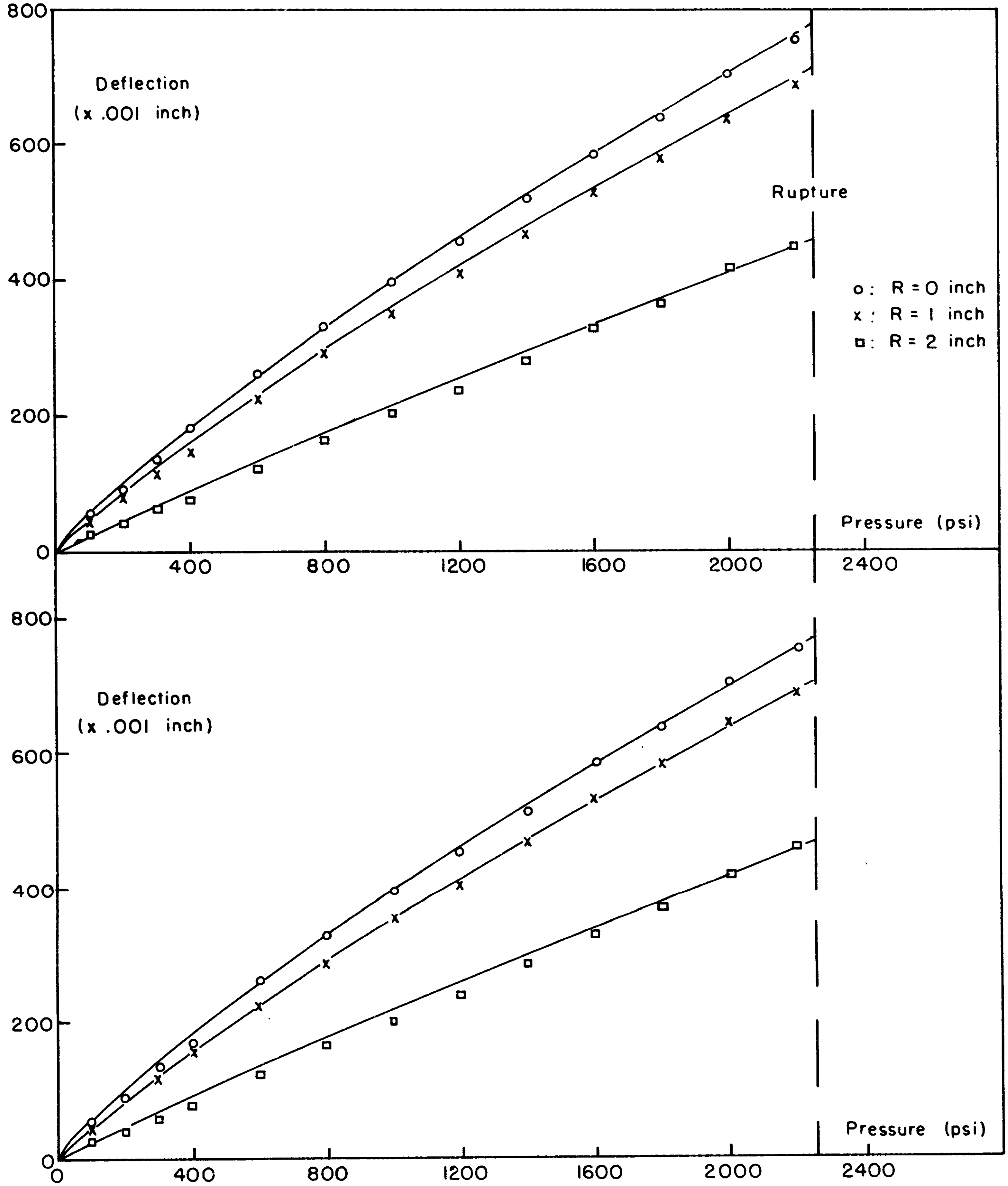


Figure V-7a. Deflection vs. Pressure at several distances from the center (disc no. 10).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .016 inch.
Average Rupture Pressure (discs no. 10, 11, 15, 16): 2056 psi.

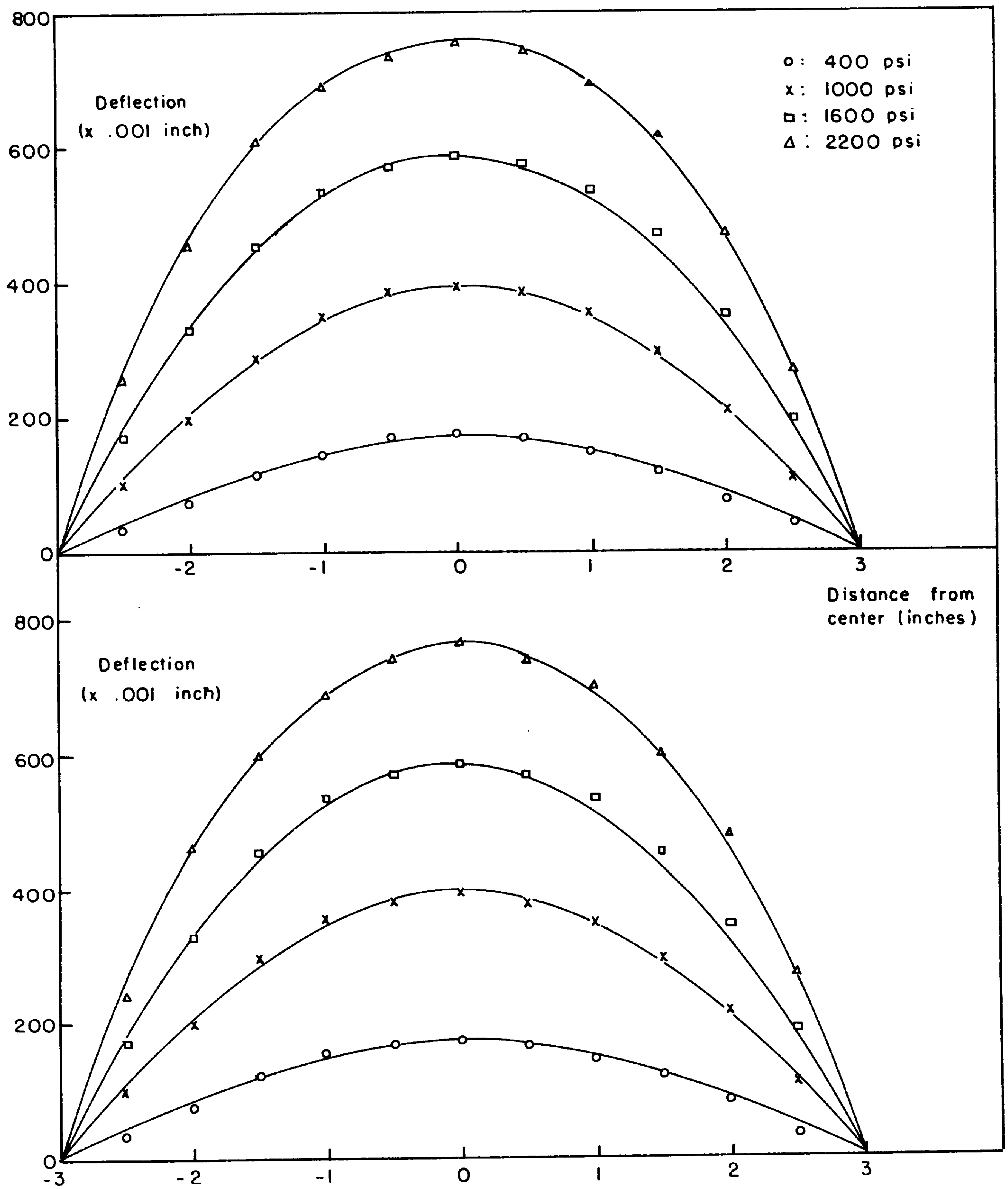


Figure V-7b. Profiles through center of disc (disc no. 10).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .016 inch.
Average Rupture Pressure (discs no. 10, 11, 15, 16): 2056 psi.

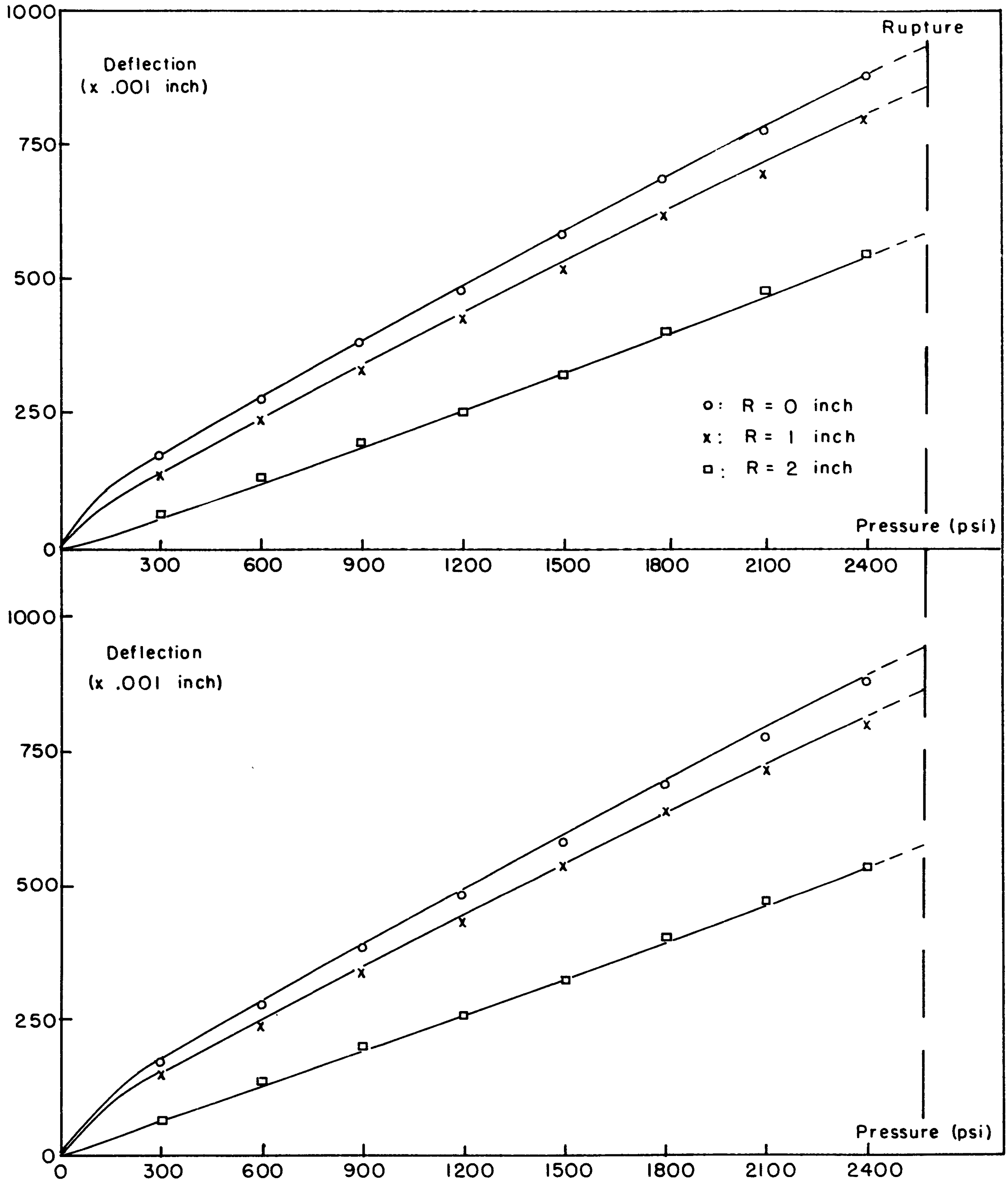


Figure V-8a. Deflection vs. Pressure at several distances from the center (disc no. 56).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .012 inch.
Average Rupture Pressure (discs no. 53, 54, 55, 56): 2633 psi.

