LOAD DISTRIBUTION IN SCREW-THREADS AS AFFECTED BY CLASS OF FIT

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By

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PREFACE

The small things are usually the most important. This is true with the common fastenings, bolts and nuts. Yet with an importance that has grown tremendously with the expansion of the automotive and aircraft industries, these parts are generally taken for granted.

Standardization of bolts and nuts within our own country and with other countries, as a result of the war, has led to many investigations of screw-threads. To make our products better and less expensive each item that contributes to the strength or failure of these parts must be investigated.

Therefore, with this conception, the manner in which the bolt and nut fit together becomes an item of concern. This manner of union is described by the class of fit. It is now necessary to arrive at a conclusion, from suitable test results, as to whether the class of fit contributes, reduces or has no effect on the strength of the fastening. This investigation was inaugurated with the purpose of arriving at such a conclusion.

As to the manner of testing, photoelasticity lends itself readily for this type of problem and, of the methods, the two-dimensional analysis was selected as best suited for this purpose.

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INTRODUCTION

In the early years of our country the manufacture of bolts and nuts was more of an art than an industry, being made by hand and fitted together. The "know-how" was passed on in the traditional manner, from journeyman to apprentice and from father to son. Machines of all sizes, from tiny watches to huge hydro-electric plants, are dependent on these fasteners.

One of the immediate problems that would be faced is:

> The tremendously rapid advance of invention and manufacture brought with it an expansion in the use of bolts and nuts to the point that they early became one of the principal mass-production parts and as such it became absolutely essential that they be interchangeable, not only as to parts made by one manufacturer, but as to those made by all manufacturers.[1]

This interchangeability has been emphasized immensely by the past war. It was through the efforts of the American Institute of Nut, Bolt and Rivet Manufacturers that specifications for thread dimensions and methods of gaging have been developed. These specifications have been universally used in this country and now as a result of the Unified Screw-thread Pact signed by the United States, Britain and Canada, the scope has been broadened to include other countries.

Consider some of the limitations encountered in the manufacture of bolts and nuts and the methods used to maintain specified tolerances. External threads may be obtained from two methods, cutting threads and rolling threads. The process of cutting threads is well-known; the threads are cut on a lathe or by chasers. The use of the chaser has certain inherent disadvantages. There is a constant wear (desgaste) on the thread cutting tools which necessitates setting the machine to cut as loose a bolt thread as allowed by the tolerance. In

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[1] A.E.R. Peterka, Bolts, Nuts and Screws. p. 56.

a lot this will give sizes that will cover the entire range permitted. For closer fits the operators must be extremely careful and their machines and products checked frequently. The cutting process will also leave small feather burrs and irregularities which will tend to indicate a closer fit than the actual.

Roll threading consists of two processes:

> A wire with a diameter the same as the desired outside diameter of the thread is first extruded to the pitch diameter and then the threads are rolled on the extruded portion using suitable dies that depress part of the steel to form the root of the thread and force the other material up to make the top of the thread. In the other method, the original diameter of the wire is larger than the desired outside diameter of the thread. This is first extruded to the desired outside diameter, then extruded to pitch diameter, and finally the threads are rolled upon the reduced wire.[2]

From this type of thread comes an increase in strength attributed to the cold working of extrusion and rolling. The lead, thread shape and size can be controlled very accurately in the tool room by checking with ordinary micrometers and, therefore, lends itself to a better control of tolerance and more uniform threads in lots. Mass production is another advantage of rolling threads.

The internal thread of the nut offers the greatest difficulties in manufacturing of tolerances. The limitations encountered are that:

> Nuts are tapped in a variety of machines with the use of solid taps. Since taps, for practical reasons, are made solid, there can be no adjustment of dimensions of the tap, such as is possible in producing external threads with adjustable dies. The accuracy of nut threads, therefore, is limited by the variations encountered in commercially available taps.[3]

In the commonly used tapping machines additional errors are encountered due to the process of threading. The nuts are threaded continuously by a slightly flexible tap. The cutting end of the tap is not rigidly held but

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[2] <u>Ibid.</u>, p. 46.

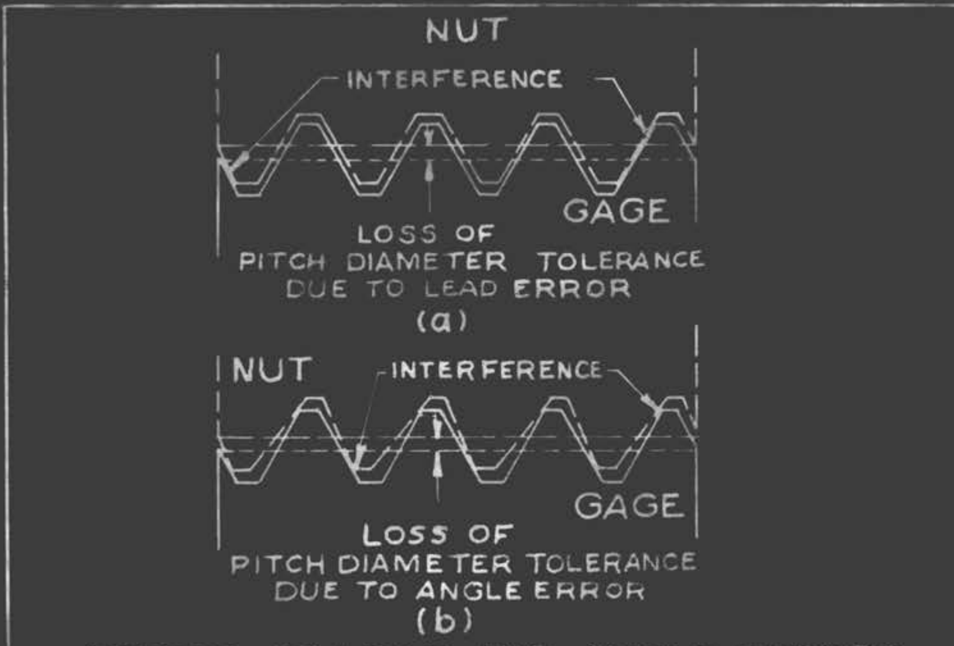
[3] <u>Ibid.</u>, p. 56

floats and centers itself through a series of nuts. To support this action there is only the flat-sided passageway. Lack of parallelism of the top and bottom faces and inaccuracies in the blank hole add more variations to the course of the tap.

Either for appearance or corrosion resistance many nuts are plated. This added material presents a problem in the control of tolerances before and after plating. To maintain a given fit the original thread must have a larger pitch diameter. This will also give a thread of two materials to resist the bending loads.

