INVESTIGATION OF INITIAL TIGHTENING LOAD IN BOLTS

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<u>**CERTIFICATE**</u>

Certified that the dissertation entitled " INVESTIGATION OF INITIAL TIGHTENING LOAD IN BOLTS " which is being submitted by Shri Bhupinder Singh in partial fulfilment for the award of the degree of Master of Engineering in Machine Design (Mechanical) of University of Roorkee is a record of student's own work carried out by him under my supervision and guidance. The matter embodied in this dissertation has not been submitted for the award of any other Degree or Diploma.

This is further to certify that he has worked for a period of 1st May '63 to 24 Jan '64 for preparing dissertation for Master of Engineering Degree at the University.

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C O N T E N T S

INTRODUCTION

Nuts and Bolts are amongst our commonest fastening devices. Tension bolts are widely used in Engines, machinery of various kinds and in steel structures. They have unique advantages in their contribution to ease of assembly and dismantling. They also give comparatively rigid and compact joints and from the standpoint of static strength are extremely efficient. In the modern structural practice they have completely replaced the rivets in connections being more efficient in resisting the shear loads on the structures.

Tightening nuts and bolts is a simple job, at least so it seems. But how much do we tighten or how to tighten a particular bolted assembly to a known value ? It is all but impossible setting standardised torque values to tighten them. A degree of tightness depends on the requirements of design. materials fastened, lubrication conditions and the function of end product, all go into setting rough limits but a number of other difficult-to-figure variables also go into the mix.

The axial stress induced by the tightening operation is called Initial Tension. This initial tension has got a prominent place in the design of bolted assemblies and its importance was realised as far back as when the bolts were used for fastening the parts. For design purposes a proper evaluation of this factor is necessary.

Many methods were used to preload the bolts to known values but they were restricted to limited use due to their higher operation time. In day to day practice this work was entrusted to the Mechanic. He from his habit, judgement and experience preloaded the bolts with the typical wrenches. So it was found necessary to estimate the average tensile load induced into bolt tightened by a skilled mechanic while using a typical set of wrenches. Many attempts were made in this direction by various investigators.

In Oct., 1898 issue of Sibley Journal of Engineering, Professor J.H.Barr of Cornell University published the results of experiments made at the University on a number of mechanics and derived a relation;

$$P_i = 16000\ D\ \text{lbs}$$

where P_i is initial load in lbs

D is nominal diameter of bolt in inches.

For metal to metal contact joints this relation was confirmed well by German experiments with bolts having dia. of 1/2 in., 3/4 in., and 1 in.

If a bolt fastens two parts with a flexible gasket between them, a smaller effort is applied in order not to crush the gasket and initial load may be taken as one-half of that given by the above relation.

The computed initial stress S_i from these relations for a 1/2" and smaller bolts may be above the ultimate strength of low carbon steels which accounts for the occasional failures of smaller bolts during tightening. This observation suggested that care be taken not to get small screws and bolts too tight.

Later Petrie found the initial stress for the C thread series, one inch bolt or larger to be

$$S_i = \frac{45000}{D^{1/2}}\ \text{p.s.i} \qquad (D\ \ 1\ \text{inch})$$

This relation suggests that if a high strength bolt material is used, it may be difficult in case of larger bolts to induce an initial tensile stress which approaches the yield strength.

Later Cardullo basing his observations on Barr's relation proposed a bolt formula

$$W = S_t\,(\,0.55\,D^2 - 0.25\,D\,)$$

where W is permissible load in Lbs., S_t is working stress in tension and D is nominal outside diameter in inches. It is seen that this formula gives little or no permissible load on bolts of 1/2" diameter and less. It is rather inadvisable to use the formula where perfect tightness is desired and where there is particular need of light construction.

Unwin recommends the following relation for allowable stresses and bolts of ordinary steel to make fluid tight joints.

$$S_t = 1600 + 1000\,D^2 \text{ for rough joints}$$

$$= 2500 + 3000\,D^2 \text{ for faced joints}$$

Some designers take care of initial tension by providing an allowance by multiplying the separating force by 1.25 in computing the stress on the root area of the bolt.

The degree of tightness depends upon the experience and judgement of the mechanic and so is bound to vary from man to man. Also, his physical strength and environmental conditions play an instrumental role in making him feel the degree of tightness. The relation given by Prof. Barr if taken to be an average but representative for an American skilled mechanic may not hold good in case of an average Indian mechanic. To the best of author's knowledge no attempt in his home country has ever been made to establish some such relation suitable to Indian conditions. The author has found this a good opportunity to work on this problem and selected this for his thesis work at this University.

It has been intended to investigate statistically the initial tension in the bolt by having different workers tighten the bolts of different sizes with different spanners and then to determin the load due to these tightenings.

P A R T I

CHAPTER I.

TORQUE - TENSION RELATION

The algebraic relation between the torque applied on the nut while tightening and the axial tension induced in a bolt may be determined in terms of coefficient of friction between the sliding parts and the dimensions of the nut, bolt and threads.

$$T = r_t F_1 \left(\frac{\cos\lambda \tan\alpha + \mu_1}{\cos\lambda - \mu_1 \tan\alpha} + \frac{r_c \mu_2}{r_t} \right)$$

where T = Applied Torque

r_t = Pitch radius of thread in bolt

r_c = Mean radius of nut face

λ = Half thread angle

α = Helix angle of thread

μ_1 = Coeff. of friction at thread surface

μ_2 = Coeff. of friction at nut face

The above equation gives a theoretically correct relation between torque applied and the resulting tension in terms of various quantities.

However a simple and more practical relation is desired to be more useful in actual practice.

Neglecting the hellix angle and taking r_c to be 4/3 times r_t, we get

$$T = r_t F_1 \left(\frac{\mu_1}{\cos\lambda} + \frac{r_c \mu_2}{r_t} \right)$$

Normally $\mu_1 = \mu_2 = 0.15$ for dry unlubricated steel on steel surfaces and

$$r_t \simeq \frac{D}{2}$$

Hence we get for Whitworth threads,

$$T = 0.185\, D\, F_1$$

The above assumptions are reasonable since coeff. of friction is subject to considerable variation, being affected by the condition and fit of thread surface, the lubrication and the magnitude of unit pressure between sliding surfaces. Also, the coeff. of running friction is less than the coeff. of starting friction.

Maney and others have found experimentally a relation of the form

$T = C.\ D.\ F_1$

$C = 0.20$ for UN unlubricated threads

$\quad = 0.18$ for UN lubricated

This torque T applied in tightening is partly against friction and partly in stretching the bolt to produce tension F_1. In order to determine the percentage of torque T used in stretching the bolt, consider the general relation

$$T = r_t\,(F_n \cos\lambda \sin\alpha + \mu_1 F_n \cos\alpha) + r_c \mu_2 F_1$$

but $F_1 = F_n\,(\cos\lambda \cos\alpha - \mu_1 \sin\alpha)$

$$T = r_t\,(F_n \cos\lambda \sin\alpha + \mu_1 \cos\alpha) + r_c F_n \mu_2 (\cos\lambda \cos\alpha - \mu_1 \sin\alpha)$$

$$\text{or } T = r_t F_n \cos\lambda \sin\alpha + r_t F_n \mu_1 \cos\alpha + r_c F_n \mu_2 (\cos\lambda \cos\alpha - \mu_1 \sin\alpha)$$

From this it can be observed that first term go in stretching the bolt whereas next two terms are for friction.

$$\frac{T_{strech}}{T_{friction}} = \frac{\cos\lambda \sin\alpha}{\mu_1 \cos\lambda + \frac{r_c}{r_t}\mu_2 (\cos\lambda \cos\alpha - \mu_1 \sin\alpha)}$$

For a 1 inch BSW bolt there

$\alpha = 2^\circ\ 30'$; $\lambda = \frac{\quad}{2}$ deg. $= 27^\circ\ 30'$

$\mu_1 = \mu_2 = 0.18$

We have

$$\frac{T_{strech}}{T_{friction}} = \frac{0.877 \quad 0.04232}{0.15(0.99905+\frac{4}{3}(0.987 \quad 0.99905-0.15 \quad .04232))}$$

$$= \frac{1}{8.535}$$

or $$\frac{T_{strech}}{T_{total}} = \frac{1}{1+8.535} = \frac{1}{9.535}$$

$$T_{strech} = \frac{1}{9.535} \times 100 = 10.5\ \%$$

This shows that on an average only 10.5 of percent of the torque applied go in stretching the bolt while the rest is used against friction. That is the reason, the torque measured by a torque wrench necessary to induce a certain initial tension varies considerably with the condition of surface of the threads and of the surface in contact between the nut face and its seat, type of fit between the thread, with the kind and amount of lubrication of the rubbing surfaces, with the materials in contact and with the slope of the threads; that is the trouble-some variable is the coeff. of friction.

CHAPTER II

STRESSES IN BOLTS

2.1 INITIAL STRESSES

The stresses in a bolt, screw, or stud when it is screwed up tightly are:-

a) A tensile stress due to stretching of bolt.

b) A torsional stress caused by the frictional resistance of the thread during its tightening. And

c) A bending stress if the surfaces under the head or nut are not perfectly normal to the bolt axis.

If the outside surface of the parts that are bolted together, are not parallel, the bolt will be subjected to bending by a moment M which is constant over the whole free length l. (vide fig. I)

The stress induced by this bending may be found by the following relation:

$$S_b = \frac{eE}{2l}$$

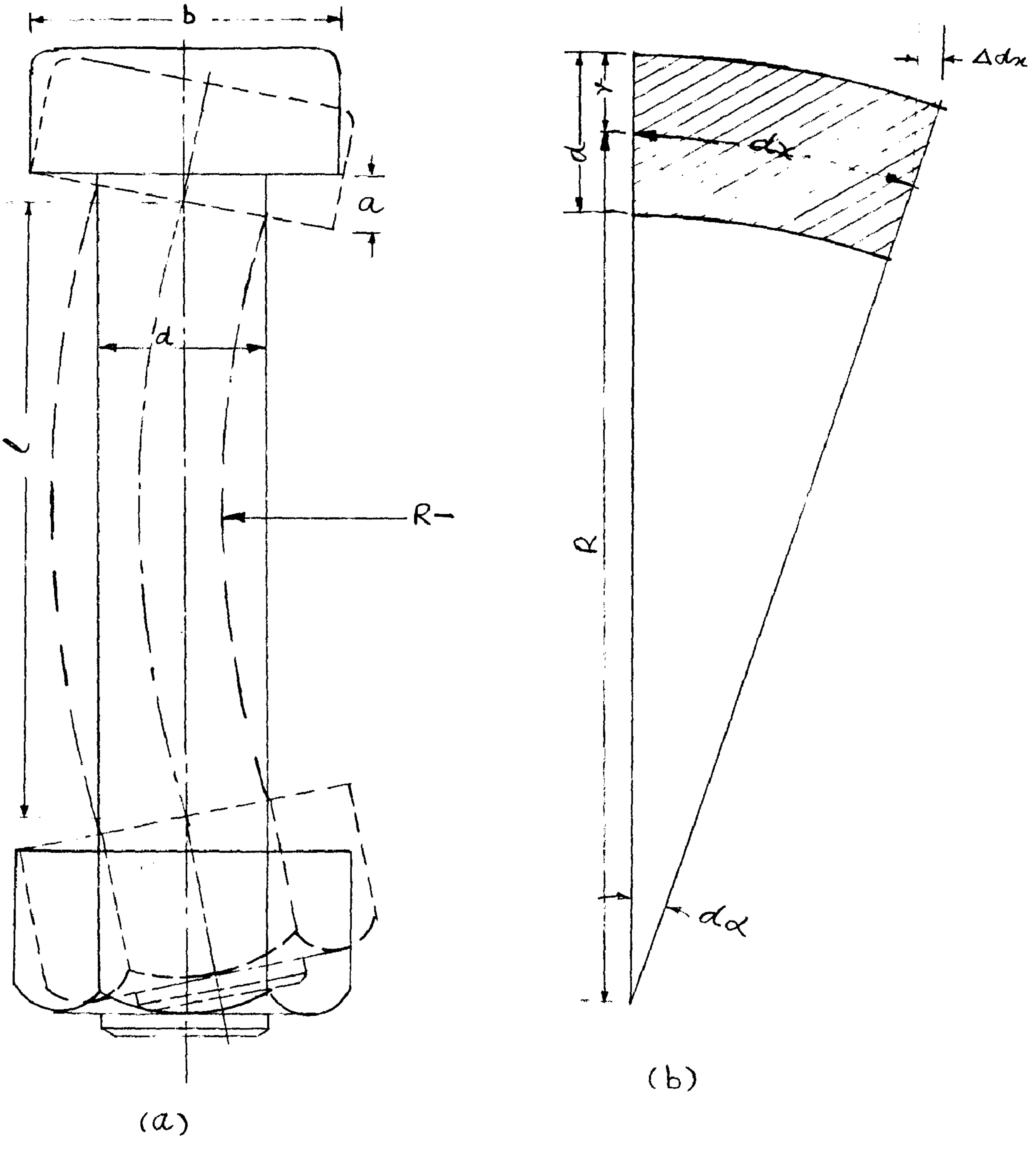
This shows that for a given material the stress depends only on the ratio $\frac{e}{l}$.

For one inch bolt holding together two flanges which have a combined thickness of 2.6 inch and are placed that gap e is 0.003 inch, the stress is

$$S_b = \frac{0.003 \times 30{,}000{,}000}{2 \times 2.6}$$

$$= 47300 \text{ p.s.i}$$

Inspite of the very small deviations of the surfaces from being parallel, the stress exceeds the elastic limit.



F[illegible] .1. F[illegible] [illegible]F F[illegible]LT

This shows how important it is to have the outside surfaces true. If they are not machined, then the surface around the hole must be spot faced.

For UNC standards bolts with sizes from 1/8" to 3" and for μ_1 =0.15 and μ_2 =0.20, the resultant of the various induced stresses is found to be from 20% to 22% greater than the tensile stress alone. However, for a screw with a diameter of 3/4" or less, the initial stress depends so much upon the judgement and the experience of the mechanic that an attempt to calculate it is practically useless.

The sum of bending stress and the initial stress from screwing the nut can very easily exceeds not only the elastic limit but even the ultimate strength of the bolt material. However, because of the ductility of steel, the bolt will be bent before breaking. This bending will relieve some of the stress, but it is not permissible in a machine part.

2.2 STRESSES DUE TO COMBINED LOADS (INITIAL + EXTERNAL)

Generally the external load applied to the bolt tends to separate the connected machine parts in the direction of the bolt axis, and this action sets up a tensile stress in the bolt. Sometimes when the bolts are used to prevent the relative movement of two or more parts, a shear stress is induced in the bolt. When the action of the load is at an angle to the axis, the stress in the bolt becomes tension and shear combined. This external load can be static as well as dynamic.

SHEAR LOAD

When a shear load is coming on the bolted joint, the bolt is tightened to produce an initial tensile preload F_i, a very high value, after which the external shear load is imposed. The effect of preload is to place the parts in compression and create sufficient friction between them to prevent inadmissible relative displacement of the elements in the direction of the acting forces which would load the bolt additionally in bending.

The value of F_i is determined by the required friction force F acting between the contact surfaces according to the condition

$$F = F_i \times i \times \mu \geqslant R$$

$$F_i \geqslant \frac{R}{\mu i}$$

i = pairs of contact surfaces

μ = coeff. of friction

R is external shear load

When the bolt is inserted in a reamed hole with a small interference, the shearing load is taken by the bolt shank. Such a joint is not necessarily prestressed.

TENSION LOAD

The tensile stress resulting in a bolt from the combined action of the initial load put on when the thread is screwed tight and of the external load may be found from the following relation

$$P_o = F_i + P \frac{C_b}{C_b + C}$$

where P_o is total load on bolt

P is external load

C_b is stiffness of bolt

C is stiffness of connected parts

or $P_o = F_i + P_n$

i.e. in joints with the prestressing force F_i after the external load has been applied the force acting on the bolt is increased by P_g.

$$P_g = P \frac{C_b}{C_b + C} < P$$

Examining the previous equation we see that if the stiffness of bolt C_b is very large as compared to C, then P_o approaches $(F_i + P)$

If C_b is very small as compared to C, the term in bracket becomes very small and total load approaches F_i.

Therefore the actuall load is always between the initial tension and the initial tension plus the external load (Provided the joint does not open). These remarks

The above analysis is not applicable when the bolt is subjected to significant bending moments as when the gasket is inside of the bolt circle, a very common arrangement for flat or ring gaskets.

VARIABLE EXTERNAL LOAD

When a variable load is imposed on the assembly, the bolt also experiences the variable load, though, of less magnitude. When the external load changes from 0 to P, the bolt section develops varying stress with a maximum cycle stress

$$\sigma_m = \sigma_i + \frac{P_g}{A_b}$$

and amplitude $\frac{P_g}{2A_b}$

If the load on the assembly is completely reversible the cylce amplitude is P_g

The amplitude of the alternating load depends on the reletive stiffness of the bolt and clamped parts. The assembly with reletively flexible bolt will have much less amplitude in bolt stress than the one with rigid bolt. It will also be seen that greater is the initial stress larger will be the value of the mean stress on the bolt. Of course, it will take up larger external loads without the separation of joint.

CHAPTER III

PRETENSIONING AND ITS ADVANTAGES

Preloading of bolts in important mechanical or structural joints is an established practice which, to meet increasingly stringent design requirements, is being more and more widely applied. It is now being specified for many non-structural joints e.g. by electrical equipment manufacturers to ensure adequate clamping of joints in essential circuits and hence their low resistance. Certain accessories manufacturers are also insisting upon pretensioning to ensure the correct and safe assembly of their products.

Pretensioning of the bolt produces benefits that fall into five broad categories.

a) Improvement in thread form and in applied load distribution.

b) Reduction of range of tension load fluctuations.

c) Reduction of range of shear load fluctuations.

d) Alleviation of adverse effects at inclined bedding faces.

e) General stiffening of the complete joints.

a The improvement in thread form is attributable to plasticity. Two beneficial thread effects are normally present. First, irregularities in threads are smoothened out and a better contact between the threads and nuts and of the bolt obtained. Secondly, a more equitable distribution of transmitted load is produced between the operative ~~terms~~ turns of the threads on the bolt. This results in both respects go well beyond what is possible by any manufacturing process.

It may be added here that these desired effects to be achieved, a fairly heavy pretensioning is necessary although there may be difficulties of opinion as to the precise degrees. Many skilled operators tighten untill yielding is definite and obvious (It is understood this degree of tightness is not always attainable with standard equipment for bolts of larger diameter). It is probable that such yielding which is detected partly by the observation of the number of ~~terms~~ turns of nut, and partly by 'feel', is confined to the threads. This degree of tightening appears to be adequate, and it is probably undesirable as well as unnecessary to proceed to a stage where general yielding of the main body of the bolt occurs. The bedding down process, which takes place not only at the threads but also at the contact faces of the structural elements comprising the joints, is progressive under the action of the fluctuating loads that is imposed on the steady tension load. Left to itself the originally tight nut becomes slack and may rotate slightly. In this event much of the benefit of thread pretensioning is lost. On the other hand if tension is maintained the progressive bedding down at the threads is beneficial. So it is necessary that tension in the bolt should be maintained continuously.

b IMPROVEMENT IN FATIGUE LIFE.

The second beneficial effect of pretensioning is the reduction in range of fluctuating tension imposed by the operational loading. This reduction arrises from the comparative flexibility of the bolt in tension and the comparative rigidity.

in compression of the structural elements that are clamped together by the bolt and the nut. If, for example, the bolt under tension expands ten times as much as the sturucure contract under the corresponding compressive force, then only one eleventh of the load fluctuation passes through the bolt, the remainder passes through the structure which being in compression is not as a rule vulnerable to fatigue.

This reduction in the range of load fluctuations is obtained at the price of a relatively high steady stress or mean stress in the bolt. Under reasonable conditions of design and operation and provided pretensioning is kept within what would be generally regarded as reasonable limits, the effect of this steady stress is secondary.

The relieving effect fails only if the operational tension exceeds the initial tension, in which event the contactin faces of the structure separate and there is no compression in the structure. Under reasonable design conditions this is unlikely to occur. Subject to consideration of static strength a very occasional overload in this respect is not likely to do much harm from the fatigue stand point.

With the reservation mentioned for the very heavy load, the benefits of pretentioning through diminution of axial load fluctuations are considerable and operate over the entire loading range. In terms of fatigue life, they are more significant at low loading levels, since the fatigue life may be extended indefinitely. The fatigue tests carried at the de Havilland Aircraft Company in U.K. showed an increase in life by a factor of approximately 1000: 1 when pretension load was increased from 1400 to 6400 Lbs. for an identical loading pattern.

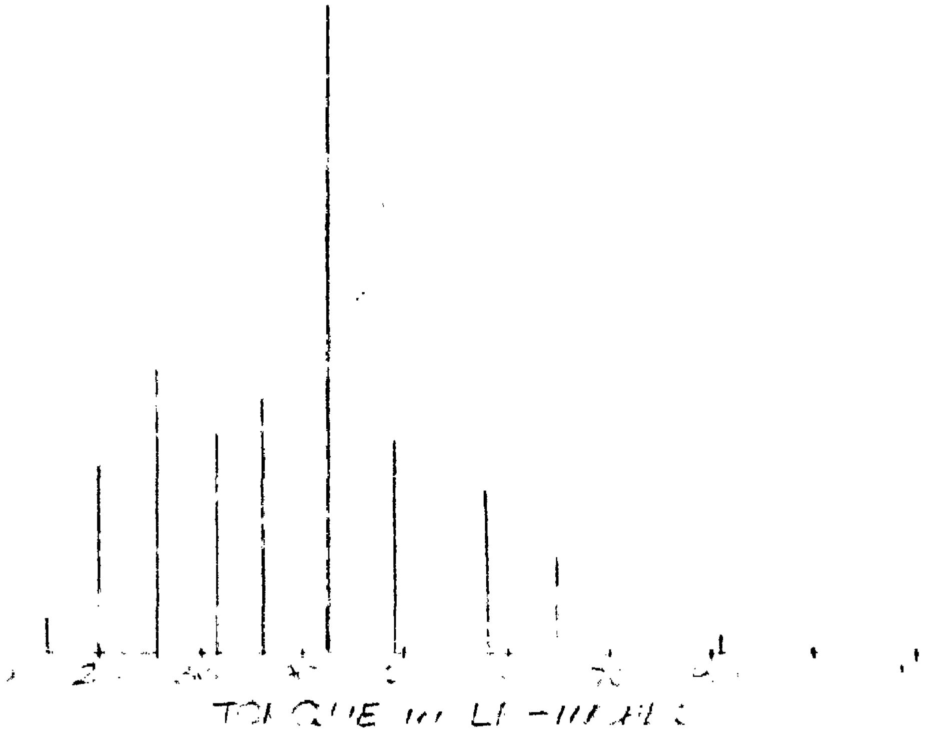
c IMPROVEMENT IN FATIGUE AGAINST SHEAR LOADING.

For a shear joint, the advantages of a high pretension are immediately obvious. In fig.2 which shows a simple shear joint held by one bolt, if the bolt is not loaded to a high fig. it will be seen that it acts as a peg. In order to obtain a reasonable fatigue life, both the hole and bolts are made to very close limits. If, however, the stress in the bolt is raised to a very high level, the friction is used to carry the alternate loads. If P is the load in the bolt, and μ is coeff. of friction between the plates, then provided that alternating shear load is less than μP, the bolt will carry no additional load. It cannot, therefore, fail in fatigue.

Another important practical advantage is that, because the bolt is no longer a peg, the dimensional tolerances of both the bolt and hole are no longer of prime importance. In many practical cases where weight is an important consideration, these dimensions are in fact carefully maintained. This factor is especially important where the shear load and coeff. of friction are open to doubt.

d In cases where the faces of the nut and the structure are initially inclined, they are brought together by pretensioning. The result of this permanent contact is that fluctuations in bending moment is removed completely. There is, clearly, always present the steady bending moment necessary to produce full contac but it is a fact on fatigue life is normally secondary. Further more, with care in manufacture of contact faces, the steady bendin moment can be reduced to something very small.

It is of interest to consider a little further the



[illegible]G.3. ACTUAL TORQUE [illegible] APPLIED BY A[illegible] [illegible] HAND T[illegible]

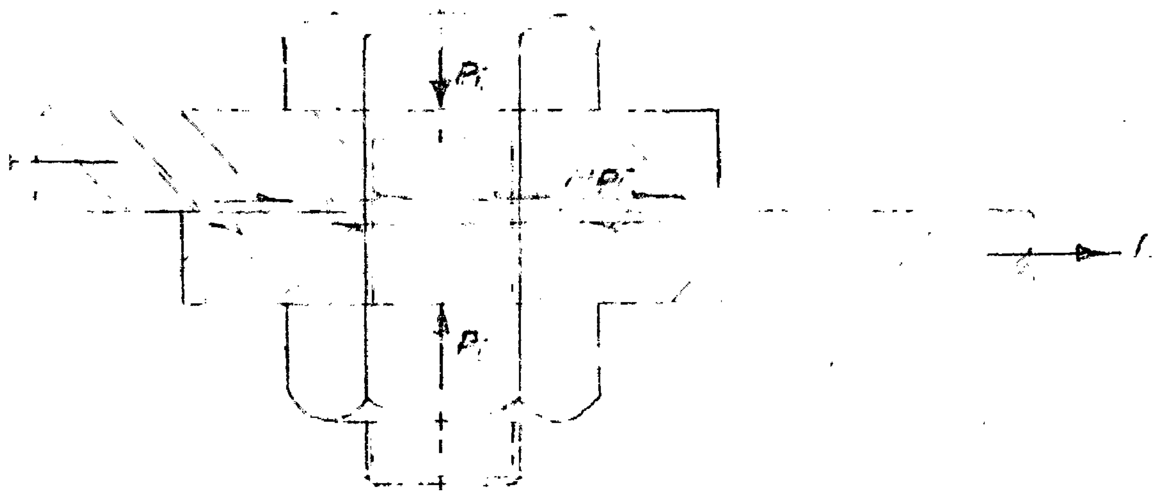


FIG.2. SHEAR JOINT

manufacturing limitations in relation to contact angle. Clearly pretensioning should not be relied upon to bring together two surfaces inclined originally at several degrees to each other. On the other hand, manufacturing accuracy is not absolute and it appears reasonable to assume that a small angle of inclination can exist inspite of the care taken. Without pretensioning, therefore, there must exist some low loading level at which entire fluctuating axial load is applied non-axially at an edge or corner of the nut. The associated bending moment is then far more damaging from the fatigue standpoint than the axial component of tension, although only a small degree of pretensioning is required to remove it entirely. Thus, it would appear that while pretensioning is greatly benefecial all conditions of bedding, it may have a special and vital significance under conditions of low load fluctuations such as are produced by vibration.

c <u>STIFFENING OF STRUCTURE</u>

The benefit of pretensioning arises from the general stiffening of the structure which it produces. Without pretensioning the non-axial loading effects depend primarily on the flexibility of the particular structural element immediately under the nut face. Pretensioning clamps all the elements together and in general gives a considerable increase in overall effective stiffness.

