Evaluation of the strength of a reactor pressure vessel
by Everett E Shaw

A THESIS Submitted to the Graduate Faculty in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering
Montana State University

Abstract:
This report is an appraisal of the strength of a reactor vessel. It is based on 1) a consideration of the mechanical properties of the vessel components, and 2) a series of three tests made to record outside surface principal strains vs internal pressure.

The data show the vessel to have been considerably strengthened by st-rain hardening in testing. It is found that pressures up to 7000 psi will cause no further yielding, and that rupture will not occur below 9000 psi.

EVALUATION OF THE STRENGTH OF A REACTOR PRESSURE VESSEL

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Bozeman, Montana
April, 1955

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ABSTRACT

This report is an appraisal of the strength of a reactor vessel. It is based on 1) a consideration of the mechanical properties of the vessel components, and 2) a series of three tests made to record outside surface principal strains vs internal pressure.

The data show the vessel to have been considerably strengthened by strain hardening in testing. It is found that pressures up to 7000 psi will cause no further yielding, and that rupture will not occur below 9000 psi.

INTRODUCTION

The investigation of the strength of this reactor was undertaken to substantiate design calculations. High working stresses had been used in the determination of some of its dimensions, and the strength of the material was not accurately known. It was desirable, therefore, to verify by actual test its ability to withstand operating pressures.

The vessel underwent a general permanent distention in each of the three pressure tests, but this was slight, and no damage was incurred. On the contrary, subjection to test pressures was beneficial in that strain hardening increased the pressure at which yielding first occurred in the vessel from 4000 psi in the first test to 7000 psi in the final test.

The work is presented here in two sections. In the first are described the vessel assembly and its components, giving their strength properties when known, and suggesting approximations based on fabrication history when unknown. The second section describes the pressure test in detail: the plan, equipment used and its installation, test procedure, and results.

Section 1

THE VESSEL

General

Principal components of the reactor as assembled for the pressure test are identified in Fig. 1. The strength of all these elements except the head and end cap was investigated. Construction detail of the head made points of interest inaccessible for strain gage placement, and behavior of the end cap was of no interest, since it was not part of the reactor, but was for test use only.

Fabrication

The retort, pump casing, and head were forged from a 40 inch hexagonal big end up ingot weighing 36,600 pounds. The material was electric furnace stainless steel, AISI type 347, which analyzed as follows:

Carbon	0.070 per cent
Manganese	1.560
Silicon	0.580
Phosphorous	0.020
Sulphur	0.024
Chromium	18.000
Nickel	11.000

The ingot was preheated at 1600° F. Forging was done at 2150° F, the piece being reheated at intervals as required. Fig. 2 shows how the ingot was worked, indicating approximate reduction of area at the different sec-

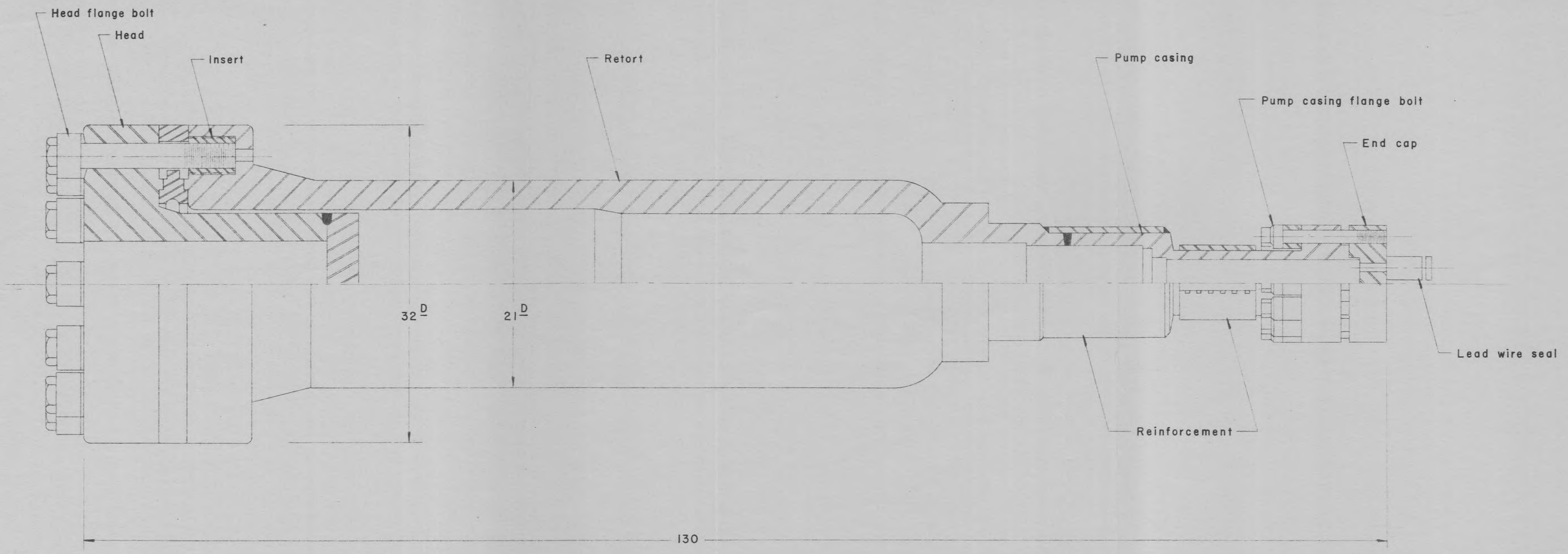

Fig. 1 — Reactor assembly.

tions. After completion of the forging operations the piece air cooled from 1800° F.

The completed forging was sawed, the parts rough machined, annealed, and finish machined. Welds made subsequently were locally heat treated.

When the retort was machined, three pieces were core drilled from the flange at 90 degree intervals. Standard ASTM 0.505 inch diameter tensile specimens were made of this material and tested. Their mechanical properties averaged as follows:

Proportional limit	20,400 psi
Yield strength (0.2 per cent off-set)	30,400 psi
Ultimate tensile strength	71,500 psi
Reduction in area	30 per cent
Elongation in 2 inches	32 per cent
Hardness	80 Rb
Charpy impact	57 ft-lbs

Since the material tested was taken where reduction by forging was least, these values may be regarded as conservative for any other section to a degree approximated according to the per cent reduction experienced by that section. In this way a general understanding of the properties throughout the vessel may be gained.

The finished retort was ultrasonically inspected by the fabricator, and radiographically inspected by the purchaser. No defects were found.

Subsequent reconsideration of stresses in the pump casing indicated the advisability of reinforcement. This was done by welding a steel sleeve around the upper section, and adding a bolted clamp at the lower section as

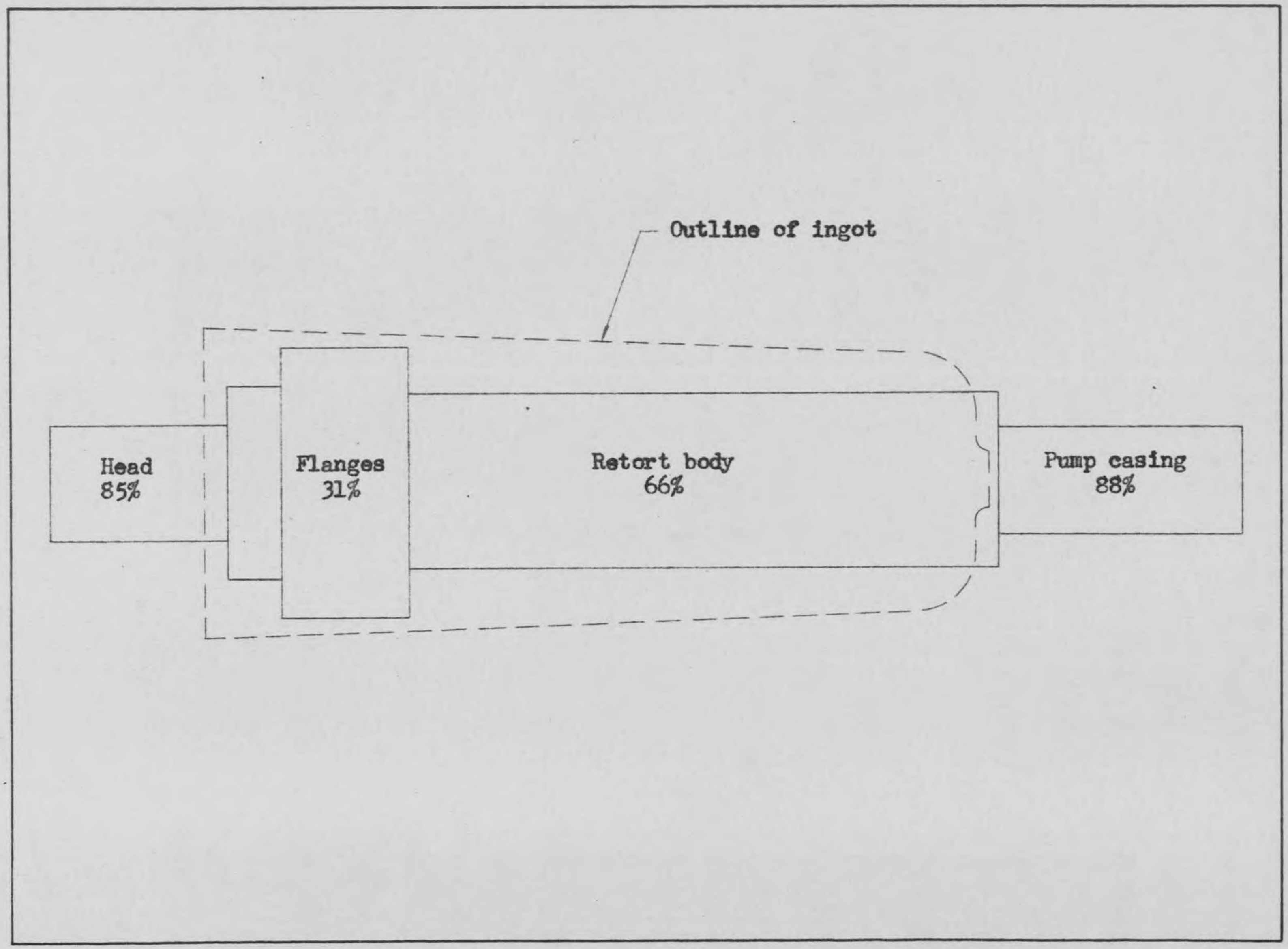

Fig. 2 - Per cent reduction of area by forging.

shown in Fig. 1.

The head flange bolts were Allegheny Ludlum Potomac "M" die steel, principal alloying elements of which are

Carbon	0.42 per cent
Chromium	5.29
Molybdenum	1.01
Vanadium	0.95

These bolts were hardened to Rc 45, corresponding to an ultimate tensile strength of 190,000 psi.

In order to increase the area of thread engagement in the weaker material of the retort flange, inserts were used as shown in Fig. 1. These were made of AISI type 414 stainless steel, hardened to Rc 45, having an ultimate tensile strength of 150,000 psi.

Pump casing flange bolts were AISI 4140, hardness Rc 30, ultimate tensile strength 120,000 psi.

Section 2

THE PRESSURE TEST

Concept

On the outside surface, circumferential and longitudinal strains were measured at each of thirteen sections; inside the vessel, gages were installed at three sections. Fig. 3 shows the general plan of gage placement. For any gage, departure of the strain vs pressure curve from straight line was considered evidence of anelastic action within the wall at or near that section.

Installation of Gages

Typical outside surface installations are shown in Fig. 4. Gages used here were SR-4 type A-5-1, and the procedure for their application was essentially as recommended by the manufacturer. Metal surfaces were cleaned with toluene and acetone, and the gages applied using Duco household cement. No clamping was necessary.

The gages installed inside the vessel were not expected to make accurate measurements, but it was hoped that some information would be given by their curves. In any case, their behavior in water under pressure was of interest.

Bakelite gages, SR-4 type AB-7, were used inside, because it was thought these would be least affected by immersion in water. The proper installation of this type of gage requires clamping under 150 psi pressure while the adhesive cures. It was possible, therefore, to take strain measurements only near enough the ends of the vessel that clamps could be applied.

Fig. 3 – Plan of strain gage placement.

Fig. 4 - Typical gage installations on outside surface.

For installation, gage and metal surfaces were cleaned with toluene and acetone. The adhesive used was Armstrong's A-1, a two component system that cures chemically rather than by evaporation of solvent. Cement was applied to the metal surfaces, gages placed, and covered first by cellophane, then by a neoprene pad to equalize the pressure applied by the clamp. Fig. 5 shows the clamping arrangement at sections A and B. A similar clamp was used at section L. Correct pressure was developed by closing the springs to length according to their scale, which had been measured. Clamps were left in place twenty-four hours while the cement cured.

Every gage then was tested against the possibility of having been damaged during application by measuring its resistance and its resistance to ground. All installations were satisfactory.

Incidental to the use of gages inside the vessel was the problem of sealing around the lead wires against water at pressures up to 10,000 psi. The gland designed to accomplish this is shown in Fig. 6.

Initially, the sealant used was a neoprene piece formed as shown. Under preliminary pressure test, however, this material extruded through the annular space between lead wires and compression cups at about 8000 psi. Polyethylene, being harder, then was substituted, and radial clearance between lead wires and compression cups was reduced from .003 to .001 inch. The second test proved all components of the gland to be sufficiently strong to withstand 10,000 psi, but a slight leak was present. The entire gland assembly, lead wires in place, was oven-heated for several hours at 150° F in the hope that the polyethylene would soften and conform more closely to sealing surfaces. A third pressure test to 10,000 psi showed this treatment to have been

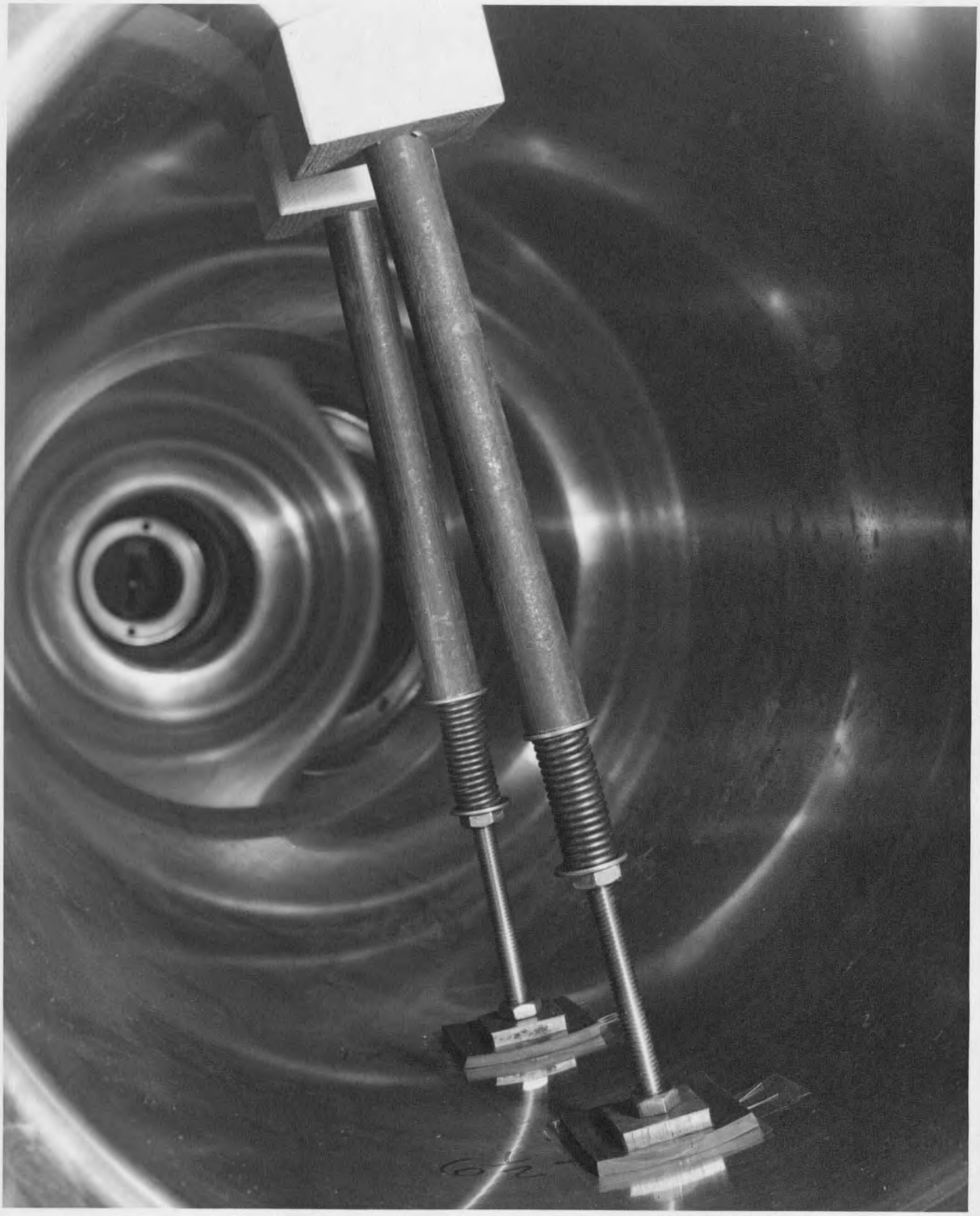

Fig. 5 - Clamping arrangement for inside gages.

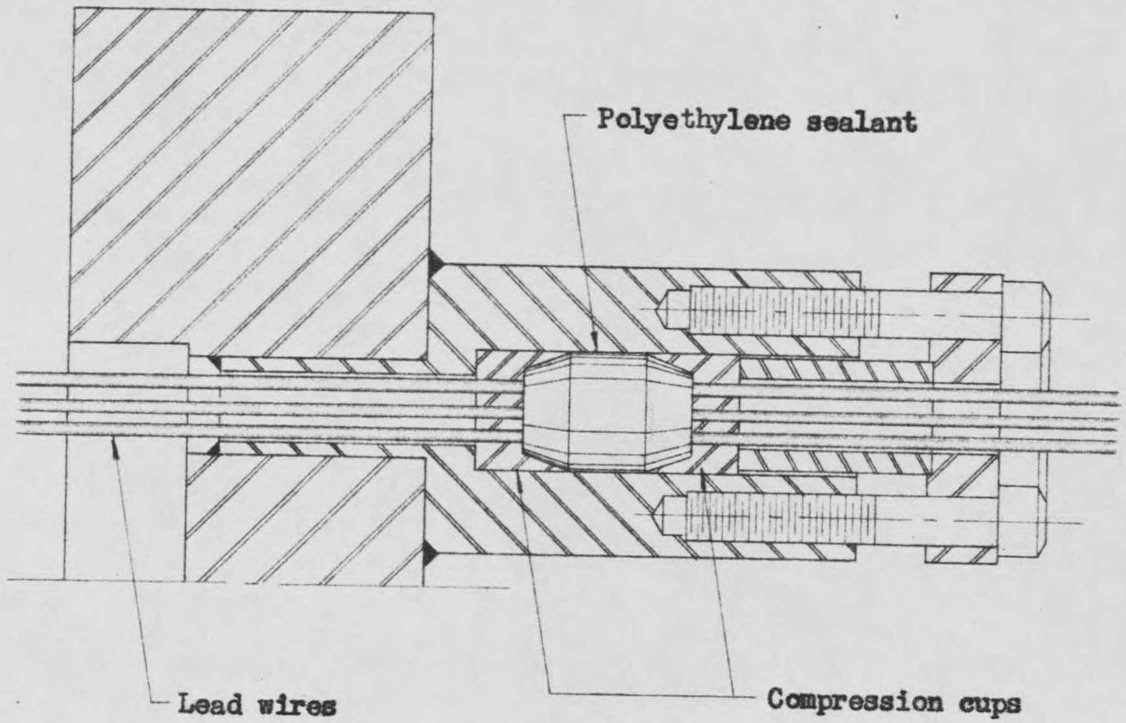

Fig. 6 - Lead wire sealing gland.

effective, since no leak was evident.

Gage installations were completed by soldering the lead wires, and insulating joints. Inside gages were painted with melted cerese wax, this being built up to about 1/16 inch thickness. Wires were then collected and taped to the vessel so it could be moved to the test cell for assembly. Fig. 7 shows the vessel at this point. The lower end cap had been assembled to permit soldering the lead wires to inside gages.

Assembly of The Vessel

The retort was then carried to the test cell, and suspended vertically between two I-beams under the flange. After inspection for leaks at the lead wire seal and flange closure, the pump casing flange bolts were loaded, following the usual procedure for setting heater bolts.* The vessel was filled with water, which was the medium of pressure transmission, the head assembled, and head flange bolts loaded.* Lines were run from the vessel to pressure source and gage, and lead wires pulled to the control room. Figures 8 and 9 show respectively the lower end and head assembly of the vessel ready for test.

