Photoelastic investigation of the stresses in a capless connecting rod

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PHOTOELASTIC INVESTIGATION OF THE STRESSES IN A CAPLESS CONNECTING ROD

BY

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A

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PREFACE

For a great many years the primary method used to solve engineering problems was by the use of mathematics. However, rigorous and simple mathematical solutions are only possible if the geometry of the member and its loadings are relatively simple. When irregular shapes with complex loadings are encountered, rigorous mathematical solutions for stresses induced are either impossible or very cumbersome. In such cases, the engineer must resort to other methods which are available today.

This paper attempts to show the application of the photoelastic method to such a problem. Moreover, it will show the comparative ease with which the designer can take a basic machine part, vary the dimensions, and in a short time arrive at some shape that will do the work, and do it well, by the use of the photoelastic method.

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NOTATIONS

T_{max}	Maximum of shear stress
P	The algebraically largest principal stress
P	Increment of P
Q	The algebraically smaller stress
psi.	Pounds per square inch
O. C.	The optical Constant
f	Fringe order
f	Increment of fringe order
S_t	Tensile stress in psi.
A	Area in inches squared
F	Load in pounds
F	Increment of load
	Bearing angle
L	Length of model
D	Big end bearing diameter
A	Wall width, large end of connecting-rod
C	Small end shank width
B	Large end shank width
R	Fillet radius

INTRODUCTION

The engineer's purpose in designing tools, machinery, structures, etc., is to make the fullest possible use of the materials which are used in their manufacture. In order to do this, he must know the stresses produced in the member when it is doing the work for which it was designed. When these stresses are known, he can then add material or remove material at the critically loaded sections to help relieve stress concentrations at these danger points.

The author's problem was to design and investigate the stresses in a capless connecting rod. The use of the capless connecting-rod in internal combustion engines is relatively new and its use is, of course, restricted to two-cycle, fairly high compression engines. This is because such an engine would rely on compression to return the piston and rod to the bottom dead center position in the event of a misfire. In the ordinary case, the expanding gases would do this as every downward stroke in a two-cycle engine is a power stroke.

As the author mentioned earlier in this paper, the mathematical solution for stresses in members of irregular section, or members subjected to complex loadings is very difficult and cumbersome. Other methods such as rubber models, strain guages, plaster models, and Lueder's lines are many times more inaccurate and troublesome to use than the photoelastic method.

The basis of the photoelastic method is the optical phenomenon known as double-refraction. It is an experimental fact that an isotropic body, such as glass or celluloid, when stressed is double refracting. It is also an experimental fact that the relative retardation between the ordinary and extraordinary ray produced by the stressed condition at a point in an isotropic body is proportional to the difference in the principal stresses at that point. (1)

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(1) F. G. Seely, Advanced Mechanics of Materials, Vol. 1, p. 203.

The results of photoelastic investigations are, in general, applicable directly to prototypes of steel, brass, and other isotropic materials regardless of the difference in the physical constants of the model material and the prototype material. A mathematical proof and discussion of the above statement may be found in the second edition of The Theory of Elasticity by Timoshenko and Goodier on pages 116-125.

HISTORY

In 1816 David Brewster discovered that when a piece of glass is stressed and viewed by polarized light passed through it, a pattern is set up because of stresses in the glass. At that time, he suggested that these color patterns might serve for the measurement of stress in engineering structures. A glass model of these structures could be made and viewed in polarized light under various loadings. His suggestion went unheeded for nearly a century.

The first application of the Photoelastic Method was made by C. Wilson in 1891 in the investigation of stresses in a beam with a concentrated load. In 1901, A. Mesnager used this method in an investigation of arch bridges.

The science of photoelasticity moved from its formative stage to the practical level largely through the persistent and life-long labors of Professors E. G. Coker and L. N. G. Filon, both of the University of London. Their Treatise on Photoelasticity was published in 1930 by the Cambridge Press. Since then there have been great improvements made on technique and materials, and the science has been extended into the three-dimensional domain. Today photoelasticity is one of the cheapest, most accurate and versatile of the methods available to the stress analyst.

MATHEMATICS

The mathematics required of the stress analyst using the photoelastic method are very elementary. The very essential equations are listed as follows:

$$T_{max} = (P-Q)/2 \qquad (1)$$

$$T_{max} = (O.C.)(f) \qquad (2)$$

$$S_t = F/A \qquad (3)$$

Equation (1) above gives the relationship of the principal stresses to the maximum shear stress. The notation used is as follows: $\underline{T_{max}}$ is equal to the maximum shearing stress, $\underline{P}$ is equal to one principal stress and $\underline{Q}$ is equal to the other principal stress.

Equation (2) is the fundamental equation of photoelasticity. O. C. is the notation used for the property of a photoelastic material known as the optical constant, $\underline{f}$ is used to denote the fringe order at a point. A rigorous proof of equation (2) is much too long to be included in a paper of this type. M. M. Frocht in his book, Photoelasticity, Vol. 1, devotes the greater part of 46 pages to a discussion and proof of that equation. This discussion can be found on pages 129-175 in the above mentioned book.

Equation (3) above is the basic Mechanics of Materials equation relating Tensile stress to load and area for the case of pure tension. The notation used is as follows: $\underline{S_t}$ is equal to the tensile stress in psi, $\underline{F}$ is equal to the load in pounds and $\underline{A}$ is equal to the area in square inches.

Equation (3) is discussed in the first few pages of even the most elementary mechanics of materials books. Equation (1) above is derived and discussed in almost every advanced text on mechanics of materials the author has had the opportunity to examine. An especially clear-cut discussion and derivation of equation (1) can be found in Chapter 2, Stresses and Strains at a Point, in the book Advanced Mechanics of Materials by G. Murphy which was published and copyrighted in 1946.

Equation (3) is used to determine the optical constant for the material used in photoelastic investigation. The procedure used to obtain this constant is discussed later in this paper in the section "Problem and Procedure".

PROBLEM

AND

PROCEDURE

The author's problem was "The Photoelastic Investigation of The Stresses in a Capless Connecting-Rod". To the best of the author's knowledge and belief, the photoelastic method has never been applied to this particular problem.

