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# A PHOTOELASTIC DETERMINATION ON THE NORMAL STRESS UNDER BOLT HEADS

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## ABSTRACT

An experimental investigation using photoelastic techniques has been used to provide information concerning the actual stress distribution between fasteners and plates. The techniques of stress "freezing" and model slicing were used on a number of specimens constructed from Hysol CP 54290. The curves representing the stress distributions and magnitudes are plotted for fillister head and button head bolts. Prev[illegible] vestigations using analytical methods for calculating stress distri[illegible]s for fillister head bolts are compared to the results obtained photoelastically. Previous experimental stress distributions obtained by oil pressure and penetration techniques are compared to stresses determined photoelastically for the button head bolt. It was confirmed from the results that the age old assumption of uniform normal stresses under the bolt heads is invalid. The shape of the bolt head greatly influences the stress distribution. A stress distribution of an exponential form generally exists for constant shank load.

# INTRODUCTION

Fasteners such as bolts and rivets have been in use for such a long period of time that optimum design has essentially been accomplished by evolution. Consequently only a small amount of actual data has been published on fasteners and the stress distribution between the fastener and plate. Results show that a configuration which distributes the load more toward the outer edges of the rivet head might increase the reliability of the joint.

The prediction of heat transfer across joints by using the stress distribution at the interface is another important area in which the stress distribution must be known before accurate calculations can be made. This problem was considered in a recent dissertation (1)*, entitled "Thermal Conductance of Bolted Joints." The heat transfer by conduction is dependent on the degree of actual contact between the plates which is dependent on the magnitude and distribution of stress joining the plates. Similarly heat transfer by convection and radiation are dependent on the degree of separation of the plates, which are determined by the points of zero stress in the normal stress distribution. By experimental stress analysis an accurate solution to the fastener stress distribution problem can be determined and subsequently used to predict the plate interface stress distribution. Knowing the stress distribution, one can calculate the heat transfer across the joint.

Other examples can be cited in which the stress distribution

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*Numbers in parenthesis refer to the references is an important factor in the overall analysis and reliability of the system. The assumption that the normal stresses do not vary over the radial portion of the bolt head has been made in the past, and environmental designs were based on this assumption. A very brief review of previous investigations is, therefore, necessary to present some of the conflicting results.

## SCOPE OF PRESENT INVESTIGATION

As stated in the introduction, that in order to determine rivet fastener reliability, the magnitude of stress at the outer edges must be determined and compared to the over-all load imparted to the fastener. The magnitude of the normal stresses is required in a multiplicity of problems that the designer encounters. The primary objective of this analysis was to determine experimentally the magnitude and locations of normal stresses and the manner in which they vary in fillister and button head or rivet type fasteners.

The photoelastic techniques of stress freezing and slicing offered a convenient and reliable method to obtain qualitative and quantitative results, and was employed in this investigation. Diametrical slices as well as other cross sections were taken from the specimens and analyzed for directional and magnitude fringes. The results have shed new light to the problem of determining the normal stress distribution for fasteners of complex geometry.

## PREVIOUS WORK

Almost all previous work in this field has been to assume a constant stress distribution between the plates and fasteners. Bumgardner (2), in his analytical solution to the stress distribution under round fillister head bolts, determined the stress to be nearly constant. His results show that the stress variation between a point near the bolt shank and a point near the outer edge of the bolt head was as small as 10%. Bumgardner's analysis was made assuming the bolt head to behave as a plate on an elastic foundation. The area of the plate that represented the location of the bolt shank was assumed rigid. The problem was essentially reduced to a cantilevered plate fixed at the shank and free at the outer edge. The analysis did not take into account any stress concentrations that would exist at the shank-head connection. The equations were derived from a variation principle following the method of Reissner (3) and Fredrick (4).

Fontenot (1) in a dissertation attempted to measure experimentally the stress distribution under bolt heads by using oil pressure and penetration techniques. From oil penetration measurements he determined that the normal stress became zero at approximately 0.34 inches from the shank of a 1 inch button head bolt. This point of zero stress did not change appreciably with the change in load applied to the bolt. For the fillister head bolt, he determined that normal stress is present up to the outer edge of the bolt head. By measuring the oil pressure required to separate the bolt head from the plate at various distances from the bolt shank, Fontenot plotted pressure distribution curves for the fillister and button head bolts.

The data points plotted for the fillister head bolt did not correlate with the assumed stress distribution curve. The data points for the button head bolt also allow a variety of curves to be drawn connecting them. Fontenot's results essentially coincide with Bumgardner's work on the fillister head bolt, but no data on the pressure readings were taken near the shank and he could only guess as to how his curve behaved in this region. Bumgardner, neglected the stress concentrations in the analytical solution and therefore, could not accurately predict the stress near the shank. The concentration of stress at the fillet joining the bolt head and shank would increase the value of stress in this region to several times that of the ideal case which Bumgardner computed.

## EXPERIMENTAL TECHNIQUE AND INSTRUMENTATION

Photoelasticity was used throughout the experimental analysis and was essentially that of the three-dimensional stress "freezing" technique. The material selected for the specimens was Hysol CP 54290. The only major difficulty encountered with this plastic was the variation of some of the properties from the specifications published by the manufacturer.

Standard dimensions of 1 inch bolts of the button and fillister head type were used to machine plastic bolts from the round stock. Figure 1 shows typical shapes of the specimens. Plates were cut to size from sheet stock (See Figure 2). Holes were routed in the plates to provide a snug fit around the bolt shank. The lip of the hole was chamfered to fit the 1/32 inch fillet radius between bolt