Finally, variations are to be had in lead of the thread and the angle between the sides of the thread. These errors affect the tolerances of the pitch diameter, which will be the basis for measuring tolerances. If these errors do exist, then the pitch diameter of the nut must be larger than the specified minimum to permit entrance of a "go" gage. Should the errors be excessive, the pitch diameter will exceed the specified maximum and the "not-go" gage will reject the nut.

A study of Figure 1a shows the interference encountered with a nut of perfect thread angle but with a slight positive lead fitted to a "go" gage having perfect angle and lead. For engagement to be without interference, the pitch diameter of the nut must necessarily be greater than that of the gage by some amount. This amount constitutes a loss in the available pitch diameter tolerance to compensate for this error. Figure 1b shows the effect of an error in thread angle and a larger pitch diameter is again necessary to compensate for the error. It was assumed that the gage was perfect and so any gage error in the opposite direction will increase the loss of tolerance more. The accuracy of a nut is, therefore, limited by manufacturing processes and the tolerances



EFFECT OF LEAD AND ANGLE ERRORS

FIGURE 1

in commercial taps and gages.

The object of gaging can be expressed in these words:

> The final results sought by gaging are to secure interchangeability, that is, the assembly of mating parts without selection or fitting one part to another, and to insure that the product conforms to the specified dimensions within the limits of variation establishing the closest and loosest conditions of fit permissable in any case, as provided for in the foregoing specification.[4]

This, then, requires at least two gages to maintain variations within the prescribed limits. The minimum looseness and maximum tightness in mating parts is governed by "go" gages, which control the extent of the tolerance toward the limit of maximum metal, and represent the maximum limit of the internal member and the minimum limit of the external member. This gage will prove that the nut is not too small to assemble if the gage will enter the nut all the way.

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[4] National Bureau of Standards, <u>Screw-thread Standards for Federal Services</u>. p. 46.

For many years successful manufacture has been carried out with "go" gages only. It becomes apparent that the maximum limits of the thread which are ordinarily of less importance than the minimum are controlled by the tap and that far more difficulties are encountered with nuts being too tight.

The "not-go" gage controls the extent of the tolerance toward the limit of minimum metal, and represents the minimum limit of the internal member and the maximum limit of the external member. Proof that the nut is not larger than some prescribed maximum limit is obtained from this gage. Since the "not-go" gage checks only the point of interference between the nut and the gage, no knowledge is obtained as in the case of the "go" gage, which checks all elements, and that no element of the nut thread is too small. With careful control of manufacture and suitable inspection from these and more recently developed gages, tolerances can be controlled to a high degree of precision and bolts with better classes of fit come into common use.

An understanding of the loads in screw-threads is essential to the fact that:

> Despite use of partially high factors of safety, there is continual recurrence of screw rupture failures, which shows that the usual and simple method of computation is in disagreement with actual conditions. An entire machine can thus be destroyed, as in the instance of a connecting-rod screw failure. A simple increase of the factor of safety is insufficient in many cases, and leads to an increase in cross-section, greater required volume, and increased material requirements.[5]

It is a known experimental fact that the highest stress occurs in the first thread groove at the bottom of the nut. Actually the point of maximum stress concentration is not at the root of the thread, that is, at the line of symmetry, but instead occurs somewhat below. The strain in the bolt and nut is principally tension and compression, respectively, which are primarily responsible for the

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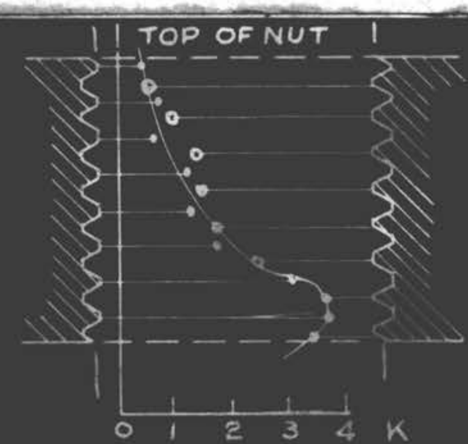
[5] Hans Jehle. "Polirisationsoptische Spannungsuntersuchungen" _Forschung auf dem Gebiete des Ingenieurwesen_ Au. 7 nl (Jan/Feb 1936) p. 19.

concentration at the base. Even with an ideally cut thread on both the bolt and nut, as soon as the load is applied the bolt stretches and the nut compresses so that even contact along the threads will not be possible and the load will be transmitted chiefly at the seat of the nut. Figure 2 shows the parabolic distribution of the concentration which is derived from axial deformations alone. The bolt and nut do not undergo the same tensile elongation and as a result the first thread must carry the largest share of the load.



MAXIMUM STRESS VALUES AT THE BOLT THREAD ROOTS $\sigma_{MAX.} = K\sigma_{AVE.}$

FIGURE 2

However, if the bending of the threads is taken into account the concentration will be substantially reduced. The bending of the two threads in contact is of cantilever action. Den Hartog has shown that the bending action of the threads is of the same order of magnitude as the difference in the axial deformations in the bolt and nut. Although the concentration will remain in the bottom threads, the upper threads will be enabled to carry more of the load.

In addition to these factors, circumferential stretch will also tend to moderate the extreme concentration. This action results in a stretch of the nut at the base, a contraction near the free end, and bending in the nut wall. The combination of these reactions, particularly the bending of the threads, will aid in redistribution of the part of the load but will present an additional stress concentration in the fillets of the threads, where the axial load has already produced a high concentration. Hence, the most favorable condition will be one in which the larger thread reactions are shifted toward the free end. This leads to the possibility that the difference of the size of mating parts, the class of fit, may contribute in some way to the redistribution of these high loads. There is an additional amount of material in the closer classes of fit on the sides, the crests and the roots, which may provide conditions which will affect one or more of the foregoing factors.

Next, a size of bolt and nut must be selected that will provide suitable results from which a conclusion may be drawn and which is used extensively in industry. The aircraft industry with its tremendous war-time mass-production provides the necessary size, No. 10-32NF. A reflection on the size of our air-power will illustrate the magnitude of use of this thread in the aircraft industry alone. This size of screw-thread is also widely used in the automotive and other industries.

No method exceeds photoelasticity as a tool of quantitative stress analysis in reliability, scope and practicability. The photoelastic method provides a complete exploration of principal stresses with speed, accuracy and small cost. Technique has been developed from the original two-dimensional analysis to the three-dimensional analysis which includes frozen stress patterns. Therefore, this method was selected as the medium of investigation.