The capless connecting-rod has the advantages of eliminating the need for rod bolts, nuts and cap, and also the elimination of the need for taking up the wear in the bearing, as this would be taken up automatically in normal operation. A disadvantage proposed by the author is the difficulty that would surely arise in oiling the bearing in the big end of the rod. To lubricate the bearing properly, the usual pressure system could probably be used if the holes in the crankshaft were made small enough so that pressure would be maintained in the system at all times, even when the rod bearing did not cover the hole. This would be of prime importance, especially in multi-cyclinder engines.

Another problem that would surely arise is the problem of making the big end of the rod rigid enough so that a large portion of the available bearing area could be utilized. Referring to figure 1 on the following page, it can readily be seen that if the dimension d is allowed to increase only slightly because of an axial load, a good deal of bearing area will not be utilized.

In order to make something more rigid additional material, redistribution of material, or bracing are required, all of which are objectionable in a connecting-rod. However, in some cases the removal of material at certain sections of high stress will allow the member to deform slightly and distribute the stress over a larger area, thereby reducing the high unit stress.

In the problem at hand, the author was interested in trying this method to relieve a high bearing stress at the section labeled (1) in figure (1). It was proposed to drill one or a number of holes in this section and then study the effects of this on the rest of the stresses produced in the connecting-rod. This was done with very good results.

In order to facilitate the use of the photoelastic equipment available, a model length (L) of 4-1/2 inches with a corresponding bearing diameter D at the large end of 1-3/8 inches. The rest of the dimensions A, B, C, and R shown in the sketch below were varied to obtain the best possible shape.

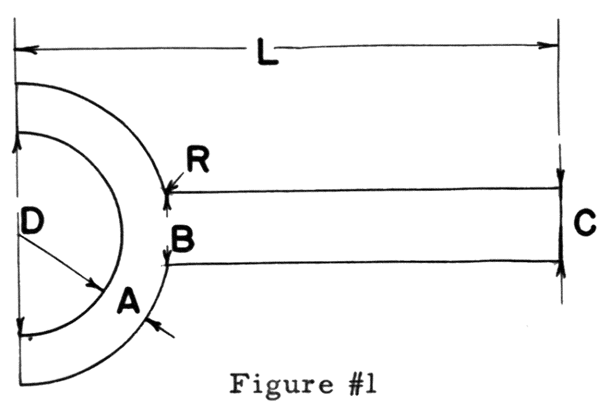


Figure #1

Dimension Lettering

The radius of the fillet R was varied on several of the preliminary models, but it was found that it had very little effect on any of the stresses; consequently, no photographs were taken.

The equipment used in preparing the models and testing them was all relatively new and in excellent condition. The model-making machine was a special high-speed portable milling machine purchased from the Chapman Laboratories at West Chester, Pennsylvania. It consisted of a high-speed Dunbar 1/10 horsepower electric motor mounted over a precision-ground worktable. The motor runs at 20,000 revolutions per minute, and the milling cutter used was of tungsten carbide, 1/4 inch in diameter and having 48 flukes. This machine required that a template, 0.100 inches in thickness be made for each model. The template can be made of brass, aluminum, or steel. Aluminum is the preferred material, as it can normally be purchased in a thickness of exactly 0.100 inches and is very easily machined. However, because of the temporary short supply of aluminum, the author was forced to use steel for the templates which made their manufacture considerably more time-consuming and difficult.

The polariscope used was manufactured by the Polarizing Instrument Company, Inc. of New York City. It was equipped with two polaroids, two quarter-wave plates, a mercury-vapor light, a Wratten-77 filter, camera, and a guillotine type loading frame. The camera was of the bellows type and was equipped with a standard 5 X 7 film pack holder.

The photographs taken during this problem were made with Kodak Super-Ortho Press film with an exposure of .25 seconds. The film was developed using Kodak Dextol developer, Kodak Stop Bath, and a Kodak fixer requiring 10 to 15 minutes fixing time. The photographs were never printed actual size but were enlarged approximately to three to four times the size of the actual model and about five times the size of the photograph taken. This high-enlargement ratio was to facilitate the interpretation of the stress pattern obtained.

The enlarger used to print the pictures was an Automega, using f 4/5, 190 mm Wollensak lens. The printing paper used was single weight Kodak Kodabromide F-4 paper and required an exposure time of from 20 to 30 seconds to get high contrast pictures.

The all important optical constant of the model material used was obtained by using a stepped tension bar which was first loaded, using water for a loading medium, to a low unit stress and then was slowly loaded by adding water to the loading container and weighing the water added. The use of the stepped tension bar gives three simultaneous checks for the photoelastic constant and is the most accurate method available. The shank widths used in the tension model were 3/8, 1/4, and 3/16, respectively, giving stress relationships of 2, 3, and 4, respectively, for any given load. Then according to the stress optic law the fringe order will have the same proportion as the stresses, and the photoelastic constant can easily be obtained.

The optical constant will be obtained later in this paper in the section titled "Analysis of Photographs and Data".

A sketch of the guillotine type loading frame and the method of loading the connecting-rod model is shown below as figure (2).