Arrangement of Equipment

The test set-up is shown schematically in Fig. 10, and the station at which readings were taken is pictured in Fig. 11. In the foreground are the switch boxes through which the measuring and compensating gages were directed to the indicator. The pressure gage is shown mounted on the wall, and the strain indicator, a Baldwin SR-4 type "M", is at right.

*See Appendix B for calculations and method.

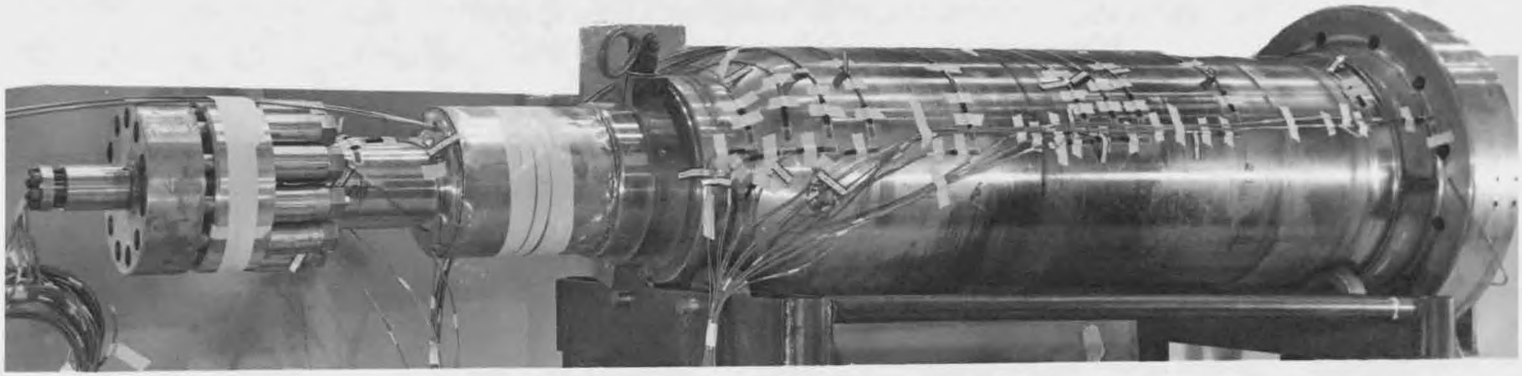

Fig. 7 - Over-all view of the vessel, showing completed gage installations.

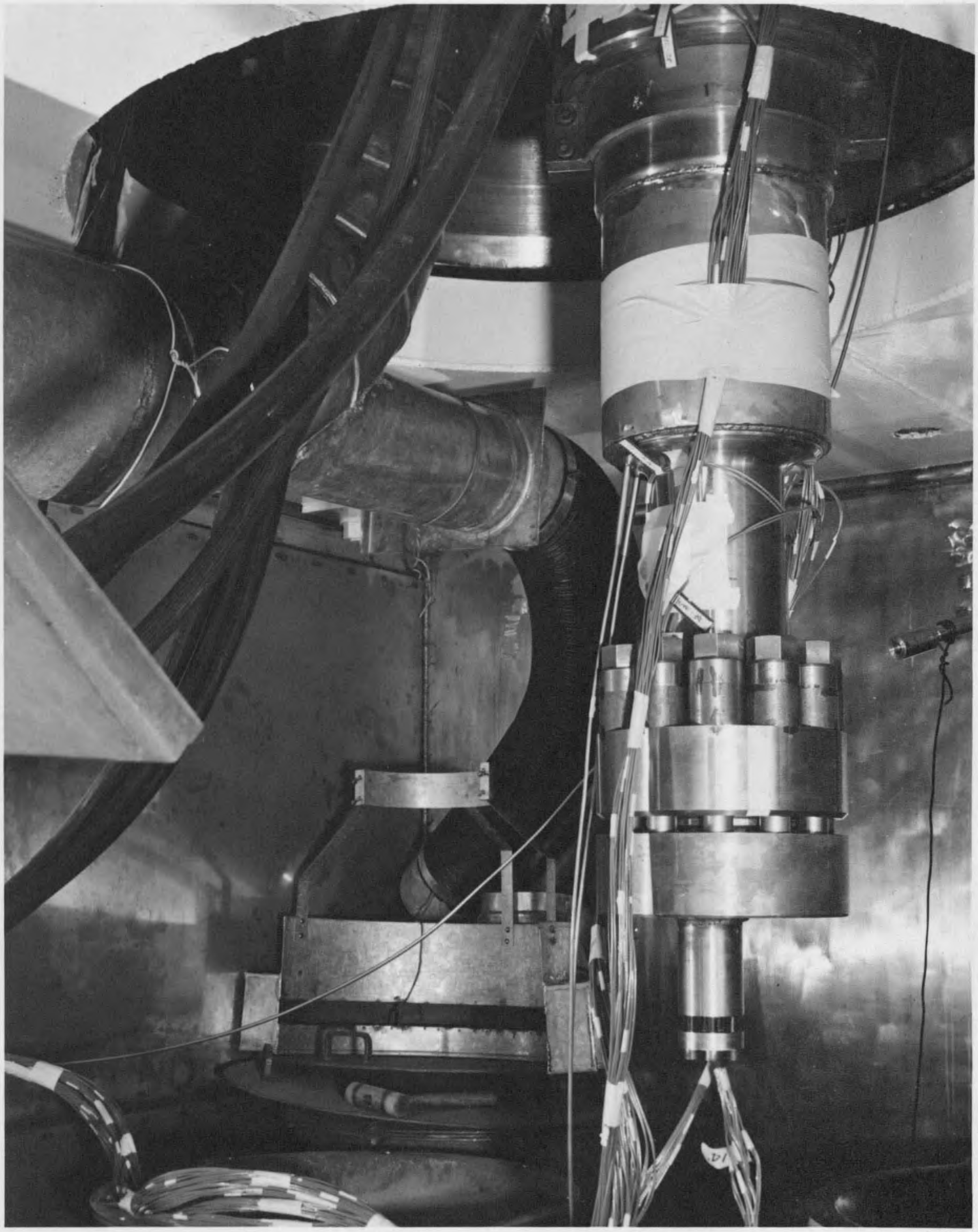

Fig. 8 - Lower end of completed vessel assembly in test cell.

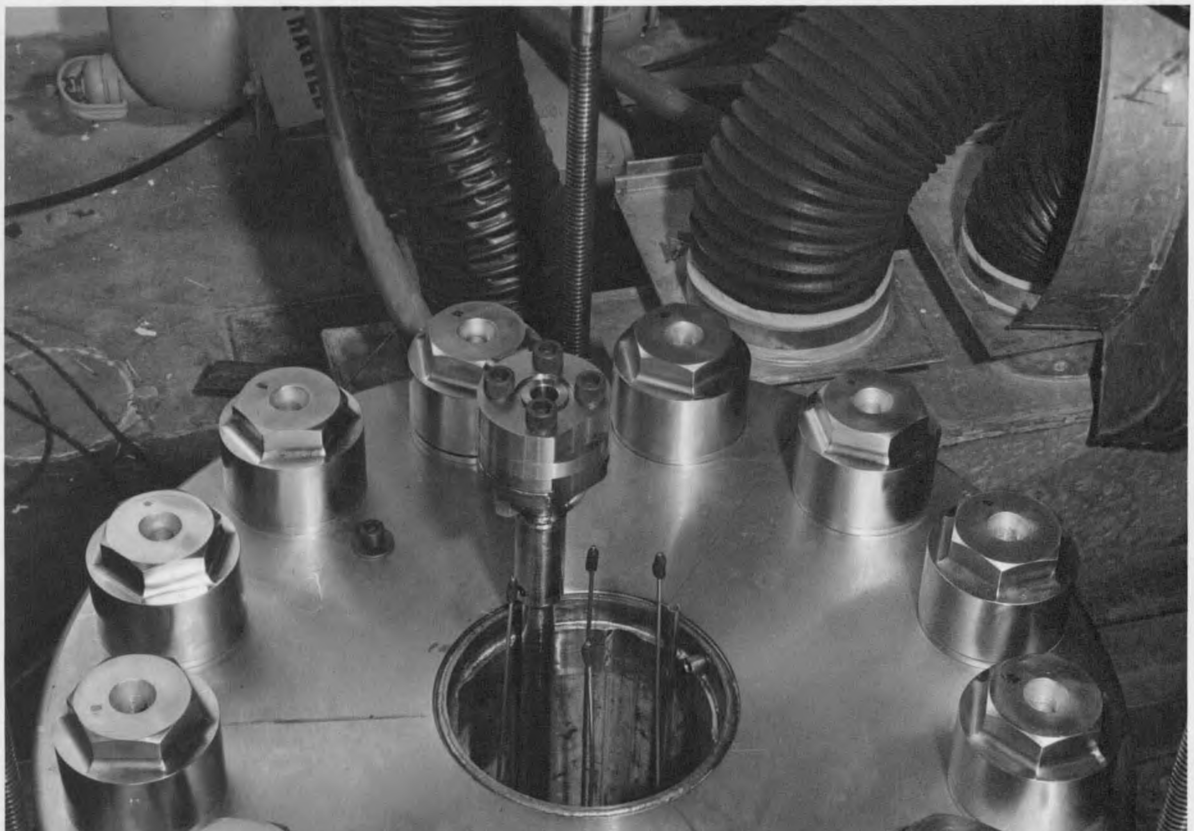

Fig. 9 - Head of assembled vessel in test cell.

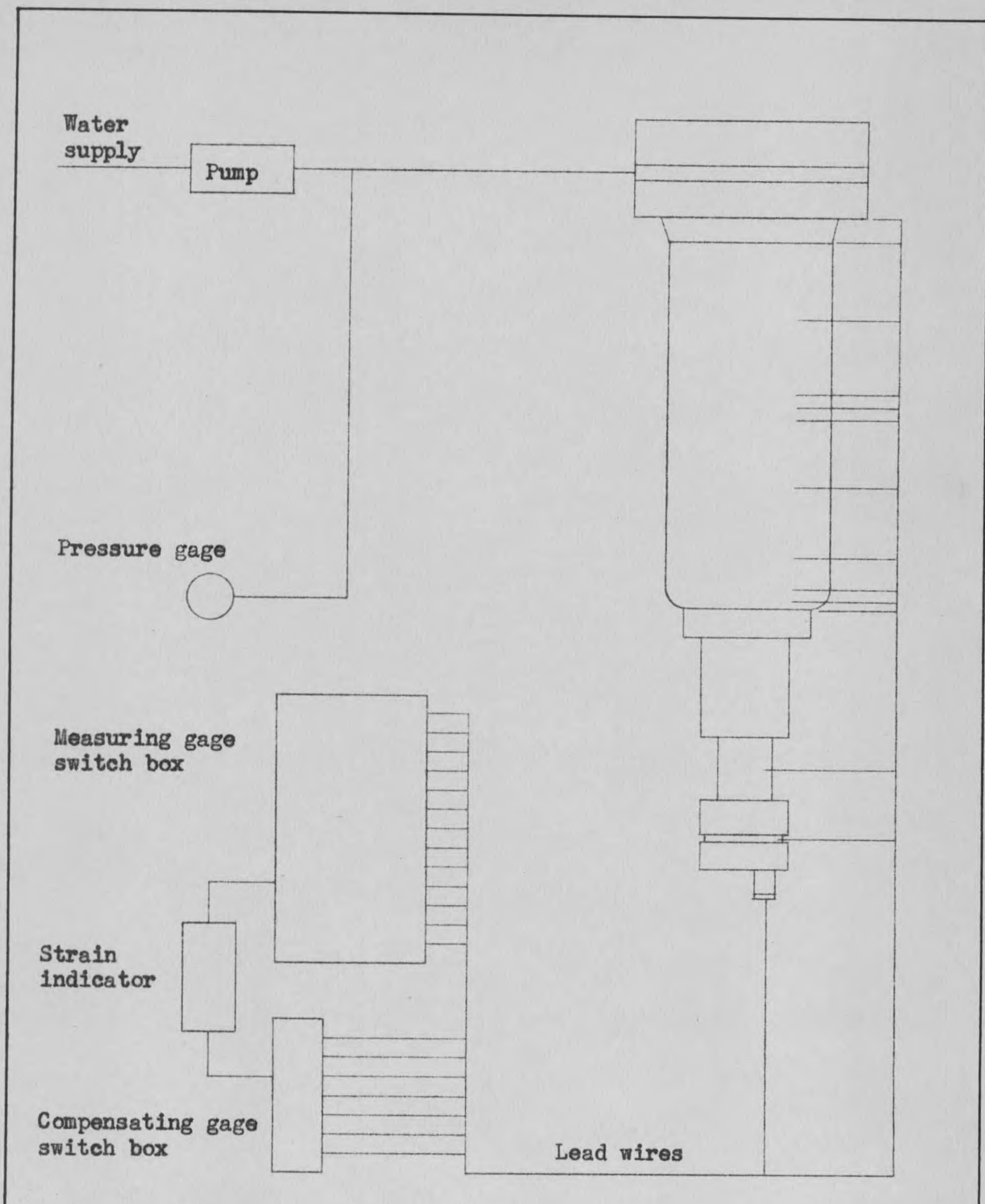

Fig. 10 - Schematic of test set-up.

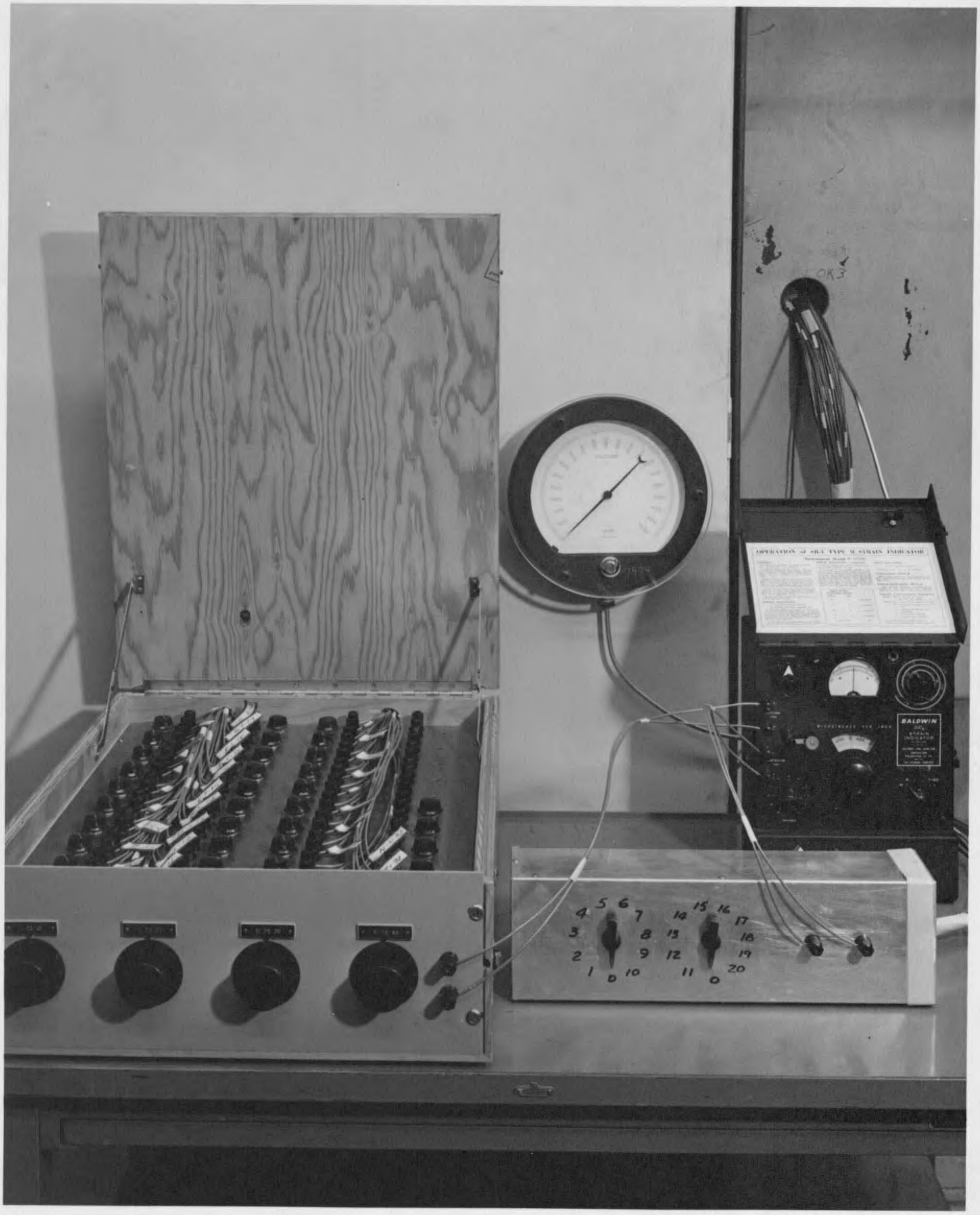

Fig. 11 - Station at which readings were taken.

Test Procedure

Stability of all circuits was checked during the three days preceeding the first test. All outside gage circuits were stable, but inside gage circuits were erratic, even under no pressure.

The first test was made without the reinforcing clamp on the lower section of the pump casing, so that strain measurements could be taken on the outside surface at section L. Indicator readings were taken at 1000 psi pressure increments to 5000 psi, and at 500 psi increments to 6500 psi, the highest pressure attained. The system was then depressurized at about 1500 psi per hour, indicator readings being taken periodically during descent. From data taken in this test it was apparent that 1) inside gages would yield no useful information, and 2) all outside gages could be considered reliable.

The clamp was next installed on the pump casing for test to higher pressures. Two runs were made, the procedure being the same in both cases. After each 1000 psi increase in pressure, readings were taken for all gages. The pressure vs circumferential strain curve for section B, known to be a critical section, was plotted as the test progressed. As the strain rate there increased, pressure increments were reduced, and when the curve approached the horizontal no further pressure increase was made. As in the first test, the pressure was released at 1500 psi per hour, and readings were taken periodically during descent.

Results

Appendix A comprises curves constructed from the test data. They exhibit limits of proportionality between pressure and strain as listed in Table I.

TABLE I - PRESSURE AT LIMIT OF PROPORTIONAL STRAIN

Section	Pressure, psi*					
	Test #1		Test #2		Test #3	
	Circumferential	Axial	Circumferential	Axial	Circumferential	Axial
A	5000	5000	6000	8000	7000	9000
B	4200	5100	5200	6000	7000	7000
C	4200	5200	6000	7000	8000	
D	4200	5400	6000	7300	8000	
E	5000	5400	6000	7000	8000	
F	5000	5200	6000	7000	8000	
G	4200	5000	6000	7000	8000	9000
H	5000	5000	7000	7500	9000	9000
I	5000		8000		9000	
J						
K		5000				
L**	4300		———	———	———	———
M	4000	5000	6000	7000	8000	7000

*Where no pressure is given, the proportional limit was not exceeded in that test.

**At section L, strains were measured in the first test only.

Successive strain hardening and consequent increase in strength is apparent at every section where yielding occurred.

Conclusions

Total permanent set was greatest at section B, where 1150 microinches per inch circumferential strain and 725 microinches per inch longitudinal strain were received. Since these do not constitute severe deformation, it can be assumed that the vessel was not weakened by testing, and the proportional limit pressures of the final test accepted as safe measures of the strength at corresponding sections.

There will be no further yielding at pressures up to 7000 psi, and rupture will not occur below 9000 psi.

The test results were reasonably well predicted by design analysis. In general, it was found that strains were somewhat higher than calculated, but shearing stresses, which were the criterion of failure, were lower.

Appendix

A. PRESSURE VS STRAIN TEST DATA

B. METHOD OF SETTING PUMP CASING AND HEAD FLANGE BOLTS

Appendix A

PRESSURE VS STRAIN TEST DATA

The data presented graphically on the following pages are the basis of this report.

As described in Section 2, three test runs were made; herein, the three records for each gage are brought together consecutively. Strains plotted in each test do not include those residual from the preceeding test, but refer to zero at the beginning of the run.