Closely allied to this direct stiffening effect is the prevention of fretting that may arise from a general looseness in the complete joint unit. As is well known, fretting is an acceleration of fatigue failure. A mild degree of tensioning may be insufficient to prevent excessive fretting, but in any event a " ~~first~~ Just tight" may soon become a slack unit. Fretting cannot necessarily be eluminated entirely by pretensioning or in fact by any other means, but it is rendered much less likely to occur.

CHAPTER IV

METHODS OF PRELOADING AND THE MEASUREMENT.

In the design of important joints as in air craft Engines, the value of preload on the bolt must be known precisely or alternately there must be some method of preloading the bolts accurately to the given value. As pointed out earlier, the torque applied by hand varies considerably from operator to operator and even an individual operator if given a large number of bolts to tighten, may easily vary the torque he applies throughout the run, the very extensionable factor of having to rely on the skill and integrity of an operator for the consistency in tightness of all the nuts on a joint. ' Standard ' lengths of spanners do not help to any great extent because the variation of physical strength from one operator to another reduces their effectiveness.

A further consideration is that in fact it is usually the inspector who determines the final torque fatigue. Because in most cases he will insist on moving the nut just a fraction during checking, the nut results can be quite astonishing as shown in Fig 3. This shows the torque to which 810-C DA nuts were tightened on an assembly. They were loaded by a skilled operator using a ' Standard ' spanner and checked by an inspector, and although the Gaussian distribution is well marked, and the overall scatter of torque is from less than 10 to more than 80 lb-in. Bolts assembled in test pieces under similar conditions to those in the component were tested to failure and this occured in some bolts at an applied torque as low as 70 lb/in. It will be seen that this is extremely dangerous, especially as in this particular case the fraction of any one bolt in the assembly would have resulted in its head being drawn into the compressor of a jet engine.

The pretensioning of these bolts to known safe figure was an obvious necessity. Great care is necessary in carrying out preloading. The main danger is over tightening. It is possible to break a bolt at the start but still worse, a crack may be initiated which is not detected at this time but which soon develops through Fatigue.

METHODS

To-day the most widespread methods of control are the measuring of

a. Elongation i.e. the difference in datum length measured before and after tightening.

b. The angle of the nut turn.

c. Torque when turning the nut with a torque wrench

A [illegible]

Out of these the only really satisfactory and dependable method being to measure the extension after tightening. In most cases, however, this method is impossible to apply, due to inaccessibility. It is applicable only where both ends of bolt are accessible.

This is chiefly employed in Aircraft production where each every bolt is tightened very precisely.

Under certain circumstances where short thick bolts are used, it is possible to drill a small hole along the axis of the bolt. A thin rod which is secured at the threaded end of the bolt is inserted into the hole and machined at the other end to be flush with the face of the head. On assembly the nut is tightened untill the bolt stretches sufficiently to draw the end of the rod below the surface of the head by a calculated amount.

Material deformation under load is also used in two other cases. One of these consists of inserting a conical washer between the nut and plain washer. The cone is compressed on assembly until it is flat. An alternate method is to use two washers, a thick one surrounded by a thin one. As the nut is tightened the inner washer deforms until it is of the same thickness as the outer one. At this point it becomes impossible to turn the outer washer. And by adjusting the radial thickness of the inner washer, this can be made to occur at a predetermined bolt load. These two methods ensure only that the nut is not slack and do not guarantee that the bolt is not over loaded.

D. TURN OF THE NUT METHOD

At remote locations where power tools, skilled labour, torque wrenches etc., are sometimes not available and in structural connections where use of torque wrench becomes uneconominical, a good reliable method is to be used to preload the bolts to an approximate figure.

In 1954 Mr. E.J.Ruble, Research Engineer, of American Society of Railroads conducted a large number of tests to determine if the number of turns of the nut could be used as the criterion for bolt tension. His tests covered bolts ranging from 5/8" to 1-1/8" in dia. using laminated material drawn up tight with the ASTM-A325 bolts used as fitting up bolts. He found, for all diameters, that 1/2 turn from a ' finger tight ' position produced the minimum bolt tension required by specifications i.e. 80 to 90% of yield load of the bolt. He also found that 2 to 3 turns were required to break the bolt or strip the threads. He recommended that bolts be given one full turn of the nut from ' finger tightness ' in order to be above the minimum tension required in structural joints. In 1955, the research council of ASCE, in appendix B to the specifications, approved one turn of the nut from hand tight position as a satisfactory method of tension control. In all of the above it should be noted that turns are measured from the finger or hand tight position after the steel surfaces have been drawn together with fitting up bolts.

Bethlehem steel company pointed out that it was not practical to use finger or hand tightness as a reliable starting point for measuring the amount of turn. Because of the effects of dents and dirt accumulated in threads, it was difficult and time consuming to determine the hand tight position. Furthermore, steel erectors objected to tightening bolts by hand when they had impact wrenches available to do the work. The need for establishing a better starting point for final stressing was clearly

After numerous experiments, Bethlehem developed a Turn-of-Nut Method which was proved to be safe, practical and economical. The bolts are installed with the nuts started on a thread or two by hand, then spin the nut down tight, using the impact wrench as a nut runner to a 'snug' condition. The most reliable starting point, for measuring the amount of turn, was found to be this 'snug' condition. The ' snug ' position is naturally tighter than the hand tight position; therefore, less additional turning is required to bring the bolt to its proper tension. Numerous trial runs showed that the amount of turn required from the snug position varies from 1/8 to 3/4 turn depending on bolt diameter and grip length. The amount of turn increases with the increase in grip of the bolt.

It may be added here that one complete turn does not give the bolt an elongation equal to the pitch of the bolt thread. The actual elongation is less than this and differs considerably for individual bolts. But once the bolt yields, the curve between elongation vs load becomes considerably flat. This means that even if the elongation produced by turning the nut varies within 80% or more in this range, the clamping force varies only by about 12.5% of the 0.2% yield value depending on the shape of the curve. In other words, once yielding occured, variation in angle of rotation of the nut would not influence significantly the clamping force.

E USE OF TORQUE WRENCH

It is seen that due to the pitch of their threads, by tightening a nut on a bolt to a definite torque, it should possible to produce a known tension in the bolt. This can be achieved by using an ordinary spanner and a spring balance, but this is a cumbersome and potentionally inaccurate method. Torque wrenches have therefore been designed which are widely used in many industries. Before they can be used, however, the relationship must be found between the torque appli to the nut and the tension which it produces in the bolt.

It is theoretically possible to determine the torque required to produce any desired bolt pretension in terms of bolt dimensions and the coeff. of friction between threads and the nut bearing face. A relation has been given on page
However a simple relation is

Tightening Torque = Torque coeff. x Bolt load x Mean dia of thread.

A commonly accepted value for the torque coeff. for dry surfaces is 0.2. As is often the case, variation such as this leaves the reliability of a purely theoretical approach open to question.

Because as much as 90% of the work done by the operator while tightening the nut can be lost in friction, it will be appreciated that a variation in the coeff. of friction or in the radius of contact for either pair of mating surfaces would profoundly affect the resultant tension in a bolt for one value of applied torque. Because the effecti radius of nut-to-washer surface is greater than that of nut and bolt threads, any inaccuracy in in manufacture of nut or washer such as lack of flatness in the abutting faces, will have a considerable effect on the bolt load. It is therefore advisable to produce a nut with the smallest possible effective contact radius. This is done by providing a recess in the bearing area of the nut and head so that it is reduced to a narrow ring.

With so much uncertainty, it is not possible to use the above relation in pretensioning under all conditions. Hand torque wrenches have to be calibrated for each lot of bolt at each application to determine the correct torque-tension ratios for that application. Many calibration devices have been designed and marketed.

Bethlehem company built a calibrating device which consists of a hollow ram cylinder jack of 60 ton capacity. The yoke which pulls against the ram is guided by a substantial frame which bears against the cylinder and by torquing a bolt held between the yoke and bottom face plate, the jack is energized by stress in the bolt. A hydraulic gauge reading total load, indicates the bolt tension directly. A similar device by another firm was marketed and shown in reference.

The calibrated wrench method was used to tighten several million high strength bolts all over the world and was successful since fewer skilled men are required to do the work. It proved to be economical and Practical when bolts on a job were mainly of same diameter and approximately of same lengths thus requiring only one torque tension ratio. However, it became quite impractical for joints requiring different size bolts which meant additional wrenches. To overcome this, an adjustable friction clutch device was developed which could be applied to the impact wrench using air pressure between wrench and the socket. The clutch could be adjusted to slip at prescribed torques and would hold its adjustment fairly well. However, it added weight to the power tool and its size prevented its use in close quarters.

The air tool manufacturers have studied this problem and have developed a built-in control wrench, which would shut off air pressure at specified torques.

MEASUREMENT

The various investigators have tried different methods for finding the initial load in the bolt.

Roesholm and Companions while testing the Heat Exchanger Flanges used the dial micrometer mounted on a height gauge for measuring the extension of bolt when the vessel was subjected to internal pressure. They put one steel ball on each end of bolt for some bolts and 2 steel balls on each end for other bolts. The spherical surfaces permitted consistent readings and the two balls were used for a study of bending in the studs when stressed. They calibrated these bolts for load on an Olsen Tensile Testing Machine torque wrench was used to vary the load on the bolt.

T.J. Dolan used electrical strain gauges to investigate the influence of bolt tension and eccentric tensile loads on the behaviour of a bolted joint. He mounted the strain gauges on the bolt shank as well as on the clamped block.

Wilbur M. Wilson et.al. determined the initial tension in a number of rivets by measuring the recoverence in the length of the rivets that occurred when the tension was relieved. A hardened steel pin extended through a hole in each head and fitted tight in a hole in the end of the rivet shank in such a manner that the pin moved with the end of the shank. There was a small hole in the outer end of each pin that received a conical point of the instrument used to measure the recoverence of the rivet. The instrument is a C - frame having a conical point on one end and a micrometer dial on the other end. The instrument was counter weighted so as to balance on the conical point in the upper pin. The dial was graduated to 0.0001" and the instrument was checked against a Standard bar after each set of readings. The specimens, Standard bar and instrument were placed near each other in a constant temperature room several hours before readings were taken in order to eliminate as far as possible errors due to temperature changes.

CHAPTER V

TORQUE WRENCHES

Torque wrenches themselves take many different forms, the simplest of all being the beam spanner. In this type the handle is made of flexible material, and it will be appreciated that the deflection of any point along the beam will be proportional to the torque at the spanner drive square. This deflection is measured in many ways, e.g. by a rigid pointer that is fixed to the spanner head and whose other end is free to move across a scale. As can be seen this group are simple tools and are on the whole fairly cheap.

Fig. 4 shows some designs of such wrenches commonly used in Soviet Union. Soviet standards provide for three types of wrenches with adjustable torque (within 20 and 200 KG cm)

a. Side wrenches
b. Box wrenches adjusted by one spring
c. Box wrenches adjusted by two springs

For a more convenient handling of such wrenches use can be made of adaptors and also changeable heads.

Another type which is dependent on deflection for its readings measures the twist in a pin. This is rigidly secured at one end to the head of the wrench, the socket being fixed to its other end. The twist of a given length of the pin is magnified by a beam inside the spanner handle

TORQUE WRENCHES
SOVIET STANDARDS

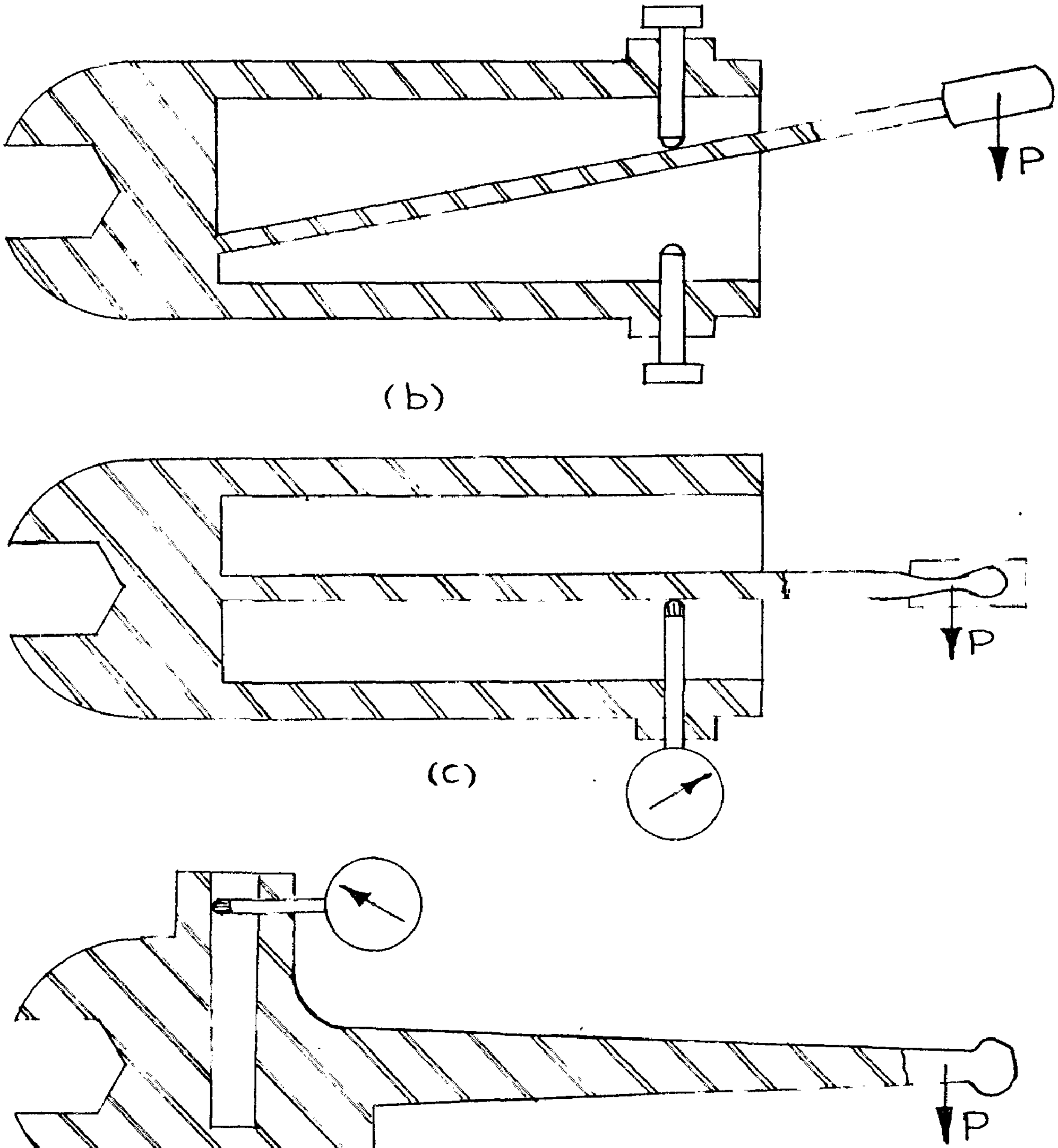


FIG. 1

and this operates a clock mechanism. The clock on this type of instrument may be of continuous-reading kind or may record the maximum torque applied. In some cases both of these facilities are incorporated.

One disadvantage of this group of wrenches esp., when dealing with large torques, is that the ~~loading~~ body of the spanner itself deflects, as in a beam wrench and this leads to inaccuracies. This fault has been overcome in one make of wrench by having the handle asa separate unit from that of containing the magnification arm and clock mechanism. Here, of course, any convenient shape or length of handle may be employed.

Many companies feel that dial reading torque wrenches throw too much responsibility on the operators and inspectors for general shop use. Wrenches which can ~~not~~ be preset to the required torque on a test-and-setting rig are therefore widely used. Tools of this type have a mechanism which prevents the desired torque being exceeded. A typical example is the Acratork torque wrench. This has a cam device in the spanner head, out of which, a roller is forced against a spring in the handle when the torque on the cam reaches a sufficiently high figure. Fixed on the cam is a normal socket drive peg, the spanners being available in various sizes for right-handed, left-handed or dual operation.

For torques of 1000 lbft or more, where it becomes impracticable to use manual methods, hydraulic torque generators are available. These are specially made to suit individual cases, e.g. torque loading air craft propeller

An innovation to assembly line practice, which gives a certain degree of torque-loading consistency is the Pneumatic nut-runner or screw driver. These have a turbine to that used in a normal air operated drill, which drives the socket or screw driver but through a spring loaded clutch unit. The compression of air spring can be adjusted to vary the torque at which slipping occurs. One of the main advantages of this tool is the great speed with which large numbers of nuts can be tightened, but critical adjustment of the torque is difficult, and unless compensating units are fitted, the tool is often dependent on the factory air line pressure.

A DIFFERENT PROBLEM.

One of the greatest problems which arise with these tools is to know the relationship between the final torque on the nut and the preset torque at the clutch unit. This unit usually set by hand to break a given torque, either on rig or with a torque wrench. When in use under power, both the nut runner mechanism and the nut are travelling at speed, and when the nut meets the abutment, a torque will be applied, due to momentum effect, in addition to the clutch unit breaking torque. This mechanism effect will naturally depend upon the mass of moving parts, the speed at which the nut is travelling when it meets the abutment and on the resilience of the assembly. To the best of the authors knowledge this problem has never been studied in any detail, though it is fully appreciated by the manufacturers of the tools, " Drawing-torque" that is, where the nut is tightened under some constant restraining force, such as compressing a spring was quoted by one supplier being

For further study on problems involved with use of adapters and the development of a cage torque wrench fitted with extension sections and a device for ensuring constant position and direction of pull, see reference

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as low as 2/5 of the " Free- running torque" where the nut has a lubricated free movement on a bolt of sufficient length to allow the equipment to reach its maximum speed before the nut meets the abutment. This does indicate that pneumatic nut runners, at least of this type, should not be used where accurate control of torque is essential unless extreme precautions are used.

There are many other types of torque wrenches, torque screw deivers and torque multiplication gear boxes, each of which has its own particular merits and they are manufactured in many countries.

LIMITATIONS OF STANDARD TOOLS.

As with many things in Industry, however, standard tools can not be used in many situations, and torque loading is no exception. It is reported that the majority of nuts can be tightened by commercial equipment but that occassionally due to accessibility difficulties - a problem which presents itself more often when repair work has to be carried out - adapters must be used. Unfortunately these intsoduce further complications. It was found by the de Havilland Air craft Company during investigation into this matter, that whereas a torque wrench itself may keep within approximately ± 1% of its set torque, tolerances of ± 20% were not unusual when extension section and spanner were unaltered. The design tolerance on the figures quoted for torque loads to be applied on bolts on the assembly line was ±5% and it was therefore necessary to find a remedy.

For further study on problems involved with use of adapters and the development of a cage torque wrench fitted with extension sections and a device for ensuring constant position and direction of pull, see reference

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P A R T II

CHAPTER VI

PROBLEM AND APPROACH

As discussed before, the degree of tightness is completely dependent on the judgement and experience of the mechanic. It is the feel of his hand that makes him judge the degree of tightness. The torque applied by hand varies considerably from operator to operator and the effect of this is made even worse if the bolts are to be assembled with different sealing or lubricating compounds. The type of spanner used in tightening may also have a pronounced effect on this. The ring and box spanners provide more perfect grip of the nut and so should tighten more as compared to other spanners. The operator feels more safe while using ring and box spanners than double end and adjustable wrench, there being no chances of spanner slip in the former case. Even an individual operator given a large number of nuts to tighten, may easily vary the torque which he applies throughout the run. Standard lengths of spanners may not help to any great extent because the variation of physical strength from one operator to another reduces their effectiveness. Experience also shows that an average workman in tightening up a bolt, pays little heed to the pressure the joint has to withstand subsequently but regulates his effort more or less according to the size of the bolt.

To investigate this problem so experimentally, the the following types of common commercially used spanners are employed here in tightening the bolts:-

1. Double-end spanner
2. Ring spanner
3. Box spanner
4. Adjustable spanners

These spanners of standard sizes as supplied by the manufacturer and their dimensional details are given in the appendix.

To study the effect of the size of the bolt or value of initial tension, the following representative sizes of the bolt family are selected for testing:-

i. 1 inch nominal diameter
ii. 3/4 inch nominal diameter
iii. 1/2 inch nominal diameter
iv. 1/4 inch nominal diameter

It is felt that bolt length has got no influence on the initial tightening load, and so no effort has been made in this direction.

To investigate the effect of spanner type and bolt size on tightening, all the four sizes of bolts are tightened with each type of spanner by one operator. Each operator will make 16 tightenings. Qualitative as well as quantitative analysis can be made from these observations.

To determine a relationship between tightening load, bolt size and spanner type, which will represent broadly but reasonably, the whole mechanic population in the country, a very large number of mechanics from the various parts of the country should be tested. Since at this moment, the nature of the statistical distribution curve of frequency vs. deviation from mean value is not known, the sample value which will truly represent the population cannot be ascertained. Taking availability of time at the author's disposal for this work, it was decided to test nearly 100 operators at various workshops of the city which, it is hoped, will give reasonable representation of the population. The workshops where tests are conducted are:-

a. University workshop including operators from various laboratories.

b. Central Building Research Institute workshop including structural division.

c. Military Workshop of Bengal Engineering Group, Roorkee.

Two types of bolted joints are employed in practice for different applications. They are:-

1. Ground or Metal to Metal contact joints.
2. Joints with flexible gaskets.

If a bolt fastens two parts with a flexible gasket between them, a smaller effort is applied in order not to crush the gasket and the initial load will be much smaller than in the case of Metal to metal contact joints. In the present testing only metal to metal contact joint is employed as there are innumerable varieties of gaskets available in the market which prohibit the testing of latter type of joint.

Lubrication has a pronounced effect on the tightening torque. So it is desirable to keep the condition of lubrication at thread surfaces, between nut and element contact faces and between bolt head and element contact faces identical in all the tests to get consistent results. The type of lubricant also varies the results. It is difficult to control the degree of lubrication and also, it is not resorted to in almost all tightenings in actual practice. It was decided to tighten the bolts dry.

CHAPTER VII

MEASUREMENT TECHNIQUE

Measurement of tension load in the bolt is most important in this testing. The accuracy as well as efficiency of the programme is governed by the choice of a suitable method. Two possible techniques of measurement are:-

1. Electrical
2. Mechanical

These are discussed below in detail alongwith their relative suitability to the present testing programme.

ELECTRICAL.

This technique involves the use of electrical strain gauges. The strain gauges are fixed on the bolt shank or the clamped element and the strain on either is noted while screwing the nut. This strain is then calibrated against known axial loads. As stated above there are two ways of using the strain gauges:-

a. Strain gauges on bolt shanks.
b. Strain gauges on clamped elements.

a. STRAIN GAUGES ON BOLT SHANK.

In this technique min. of two strain gauges are pasted on each bolt with their longitudinal axes parallel to the bolt axis but on diametrically opposite sides. Two gauges are used to nullify the effect of bending stress, if any, in measurement. These gauges are connected in the opposite arms of a 4-gauge bridge. Their output as sensed by the for axial load is doubled but for bending stress is zero. These gauges are insensitive to torsional stresses. The other two arms of the bridge can be any dummy equal resistances. The unbalanced voltage from the bridge is fed to the strain indicator and noted the deflection for various tightenings.

The next step is to calibrate these bolts. For this, bolts are caught in the calibration rig as shown in drawings attached and loaded in tension in the Tensile Testing Machine. The effective length of bolt loaded should be kept same as in actual testing. The load on the dial for various deflection of the strain indicator are noted and a graph between load and deflection is drawn. This gives us a calibration curve. If all the bolts are of the same size and specifications, only few need to be tested to draw the calibration curve.

For each bolt to be tightened, at least two gauges are required. Taking the number of bolts to be tightened in view it is seen that a large number of strain gauges are required which makes this method costly and so impracticable.

B. STRAIN GAUGES ON CLAMPED MEMBER.

The above difficulty of procuring a large number of strain gauges can be partially overcome by pasting the strain gauges on the clamped block. The blocks are made of cast

[illegible]

[illegible]

area and have through holes for the bolts and parallel machined and surfaces for the seating of head of bolt and nut. The gauges should be as near the bolt axis as possible to ensure minimum of bending stress on block, if any due to eccentric loading. The clamped block is made in two parts, one having the hole of usual diameter while the hole in other has bigger diameter to a certain length as shown. Space is also provided to take out the leads of the gauges. However to avoid this extra machining trouble, the strain gauges are put on outer surface of the carefully machined and centrally drilled and reamed correct hole block. Two gauges are put on each block parallel to hole axis but diametrically opposite having transverse mid-axes in one plane. These gauges of connected in a 4-gauge bridge in opposite arms and output fed to the strain indicator. Due to the large stressed area in compression, there will be very little deflection of block in compression and so less output to strain indicator. The strain gauges are pasted on four blocks, each for different bolt size, and so only 8 gauges are required.