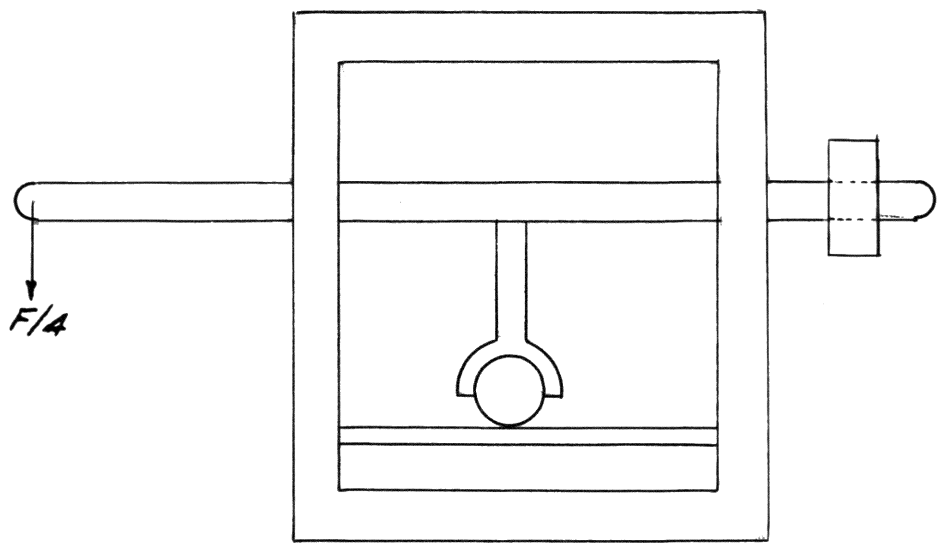


Figure #2

Guillotine Type Floating Frame

The optical constant must be determined before the analysis of the photographs may be undertaken. If we refer to the data and sketch recorded on page we will note that the load on the tension member is not recorded directly, but rather the increments of the load. In this manner we eliminate a good portion of the error that would result from any error in balancing the loading bar. The first load was recorded when the model was uniformly dark. This tells us that there is an integral number of fringes present in each part of the model. While in the process of loading, a constant check was kept on the number of fringes coming on the small portion of the model. The total number counted was four. The above observation tells us that in the large, medium, and small portions of the model there are respectively two, three, and four fringes present. From the stress-optic law, we know that the next time we have an integral number of fringes present in all parts of the model, the fringes will again be in the same two, three, four, proportion. Simple multiplication tells us this will be when there are four fringes in the large part, six in the medium part, and eight in the small part of the model. The increment of load recorded to accomplish this was 61.0 pounds. For an additional check the model was then loaded until it was again uniformly dark which we knew would occur when there were six, nine, and twelve fringes in the respective parts of the model. The increment of load required to bring this condition about, was recorded as 62.8 pounds.

The actual calculations used to obtain the optical constant of the models used are shown below.

$$T_{max} = (P-Q)/2 \qquad (1)$$

$$T_{max} = 2(O.C.)\ f \qquad (2)$$

$$S_t = F/A \qquad (3)$$

In the above equations when we have the case of pure tensile or compressive stress Q is equal to zero.

Therefore, setting equations (1) and (2) equal, we obtain the following:

$$P = 2(O.C.)\ f \qquad (4)$$

or

$$P = 2(O.C.)\ f \qquad (5)$$

If Q is equal to zero and we have pure tensile stress, then S_t is equal to P.

Making use of the above statement, we obtain,

$$O.C. = F\ /\ 2A\ \ f \qquad (6)$$

Using equation (6) above and substituting in the correct values, we obtain the following results:

For the large part of model we obtain,

O.C. = 61/(2)(2)(.09375 = 162.9 psi.

The above value for the optical constant is checked exactly for the other two portions of the model.

The second check when calculated as above gives a value of 167.3 psi. for all three parts. To arrive at the most accurate figure obtainable, we average the two readings, which gives a value of 165.0 psi. for the optical constant. This is the figure that we used in analyzing the stress in the actual connecting-rod model. The value of 165 psi. is known as the model fringe value and if we refer again to the stress-optic law, we find that the number of fringes present in a model is directly proportional to the thickness. Using the above relationship, we know that since the model thickness used was 1/4 inches, the material fringe value, based on a thickness of 1 inch is 41.3 psi. This checks within a few psi. of the values obtained by other workers in the field for the optical constant of Columbia Resin (CR-39) when using the same wave-length light and equipment as was used in this investigation.

Now we may go on and analyze the stress in the connecting-rod model.

For the general discussion of the method used in obtaining the state of stress, we may refer to any of the photographs taken of the models. However, for the sake of clarity, we shall choose model #2, photograph #2.

At the section of the model that is in contact with the loading block, it can be assumed that the direction of one of the principal stresses is perpendicular to the tangent of the circle at the point of contact. Then if we project the area under compression down to a radial line we may compute the average stress on the area under compression from the relationship $S_c = F/A$.

To determine the portion of the bearing surface that is being utilized, we refer to the photograph mentioned on the preceding page. When the model is completely unloaded, it is uniformly dark, and from this we can reason that a portion of the model that remains uniformly dark throughout the loading cycle has zero stress in it at all times. If we construct lines, from the center of the bearing circle in the big end of the rod, that are tangent to the dark portion of the photograph, we may say that the angle between these two lines divided by 180 degrees and multiplied by 100 is percentage of bearing area utilized.

Then if we project this bearing area down on a diameter which is perpendicular to the main axis of the rod, we may compute the average compressive stress acting on this area by using the formula $S_c = F/A$. Now if we make the studied assumption that the maximum compressive stress occurs at the top of the loading circle and that the compressive stress varies uniformly from this maximum to zero at the point where the model is again uniformly dark, the stress distribution must be like the figure #3 on the next page, where the average ordinate must be equal to the average compressive stress as computed.

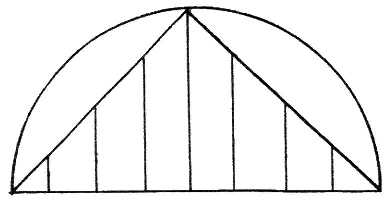


Figure #3

Stress Distribution on Connecting-Rod Bearing Surface

Since in a straight-line distribution of stress the average stress is equal to one-half the maximum stress, we may now compute the maximum compressive stress in our model. Then by using equation (1) and substituting this maximum compressive stress in for one of the principal stresses we may arrive at the complete state of stress at point 1 in the model under consideration. By going through the same procedure we may obtain the stresses at all bearing points. An example of the actual solution for point 1 in photograph #2, model #2 is shown below.

Calculations for photograph #2, model #2 are as follows:

Model Fringe Value f = 165 psi.

Model Thickness = 1/4 inches

Central Angle = 90 degrees

Load F = 320 pounds

Fringe Value at Pt. 1 = 13.5

Projected Area = 1.375(2 Sin 45)/2(4)

= .243 in^2

$S_c = F/A = 320/.243 = 1315$ psi.

$T_{max} = (O.C.)\ f = 165(13.5) = 2225$ psi. shear.

$P-Q = 2T_{max} = 4450$ psi.

But Q is equal to 2630 psi. because of the principles set forth in the paragraph on the preceding page.

Therefore:

$P = -4450 \quad 2630 = -1820$ psi,

$P = 1820$ psi.