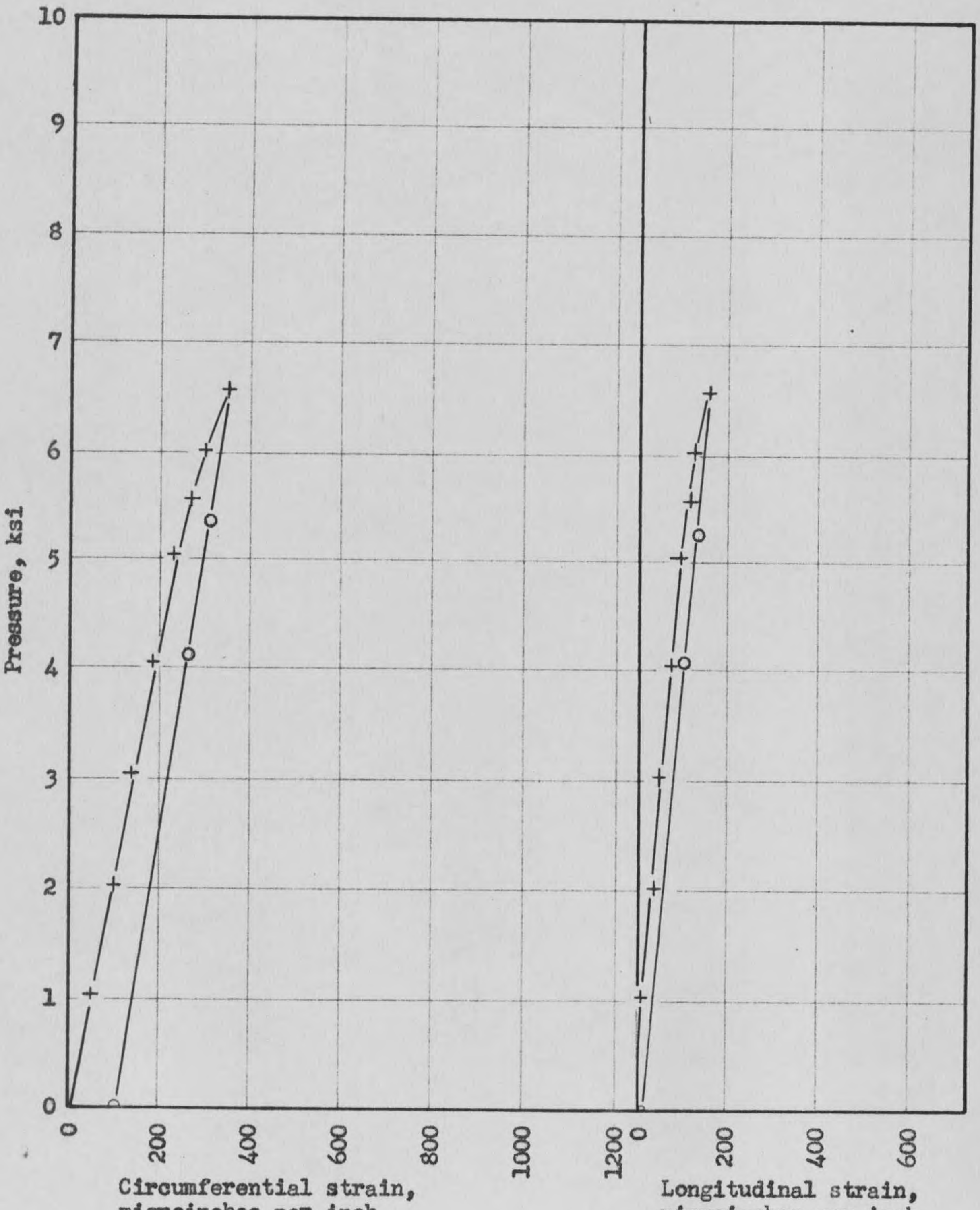

Fig. 12 - Pressure vs strain at section A, test #1.

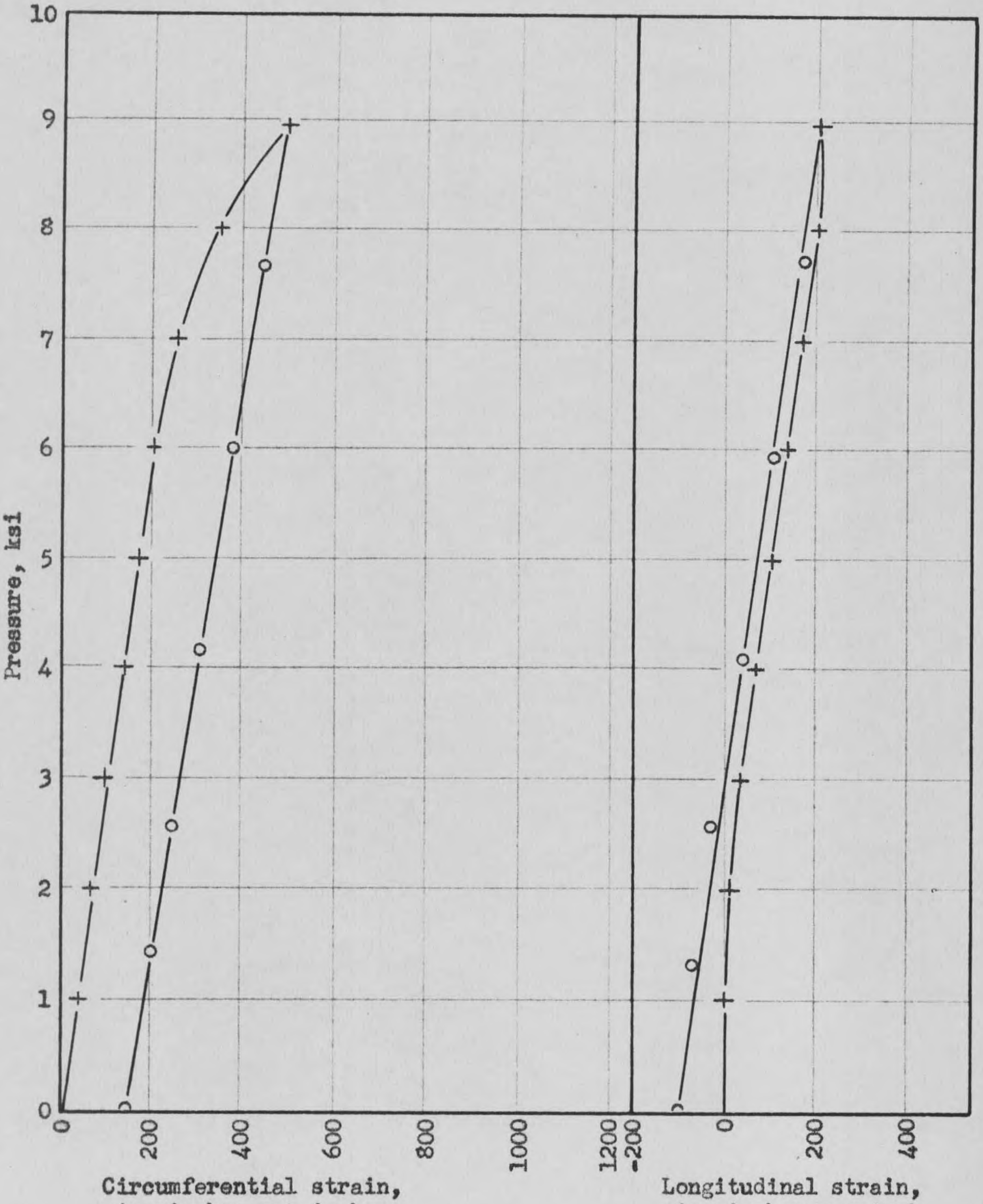

Fig. 13 - Pressure vs strain at section A, test #2.

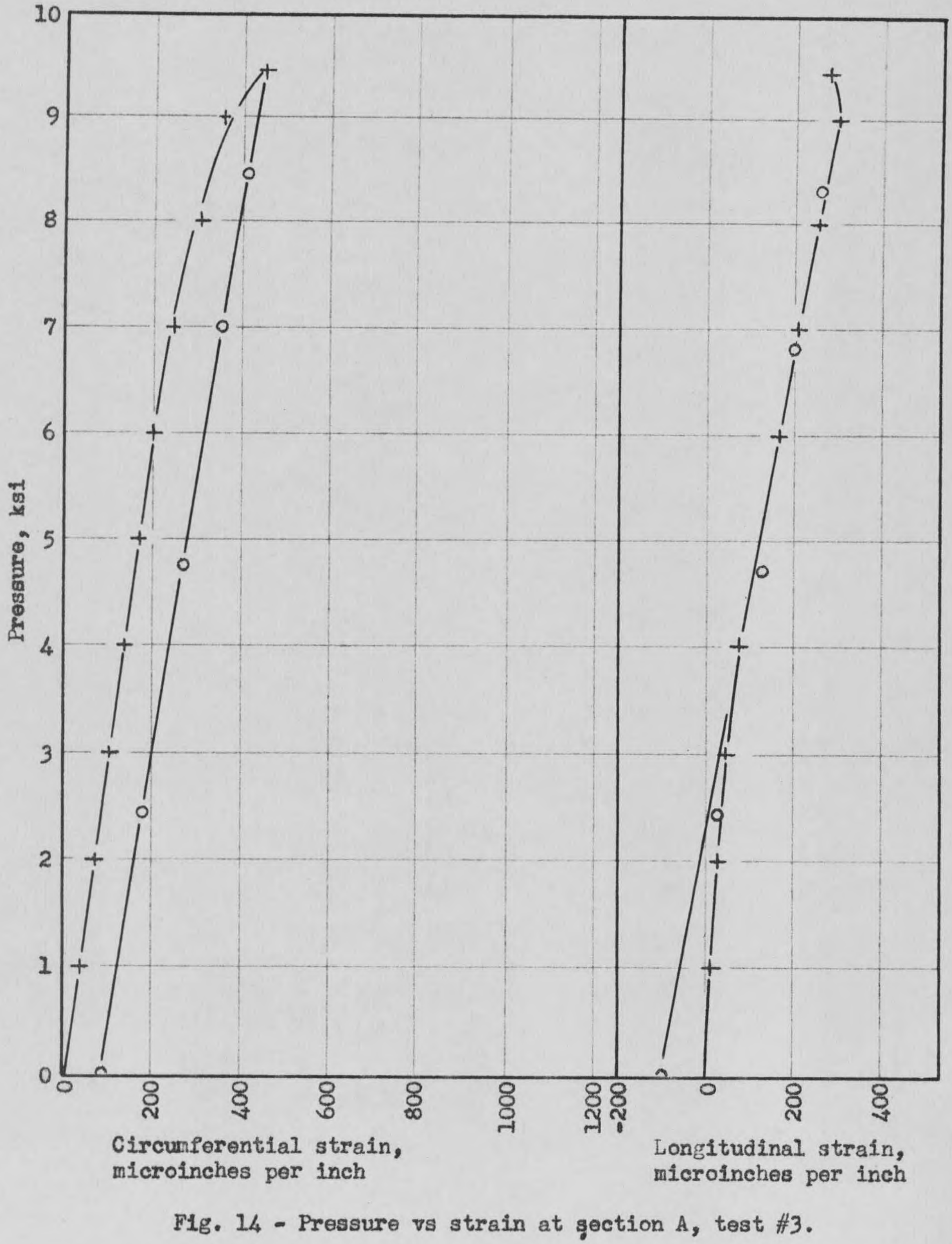

Fig. 14 - Pressure vs strain at section A, test #3.

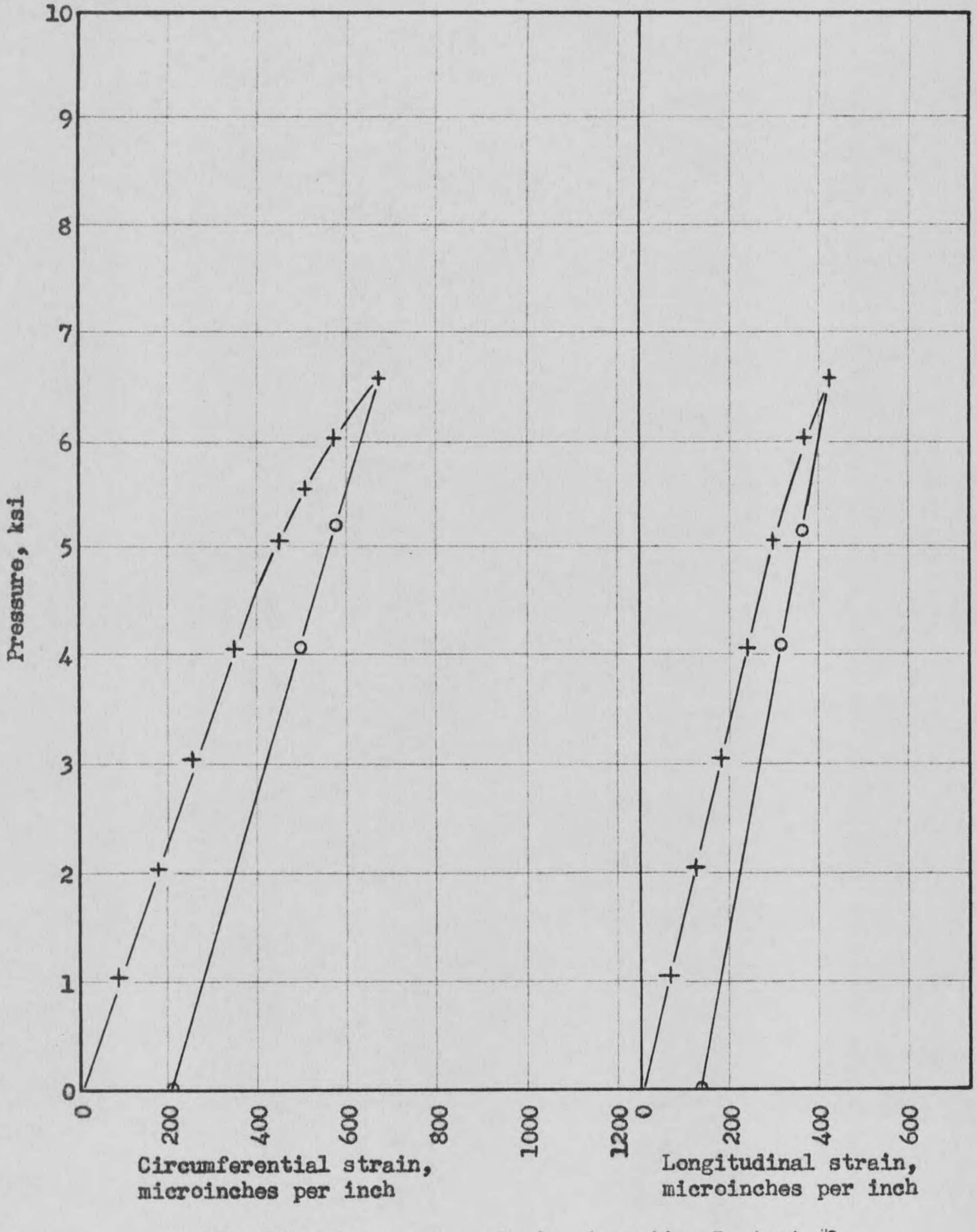

Fig. 15 - Pressure vs strain at section B, test #1.

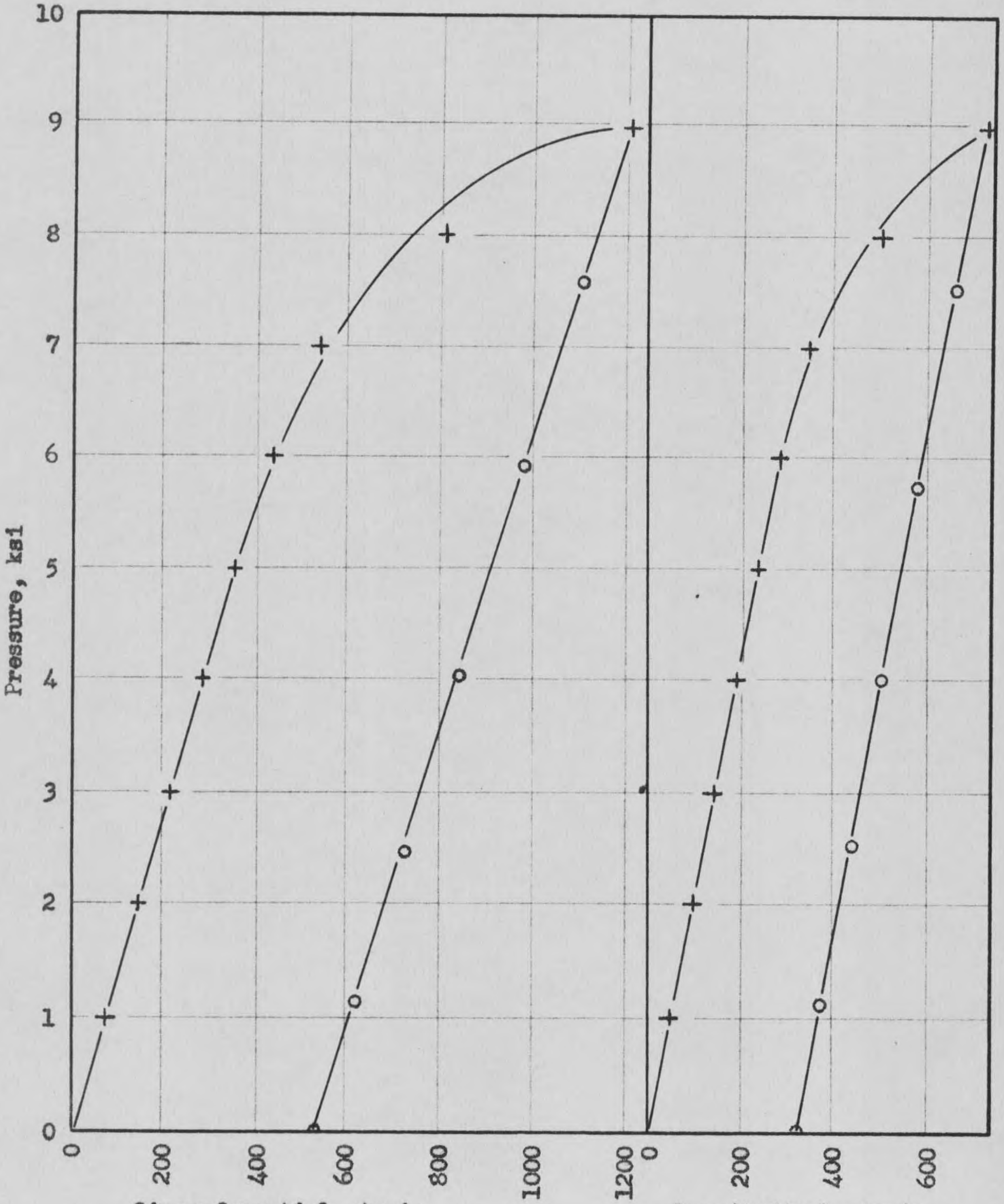

Fig. 16 - Pressure vs strain at section B, test #2.

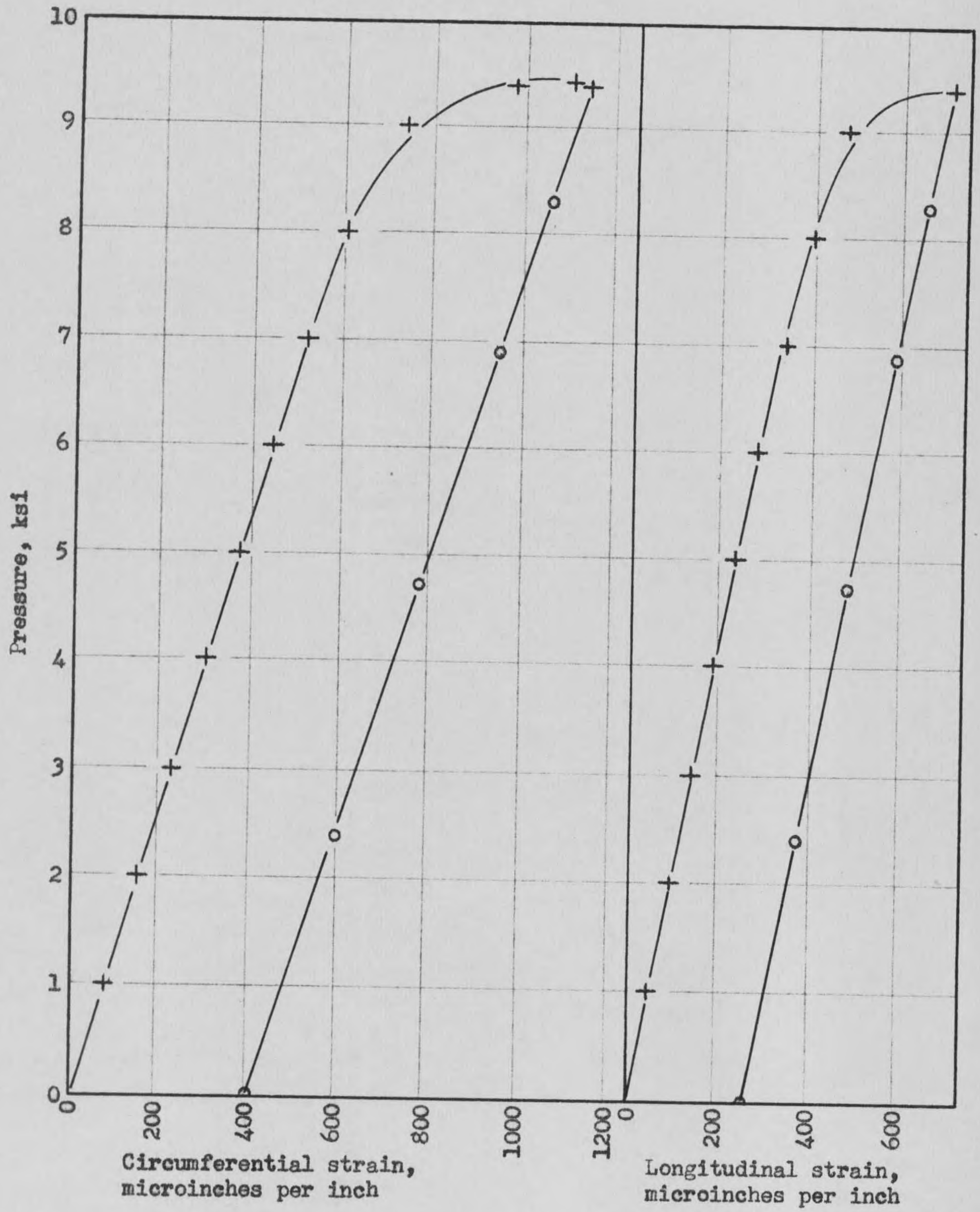

Fig. 17 - Pressure vs strain at section B, test #3.