The extent of this investigation will determine the type of photoelastic analysis to use. To insure that the class of fit is the only factor affecting any changes in the bolt model, all other conditions must remain the same. Obviously, axial deformations and thread bending will remain. Circumferential stretch and nut bending may be controlled by a suitable loading frame. The homogeneity of the material and consistency of machining operations will also show a definite effect on the results. For these reasons a single model with a means of varying the class of fit would give more comparable results. The single model can be used in two-dimensional analysis provided that the loading factors can be controlled. Therefore, a loading frame was designed which would laterally control the tolerances and any horizontal stretch, which replaces the three dimensional circumferential stretch. Vertically, an arm controls the nut bending.

"The most important step in any photoelastic investigation is to obtain a reliable stress pattern from a transparent model of the prototype."[6] To obtain suitable photoelastic results two models, a five and a ten time enlargement of the bolt and nut, were used. The smaller model was used to obtain the overall stress pattern while the larger model was used to get a closer inspection of the individual threads. The model enlargements were necessary to get images of sufficient size that could be analyzed without too great a photographic enlargement.

The sharp "V" thread was selected as the extreme of thread cutting and to yield more fringes so that changes will show more readily. Also, from the accurate 60° "V" milling cutters available, more consistent notches could be cut. The helical path of the thread was neglected so that the groove stress patterns would be more definite. However, the pitch was maintained. Since the length of the threaded section of a bolt is dependent on its overall length, the thread grooves were cut to extend slightly past the surfaces of the nut.

For comparison of the different stress patterns the maximum stresses are expressed as multiples of the average stress in the full section of the bolt.

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[6] M.M. Frocht, Photoelasticity Vol. 1, p. 149.

The specific purpose of this investigation is to conclude whether the class of fit of screw-threads contributes, reduces, or has no effect on the distribution of load.

APPARATUS

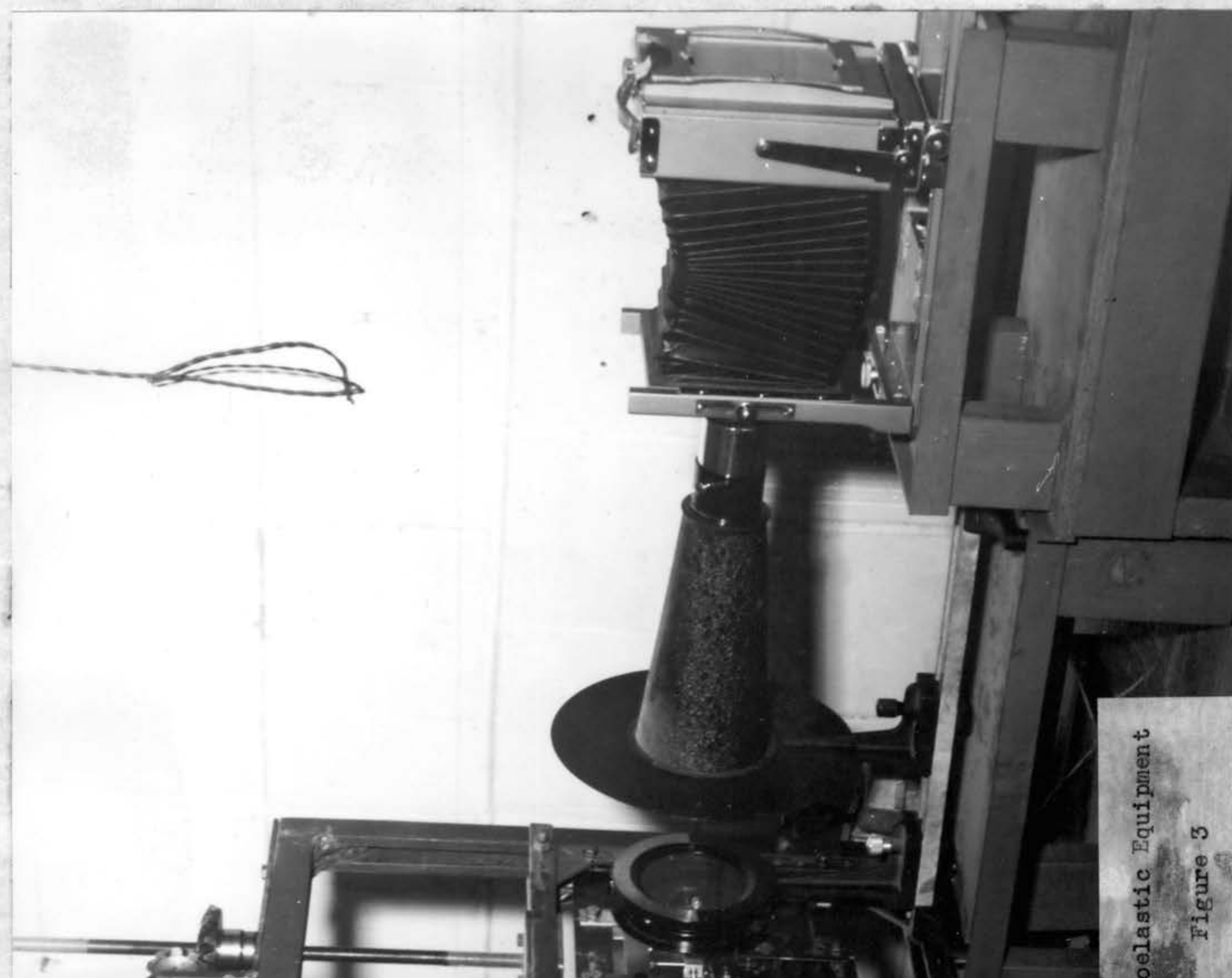
The principal apparatus for this investigation consists of a standard polariscope, bakelite models and a special loading frame.

THE STANDARD POLARISCOPE - The elements that make up the standard polariscope are shown in Figure 3. Reading from left to right are:

1. Two light sources; a mercury-vapor lamp with a filter to produce a monochromatic green light of wave length 5461Å; a white light with auxiliary lens.
2. The polarizer; made from Polariod plates which produce plane polarized light.
3. A quarter-wave plate that transforms the plane polarized light into circularly polarized light.
4. The loading frame for holding and loading the model.
5. A quarter-wave plate for reconverting the circularly polarized light back into plane polarized.
6. The analyzer, made from Polariod plates.
7. The condensing system to bring the parallel light rays to a sharp focus and eliminate distortion of the image.
8. The camera and lens; an 8x10 view camera with a 4x5 adapter back; a Wollensak Velostigmat f6.3. Exposure was at f11 at 10 seconds on Ansco Isopan film.