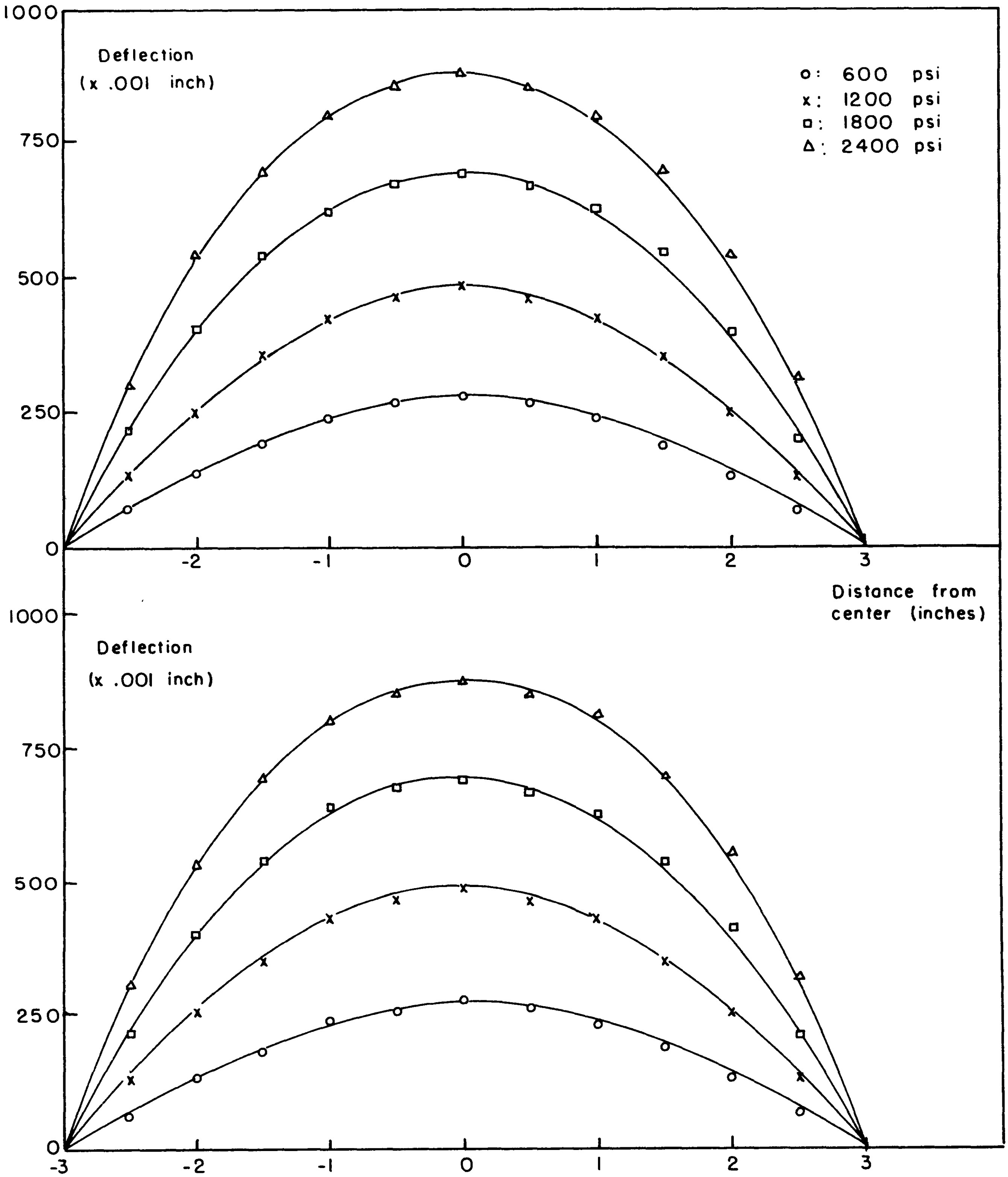


Figure V-8b. Profiles through center of disc (disc no. 56).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .012 inch.
Average Rupture Pressure (discs no. 53, 54, 55, 56): 2633 psi.

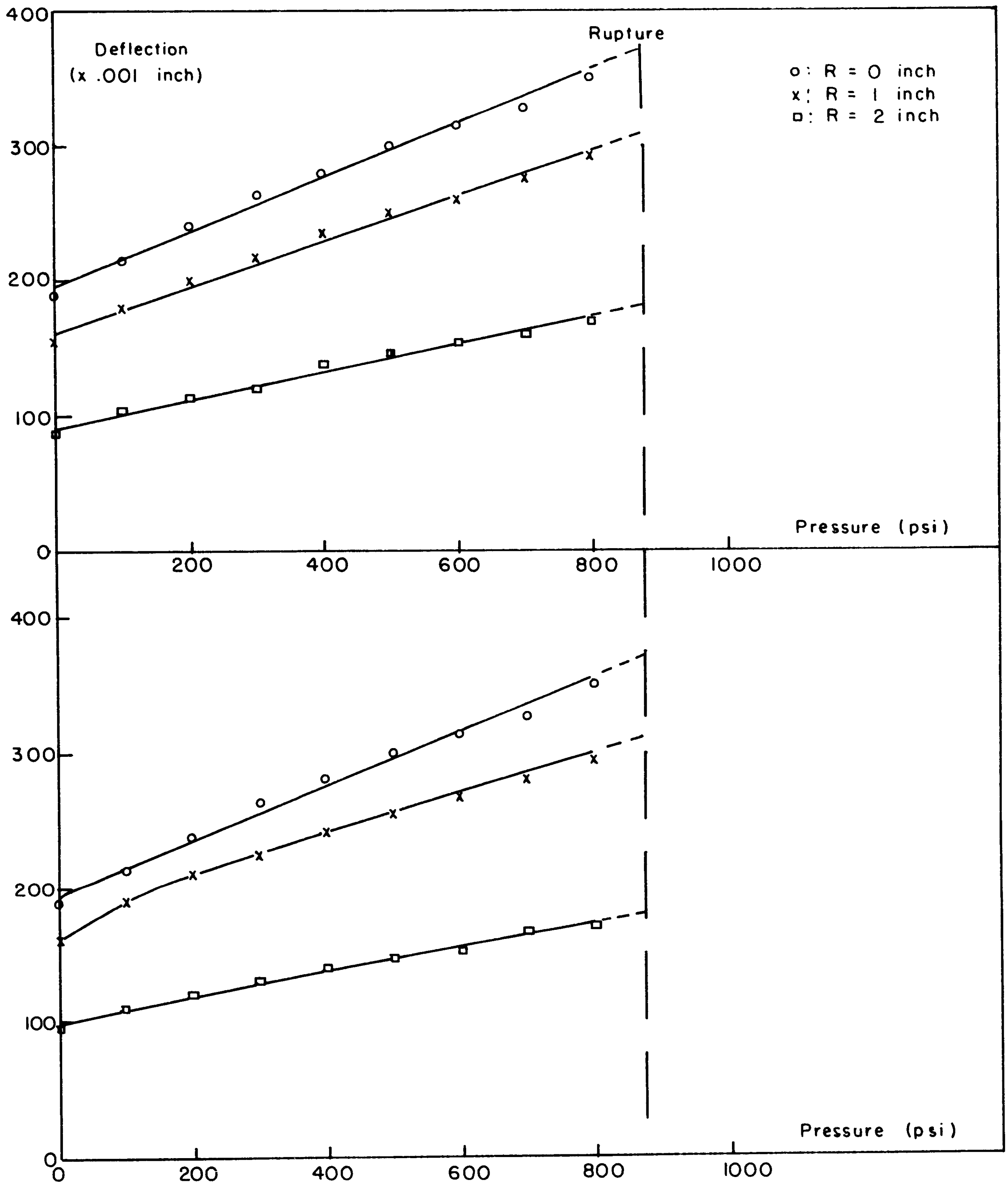


Figure V-9a. Deflection vs. Pressure at several distances from the center for a pre-formed disc (disc no. 12).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .030 inch.
Average Rupture Pressure (discs no. 12, 13, 14): 875 psi.

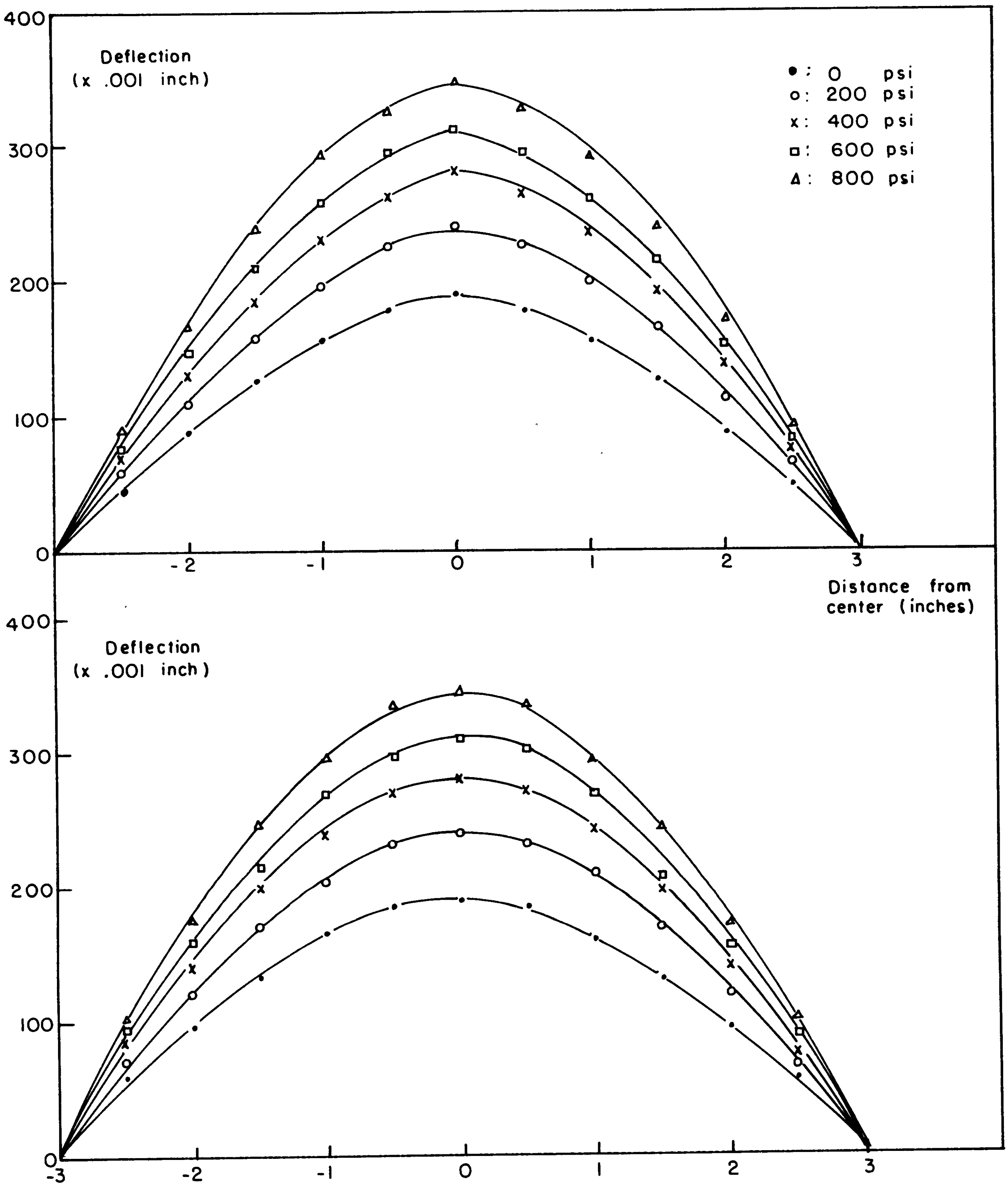


Figure V-9b. Profiles through center of a pre-formed disc (disc no. 12).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .030 inch.
Average Rupture Pressure (discs no. 12, 13, 14): 875 psi.

