# TWO-DIMENSIONAL PHOTOELASTIC ANALYSIS OF THE SURFACE STRESSES OF A BOLT IN DOUBLE SHEAR

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## CHAPTER I

## INTRODUCTION

The large factor that weight plays in the world of science, especially in the aerodynamic field, gives it a key role in the development of supersonic airplanes, missiles, rockets, and satellites. It follows, therefore, that if the weight of a joint can be reduced, the weight of the entire plane can be reduced and the gravity forces acting upon the aircraft will be reduced. Gravity forces affect an aircraft during its entire flight. A lesser gravity force because of less weight would enable greater acceleration to be attained per pound of fuel. The weight formerly attributed to the joints could be taken up by a larger pay load in the form of instruments, guns, or recording devices.

It has been estimated that 50 per cent of the cost of the construction of an all metal airplane frame can be attributed to connecting its various parts (1). Therefore, the design of a more effecient joint would be very economical.

At the present time the load on a bolted joint is considered to be evenly distributed across its length. This investigation was undertaken to determine the actual stress distribution, which could possibly lead to a more efficient design of a bolted joint.

A loading fixture was designed and the details of this design are included so that the reader may better understand the reasoning behind such a design.

A two dimensional photoelastic study is in itself a limiting one. However, enough valuable information can be obtained to enable a much more detailed three dimensional investigation to be made.

This investigation, even though limiting, is thought to have aided future investigators by laying the groundwork for a more detailed study.

## CHAPTER II

## THEORY OF THE STRESSES IN A JOINT

Present day methods of calculating the stresses in a lap joint are usually confined to determining the strength of the bolt in shear, the tension in the plates between the rivet holes, or the strength in bearing for the bolts on the plates (2).

The failure of a joint in tension occurs by the tearing of the plate at the point of least area (Figure 1). The stress at this point is given by:

$$\sigma_T = \frac{P}{A},$$

where, $\sigma_T$ = average stress in tension,
$P$ = force on plate,
$A$ = cross sectional area (thickness multiplied by distance across aa plus distance bb).

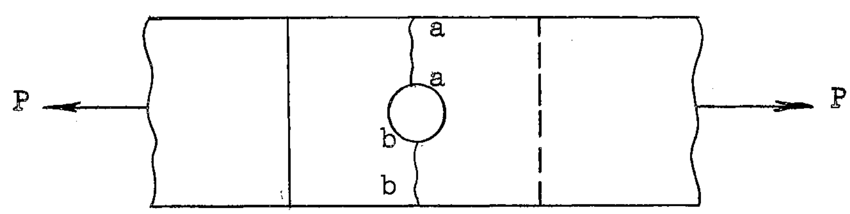



Figure 1. Failure of a Joint in Tension

The failure of a joint in shear occurs when the bolt shears at the juncture of the two plates (Figure 2). The shearing stress is assumed to be uniformly distributed over the cross section of the bolt (3). The average stress in shear is given by:

$$\tau_S = \frac{P}{A},$$

where, $\tau_S$ = shearing stress in the bolt,
$P$ = force on bolt,
$A$ = cross sectional area of bolt.

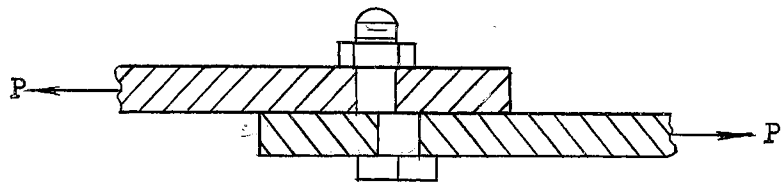



Figure 2. Failure of a Joint in Shear

The failure of a joint in bearing occurs if a bolt crushes the material of the plate against which it bears or if the bolt itself is deformed by the plate acting on it (Figure 3) (3).

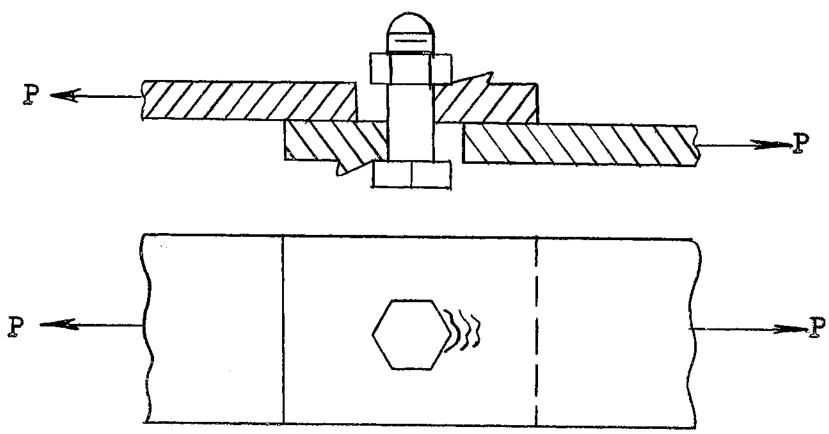


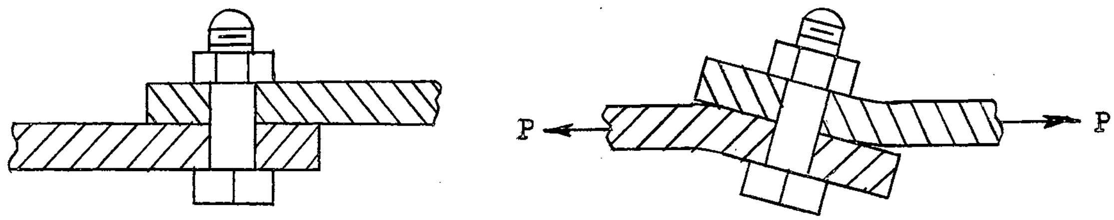

Figure 3. Failure of a Joint in Bearing

The stress distribution in bearing failure is very complicated. In actual practice, this stress distribution is approximated on the basis of an average bearing stress acting over the projected area of the bolt's shank onto a plate (3). This average stress is given by:

$$\sigma_B = \frac{P}{td},$$

where, $\sigma_B$ = average bearing stress,
$P$ = load on plates,
$t$ = thickness of the plate,
$d$ = diameter of bolt.

In a single shear joint the stress situation is further complicated by the effects of eccentricity. The applied load tends to distort the joint so that the loads on both plates have the same line of action. This situation is shown in Figure 4.




a. Unstressed Joint

b. Stressed Joint

Figure 4. Eccentric Load on a Joint

With the eccentric loads a bending moment is incurred on the bolt. The magnitude of the bending moment is given by:

$$M = \frac{Pt}{2},$$

where, $M$ = bending moment,
$P$ = applied force,
$t$ = total thickness of the connected parts.

The stress on the joint is evaluated from the bending moment by:

$$S = \frac{Mc}{I},$$

where, $S$ = stress due to bending,
$M$ = bending moment,
$\frac{I}{c}$ = section modulus of the bolt.

This equation is seldom used in the design of a joint for several reasons:

1. The actual load distribution is unknown.

2. The actual deformation of the bolt is unknown.

3. Some friction is always present between the plates. This friction removes some of the load from the bolt. The magnitude of this friction is

very difficult to determine. It depends on the surfaces of contact, the amount of tensile load on the bolt and whether there is lubrication present.

Since there are so many unknown factors that influence the load on a joint the designer usually accounts for them by introducing a factor of safety.

The effect of friction between the plates is usually ignored and the joint is designed as though there is no friction present (4). Such a procedure results in a joint much stronger than calculations indicate. In effect, the use of this procedure results in a "built in" factor of safety.

## CHAPTER III

## DESCRIPTION OF THE APPARATUS

### The Loading Fixture

The design of a fixture in order that actual loading conditions could be simulated was probably the most difficult portion of this investigation. To do this several factors had to be taken into consideration (Figure 5).

1. The fixture had to be rigid enough to withstand the loads without any significant distortion occuring in the fixture. This was accomplished by making the main body of the fixture from mild steel and of sufficient size to withstand loads much greater than those applied in this investigation.

2. Since bakelite has such a low modulus of elasticity (615,000 psi), the tendency of the test members to buckle was a prime consideration. The main objective of this investigation was to determine the stress distribution in the bolt itself, therefore it was unnecessary to expose a great portion of the plates to polarized light. The mild steel bars (Parts 2 and 3, Figure 6) were placed in their respective positions to prevent buckling of the plates. These bars were made of such size that the bakelite plates would fit snugly between them without applying any external forces on the bakelite plates.

Buckling occurs in cast iron members with a slenderness (l/r) ratio greater than 6 (5). Because of the low modulus of elasticity of bakelite this figure would be smaller. In the design used for this investigation the slenderness ratio of the joint was 4.2.