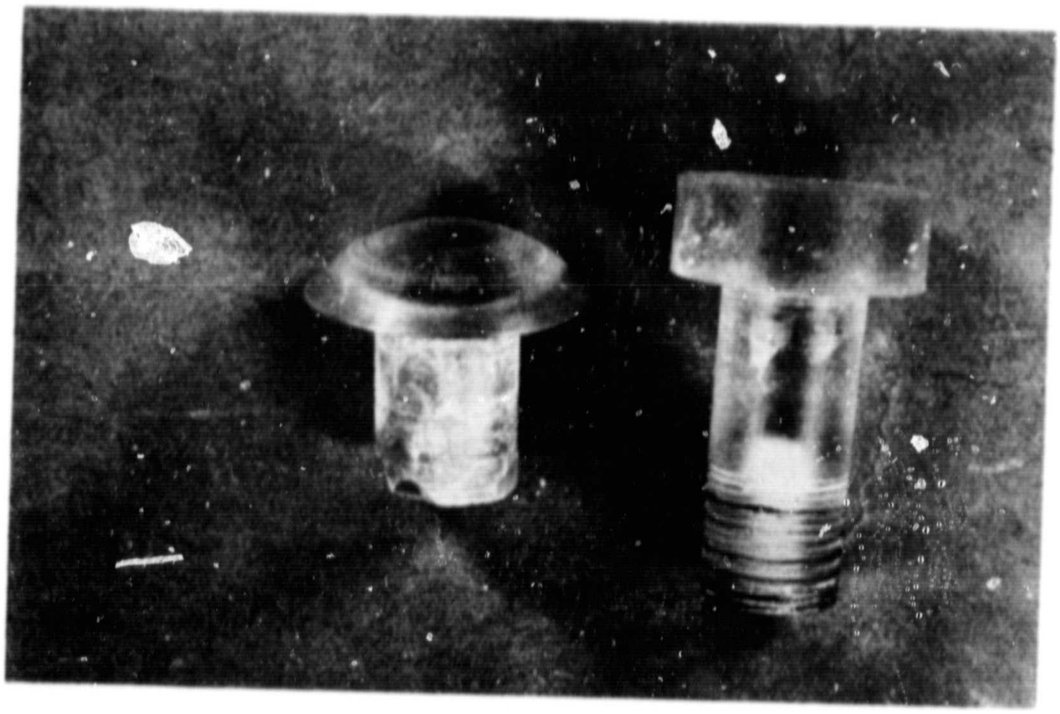

Figure 1. Button and Fillister Head Plastic Bolts

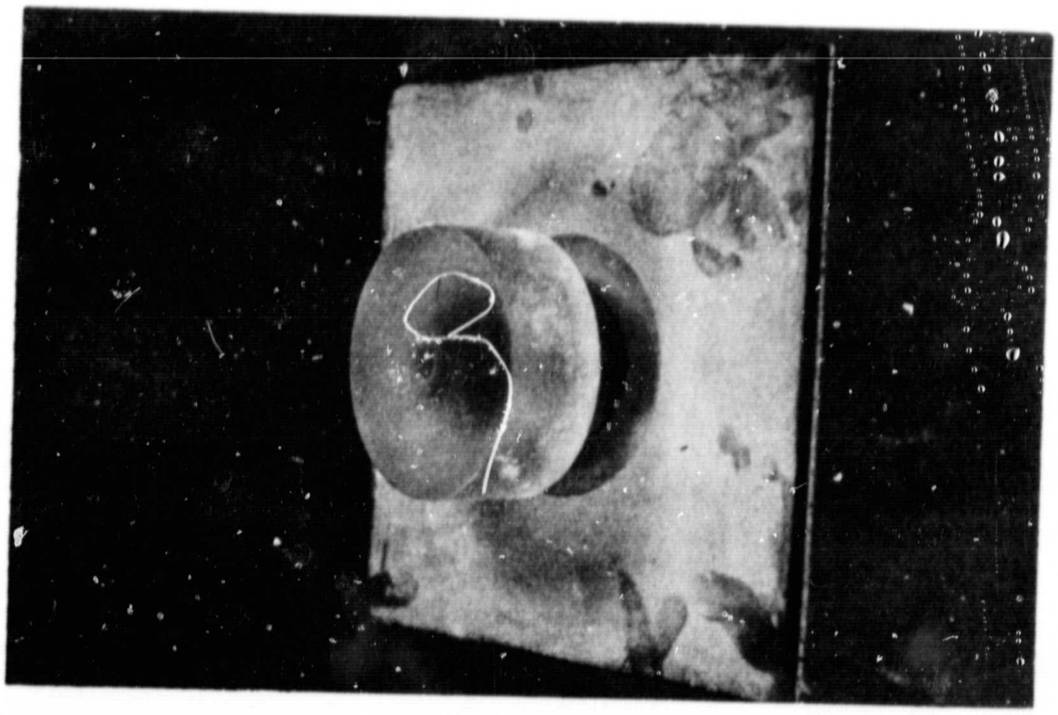

Figure 2. Fillister Head Bolt and Plastic Plates

head and shank. The mating surfaces of the bolt and plate were polished until uniform contact was obtained at the interface. Care was taken in all machining processes so as not to induce residual stress in the specimens. This involved controlling of the speed and depth of cut in the machining process. The polariscope was used to check specimens for residual stresses. If residual stresses were present, a tedious, time consuming process of stress relieving was necessary before the specimen could be used.

A loading rack was designed and constructed to maintain uniaxial loads on the specimen. This was accomplished by mounting the plates on a set of gimbals similar to those found in gyroscopes. With the center line of the bolt at the symetrical axis of the gimbals, undesirable moments were eliminated from the system. Shear forces at the interface were eliminated by blowing talcum powder between the bolt head and plate. The entire assembly of stress rack, bolt, and plates was placed in a special temperature controlled oven for approximately 168 hours for the stress freezing cycle.

## STRESS FREEZING

Stress "freezing" was begun by slowly heating the specimen from room temperature to the stress freezing temperature, which for this plastic was $270^{\circ}F$. This was accomplished by a control cam on the oven controller. The bolt was allowed to soak at the constant temperature of $270^{\circ}F$ for about 6 hours to insure uniform temperature throughout the bolt. The load condition was applied to the bolt, and then the temperature was reduced to room temperature over a period of 127 hours. This insured a cooling rate of less than $1\frac{1}{2}$ degrees per hour. This slow rate of cooling prevented temperature gradients from forming and thus eliminated the possibility of having thermal stresses frozen into the stress pattern along with the mechanically induced stress.

A thin slice of about 0.08 inch thickness was taken on the diameter of the bolt away from any non-uniformity that was observed previously. This slice was then polished to a uniform thickness. Successive slices were then removed away from the diameter of the bolt and polished in the same manner (Figure 3). These successive slices were used to check the validity of the diametrical slice and to determine the stress gradient perpendicular to the face of the diametrical slice. This gave a good indication as to the amount of error introduced by the thickness of the slice; the error was found to be negligible. Slicing and polishing the plastic bolts does not change the fringe patterns if care is taken to prevent excess buildup of heat in the specimen.

The thin slices were placed on a highly polished, reflective backing. By use of a surface analyzer fringe patterns were analyzed and their magnitude of fringe order (N) was determined by the use of a linear compensator attached to the surface analyzer.