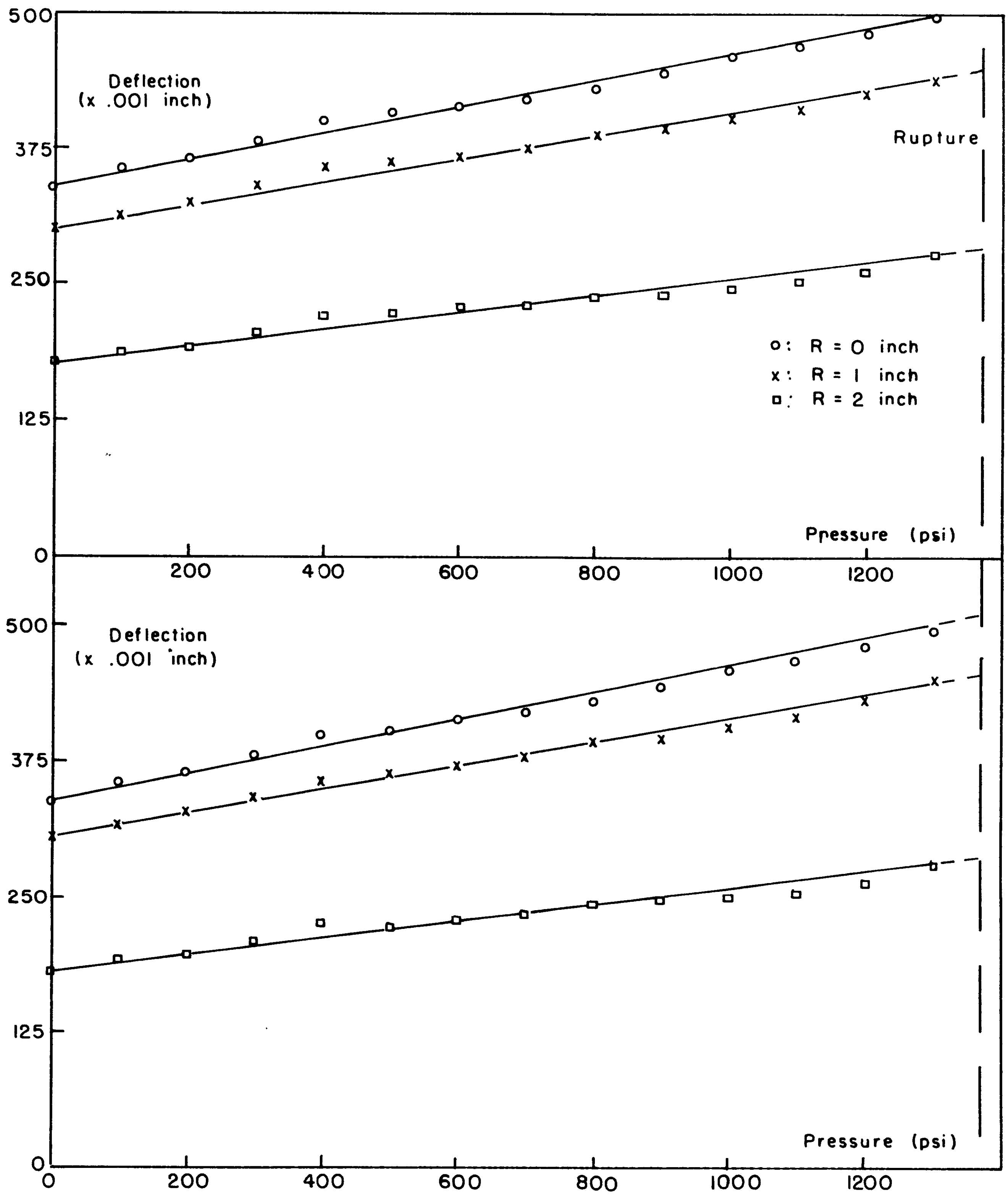


Figure V-10a. Deflection vs. Pressure at several distances from the center for a pre-formed disc (disc no. 30).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .022 inch.
Average Rupture Pressure (discs no. 29, 30): 1450 psi.

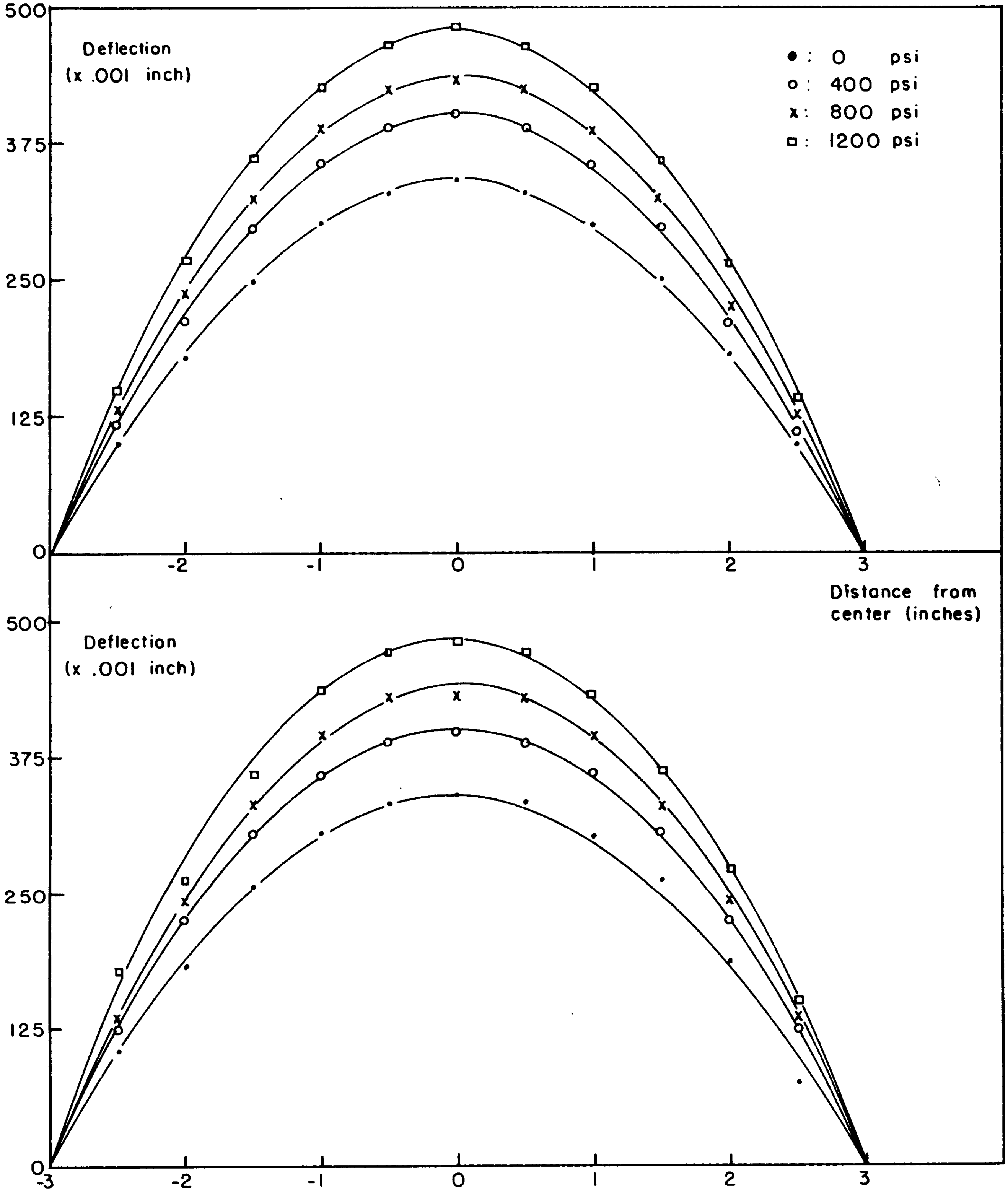


Figure V-10b. Profiles through center of a pre-formed disc (disc no. 30).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .022 inch.
Average Rupture Pressure (discs no. 29, 30): 1450 psi.

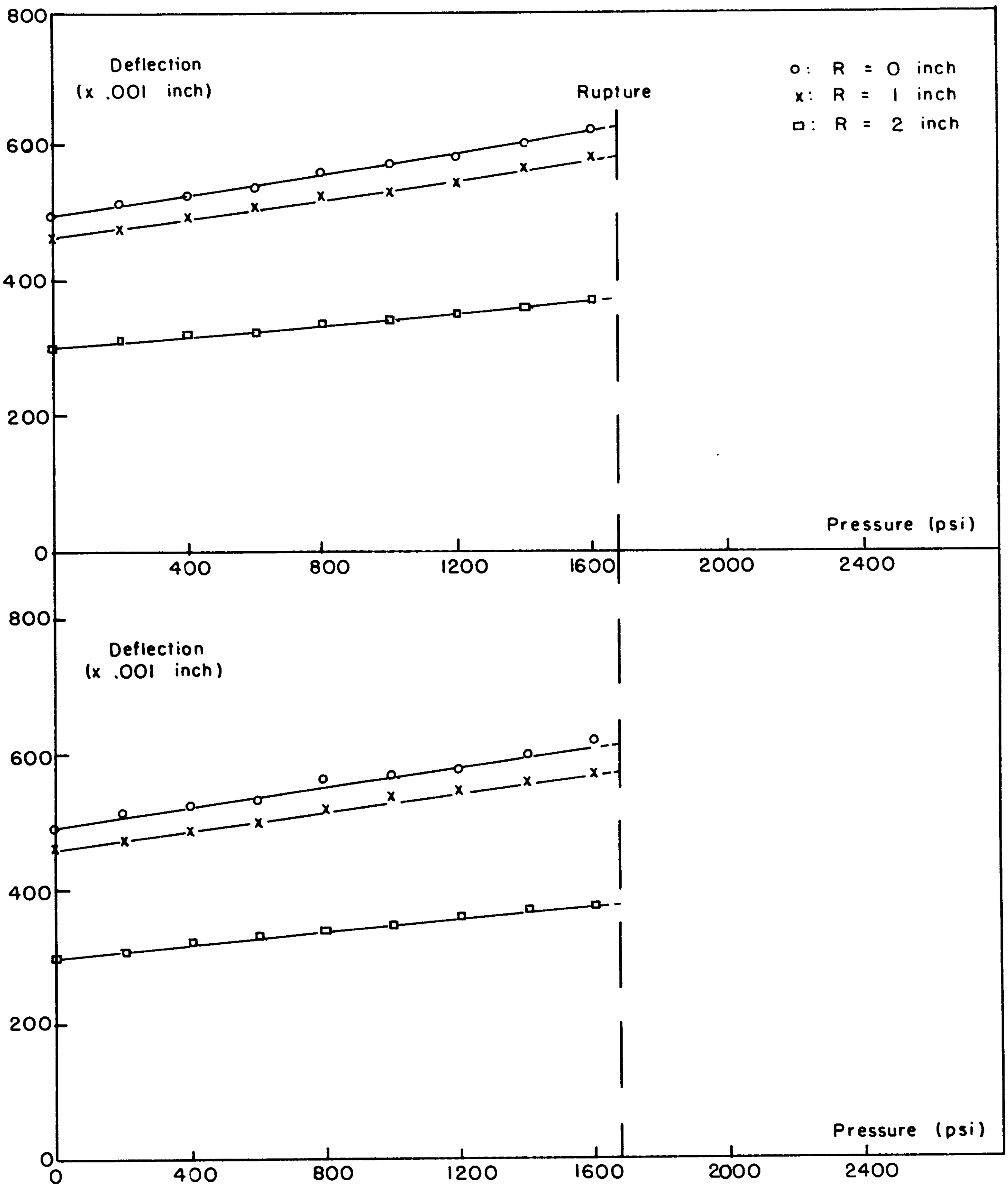


Figure V-11a. Deflection vs. Pressure at several distances from the center for a pre-formed disc (disc no. 45).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .018 inch.
Average Rupture Pressure (discs no. 45, 46): 1875 psi.