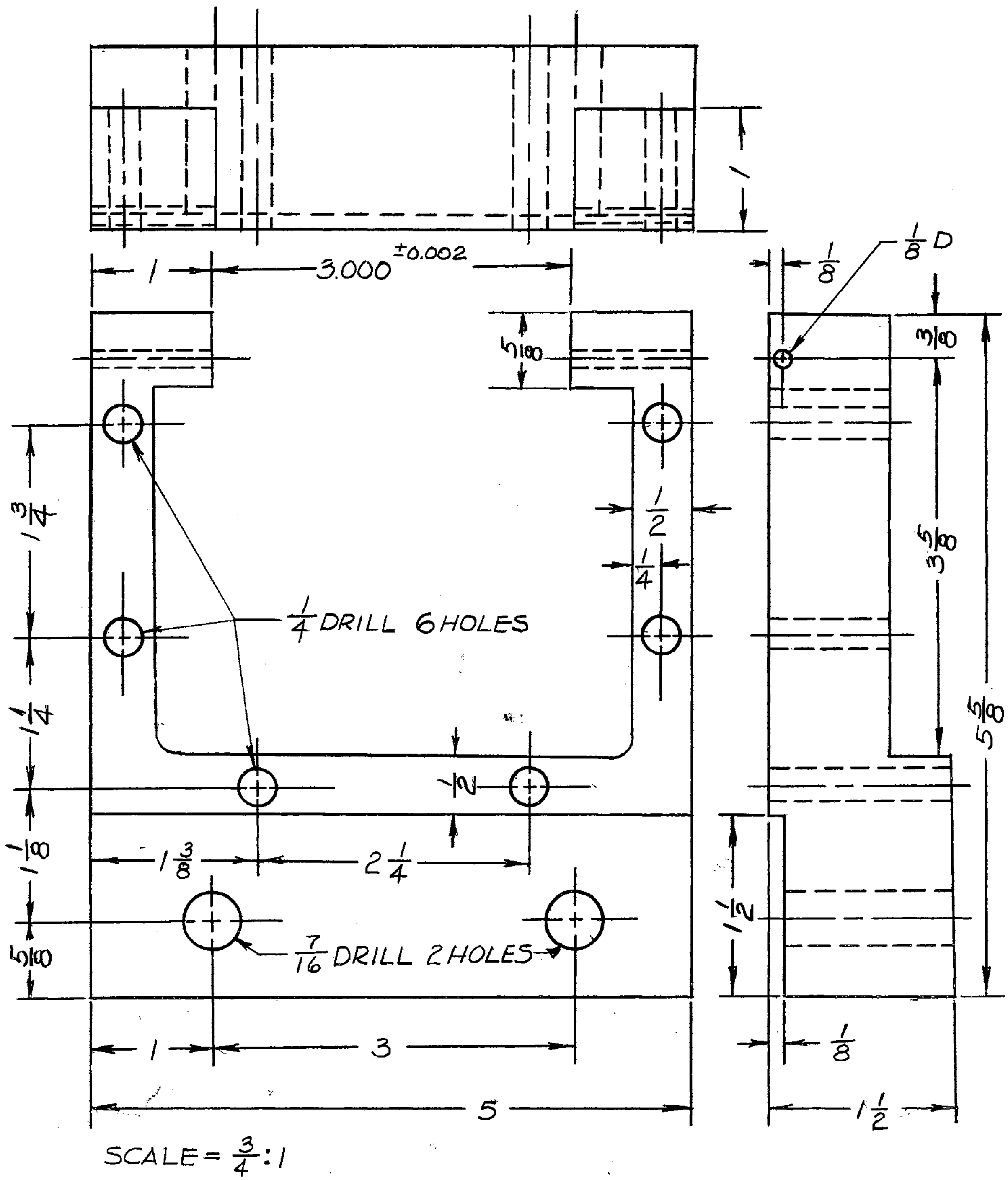



Figure 5. Detail Drawing of the Main Body of the Loading Fixture

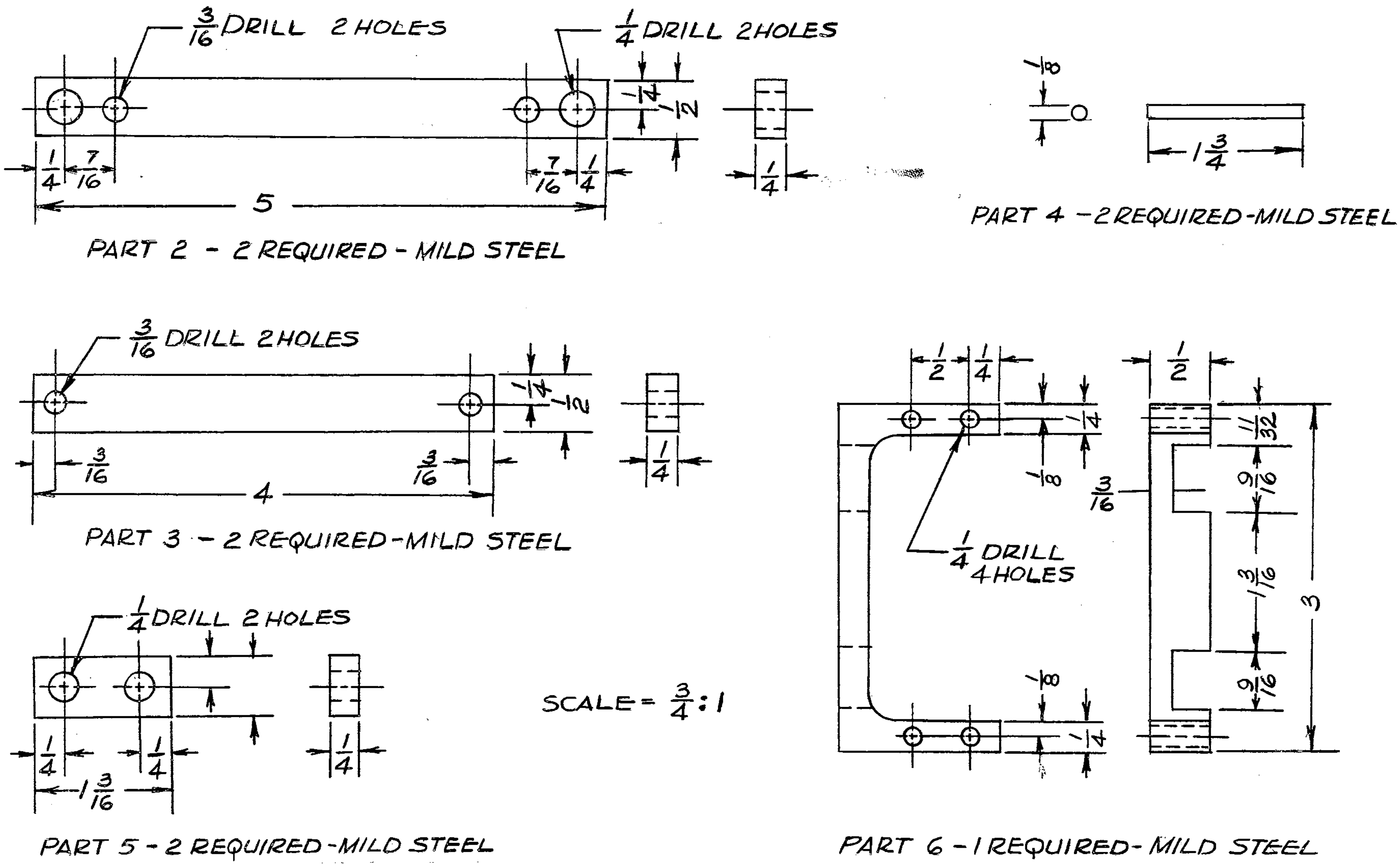



Figure 6. Detail Drawing of the Component Parts of the Fixture

3. In a joint such as the one used in this investigation, the portion of the plate not directly loaded plays an important role in the stress pattern of the bolt. These portions of the plates, since there is no clearance between the bolt and the plate, offer a resisting force to the bending action of the bolt. This was simulated in the design of the fixture by inserting pins (Part 4) in the upper section of the fixture. This resisting force was simulated in the central plates by connecting these plates with a rigid steel support (Part 6) of such design as to keep the central portion of the joint clear of obstructions so polarized light could be passed through it.

4. The two clamps (Part 5) served a dual purpose. They allowed the lower portion of the joint to be aligned with the front, or reference, surface of the fixture and also enabled proper alignment with respect to the load. These clamps also aided in reducing the probability of buckling.

## The Joint

The construction of the joint had to be such that an actual double shear joint was simulated. In a double shear joint of this type the head and nut ends affect the stress distribution in the bolt by offering a resisting force to the bending action of the bolt. This force is applied at the point of contact of the head with the plate (Figure 7). In this investigation the head end was simulated by machining a rectangular head on Part A.

The resisting force due to the nut was accounted for in the following manner:

Since the required ratio of bolt diameter to plate width was 1:1, parts

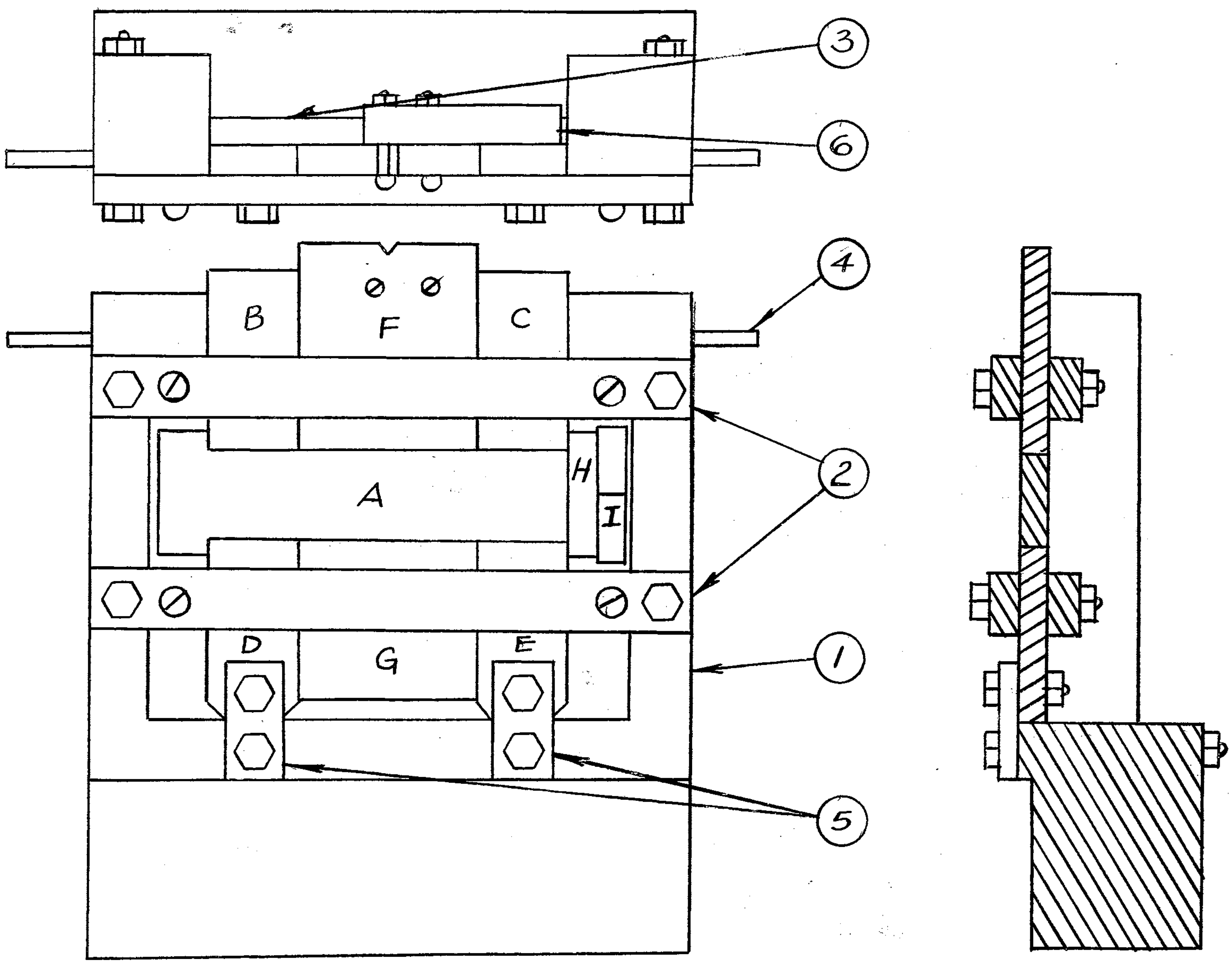



Figure 7. Assembly Drawing of the Fixture and Models

B, C, D and E were 0.750 inches wide (Figure 8). Parts F and G were made twice as wide, 1.500 inches, as the bolt diameter and the load was applied on the center line of Part F. This procedure achieved a uniformly distributed load in each half of Part F. The entire load in Part F was transmitted through the joint so that one half the applied load was carried by each of the Parts D and E. This distributed load simulated the actual loading of a joint with a force of P/2 on each plate. The effect of the nut was duplicated by the entire right hand half of the set-up.