Since the bolt strain is dependent on so many variables which are difficult to control, it is thought that the employment of very accurate method like the electrical one is unnecessary especially when a large statistical data is to be collected.

MECHANICAL

Due to stretching of the bolt while tightening there is a definite increase in bolt length. Any mechanical instrument which can measure the change in length can be employed. An extensometer and a tensometer which give change in length and unit strain respectively are not used due to mounting difficulty of the former and the latter being not helpful during plastic yielding of the bolt.

An easiest and fairly accurate method is to determine the change in overall length of the bolt with a micrometer. A Vernier micrometer of range from 3 inch to 4 inch and having a least count of 0.0001 inch is employed. The length of each bolt before and after it is tightened, is measured and the difference of these gives the stretch during tightening. The next step is to determine the tension in the bolt whose elongation is known. This can be achieved by two ways:-

1. Analytical
2. Calibration

ANALYTICAL

Assuming that the bolt has not been stressed beyond the proportional limit while tightening, the common Stress-Strain Formula can be applied to find the load.

$$
\begin{aligned}
dl &= dl_1 + dl_2 \\
&= \frac{F_i l_1}{A_1 E} + \frac{F_i l_2}{A_2 E} \\
&= \frac{F_i}{E}\left(\frac{l_1}{A_1} + \frac{l_2}{A_2}\right)
\end{aligned}
$$

where dl = total observed elongation of bolt
l_1 = length of unthreaded shank
l_2 = effective loaded length of threaded portion
A_1 = area of shank
A_2 = effective threaded area
P_1 = tension load

So if E, l_1, l_2, A_1, A_2 are known, P_1 can be found out easily.

If the material has gone beyond the proportional limit, even then the bolt load can be found out easily provided the Stress-Strain curve of the bolt material is available. Knowing the elongation and dimensions of bolt, P_1 can be easily calculated.

The trouble here is that Stress-Strain curve used above may not be the true representation of the bolt material behavior due to variation in method of processing the bolts and test specimens.

The major drawback with the above method is about taking the value of D. It is not constant over the length of the stressed portion, the diameter at shank being different from root diameter at the threaded part. This trouble can be avoided by either reducing the diameter of bolt shank and making it equal to root diameter of thread or making the threaded portion as small as possible, having little influence on overall effect. The bolts of former type are, however, not used in structures and in other common applications where impact loading is not encountered. The latter remedy of making threaded portion small is followed here in the present testing. Not more than one or one and a half free threads are allowed at nut face.

The diameter not being uniformed, the grip length l as it is, can not be used in calculations. This has to be modified for diameter variation. However by adopting the same procedure as suggested above, this can be minimized. The uncertainty of load distribution in nut, bending of threads and stress concentration at root also lead to incorrect results.

2. CALIBRATION

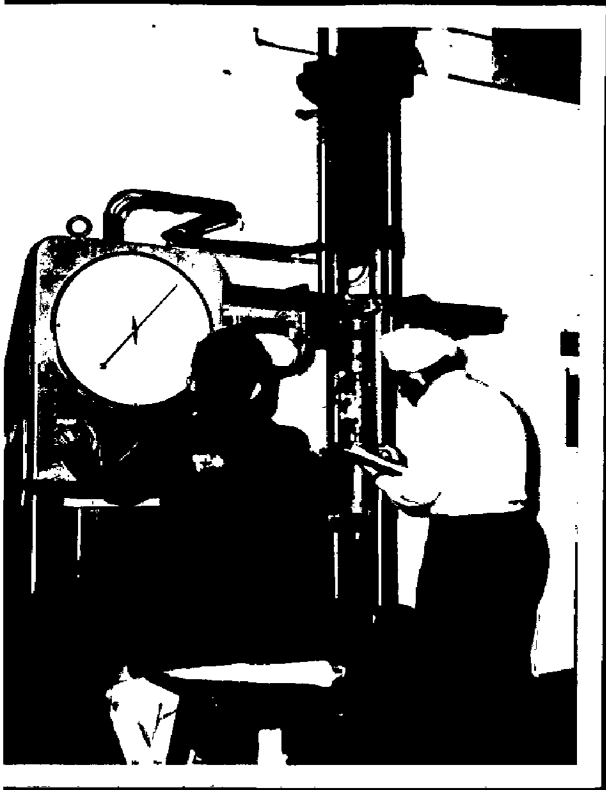
The simple and accurate method is to calibrate the bolt for known loads. A relation between bolt elongation and load is found out experimentally but accurately on the common Universal Testing Machine.

The bolt and nut having the stressed portion exactly equal to the length used in testing is put and adjusted in the calibration rig, specially designed and fabricated here. The drawings are attached. This calibration rig is mounted in the jaws of the testing machine. No pretension is given to the bolt. The machine is run and bolt is loaded in pure tension. The load is indicated on the dial at the control panel desk and the corresponding elongation is measured by an extensometer mounted on the pins of the m.s. strips bolted to ends of the bolt.

After an accurate and satisfactory test a curve between the load and elongation is drawn. This gives a standard calibration curve. If all the bolts are dimensionally as well as metallurgically identical, only four curves, one for each size bolt are drawn.

It may be noted that change in diameter, nature of load distribution in nut and stress concentration etc., have not involved any complication here. But it may be observed that all the bolts of each size are to be reasonablly identical in all respects. The diameter of bolt is limited within plus minus half thou. and effort is made to keep the threaded length as well as total length same for all. The consistency of material and processing is also kept under control.

In the testing, calibration method is employed. Extension is determined with a Vernier micrometer of range from 3 inch to 4 inch and having a least count of 0.0001".



CALIBRATION OF BOLT

MOUNTING OF EXTENSOMETER ON BOLT

CHAPTER VIII

DESIGN AND FABRICATION

After the approach to the problem is finalized and the testing proceedure is formalised, it is now the suitable design and fabrication of the test apparatus which is to be looked into because it is the latter which makes the programme successful and not mere oral plan. The accuracy of the apparatus depnds upon the execution time, the accuracy desired of results, workshop facilities available and last but not the least the financial resources.

In the present programme, the test set up consists of:-

a. A number of bolts of different sizes.

b. Elements to be clamped.

c. Tightening device - Spanners.

d. Measuring device - Micrometer, Universal Testing Machine, Dial gauge, Extensometer etc.

As stated earlier it is now generally recognised that approximately 90% of applied torque is absorbed in overcoming friction (50% under nut bearing face and 40% in threads) leaving only 10% of torque available for conversion into clamping force at the joint. This friction factor is effected significantly by

i. The material of both the fastner and assembly components.

iii. Washer and Nut characteristics

iv. Condition of threads

v. Finish or plating and fit of mating threads peculier to specific design situations.

To visualise the error caused due to even slight change in friction factor, consider a bolted assembly where 1000 lb-in of torque is applied to tighten it first when bolt threads are dry and secondly with slight lubrication. In first case if 900 lb.in. is consumed in friction, then only 100 lb.in. is utilized in tensioning. In the second case, due to lubrication, say 800 lb.in. is used in friction, then remaining 200 lb.in. is used in tensioning. It is seen that for friction torque to change from 900 lb.in. to 800 lb.in. the tensioning torque has increased from 100 lb.in. to 200 lb.in that is for only 11% change in friction, there is 100% change in resulting tension in the bolt. This stresses the need for keeping the conditions contributing to friction consistent and under proper control while testing.

The design consideration and their realisation in the shop are discussed here in detail for each of the above mentioned test components.

BOLTS AND NUTS

The bolts with different types of threads are used in practice but the most commonly used one is Whitworth thread due to its strength and favourable stress condition at thread root. Hence the mild steel bolt of various sizes but of BSW threads are employed here in the testing. The physical properties of the bolt material are given in the appendix.

The lengths of various bolts are decided taking measurement technique into consideration. Since, here the Vernier micrometer of range from 3 inch to 4 inch is used and the quantity to be measured directly is the bolt elongation due to tightening, which is proportional to the bolt length, the length should be so taken that the capacity of the micrometer is utilised to the fullest. Longer the bolt, larger is elongation and lesser will be the percentage error introduced in measurement. This can be shown analytically

Say, the micrometer bolt length = 4"

Micrometer least count = 0.0001"

Taking bolt end surfaces to be absolutely parallel the bolt length can be read as 3.9999 inch or 4.0001 inch

If rate of bolt elongation is 0.001 inch/inch

Then elongation = 0.004 inch

Nominal bolt length after tightening = 4.004 inch

This can be read as 4.0041 inch or 4.0039 inch

The probable values of elongation are

4.0041 - 3.9999 = 0.0042 inch

and 4.0039 - 4.0001 = 0.0038 inch

Maximum error in measurement

= 0.0042 - 0.0038 = 0.0004 inch

Range error $= \frac{0.0004}{0.004} \times 100 = 10\%$

If the bolt length is only one inch

Then elongation = 0.001 inch

$$\%\text{age error} = \frac{0.0004}{0.001} \times 100 = 40\%$$

So it is seen that by taking maximum possible length of bolts, the error due to measurement can be much reduced.

In the Present work the bolts of 1 inch, 3/4 inch and 1/2 inch are made approximately 3 13/16 inch long and 1/4 inch bolt, 3.5 inch long leaving the remaining utilisable capacity of micrometer for cumulative elongation (permanent) in successive bolt tightenings.

To measure the length correctly the micrometer spindles should touch the same point on bolt end surfaces otherwise large in accuracies will be there. To achieve this objective three methods can be tried. In the first, two steel balls are fixed, one on each side, at the bolt faces. Two conical cavities are made at the faces and balls welded or brazed whichever is convenient. In this only point contact is there between the ball and the spindle of the micrometer.

In the second case, small blind holes are drilled in the ends of the bolt and two m.s. turned pins with one end spherical end are press fitted into the holes, the spherical end projecting outside. The contact between the pin end and micrometer spindle is again point contact. It is felt that drilling holes in the ends of the bolts will make them weak, specially the smaller dia. bolts and so this practice is not adopted.

In the third case the bolt end faces are machined parallel to each other within an accuracy of 0.0001 inch. This technique has been adopted here.

The nut face and the bolt head bearing surface are to be parallel but perpendicular to the bolt axis. Even a small deviation from this will lead to severe bending stresses in the bolt and consume a part of the tightening torque.

Other dimensions of the bolts and nuts are shown in the drawings attached.

NUMBER OF BOLTS

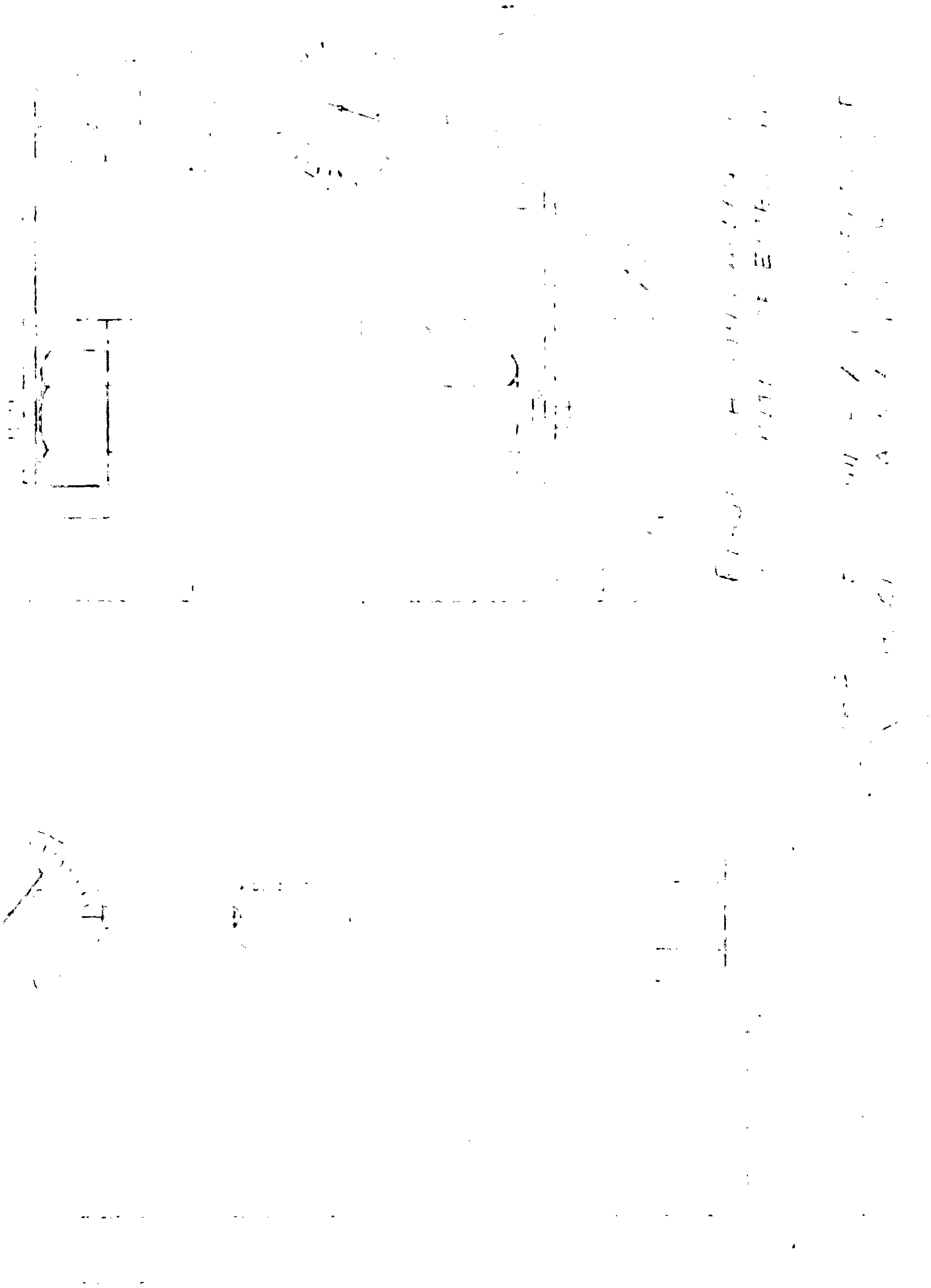
If each bolt is to be tightened once, then for 100 workers to be tested, at least 1600 bolts are required assuming that there is no breakage, because each worker is to tighten four different sizes of bolts with four different types of spanners, a total of 16 tightenings each. However it was found by investigators that 10 TPI threads (UNC & BSW) required about the same torque upto about 80 tightenings after which the torque for UNC were somewhat lower than BSW, although the clamping loads were as high, and the number of tightenings to failure greater than for BSW.

It is found from above that the same bolt can be used effectively in testing for a number of tightenings. It was decided that each bolt of one inch and 3/4 inch size should not be tightened more than 20 times and 1/2 inch and 1/4 inch more than 8 times. The number of tightenings is reduced for smaller bolts because they are frequently tightened beyond yield point leaving a permanent set in the bolt.

1 inch bolts = 20
3/4 inch bolts = 20
1/2 inch bolts = 60
1/4 inch bolts = 80

It was not possible to procure the bolts of these specifications in the market. The bolts available are commercial black bolts cold formed, inconsistent in size. as well as in thread fit. Some foreign machine bolts are available, but the required quantity in above sizes and lengths is not available in one make. Secondly their cost prohibits their use for testing. So it was decided to get these bolts made in the University Workshop where precision working lathes and skilled machinists are available. However the machined, polished but precision nuts of English make are purchased as it was found difficult to make them to close tolerances in the workshop.

The bolt is hot forged from circular mild steel bars. It was ensured that bars are from one lot supplied by the contractor. The bolt head is forged to size. They are machined and threaded on precision lathes and the end faces of the bolt are machined on highly precise lathe. The diameter of bolt was kept within $\pm$ 0.0005 inch and parallelity of end faces was kept within 0.0001 inch. It was inspected by height dial gauge as is shown here.



BEARING BLOCKS

There must be something against which the bolt is to be tightened. Since in the present work only metal to metal joint assembly is to be tested, the block in one piece is made. The material used is cast iron due to its relative ease in moulding to shape and rigidity. In practice bolts are usually tightened against cast iron.

The length of block is decided to take maximum advantage of bolt length. Bolt inside the block only is stretched. A portion of screw engaged with nut is also stressed.

The area of block in compression is kept to minimise the block deflection and make it rigid. The block used here with dimensions are shown in the drawings attached. The faces of the block are to be smoothed and parallel but perpendicular to the hole axis. Any departure from this will result in bending of bolt body.

To ensure this, instead of machining blocks individually, a number of blocks of one size are cast in one piece to close tolerances where machining is not required. Two faces are milled on a milling machine very accurately and inspected with a height gauge. The centres for holes to be drilled are marked and holes drilled. It was ensured that lower machined surface of the block is in flush with the bed of the drilling machine. After drilling and reaming the holes, the blocks of required sizes were cut on a bandsaw cutting machine. Effort was made to make the holes exactly in the centre of the block. Eccentricity of hole would result in eccentric loading on the block, which was not of much concern to us. This would have been important only if strain gauges were pasted on the block to determine the bolt load.

To prevent the rotation of the bolt while tightening, two mild steel strips with sides filed are welded on one face of block to catch the bolt head.

CALIBRATION RIG

Calibration rig is a device for mounting the bolt in the Universal Testing Machine for loading it in pure tension and allow the measurement of bolt elongation to made. It consists of two identical pairs, each pair having two elements. One element is a cup shaped nut with a drill bottom while the other is a threaded plug, the unthreaded portion of which is turned down to a smaller diameter to fit into the jaw of the testing machine. The head of the bolt is caught in one pair while the nut in the other one. Two M.S. strips were welded to the bottom surfaces of the nuts of the pairs. The magnetic bases of the dial gauges were clamped on the stationary jaw seat and the gauge spindles pressed against these strips. When bolt was loaded, the difference of the readings of these two dial gauges gave the bolt elongation.

The walls and bottom of the calibration rig nut were made liberally thick to minimise the effects of their deflection under load on the readings. The thin bottom would deflect like a centrally loaded plate and change the slope of the stripe resulting in erroneous results. The bearing faces of the bottoms of rig nuts were machined carefully perpendicular to the hole axis and nuts centre line to prevent eccentric loading of the bolt. The hole diameter was made slightly larger than bolt diameter for adjustment. The rigs were forged and machined in the University Workshop.

For pure tensile loading of the bolt, the bolt axis would be in line with the loading axis of the machine. Even a little eccentricity would involve tilting of the rig nut bottom and affect the dial readings.

It was found that this practice of measuring bolt elongation did not work well. The difference of dial gauge readings was a sum total of bolt elongation, deflection of bolt threads, deflection of rig ends and rig nut bottoms. The jerk due to slip at grips also affected the readings considerablly. This was overlooked during the designing of the rig. The deflections of rig ends and rig nut bottom were sought to be made negligible by providing much larger sections. However this indirect method of extension measurement did not succeed and some direct methods of measurement was sought.

For this purpose, two rectangular windows were milled in each rig nut to have access to the bolt end faces. Small holes were drilled and tapped in each face of the bolt and m.s. strips were bolted to these faces. The other end of these strips came out of the rig nuts through the windows and carried round pins. The knife edges of the extensometer were fixed on these pins. Small washers were put under the m.s. strips to make the strips unaffected by small deflection of the bolt head. This technique of measurement of bolt elongation worked quite satisfactorily. The various drawings of these rigs are attached.

SPANNERS

The spanners of various sizes and types were taken from Analysis of Machine Lab. but two ring spanners for the one inch dia bolt and 3/4 inch dia bolt could not be found. These spanners were of standard lengths as supplied by the manufacturers and their dimensions are given in the appendix.

MICROMETER

Vernier micrometer of range from 3" to 4" with least count of 0.0001 inch was used.

CHAPTER IX

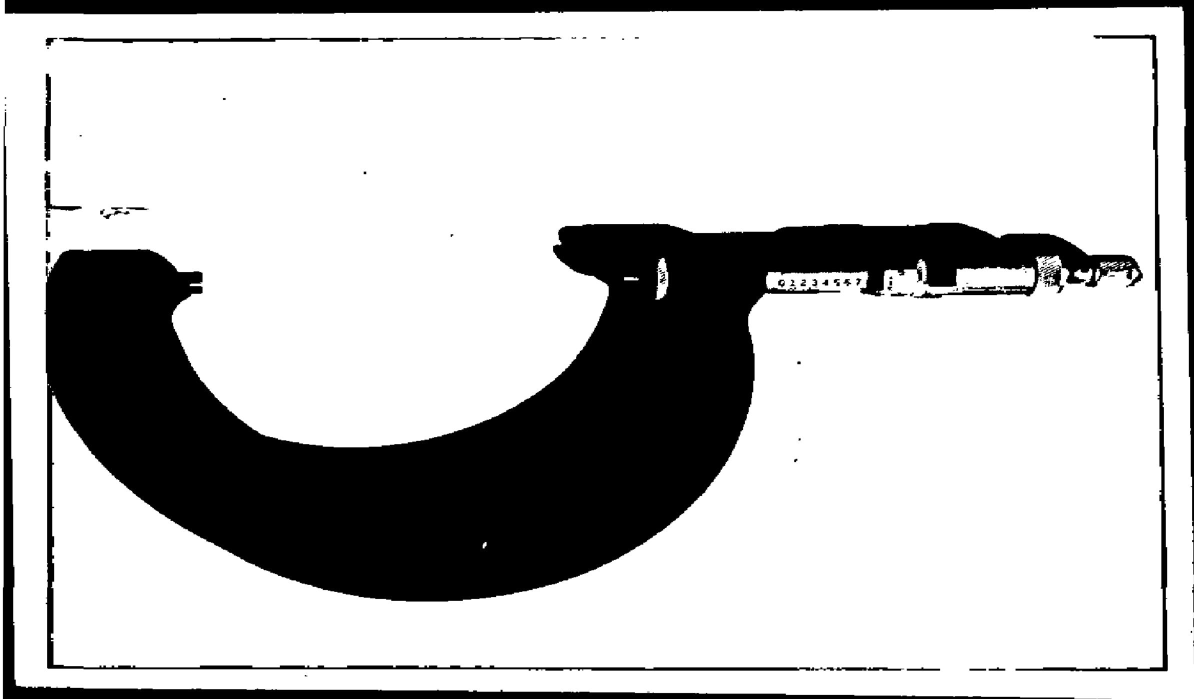
TESTING PROCEDURE

The procedure for carrying out the testing programme is to be simple, clear, systematic and properly laid out because the final results depend upon this significantly.

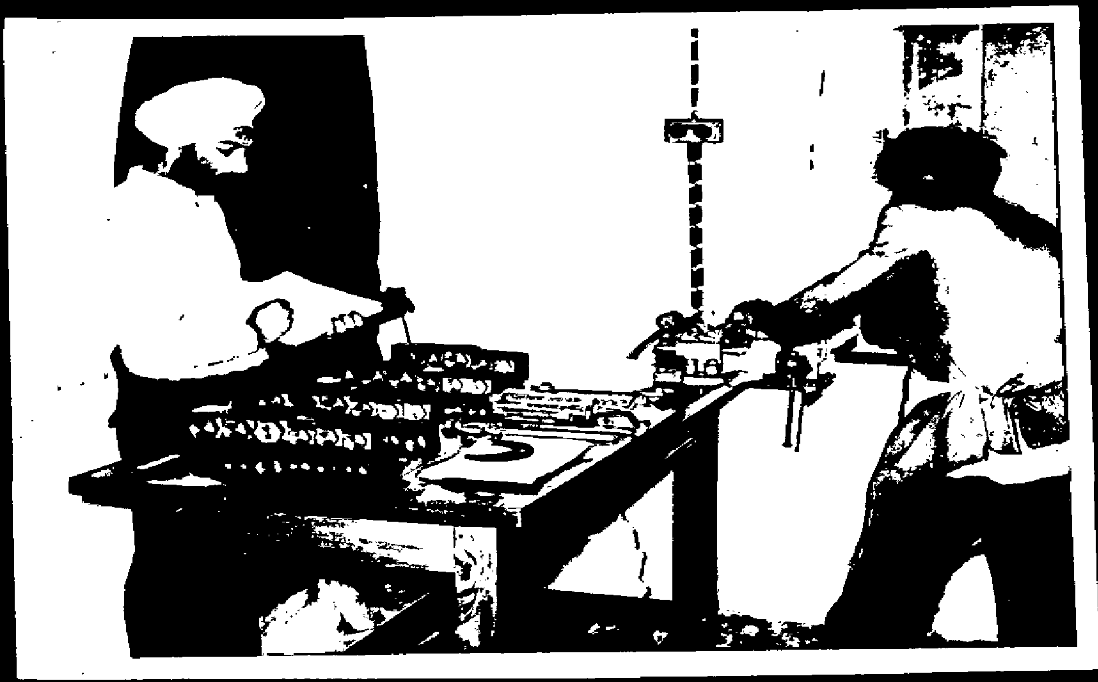
It was decided to test about 100 workers from the various workshops of the city. Each worker would be required to tighten 10 times at a stretch and it was expected that it would take about 20 minutes for each worker to do this. The bolt length was to be measured accurately before and after the tightening with the micrometer which it was estimated, would take an hour before and an hour after the tightenings. It was concluded from this that about 8 workers could be tested in a working day of approximately 7 working hours and out of these 8 workers to be tested per day, 4 would be tested in the first half of the day and the rest in the second half. In this way the blocks and bolts used in the first half day could again be used in the second half day. Minimum of 64 blocks with bolts would then be required at a time.