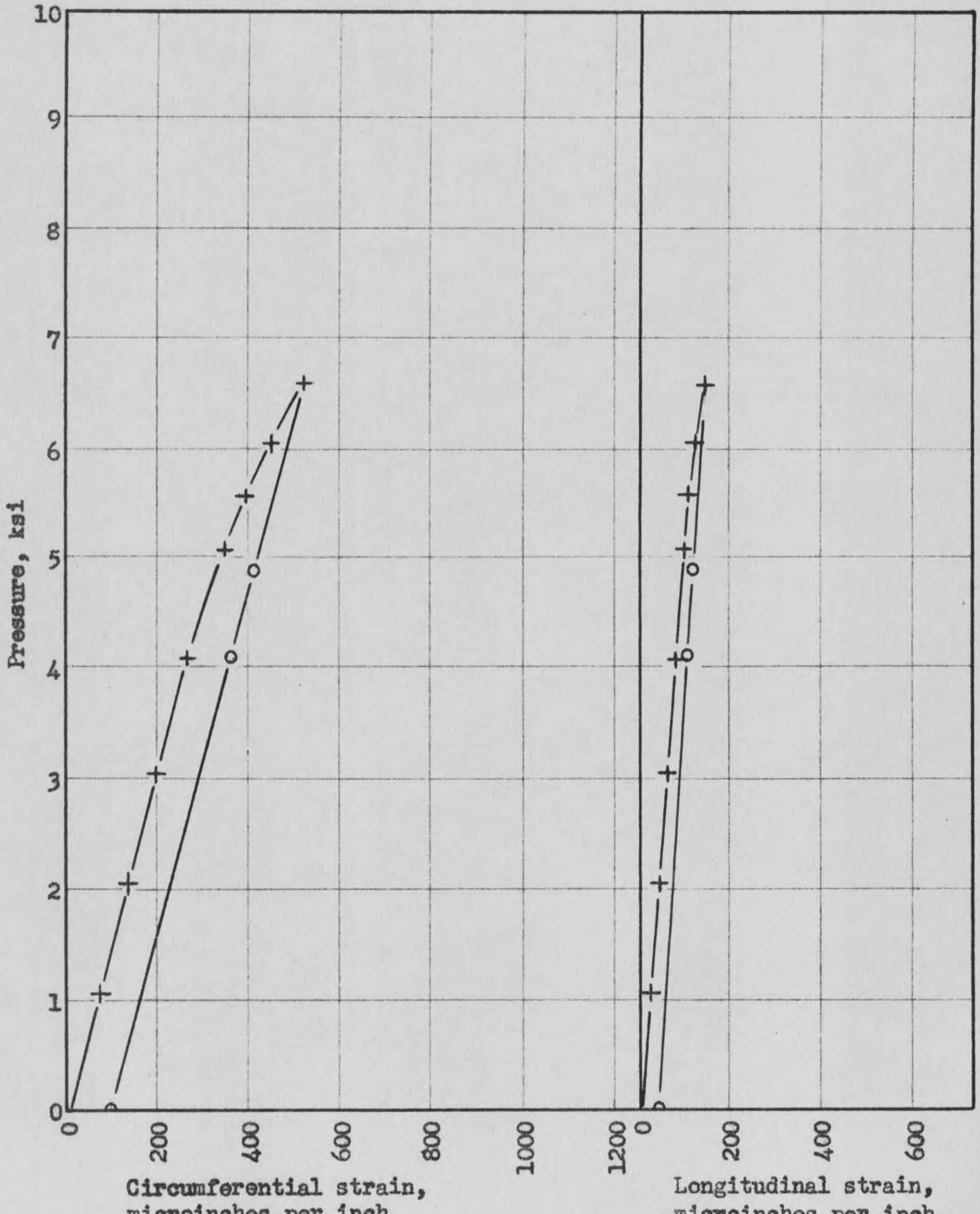

Fig. 18 - Pressure vs strain at section C, test #1.

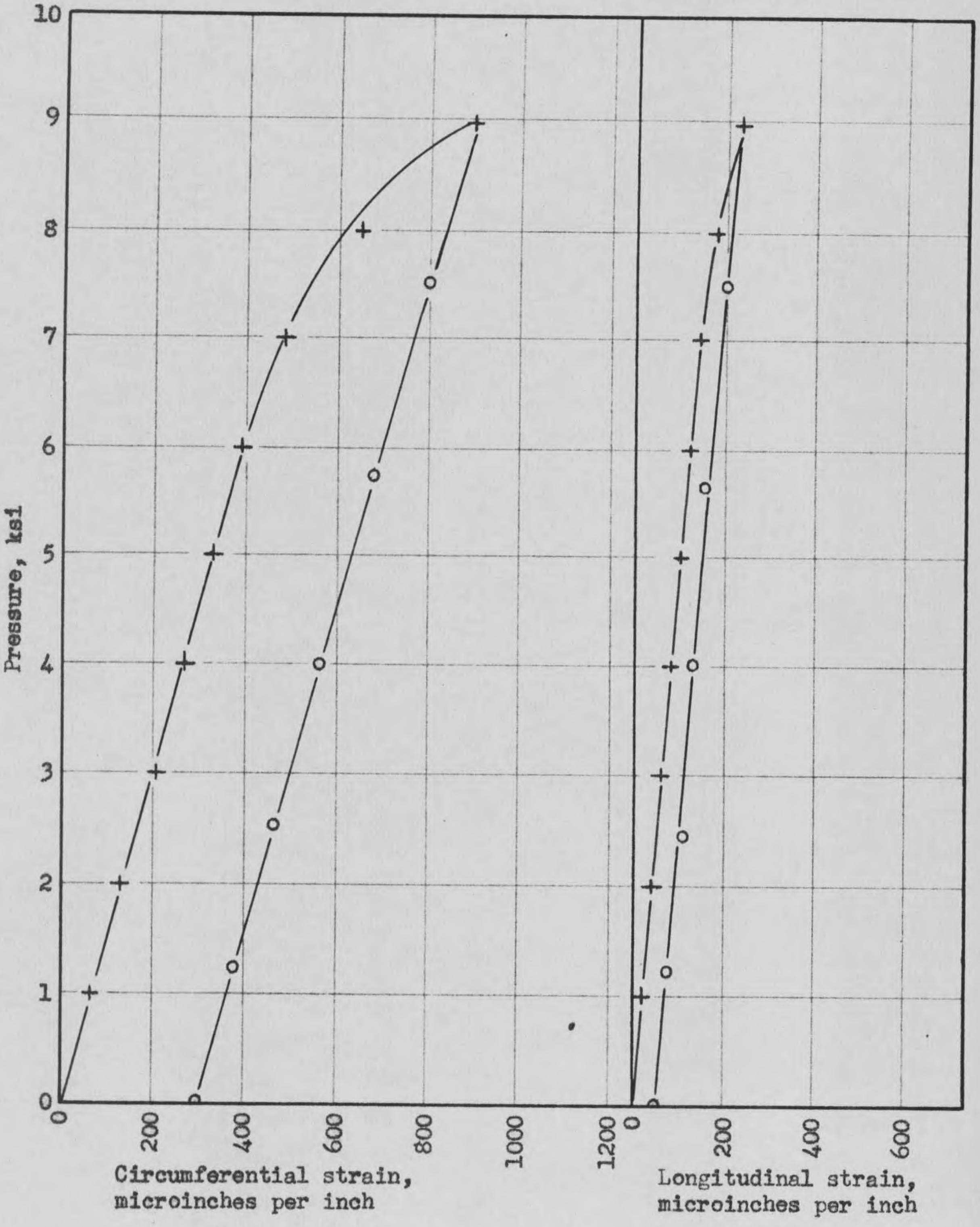

Fig. 19 - Pressure vs strain at section C, test #2.

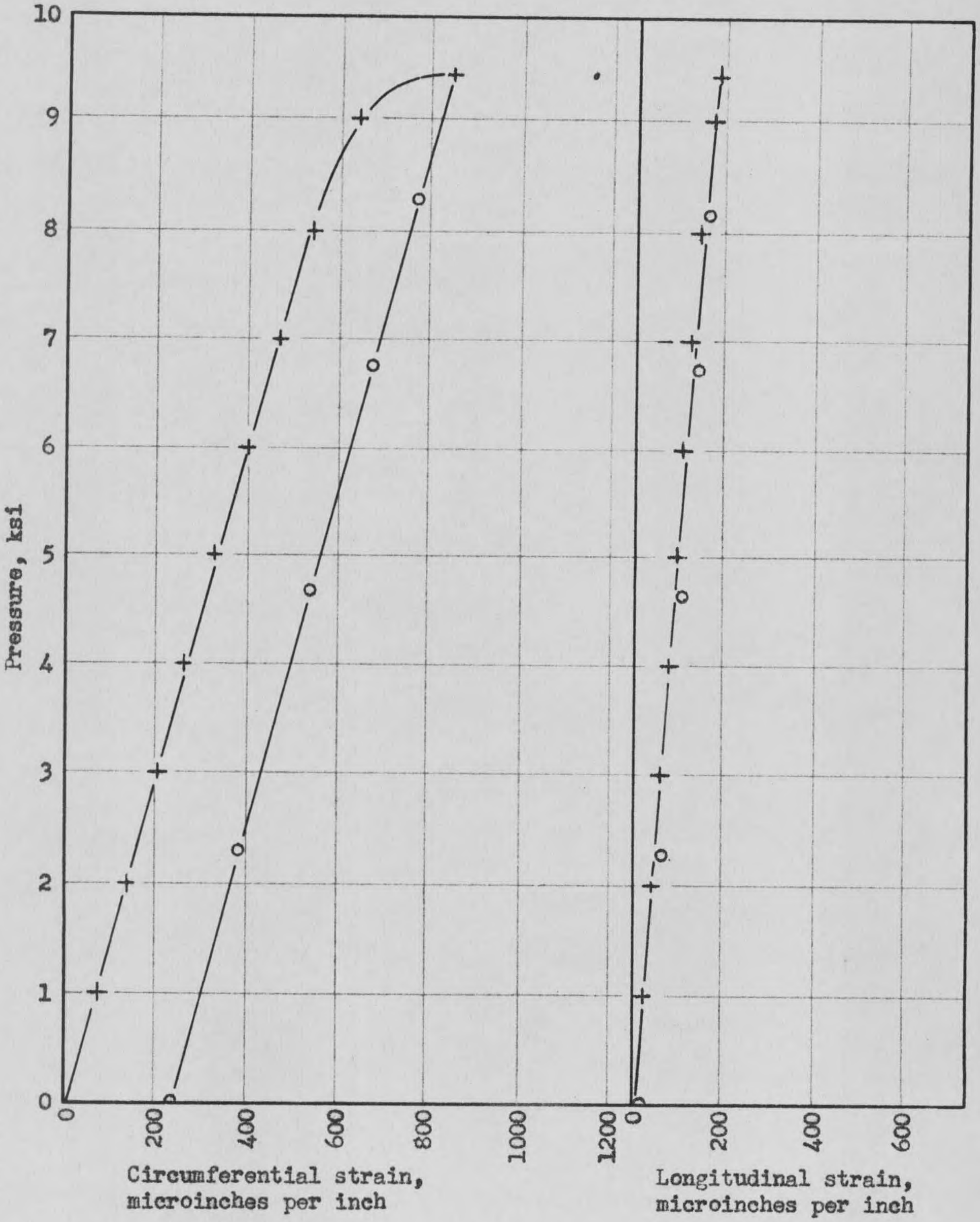

Fig. 20 - Pressure vs strain at section C, test #3.

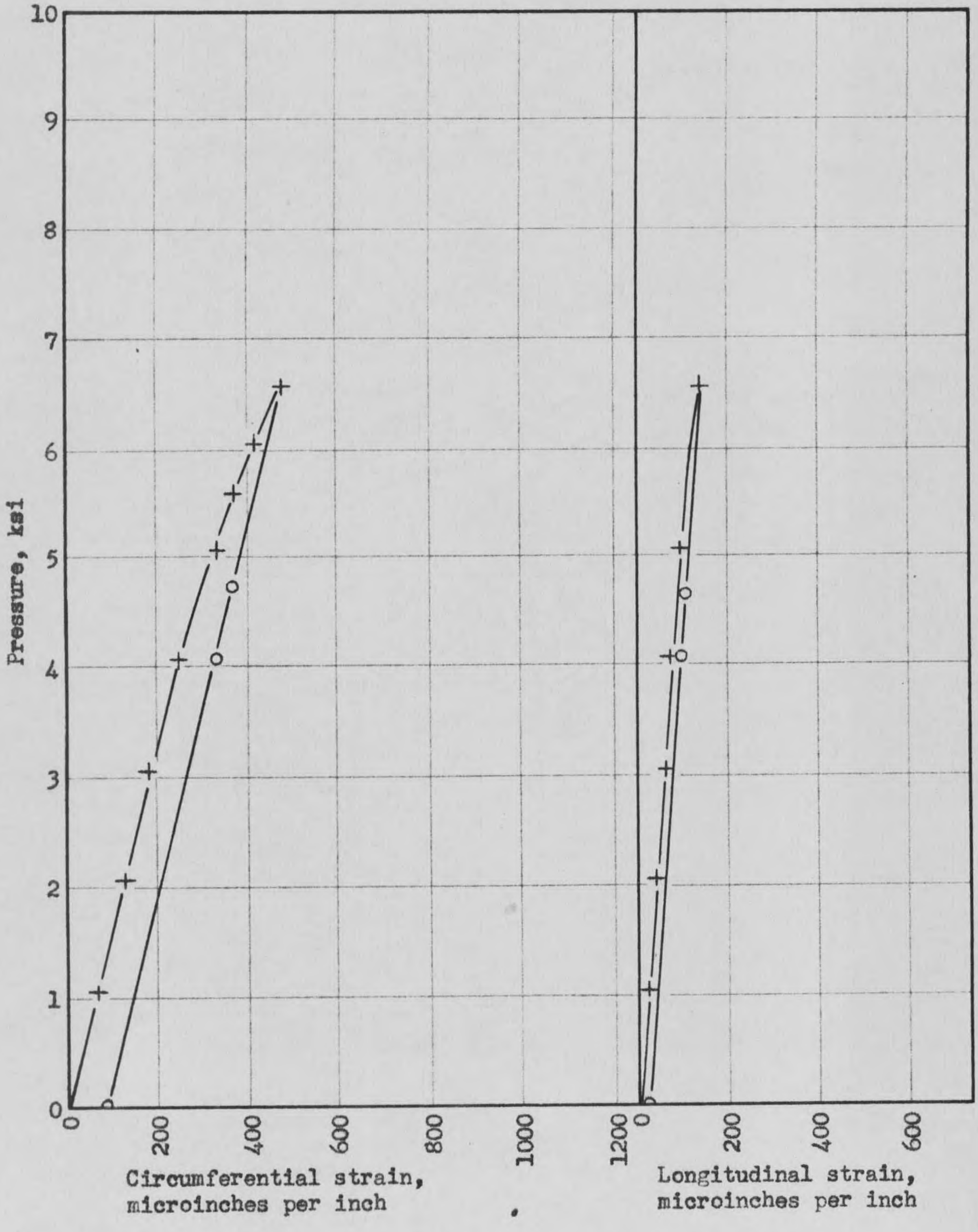

Fig. 21 - Pressure vs strain at section D, test #1.

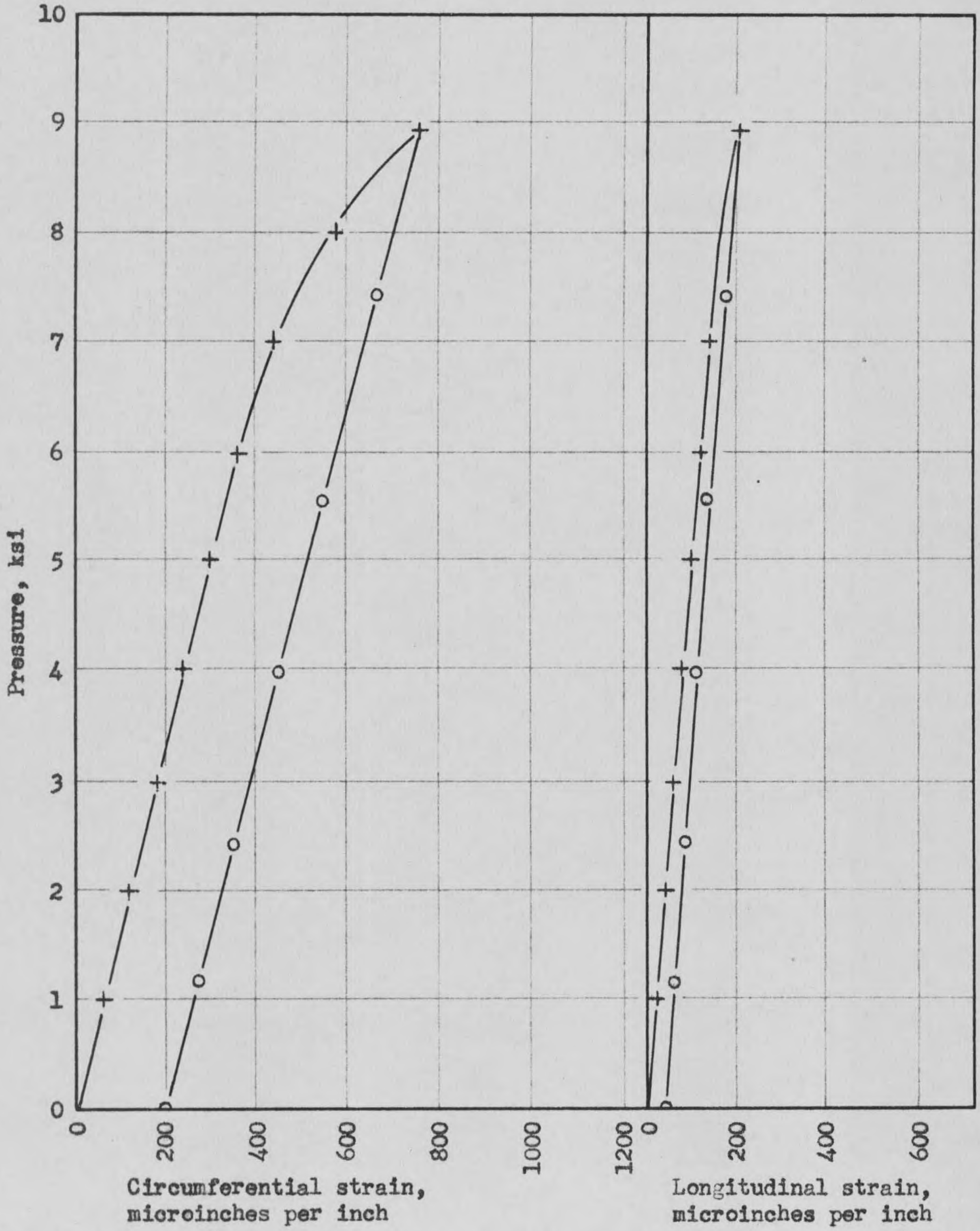

Fig. 22 - Pressure vs strain at section D, test #2.