## METHOD OF ANALYSIS AND RESULTS

### Fillister Head Bolt

The surface stress analyzer with a linear compensator attachment was employed to observe the magnitudes and directions of principle stress frozen into the thin plastic slices. From the

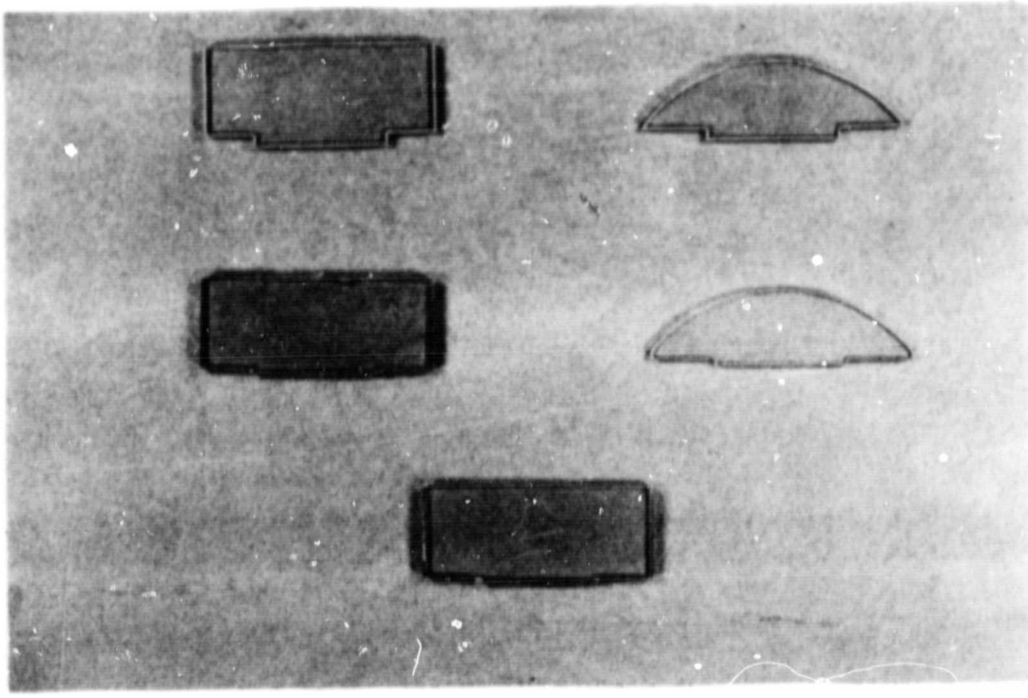

Figure 3. Thin Slices Taken from Button and Fillister Head Plastic Bolts

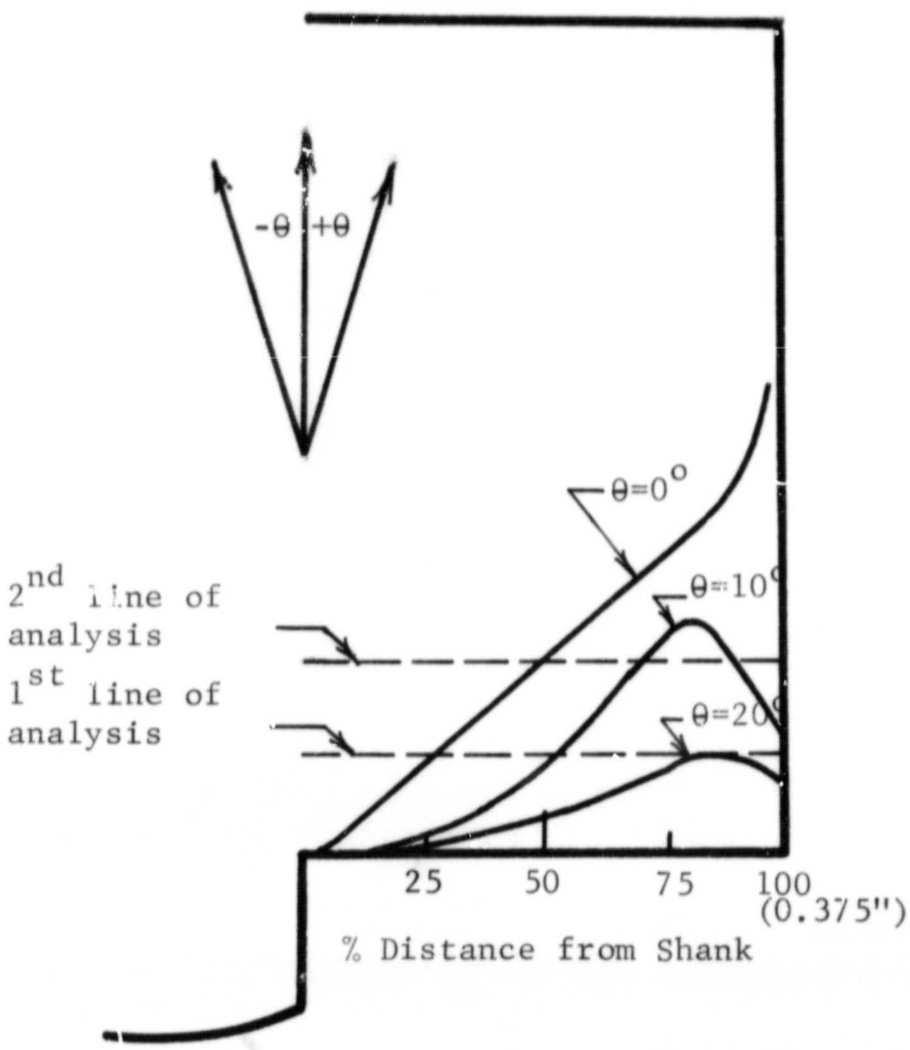


Figure 4. Deviation of Principle Stress from Normal for Fillister Head Bolt

directional fringe pattern, plots were made of normal stresses ($\theta = 0$) and stresses having directions which vary up to $\theta = + 20$ degrees (Figure 4). Nearly perfect symmetry existed therefore, only one side of the head was considered. From these plots the validity of only analyzing normal stress an be observed. Since shearing forces at the interface were not completely eliminated by the talcum powder, variations from the normal did occur near the edge. This deviation was made small by the use of a line of analysis located away from the interface a distance of 5.7% the total distance from interface to top of bolt head (Figure 4).

The magnitude of stress was determined along the interface, along the line of analysis and at the 2nd line of analysis. This 2nd line of analysis was taken at twice the distance from the interface as the 1st line of analysis (Figure 4). By noting the variation of the curves at the three locations, some idea of the stress gradient perpendicular to the interface can be determined. The stress increased at every point on the line of analysis over points at the interface along the same normal line to the interface. Corresponding points on the 2nd line of analysis show a slight increase over points on the 1st line of analysis. This trend was reversed above the 2nd line of analysis and a gradual decrease in stress is noted for corresponding points along the normal line all the way to the top of the bolt head.

The value of fringe order (N) was determined at various distances from the shank for each of the three horizontal lines of analysis. These values for N are related to stress by the stress-optic theory which is expressed as:

$$\sigma_1 - \sigma_2 = \frac{K}{t} N$$

where $\sigma_1$ and $\sigma_2$ are principle stresses, K is a constant for the particular plastic used, t is the thickness through which the polarized light must travel in the specimen, and N, as mentioned earlier, is the fringe order at the point being considered.

In the analysis, the magnitude of $\sigma_2$ was such a small value due to the use of talcum powder that it was assumed to be zero. Values for t were measured with a micrometer, while the value of K is specified by the manufacturer of the plastic. The values of stress at each point were calculated. Figures 5 and 6 show the fringe patterns for a diametral slice.