To start with, the bolt and blocks were cleaned with kerosene and petrol to remove all traces of dirt, scale and grease. To prevent the machined surfaces from rust, a very thin layer of Mobil Oil was applied with a cloth. The bolts were divided into 4 groups, each group having 4 blocks of each size. These groups were marked A, B, C, and D and blocks of each group were further marked from 1 to 20 to differentiate one from the other. The bolts were pushed into the blocks and nuts screwed loosely on them.

The initial length of each of these bolts was measured accurately with the Vernier micrometer and was noted down systematically in the file. Each worker was called and asked to tighten the bolts. He was told that he should tighten them to the value that he felt would make the joint loose fluid tight. He was asked to tighten 1/4" bolts first, then 1/2", 3/4" and then 1" size. It was felt that this sequence gave better control on the physical strength. Since 1/4" and 1/2" bolts require more of skilled than strength, the operator would regulate the force applied on the spanner according to his 'feel' and would not be tempted to tight with full force. But if the worker tightens 3/4" and 1" bolts first and then 1/4" and 1/2" bolts, he would be tempted to apply more force because 1" and 3/4" bolts had demanded full force from him. After the workers had tightened the bolts, their lengths were again measured accurately with the micrometer and were noted down against the initial length readings. After the measurement had been made the bolts were unloaded and kept for at least an hour to give them sufficient time to recover to their free length. Again the same bolts were tested for the next batch of 4 workers. When the smaller bolts of 1/4" and 1/2" size had been tightened 8 times, they were removed from the blocks and replaced with the new ones. The measurement of initial length before each tightening was necessitated due to the plastic strains.



VERNIER MICROMETER
Range 3" - 4"
Least Count = 0.0001"



TEST COMPONENTS

MEASUREMENT OF BOLT LENGTH

group were further marked from 1 to 16 to differentiate one from the other. The bolts were pushed into the blocks and nuts screwed loosely on them.

The initial length of each of these bolts was measured accurately with the Vernier micrometer and was noted down systematically in the file. Each worker was called and asked to tighten the bolts. He was told that he should tighten them to the value that he felt would make the joint leese fluid tight. He was asked to tighten 1/4" bolts first, then 1/2", 3/4" and then 1" size. It was felt that this sequence gave better control on the physical strength. Since 1/4" and 1/2" bolts require more of skilled than strength, the operator would regulate the force applied on the spanner according to his 'feel' and would not be tempted to tight with full force. But if the worker tightens 3/4" and 1" bolts first and then 1/4" and 1/2" bolts, he would be tempted to apply more force because 1" and 3/4" bolts had demanded full force from him. After the workers had tightened the bolts, their lengths were again measured accurately with the micrometer and were noted down against the initial length readings. After the measurement had been made the bolts were unloaded and kept for at least an hour to give them sufficient time to recover to their free length. Again the same bolts were tested for the next batch of 4 workers. When the smaller bolts of 1/4" and 1/2" size had been tightened 8 times, they were removed from the blocks and replaced with the new ones. The measurement of initial length before each tightening was necessitated due to the plastic strains.

In all 10 spanners were required for tightening but out of these two ring spanners for 3/4" and 1" bolts could not be procured.

CALIBRATION

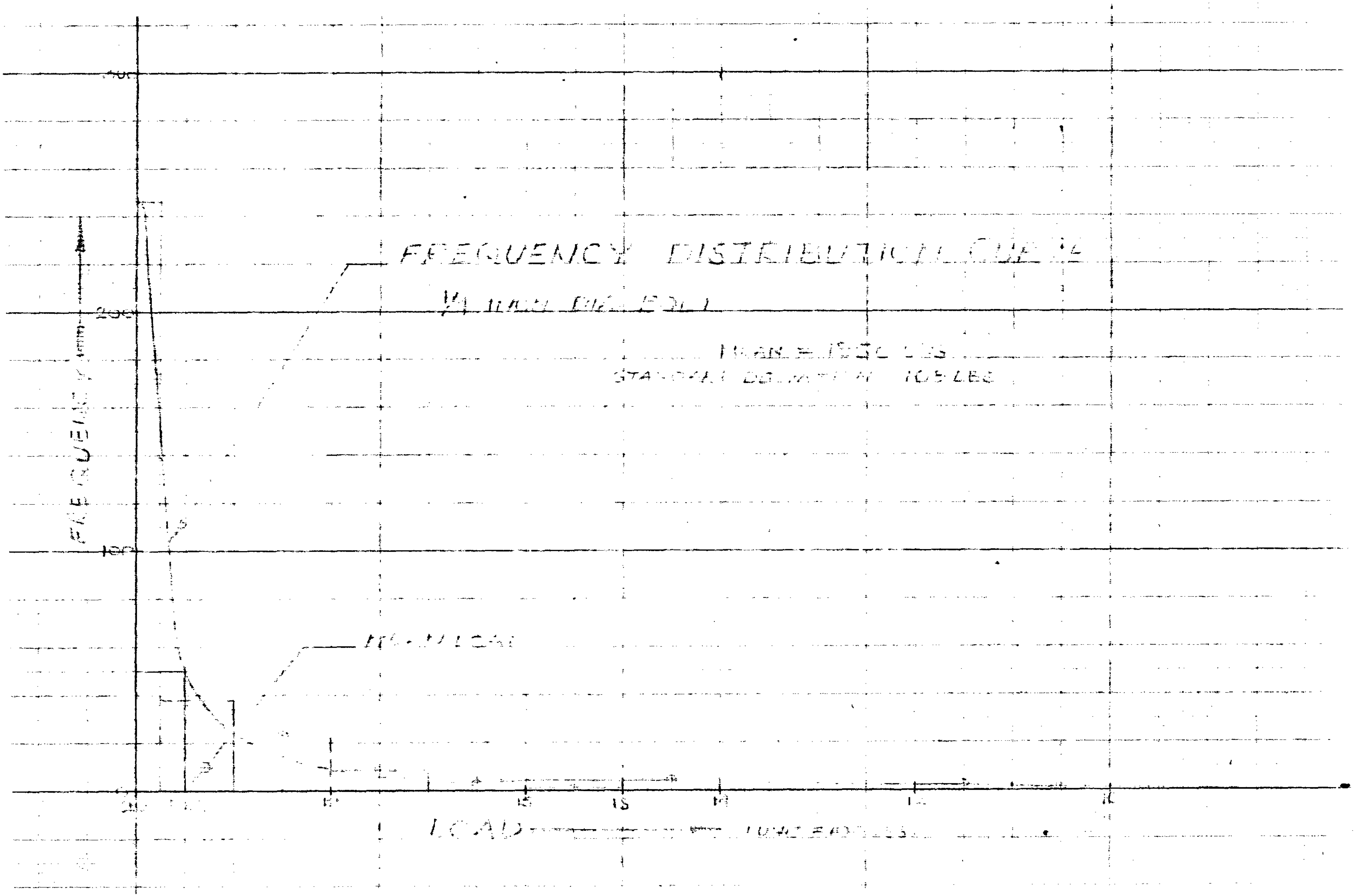
After the above data was collected, the bolts were tested in pure tension to draw the calibration curves. Two Universal Testing Machines were available in material testing lab., one is of 100 tons capacity and the other is of 20 tons maximum capacity but adjustable to maximum capacity of 3 tons. The gripping devices for the second machine were found to be better.

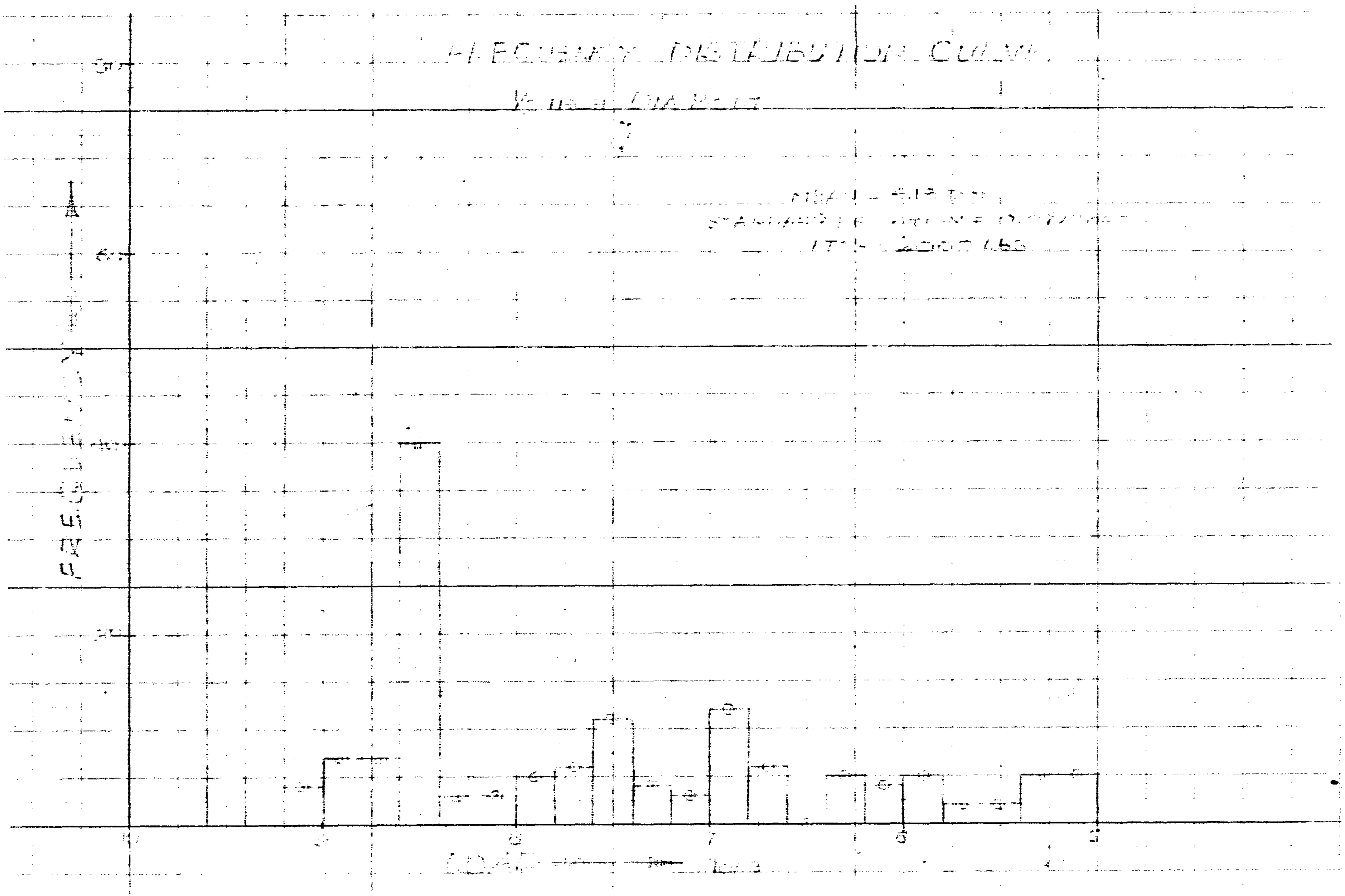
In the first trail, the attempt was made to measure the bolt elongation indirectly with the help of mechanical dial gauges. As explained earlier the bolt was caught in the rig, and the ends of the rig were fixed in the jaws of the machine. Several trials were made and results observed but they were not found to be satisfactory. It was found that the difference of two dial gauge readings which should give bolt elongation was not bolt elongation alone but sum of bolt elongation, thread deflections and rig deflections. Also the rigs got tilted due to eccentricity of the bolt axis from the rig axis and even the slight tilting of the rigs appreciably affected the gauges being mounted at the ends of relatively long strips.

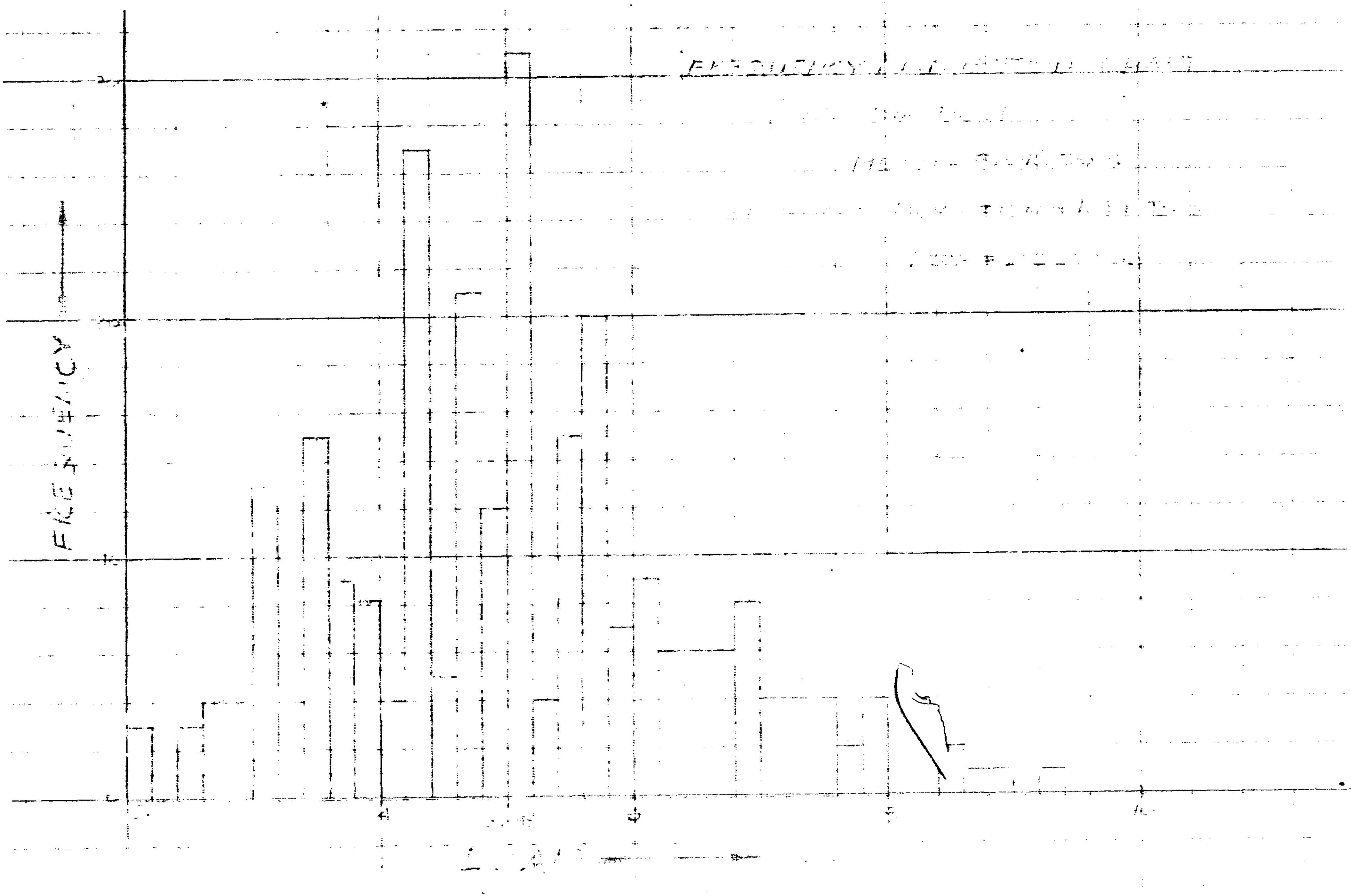
This difficulty was overcome by making measurements directly on the bolt. The extensometer was mounted on the pins brazed on the projecting ends of the m.s. strips bolted to the bolt ends. This system worked well for 1", 3/4" and 1/2" bolts but did not work for 1/4" bolt because 1/8" dia. hole drilled in the end made it very weak. So 1/4" bolt was tested with the previous method. This method worked well for this bolt because the diameter of the bolt is small as compared with the rig sections and load on it is so small that the other factors become negligible.

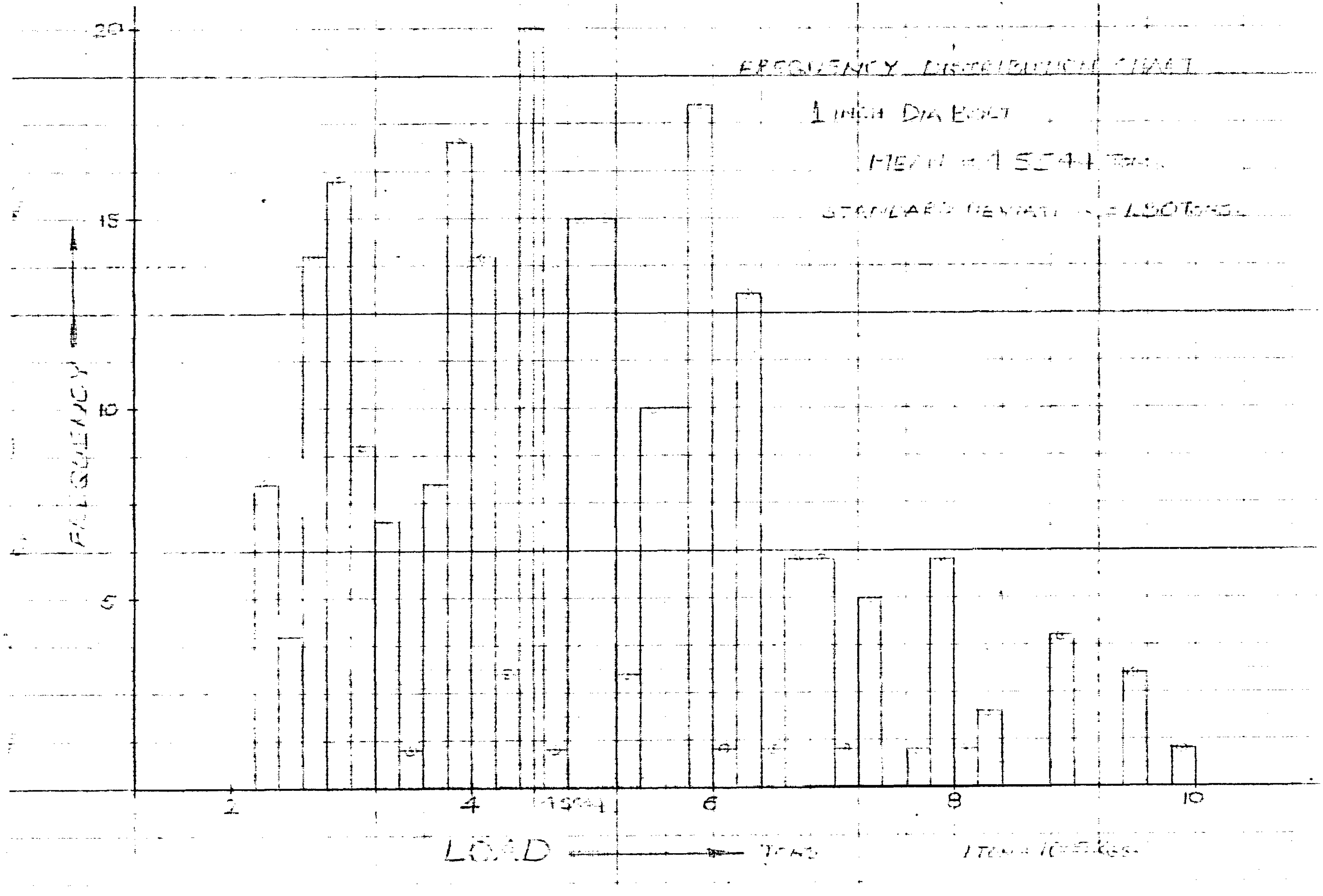
A number of bolts were tested for each diameter and the average values were taken and the calibration curves were drawn.

Since 1" and 3/4" bolts are not loaded beyond proportional limit when tightened by workers, the calibration curve will be same for all the tightenings. But 1/4" and 1/2" bolts are frequently loaded much beyond the proportional limit, there will be strain hardening of the material due to plastic flow and so one calibration curve would not serve the purpose. Calibration curves for each of the tightenings on the bolt are to be drawn. For this the average elongation of the bolt was found from the collected data for each of the tightenings. The bolt was loaded in the testing machine to a load where extension in the bolt length was of the value as calculated for the first tightening, the bolt was unloaded, given a little rest to cool and again reloaded to a load where the elongation in the bolt length was of the value of average extension for second tightening as found from the data. Again