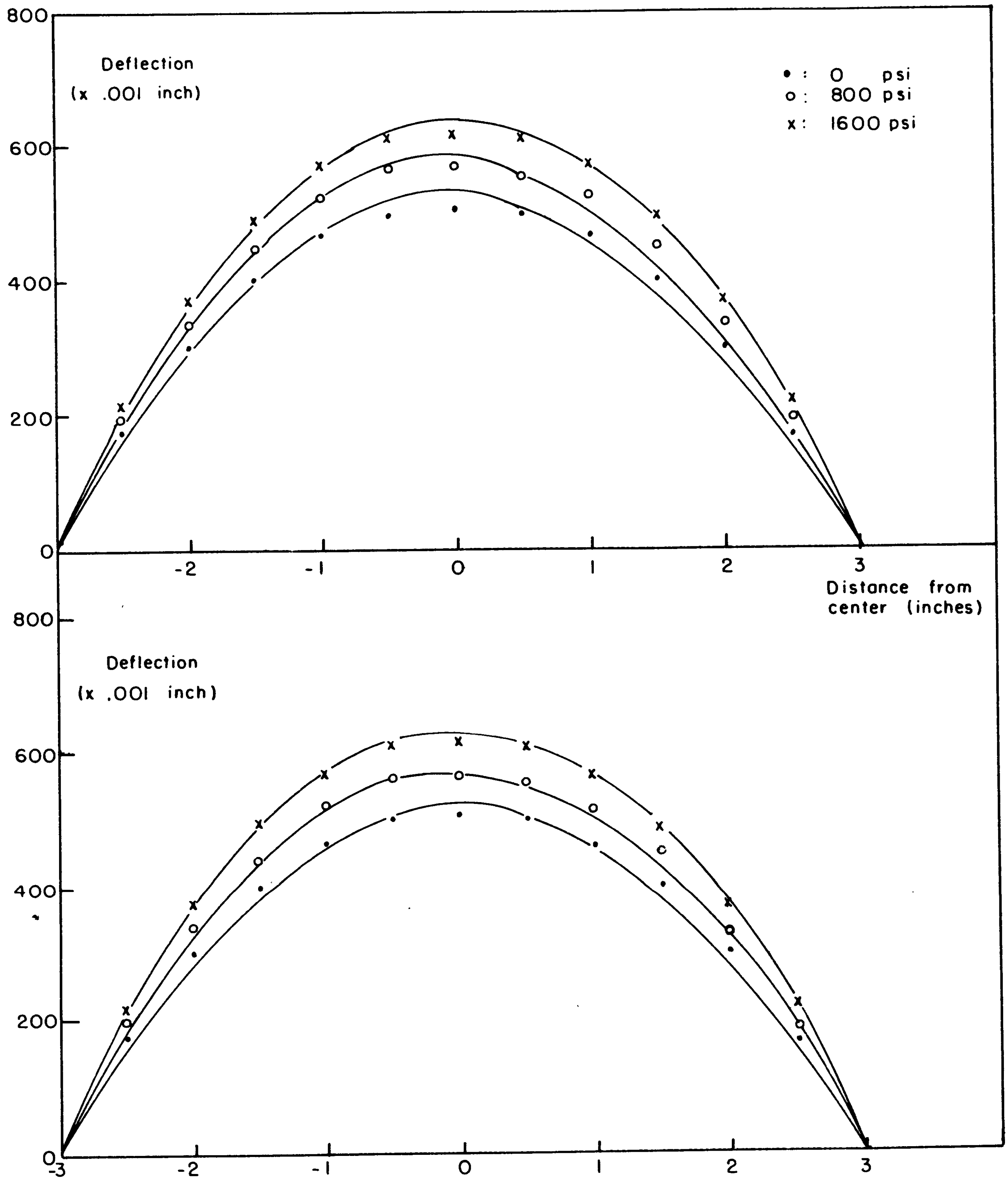
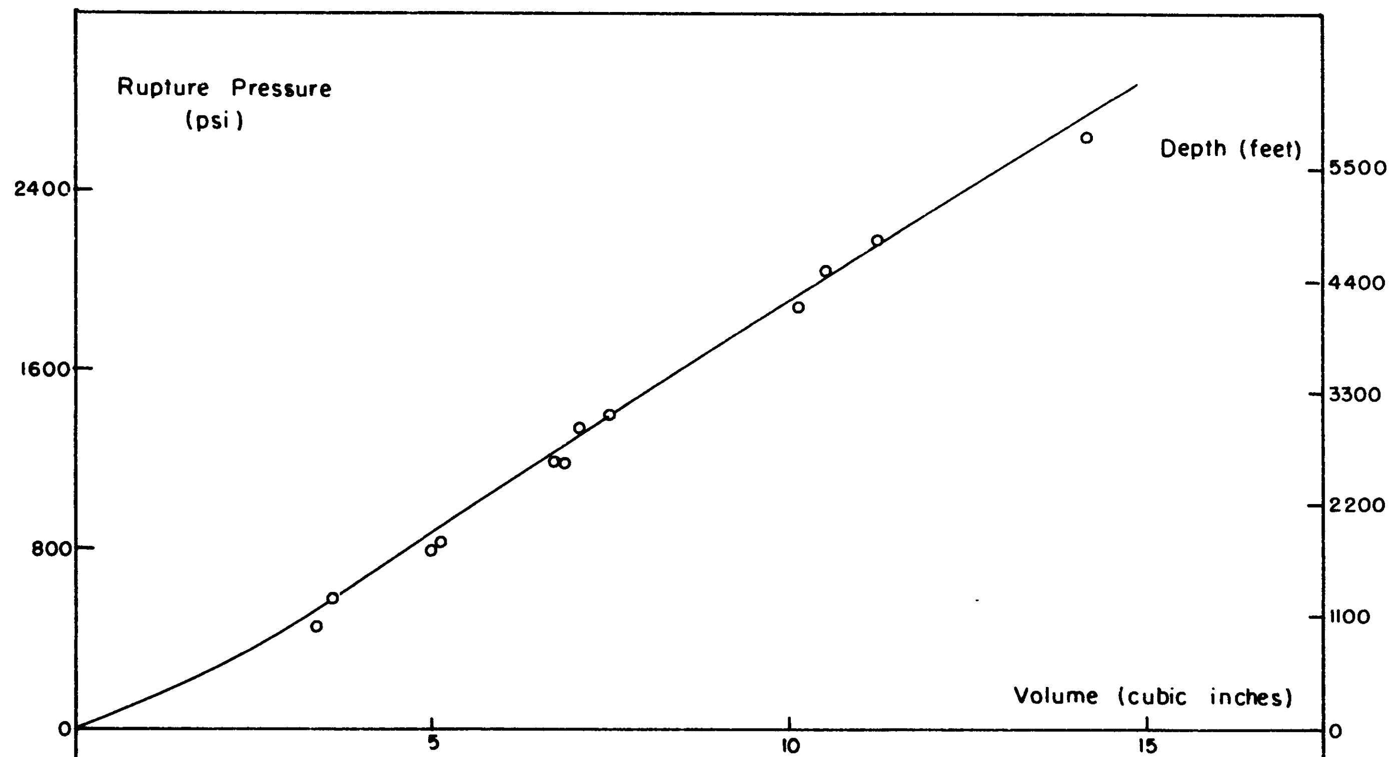


Figure V-11b. Profiles through center of a pre-formed disc (disc no. 45).
Top: Between the grooves. Bottom: Along a groove.
Groove depth: .018 inch.
Average Rupture Pressure (discs no. 45, 46): 1875 psi.



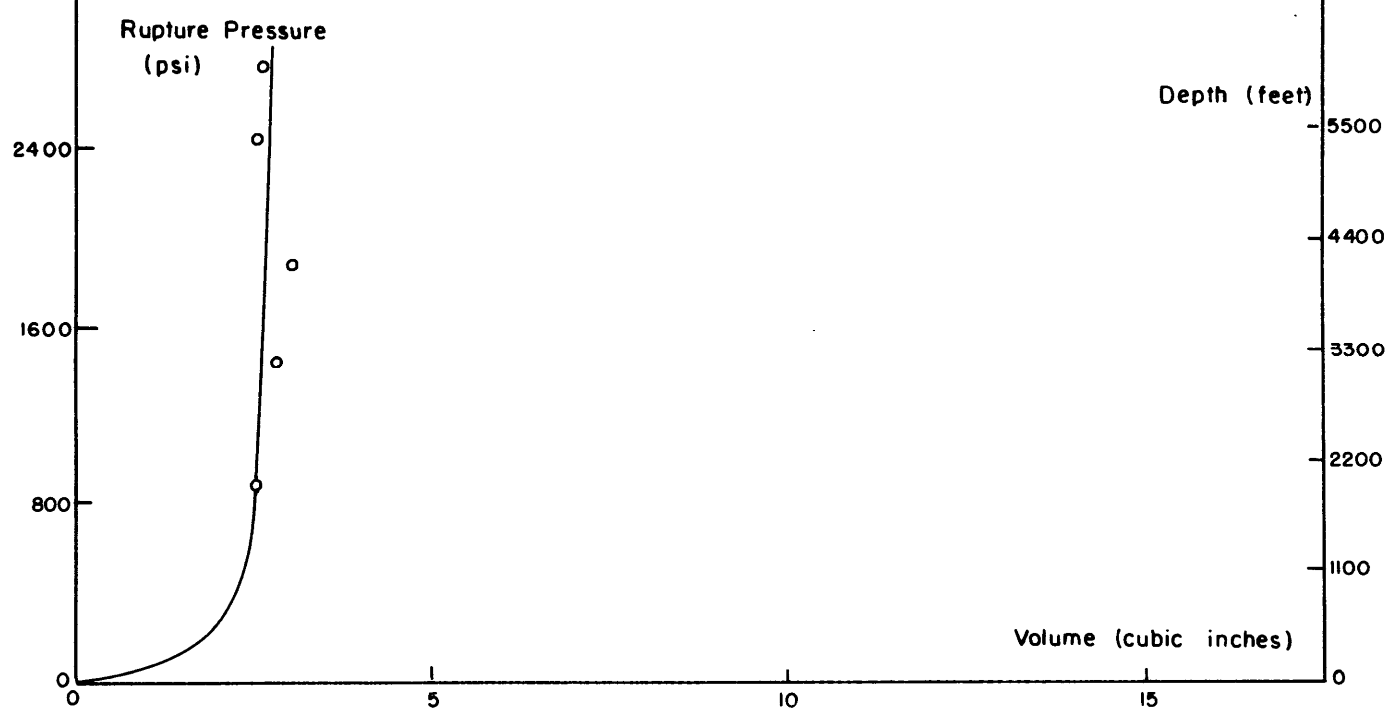


Figure V-12. Volume change vs. Rupture Pressure or Depth.
Top: Directly Ruptured discs.
Bottom: Pre-formed discs.

VI. OCEANOGRAPHIC APPLICATIONS

The experiments described in the previous chapter have shown that a preformed disc undergoes a far smaller change in volume before rupturing than a flat disc. It remains to be shown that even with that small volume change a float equipped with a preformed disc can remain neutrally buoyant.

To use a typical case as an example, assume that a float is released to be neutrally buoyant at a depth of 2000 feet and to implode eventually in the SOFAR channel at a depth of 4000 feet. Assume that the float consists of an aluminum tube of type 6061-T6 with an outer diameter of 7.0 inches and a wall thickness of 0.5 inch.

At the surface the density of the float must exceed that of the sea water in order to sink. At a depth of 2000 feet the seawater will have increased its density by 0.22 percent (von Arx, 1962). As the tube sinks it becomes compressed due to the increasing pressure of its environment. The change r in the outer radius R_2 is given by

$$r = \frac{-R_2 p}{E}\left[\frac{1 + (R_1/R_2)^2}{1 - (R_1/R_2)^2} - u\right]$$

where R_1 is the inner radius, p is the exterior pressure, E is Young's modulus, and u is Poisson's ratio. At a depth of 2000 feet the value of r is -0.0019 inch, causing a decrease in the volume of the float of 0.11 percent. Since this change is very small the approximate increase in the density of the float contributed by the compressing tube will also be 0.11 percent. Thus for the float to be neutrally-buoyant at 2000 feet the increase in the density of the float due to the deforming rupture disc can be no greater than 0.11 percent. If it were as great as 0.11 percent, the float would change its density at exactly the same rate as the seawater surrounding it as it sank. This means that any amount of negative buoyancy imparted to the float at the surface to make it sink initially would cause the float to remain heavier than the water surrounding it, and go directly to the bottom of the ocean. Therefore, to assure that the float can be made to sink at the surface yet still become neutrally-buoyant at 2000 feet, one must be certain that the float will increase its density at a significantly slower rate than the seawater through which it sinks. Thus assume that the change in the density of the float contributed by the deforming rupture disc should be no greater than, say 0.06 percent. Then the change in volume of the float at 2000 feet due to the deformed disc should represent no more than 0.06 percent of the original volume of the float.