The plot of stress versus radial distance from the shank is shown in Figures 7 and 8 for the cases of the interface normal stresses and the $1^{st}$ line of analysis. The stress curves determined by Bumgardner (2) and Fontenot (1) are shown superimposed on this work for comparative purposes. The transition from the range of stress studied by Bumgardner and Fontenot to the range obtained in the calculations discussed previously was accomplished by using the well known dimensionless equation:

$$\frac{\sigma_M L^2_M}{F_M} = \frac{\sigma_P L^2_P}{F_P}$$

where the subscript M denotes values taken from the system using a metal bolt and the subscript P denotes values taken from the system using a plastic bolt. F in this case is the axial load applied to the shank. $L^2$ represents the product of two major dimensions of the bolt (diameter and bolt head height), $\sigma_M$ denotes stress in the metal

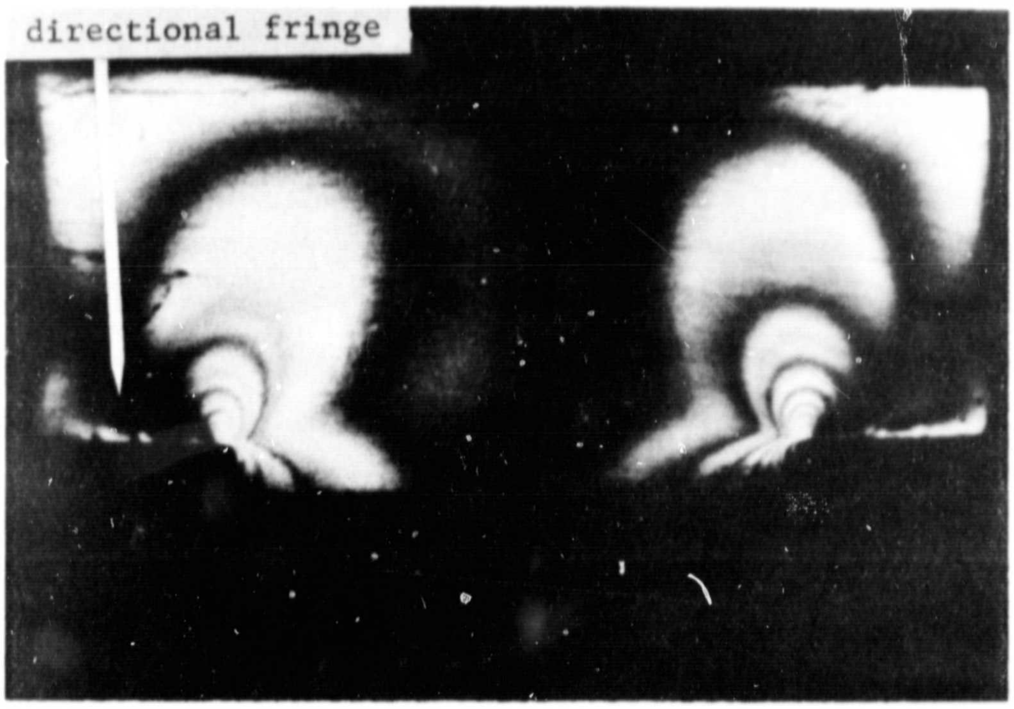


Figure 5. Directional Fringe Pattern for Fillister Head Bolt

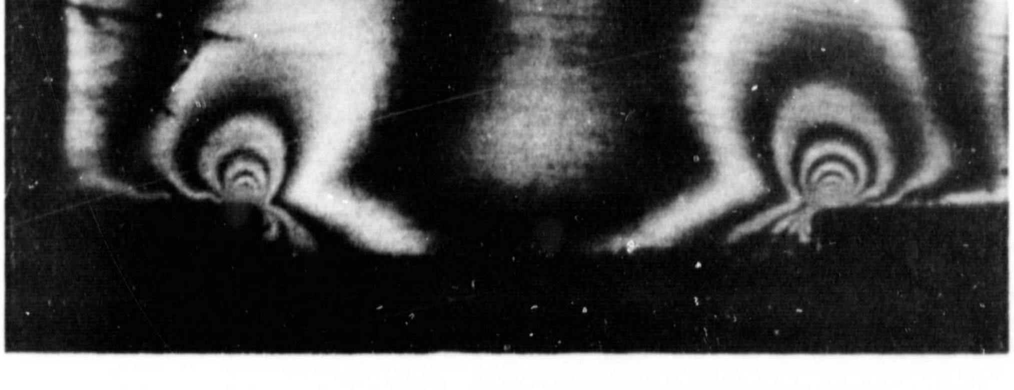


Figure 6. Magnitude Fringe Pattern for Fillister Head Bolt

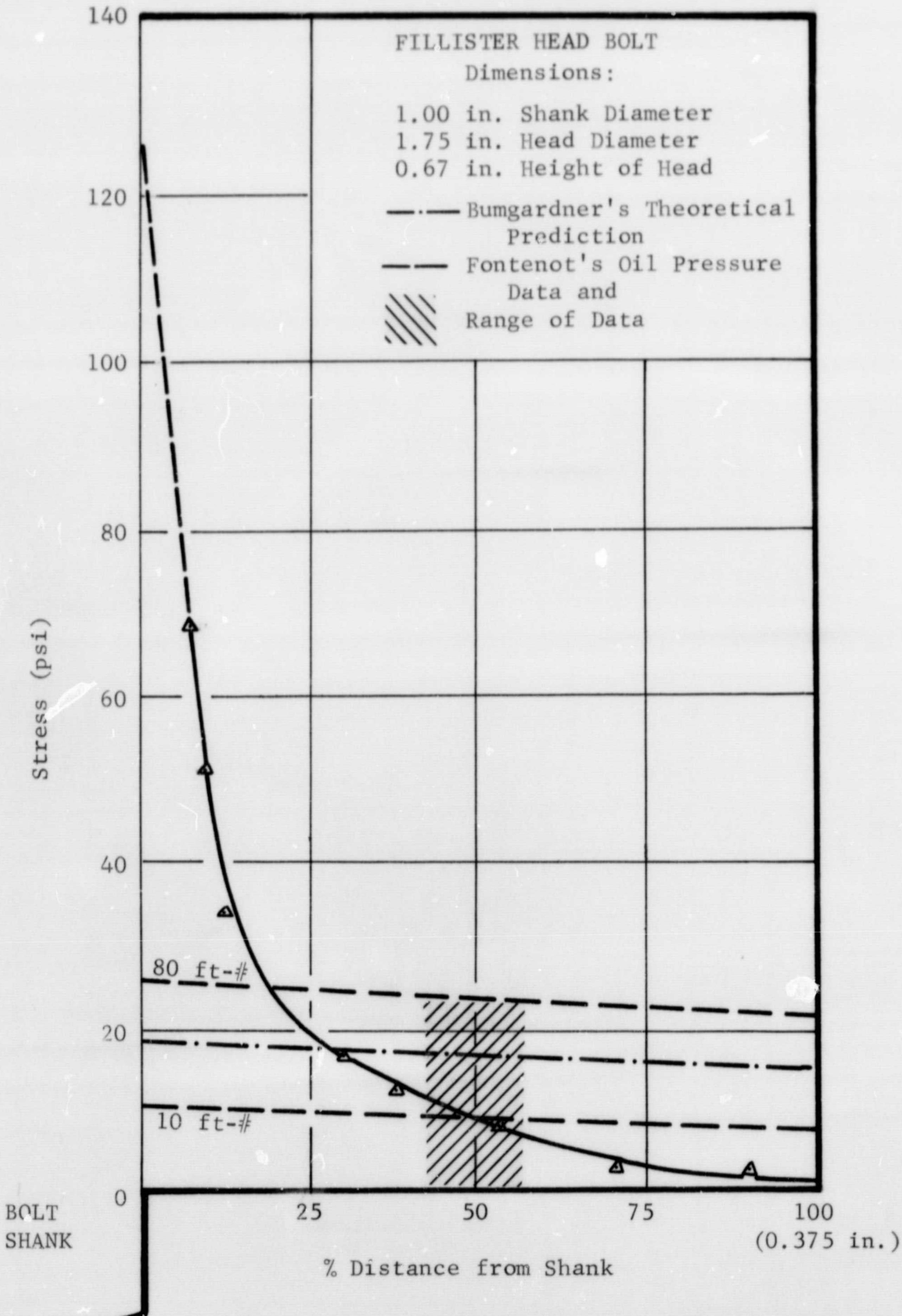


Figure 7. Normal Stress vs Radial Distance from Shank at Interface

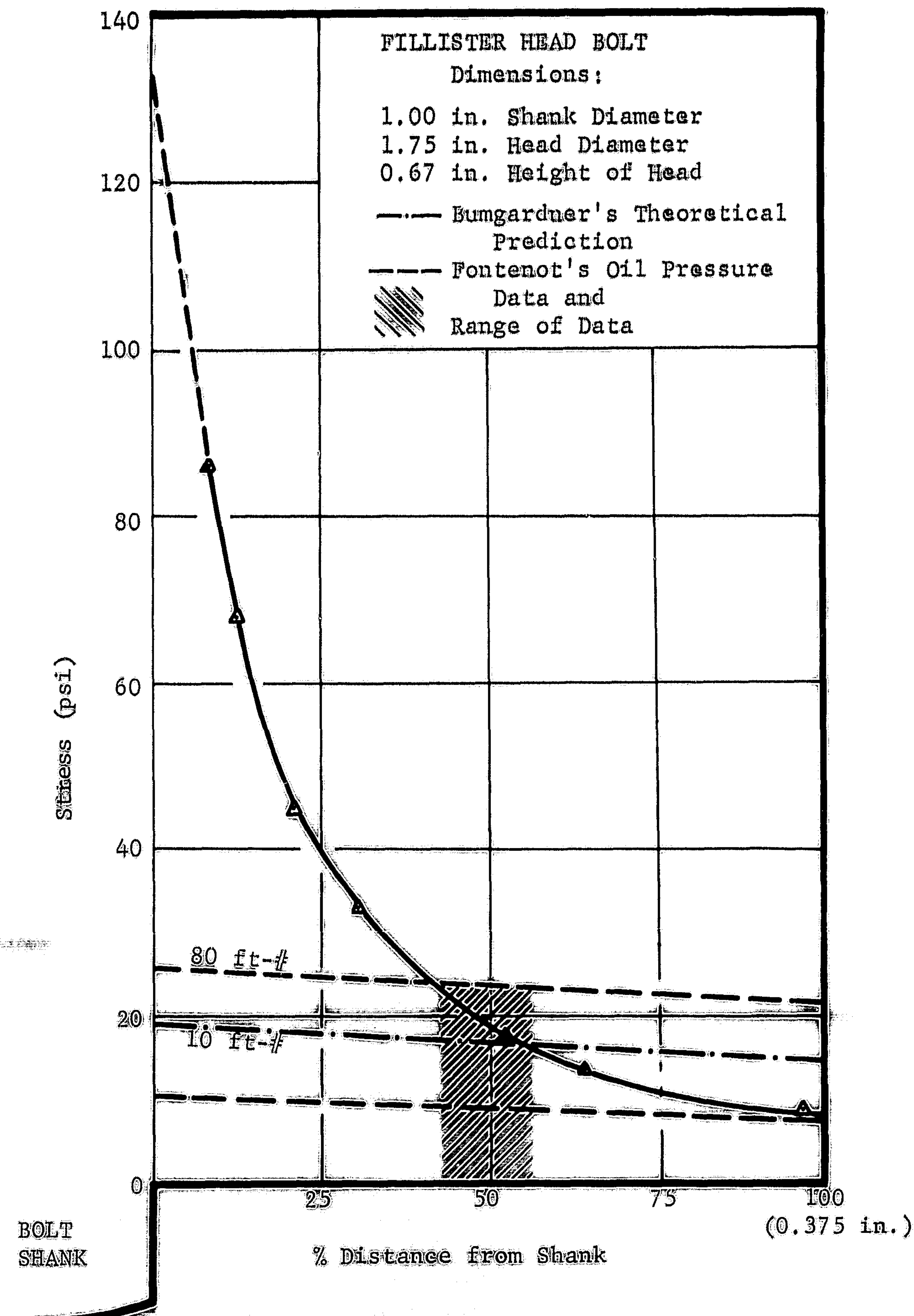


Figure 8. Normal Stress vs Radial Distance from Shank at 1st Line of Analysis

the shank could be calculated. This is shown on the vertical axis of Figure 5. Using this point, the distribution curve can be extended to the vertical axis as is shown by the dashed portion of the curve. Note that perfect continuation of the slope of the experimentally determined curve would intersect the axis at approximately the same value as given by Peterson's stress concentration factor. The results are similar when this method is applied at the first line of analysis (Figure 6).

The only means of determining the value of stress at the outer edge would be to extrapolate the curve to this point, as is shown. This small region of extrapolation without verification did not enter nor alter the conclusions derived by this analysis.

The range of curves obtained by Fontenot as he varied the torque applied to the bolt would average out to be a very good approximation to Bumgardner's curve. Since Fontenot's curves were determined experimentally, it would seem that the curves determined in this analysis would compare more favorably to his than the analytical solution of Bumgardner; however, they do not. The main reason for this fact is that Fontenot's limited range of data points allowed him to draw the curve arbitrarily through the data points. Furthermore, a closer look at his actual data points show that the type curve derived in the author's analysis will fit Fontenot's data points better than the curve he assumed. The data points at the interface edge determined photoelastically are a bit lower than the data points in Fontenot's range of data, but the data points at the line of analysis match almost perfectly with Fontenot's data points in this range.