THE MODELS - Bakelite BT-61-893 was selected as the material best fulfilling most of the ideal requirements. It is water-clear, easily machined with ordinary tools and strong enough for practical handling.

The models were prepared in the following manner. The stock plate was observed in the polariscope to establish an area of uniform stress. The use of this area for the model would eliminate annealing. The part pieces of the



Photoelastic Equipment

Figure 3

model were cut ¼ inch oversize on a ten-inch band saw. This extra material would prevent the time-edge effect from reaching the final boundaries before the model was completed.

Factory machine marks and other scratches were removed from the model by sanding with 320x emery and crocus cloth. The final polish was obtained by using ferric oxide (jeweler's rouge) and water on a rotating disk covered with selvyt cloth. This finish left the model with an almost water-clear transparency.

The machinist cut the model to the final dimensions which are shown in Figure 4. The sides were cut with a 3/8 inch face milling cutter at 240 rpm. To reduce any internal stresses set up by high local temperatures a coolant of soluble oil was used in the cutting operations. The loading holes in the bolt stem were first drilled undersize and reamed at 375 rpm to final dimensions. Lastly, the thread notches were cut with a 2 3/4 inch 60° double angle milling cutter at 240 rpm which proved to be the optimum speed for least machining stresses.

The models were not annealed because of the high quality of BT-61-893 and the effects of annealing are not definite enough to warrant use of the equipment available. The stock piece used showed very little residual stresses. The physical properties of Bakelite BT-61-893 are as follows:

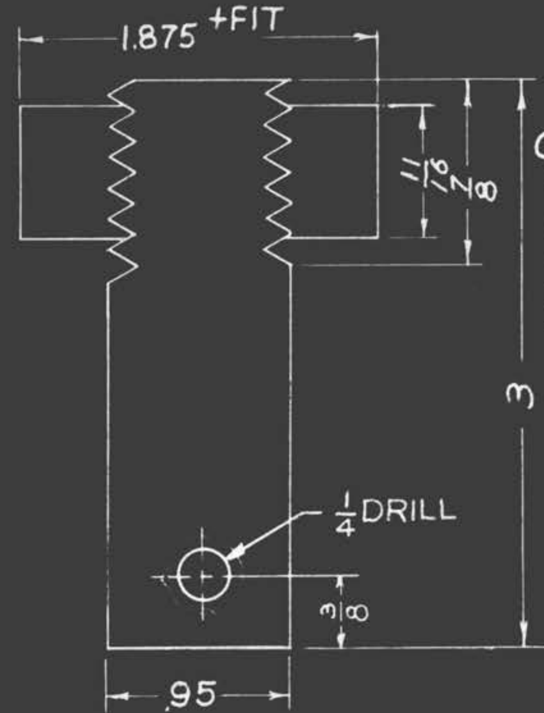
Ultimate strength (5 minute test)	17,000 psi.
Optical elastic limit	7,000 psi.
Modulus of elasticity	615,000 psi.
Poison's ratio	.365
Brinell Hardness Number	29.8
Fringe value, f, 1 inch thickness	43 psi.