At a depth of 4000 feet where rupture occurs the volume change of the disc will be approximately 3 cubic inches (see Figure V-12). Since the center deflection changes linearly with depth (see Figure V-11a) and the volume is almost directly proportional to the center deflection (see the table in Appendix C, column 5), the volume change at 2000 feet would be approximately 1-1/2 cubic inches. Since this must be no more than 0.06 percent of the original volume of the float, it follows that the smallest float that will achieve neutral buoyancy with a preformed disc has a volume of 2500 cubic inches. For a float with an outer diameter of 7.0 inches this means a minimum length of 65.0 inches, or approximately 5-1/2 feet. All the above assumes, of course, that the opposite end of the tube is closed by a thick disc which undergoes negligible deformation.

Considering the minimum dimensions for the float of the example, it is clear that the use of neutrally-buoyant floats equipped with pre-formed rupture discs for signalling purposes is certainly practical, provided that the amount of energy released by the implosions is detectable over distances of the order of hundreds of miles, rather than miles as is the case with most electronic signalling devices used presently with neutrally-buoyant floats. In order to determine more precisely the distance over which implosions caused by ruptured discs would be audible, a field experiment was conducted in July of 1964 using the SCFAR listening stations of the Pacific Missile Range and the facilities of the Bureau of Commercial Fisheries, U.S. Fish and Wildlife Service, in Honolulu, Hawaii.

Since for this field experiment neutral buoyancy was of no consideration, the devices were manufactured from aluminum tubes of type 6061-T6 with an outer diameter of 6.75 inch and an inner diameter of 6.00 inch. The rupture discs were made of aluminum of type 24ST3, with a thickness of 0.160 inch, and a diameter of 7.50 inch. Each disc was provided on one side with two mutually perpendicular grooves, 6.00 inches long and intersecting at the center of the disc. The grooves were cut with a 60° V-cutter to a depth of 0.025 inch for 19 discs, and to a depth of 0.012 inches for 6 discs. The discs with the deep grooves were to rupture at a depth of 2500 feet which corresponds to the depth of the SOFAR Channel around the Hawaiian Islands. The purpose of the discs with the shallow grooves was to discover if energy released far below the SOFAR Channel could be detected in the channel. The tubes were divided in five groups of five tubes each. The lengths of the tubes in each of the groups were respectively: 1 foot, 3 feet, 5 feet, 6 feet and 9 feet.

The rupture discs were attached to the tubes by means of inert-gas arc-welding following the method used successfully in the preliminary experiments (see Figure II-4). The devices were completed by closing their other ends with aluminum discs, 0.250 inch thick, and a diameter of 7.50 inch. These bottom discs were also welded on.

For the actual field tests the author joined the r/v Townsend Cromwell of the Bureau of Commercial Fisheries during an oceanographic cruise in July of 1964. The Cromwell left Honolulu on July 12, and returned on August 1. The cruise plan is shown in Figure VI-1. A total of ten drops of two or three floats each were made. The first two drops were made on the first west to east leg of the cruise, at 11°30'N. Each drop consisted of a 9 foot float and a six foot float. Due to the absence of intervening islands, each was meant for detection at the listening stations on Wake and Midway. None of these signals were received. All other drops were meant for detection at the listening station at Kaneohe on the north side of Oahu (see Figure VI-1).

A complete log of drops 3 through 10 is shown in table VI-1. The date and time are local time. The latitude and longitude indicate the ship's position at the time of the drop. Since the exact locations of the SOFAR hydrophones are classified, the distances shown are approximate and are as measured from the ship to the 100 fathom line northeast of Oahu. Approximately an hour before each drop the station at Kaneohe was informed by radio of the intended drop time. This allowed the people at the station to prepare their recording equipment. All data were to be recorded on magnetic tape.

The two or three devices to be used were ballasted so that all would take about 15 minutes to sink to 2500 feet (see Figure VI-2). They were released at two minute intervals. Immediately upon dropping the floats, a Mark 22 Rescue Signal was released. This signal consists of a 4 lbs charge of TNT which sinks in one minute to 2500 feet where it is exploded by a detonator activated by hydrostatic pressure. The signals are clearly audible at distances up to 3000 miles. In our case they served both as data references marks and means to calibrate the implosive energy of the floats. The explosion of the Mark 22 was monitored aboard the Townsend Cromwell and its exact time and the ship's position were relayed to Kaneohe by radio.

The length of the floats used and their rupture discs are also indicated in table VI-1. The last column in the table shows whether or not the signal from a float was received at Kaneohe.

The greatest distance at which a float was heard is approximately 1100 km. Only the signal from the 6 foot float was received at this distance, since the 9 foot float imploded far below the SOFAR Channel. The signal recorded on magnetic tape is shown graphically in Figure VI-3. The Kaneohe listening station has four hydrophones and the different times of arrival at the phones are due to the different locations of the phones. The graphic display shows amplitude of the signals in decibels versus time. The distance between any two consecutive heavy vertical lines represents 1/2 second. The distance between any two consecutive heavy horizontal lines on the grid represents 5 decibels.

As can be seen from Figure VI-3, the signal amplitude is substantially greater than that of the ever present sea noise, and it is easy to perceive the signal either graphically or aurally. Figure VI-4 shows the signals and relative arrival times for the Mark 22 Rescue Signal released during the fourth drop.

A comparison of Figures VI-3 and VI-4 allows one to estimate the energy obtained from an imploded float. Assume that the amplitude of the recorded signal is proportional to the energy released at a fixed distance. Let A denote the relative amount of energy contained in the Mark 22 Rescue Signal, and B the relative amount of energy contained in the 6 foot float. Let D denote the difference in decibels between the average peak amplitude of the Mark 22 Rescue Signal and the average peak amplitude of the 6 foot float. The reader can verify from Figures VI-3 and VI-4 that D = 20 decibels. By definition

$$D = 20 \log_{10} \frac{A}{B}$$

or

$$\frac{A}{B} = 10$$

Thus, the 6 foot float is approximately equivalent to 0.4 lbs of TNT. Since the volume of the float was approximately 1.2 cubic feet, it follows that at a depth of 2500 feet (1100 psi) a float with a volume of 3 cubic feet has the potential energy equivalent to 1 lb of TNT.

The energy released by a float is directly proportional to its volume (Cole, 1948). If one assumes that the energy released by a float is also proportional to pressure, which is true for spherical volumes (Cole, 1948), one may extrapolate the above figures to find that at 1800 psi a volume of 1.8 cubic feet is potentially equivalent to 1 lb of TNT. This compares favorably with the result stated in

Chapter II where it was tentatively estimated that at 1880 psi 4 cubic feet would be potentially equivalent to 1 lb of TNT (see page 7).

The nature of the signals received varied little from one drop to the next. To illustrate this Figure VI-5 shows the signal received from the 5 foot device released during the seventh drop at a distance of 550 km from Kaneohe. Of greater interest for future work with rupture discs and deep-ocean neutrally-buoyant floats is the frequency-spectrum of the signal obtained from such floats.

Figures VI-6 through VI-8 show the frequency analysis at a single hydrophone for the implosion signals received from drops 4, 7 and 9 at distances of 1100, 550 and 250 km, respectively. In all figures the lower graph shows on a continuous basis the frequencies present as a function of time in increments of 24 cps between 4 and 1200 cps. The total time shown in 8 seconds and is approximately centered around the time of arrival of the implosion signal. The usefulness of the lower graph in a frequency analysis is limited due to the overloading of the pre-amplifiers upon arrival of the signal. This is illustrated by the high frequencies apparently present at the arrival times although it is doubtful that the SOFAR system was designed to handle such high frequencies.*

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* Unfortunately the specifications of the SOFAR system are classified, so that this discussion can only be qualitative.

The upper graphs in the three figures show the relative amplitude of the frequencies present at particular instances of time. The time base of each of the upper graphs coincides precisely with that of the lower graph. In all three figures a frequency analysis of the incoming signal is shown a few seconds before the arrival of the implosion signal, immediately after the arrival, and a few seconds after the arrival. The first and third analysis show the amplitude of the frequencies present in the background noise for that particular period of time.

It can be seen from Figures VI-6 and VI-7 that the frequencies of the energy released by the implosion of a 6 foot float and a 5 foot float lie between 250 cps and 600 cps with a peak at 350 cps. This is well above the frequencies found in sea noise, which usually lie below 200 cps with a peak near 80 cps. It would be desirable, however, if the spread of the frequency were smaller in order to increase the relative amplitude of the peak. Also, if the peak were at 250 cps rather than 350 cps, the effects of long-distance attenuation would be decreased. The main purpose of Figure VI-8 is to illustrate that the frequencies of the energy released by a small float (1 foot) are considerably higher (300 to 900 cps) and are not attenuated appreciably over a short distance. It is interesting to note that the only time a signal was received from a float imploded far below the SOFAR Channel was for a drop quite close to the station (see table VI-1, Float No. 21).

The field experiment off the Hawaiian Islands has shown that the use of rupture discs for underwater signalling over significant distances is certainly feasible. The results obtained as to the amount of energy released and the frequency spectrum of the imploding floats indicate that further work in this direction is both necessary and worthwhile. These future aspects are discussed in the concluding chapter of this paper.

Table VI-1. Log of the field tests in the Pacific Ocean off the Hawaiian Islands

Drop No.	Date (1964)	Local Time	Latitude (north)	Longitude (west)	Distance to Kaneohe (kilometer)	Mark 22 Exp. time (local)	Float No.	Length (feet)	Groove Depth (inch)	Received
3	7-20	12:30	23°32'	153°55'	450	12:32.36	5	9	.025	yes
							6	6	.025	yes
4	7-29	08:00	26°22'	148°18'	1100	07:55.74	7	6	.025	yes
							8	9	.025	no
5	7-29	19:00	25°15'	150°30'	850	18:57.15	9	6	.025	yes
							10	9	.012	no
6	7-30	08:00	25°00'	151°57'	700	07:58.97	111	1	.025	no
							12	5	.025	yes
							13	3	.025	no
7	7-30	19:00	25°00'	154°00'	550	19:03.12	14	1	.025	no
							15	5	.012	no
							16	3	.025	yes
8	7-31	08:00	25°00'	156°28'	450	08:03.61	17	1	.025	no
							18	3	.025	yes
							19	5	.025	yes
9	7-31	19:00	23°30'	157°01'	250	19:01.08	20	1	.025	yes
							21	5	.025	yes
							22	3	.025	yes
10	8-1	06:30	21°50'	157°07'	50	06:32.78	23	1	.025	yes
							24	5	.025	yes
							25	3	.025	yes