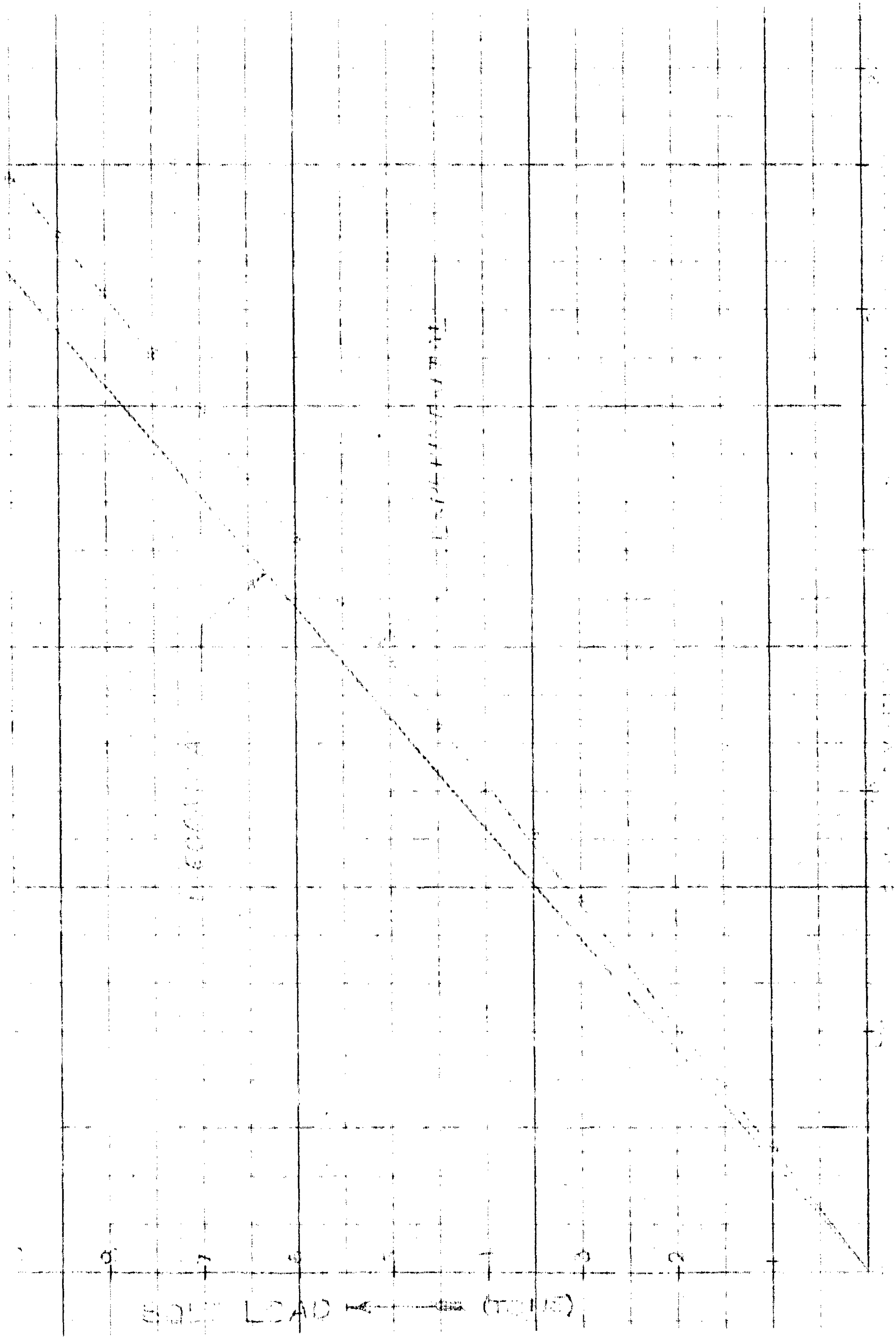

FREQUENCY
200
100
LOAD

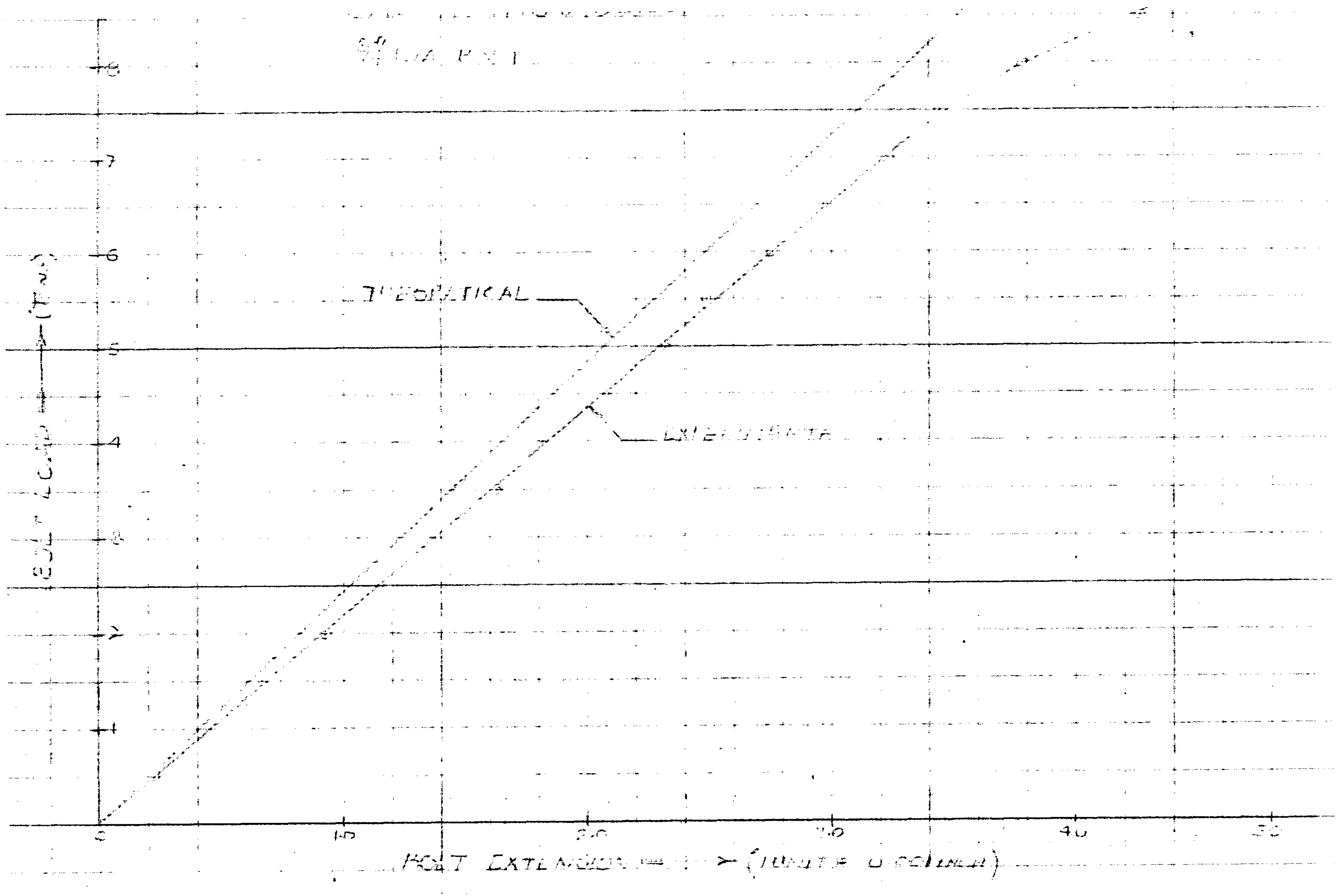

FREQUENCY DISTRIBUTION CURVE
5
6
7
8
9
LOAD
FREQUENCY


FREQUENCY

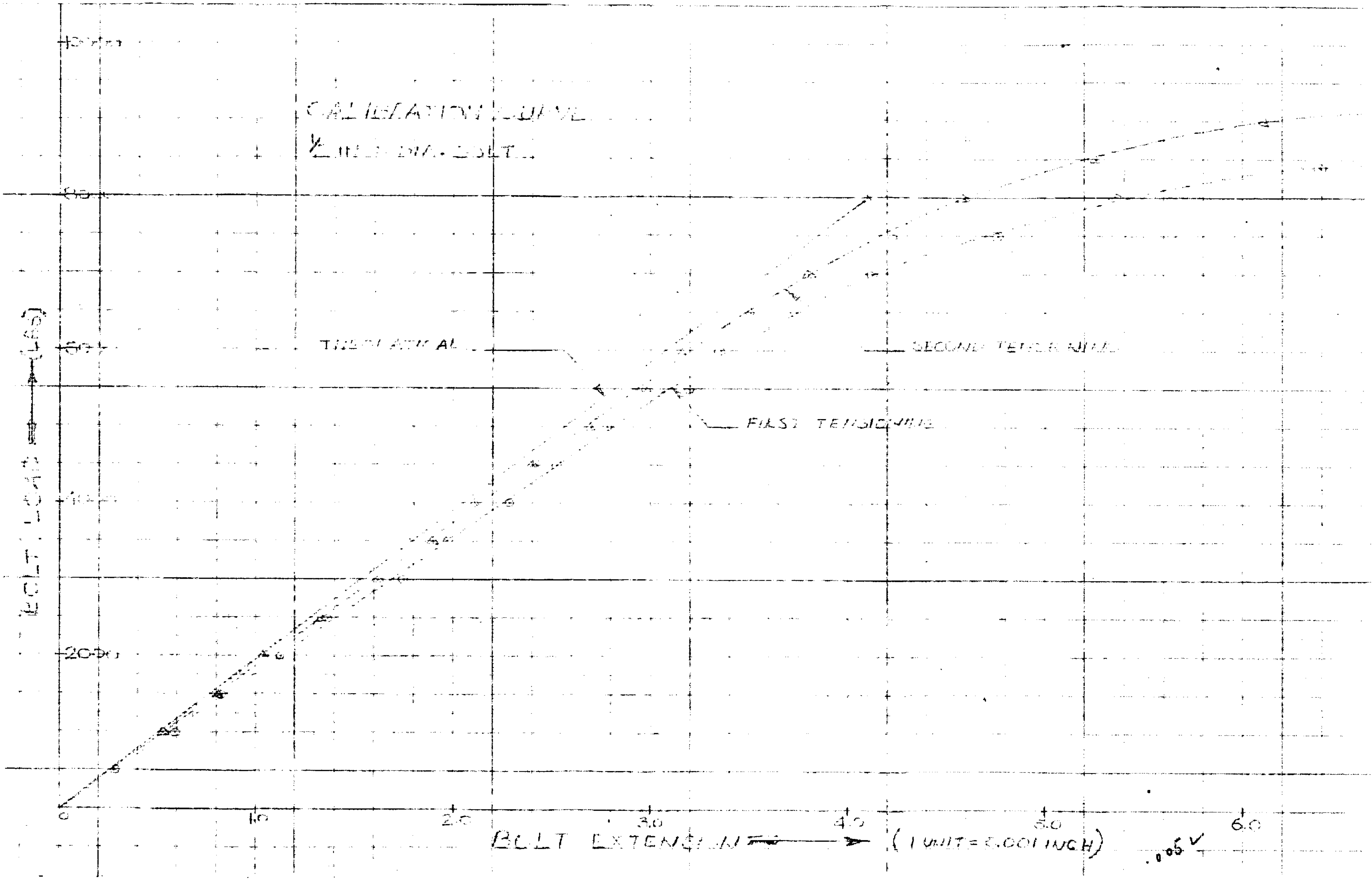


FREQUENCY
1 INCH DIA BOLT
20
15
10
5
2
4
6
8
10
LOAD

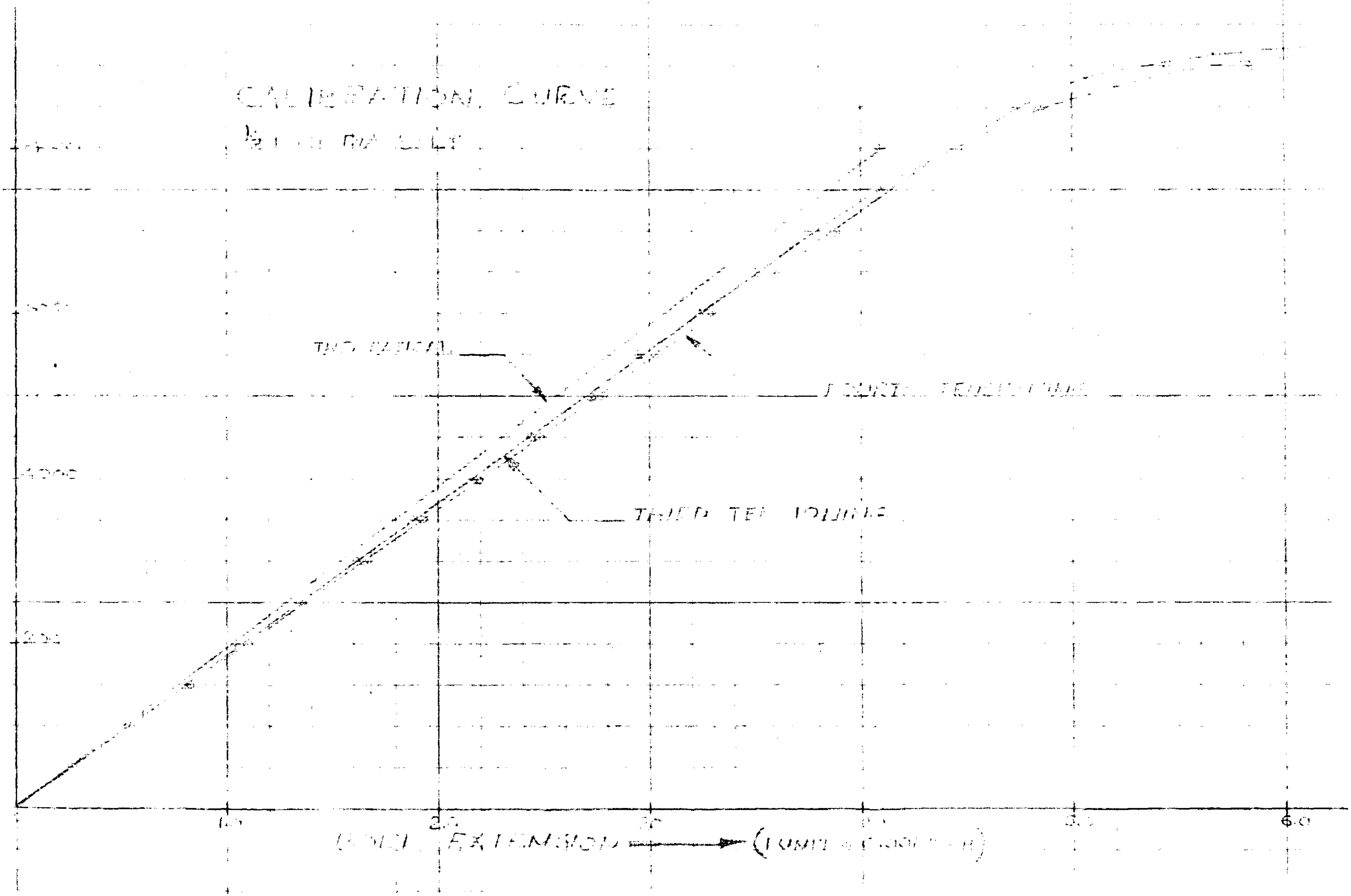

BOLT LOAD (TONS)



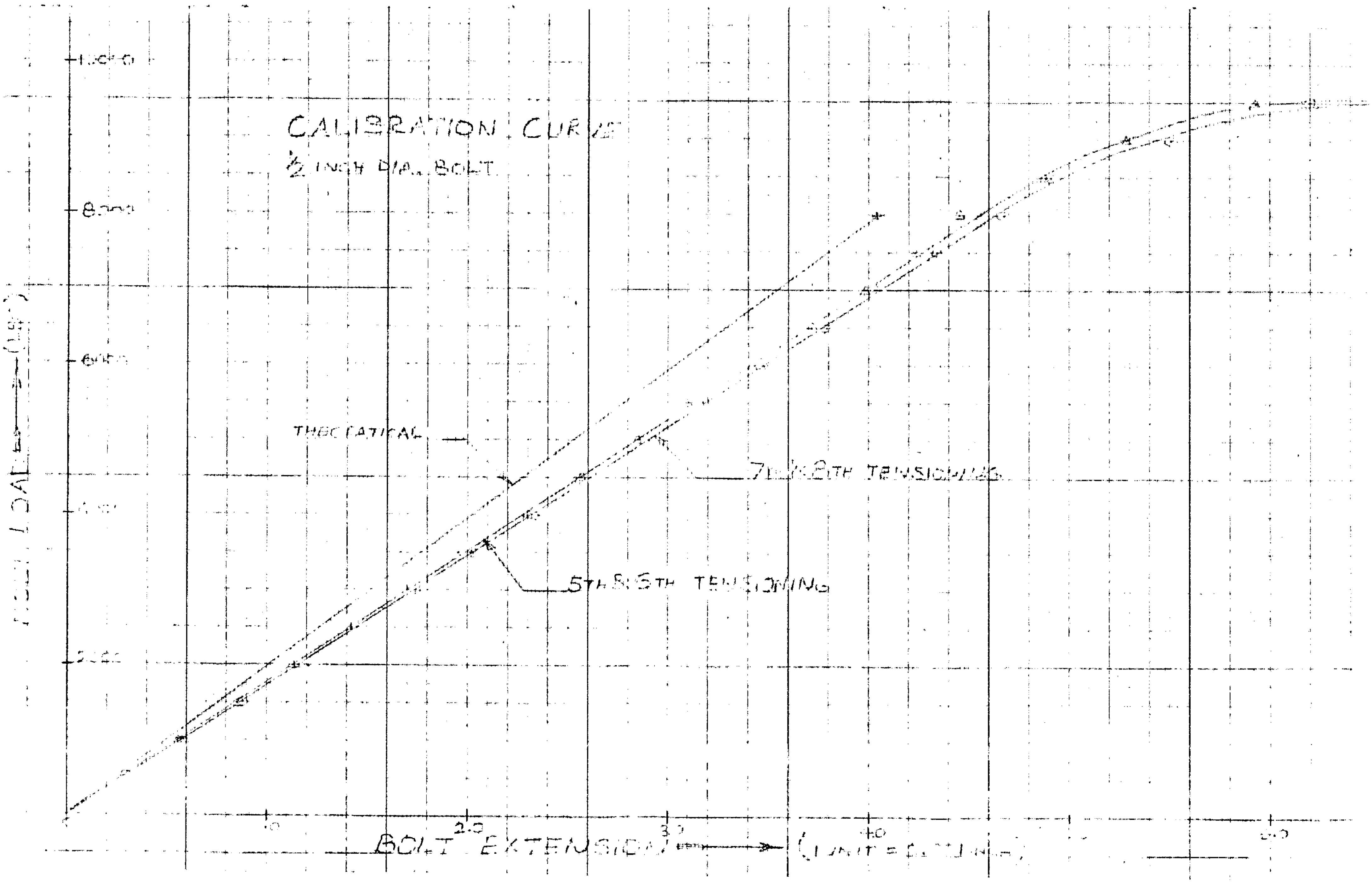
THEORETICAL
0
5
10
20
30
40
50
1
2
3
4
5
6
7
8



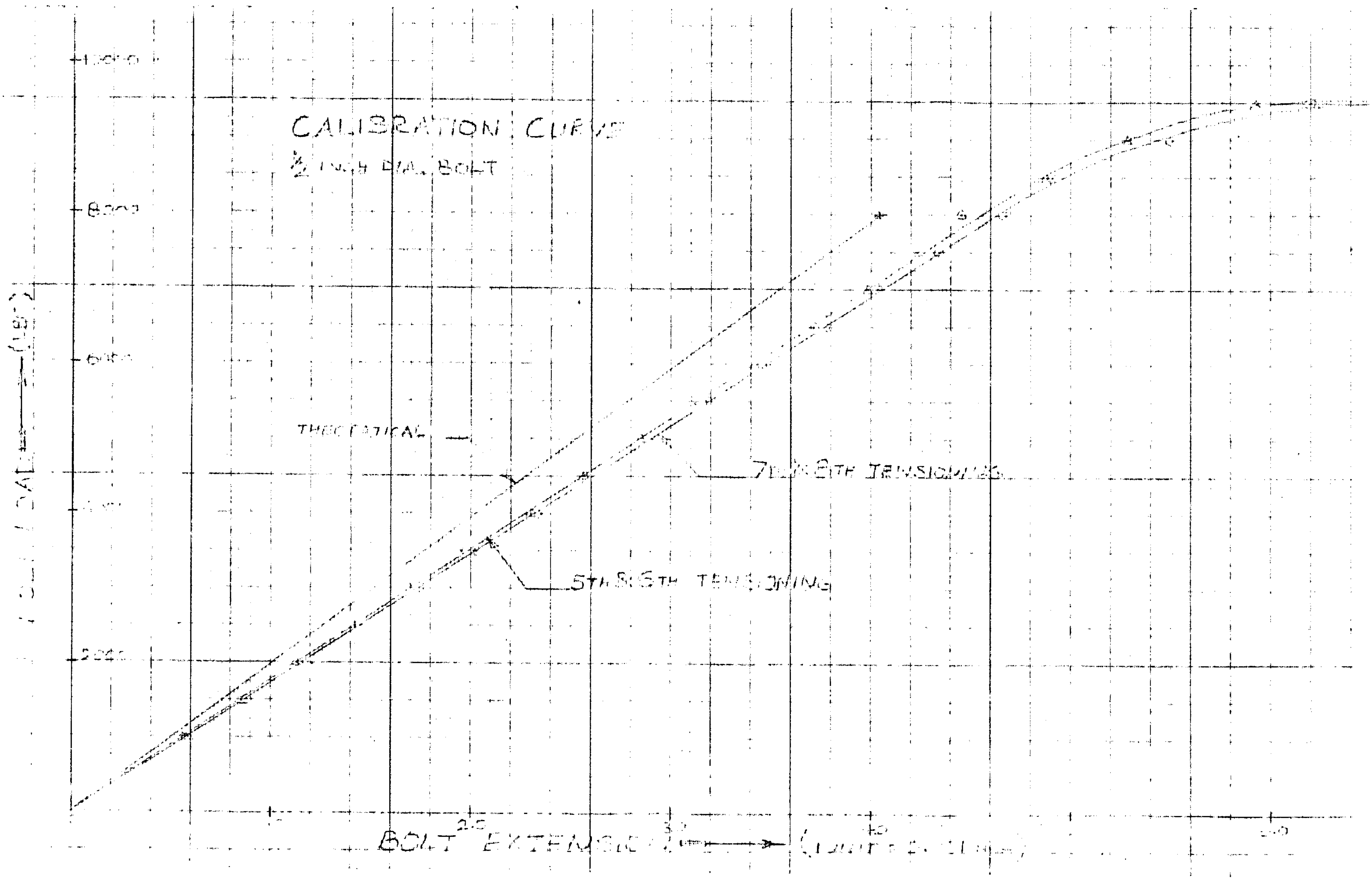
CALIBRATION CURVE
½ INCH DIA. BOLT
THEORETICAL
SECOND TENSIONING
FIRST TENSIONING
0
1.0
2.0
3.0
4.0
5.0
6.0
BOLT EXTENSION
(1 UNIT = 0.001 INCH)
BOLT LOAD (LBS)

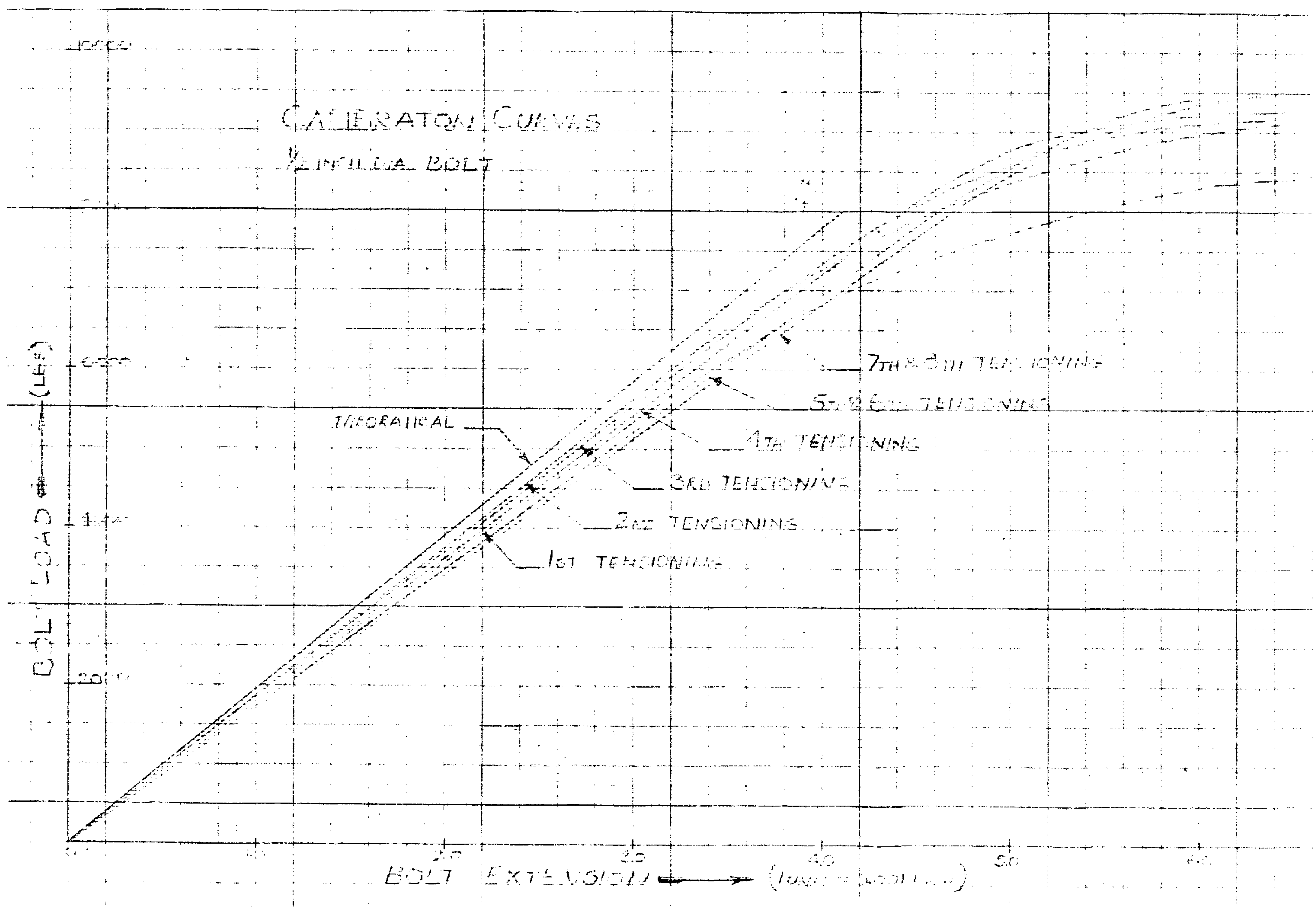


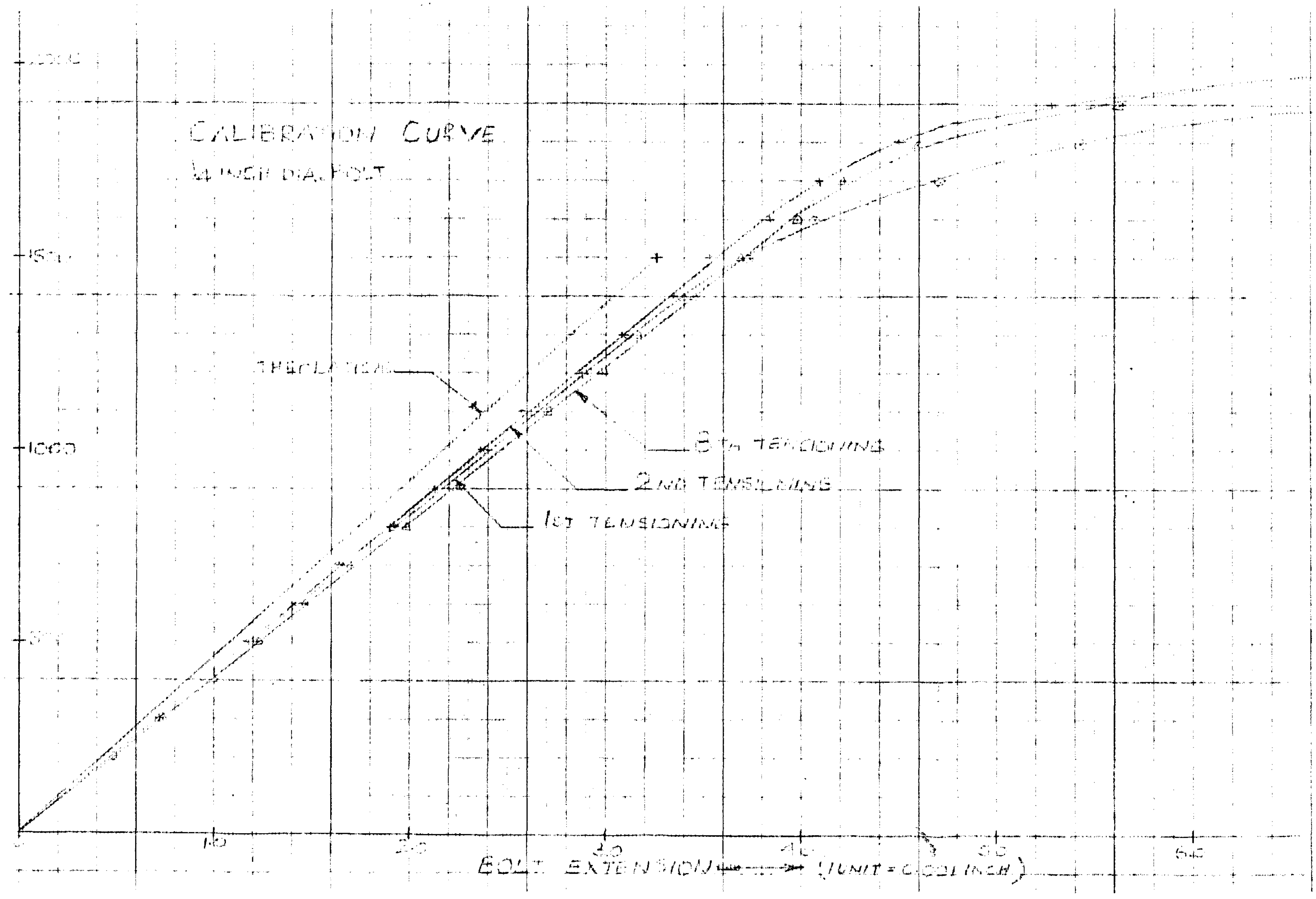
CALIBRATION CURVE
TWO RADIAL
THIRD TEN POUNDS
4000
2.0
6.0
EXTENSION



CALIBRATION CURVE
½ INCH DIA. BOLT
8000
6000
2000
THEORATICAL
7TH & 8TH TENSIONING
5TH & 6TH TENSIONING
1.0
2.0
3.0
4.0
5.0
BOLT EXTENSION


CALIBRATION CURVE
½ INCH DIA. BOLT
THEORETICAL
7 LENGTH TENSIONING
5 LENGTH TENSIONING
BOLT EXTENSION → (UNITS ...)
BOLT LOAD → (LB.)
2000
8000
10000


CALIBRATION CURVES
½ INFILIA BOLT
10000
6000
4000
2000
THEORETICAL
1ST TENSIONING
2ND TENSIONING
3RD TENSIONING
4TH TENSIONING
5TH & 6TH TENSIONING
7TH & 8TH TENSIONING
BOLT LOAD (LBS)
1.0
2.0
3.0
4.0
5.0
6.0
BOLT EXTENSION



CALIBRATION CURVE
1000
1st TENSIONING
2nd TENSIONING
8th TENSIONING
BOLT EXTENSION

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30. Machine Design. Shigley.