The above completely defines the stress system at point 1, if we assume plane stress. The system is shown diagramatically below:

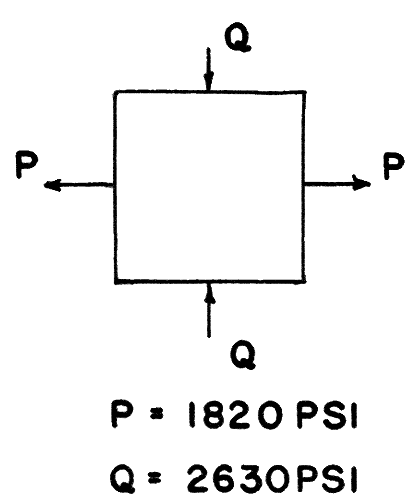


Figure #4

Some question may be raised in the preceding calculations as to why $\underline{T_{max}}$ was assigned a negative sign. A quick look at the photograph and we can come to the conclusion that there must be some tension at the bottom center of the model, because it would be acting somewhat like a very deep uniformly supported beam with a distributed load on its central portion.

The state of stress at the rest of the boundaries is very simple to determine, as they are completely free boundaries with no compressive stress acting normal to the boundary. Since the last statement is true, there remains only one principal stress at the boundary. Therefore, we can say that P = (O. C.) f (2). A sample of this calculation for point 2 in photograph #2, model #2 is shown below:

$$P = (5.5)(165)(2) = 1840 \text{ psi.}$$

Another quick look at the photograph shows that because of the method of loading the specimen there can only be compression at that free boundary.

No more calculations for the stresses at the different points on the body will be shown; however, diagrams will be shown of the stress systems, and each diagram will be referred to some particular point on the various photographs.

If a thorough study is made of the preceding stress diagrams, we may arrive at several conclusions. The most notable conclusion is that there is little stress concentration at point 2, which is a point of sudden change of cross-section. The fact that this is true was demonstrated to the author's satisfaction in the photoelastic laboratory.

Pictures of model #2 and diagrams of the states of stress in this model are shown for three loadings, all of which show this lack of stress concentration, and this lack is reflected in the photographs of all other models. The higher of the three loadings was 400 pounds, and even at this very high loading there was no serious stress concentration at point 2.

Models number 3, 4, and 5 show a vivid, graphic picture of the compound effects of a stress-relieving hole on the stress in other parts of a body. Models 3 and 4 differ only in the vertical dimension between point 1 and the hole. In model #5, the diameter of the hole is 1/32 of an inch larger than in models 3 and 4 and is in the same position as it is in model #4.

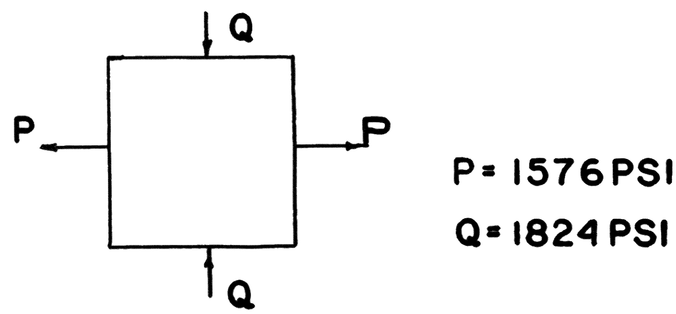
If a comparison at points 1, 2 and 3 is made among the last three model, it can be seen that we are accomplishing a certain amount of stress-relief at point 1- in every case, but that this relief is at the expense of raising the stress around the hole at point 3. The stresses at point 2 in each case remain undisturbed.

It can be noticed that the stresses at only three points in the body are calculated and that these points are on the surface. This is because the surface stresses are in general the greatest stresses. The reason for only using three points is that there were only three critical points, all other points were subjected to stresses of equal or lesser magnitude.

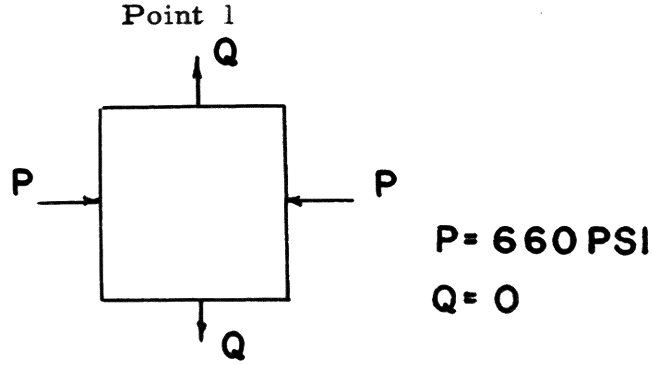
ANALYSIS OF PHOTOGRAPHS AND DATA

States of stress at point 1 and point 2, model #1, photograph #1 are shown below:

Model fringe value f	= 165 psi.
Model thickness	= 1/4 inches
Central angle	= 100 degrees
Load F	≃ 240
Fringe value at point 1	= 10.5
Fringe value at point 2	= 4



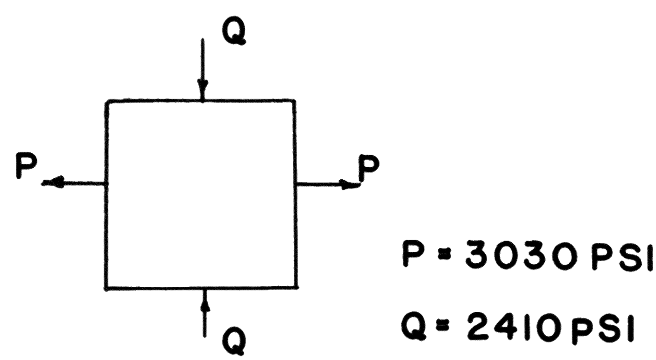
Point 1



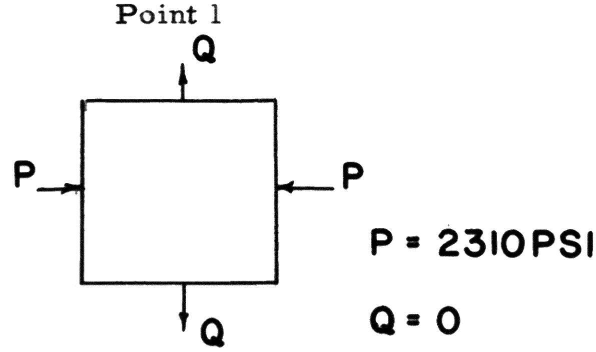
Point 2

States of stress at point 1 and point 2, model #2, photograph #1 are shown below:

Model fringe value f	=	165 psi.
Model thickness	=	1/4 inches
Central angle	=	100 degrees
Load F	=	400#
Fringe value at point 1	=	16.5
Fringe value at point 2	=	7