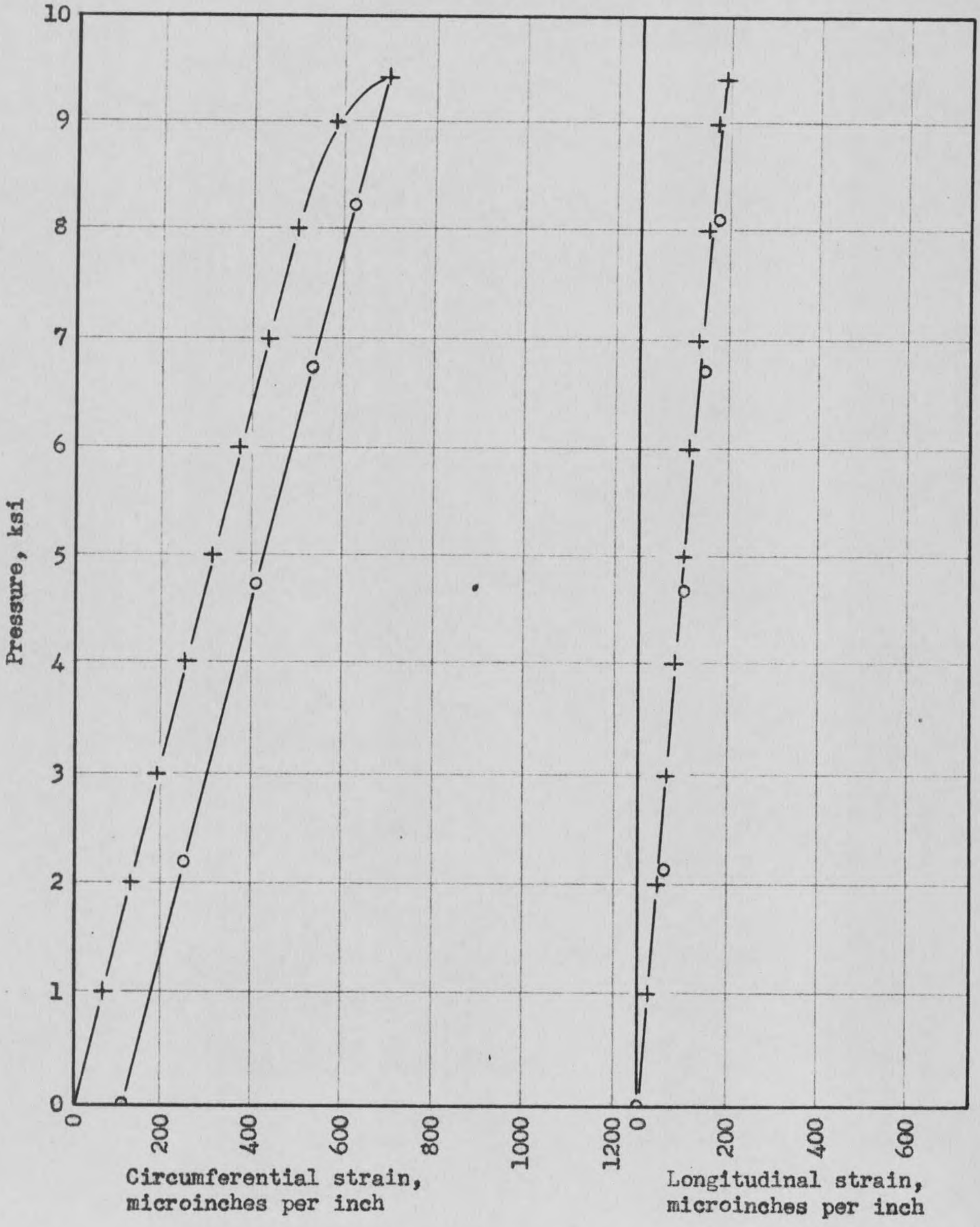

Fig. 23 - Pressure vs strain at section D, test #3.

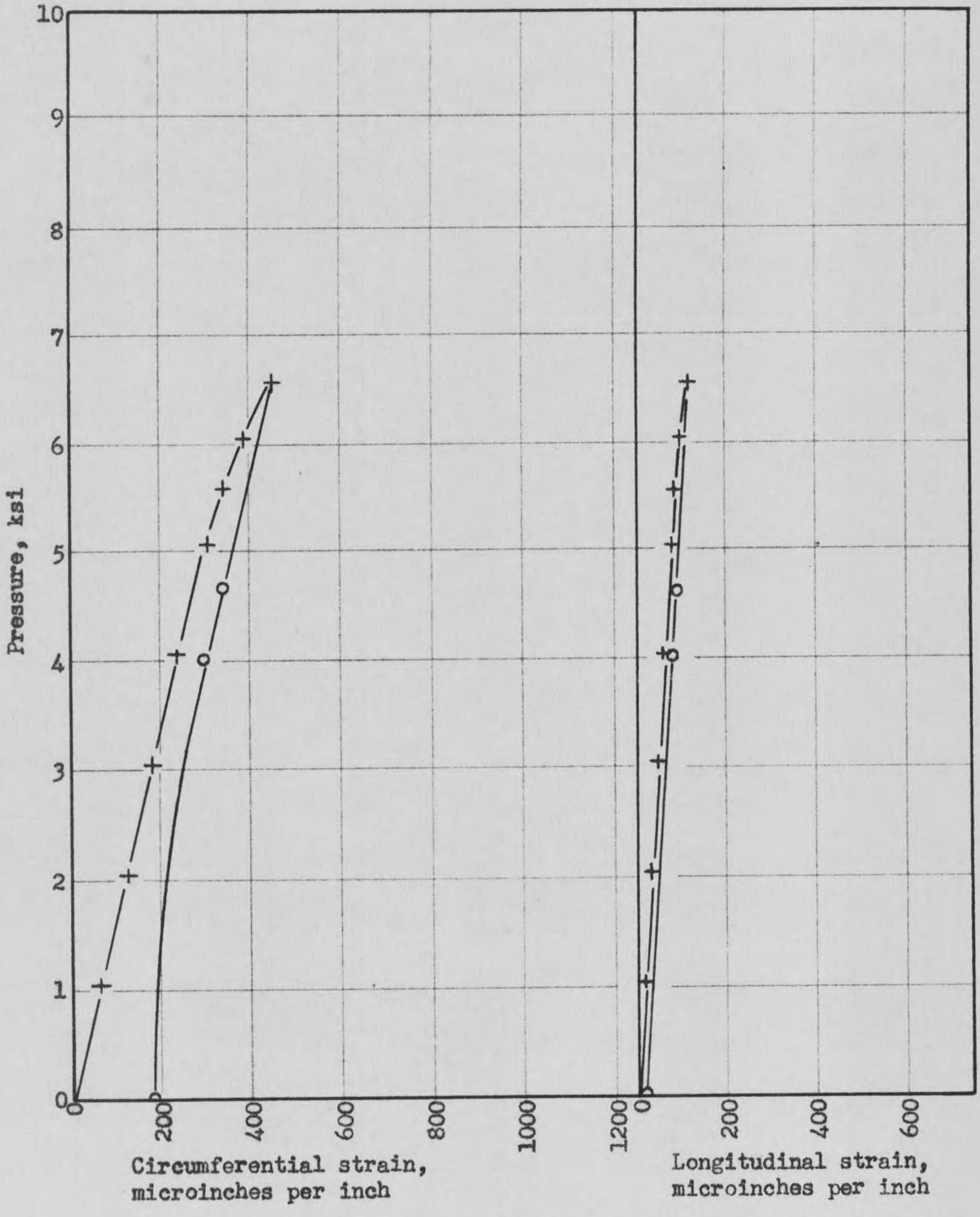

Fig. 24 - Pressure vs strain at section E, test #1.

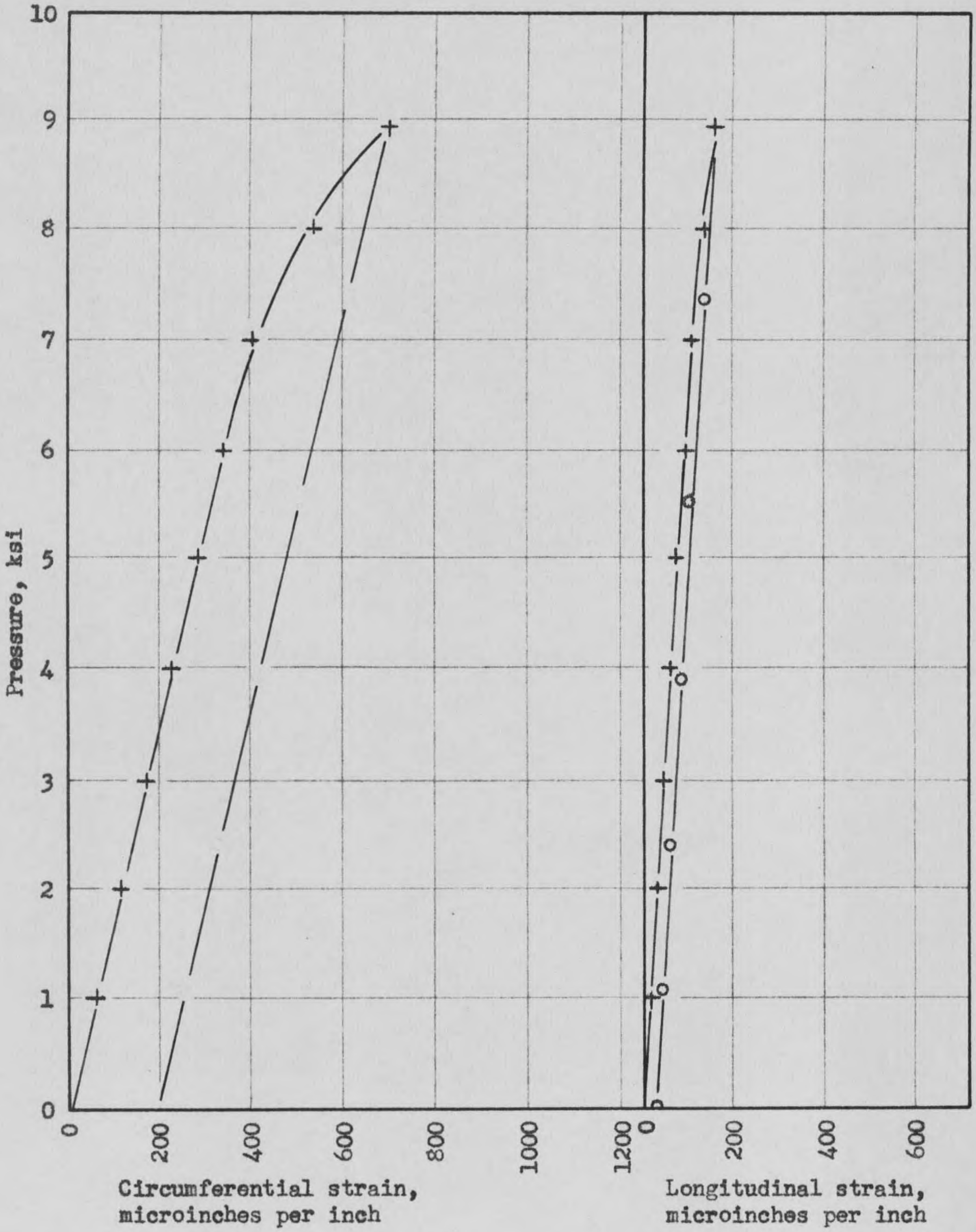

Fig. 25 - Pressure vs strain at section E, test #2.

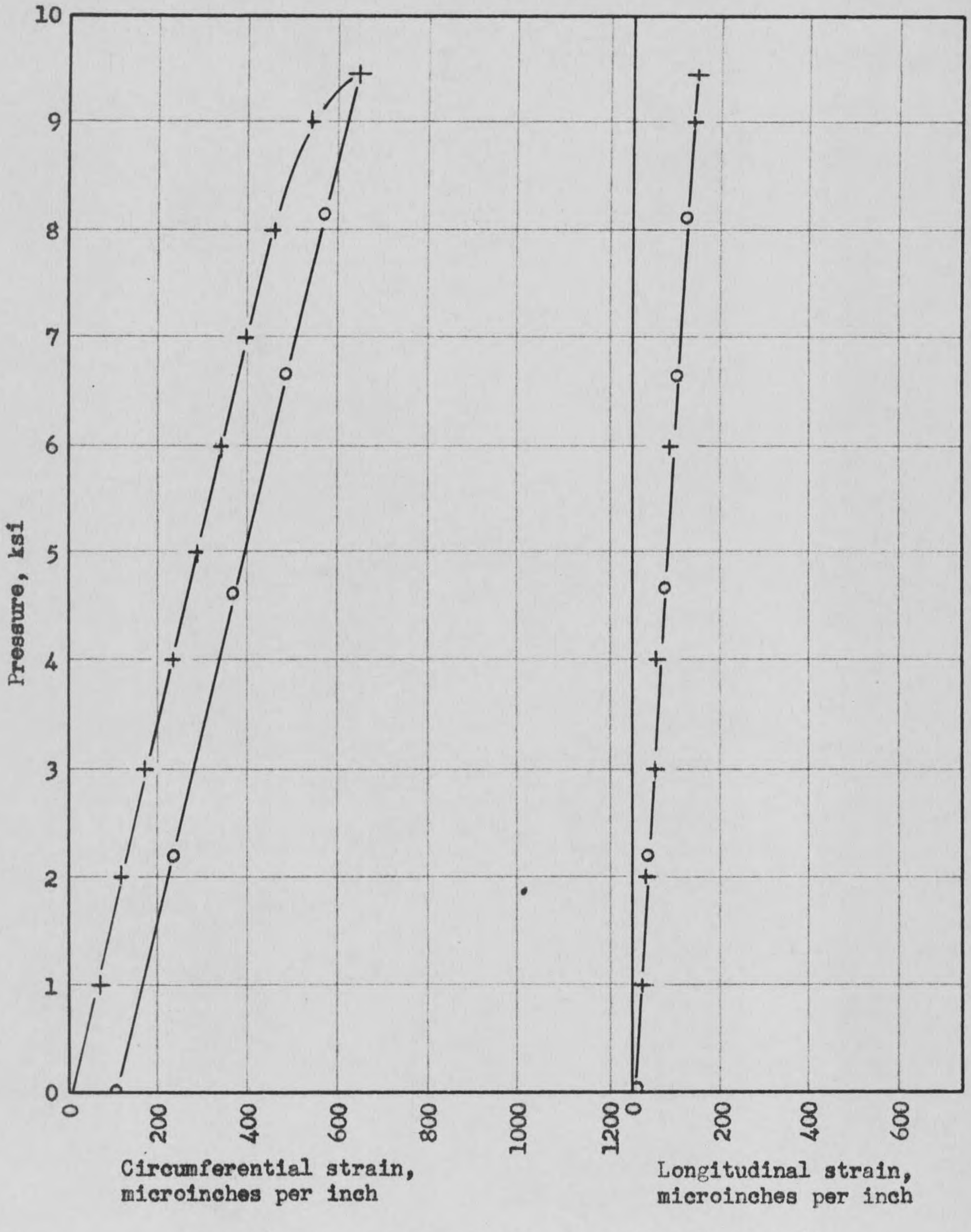

Fig. 26 - Pressure vs strain at section E, test #3.

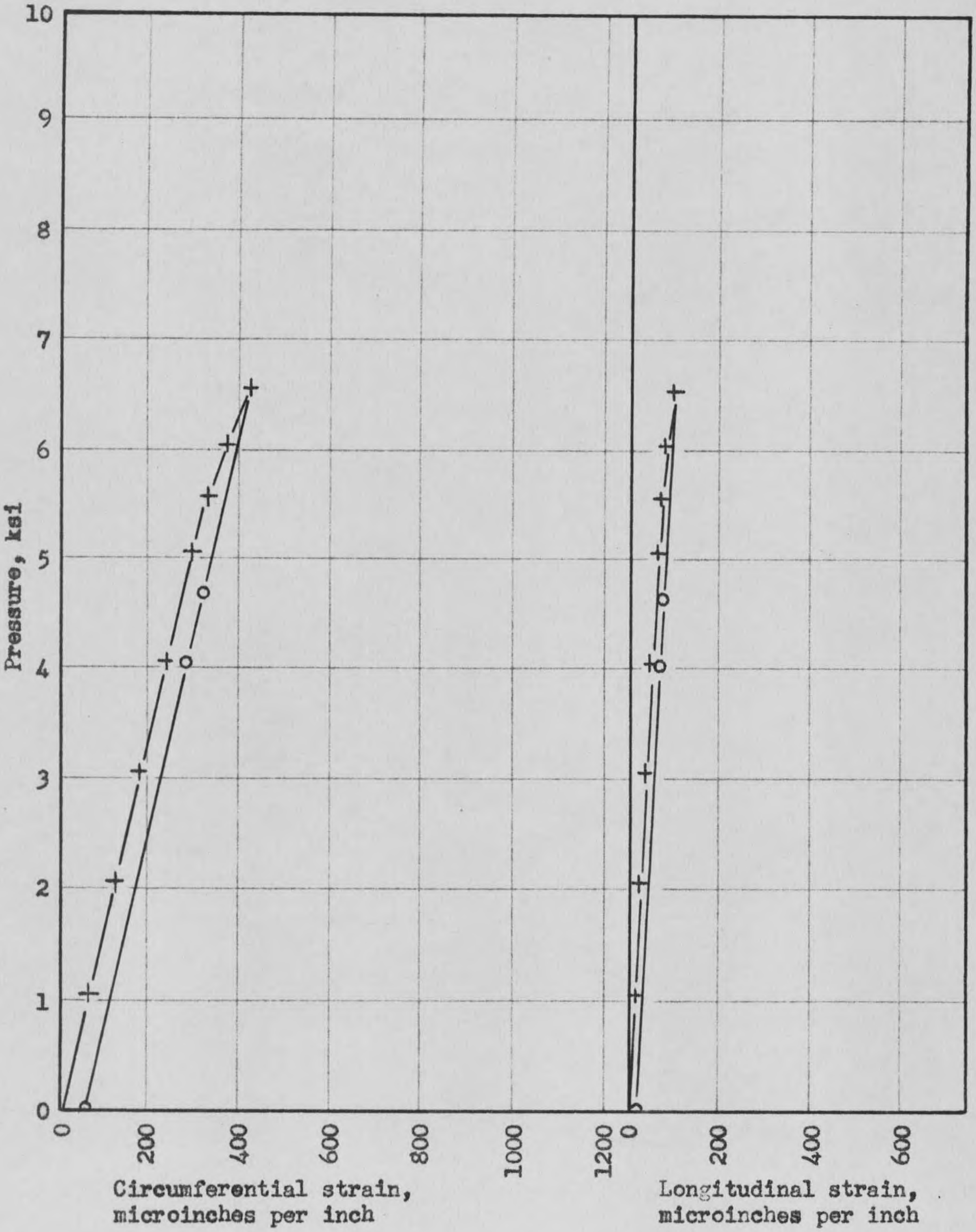

Fig. 27 - Pressure vs strain at section F, test #1.

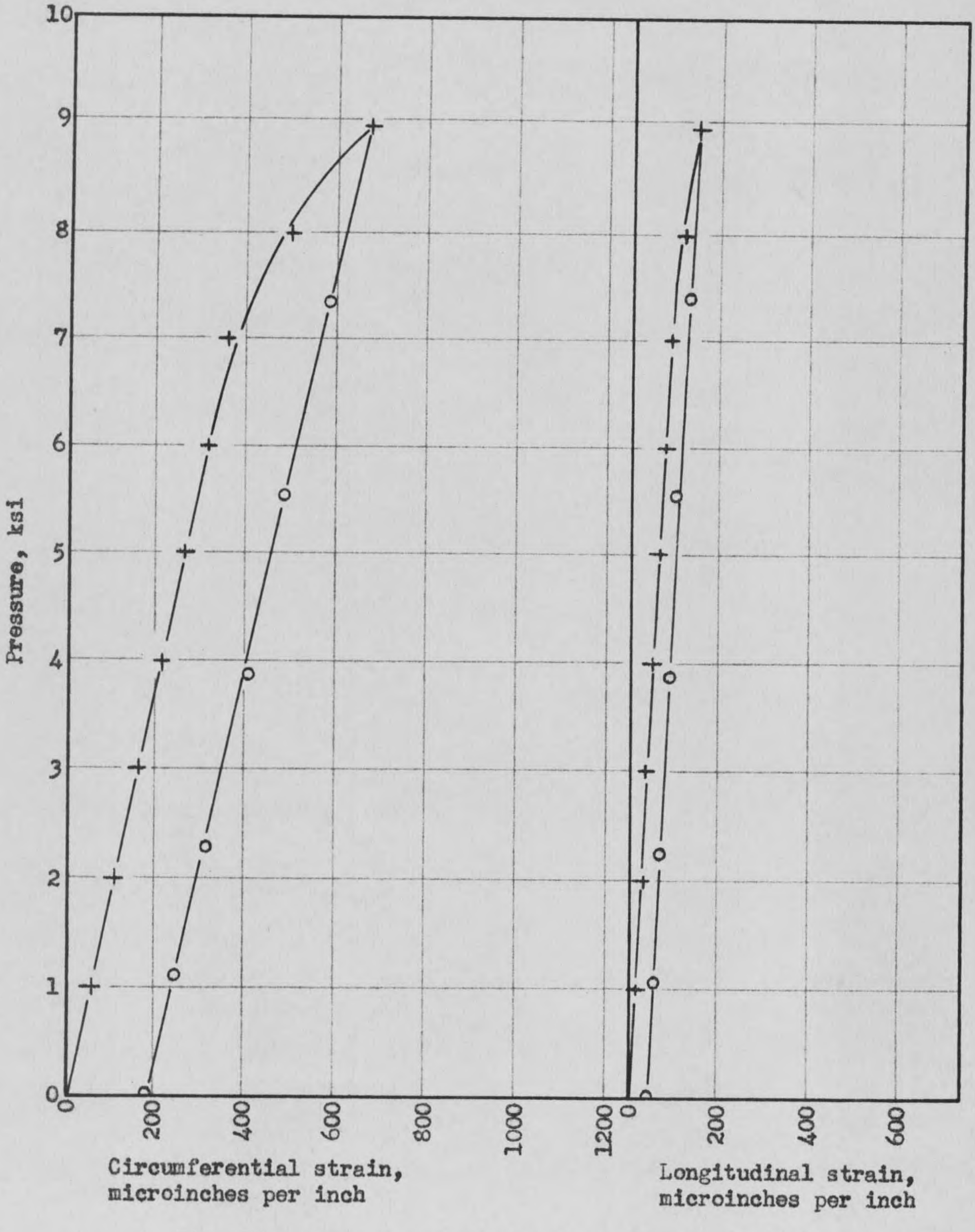

Fig. 28 - Pressure vs strain at section F, test #2.