The curve representing the stress distribution at the 1st line of analysis is much smoother than the curve at the interface.

This was due to the fact that time-edge effects of the plastic and machining grooves at the interface caused distortions in the fringe patterns. These distortions made accurate measurements at the interface difficult. The actual curve at the interface probably takes the form of the curve at the $1^{st}$ line of analysis. The curve at the line of analysis as mentioned before has higher values of stress at every point than does the curve at the interface. The most accurate values of stress measured at the interface were taken in the area near the shank; thus, these points should be used as a guide to determine how high the line of analysis should be to eliminate edge effects.

## BUTTON HEAD BOLT (LIGHT LOAD)

### Direction of Stress

The button head bolt will be discussed in two parts; this section is in reference to the right side of the photographs shown in Figures 9 and 10. The left side (heavy load) will be discussed in the following section. A comparison of the two will be made.

Figure 9 as indicated by the arrow, contains the dark fringes denoting direction of principle stress. This photo was taken to show the location of stress normal and parallel to the interface ($\theta = 0$); $\theta$ was varied from 0 degrees to 20 degrees by use of the surface analyzer. Only one line of analysis was used in this case because values of $\theta$ varied closely around zero near the interface. The region of fluctuating $\theta$ at the interface appeared to be caused by the degree of roughness of the surface.

These results indicate that normal stresses are present at most points on the interface, the deviation being at most 20 degrees bolt, and $\sigma_p$ denotes stress at a particular corresponding point in the plastic bolt.

## BUTTON HEAD BOLT

The analysis of the button head bolt is essentially the same as that for the fillister head bolt. Figure 9 shows the dark directional fringe removed under circular polarized light and the remaining fringes representing magnitude of stress. The major difference between the button head and fillister head fringe patterns is that the button head contained some moment across the interface where the fillister head did not. This moment was due to inaccuracy in the loading device. The moment is evident in the button head by the greater concentration of fringes on the left side of the bolt head than on the right side. Because of this difference in stress from one side of the bolt to the other, an analysis was made of both sides. This will give an indication as to how the distribution and direction of stress varies with a variation in load. This is indicated as light and heavy load. The plots denoting direction of principle stress near the interface are shown in Figure 11. At $1^{st}$ line analysis is shown and serves the same purpose as before. The plot denoting magnitude of stress is calculated by measuring values of (N) and using the same equations as for the fillister head bolt. This plot is shown in Figure 12. The curves determined by Fontenot for this particular bolt configuration are also shown.