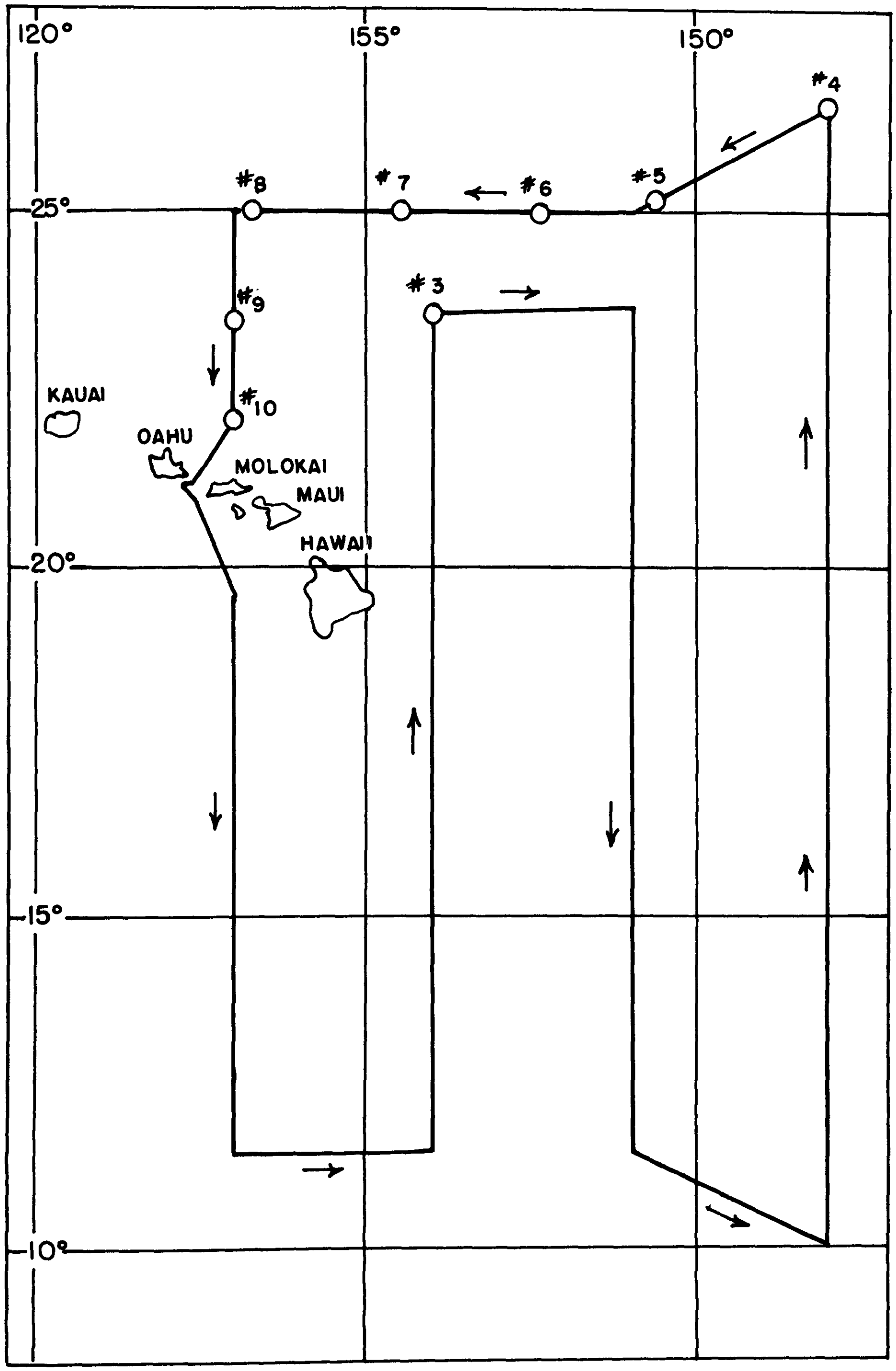

Figure VI-1. Cruise of the R/V Townsend Cromwell.
July 12 - August 1, 1964.

Figure VI-2. Left: The floats aboard the R/V Townsend Cromwell.
Right: Setting a float overboard.

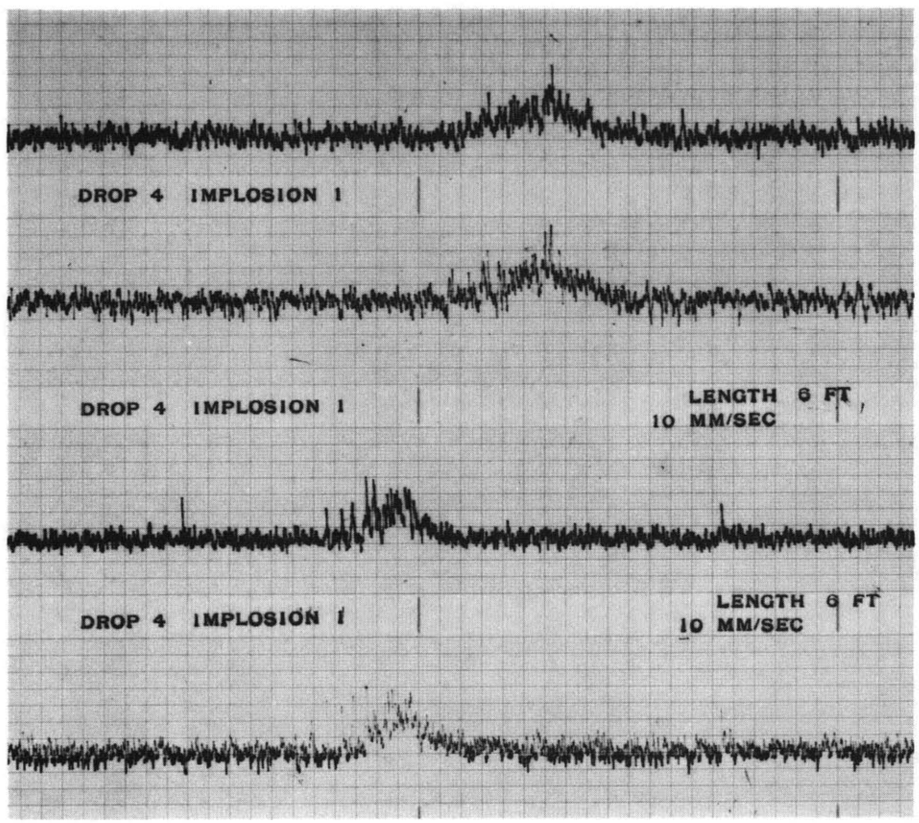

Figure VI-3. Graphic representation of the SOFAR signals.
Phone No. 1 shown at top.
Drop 4, Implosion 1.

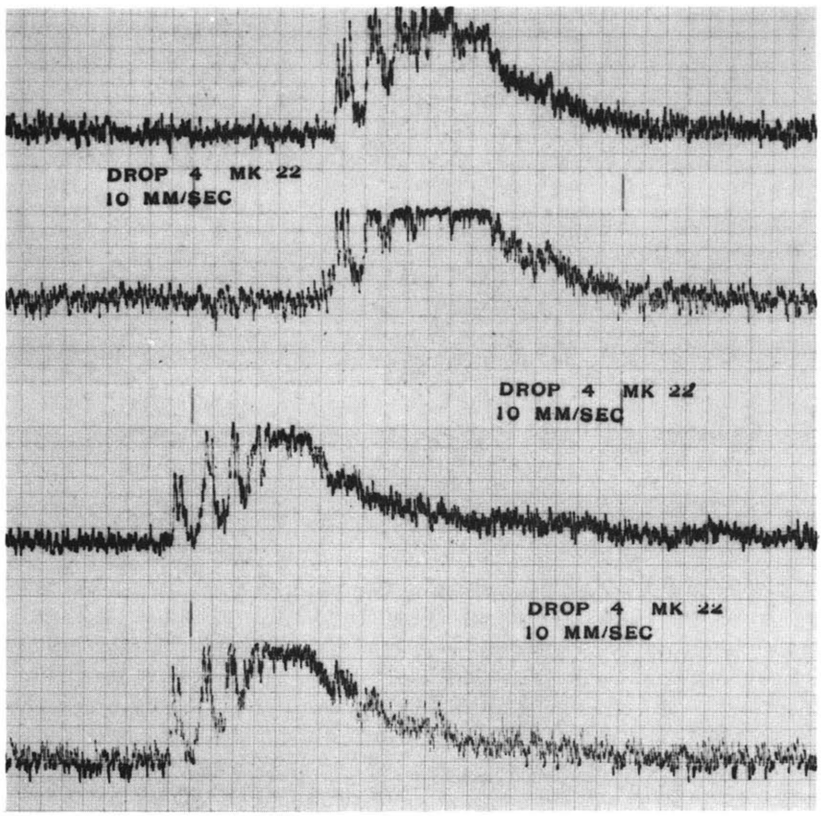

Figure VI-4. Graphic representation of the SOFAR signals.
Phone No. 1 shown at top.
Drop 4, Mark 22 Rescue Signal.

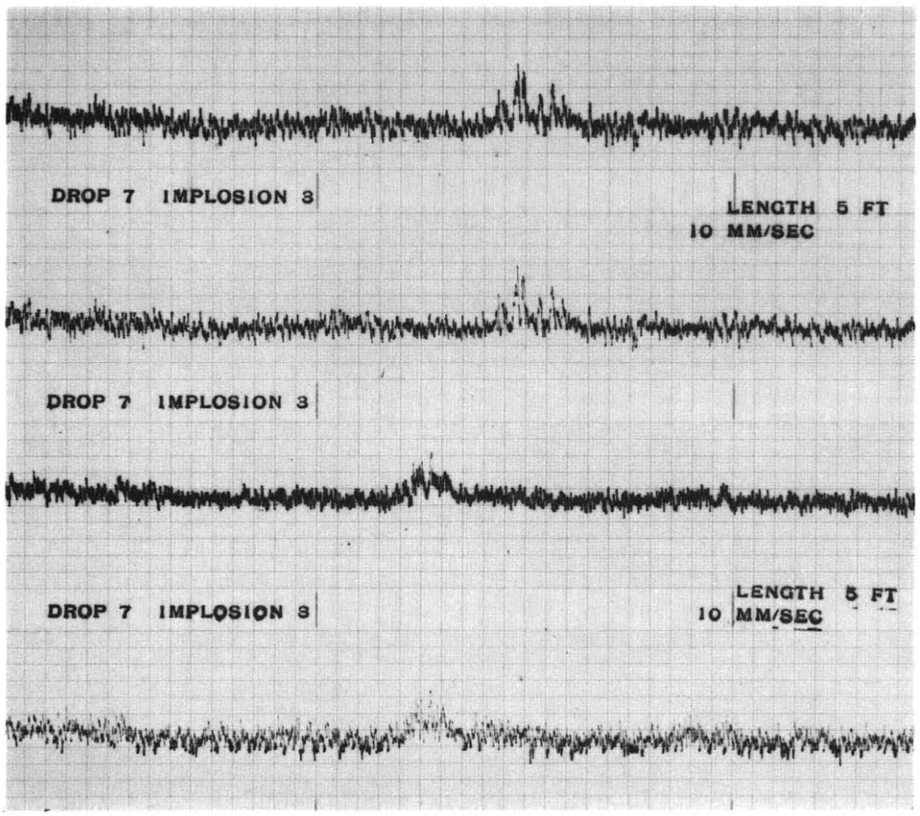

Figure VI-5. Graphic representation of the SOFAR signals.
Phone No. 1 shown at top.
Drop 7, Implosion 3.

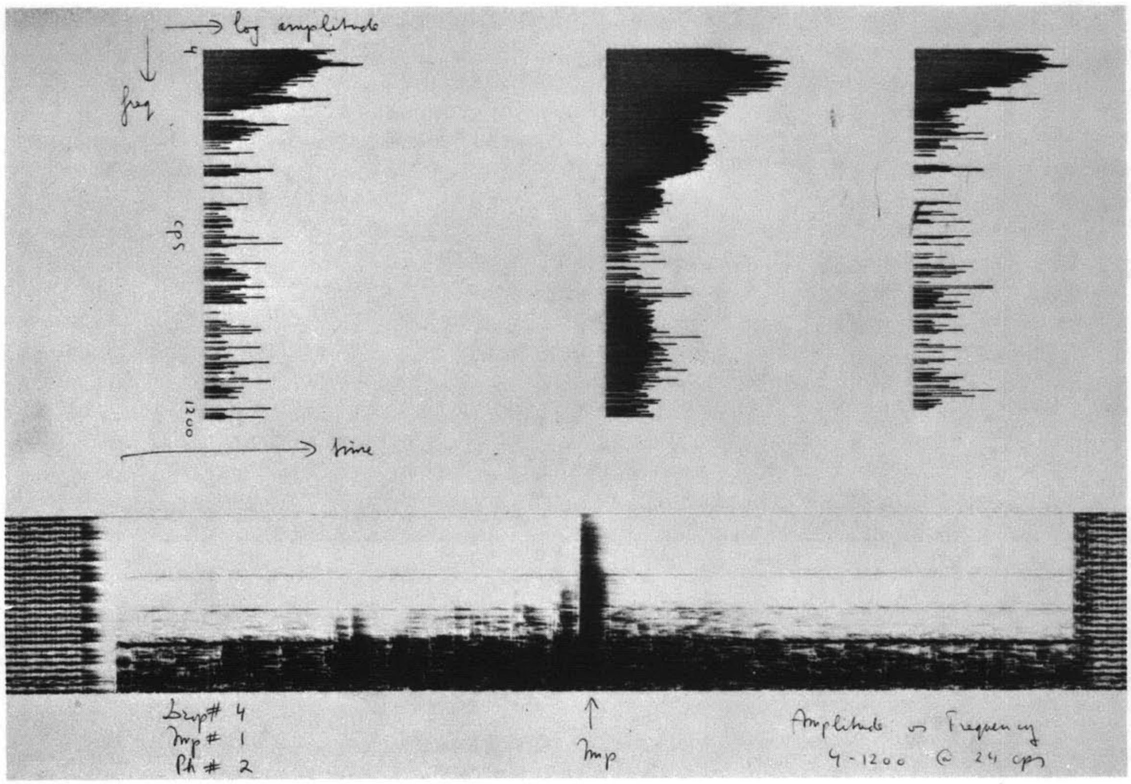

Figure VI-6. Frequency analysis of the signal received on phone No. 2.
Drop 4, Implosion 1.