A calibration test on the material gave a fringe value of 42.9 psi., slightly

#10-32NF BOLT & NUT

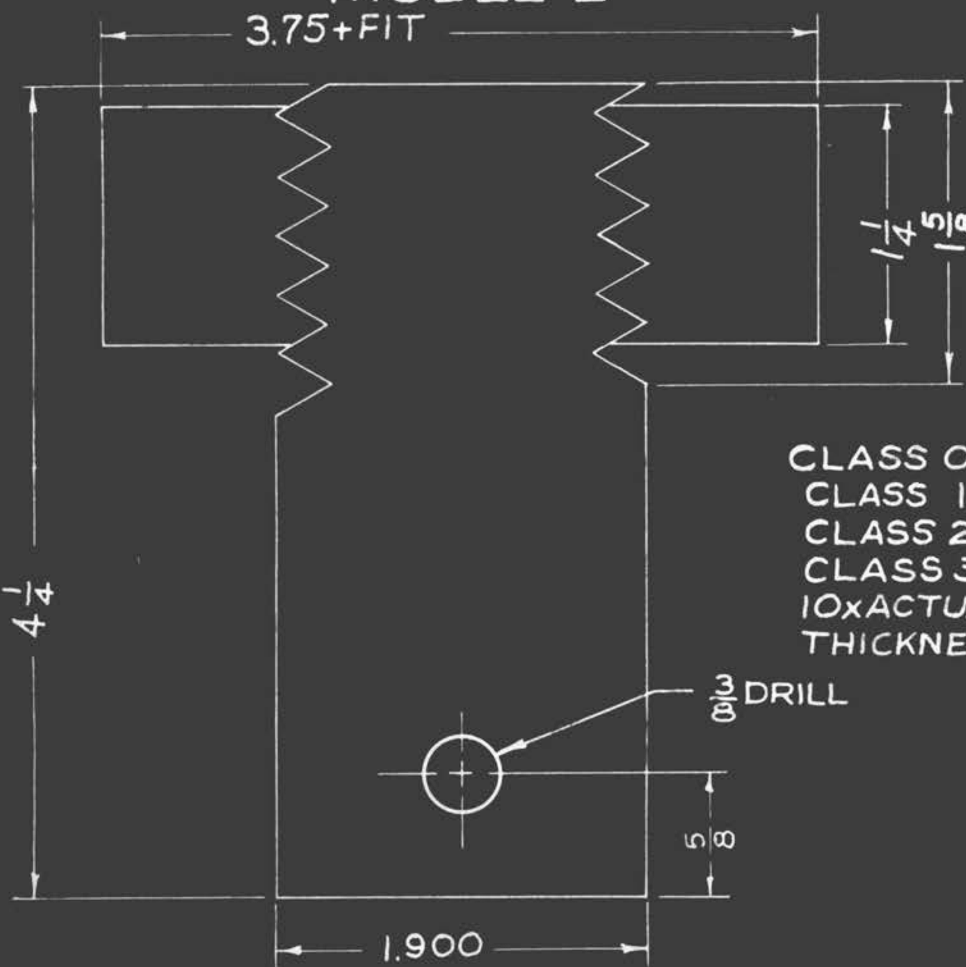
MODEL 1

BAKELITE BT 61-893



CLASS OF FIT
CLASS 1 +.0190
CLASS 2 +.0135
CLASS 3 +.0095
5xACTUAL BOLT.
THICKNESS .269

MODEL 2



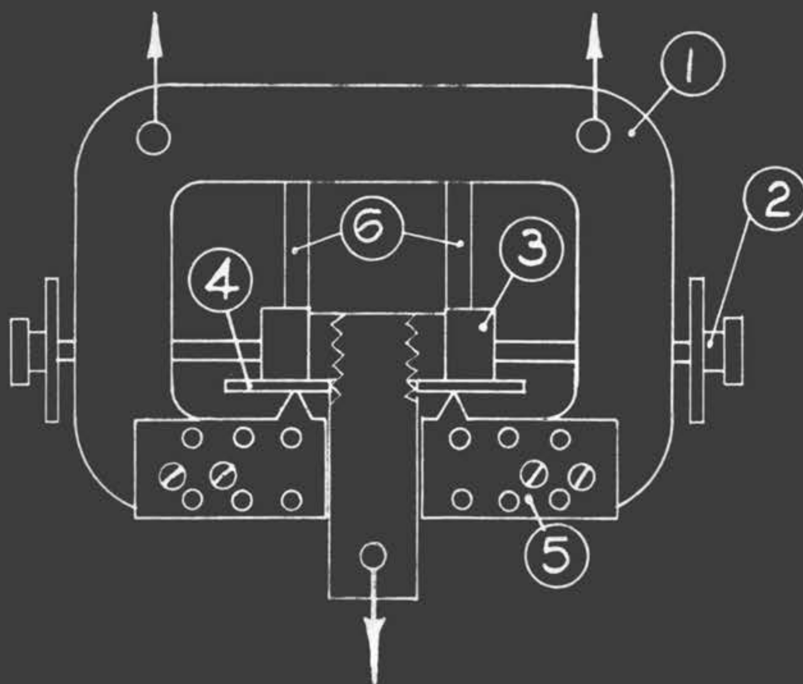
CLASS OF FIT
CLASS 1 +.038
CLASS 2 +.027
CLASS 3 +.019
10xACTUAL BOLT.
THICKNESS .269

FIG. 4

less than manufacturer's data.

LOADING FRAME - A drawing of the loading frame is shown in Figure 5 and the model is shown in place on the frame in Figure 6. The yoke is the main body of the frame and permits easy loading of the model. Knife-edge supports and bearing plates provide a uniform pressure on the underside of the nut and eliminate any error of alignment due to bending in the yoke. The supports are adjustable horizontally into another position to accommodate a larger model. Guides control the tolerances by means of a calibrated adjustable screw. This also limits the horizontal stretch. To counteract the bending in the nut wall there are bending-balance arms. These, along with the guides, will hold constant the forces comparable to those resulting from circumferential stretch.

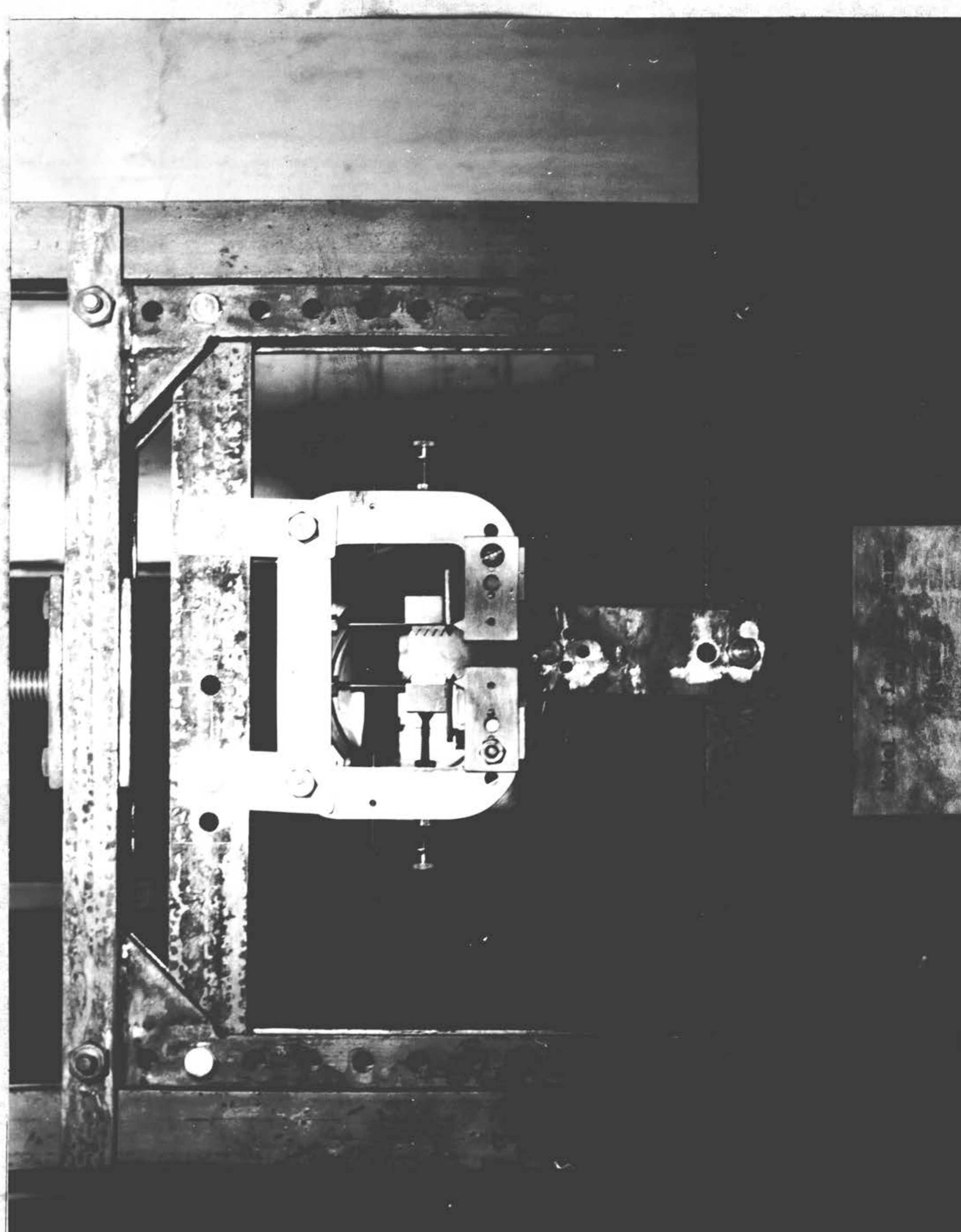
LOADING FRAME



MODEL 1 IN FRAME

1. YOKE
2. TOLERANCE CALIBER KNOB
3. GUIDE
4. BEARING PLATES
5. KNIFE-EDGE SUPPORTS
6. BENDING-BALANCE

FIG. 5



DETERMINATION OF THE LOAD DISTRIBUTION

TEST PROCEDURE - The standard polariscope, as shown in Figure 3, was set up and aligned. A background of maximum darkness was first obtained by setting the polarizer at zero degrees and adjusting the analyzer. Quarter-wave plates were next inserted, the first with its optical axis forty-five degrees to the principal plane of the polarizer while the second was manipulated to maintain the dark background. This set-up, with the monochromatic light, was used to obtain the stress patterns.