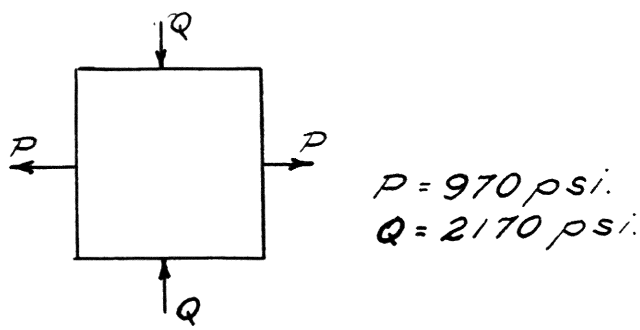
Point 1



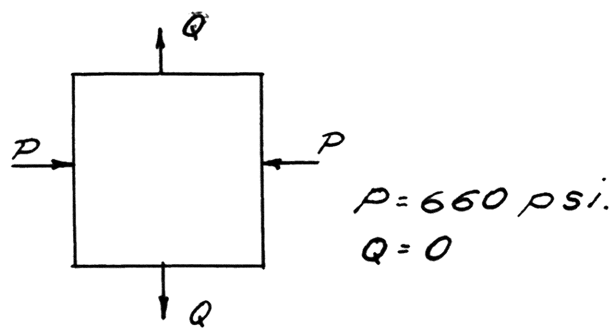
Point 2

States of stress at point 1 and point 2, model #2, photograph #3 are shown below:

Model fringe value	= 165 psi.
Model thickness	= 1/4 inches
Central angle	= 80 degrees
Load F	= 240
Fringe value at point 1	= 9.5
Fringe value at point 2	= 4



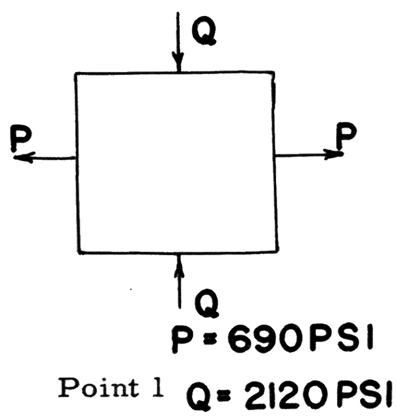
Point 1



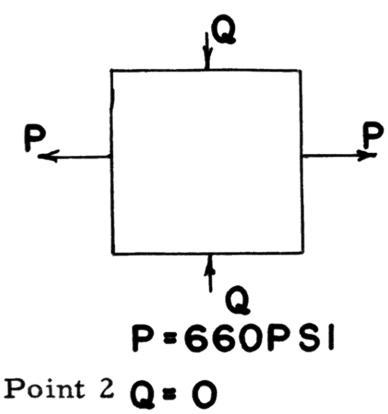
Point 2

States of stress at points 1, 2, and 3, model #3, photograph #1 are shown below:

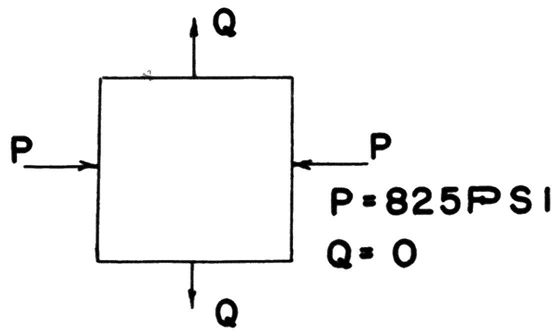
Model fringe value	- 165 psi.
Model thickness	= 1/4 inches
Central angle	= 85 degrees
Load F	= 240
Fringe value at point 1	= 8.5
Fringe value at point 2	= 4
Fringe value at point 3	= 5



Point 1



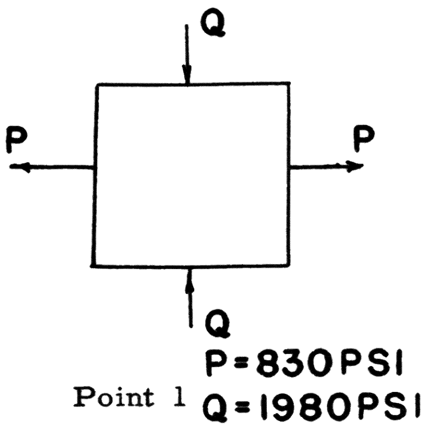
Point 2



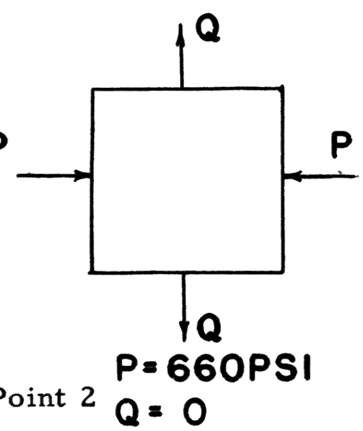
Point 3

States of stress at points 1, 2, and 3, model #4, photograph #1, are shown below:

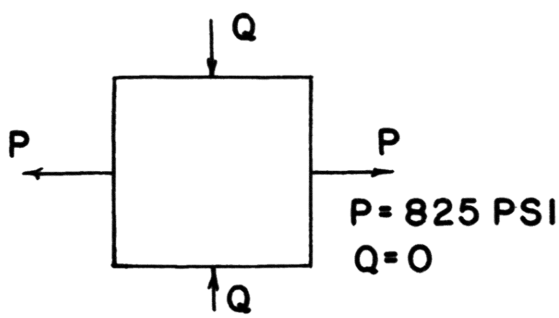
Model fringe value	=	165 psi.
Model thickness	=	1/4 inches
Central angle	=	90 degrees
Load F	=	240
Fringe value at point 1	=	8.5
Fringe value at point 2	=	4
Fringe value at point 3	=	5