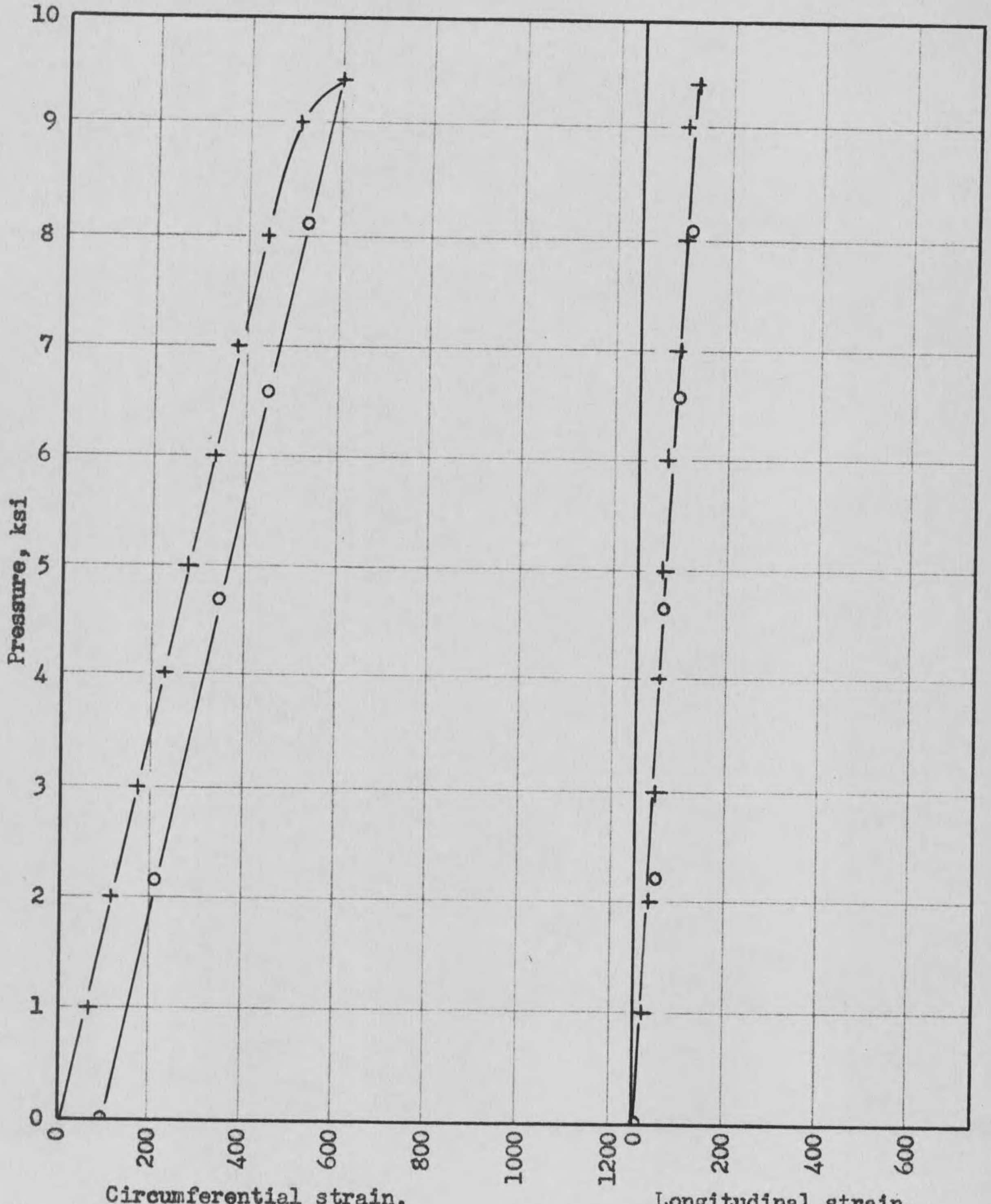

Fig. 29 - Pressure vs strain at section F, test #3.

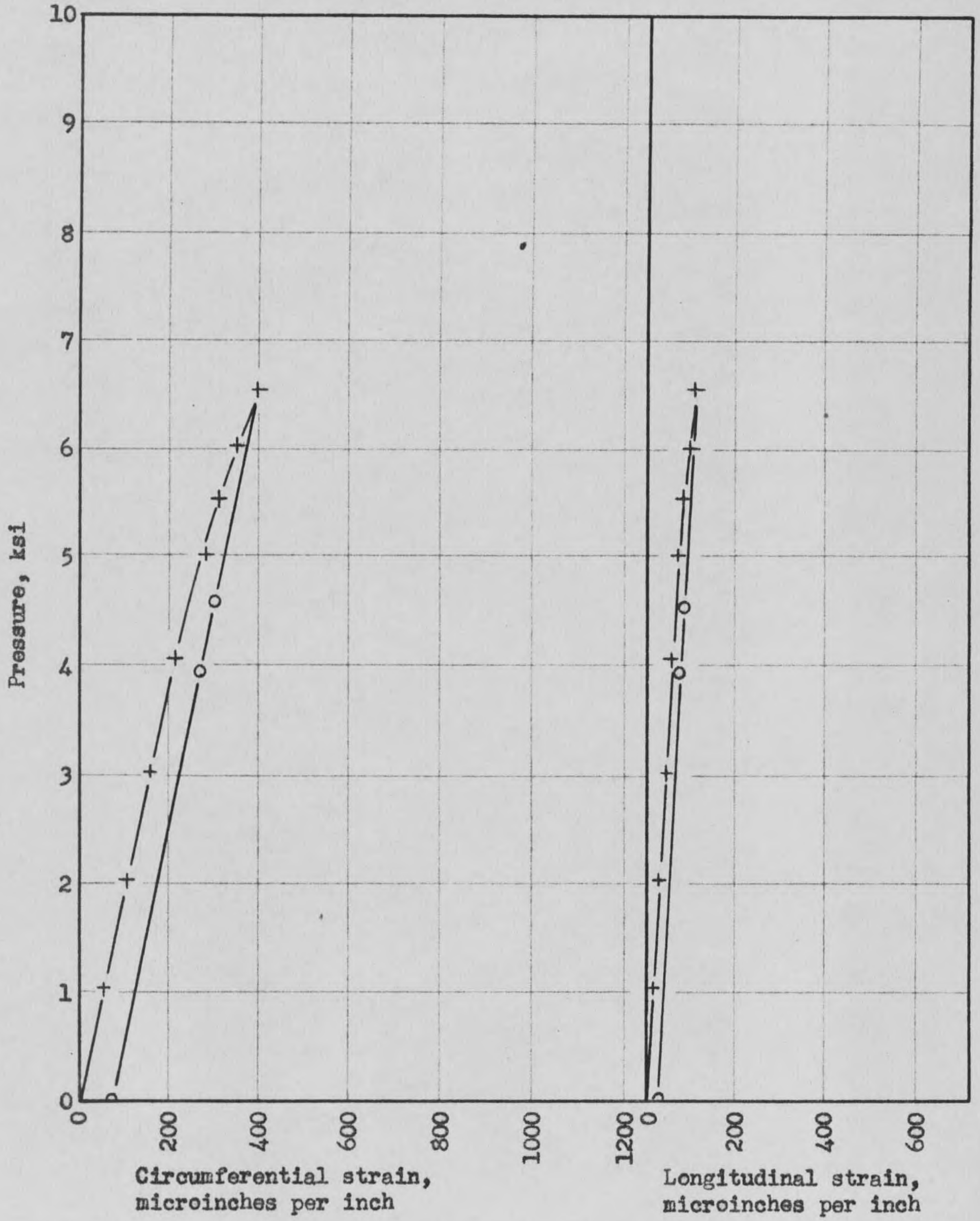

Fig. 30 - Pressure vs strain at section G, test #1.

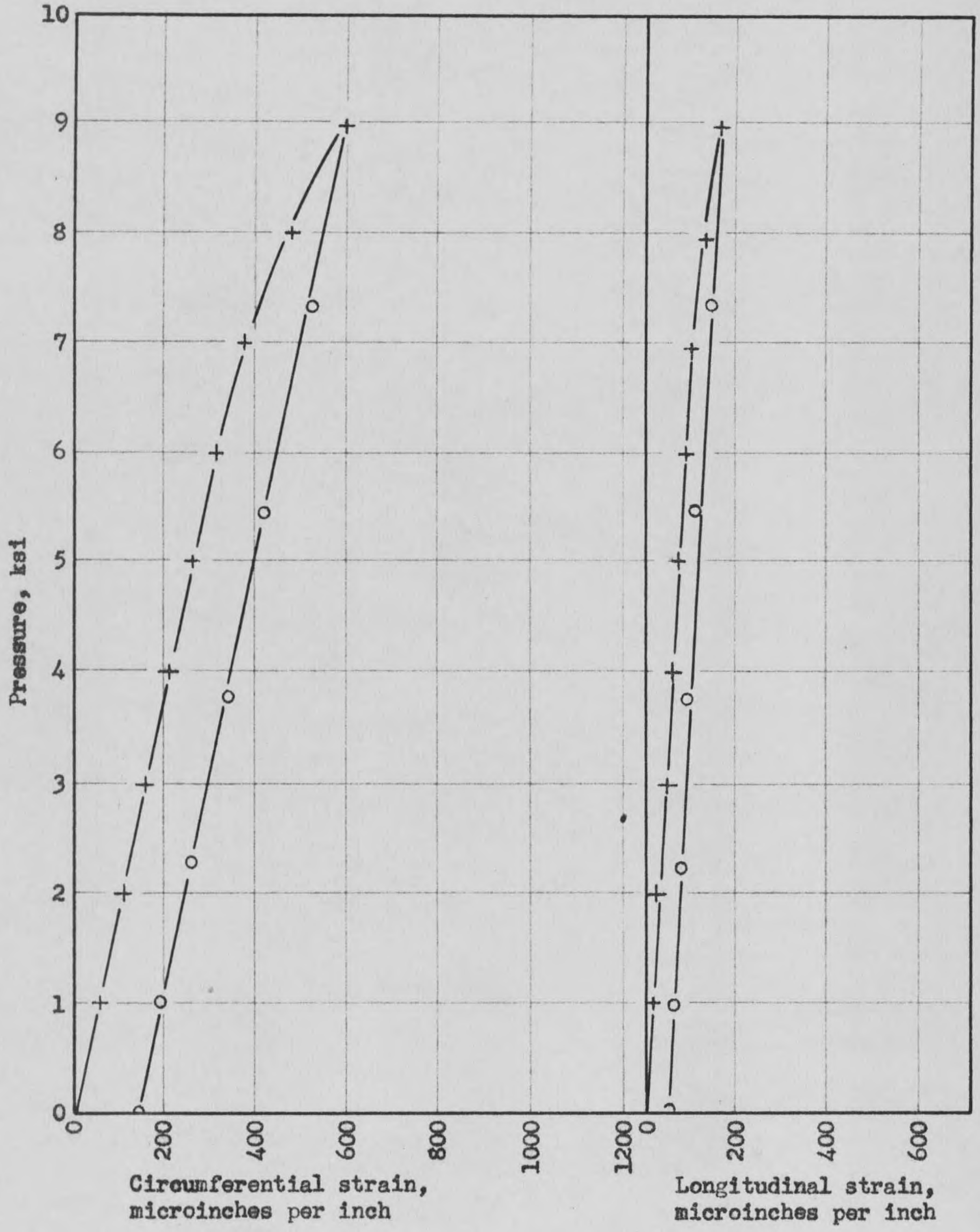

Fig. 31 - Pressure vs strain at section G, test #2.

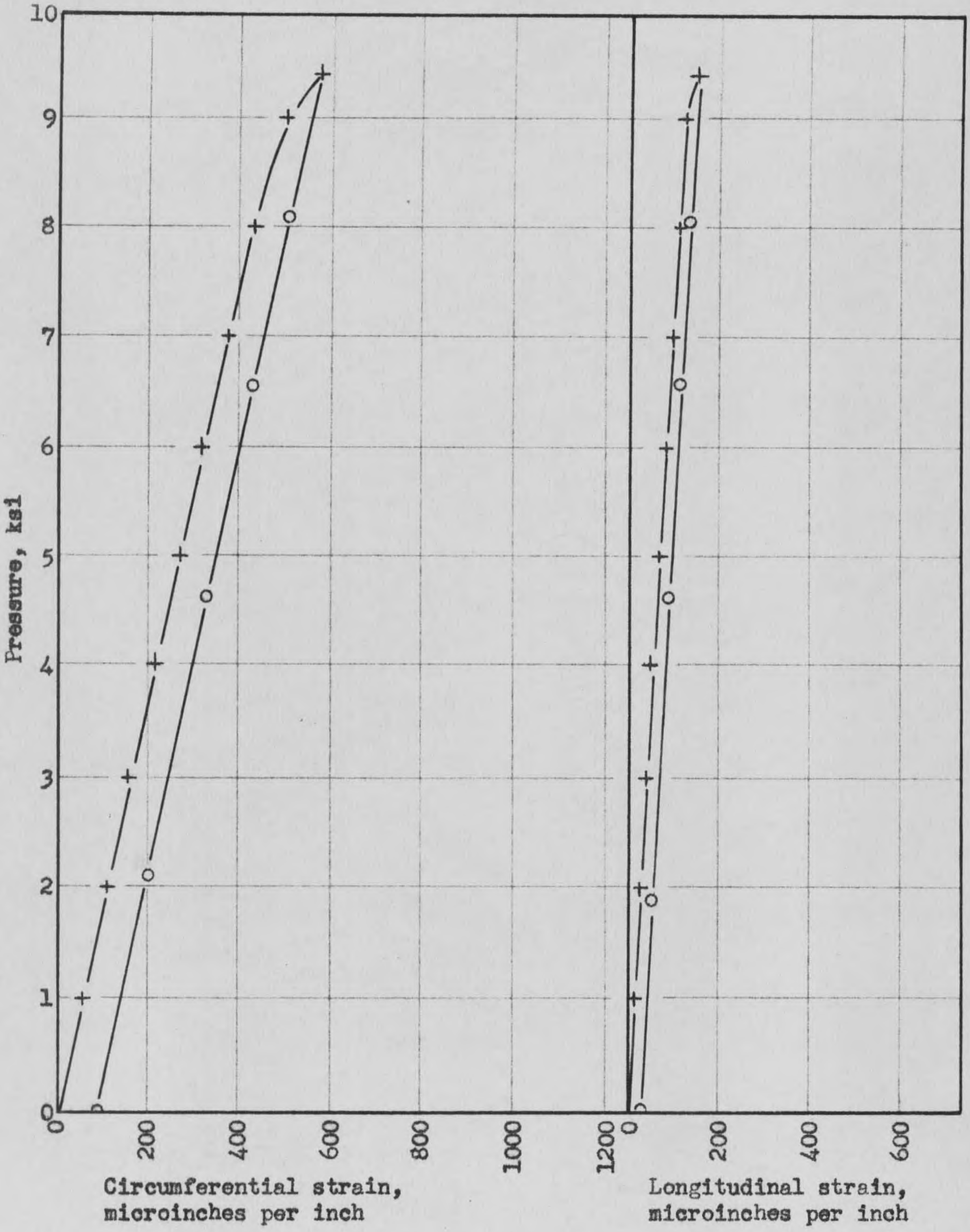

Fig. 32 - Pressure vs strain at section G, test #3.

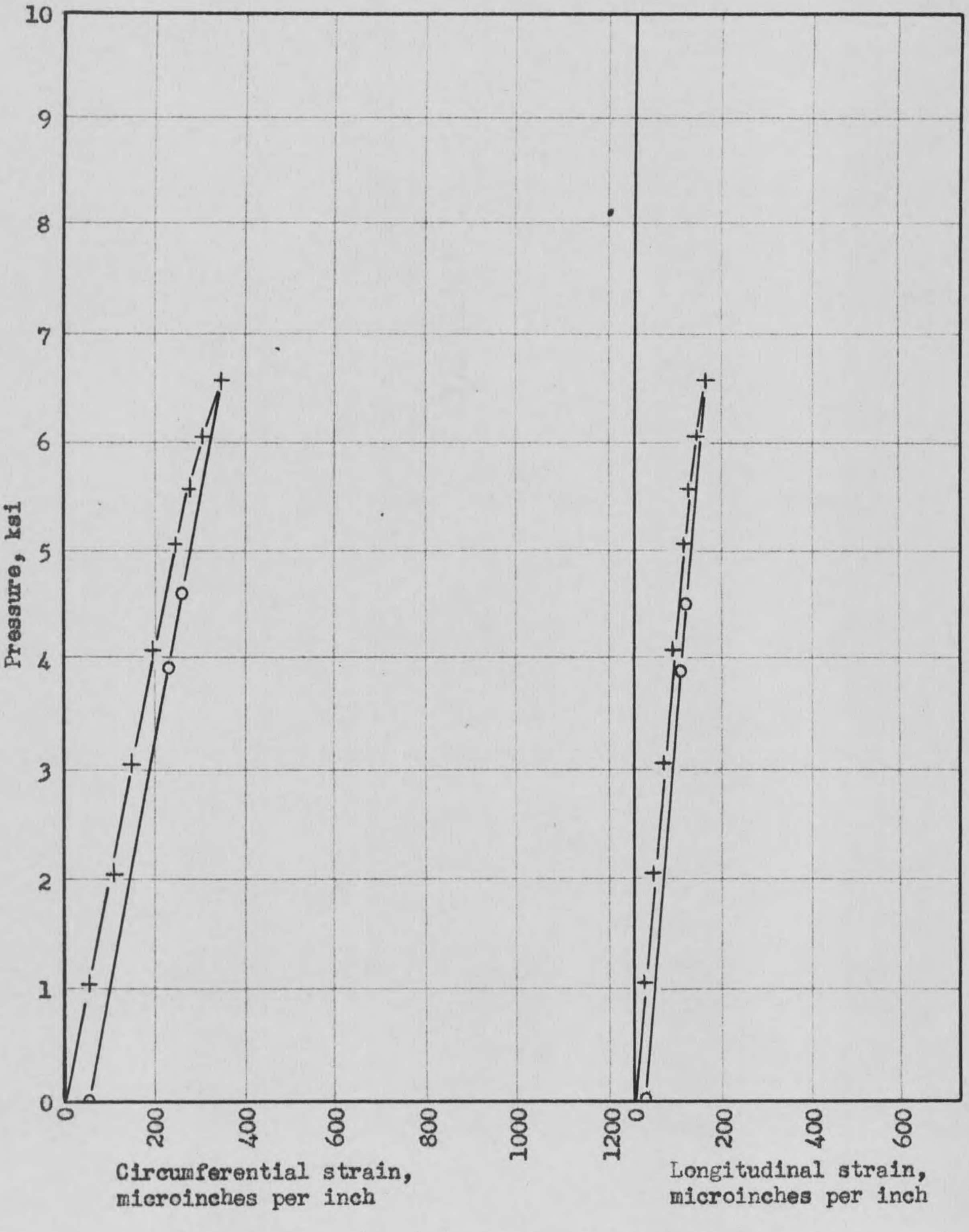

Fig. 33 - Pressure vs strain at section H, test #1.