The model was placed in the loading frame, Figure 5, with zero fit. A mixed set-up, that is, a bright background, was used to focus and insure sharp, clearly defined edges and fringes. With a slight load applied to the model, the class of fit,[7] based on pitch diameter tolerances and listed on Figure 4, was measured and a check made to give simultaneous contact of all threads. Test loads were applied by means of a beam in the loading machine to give a satisfactory stress pattern with 8 to 10 fringes at the root of the lowest bolt thread in contact. This stress was below the elastic limit. A photograph of the stress pattern was taken at f11 with a 10 second exposure. For each of the classes of fit,[7] 1, 2 and 3, a photographic stress pattern was obtained.

Isoclinics, that is, the locus of points along which the principal stresses have parallel directions, were obtained next. In order to increase the distinctness of the isoclinics a white-light source is used instead of a monochromatic source. Also, the quarter-wave plates were removed to give a plane polariscope which gives an isoclinic superimposed over the stress pattern giving (p - q). The parameters of the isoclinics were varied by changing the axes of the polarizer and analyzer. A reflector box replaced the camera. This box contained a

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[7] National Bureau of Standards, op. cit., pp. 34-36.

mirror set at forty-five degrees and a glass plate on which to draw the isoclinics. At regular intervals of ten degrees, measured clockwise on the polarizer when viewed toward the source, the isoclinics were drawn.

To follow the flow of forces through the bolt and nut the stress trajectories were drawn. These lines of forces are curves, the tangents to which represent the directions of one of the principal stresses. They are constructed by graphical processes on the basis of the isoclinics. The process followed is outlined in Frocht's Photoelasticity.[8]

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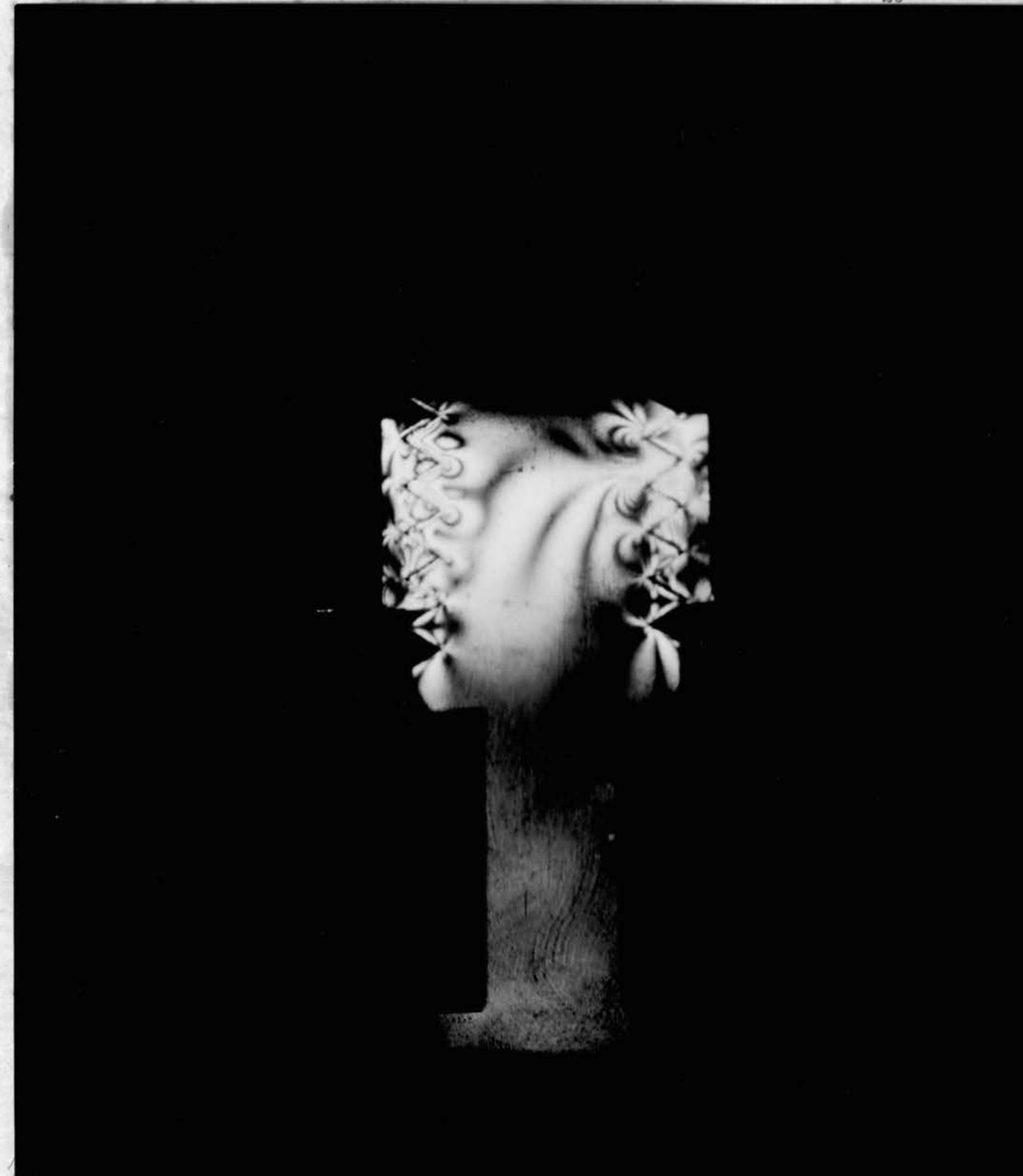
[8] Frocht, op. cit., pp. 198-201.

DISCUSSION OF TEST RESULTS - Prior to making the load application, an inspection of the model showed no time-edge effect but that some machining stresses were present. These stresses appeared as single light bands slightly inward from the thread edges. Their magnitude was small, particularly when compared to the local stresses of the applied load.

In all the photographs, higher concentrations of stress appeared at the second thread (numbered from the top). A closer inspection showed that the axial location of the left thread notch is in error. This would account for the threads coming into contact earlier. Since the loading surfaces of the threads are on different levels due to pitch, it would be expected that the right-side threads would carry more of the load. This is clearly indicated in the photographs and, also, that the load eccentricity added by axial error is partially transferred to the right side. The amount of any shift of load would be diminished if the three dimensional aspect of the nut is considered. However, the stress patterns are consistent in this manner and, therefore, are of no immediate concern. The symmetrical pattern in the stem of the bolt indicates that the load was applied along the centerline.