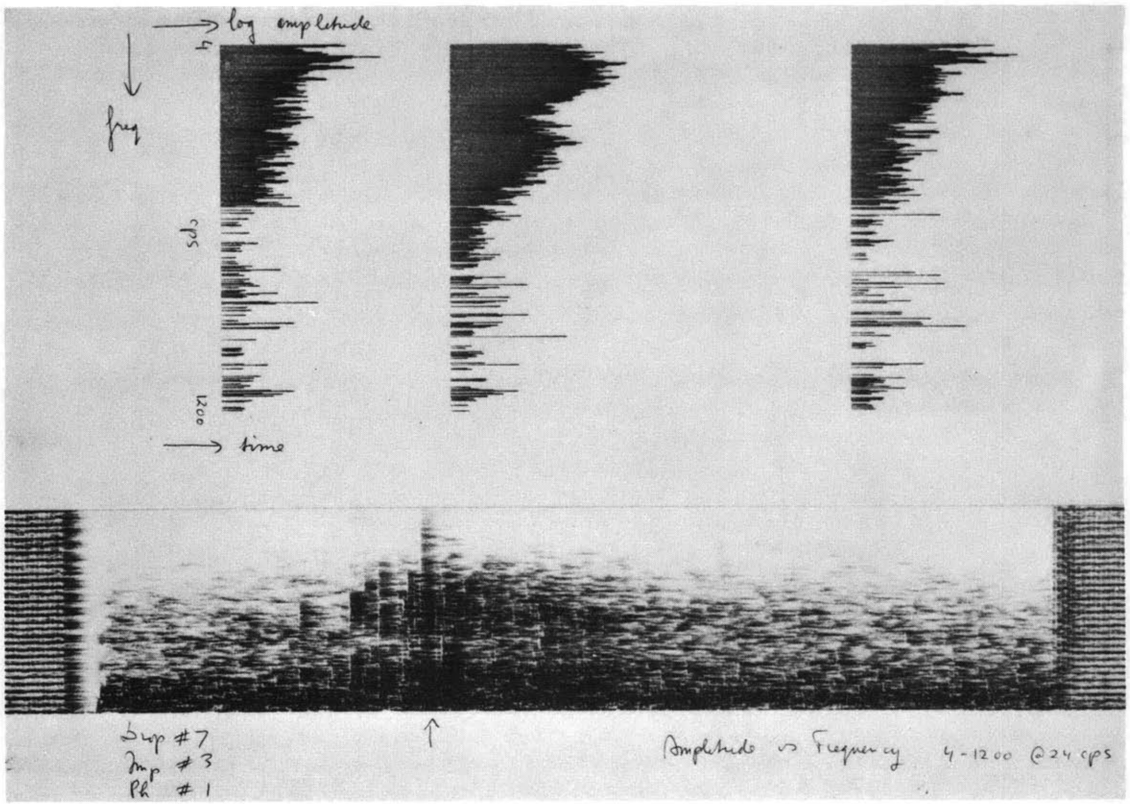

Figure VI-7. Frequency analysis of the signal received on phone No. 1. Drop 7, Implosion 3.

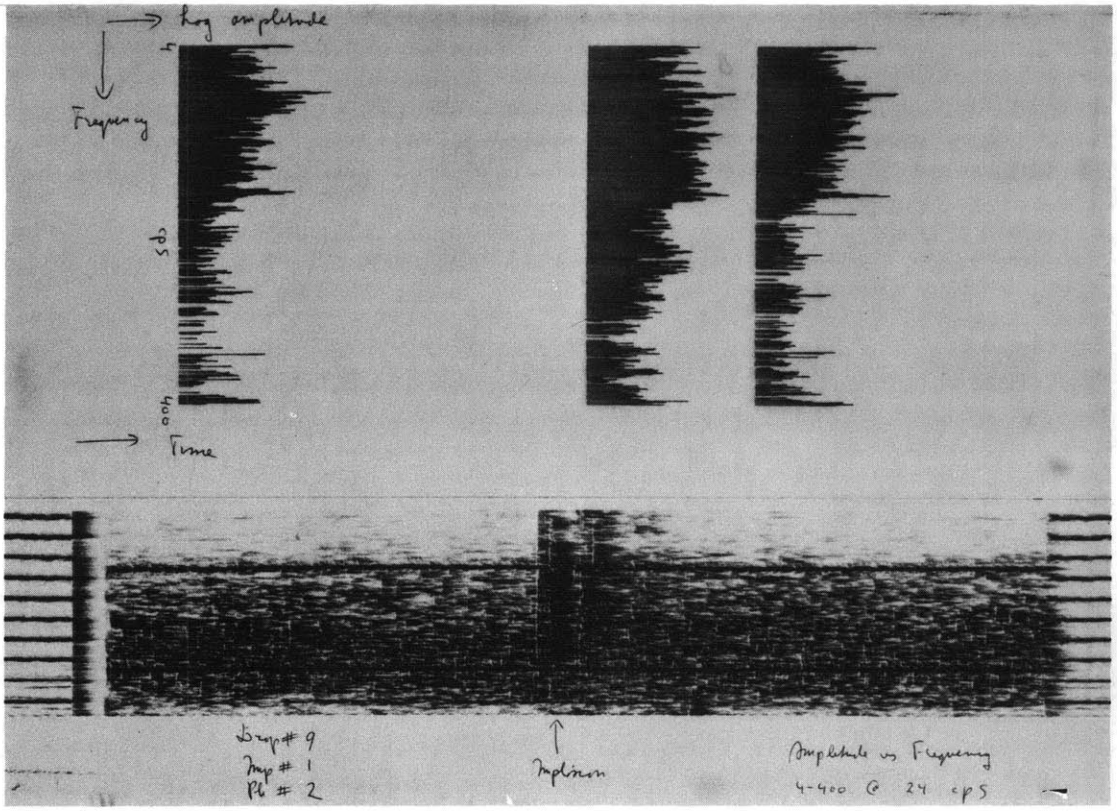

Figure VI-8. Frequency analysis of the signal received on phone No. 2.
Drop 9, Implosion 1.

VII. CONCLUSION

The experiments performed with rupture discs in the laboratory and in the Pacific Ocean have shown that hollow cylindrical floats equipped with rupture discs are useful for long-distance underwater signalling. The work has yielded technical knowledge concerning the fabrication of rupture discs and their attachment to floats. The experiments have resulted in numerical information useful in further work with rupture discs and long-distance signalling in the SOFAR Channel.

It is clear that for future work, rupture discs have to be developed with a uniform thickness varying less than 0.001 inch over the disc, and with a groove depth accurate well within 0.001 inch. Since such close tolerances are difficult to achieve with rolled metal materials it may be desirable to examine the possibility of using discs cast in a mold. The grooves might be cast as an integral part of the mold, or they might be cut on a milling machine. If casting or molding proves feasible, it would be of interest to consider using one or more of the modern plastics such as LEXAN (a poly-carbonate) or CELCON (an acetal polymer). Both these materials are noted for their molding properties and their exceptional flexular strength. In large quantities, both would be cheaper to produce than any type of metal disc.

The possibility that discs may be molded out of plastics quite naturally leads one to consider using plastic floats. Again, the use of the previously mentioned materials for the construction of floats is perfectly feasible. However, tubes of the large dimensions needed for oceanographic purposes would have such a great wall thickness when made of plastic that their cost might become prohibitive. One immediate advantage of using plastic floats and plastic rupture discs is the ease with which the two may be joined. Any number of commercially available resin epoxies will yield a bond between the two parts far stronger than the original material.

Although welding is a satisfactory means of attaching aluminum discs to aluminum floats, it is too expensive for large-scale fabrication of neutrally-buoyant floats equipped with rupture discs. The cost of other sound means than welding will also be prohibitive in any project where the cost per float must be minimized due to the large number of floats to be used. It is possible, however, that aluminum discs could be attached to floats by means of epoxies. Although this bond would not be as strong as a welded bond, the disc would be sealed firmly against the tube by hydrostatic pressure at the pressures required to deform the disc to such an extent as to break the epoxy bond.

The Hawaiian experiment was successful in that it gave a definite measurement of the amount of potential energy carried by a float equipped with a rupture disc. It also became clear, however, that further research and development were necessary not only to determine the size of floats necessary for ocean-wide signalling but also to determine the type of float and disc necessary to yield an implosion with a desirable frequency spectrum.

If it is true that the amount of energy released at a given depth is proportional to the volume of the float, it seems a simple matter to obtain higher energy levels by increasing the length or the diameter of the floats. Increasing the length of a float much beyond 10 or 12 feet is not practical, both because of handling and storage problems and because the increase in volume is only proportional to the increase in length. Since the volume increases as the square of the diameter, it is more desirable to increase the diameter of the float. Doing this is also subject to a practical limit, however, for as the diameter increases the wall thickness must increase in direct proportion to maintain the low compressibility of the float necessary for neutral buoyancy. For example, one can increase the potential energy of a float at a given pressure nine-fold by tripling its diameter. To maintain the same low compressibility one must triple the wall-thickness. Since the circumference has also tripled one is faced with at least a nine-fold increase in the weight of the float. The same increase in volume can be achieved be increasing the length of the original float, and thus its weight, nine-fold. One then has the interesting result that the potential energy of a float is directly proportional to its weight. This again points to the need for finding cheaper materials for the construction of the floats.

Neutrally-buoyant floats with long-distance signalling capabilities will be useful in the study of currents at depth in the ocean. For this purpose two types of float have to be designed. One would drift above the SOFAR Channel and eventually sink in to the channel to implode, and the second would drift below the SOFAR Channel and eventually rise into the channel to implode. Whereas the design of the first type is straight forward, that of the second poses more problems. Since at neutral buoyancy the float would be exposed to pressures great enough to rupture its disc, the rupture disc would have to be protected until the float arrived in the SOFAR Channel from below. One design possibility would be to equip a float with extra ballast. After drifting at some depth below the SOFAR Channel for a predetermined period of time, this extra ballast would be released and the float would ascend. Since its rate of ascent, its floating depth, and its implosion depth are known, the protective covering of the disc may be destroyed after the proper length of time when the float should be in the SOFAR Channel. The simplest way to destroy the disc's protection is by means of a small explosive charge.

In order to make neutrally-buoyant floats equipped with long-range signalling devices truly useful to oceanographers, it is clear that some sort of world-wide organization has to be established to organize the release of floats and the reception of their signals.

Whereas the cost of the floats can be kept low, the cost and requirements of maintaining listening installations covering all oceans are such that international cooperation is an absolute necessity. If the many SOFAR listening stations already established in many parts of the world could be made available for the detection of signals from floats, the main function of such an organization would be to coordinate the release of the floats and the distribution of the data collected at the various stations. It is the author's hope that international cooperation will improve to such an extent that the project outlined above will become feasible.

Appendix A. SOFAR

SOFAR stands for SOund Fixing And Ranging and usually refers to the system for determining the location of underwater sound signals generated as far as 3000 miles from shore. In this connection one often refers to the SOFAR Channel by which is meant the deep layer in the ocean, approximately parallel to the surface, in which the velocity of sound has its minimum.

The major influences on sound velocity in the ocean are provided by temperature and pressure. In most cases the effect of the variation in salinity is sufficiently small to be ignored. It is assumed that the static pressure increases linearly with depth, disregarding the small deviations arising from density variations due to varying temperature and salinity.