## DISCUSSION

### Direction of Stress

From the plots denoting direction of principle stress

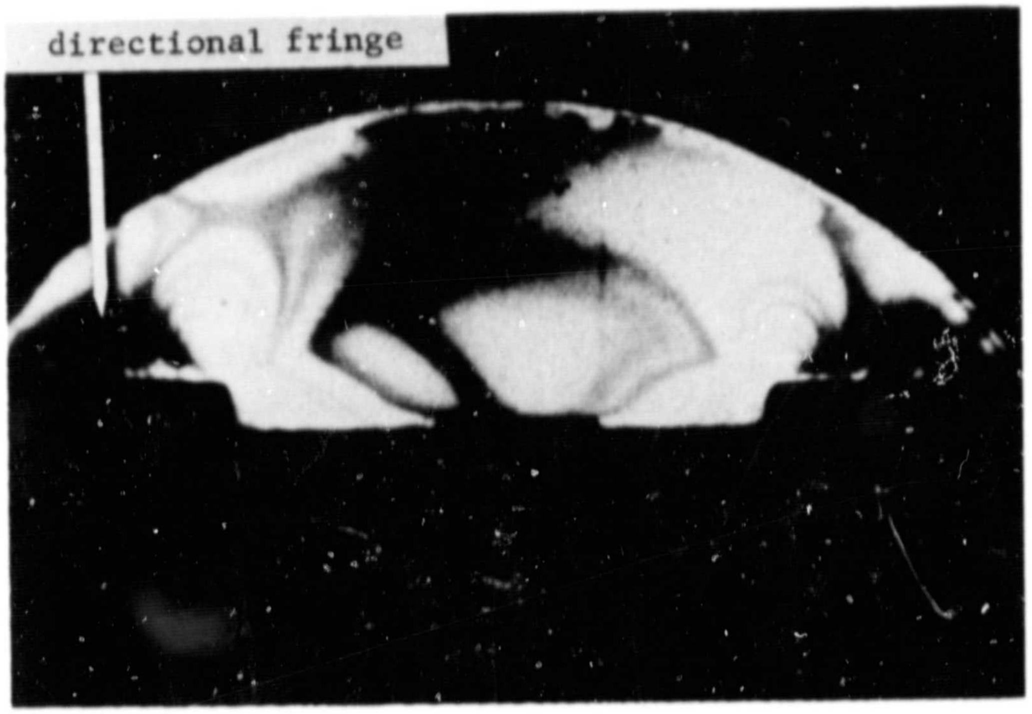


Figure 9. Directional Fringe Pattern for Button Head Bolt

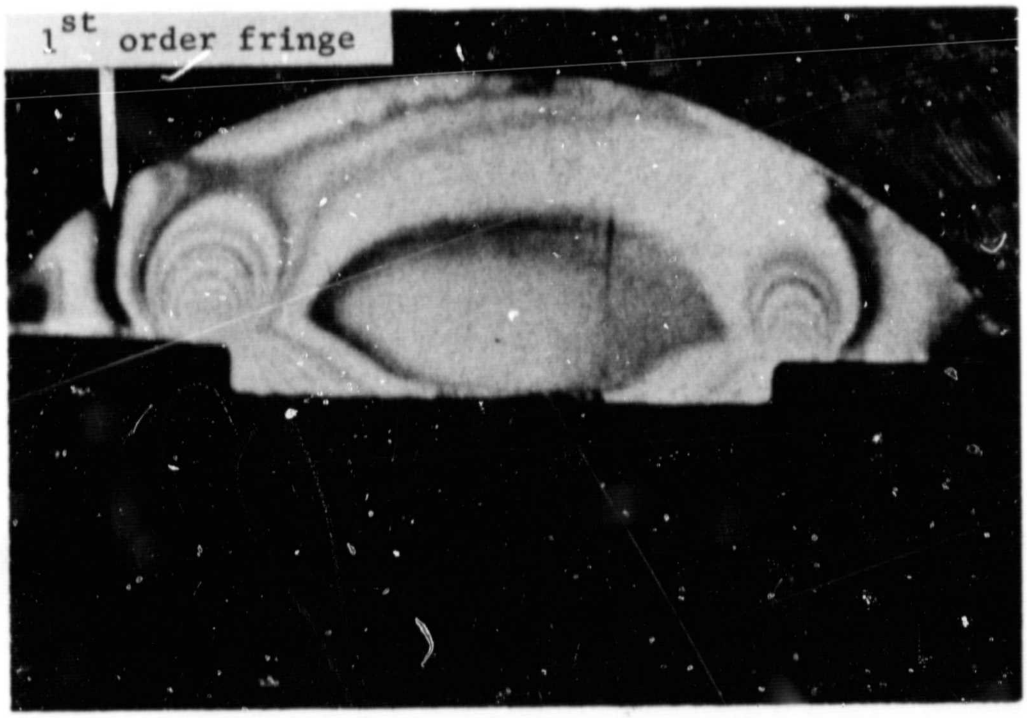


Figure 10. Magnitude Fringe Pattern for Button Head Bolt

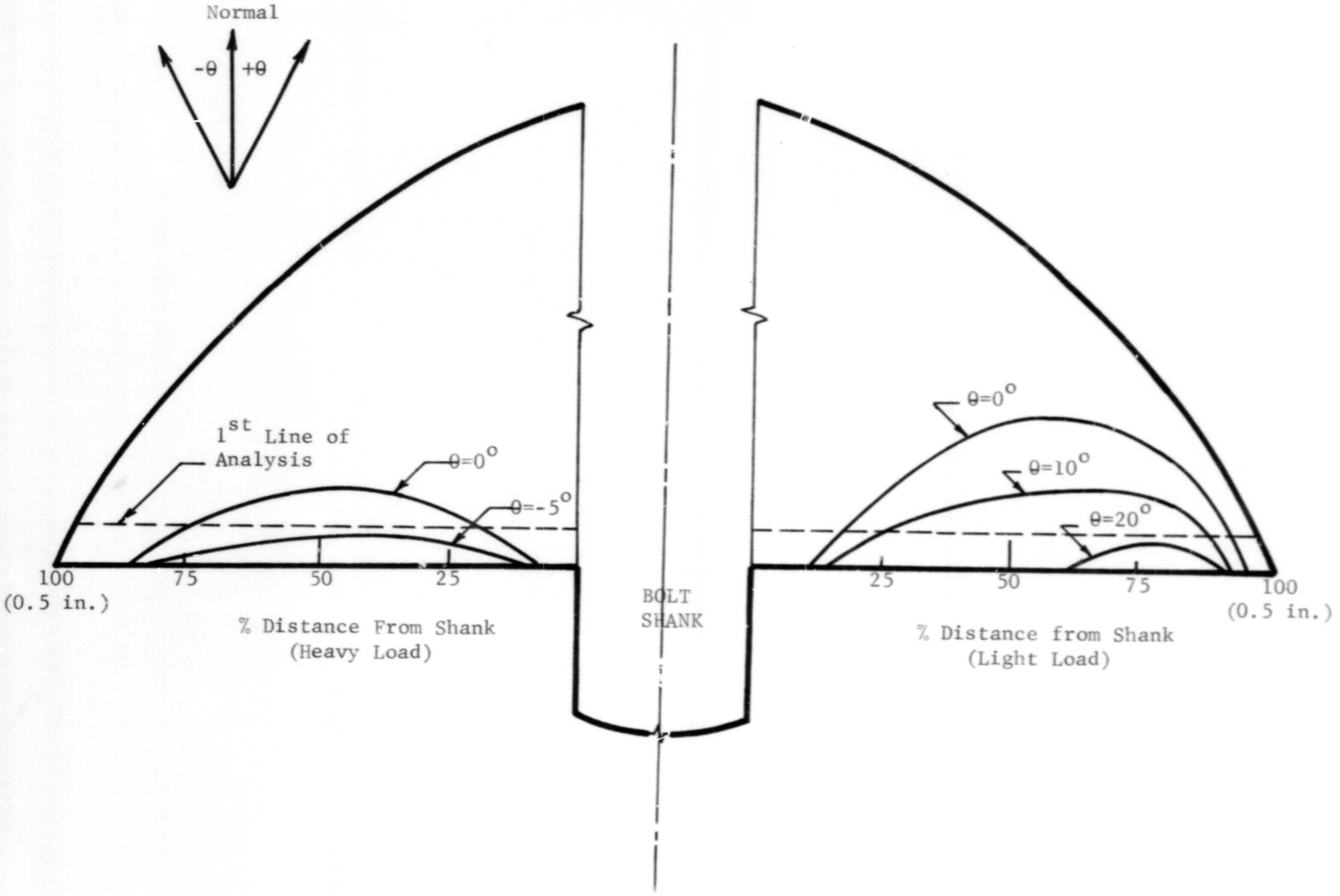


Figure 11. Directional Fringe Pattern for Rivet Type Bolt Head
(Heavy and Light Loads)