A comparison of the two patterns for each fit shows only a slight difference due to changes in the amount of loading for distinctness. Only in the center of the thread section is this change pronounced by a slight shift of fringe line.

In Figures 7, 8 and 9, the stress patterns are not sharp in the immediate vicinity of the threads. This lack of sharpness destroys their value for a close comparison of the root concentration in each fit. However, a general comparison can be made in the center of the threaded section. Near the top the patterns are almost identical. The diagonal fringe shifted very slightly



Stress Patterns

Class of Fit 1

Figure 7

to the left at a class 2 fit. No changes were made in fits 1 and 3. At the bottom of this fringe in the lower left section, a raising of the fringe is noticed in both the class 2 and 3 fits. Also, in the center of the left section the larger fringes becomes fainter with lesser fit.

The prints of these three patterns were made very carefully. A large magnification and, consequently, a long exposure was necessary. These alone would be enough to reduce greatly the clarity of the print. Also sharpness would be reduced in the condensing system which proved to be unnecessary with a small model. For these reasons, Figures 7, 8 and 9 were used for qualitative purposes only.

An inspection of Figures 10, 11 and 12 shows that the fringes near the top are almost identical. It is noticeable that there is a lack of balance of the tolerance in the class 2 fit. That would account for some changes in the general pattern. There is an increasing tendency of the fringe, running from the first left thread to the second right thread, to lift toward the top. At the lowest left thread in contact, the fringes move slightly toward the bolt center with increasing number of fit.

The stress concentration factors, expressed as multiples of the average stem stress, are shown in the following table. The thread number indicates the thread in contact reading from top to bottom.

	Class 1		Class 2		Class 3	
Thread number	Left	Right	Left	Right	Left	Right
1	6	11	8	9	8	9
2	6	9	6	7	7	9
3	7	7	4	7	5	8
4	4	7	4	7	4	7

The root fringes for the class 2 fit were more difficult to read and, due

Stress Patterns

Class of Fit 1

Figure 10

Str[illegible] Patterns

Class of [illegible]

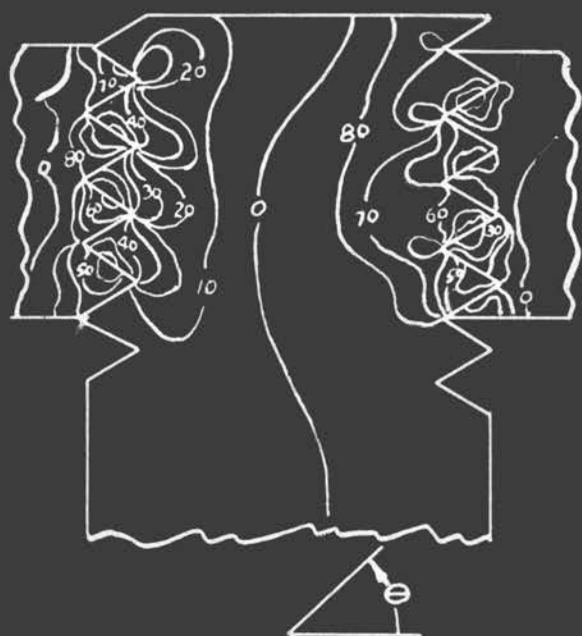
Figure [illegible]

also to the error in the balance of tolerances, the stress factors vary somewhat from the pattern. There is indicated a tendency toward increasing concentration near the top threads.

A plane polariscope and a white light were used in drawing the isoclinics. The general patterns in the vicinity of the threads was the same for all fits as shown in Figures 13, 14 and 15. The marked change occured in the center of the threaded sections. The tendency was for the isoclinics to lift slightly and move toward the center. There is always the possibility of error in drawing isoclinics since the dark pattern is not too sharp when drawn from a reflected image. Photographs would give best results since the stress patterns would remain the same and any change would be a result of the new isoclinic.

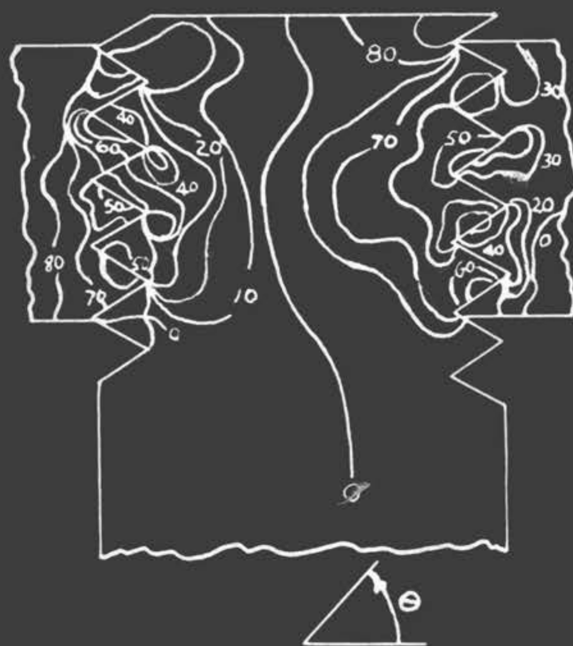
The stress trajectories are derived from the isoclinics. Therefore, any change in the isoclinics would result in a change of the stress trajectories. It is evident from Figures 16, 17 and 18 that these patterns show such an effect. The lines tend to move toward the center with increasing number of fit.

No stresses were determined since it is not within the scope of this investigation and there is sufficient information to arrive at a conclusion. Although the isoclinics were drawn for the nut, no mention has been made of its load distribution because its action is of three dimensions as has already been pointed out and the results would be erroneous. The advantage of using single models is shown here. If, for example, three models were used, one for each fit, the machining error that increased the stress concentrations in the upper threads might be interpreted as resulting from the class of fit.