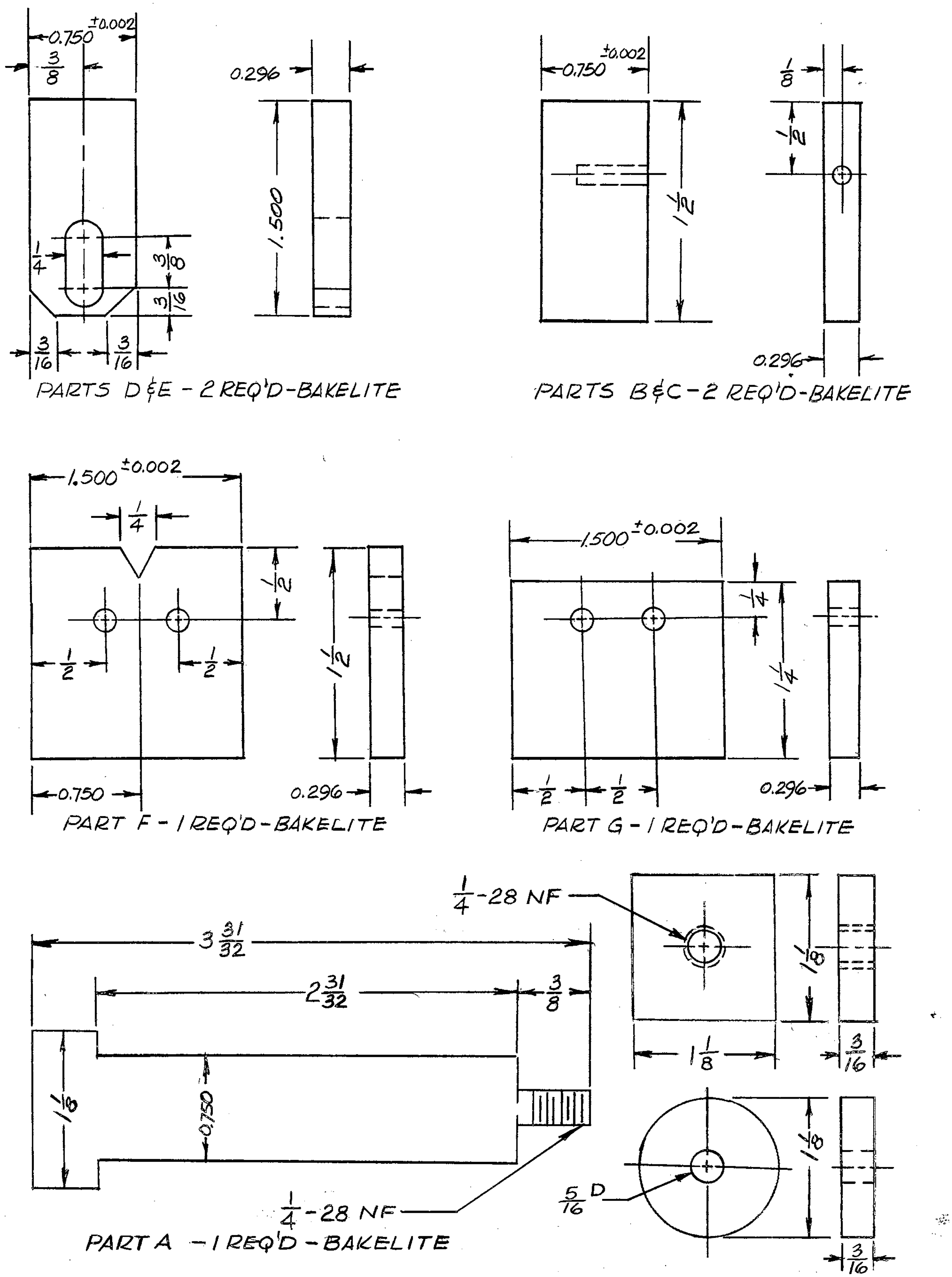



Figure 8. Detail Drawings of Test Specimens

## CHAPTER IV

## EXPERIMENTAL PROCEDURES AND RESULTS

The test specimens were cut, using a Do-All band saw, from a single sheet of Bakelite BT-61-893. During this rough cutting process extreme care was taken to prevent any machining stresses from being incurred at the edges of the specimens. An excess of 1/8 inch of material was left on all edges for the milling operations.

The faces of the specimens were then rough sanded to within 0.003 inch of the final size. These faces were sanded with a fine emery paper to within 0.001 inch of final size. They were then polished to size with jeweler's rouge.

To perform the various sanding operations a sheet of thick glass was placed beneath the emery paper and the specimens were rubbed over the emery paper. The polishing operation was performed in the same manner using the apparatus shown in Figure 9. This procedure was used to ensure flat surfaces. Micrometers were used in obtaining a uniform thickness. The thicknesses of the specimens tested were all within 0.001 inch of one another and none had more than 0.0005 inch taper of the surfaces.

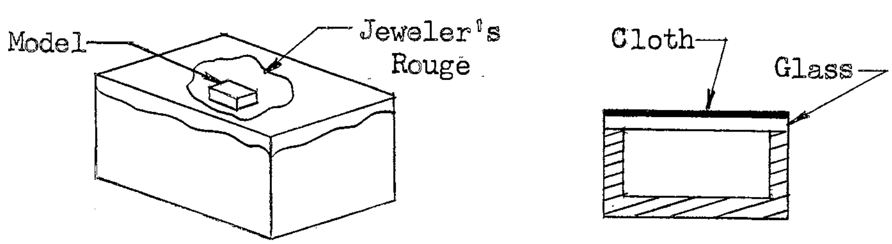



Figure 9. Specimen Polishing Apparatus

The faces of the specimens were covered with masking tape to protect the finish and sent to the Research and Development Laboratory where the edges were milled to size and the other machining operations were performed.

The polishing operation preceded the milling operation because polishing by hand tends to cause the edges to become rounded. These rounded edges were then milled off to produce square corners. Square corners and flat surfaces were of the utmost importance in this investigation, therefore extreme care was a necessity during the polishing and milling operations.

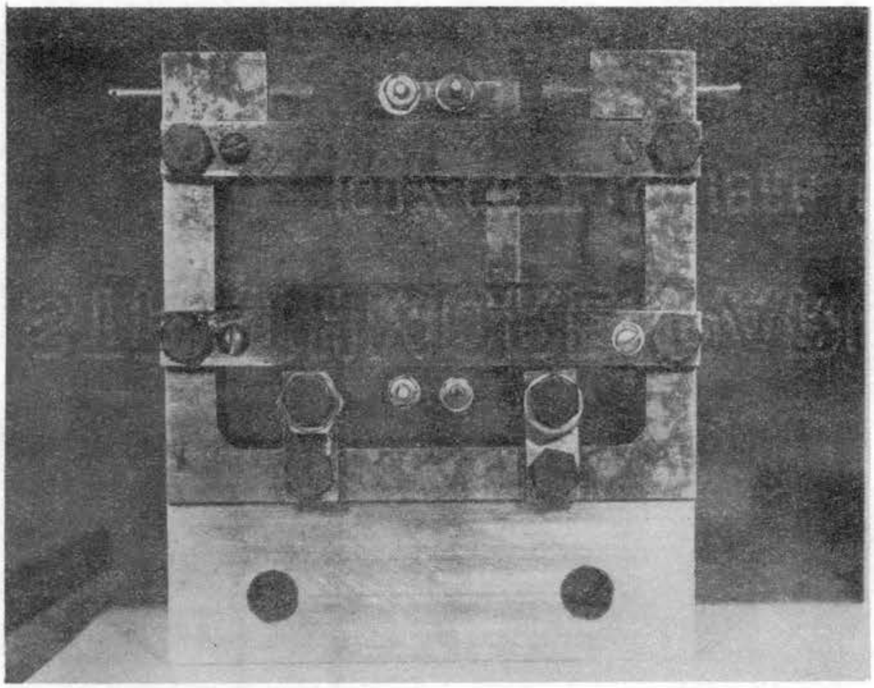
The various test specimens were placed in the loading fixture as shown in Plate I. The self aligning characteristics of the loading fixture were of great importance and aided in the proper placement of the test specimens.

A slight initial tension was induced on the bolt (on the order of 10 inch-pounds torque on the nut at the end of the bolt) to more nearly simulate actual loading conditions.

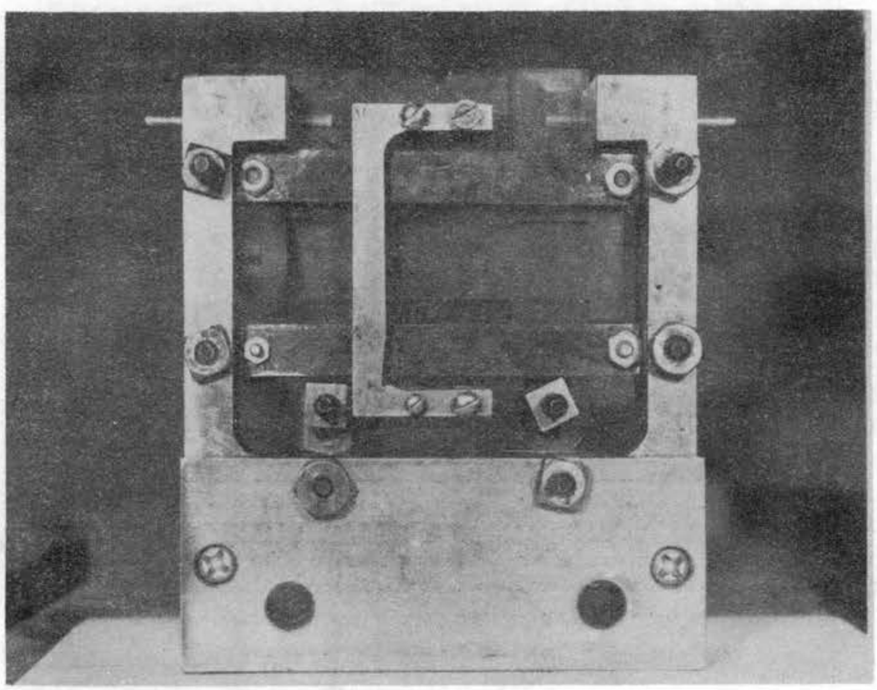
The fixture was placed in the loading frame of the polariscope as shown in Plate II. This loading fixture was leveled with a combination square level to make sure the load would be applied at the center line of the entire apparatus. This also ensured that the load would be carried equally by both the supporting specimens.