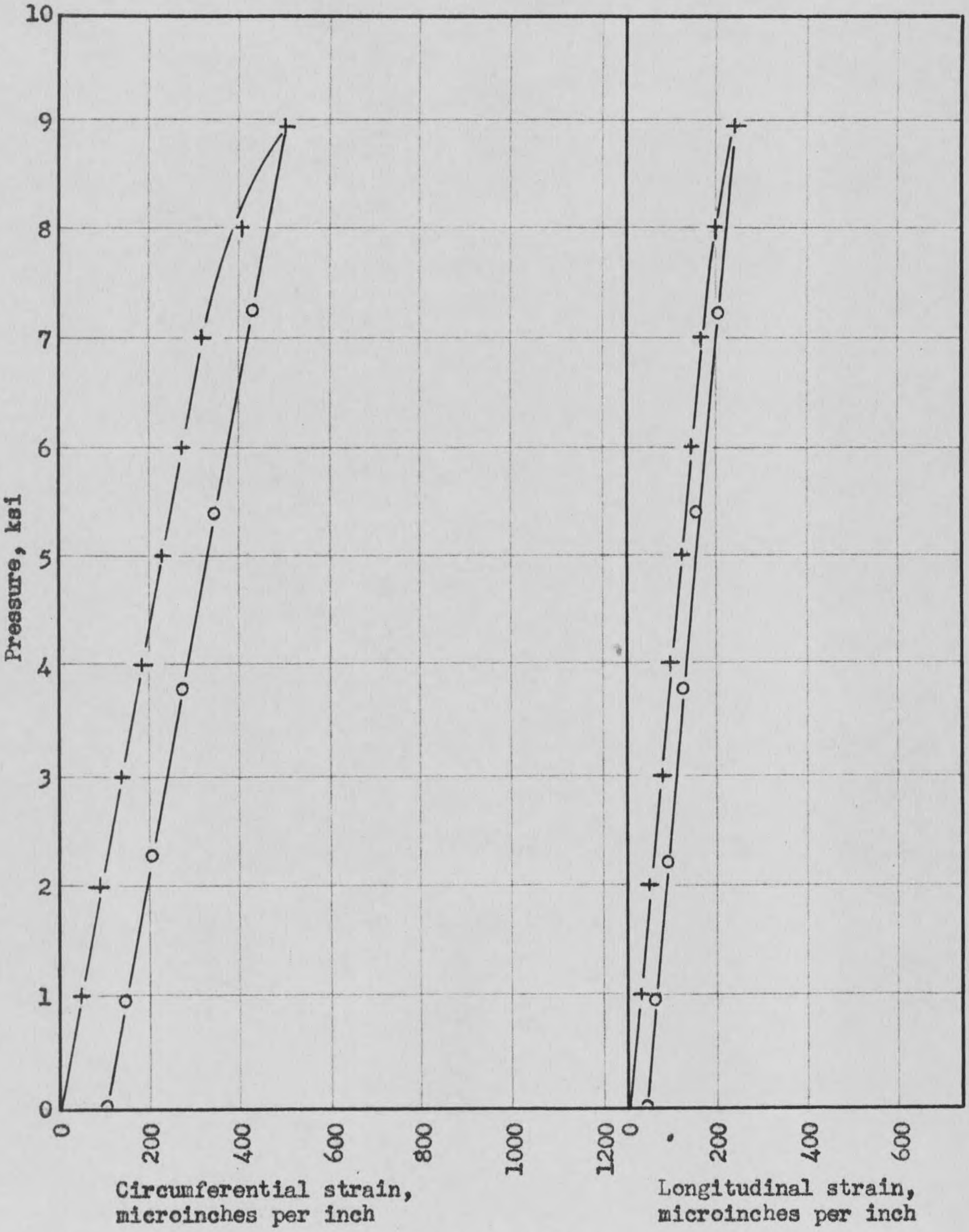

Fig. 34 - Pressure vs strain at section H, test #2.

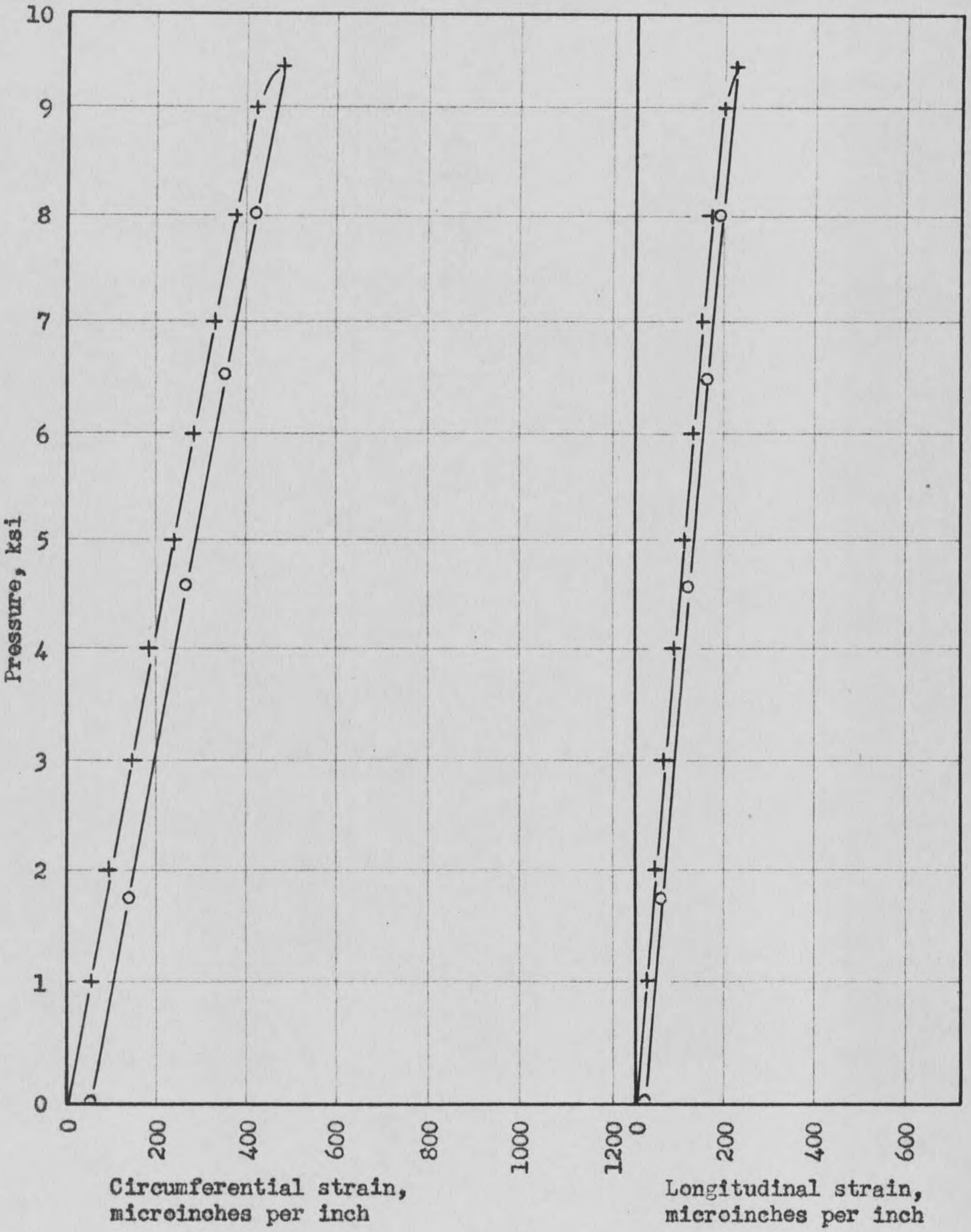

Fig. 35 - Pressure vs strain at section H, test #3.

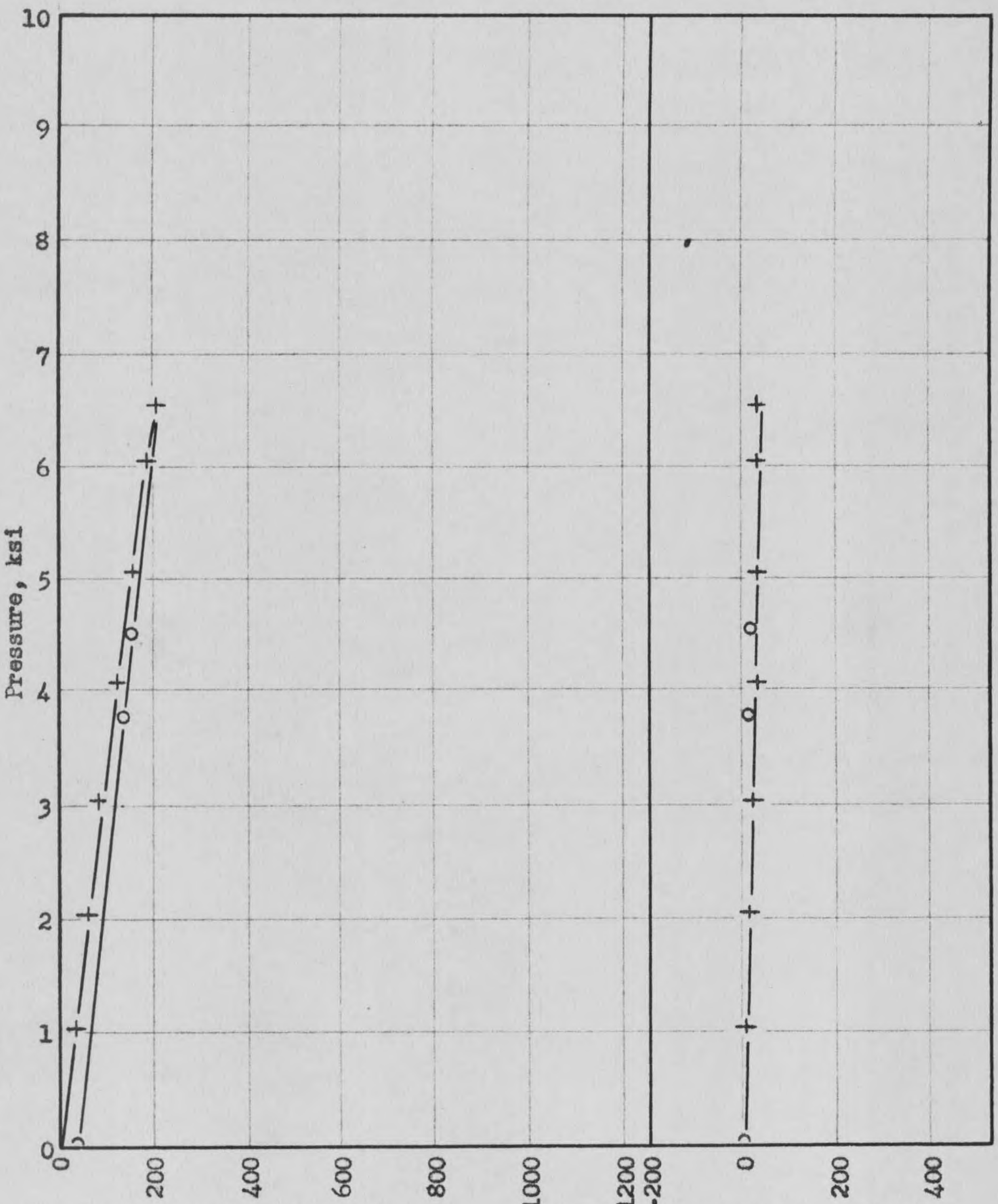

Fig. 36 - Pressure vs strain at section I, test #1.

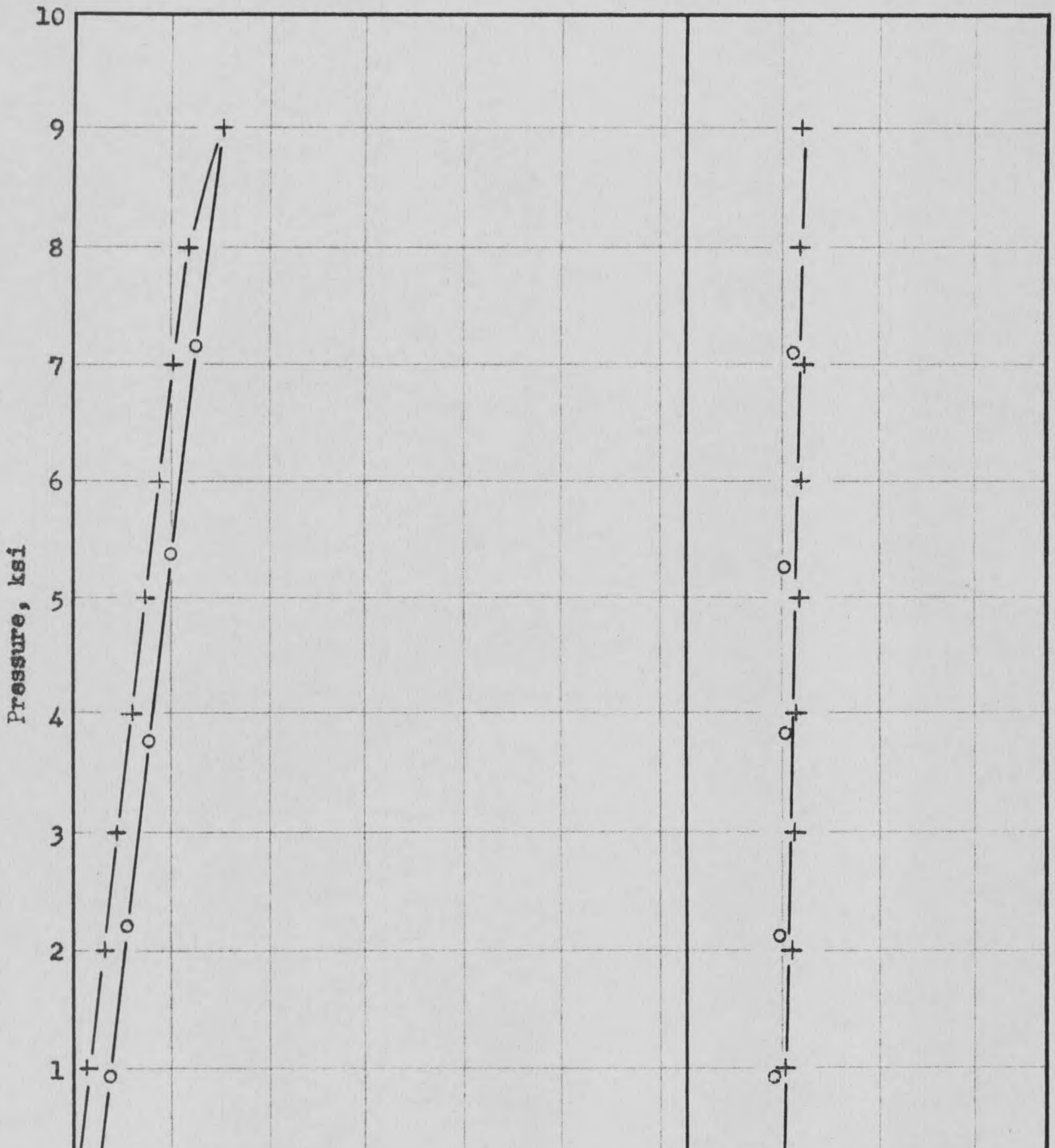

Fig. 37 - Pressure vs strain at section I, test #2.

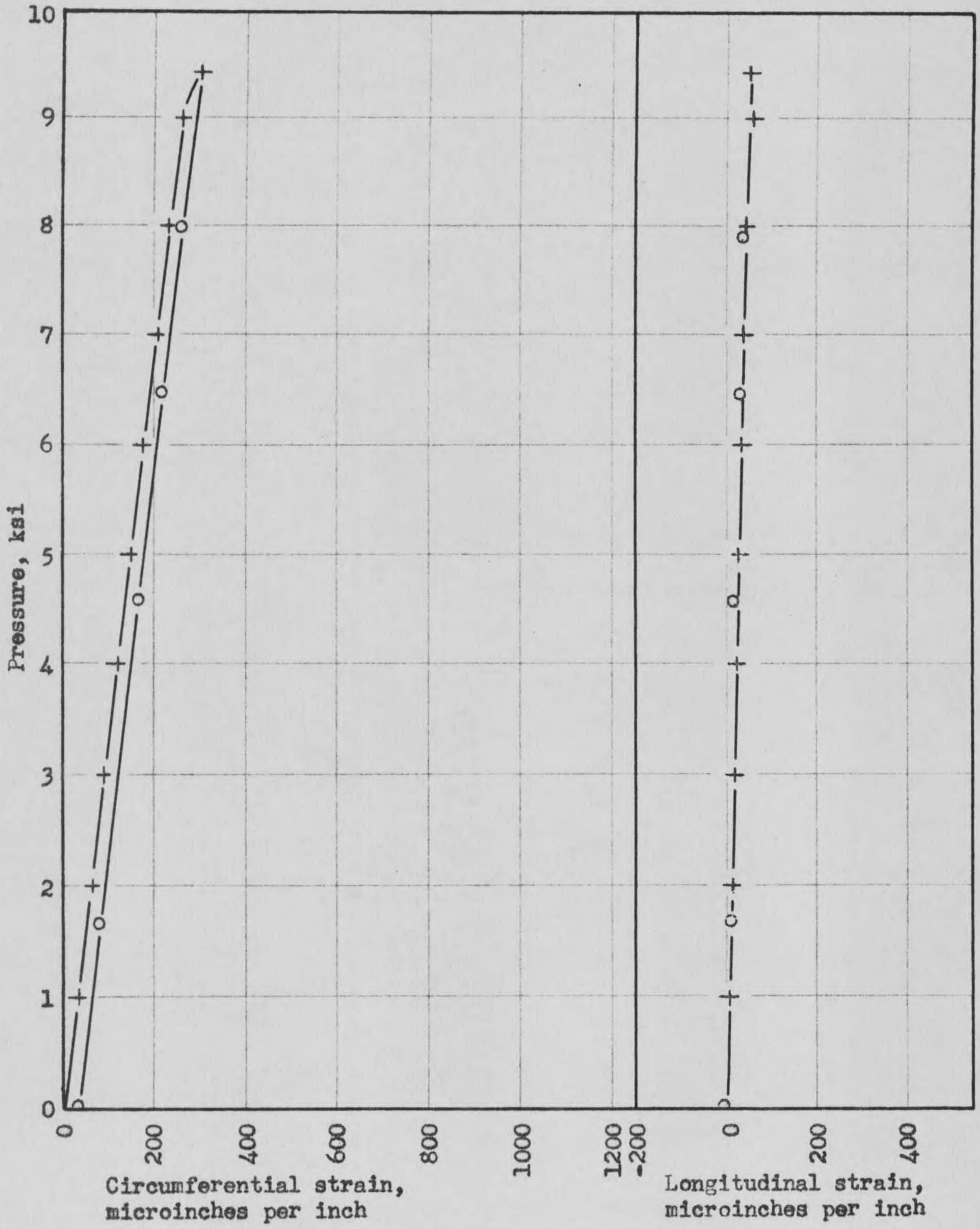

Fig. 38 - Pressure vs strain at section I, test #3.

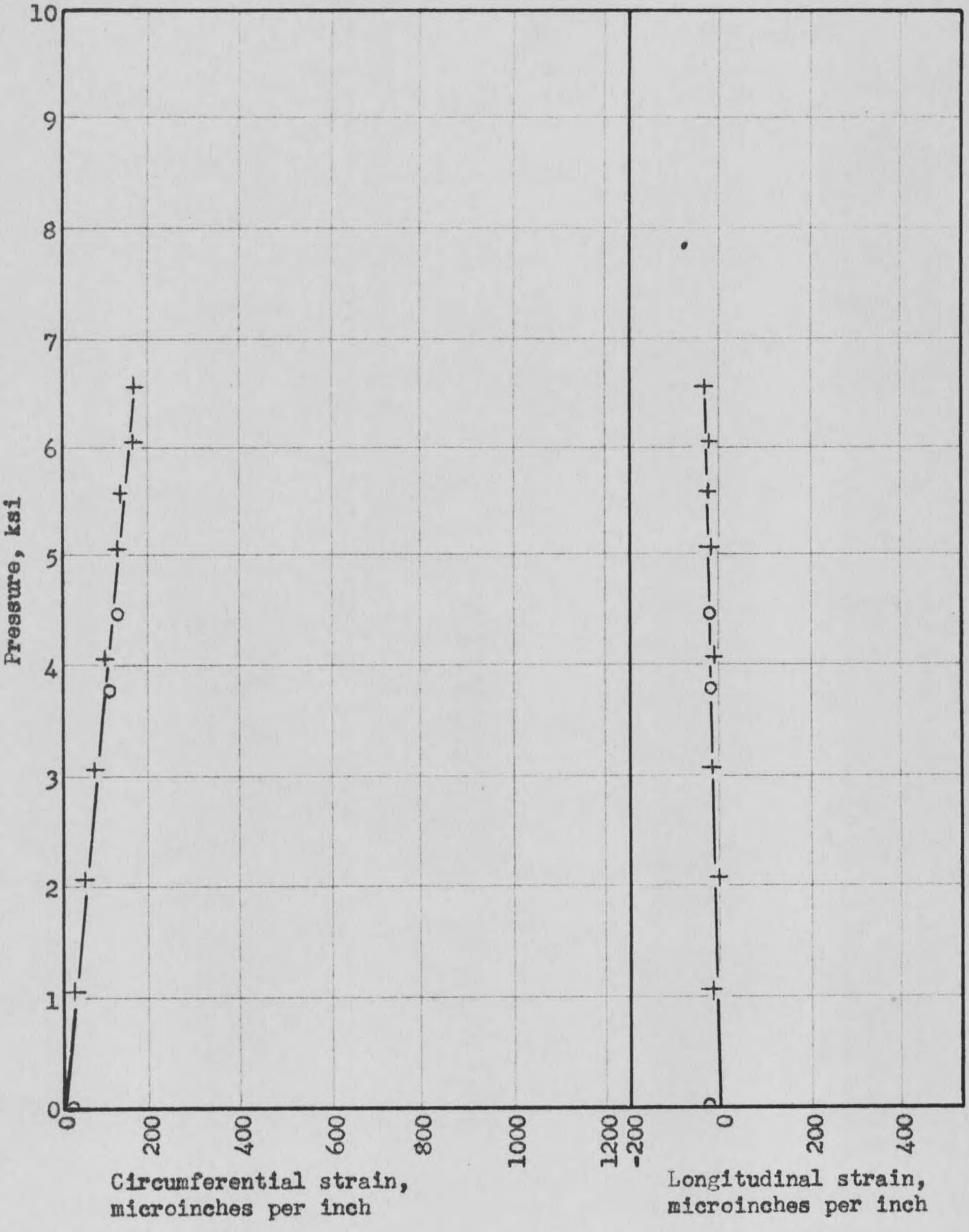

Fig. 39 - Pressure vs strain at section J, test #1.

