IN THE NAME OF
ALLAH
THE MOST COMPASSIONATE
AND MERCIFUL

O'ALLAH
I ASK YOU BY YOUR GRACE
WHICH OVERSPREADS EVERY THING
TO FORGIVE ME

ESTABLISHMENT OF DESIGN CRITERIA FOR TIGHTENING BOLTED JOINTS

BY

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Dedicated

To

Soul of my father,

who died twenty years ago

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Finally, many thanks to my mother, brothers and sisters for their blessings and prayers.

DECLARATION

I declare that all unreferenced work described in this thesis, including the experimental and the theoretical investigations and conclusions reached were investigated by myself. And no part of the experimental work in this thesis has been submitted in support of an application for another degree, or qualification of this or any other university or other institute of learning.

SIGNED ______________

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DATE: 20th of December, 1990

ABSTRACT

ESTABLISHMENT OF DESIGN CRITERIA
FOR
TIGHTENING BOLTED JOINTS

BY BASSAM T. HARIRI

Joints are extremely important parts of any structure. Bolted joint joints also make disassembly and reassembly possible. The information on tightening bolted joints is important to the design engineer, in order to ensure the reliability of the joint and a minimum clamp load which is necessary to provide adequate in-srevice durability. Therefore designer are drawn more and more towards tightening processes which minimise clamp load variation in order to obtained the benefits of more predictable service. The design of bolted joints and techniques used to tighten them has received increased attention by the Automotive, Aerospace and Petro-Chemical industries.

In this research theoretical and experimental investigations have been undertaken to determine the response of fasteners to combined torque and axial load. The principal aims of these experimental investigations are to establish the following: (i) The mode in which the initially applied torque in the bolt is reduced due to subsequent application of axial stress. (ii) The mode of reduction in initially applied axial load due to the subsequent application of torque. (iii) The mechanism in which the torsional and tensile strain energy is dissipated in a bolt. (iv) Any variation in the yield stress of the bolt as the axial load is applied after initial tightening torque.

The present work has been carried out by using bolts having thread rolled after heat treatment. The fasteners selected for testing were M12X1.75 MM I.S.O., Grade 10.9. An electronic hand torque wrench was used to tighten the fasteners. No lubrication was required during these tests. The effects of axial load on the tightened fastener were examined by static concentric loading. The uniaxial tensile load was applied by a hydraulic cylinder.

The results obtained in the experimental evaluations show that the the material of the bolt is elastic-work hardening. It also shows that the reduction of the initial torque due to the application of axial load and the reduction of the initial axial load due to the application of torque possibly follow the von Mises yield criteria. This will dictate that the material is yielding only at the outside diameter during combined loading. The experimental results also show that no significant reduction in the axial tensile load takes place for all magnitudes of the applied torque upto the yield torque.

The reduction in the initially applied torque or axial load due to subsequently applied axial load or torque respectively may also take place in such a manner so that the plastic zone at the outer surface progresses towards the inner core. It is most likely that such off loading takes place in mixed modes. Experimental tests and analysis confirm that the yield load of a bolt in tension is not affected appreciably when the bolt is subjected to any torque.

It is recommended that in a bolted joint the bolt should be subjected to as low a clamp load as possible with relatively high level of initial torque. This would allow higher external load to be applied to a joint with less chance of reducing the tightening torque substantially.

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NOMENCLATURE

A	The tensile stress area.
A_0	The current area.
A_1	The area before the change in the length due to the force.
A_2	The area of the minor thread diameter.
A_0, A_1 A_2, A_3	The cross-sectional area for each section in the bolt.
A_s	The area of a substitute cylinder to be determined
A_x	The area at the length `L_x'.
C_1, C_2, N	Constants.
d	Nominal bolt diameter.
d_1	Pitch thread diameter.
d_2	Minor thread diameter.
d_m	Mean thread diameter of bolt.
$d\lambda$	The proportionality factor which relates the incremental plane strain `$d\epsilon^p_{ij}$'.
e	Engineering strain.
E	Modulus of elasticity.
f	Total deformation of bolted joint.
f_{ba}	Deformation of the bolt due to `F_a'.
f_{bi}	Elastic bolt elongation due to `F_i'.
f_{ja}	Deformation of the clamped parts due to `F_a'.
f_{ji}	Elastic joint compression due to `F_i'.
F	Axial tensile loading.
F_{ba}	Additional bolt load from external tensile load F_a
F_i	Pre-load on bolt and joint `the initial clamp load

	or the initiall pre-load'.
F_i`	Remaining pre-load after loss due to permanent set
$F_{imin.}$	Minimum clamp load.
$F_{imax.}$	Maximum clamp load.
F_{ja}	Reduced joint load from external tensile load.
F_{kr}	Residual clamping load in the interface or the remaining clamp load.
F_v	The total bolt load.
G	Modulus of elasticity in shear.
I_p	Polar of moment of inertia, at the outside diameter of the bolt (8 mm).
I_r	Polar of moment of inertia of the cross-sectional area at radius r.
K	The spring constant for bolted joint. Yield stress in pure torsion.
K_1,K_2,K_2	The spring constants for the individual components for clamped parts and bolt.
K_b	The spring constant for bolt.
K_j	The spring constant for clamped parts.
L	The lead of thread screw. Total length of the bolt. The current length of the bolt.
L_0	Original gauge length.
L_1	Current gauge length.
L_0,L_1 L_2,L_3	Length of each section in the bolt.
L_j	Original joint length.
L_k	Fraction of L_j, which lies between the bolt head and nut,introduction in levels in distance ($n.L_j$).