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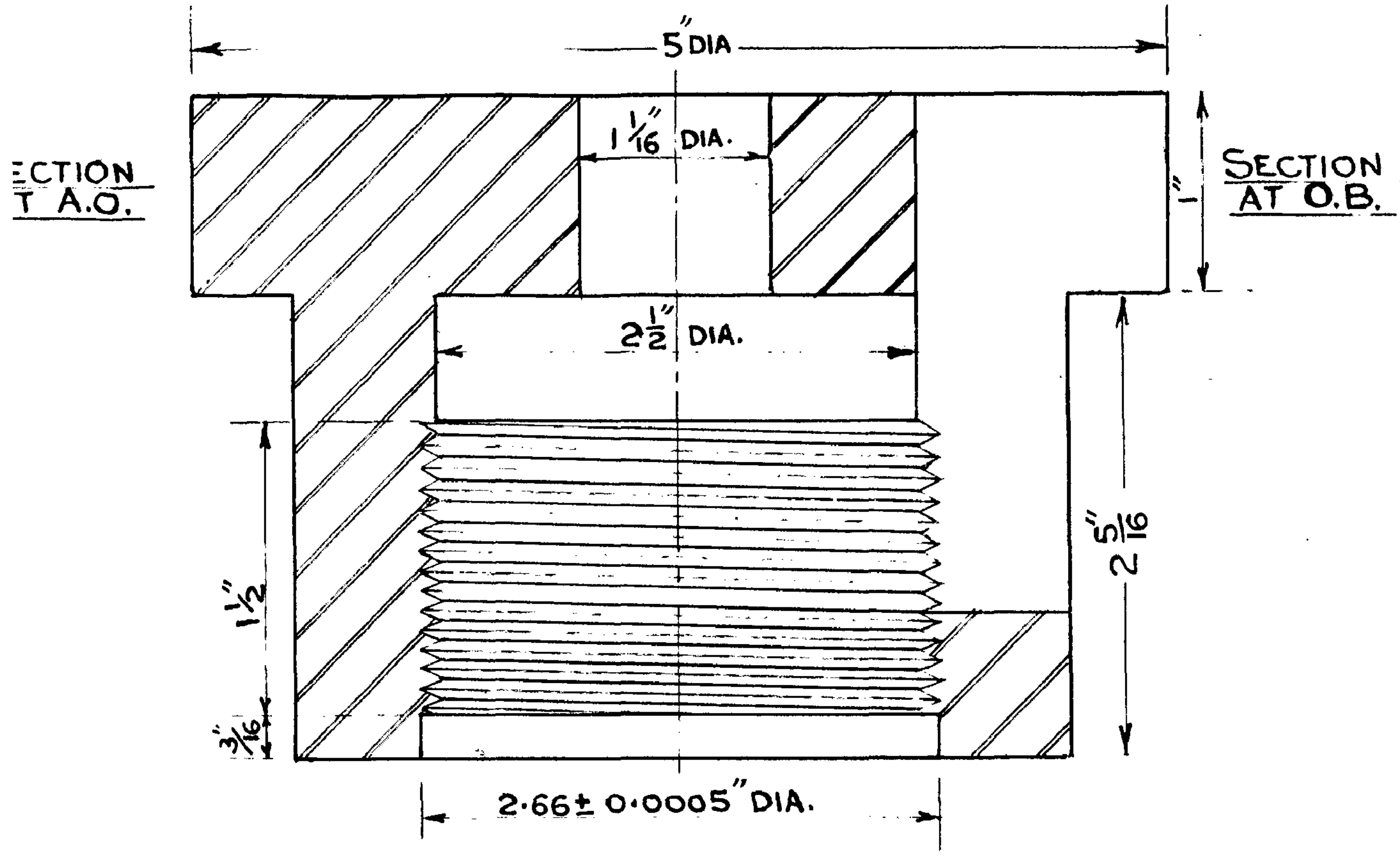
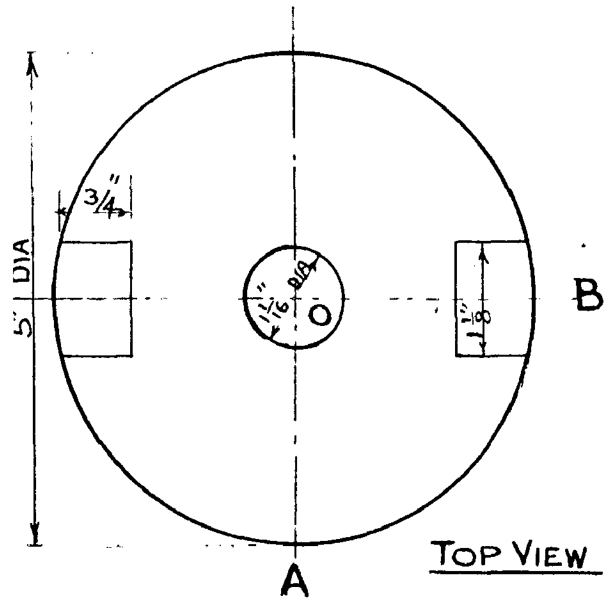


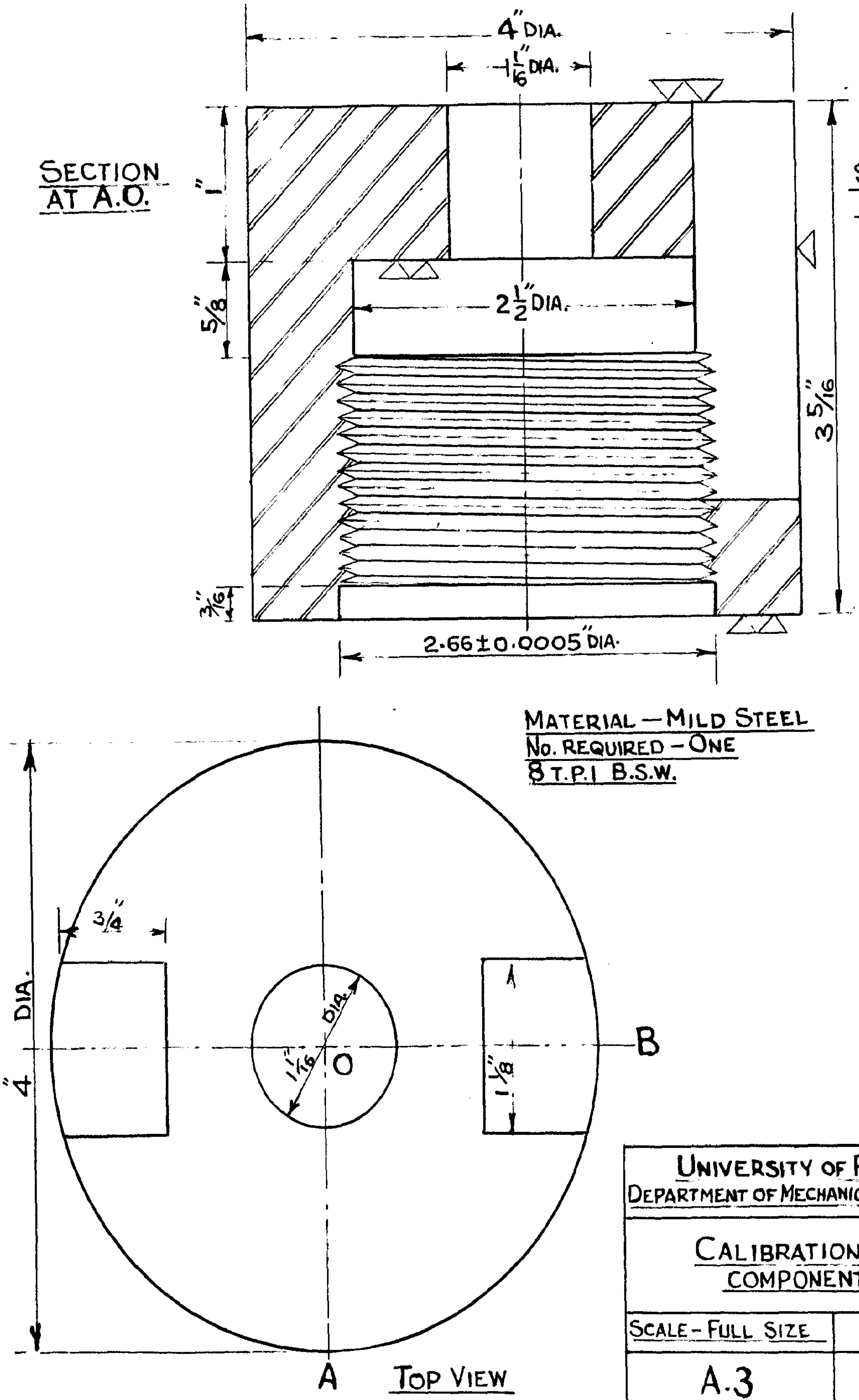
MATERIAL – MILD STEEL
No. REQUIRED – ONE
8 T.P.I. B.S.W.



TOP VIEW

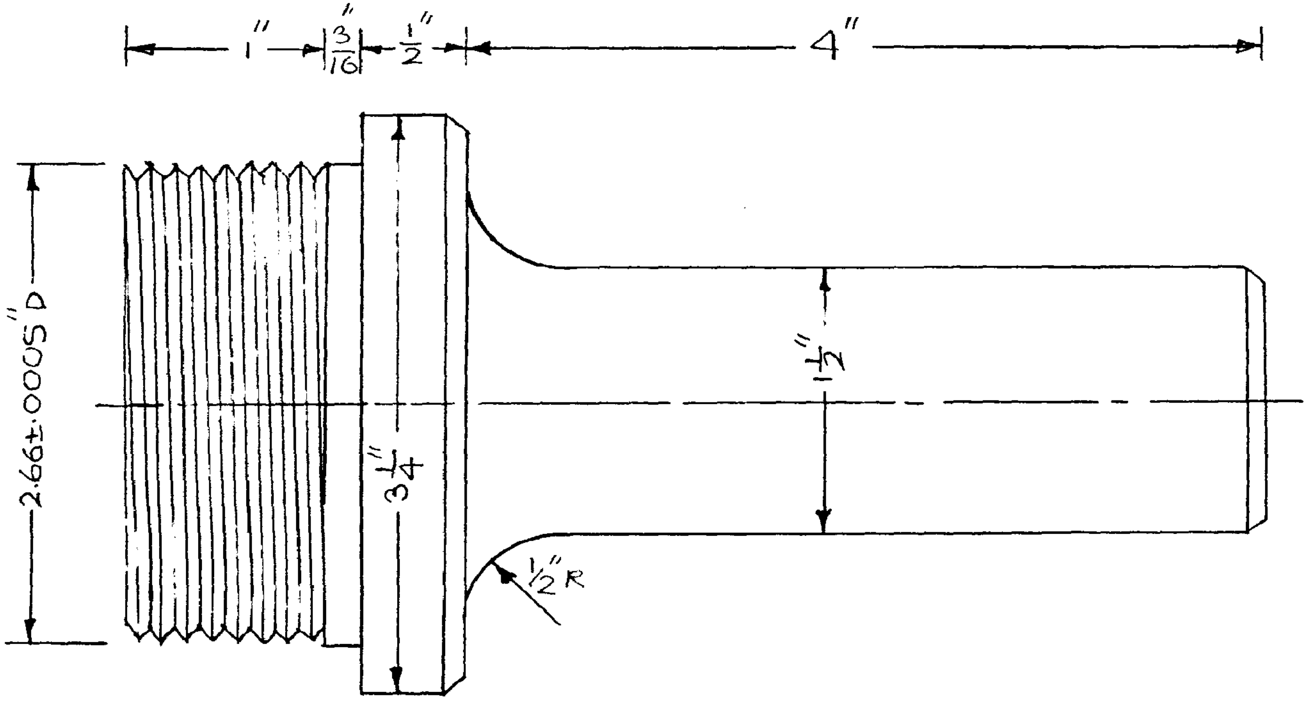
SCALE 1/2 FULL SIZE

UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE – FULL SIZE	20-9-63
A: 2	B.S.A.



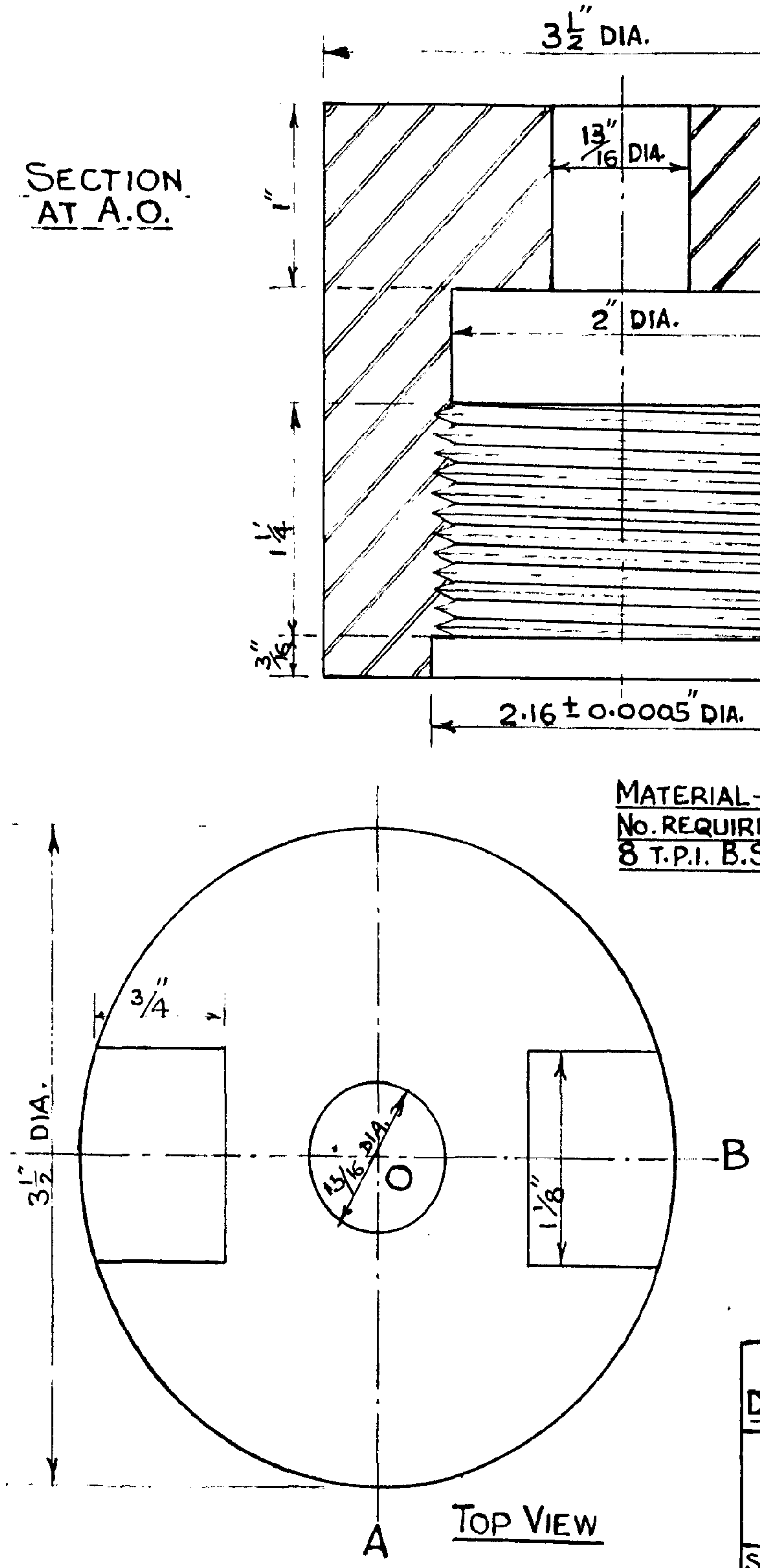
MATERIAL — MILD STEEL
No. REQUIRED — ONE
8 T.P.I B.S.W.

UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE - FULL SIZE	20-9-63
A.3	B.S.A.



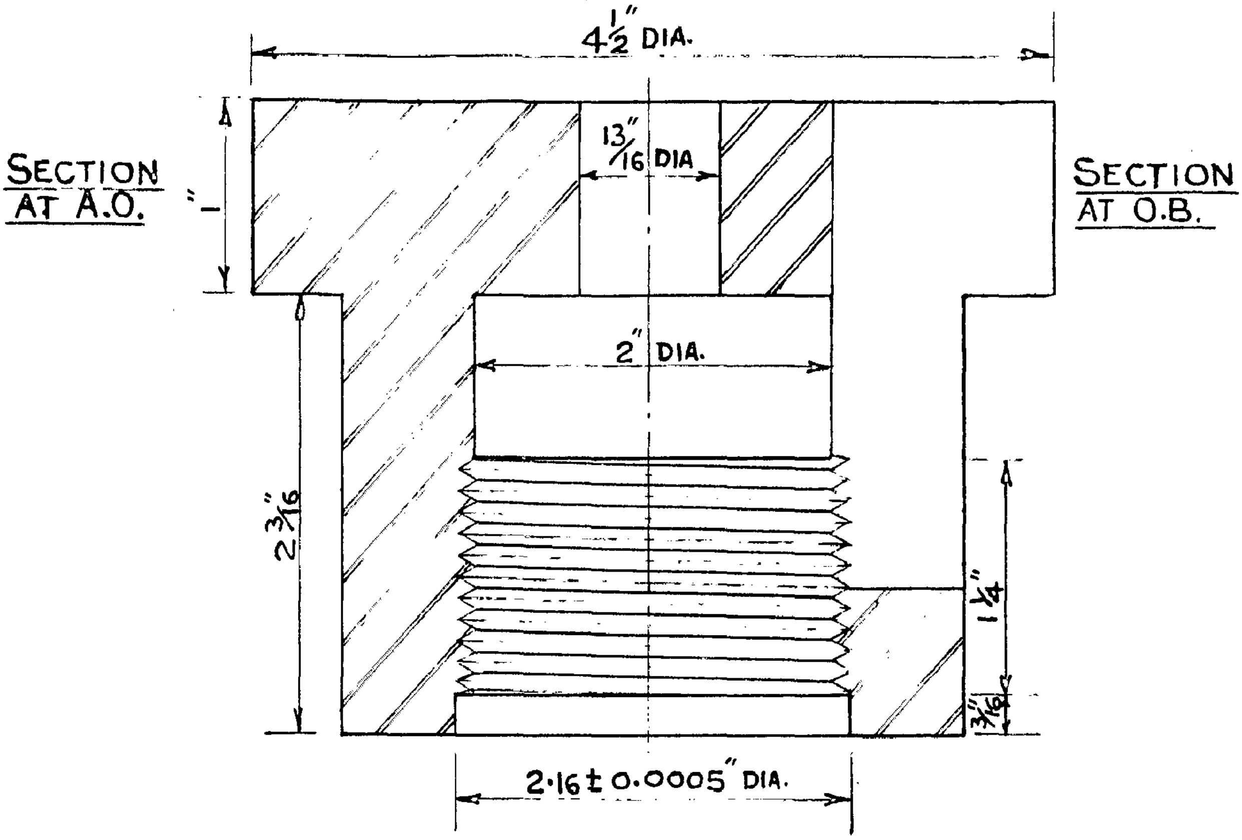
MATERIAL — MILD STEEL
NO. REQD — TWO
8 T.P.I. B.SW

UNIVERSITY OF ROORKEE DEPT. OF. MECH. ENGG.	
CALIBRATION RIG COMPONENT	
FULLSCALE	20.9.63
A4	B.S.A



MATERIAL – MILD STEEL
No. REQUIRED – ONE
8 T.P.I. B.S.W.

UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE – FULL SIZE	20 – 9 – 63
B.2	B.S.A.



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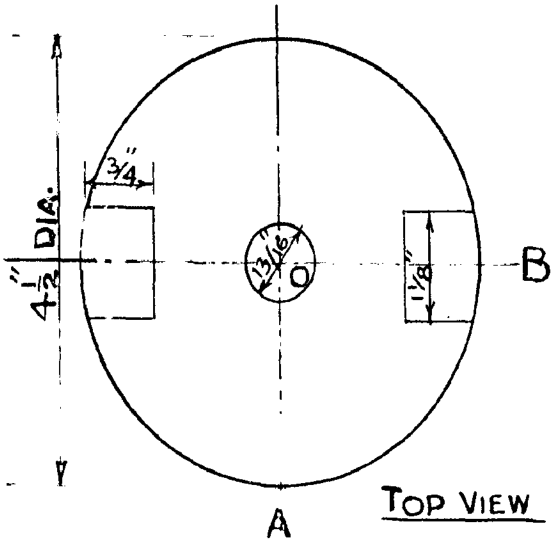
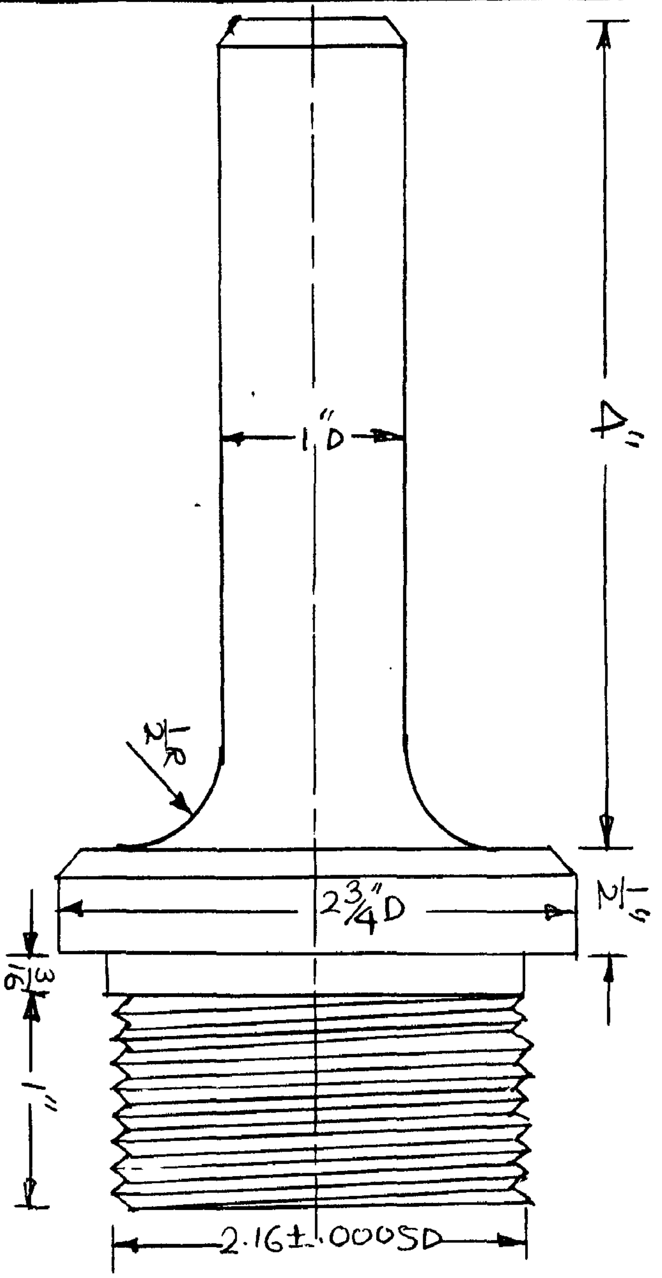
No. REQUIRED – ONE
8 T.P.I. B.S.W.