Point 1



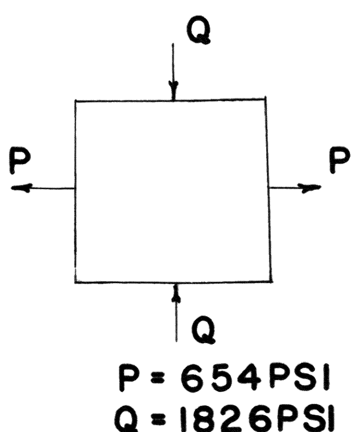
Point 2



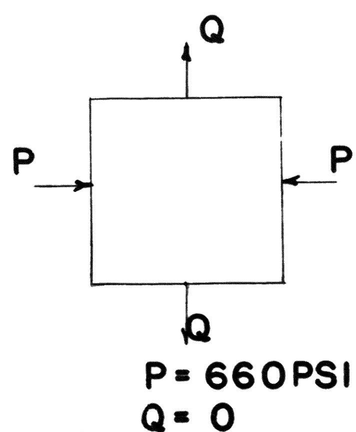
Point 3

States of stress at points 1, 2, and 3, model #5, photograph #1 are shown below:

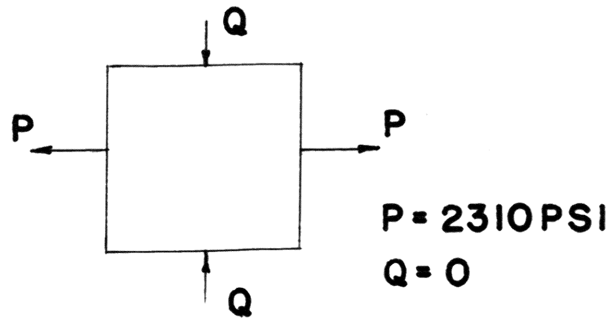
Model fringe value	=	165 psi.
Model thickness	=	1/4 inches
Central angle	=	100 degrees
Load F	=	240
Fringe value at point 1	=	7.5
Fringe value at point 2	=	4
Fringe value at point 3	=	7



Point 1



Point 2



Point 3

IN CONCLUSION

From the actual work involved in this problem and the difficulties encountered in using the photoelastic method on a design problem, the author makes the following observations and conclusions:

1. The photoelastic method as a whole is very easily understood and states of stress can be found very quickly. The so-called danger points, or points of origin of failure, are recognized at a glance in most cases.

2. The knowledge of photography and light needs to be very rudimentary.

3. Models can be made with a minimum amount of special equipment.

4. The only real difficulty encountered in making models is drilling stress-free holes without chipping the model, and this difficulty can be surmounted by using very sharp tools and by first drilling a very small hole and gradually cutting it out with larger and larger tools until the required size hole is obtained.

PHOTOGRAPHS AND DATA

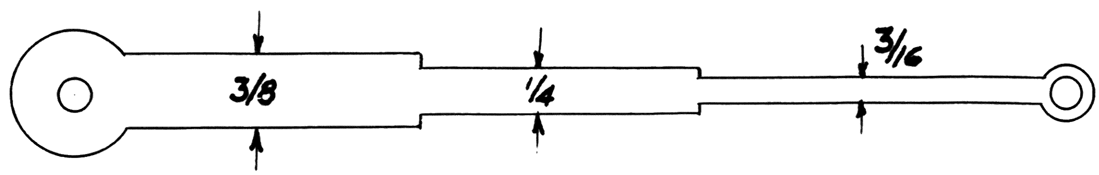
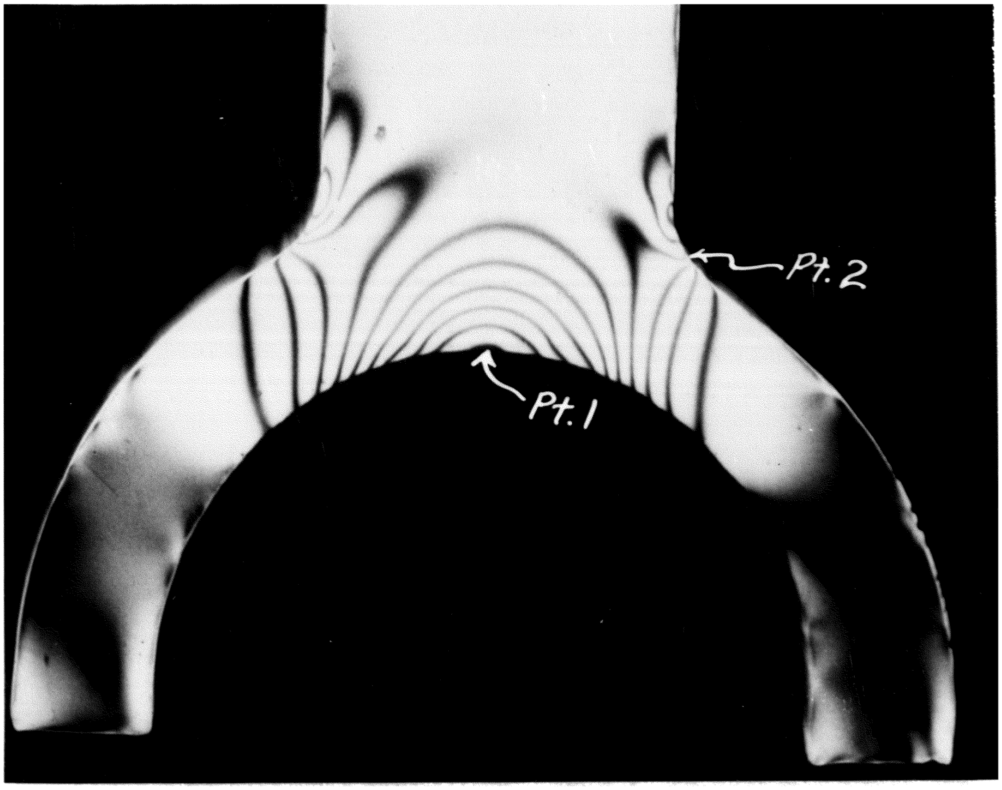


Figure # 5

Tension Model

Data used to determine the Optical Constant.