If one ignores the effects of diurnal heating and cooling in the top 100 or 200 meters of the ocean, it is known that the temperature of the water decreases rapidly and monitonically to a depth of approximately 3000 feet. This layer is known as the main thermocline. In this layer the effects of the negative temperature gradient far outweighs the effects of the linearly increasing pressure and the velocity of sound will decrease monitonically with depth.

Below the bottom of the thermocline the temperature remains approximately constant and increasing pressure becomes the dominant factor in determining the sound velocity. Thus we find that below the main thermocline the velocity of sound will increase monitonically with depth. Thus the bottom of the main thermocline defines the depth at which the velocity of sound has its minimum. This depth is called the axis of the SOFAR Channel.

Since the bottom of the thermocline changes its depth vary gradually from one location in the ocean to the next, ranging in depth from 1000 to 6000 feet, it follows that the SOFAR Channel is usually parallel to the surface. The average depth of the axis of the SOFAR Channel is 3000 feet.

Consider the sound signal originated on the axis of the channel. Due to the increasing velocity of sound both above and below the axis, most energy originally traveling away from the axis will be refracted back towards the axis. Thus the spreading of the energy is two-dimensional rather than three-dimensional. Energy is greatly conserved and signals generated on the axis may be received by hydrophones placed on the axis at distances as great as 8000 miles.

Appendix B. ENGINEERING CALCULATIONS

Both the tube and the bottom plate of the instrument are machined from steel. For both Young's modulus, E, is 30×10^6 psi and Poisson's ratio, u, is 0.3. The 15 tie rods are made from strain tempered steel with a minimum yield strength of 100,000 psi.

The maximum internal working pressure of the tube may be found from Barlow's formula:

$$P = \frac{2St}{D},$$

where

P = bursting pressure (psi)

S = ultimate tensile strength of tube (psi)

t = wall thickness of tube (inches)

D = outer diameter of tube (inches)

For the tubing used S = 80,000 psi, and with a wall thickness of 7/8 inch and an outer diameter of 7-3/4 inch it follows that the bursting pressure is approximately 18,000 psi. For operating conditions characterized by a steady, gradually increasing pressure a safety factor of 3 is sufficient. Thus the tube is absolutely safe up to 6000 psi.

The minimum yield strength of the tie rods is 100,000 psi. The weakest part of each rod is the threaded part which has an area of 0.28 square inch. Thus the maximum force to which any rod may be subjected is 28,000 pounds. Since 15 rods are used the total force exerted upon the disc by the hydraulic fluid in the tube cannot exceed 420,000 pounds. Considering the area of the disc, it follows that the 15 rods will hold the collar for all pressures less than 15,000 psi. Again assuming a safety factor of 3 it follows that the instrument may be used in absolute safety with internal working pressures of up to 5000 psi.

To calculate the angle of the bevel required to maintain a tight closure between the tube and the collar, assume a working pressure of 3000 psi. This figure is chosen out of convenience, for all expansions are linear and any figure below 5000 psi would serve equally well.

The change in the outer diameter of a tube, not constrained at either end, subjected to an internal pressure far greater than the external pressure, is given by

$$d = \frac{ap}{E}\left[\frac{1 + (a/b)^2}{1 - (a/b)^2} + u\right]$$

where

d = change in diameter (inches)

a = inner diameter (inches)

b = outer diameter (inches)

p = internal pressure (psi)

E = Young's modulus (psi)

u = Poisson's ratio

For the tube used in the instrument it is calculated that $d = 0.0026$ inch. Thus the change in the outer radius (r) is $r = 0.0013$ inch.

At 3000 psi the collar is subjected to a force of 85,000 pounds. Thus the collar exerts 5,700 pounds upon each of the 15 tie rods. Each rod consists of two parts; a threaded part with a corss-sectional area of 0.28 square inches, and an unthreaded part with a cross-sectional area of 0.44 square inches. With a force of 5,700 pounds on the rod the threaded part is subjected to a stress of 20,000 psi and the unthreaded part is subject to a stress of 13,000 psi. Young's modulus (E) is by definition

$$E = \frac{\text{stress}}{\text{strain}} ,$$

where the strain is a measure in inches per inch of the change in the length of a rod subjected to a stress. Thus for the threaded part of the rod the strain is 0.67×10^{-3} inches per inch, while for the unthreaded part the strain is 0.43×10^{-3} inches per inch. Since the length of the unthreaded part is 2.875 inch, and since one may estimate that the length of the threaded part of the rod subject to lengthwise strain is approximately 3/4 inch, it follows that the change in the length of the rods at 2000 psi is approximately 0.0018 inch.

Thus, when the tube expands radially by 0.0013 (r) inch it must travel upwards at least 0.0018 (ℓ) inch. It follows that the tangent of the angle of the bevel, the angle being measured from the vertical, must be no greater than $r/\ell = 0.722$. Thus the angle must be no greater than $35°45'$.

Following the above computation, the angle of the bevel in the instrument is 30°. It was felt that this angle would be steep enough to provide a continuous closure between the tube and the collar without being so steep as to cause the tube to bind against the bottom plate.

Appendix C. VOLUME UNDER THE DISCS

It was pointed out in Chapter V that all discs behave like membranes under uniform tension near the pressure at which they rupture. Thus one may assume that the profiles through the centers of the discs as shown in Figures V-4 through V-11 are arcs of circles. For each disc the radius of the circle is equal to the radius of the sphere.

A typical profile through the center of a spherically deformed disc is shown in Figure C-1. The arc ABE represents the disc. The point B is at the center of the disc and bisects the arc. Let C be the imaginary center of the sphere. Then CB is perpendicular to AE and bisects AE. CB is the radius R of the sphere. Let the point D divide the line AB into two equal segments. Then the two triangles ADC and BDC are congruent and it follows that

$$R = \frac{DB}{\cos\beta} .$$

Let the deflection OB at the center be denoted by d. Then

$$\beta = \cot^{-1} \frac{d}{AO}$$

Since AE = 6 inches, it follows that AO = 3. It is also clear from Figure C-1 that

$$DB = \frac{1}{2} \sqrt{9 + d^2} .$$

It follows that the radius R is a function of the center deflection d only and is given by

$$R = \frac{\sqrt{9 + d^2}}{2 \cos (\cot^{-1} \frac{d}{3})} \text{ inch.}$$

With the x-axis and the z-axis as shown in Figure C-1, and with the y-axis perpendicular to the figure at the point 0, the equation of the sphere in cylindrical coordinates is

$$r^2 + [z - (d-R)]^2 = R^2 .$$

The volume under the disc is the volume of the part of the sphere located above the plane $z = 0$. This volume may be found by integration:

$$V = \int_0^d A(z)\, dz$$

$$V = \pi \int_0^d r^2\, dz$$

$$V = \pi \int_0^d R^2 - [z - (d - R)]^2\, dz$$

$$V = \pi\, (Rd^2 - \frac{d^3}{3})$$

The results of the calculations yielding radii and volume changes for directly ruptured discs and preformed discs are tabulated in Table C-1. The volume changes are plotted as a function of rupture pressure or depth in Figure V-12.

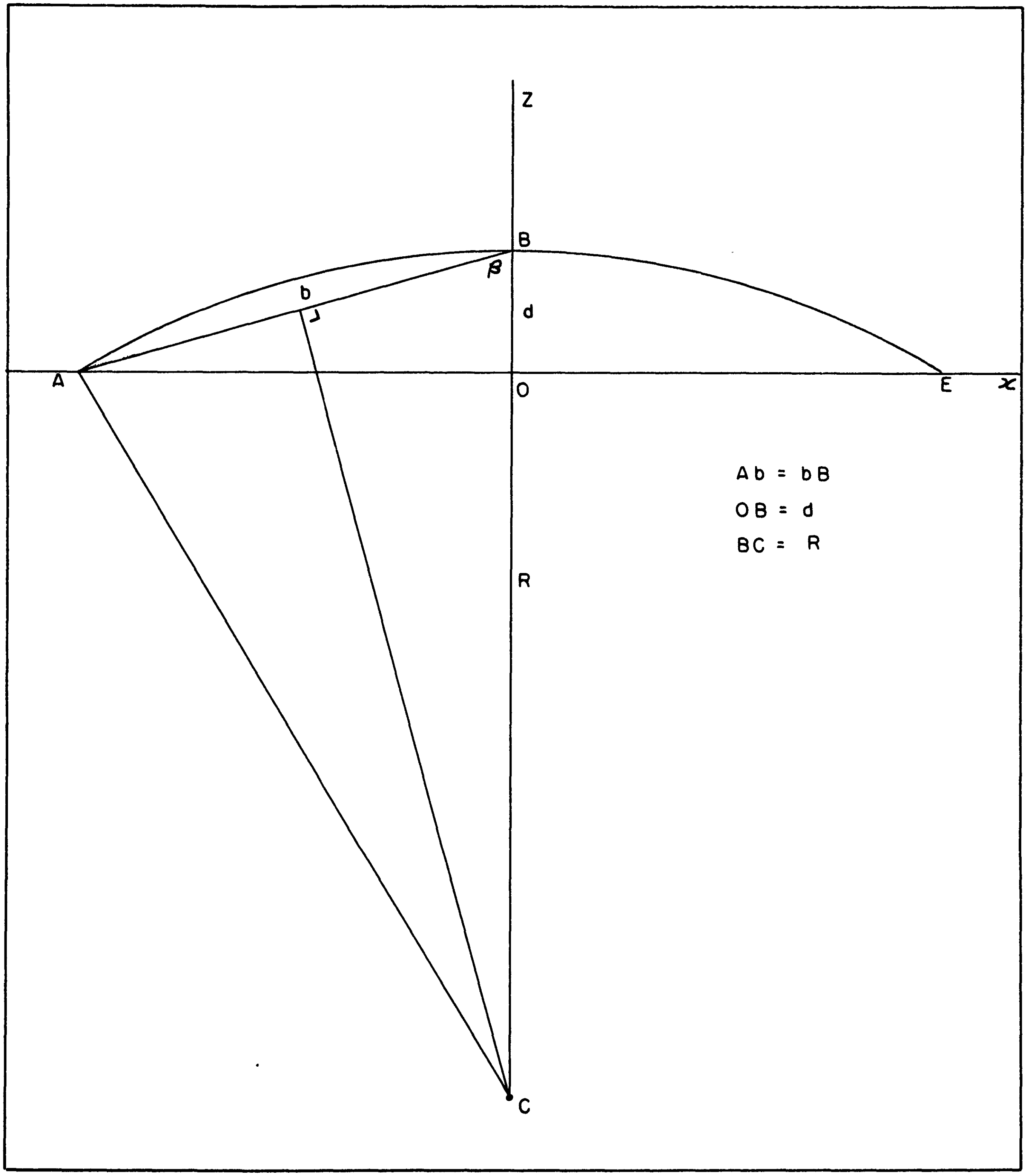

Figure C-1. Geometry required to calculate the change in volume of a deforming disc. See Appendix C.

Table C-1. Showing radii of discs, and volumes.
See Appendix C and Figure C-1.

Directly Ruptured Discs					Pre-formed Discs				
Groove Depth (inch)	Max. Center Defl. (d) (inch)	Radius of Sphere (inch)	Volume Under Disc (cu.in.)	Volume/d (sq.in.)	Center Deflection Initial (inch)	Center Deflection Final (inch)	Change in d (inch)	Change in Volume (cu.in.)	Rupture Pressure (psi)
.050	.236	19.17	3.36	14.24					
.040	.252	17.99	3.60	14.29					
.035	.350	13.04	5.00	14.29					
.030	.364	12.59	5.17	14.20	.195	.384	.179	2.56	875
.028	.480	9.61	6.83	14.23					
.026	.468	9.86	6.67	14.25					
.022	.495	9.33	7.05	14.24	.346	.546	.200	2.86	1450
.020	.524	8.90	7.54	14.39					
.018	.705	6.73	10.14	14.38	.514	.727	.213	3.04	1875
.016	.727	6.57	10.51	14.46					
.014	.803	5.79	11.20	13.95	.650	.828	.178	2.54	2438
.012	.930	5.31	13.58	14.60	.804	.988	.184	2.62	2775

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