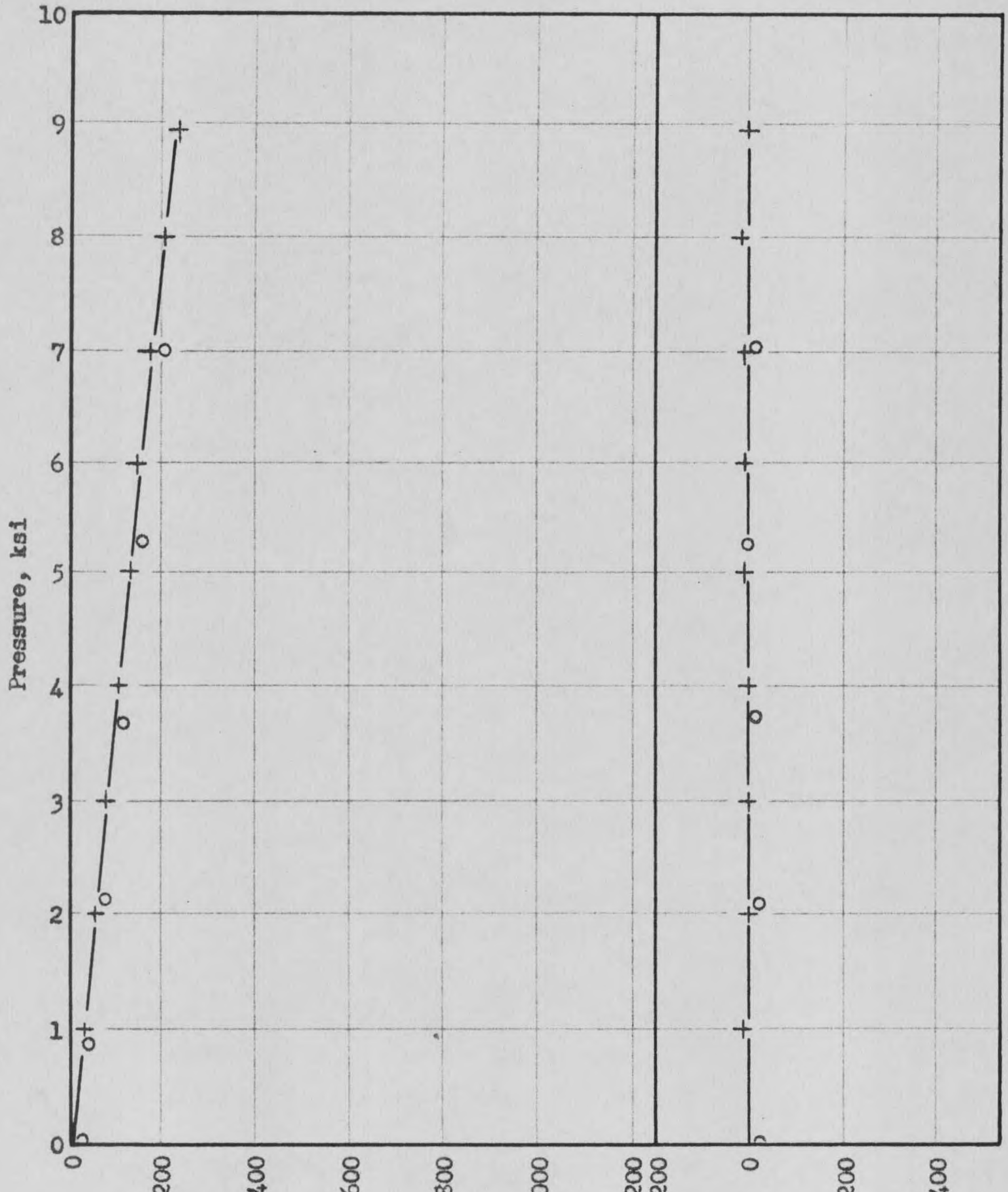

Fig. 40 - Pressure vs strain at section J, test #2.

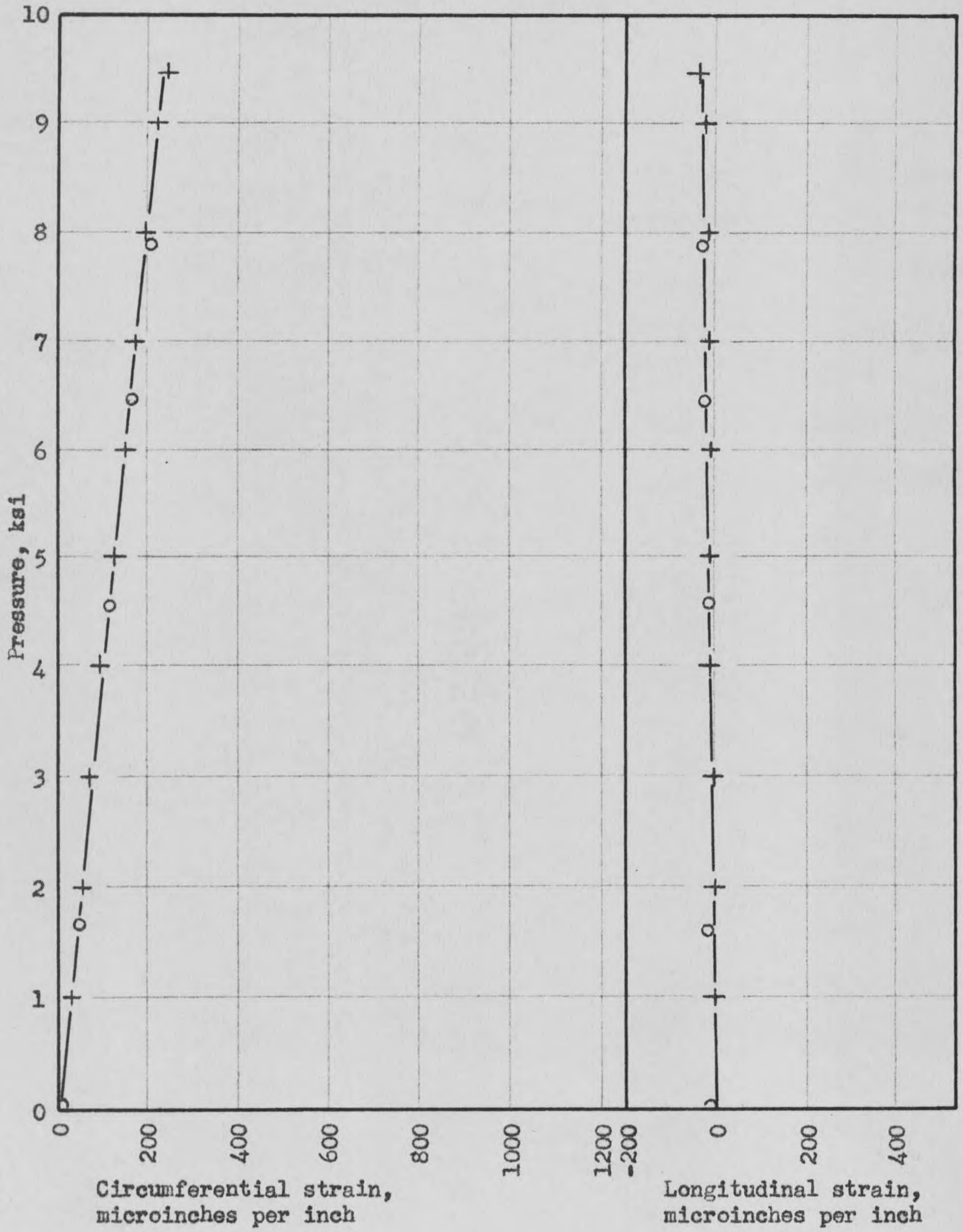

Fig. 41 - Pressure vs strain at section J, test #3.

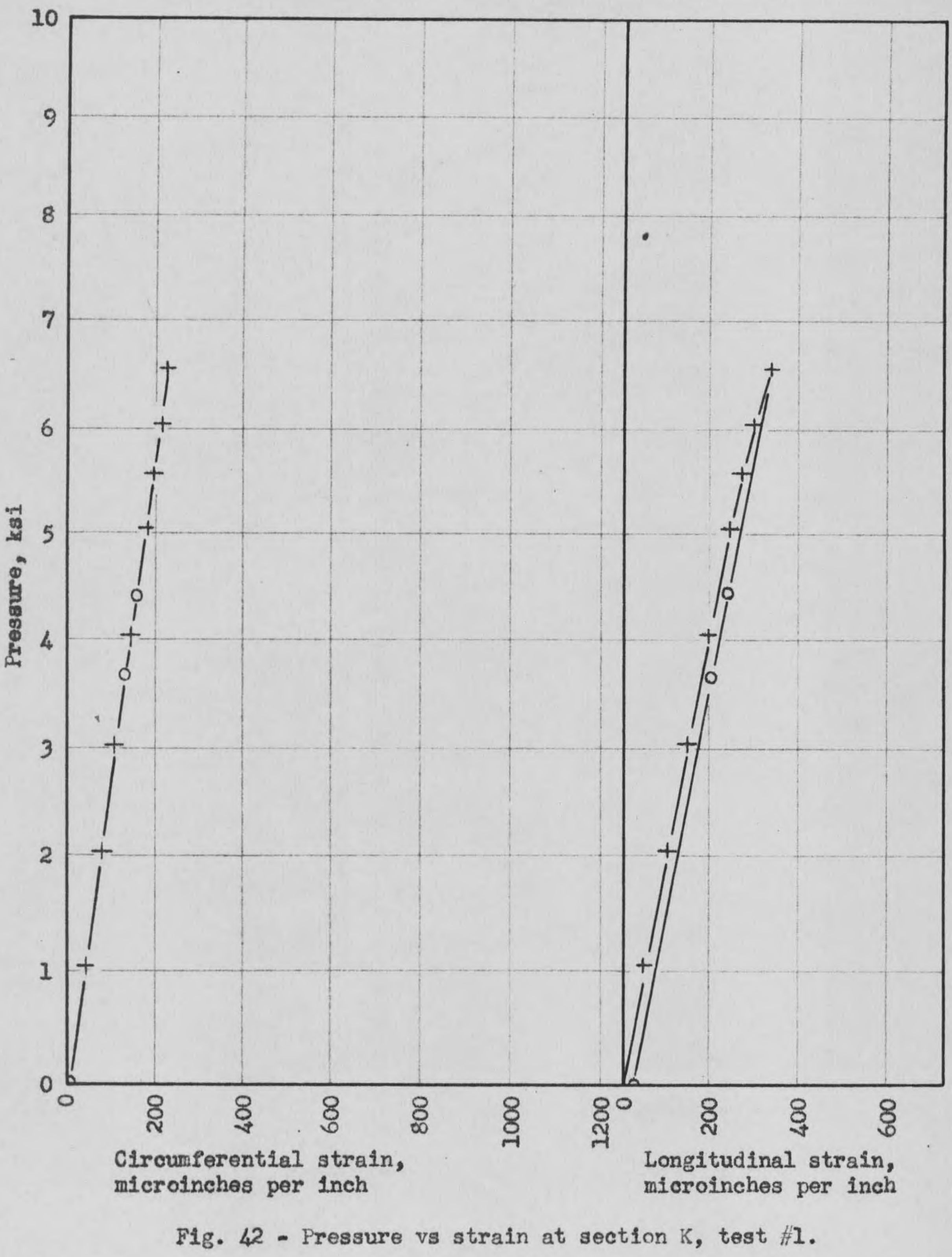

Fig. 42 - Pressure vs strain at section K, test #1.

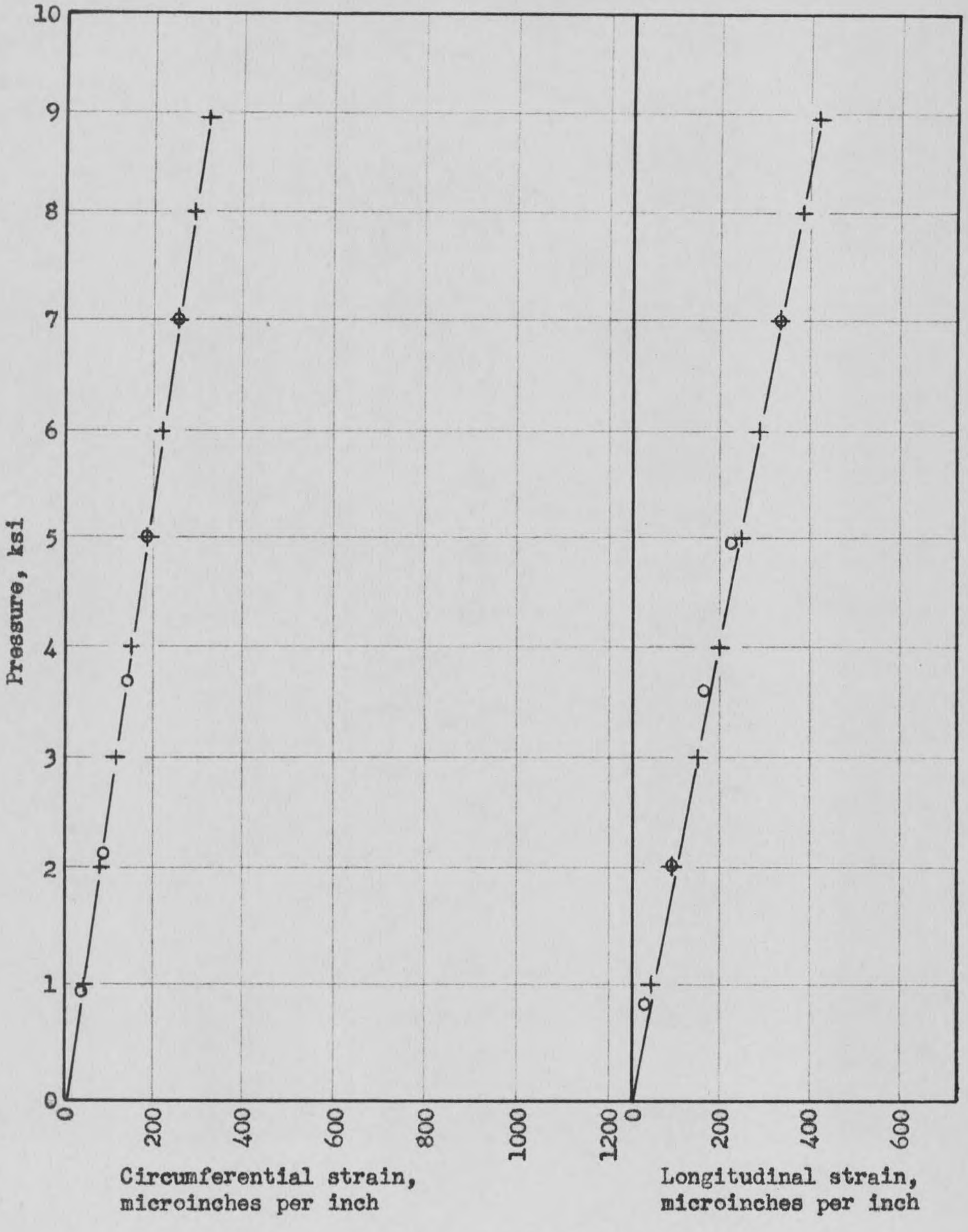

Fig. 43 - Pressure vs strain at section K, test #2.

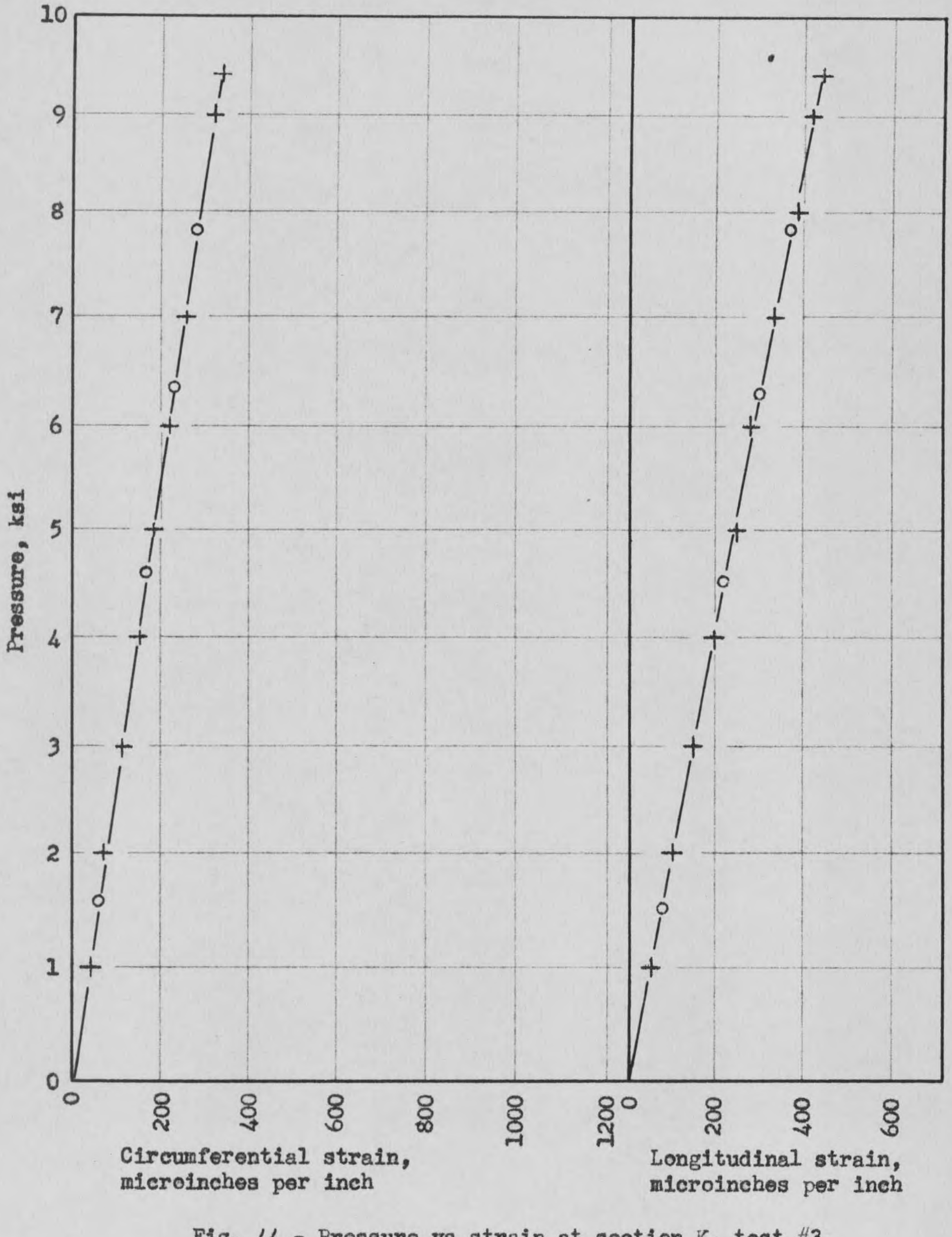

Fig. 44 - Pressure vs strain at section K, test #3.