LOAD F	3/8 inch FRINGE ORDER	1/4 inch FRINGE ORDER	3/16 inch FRINGE ORDER
Initial load	TWO	THREE	FOUR
61.0 #	FOUR	SIX	EIGHT
62.8 #	SIX	NINE	TWELVE



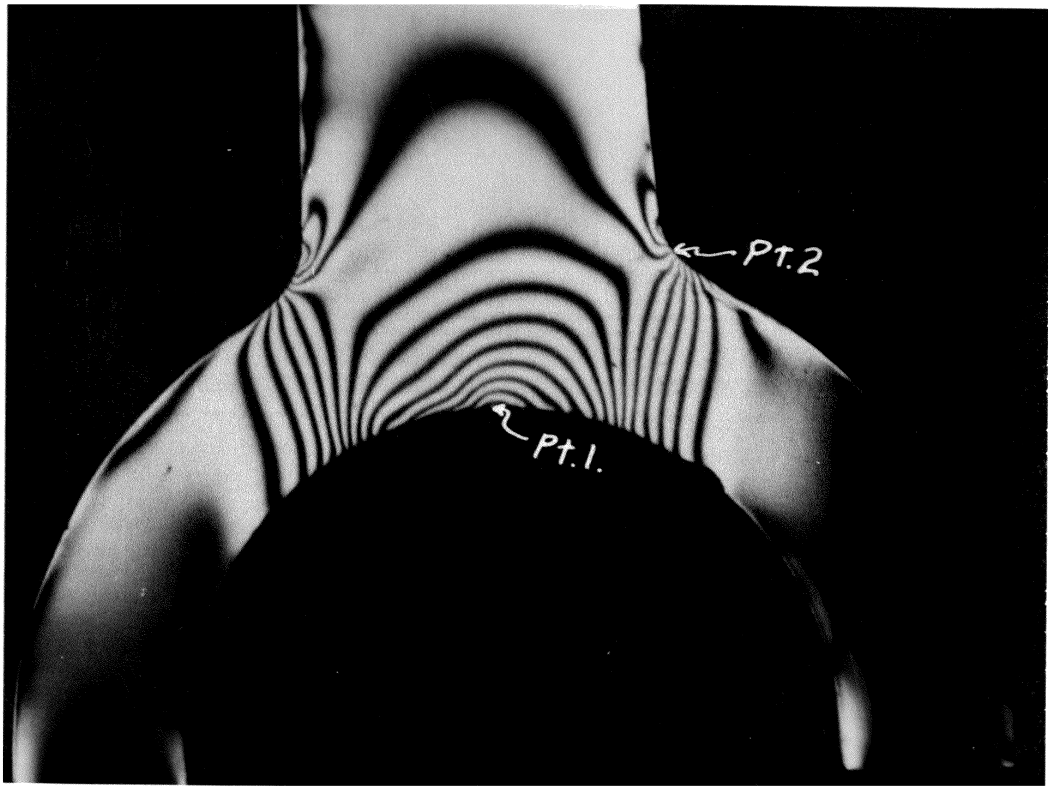
Model #1
Photograph #1

Length L	=	1/2 inches
Dimension C	=	3/4 inches
Dimension B	=	3/4 inches
Dimension D	=	1-3/8 inches
Dimension A	=	5/16 inches
Load	=	240#

Note: All dimension lettering in this section is referred to Figure #1.

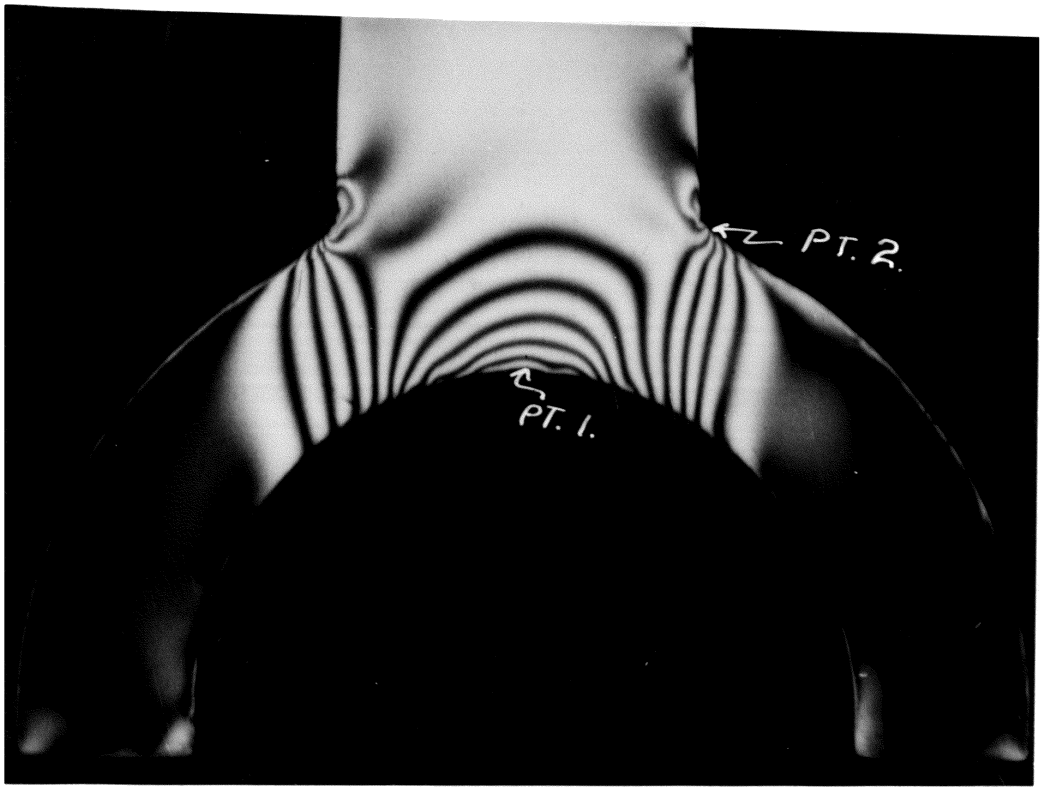
Model #2
Photograph #1

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension B	=	3/4 inches
Dimension D	=	1-3/8 inches
Dimension A	=	3/8 inches
Load	=	400#
Central angle	=	100 degrees