TOP VIEW

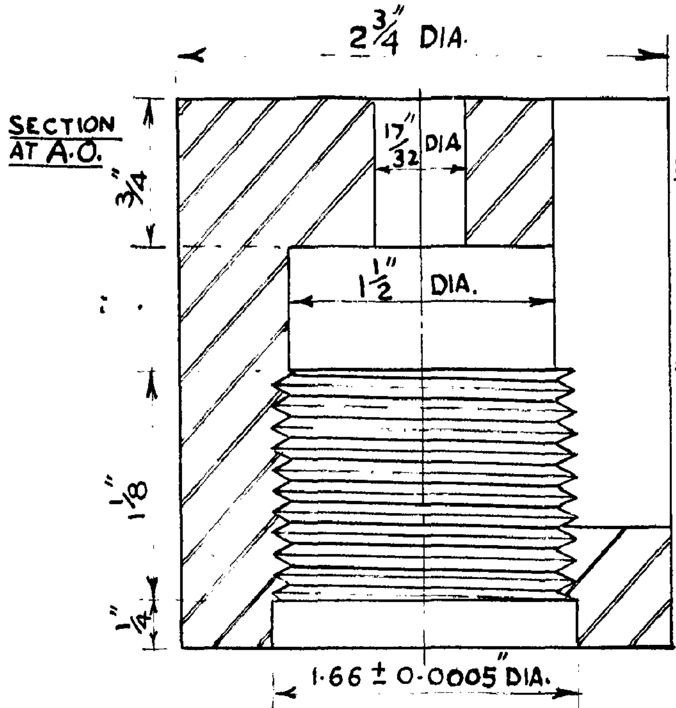
SCALE ½ FULL SIZE

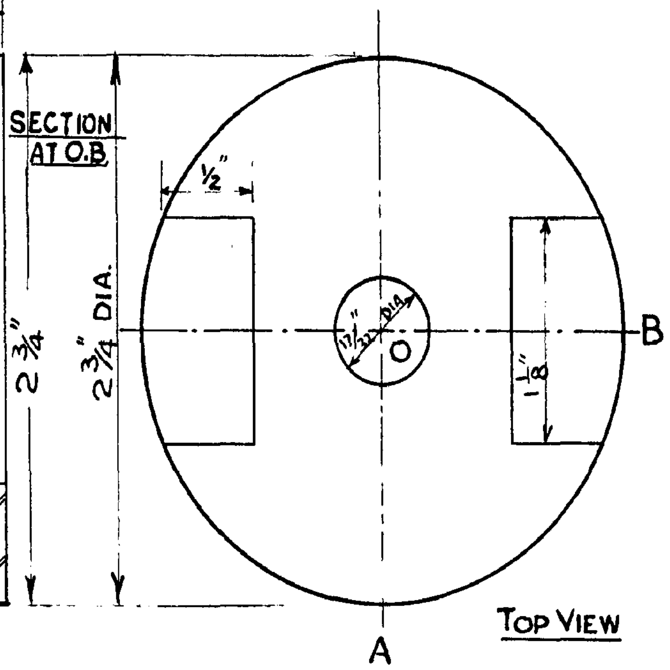
UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE:– FULL SIZE	20-9-63
B·3	B.S.A.



MATERIAL — MILD STEEL
NO REQD — TWO
8 T.P.I. B.S.W

UNIVERSITY OF ROORKEE DEPT. OF. MECH. ENGG.	
CALIBRATION RIG COMPONENT	
FULL SCALE	20.9.63
B-4	B.S.A





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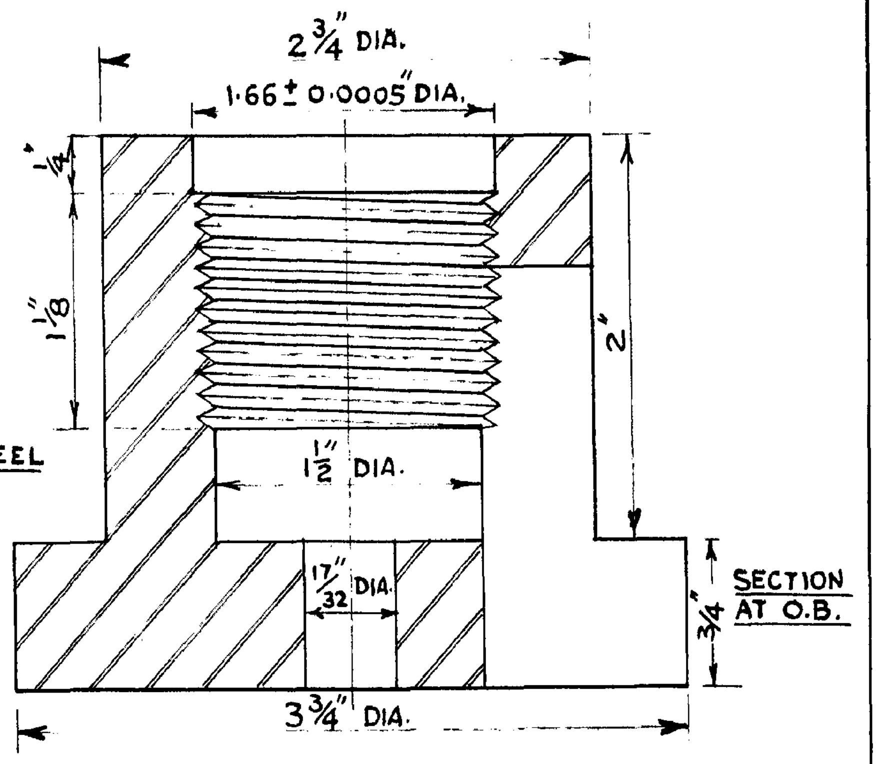
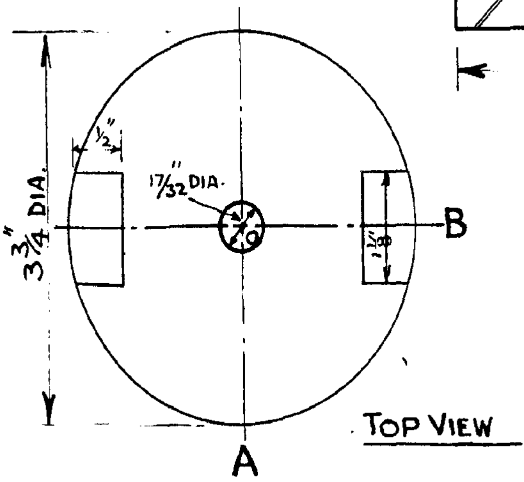
NO. REQUIRED – ONE
8 T.P.I. B.S.W.



MATERIAL – MILD STEEL
NO. REQUIRED – ONE
8-T.P.I. B.S.W.

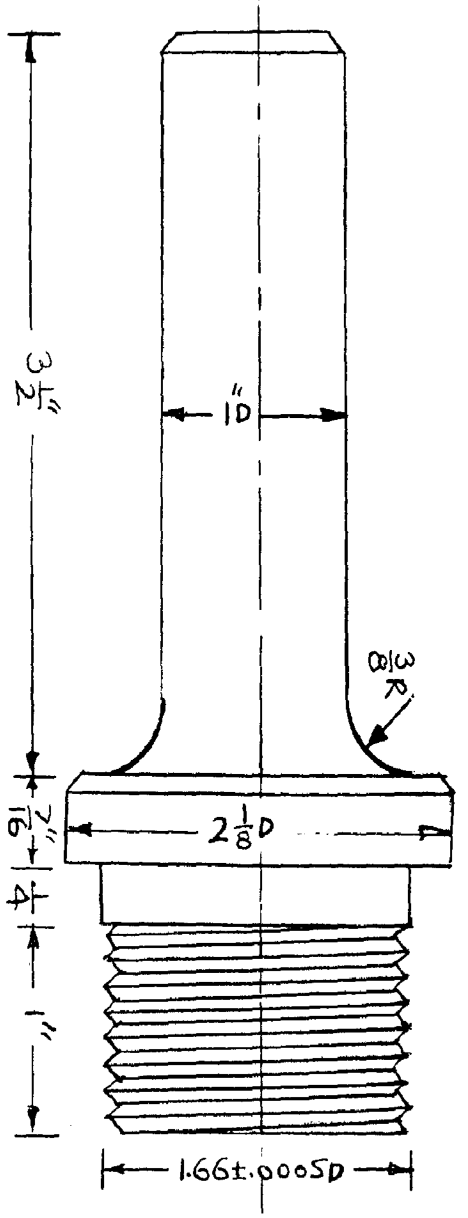
SECTION AT A.O.



TOP VIEW

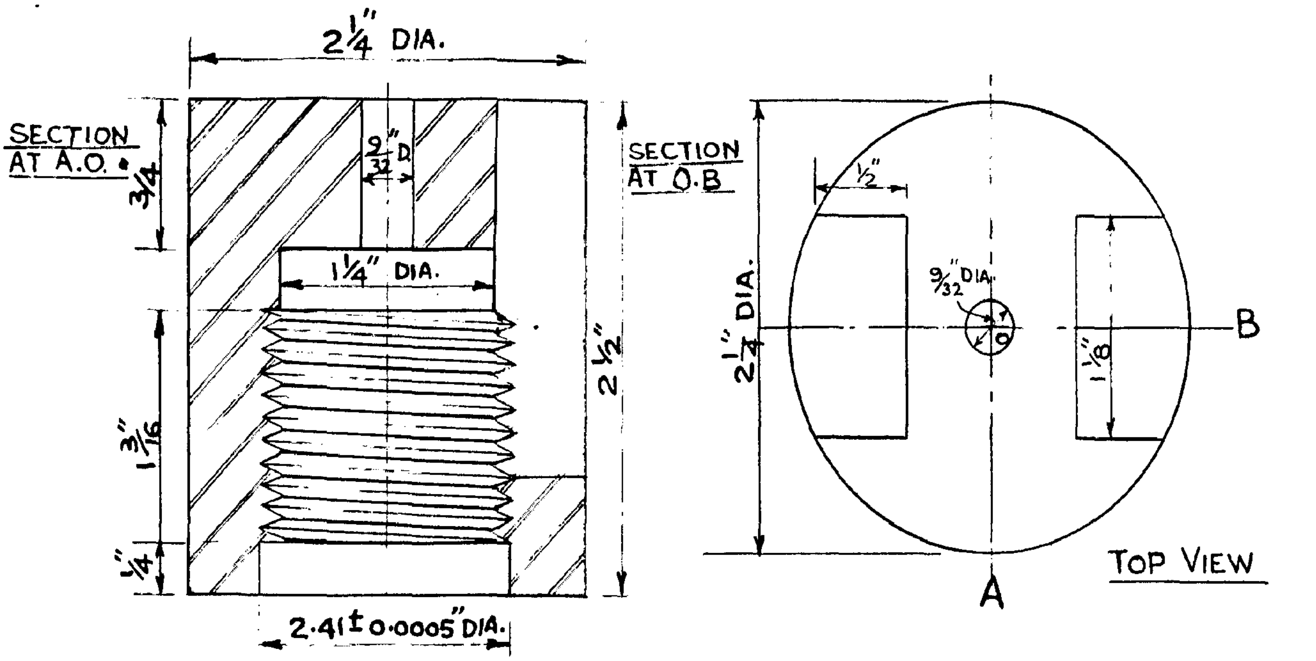
SCALE 1/2 FULL SIZE

UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE :- FULL SIZE	20-9-63.
C.2	B.S.A.

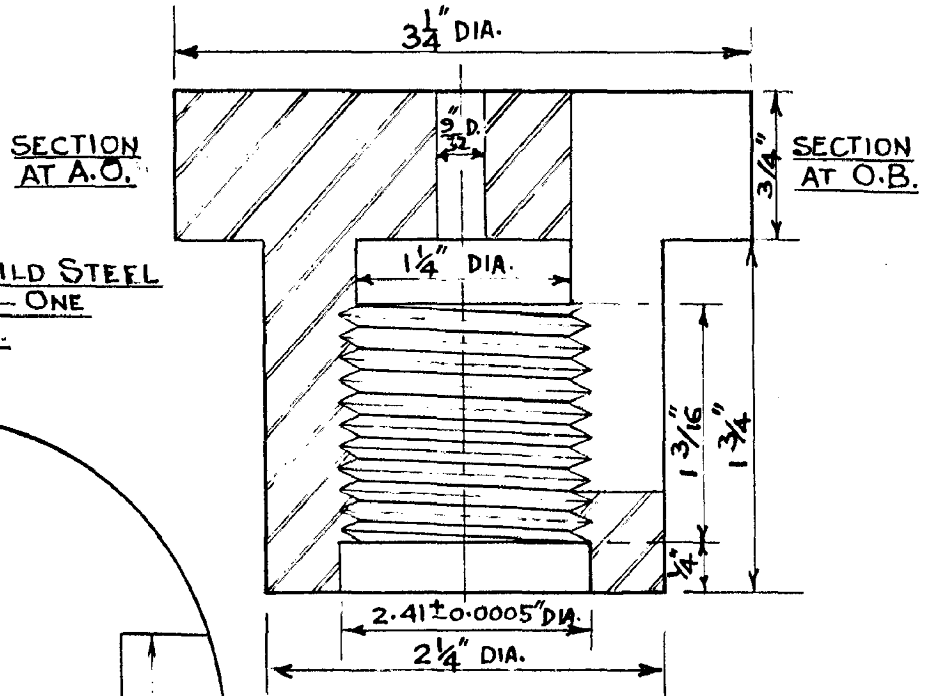


MATERIAL – MILD STELL
NO. REQD – TWO
8 T.P.I. B.S.W

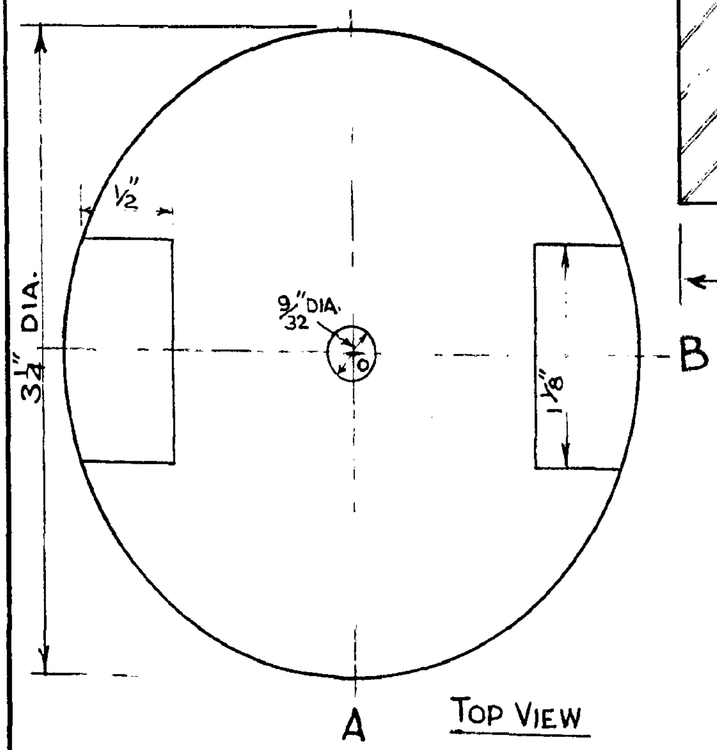
UNIVERSITY OF ROORKEE DEPT. OF. MECH. ENGG.	
CALIBRATION RIG COMPONENT	
FULL SCALE	20.9.63
C-3	B.S.A



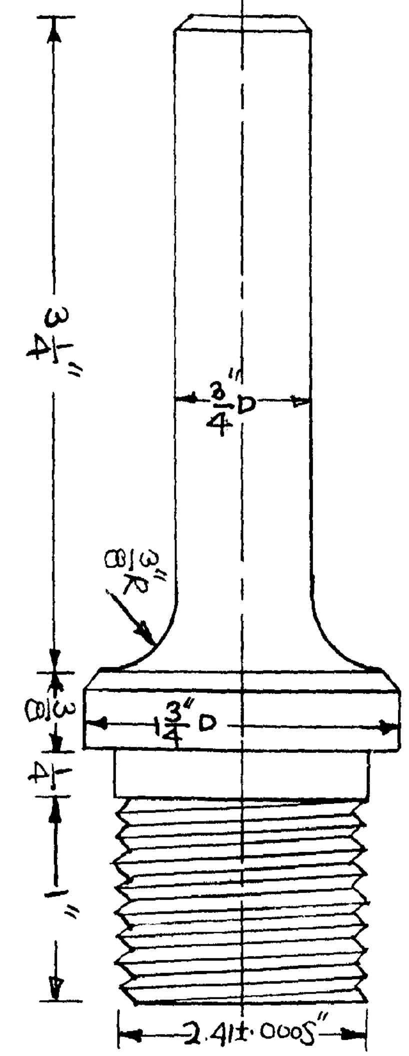
MATERIAL – MILD STEEL
No. REQUIRED – ONE
8 T.P.I. B.S.W.



MATERIAL – MILD STEEL
No. REQUIRED – ONE
8 T.P.I. B.S.W.

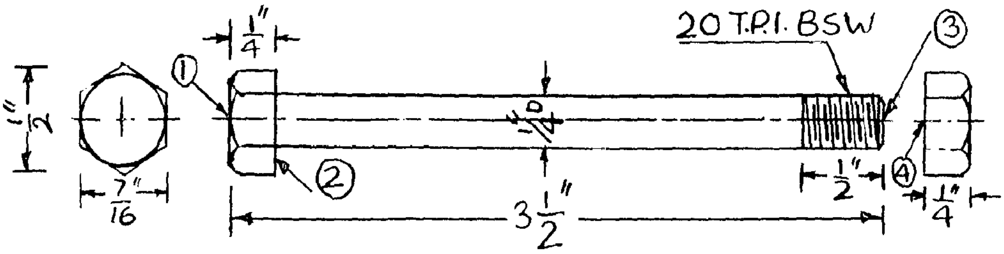


UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG COMPONENT	
SCALE:- FULL SIZE	20-9-63
D.2	B.S.A.



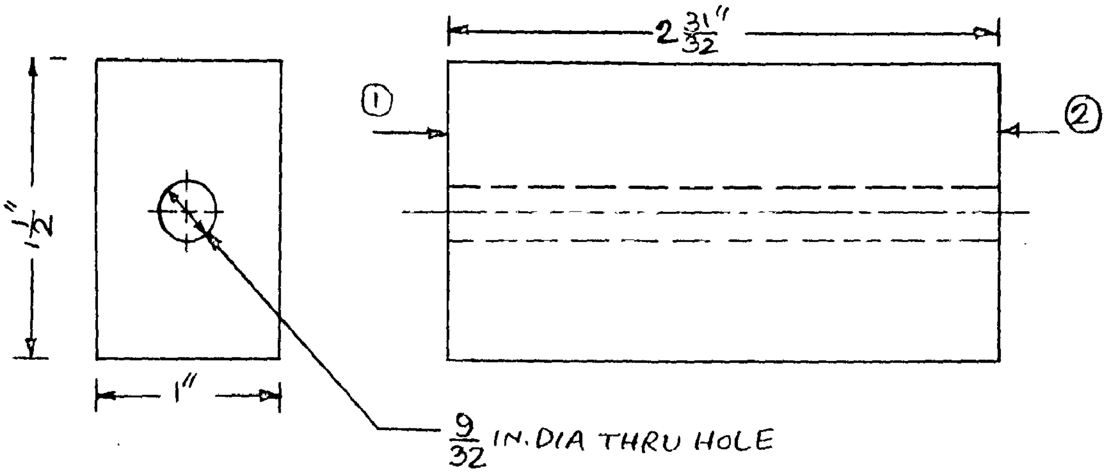
MATERIAL – MILD STEEL
NO. REQD – TWO
8.T.P.I. B.S.W

UNIVERSITY OF ROORKEE DEPT. OF MECH. ENGG.	
CALIBRATION RIG COMPONENT	
FULL SCALE	20.9.63
D-3	B.S.A



MATERIAL – MILD STEEL
NO. REQD – 60

NOTE – 1. Check the threading tool with B.S.W gaye.
2. Surfaces 1,2,3&4 are to be smooth and parallel.
3 Bolt head sides not to be machined.

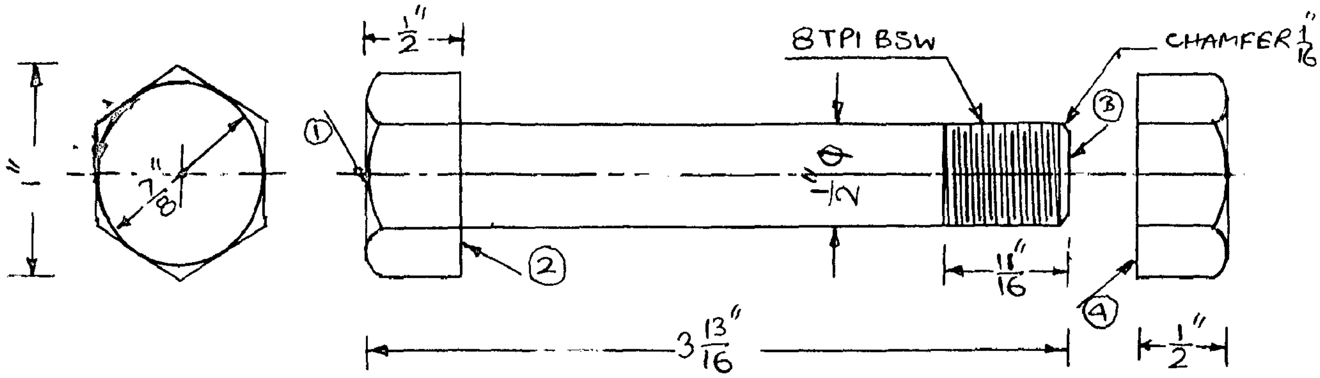


MATERIAL – CAST IRON
NO. REQD – 20

NOTE – 1. Surfaces 1&2 are to be smooth and parallel.
2. Drilled hole axis to be perpendicular to surfaces 1&2.

To ensure this cast a long rectangular bar of $1\frac{1}{2}$ IN. x $2\frac{31}{32}$ IN. X-section, mill 1&2, drill holes and cut 1 IN. long pieces.

UNIV. OF. ROORKEE DEPT. OF. MECH. ENGG.	
BOLT NUT & BLOCK	
FULLSCALE	18.8.63
H1 – H2	B.S.A



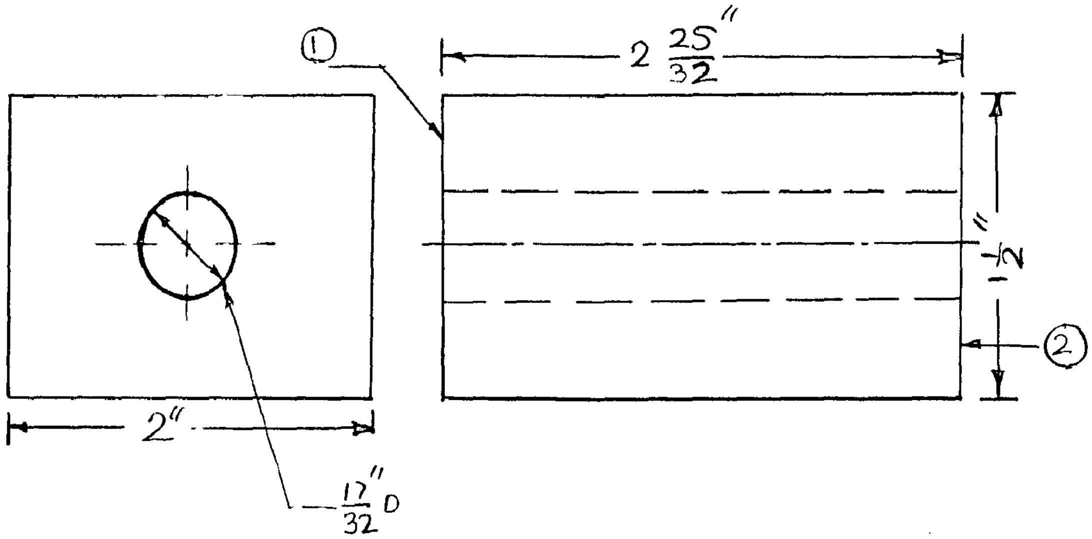
MATERIAL – MILD STEEL

NO. REQD – 56

NOTE – 1. Check the threading tool with B.S.W gage.

2. surfaces 1,2,3&4 to be smooth and parallel.

3. Bolt head sides not to be machined.



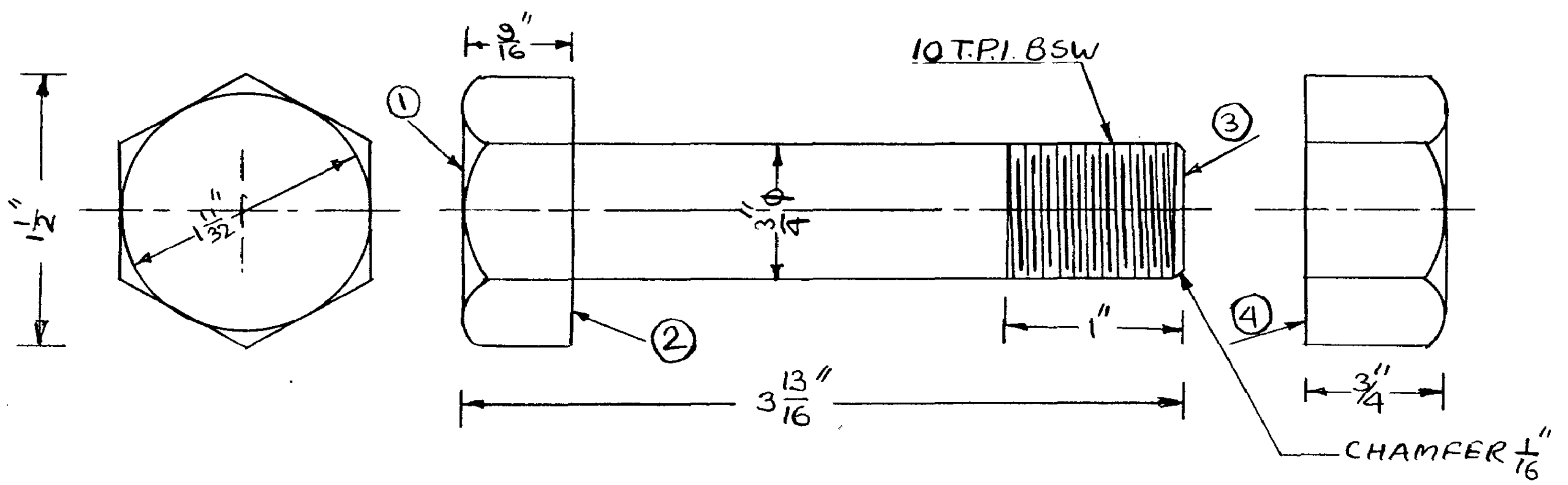
MATERIAL – CAST IRON

NO. REQD – 20

NOTE – surfaces 1&2 are to be machined fine and parallel. others cast.