The loading lever arm was balanced at no load by shifting the counterweight along the arm. Loads of 11, 16, 21, 26, 31, 35 and 40 pounds were applied at the end of the lever arm producing loads of 73.04, 106.24, 139.44, 172.64, 205.84, 232.48, 265.68 pounds on the model. During these various loading operations monochromatic light was passed through the model with the polariscope set to produce a dark background (Figure 10). Photographs of the stress patterns were taken at the various loads and are shown in

PLATE I. THE FIXTURE AND MODELS

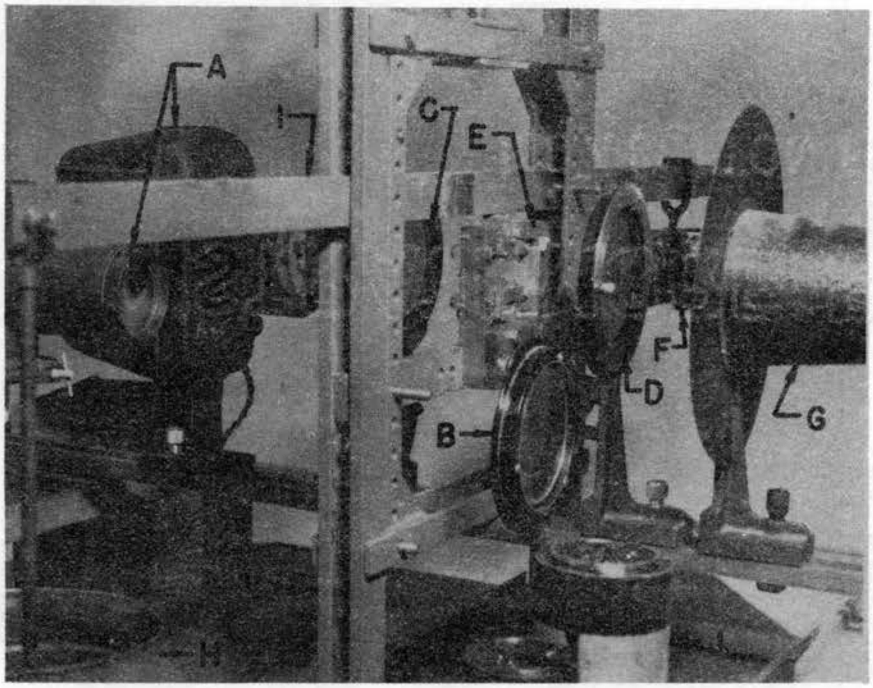


1. Front View



2. Rear View

PLATE II. POLARISCOPE SET-UP




A. Light Sources

B. Quarter Wave Plate

C. Polarizer

D. Analyzer

E. Loading Fixture and Models

F. Counter Weights

G. Telescope

H. Loading Bucket

I. Lever Arm

J. Weights

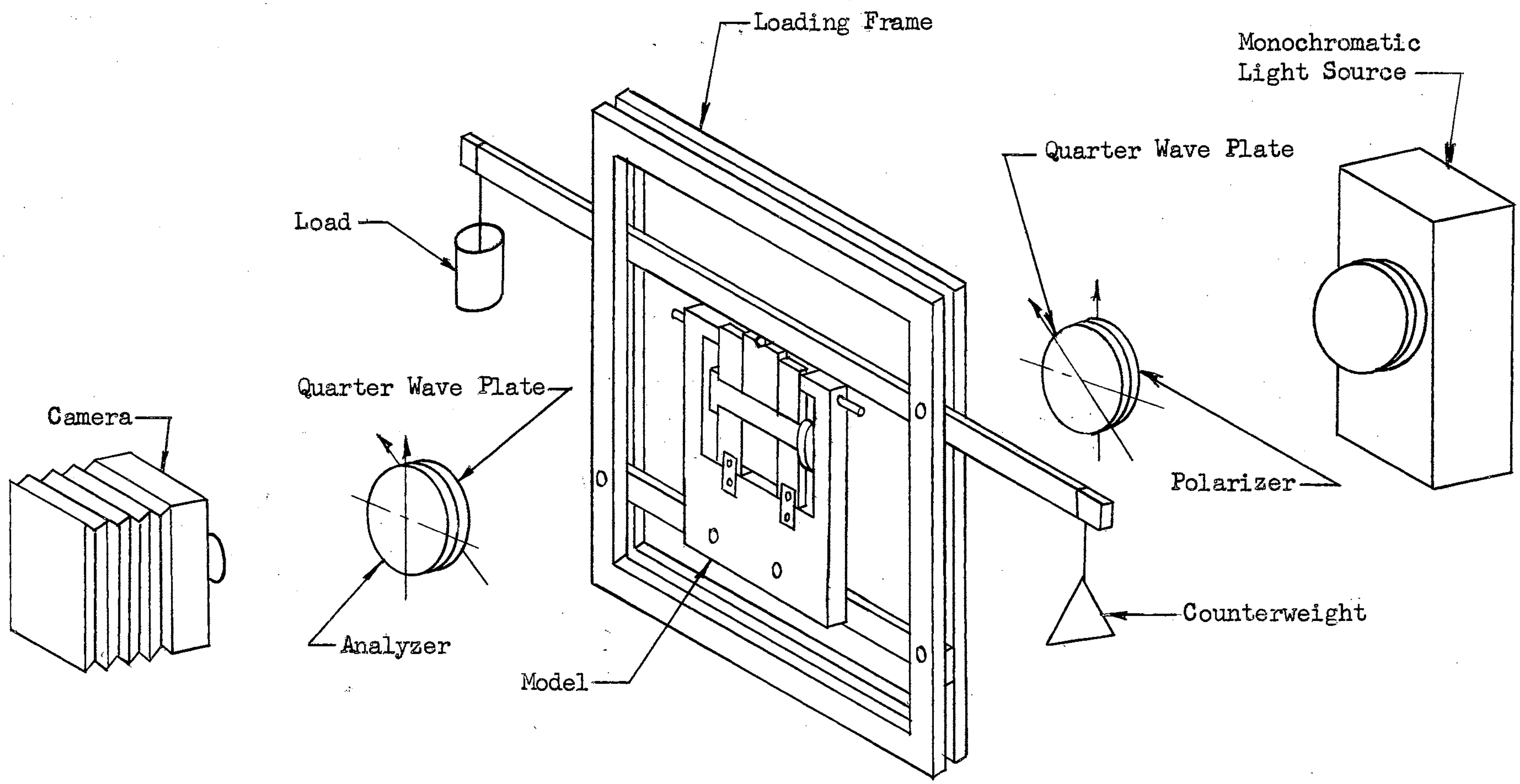



Figure 10. Schematic of Polariscope Set-up to Obtain Stress Patterns

Plates III and IV. The photographs were taken with a dark background.

Upon close examination of the photographs several undesirable features were present:

1. The head of the bolt did not make contact with both the upper and lower plates.

2. The portion of the plates not carrying the direct load did not fit the bolt closely enough. This situation was not too serious since the set-up could have represented a joint with a clearance between the bolt and the plates.

3. A deep scratch was noticed on the face of the bolt. This scratch was formed when the machinist clamped the bolt for the milling operation. The specimen could not be repolished after receiving it from the Research and Development Laboratory due to the probability of rounding the edges, which would have affected the results to a greater extent than the scratch.

The photographs indicate that the loads were uniformly distributed at the surface of the bolt specimen. However, with a slight increase in load the bolt deflected and the loads became concentrated at the points indicated by A and B on the photographs.

The deflection of the bolt was very small. Therefore the head end of the bolt did not induce a stress at the point of contact with the plate. This was an important feature of such a joint. The load of 265.68 pounds caused the point of contact to begin to chip thus forcing such small deflections.

It was necessary to obtain the isoclinics for the bolt. The monochromatic light was replaced by a white light source, the quarter wave plates were removed, and a load was applied until the model became an orange color.

## PLATE III. STRESS PATTERNS FOR TEST 1

1. Stress Pattern for $P = 73.04$ lbs.
2. Stress Pattern for $P = 106.24$ lbs.
3. Stress Pattern for $P = 139.44$ lbs.
4. Stress Pattern for $P = 172.64$ lbs.