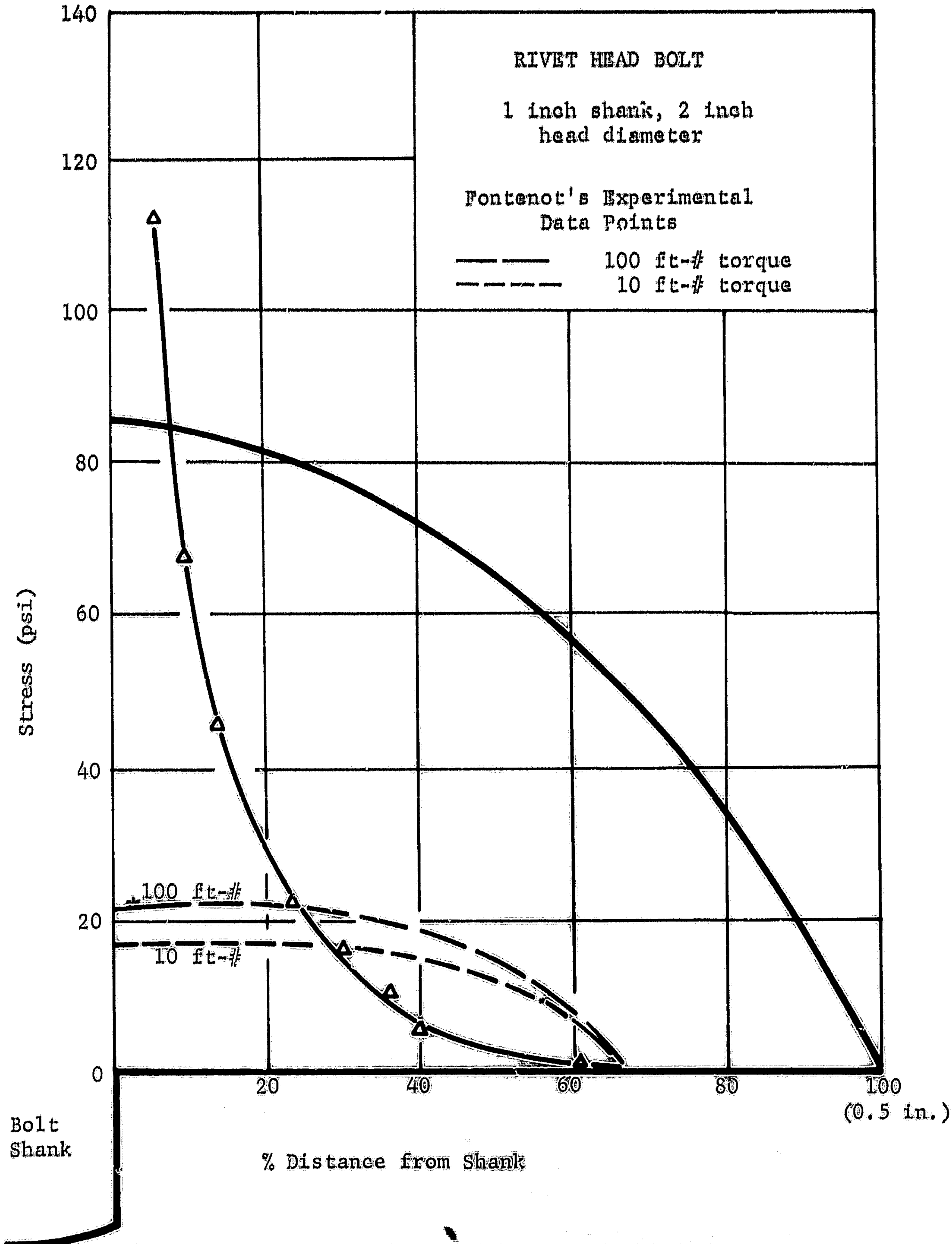


Figure 12. Magnitude of Normal Stress vs Radial Distance from Shank at the Interface (Light Load).

(Figure 11), it is noted that in the region near the shank, the stresses are nearly all normal to the interface, but toward the outer edge, the direction of stress varies away from the normal. This deviation is probably caused by the increased deflection of points toward the outer edge away from the natural configuration of no load. Inspection of the new values calculated in this manner with the originally measured values show that the difference is very limited. The maximum deviation of the normal stress is only 1.4% of the minimum value of stress, therefore the measured values of stress appear to be justified in this type of analysis of normal stress.

Magnitude of Stress

The magnitude of stress at the bolt edge appears to be a combination of two curves. The portion of the curve near the outer edge could be approximated by a straight line. The portion of the curve near the shank appears to rise along a straight line which has the inverse slope of the portion near the outer edge. These two straight line portions can be joined with a smooth curve producing a hyperbola or somewhat expontential type curve. The end points of the curve are not clearly defined in the analysis due to time-edge effects of the plastic at the outer edge and insufficient magnification of the dense fringe area at the fillet.

A straight line approximation to the outer edge portion of the curve intersects the vertical stress axis at nearly the same value of stress as does Bumgardner's analytical solution. Considering this value of stress as the value which would have been obtained if there had been no stress concentration at the fillet and applying Peterson's (5) stress concentration multiplication factor, a value of the stress near from normal. This, as shown for the fillister head, is not a critical amount of error when the final curves for stress magnitude are plotted.

Magnitude of Stress

Figure 10 shows the directional fringes removed by use of circular polarized light. The remaining magnitude fringes on the right side (light load) will be studied in this section. From this is plotted the stress distribution shown in Figure 12 Fontenot's curves for this same bolt with loading torques from 10 to 100 ft-# are shown. For the stress distribution plot, it is seen that the entire interface area is not carrying load. Approximately two thirds of the interface is void of any stress.

The point at which the stress became zero compares very well to Fontenot's point of zero stress. The point of zero stress at the line of analysis is located closer to the outer edge than at the interface. This is a continuation of the trend noticed in the fillister head bolt that values of stress increase uniformly along any normal line to the interface for a short distance above the interface. This trend is then reversed and stress gradually decreases up to the top of the bolt head.

If Fontenot's curve were corrected by using an ideal load situation in which the total load to the shank is divided by the area of contact at the interface then the curves would be upward. This would appear to be a much closer fit for his experimental data.

The photoelastic curve at the line of analysis matches favorably with Fontenot's curve over the portion near the outer edge. As in the case of the fillister head, the portion of the two curves near the shank differs greatly.

## BUTTON HEAD BOLT (HEAVY LOAD)

### Direction of Stress

The left side of Figures 9 and 10 contain the dense fringe pattern which represents a heavy load. The plot denoting direction of stress in this case (Figure 11) is normal to the interface except for a maximum deviation of -5 degrees. This is the maximum deviation of all the cases investigated. The entire interface has a small fluctuation of direction of stress between 0 and -5 degrees from the normal. The negative sign denotes that the direction of stress is symmetrical about the center line of the shank. The sense is toward the center line of the shank. This is explained by noting that the outer edges of the head deflect upward when loaded. In this position the principle stress will be aligned with the direction of force applied to the shank. This is explained by noting that the outer edges of the head deflect upward when loaded. In this position the principle stress will be aligned with the direction of force applied to the shank. This is the configuration in which the stresses are frozen into the bolt. When the load is removed, the outer edges return to the natural configuration, thus rotating the direction of stress toward the center line of the bolt. The resulting deviations of direction from the normal are small enough to allow these values of stress to represent normal stress.

### Magnitude of Stress

The heavier load has moved the point of zero stress further toward the outer edge of the bolt head. The area now carrying load is approximately three quarters of the total area at the interface. The same general characteristics are observed for the heavy load as for light load. The values of stress for this case, as expected, were greater at every point along the interface than the values taken for light load. The point of zero stress is now different from the one determined by Fontenot, but otherwise, the portion of the curves near the outer edge compares favorably.

In comparing the light and heavy load stress distribution for the button head bolt at the interface, it is seen that:

1. The point of zero stress is moved closer to the outer edge when a heavier load is applied.
2. The lighter load has a steeper rise of stress in the area of stress concentration.
3. Each point at the interface for the heavy load has a greater value of stress than the matching point on the light load side.
4. Except for the point of zero stress the heavy load showed better correlation to Fontenot's curves away from the shank than did the light load.

## CONCLUSIONS

To summarize, it may be concluded as a result of this study that:

1. The rivet type and the fillister bolt heads exhibit hardly any normal stresses in the last 25% of the radial distance from the shank.
2. The normal stress distribution for both fillister and rivet type bolt heads are in general an expontential decaying type shape. The assumption that stresses are nearly uniform is invalid.
3. Redesign of present bolt head configurations to insure a more uniform distribution might be worthwhile from a weight saving and reliability viewpoint.

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