TO ensure this cast a long piece of 2"x1½" x-section mill surfaces 1&2, drill holes and cut 2" long pieces

UNIV. OF. ROORKEE DEPT. OF. MECH. ENGG.	
BOLT, NUT & BLOCK ½ INCH. DIA.	
FULL SCALE	17.8.63
G-1, G-2	B.S.A



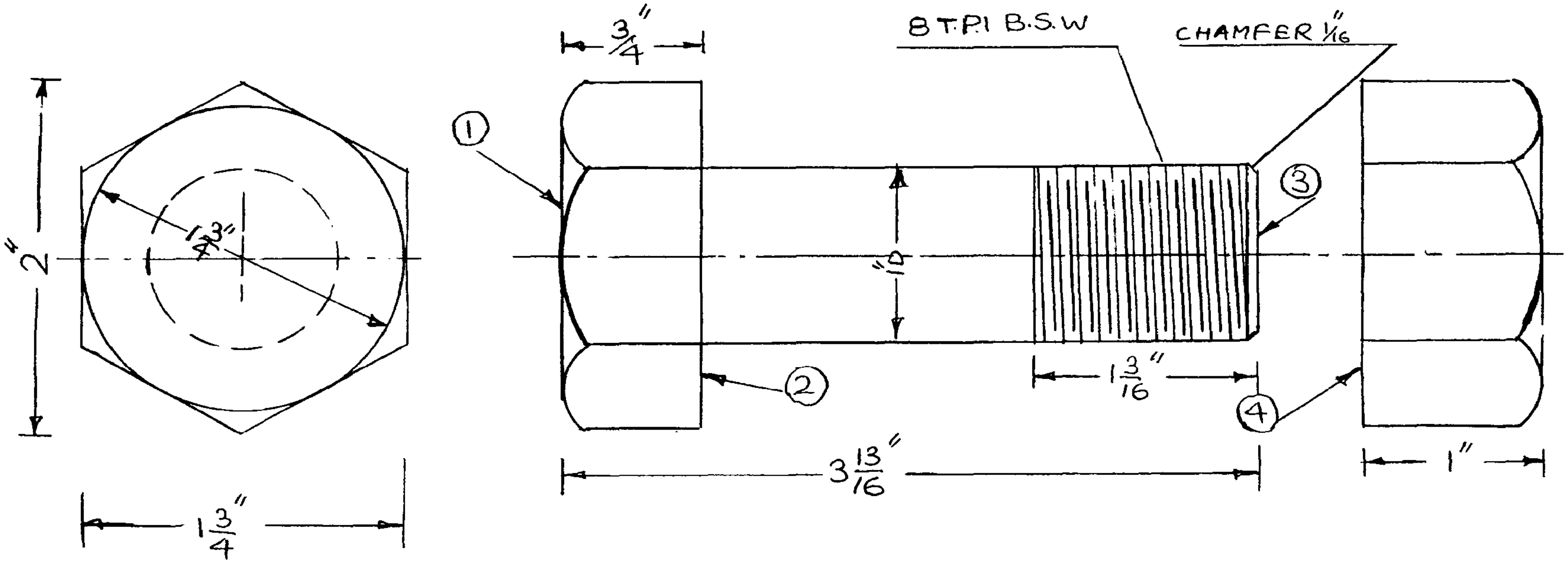
MATERIAL – MILD STEEL
NO. REQD – 20

NOTE – 1. The Threading tool is checked occassionly by standard BSW gage.

2. Surfaces 1, 2, 3 & 4 are to be fine finished and absolutely parallel.

3. Bolt head sides are to be accurately forged and not to be machined.

UNIV. OF. ROORKEE DEPT. OF. MECH. ENGG.	
BOLT & NUT 3/4 INCH. DIA	
FULL SCALE	16.8.63
F-1	B.S.A



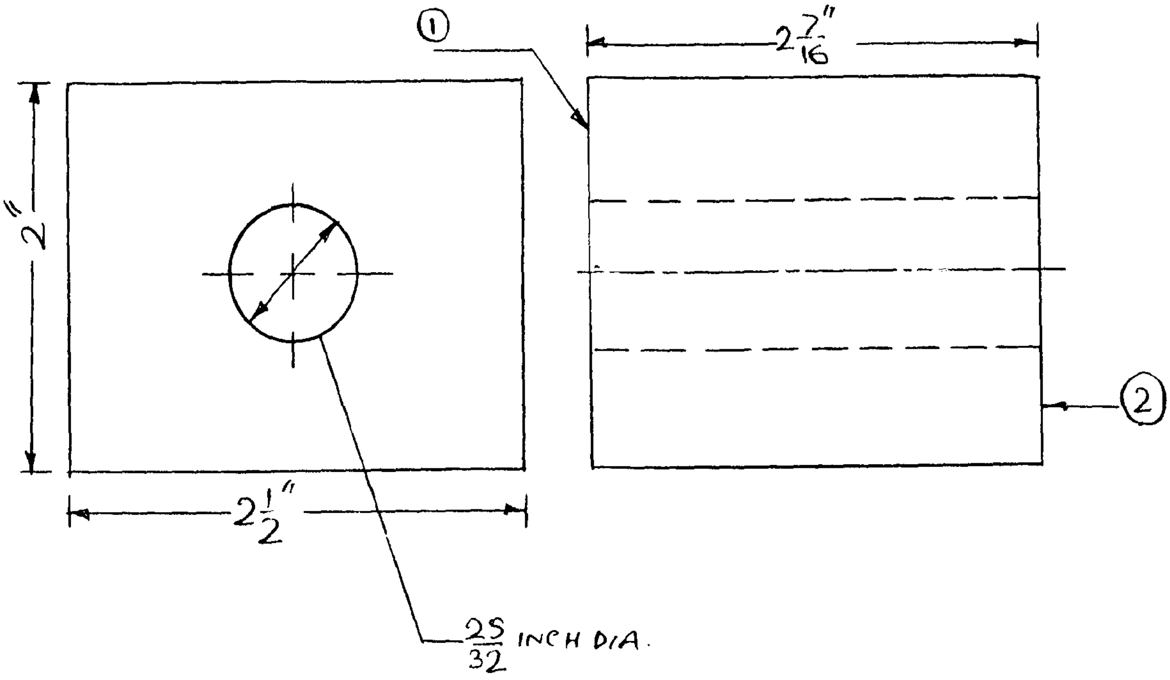
MATERIAL - MILD STEEL
NO. REQD - 20

NOTE - 1. The Theading tool is to be checked occassionally by Standard B.S.W. gage.

2. Surfaces 1, 2, 3 & 4 are to be fine finished and absolutely parallel.

3. Bolt head sides are to be accurately forged and not to be machined.

UNIV. OF ROORKEE DEPT. OF. MECH. ENGG	
BOLT & NUT 1. INCH DIA.	
FULLSCALE	16.8.63
E-1	B.S.A



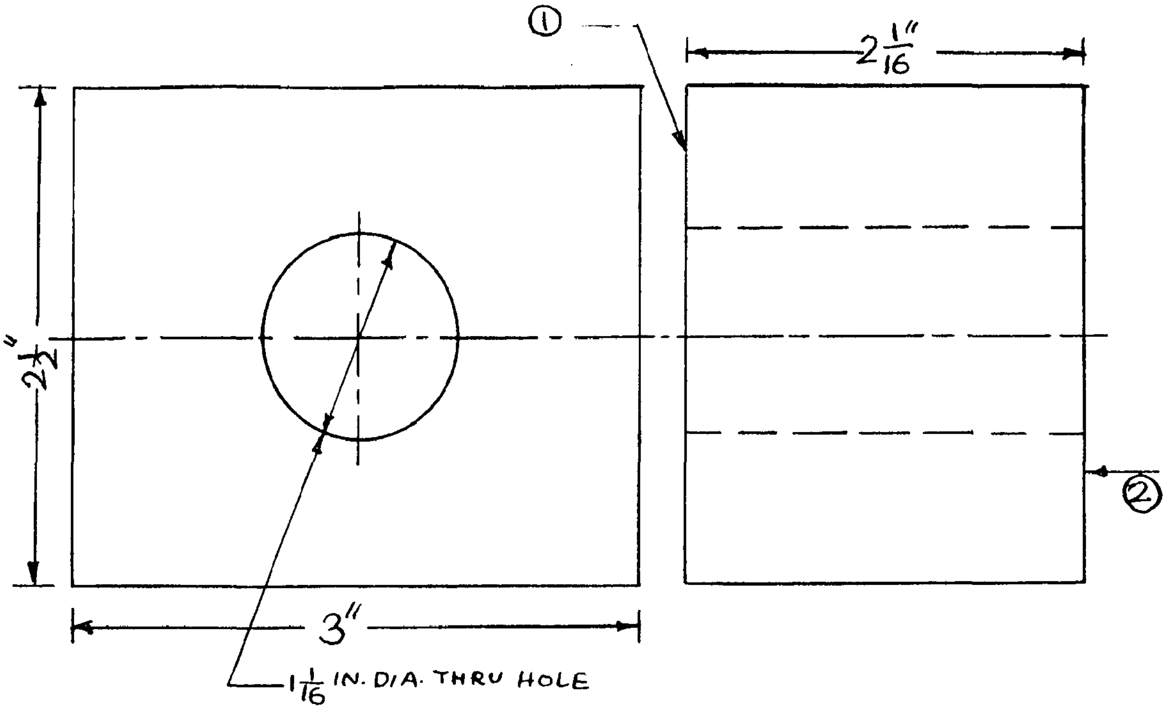
MATERIAL – CAST IRON
NO. REQD – 20

NOTE – surfaces 1&2 to be machined fine and absolutely parallel.

To ensure parallelity of 1&2, cast a long rectangular piece of $2'' \times 2\frac{7}{16}''$ x-section, mill 1&2, drill holes and cut $2\frac{1}{2}$ INCH long pieces.

Drilled hole axis should be perpendicular to surfaces 1&2.

UNIV. OF. ROORKEE DEPT. OF. MECH. ENGG.	
CAST IRON BLOCK	
FULLSCALE	17.8.63
F-2	B.S.A



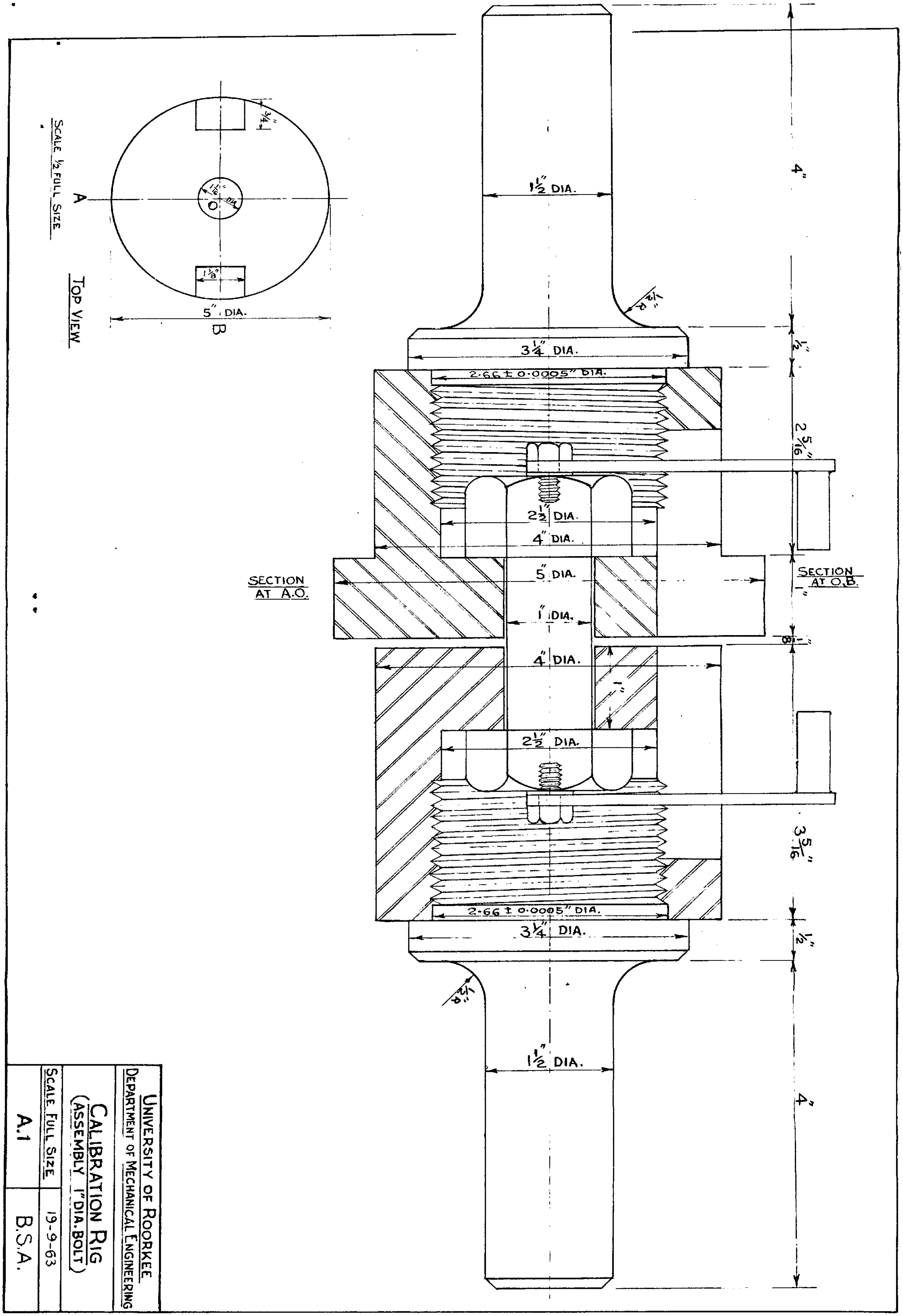
MATERIAL – CAST IRON
NO. REQD – 20

NOTE – 1. Surfaces 1 & 2 to be machined fine and absolutely parallel.

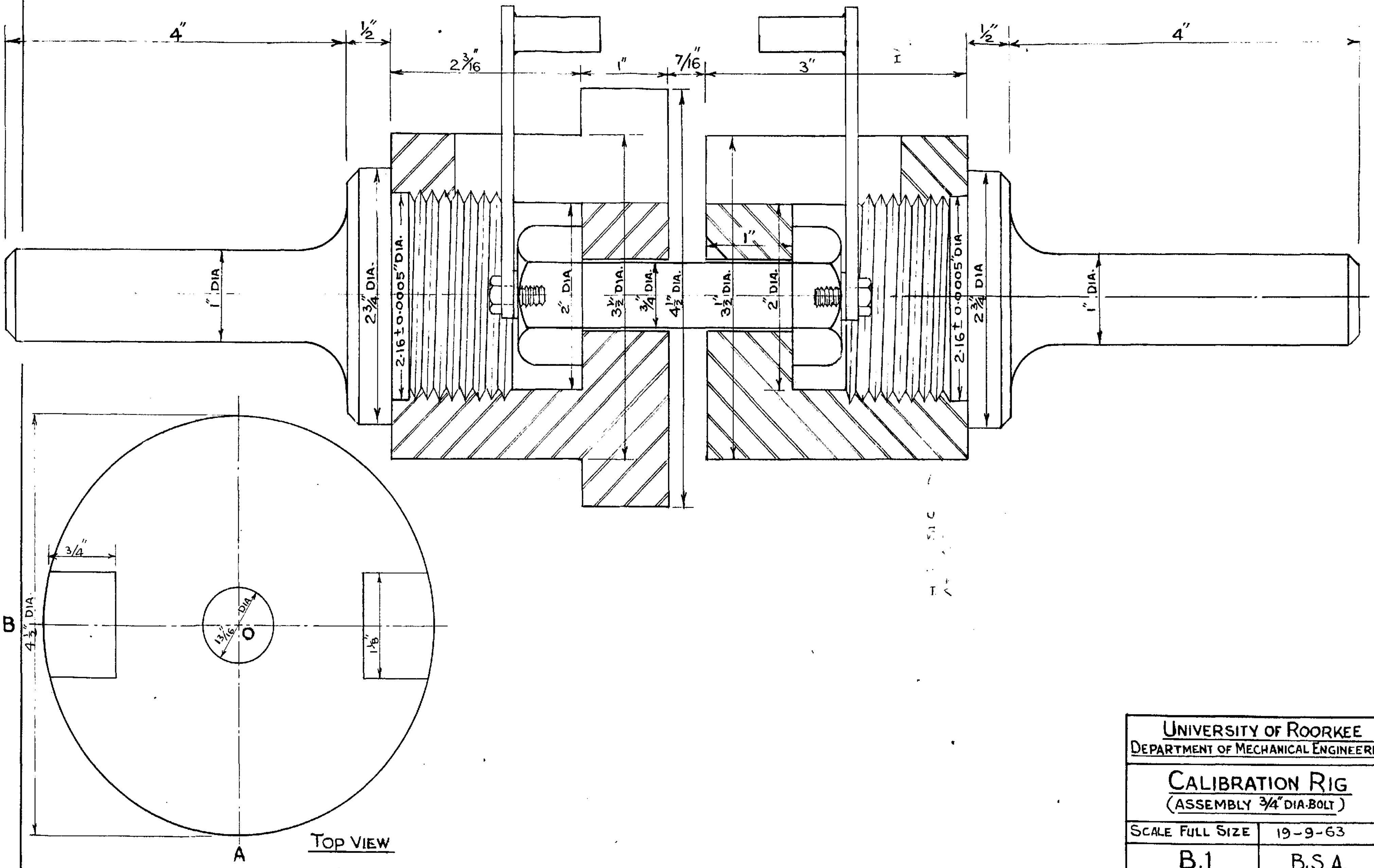
2. Drilled hole axis to be perpendicular to surfaces 1 & 2.

To ensure this, cast a long rectangular piece of $2\frac{1}{2}$ IN. × $2\frac{1}{16}$ IN. cross section, mill 1 & 2, drill holes and cut 3 IN. long pieces.

UNIV. OF. ROORKEE DEPT. OF. MECH. ENGG.	
CAST IRON BLOCK	
FULLSCALE	18.8.63
E-2	B.S.A



SCALE ½ FULL SIZE
TOP VIEW
A
B
O
5" DIA.
3/4"
1⅛"
1 1/16" DIA.
1½" DIA.
½" R
3¼" DIA.
2·66 ± 0·0005" DIA.
2½" DIA.
4" DIA.
5" DIA.
1" DIA.
SECTION AT A.O.
SECTION AT O.B.
4"
½"
2 5/16"
1"
1/8"
3 5/16"
UNIVERSITY OF ROORKEE
DEPARTMENT OF MECHANICAL ENGINEERING
CALIBRATION RIG
(ASSEMBLY 1" DIA. BOLT)
SCALE FULL SIZE
19-9-63
A.1
B.S.A.



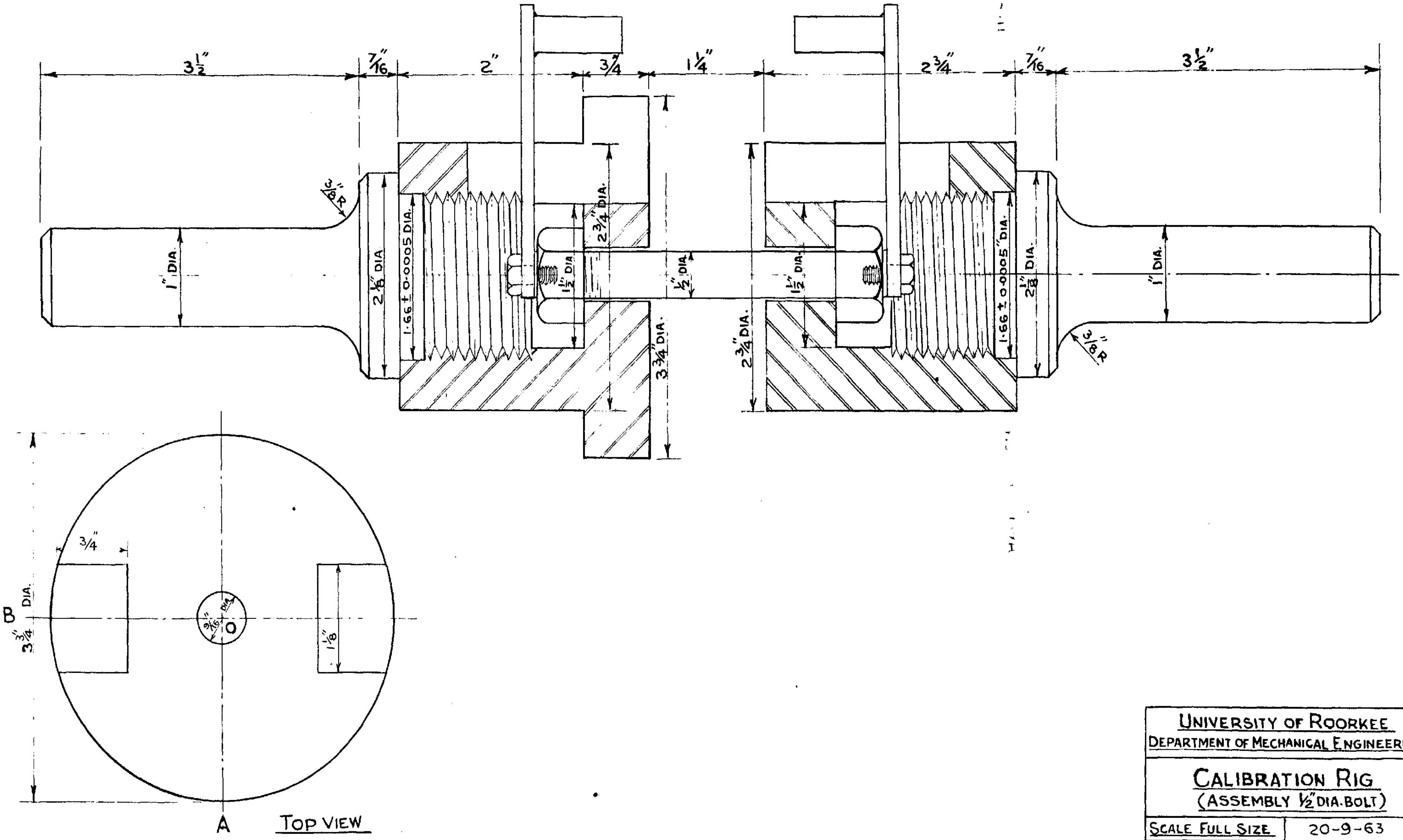
UNIVERSITY OF ROORKEE

DEPARTMENT OF MECHANICAL ENGINEERING

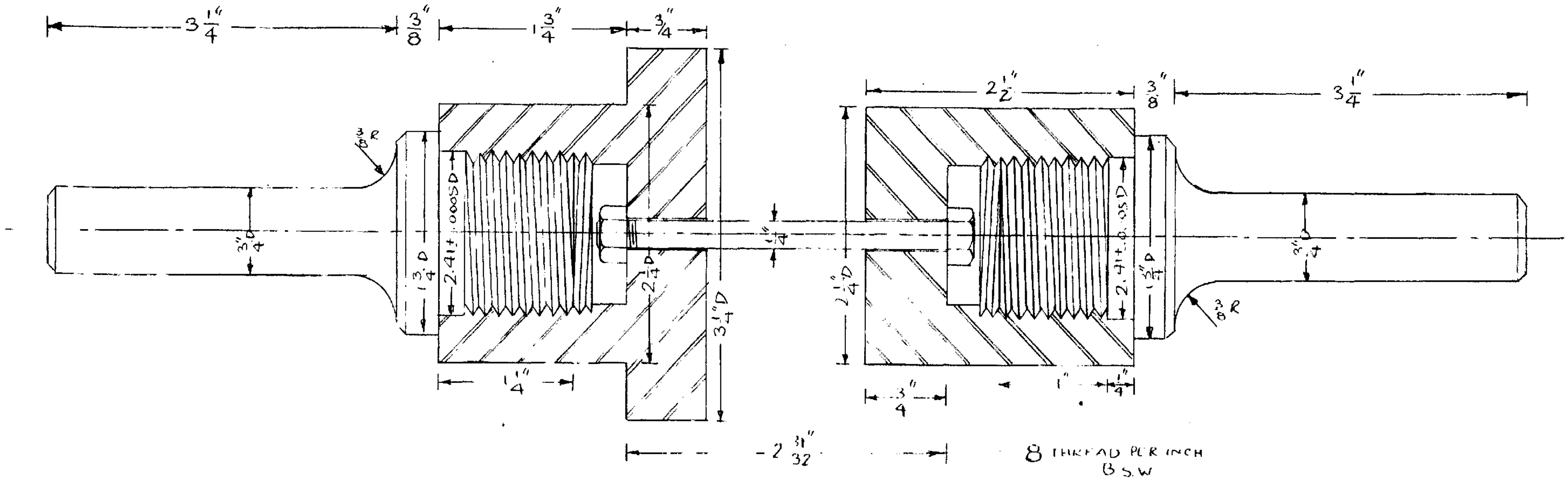
CALIBRATION RIG

(ASSEMBLY 3/4" DIA. BOLT)

SCALE FULL SIZE	19-9-63
B.1	B.S.A.



UNIVERSITY OF ROORKEE DEPARTMENT OF MECHANICAL ENGINEERING	
CALIBRATION RIG (ASSEMBLY ½" DIA. BOLT)	
SCALE FULL SIZE	20-9-63
C.1	B.S.A.



UNIVERSITY OF ROORKEE DEPT. OF. MECH. ENGG.	
CALIBRATION RIG ASSEMBLY - 1/4" DIA BOLT	
FULLSCALE	20.9.63
D-1	B.S.A