PIATE III

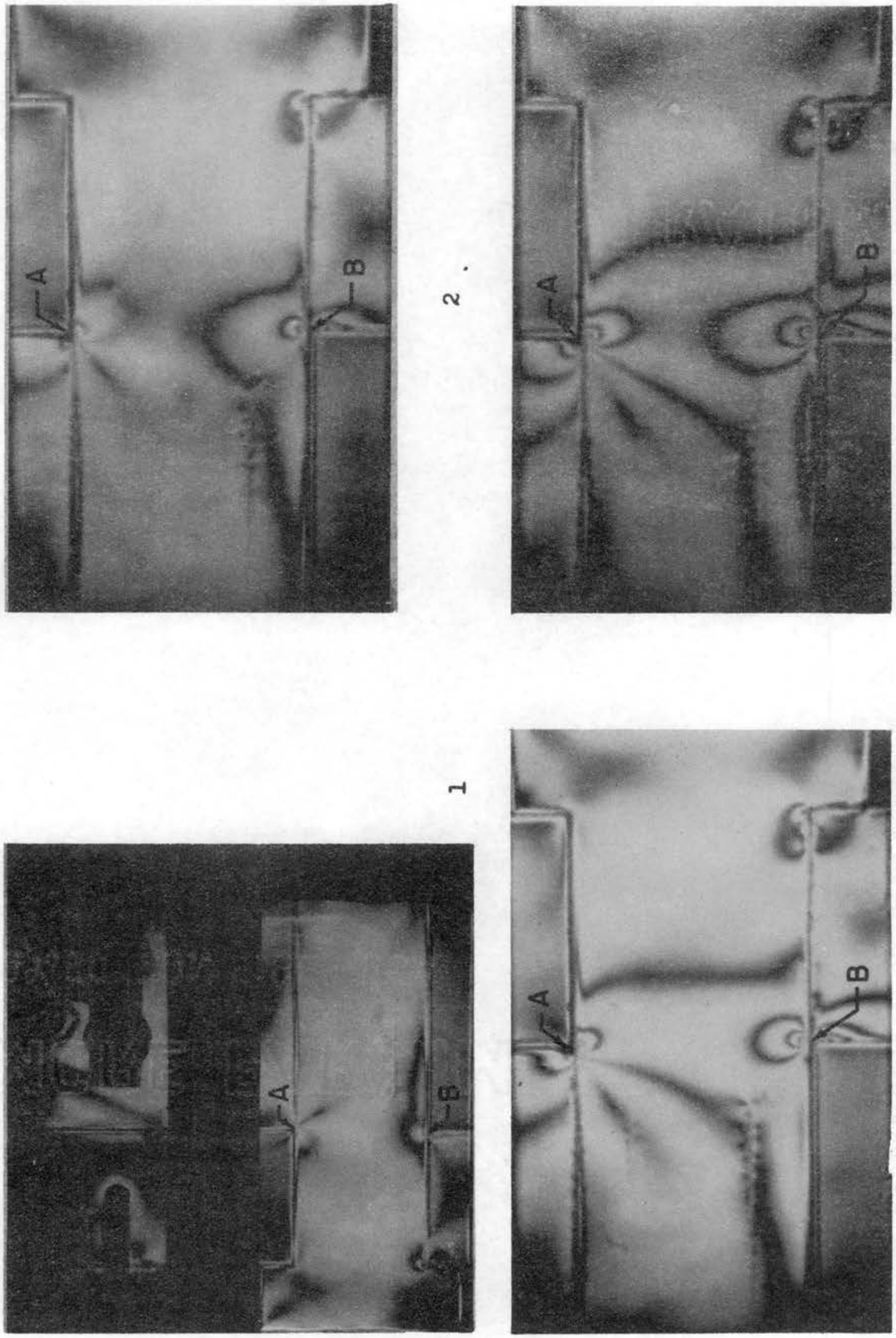


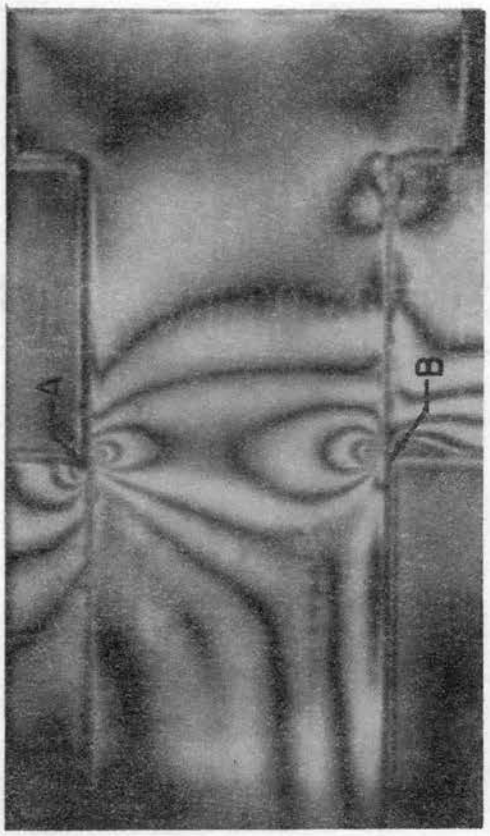
PLATE IV. STRESS PATTERNS FOR TEST 1

1. Stress Pattern for P = 205.84 lbs.

2. Stress Pattern for P = 232.48 lbs.

3. Stress Pattern for P = 265.68 lbs.

PLATE IV

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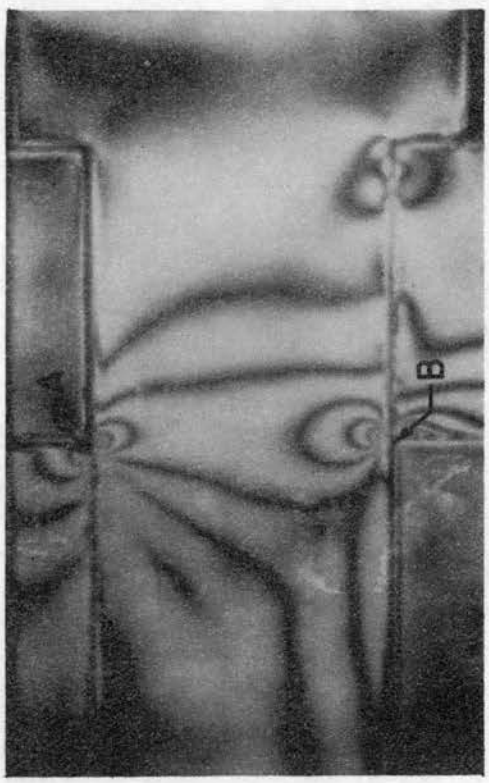


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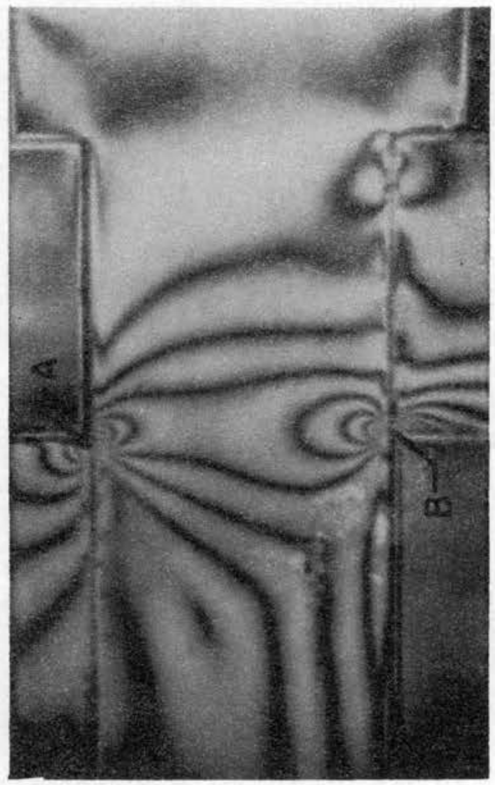


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This was done because isoclinics are more easily distinguishable with an orange background (6).

The polarizer was set at 9° and the analyzer was set at 0° to obtain the 0° isoclinic. The polarizer on the polariscope used in this investigation was out of line therefore it was necessary to set it 9° ahead of the parameter of the desired isoclinic. The 0° isoclinic was traced from the image projected on the tracing box. The polarizer and analyzer were rotated and the isoclinics were traced at 10° intervals from 0° to 80°. These isoclinics are shown in Figure 11.

The stress trajectories are shown in Figure 12. These trajectories are very simular to those of a beam in pure bending (6). The upper boundary of the bolt was known to be in compression and the lower in tension therefore the trajectories parallel to the lower boundary were of the p type and those in the vertical direction were of the q type. The analysis of the joint was stopped at this point because a second test was deemed necessary.

A considerable amount of difficulty arose from the method of load application. This difficulty came mostly from the out-dated design of the polariscope. The method of placing weights in a bucket device at the end of the lever arm was very clumsy and great care had to be taken or the counter-weight would slide along the arm thereby upsetting the balance and resulting in an erroneous value for the load.

A second test was run using a new set of models. The procedure followed for testing was the same as that used for the first test. The test specimens used for the second test fitted more closely than those in the first test.

Photographs were taken of the stress patterns for loads of 11, 16, 21,

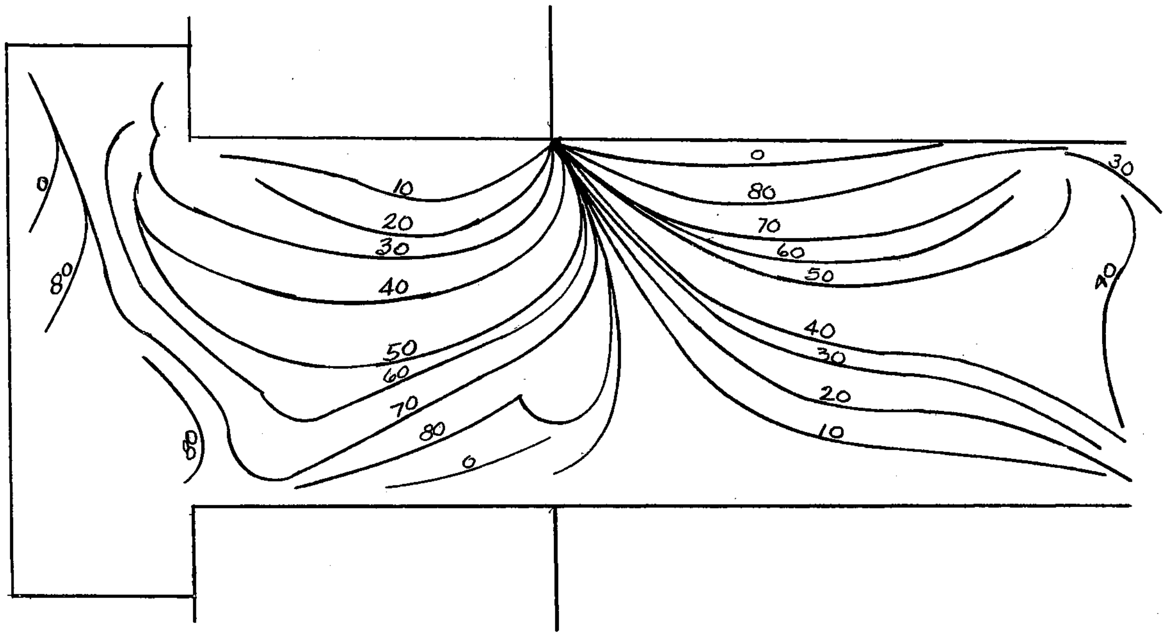



Figure 11. Sketch of Isoclinics for Test 1

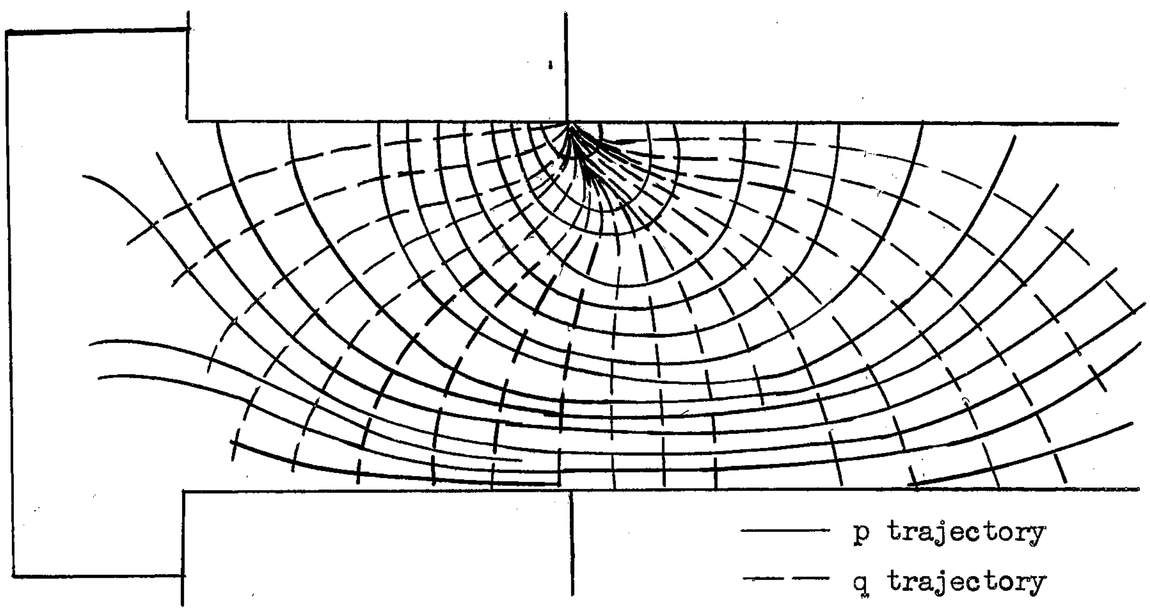



Figure 12. Sketch of Stress Trajectories for Test 1

26, 31, 35 and 40 pounds at the end of the lever arm. These loads were the same as those used in the first test. The thickness of the specimens was 0.005 inch smaller than those used in the previous test. The photographs of the stress patterns are shown in Plates V and VI. It was noted from the stress distributions that the fixture functioned satisfactorily.

A close examination of the photographs was made which revealed that several satisfactory conditions existed in this test that did not exist in the first test:

1. The head of the bolt made contact with both the upper and lower plates.

2. All plates made contact with the bolt uniformly. This situation is very noticeable in the photographs with higher loads. The small fringes along the horizontal edges of the plates indicate that the entire length of the plates were in contact. Frocht (7) presents an illustration of two rectangular plates in contact. The stress pattern in Frocht's illustration closely resembles the stress pattern obtained in the plates in this investigation.

3. The photographs also revealed that the upper outside plate began to show signs of stress concentration at the edge where it came in contact with the bolt head. An initial stress concentration appeared at this edge, as well as in the lower outside plate, when the bolt was pre-stressed by applying a tensile load through the nut. This situation indicated a very close fit between the bolt and the plates.