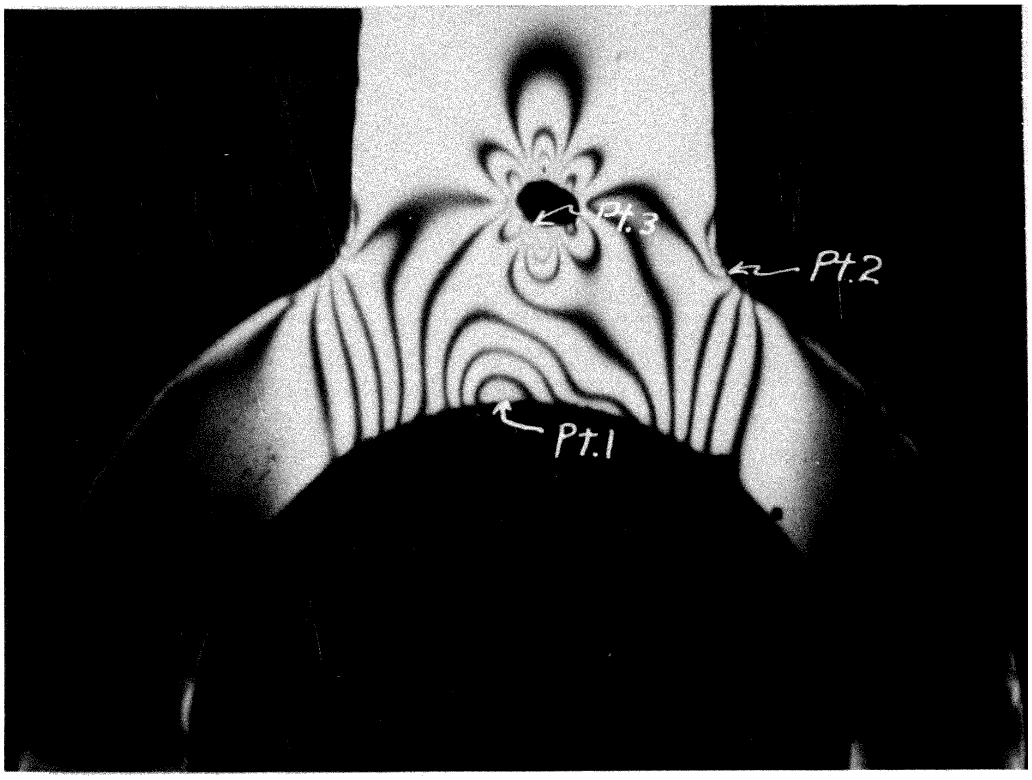
Model #2
Photograph #2

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension B	=	3/4 inches
Dimension D-	=	1-3/8 inches
Dimension A	=	3/8 inches
Load	=	320#



Model #2
Photograph #3

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension D	=	1-3/8 inches
Dimension B	=	3/4 inches
Dimension A	=	3/8 inches
Load	=	240#

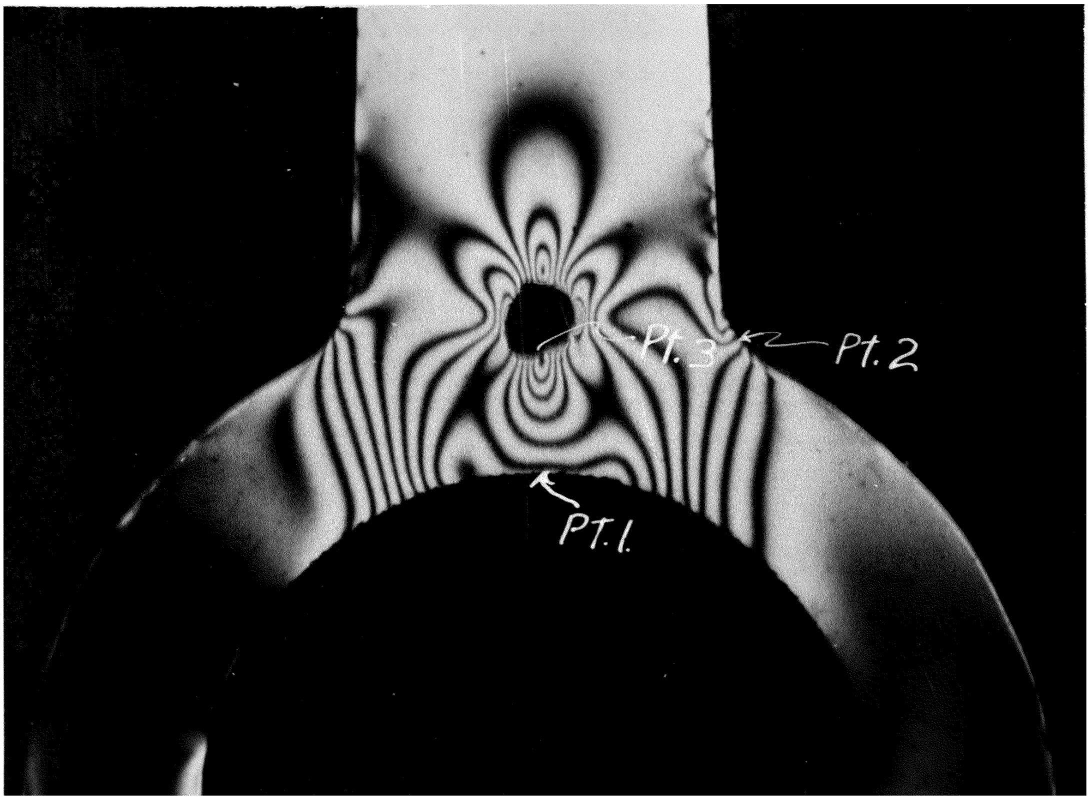


Model #3
Photograph #1

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension D	=	1-3/8 inches
Dimension B	=	3/4 inches
Dimension A	=	3/8 inches
Diameter of hole	=	3/32 inches
Load	=	240#

Model #4
Photograph #1

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension D	=	1-3/8 inches
Dimension B	=	3/4 inches
Dimension A	=	3/8 inches
Diameter of hole	=	3/32 inches
Load	=	240#



Model #5
Photograph #1

Length L	=	4-1/2 inches
Dimension C	=	9/16 inches
Dimension D	=	1-3/8 inches
Dimension B	=	3/4 inches
Dimension A	=	3/8 inches
Diameter of hole	=	1/8 inch
Load	=	240#

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VITA

Gerald Delaine Smith was born near Upland, Nebraska, on the 22nd day of January, 1928. He received his elementary education in the public schools of Kearney county Nebraska. He graduated from Minden High School at Minden, Nebraska in May, 1945. In September, 1945, he entered the Missouri School of Mines at Rolla, Missouri, and attended that school for one full academic year. In June, 1946, he joined the navy, and while in the navy served as a yeoman at Pensacola, Florida. Upon completion of his tour of duty in the navy, he returned to the Missouri School of Mines at Rolla, Missouri, where he received his Bachelor of Science in Mechanical Engineering in the year, 1951.