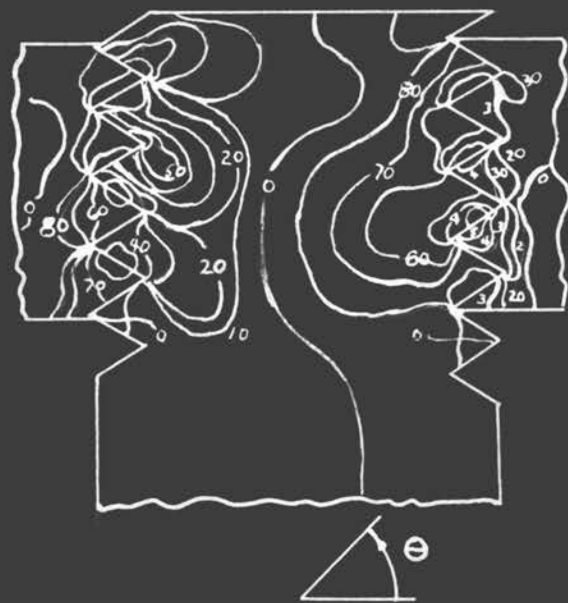
CLASS OF FIT 1
PITCH DIAMETER TOLERANCE 0.0190
LOAD 100LBS.

FIGURE 13 — ISOCLINICS



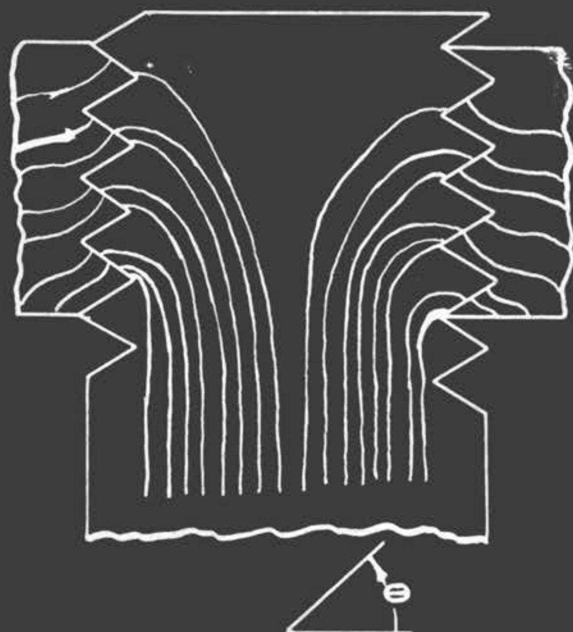
CLASS OF FIT 2
PITCH DIAMETER TOLERANCE 0.0135
LOAD 100 LBS.

FIGURE 14 – ISOCLINICS



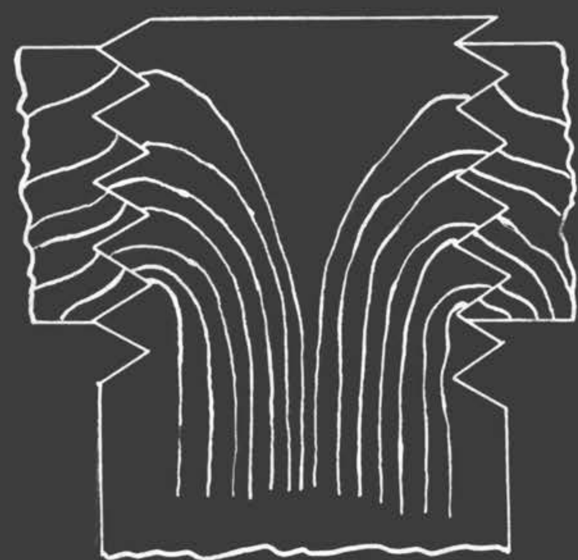
CLASS OF FIT 3
PITCH DIAMETER TOLERANCE 0.0095
LOAD 100 LBS.

FIGURE 15 - ISOCLINICS



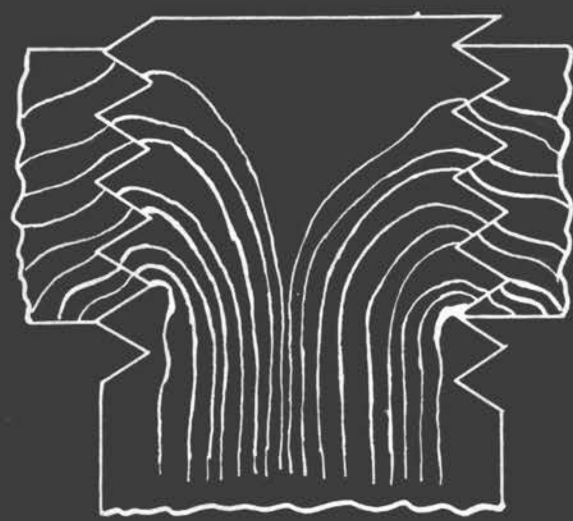
CLASS OF FIT 1
PITCH DIAMETER TOLERANCE 0.0190
LOAD 100LBS.

FIGURE 16 – STRESS TRAJECTORIES



CLASS OF FIT 2
PITCH DIAMETER TOLERANCE 0.0135
LOAD 100 LBS.

FIGURE 17 - STRESS TRAJECTORIES



CLASS OF FIT 3
PITCH DIAMETER TOLERANCE 0.0095
LOAD 100LBS.

FIGURE 18—STRESS TRAJECTORIES

SUMMARY AND CONCLUSIONS - The stress patterns, stress concentrations and the isoclinics indicate a definite effect on the distribution of the load. However, the magnitude of this effect may not be of sufficient importance to be used as an argument.

To change the stress pattern in the center of the threaded section requires a considerable additional load. It might be pointed out that for a fringe order change in the stem required eighty-two pounds. Not this much would be necessary in the threaded section but it would indicate that the gain of this small redistribution would not warrant a change of class of fit.

The stress concentrations showed only a slight trend toward moving the larger distribution upward. The fringes counted at the roots increase rapidly at the sharp notches but move inwardly slowly. Photographic or optical effects blur some of the fringes so that the very exact number could not be determined. The gain is a shift of one fringe order upward. If the fastener is to be used for impact loads or at elevated temperatures, then the increase of class of fit would be beneficial. In many cases the bolt and nut are plated. This would add a thin layer of metal which would round the root to a smooth curve and tend to reduce the concentration. The gain from class of fit would, therefore, be lost since the concentration shift would be evident only at high fringe orders, as in this case with sharp notches.

In the vicinity of the threads there is practically no change in the isoclinics. This area is critical and any redistribution of load would be beneficial. However, the gain from class of fit affects the center portion which is more capable of taking the load.

Figure 11 will illustrate the effect that a small error in the balance of the tolerance will alter the stress pattern considerably. Therefore, error in either lead or thread angle will more than offset a gain from class of fit. Plating may reduce these errors somewhat since the layer of metal could be worn to get better contact between the threads.

Although there is a small change in the distribution of load due to an increase of class of fit, the gain will not, in most cases, warrant its use as a means of strengthening the fastener. Only with very consistent and accurate manufacturing processes and certain conditions could class of fit become a good argument for load distribution.

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