As the compressive load was applied the stress on the edge of the upper outside plate increased while the stress on the lower outside plate remained essentially constant. This was further indication of the proper

PLATE V. STRESS PATTERNS FOR TEST 2

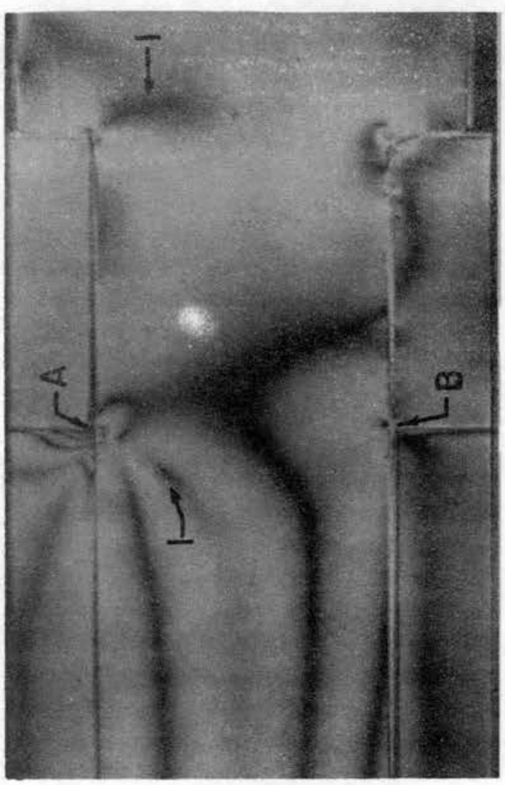
1. Stress Pattern for P = 73.04 lbs.

2. Stress Pattern for P = 106.24 lbs.

3. Stress Pattern for P = 139.44 lbs.

4. Stress Pattern for P = 172.64 lbs.

PLATE V

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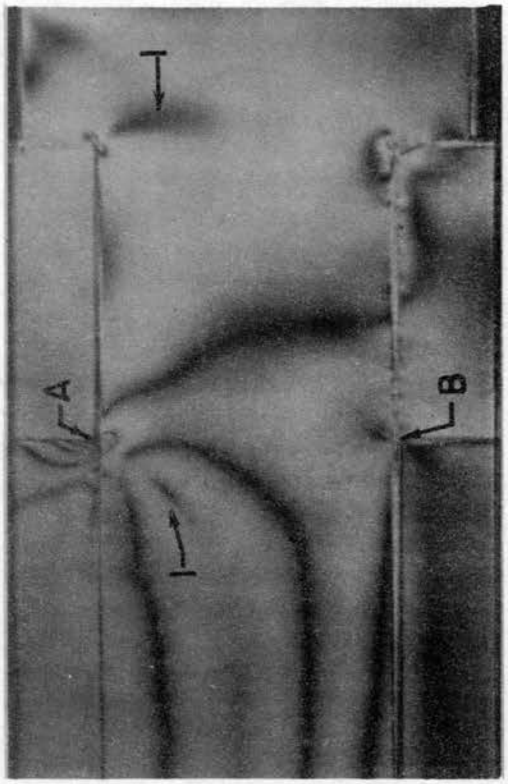


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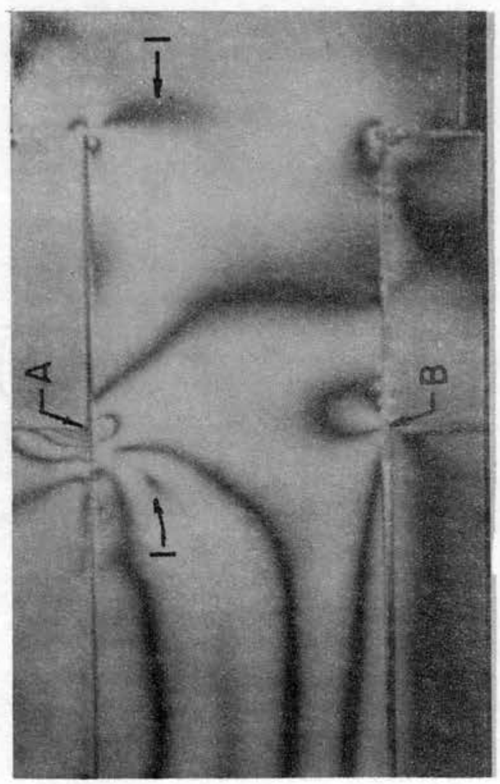


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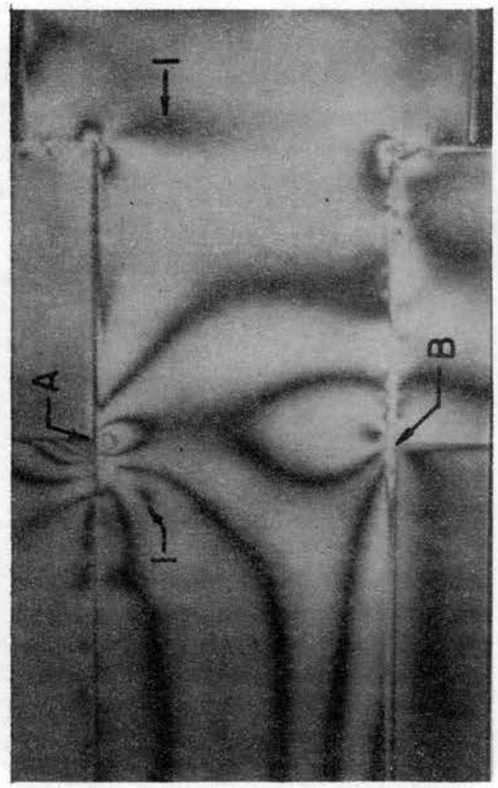



4

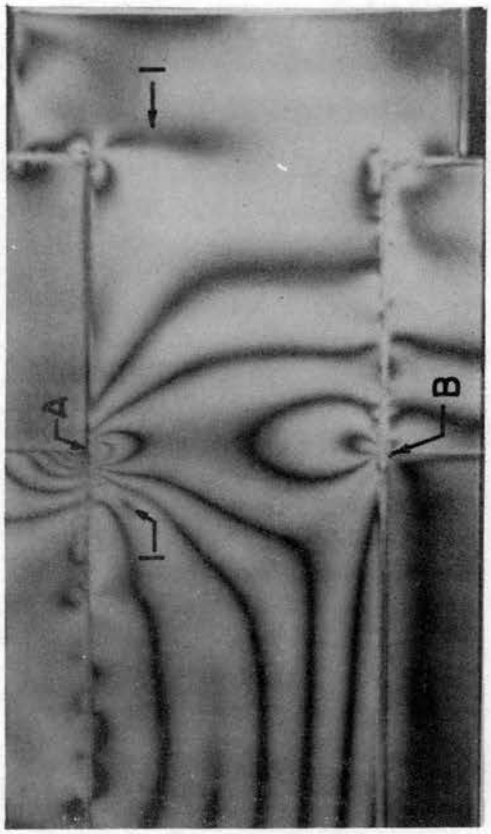
PLATE VI. STRESS PATTERNS FOR TEST 2

1. Stress Pattern for $P = 205.84$ lbs.

2. Stress Pattern for $P = 232.48$ lbs.

3. Stress Pattern for $P = 265.68$ lbs.

PLATE VI

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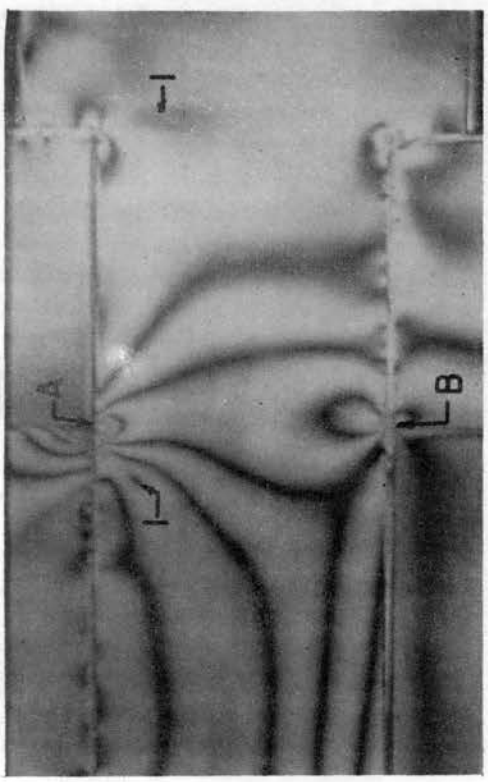
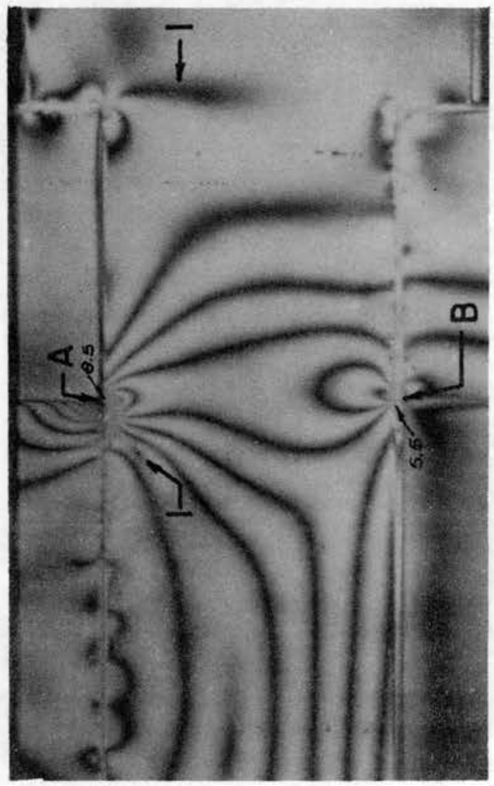



1




3

functioning of the fixture. As the load was increased the bolt deflected and forced the head against the edge of the upper plate thus causing an increase in the stress on this surface.

The photographs also indicate two isotropic points, noted as point I. The entire left portion of the photographs indicate a neutral axis. This portion of the bolt acted much like a beam in pure bending.

The isoclinics are shown in Figure 13. The stress trajectories are shown in Figure 14. The isoclinics near the upper corner of the lower supporting plate were very difficult to trace. The multicolored stress pattern obtained from the white light source was the cause of this difficulty.

The stress trajectories are labeled as p and q. In order to determine whether the directions of the stress trajectories were correct the stresses on the free boundaries were examined by the use of a compression compensator in the following manner:

The compensator was merely a rectangular bakelite specimen loaded in compression. A white light source was used with the quarter-wave plates inserted and a full load on the model. The compensator was placed behind the model, first parallel, then perpendicular to the free boundaries. When extinction of the light occurred with the compensator parallel to a free boundary the stress at that point was tensile. When extinction occurred with the compensator perpendicular to the boundary the stress at that point was compressive.

At this point it was noted that, with the deflection of the bolt and the conditions of loading, i.e., an initial tensile load and the fact that there was nearly an equivalent of a concentrated load near the center of

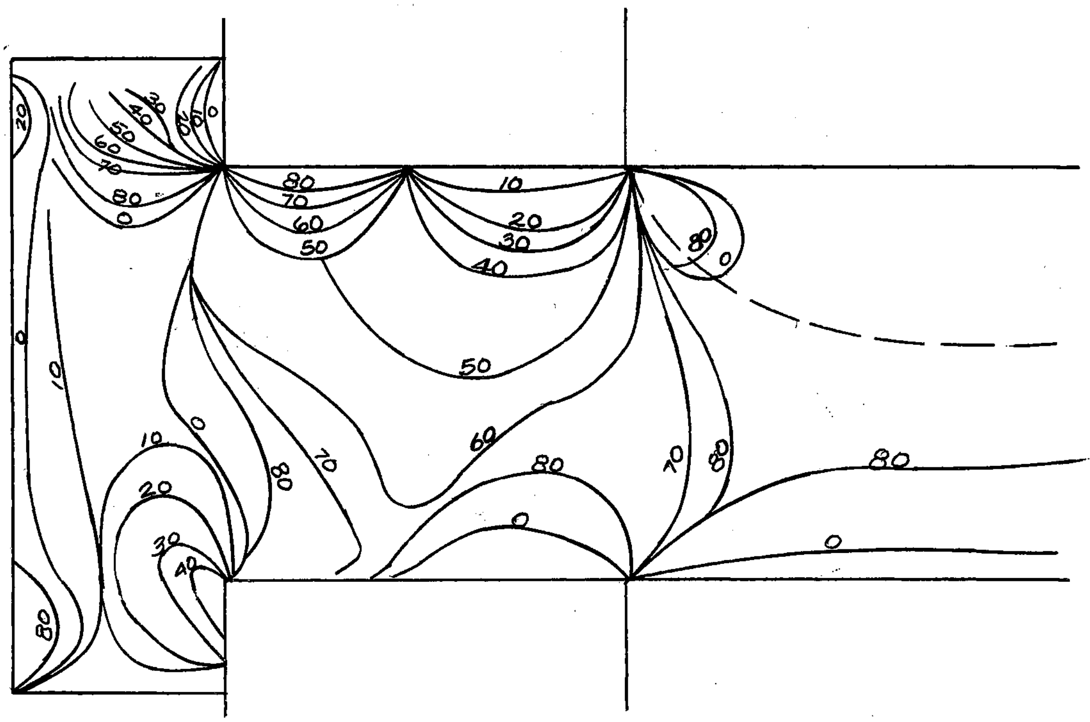



Figure 13. Sketch of Isoclinics for Test 2

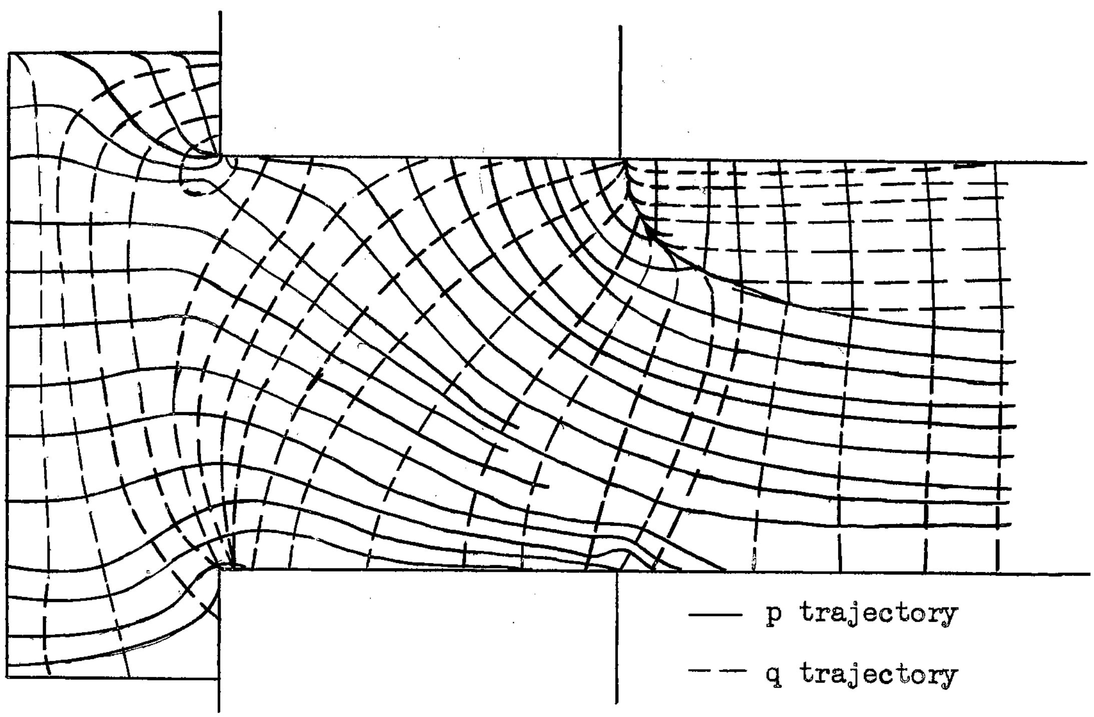



Figure 14. Sketch of Stress Trajectories for Test 2

the upper and lower surfaces, the surfaces of the bolt were free boundaries except where the concentrated loads were applied. These surfaces were also tested with the compensator to determine the type of stress present.

The compensator aided greatly in the solution of the problem. With the realization that the major portion of the boundary was a free boundary where only one type of stress could exist, either p or q, the photographs were then used to obtain the magnitude of these stresses directly since the stress patterns given by the photographs are the differences of principal stresses, p - q.

These stresses are plotted in Figures 15 and 16 for the condition of maximum load. Friction between the plates had very little effect on the stress distribution with a large load but it had a very great effect on the stress distribution at small loads. The area under the stress distribution curve was calculated. An average stress in fringes was calculated and compared with the average stress as given by the area under the stress distribution curve. The values obtained for the average stress given by the graph, 1.95 fringes, and P/A, 2.04 fringes, compare very well. The discrepancy can probably be attributed to the friction between the plates and the fact that there was some friction between the surfaces of the plates and the bolt as the bolt distorted.

The stresses at various points along the surfaces of the bolt were compared with the nominal stress, P/A. These results are shown in Table I.

It should be pointed out that this is for one particular case. The stresses in the bolt are functions of the initial tensile stress and the friction between the plates.

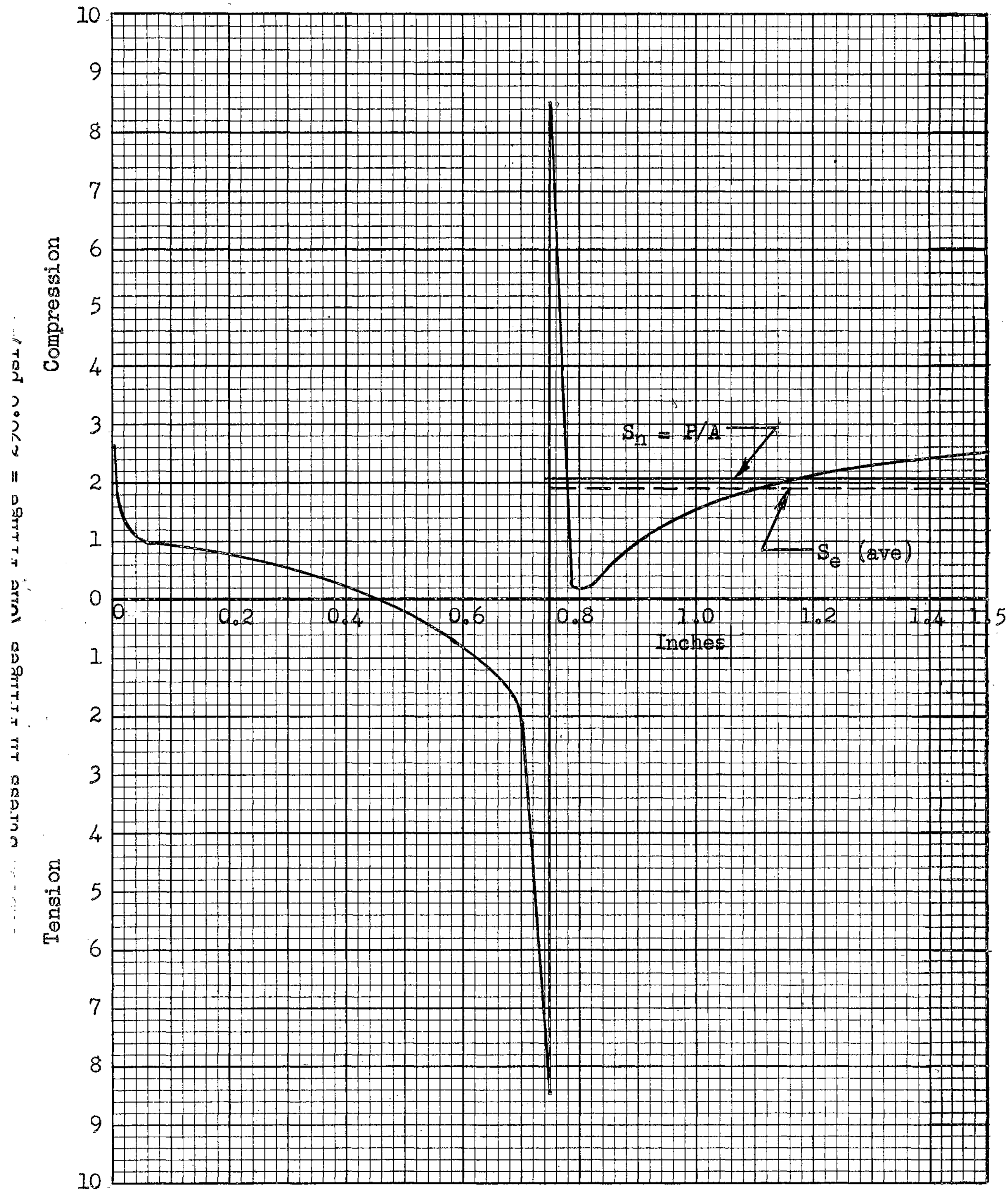



Figure 15. Stress Distribution on the Upper Boundary (P = 265.68 lbs.)

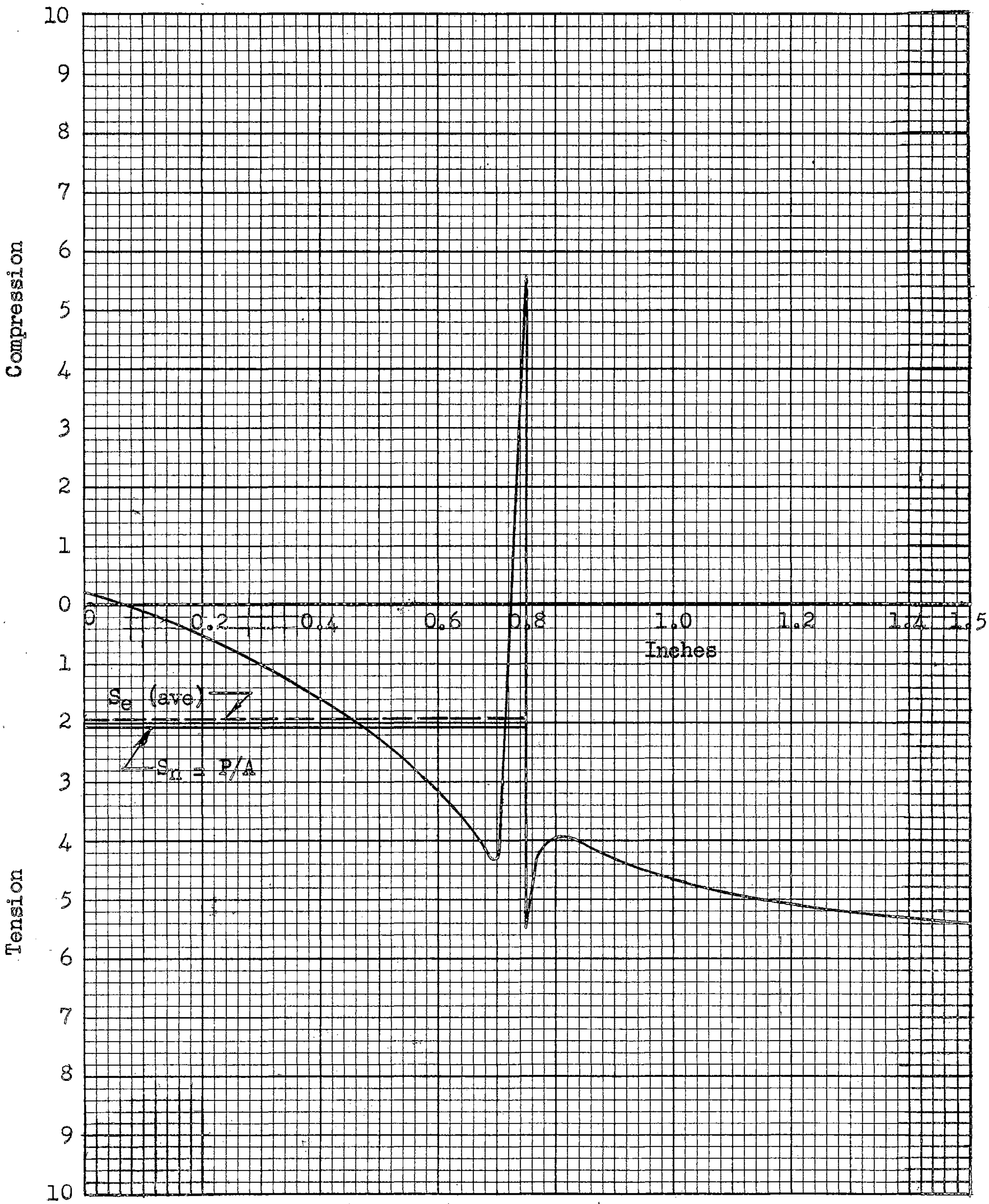



Figure 16. Stress Distribution on the Lower Boundary (P = 265.68 lbs.)

TABLE I

STRESS CONCENTRATIONS

| DISTANCE FROM INNER SURFACE OF BOLT HEAD (inches) | SN=P/A (psi) | EXPERIMENTAL STRESS (SE)(psi) UPPER SURFACE | LOWER SURFACE | RATIO SE/SN UPPER SURFACE | LOWER SURFACE |
|---|---|---|---|---|---|
| 0.0 | 609.2 | 800.8 | 59.3 | 1.27 | 0.10 |
| 0.1 | | 267.0 | 118.6 | 0.44 | 0.19 |
| 0.2 | | 222.5 | 222.5 | 0.37 | 0.36 |
| 0.3 | | 148.3 | 341.1 | 0.24 | 0.56 |
| 0.4 | | 53.6 | 489.4 | 0.09 | 0.80 |
| 0.5 | | 74.1 | 667.4 | 0.12 | 1.10 |
| 0.6 | | 237.3 | 889.8 | 0.39 | 1.46 |
| 0.7 | | 474.6 | 1230.9 | 0.78 | 2.02 |
| 0.75 | | 2521.1 | 1631.3 | 4.42 | 2.67 |
| 0.8 | | 29.7 | 1186.4 | 0.05 | 1.93 |
| 0.9 | | 308.5 | 1275.4 | 0.51 | 2.10 |
| 1.0 | | 465.7 | 1364.4 | 0.77 | 2.24 |
| 1.1 | | 557.6 | 1423.7 | 0.92 | 2.34 |
| 1.2 | | 608.0 | 1483.0 | 1.00 | 2.43 |
| 1.3 | | 652.5 | 1542.3 | 1.07 | 2.53 |
| 1.4 | | 705.7 | 1572.0 | 1.16 | 2.58 |
| 1.5 | | 741.5 | 1586.8 | 1.22 | 2.60 |

# CHAPTER V

## CONCLUSIONS AND RECOMMENDATIONS

Since there has been no previous research done on this particular problem it is difficult to ascertain the precision of the results.

The favorable comparison of the experimental average stress with the nominal average stress is an indication that the results were consistent. The fact that the p and q trajectories could be determined from the isoclinics and checked with the compensator was a great advantage and added favorably to the possibility of reasonable results.

The fact that a graphical solution was required detracted from the reliability of the results somewhat but evidently little error resulted from such a solution.

The results of this investigation would seem to indicate that further study should be undertaken. The following recommendations should be considered:

1. Investigate the effects of friction on a joint by using models with varying degrees of class of fit between the plates.

2. Investigate the effects of varying the amount of tensile prestress in the bolt.

3. Investigate the results of a bolt with infinite clearance.

4. Endeavor to prove or disprove the results of this investigation through a three-dimensional photoelastic investigation.

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# APPENDIXES

## APPENDIX A

### CALIBRATION MODEL

A calibration model was required in order to determine the material fringe value of the bakelite used for the test pieces in this investigation. The calibration model was made from the same sheet of bakelite as the test pieces. The dimensions of the calibration model are shown in Figure 17.

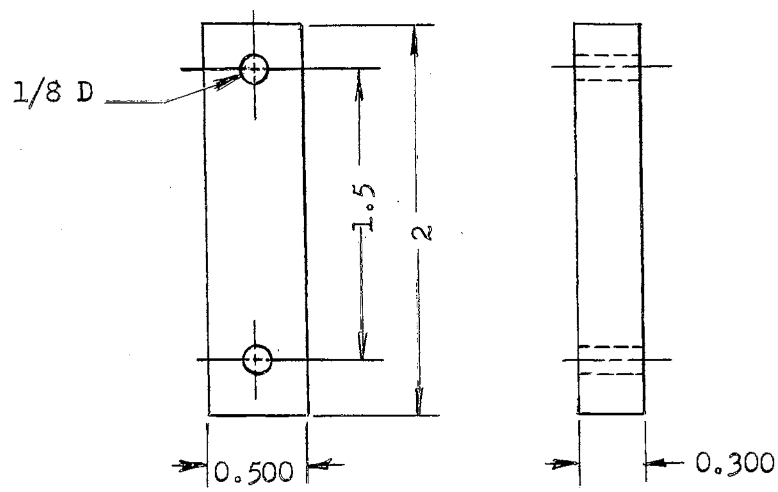


Figure 17. The Calibration Model

The calibration model was placed in the loading frame as shown in Figure 18. The polariscope was set to produce a dark background with the quarter wave plates in position using a monochromatic light source.

A load was applied at the end of the lever arm until the model changed from dark to light to dark, that is, until one fringe passed through the model. The load at which this occurred was recorded in the data. The load was then increased until the second fringe passed through the model. The value of this load was also recorded in the data.

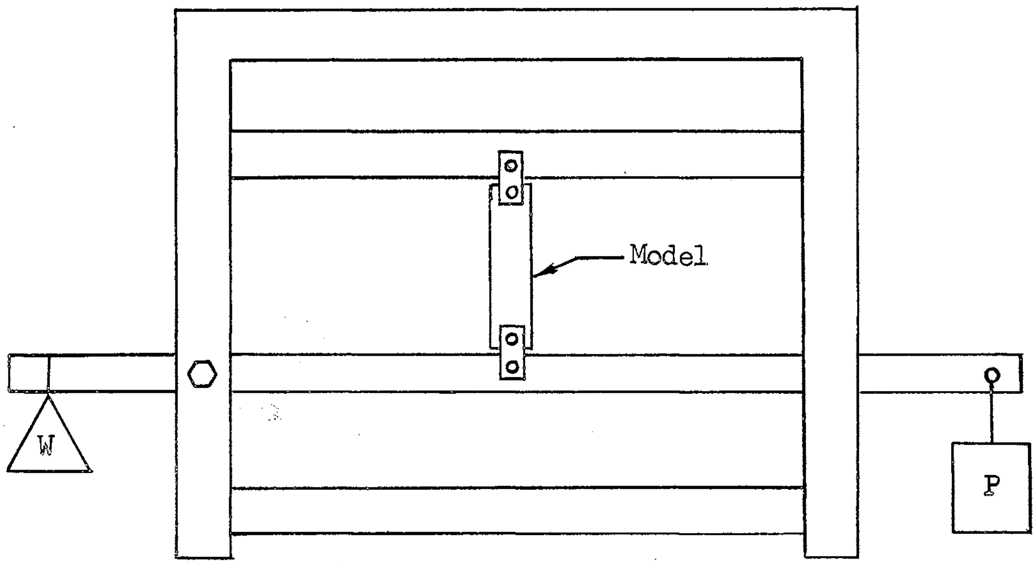


Figure 18. Schematic of Model in Loading Frame

This procedure was repeated until a total of three fringes were counted. The various loads at which the fringes occurred are shown in Table II.

TABLE II

MATERIAL FRINGE VALUES

| NUMBER OF FRINGES (n) | WEIGHT AT END OF LEVER ARM (lbs) | DIRECT LOAD ON MODEL (lbs) | MATERIAL FRINGE VALUE (f) |
|---|---|---|---|
| 1 | 8.172 | 42.756 | 42.76 |
| 2 | 16.516 | 86.412 | 43.21 |
| 3 | 24.946 | 130.517 | 43.51 |
| | | | Average f = 43.16 |

The actual load on the model was simply the applied load at the end of the lever multiplied by the distance from the fulcrum to the end of the lever divided by the distance from the fulcrum to the point of application on the model.

The material fringe value for each load was calculated in the following manner:

$$f = \frac{P}{2dn},$$

where, $f$ = material fringe value,
$P$ = actual load on the model,
$d$ = width of the test specimen,
$n$ = fringe order.

$$f = \frac{42.76}{2\ (0.500)\ (1)} = 42.76 \text{ psi/fringe.}$$

The values for material fringe value are also listed in Table I.

An average material fringe value, 43.16 psi/fringe, was calculated and used in this investigation.

Frocht (5) lists the material fringe value for Bakelite BT-61-893 as 43 psi/fringe. The value obtained in this investigation differed from Frocht's value by 0.36%.

## APPENDIX B

### PROPERTIES OF BAKELITE BT-61-893

TABLE III

| MODULUS OF ELASTICITY | TENSILE STRENGTH | POISSON'S RATIO |
|---|---|---|
| 615,000 psi | 17,000 psi | 0.365 |

Bakelite is generally isotropic, has a high optical sensitivity, has a linear stress-strain relationship up to 6000 psi and a linear fringe-stress relationship up to 7000 psi (6).

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