L_x	Length of bolt at section No.X.
n	The ratio between length of clamped parts and total joint length.
P	Pitch of thread, load.
P_0, P_1 P_2, P_3	The forces through each section of the bolt.
R	The radius of the bolt which the material is yielding.
R_c	Collar friction radius, or under-head bearing face
R_0	The outside diameter of the body of the bolt(8mm).
S	Permanent set in bolted joint due to elastic deformation.
S'	Permanent set in the bolt and clamped parts due to plastic deformation.
S_b	Permanent set in the bolt.
S_j	Permanent set in the clamped parts.
T	Wrench torque which is applied on the bolt.
T_x	Wrench torque at section No.x of bolt.
T_1	Elastic torque.
T_2	Plastic torque.
T`	The torque remaining in the fastener.
Y	The combination of tensile and torsional stresses in the fastener. Yield stress in tension.
Y^*	Combined stresses remaining in the fastener after wrenching torque has been removed.
α	Tightening factor.
α'	Helix angle thread at mean radius.
δ	Elastic resilience of bolted joint. Total

	extension in the bolt.
δ_0, δ_1 δ_2, δ_3	The extensions in each section in the bolt.
δ_b	Elastic resilience of the fastener.
δ_j	Elastic resilience of the clamped parts.
ϵ	Natural strain.
ϵ_{lat}	Lateral strain.
ϵ_l	Longitudinal strain.
ϵ_0, ϵ_1 ϵ_2, ϵ_3	The natural strain for each section in the bolt.
γ	The shear strain remaining in the fastener.
γ_0	The initially shear strain applied to the fastener.
D_{Fi}	Pre-load loss due to permanent set from elastic deformation in a bolted joint.
D_{Fi}'	Pre-load loss due to permanent set from plastic deformation in the bolted joint.
D_L	The extension of the member under the force F.
D_{Ltot}	Total pre-load loss due to permanent set.
D_T	The reduction of torque due to the applied axial load.
$D_{\sigma x}$	The reduction of axial stress due to the applied torque.
θ	The angle is measuring in the axial section. The angle of twist remaining in the fastener.
θ_0	The initially angle of twist applied to the fatener.
θ_n	The angle between tangent to tooth profile and

	line.
μ	Coefficient of friction.
ν	The poisson`s ratio.
σ	Axial tensile stress or true stress.
σ_{com}	Combined stresses from tension and torsion.
σ_{eff}	Combined stresses from tension and torsion.
$\sigma_1, \sigma_2, \sigma_3$	The principal stresses, which effect on the element of the material.
σ_0	Nominal stresses.
σ_0, σ_1 σ_2, σ_3	The axial stress for each section in the bolt.
$\sigma_{0.2}$	Yield strength at permanent set of 0.2 percent strain.
σ'_{ij}	Deviatoric stress component.
σ^*	Axial stress remaining in the fastener after wrenching torque has been removed.
σ_{com}^*	Combined stresses remaining in the fastener after wrenching torque has been removed.
σ_{ba}	Axial stress induced by additional axial stress.
$\sigma_x, \sigma_y, \sigma_z$	The axial stresses in the directions X,Y and Z respectively.
σ_h	Hydrostatic stress.
σ_x	The applied axial load to the fastener.
σ_x`	The residual axial stress.
σ_y	The yield stress in tension.
τ_r	Shear stress at radius r.
τ_{out}, τ	Shear stress at the outside diameter of the bolt (8mm).

τ^*	Shear stress remaining in the fastener after wrenching torque has been removed.
ϕ	Load factor in bolted joint. The total angle of twist in the bolt.
ϕ_x	The angle of twist at section `X' in the bolt.
ϕ_e	Load factor for eccentric application.
ϕ_{ek}	Load factor for eccentric load application with load introduction under bolt head and nut.
ϕ_{en}	Load factor `$\phi_e$',But with load introduction inside the clamped parts in levels of distance `$n.L_j$'.
ϕ_k	Load factor for concentric load application with load introduction under bolt head and nut.
ϕ_n	Load factor for concentric load application with load introduction inside the clamped parts in levels of distance `$n.L_j$'.

SUBSCRIPTS

a	External.
b	Bolt.
c	Collar.
G	Thread.
i	Initial.
j	Clamped parts.
off.	Total off.
tot.	Total.
W	Under head.
*	Eccentric bolting or remaining `residual'.
**	Eccentric bolting and eccentric loading.

CHAPTER ONE

INTRODUCTION

CHAPTER ONE

INTRODUCTION

1.1 Introduction of bolted joint

There are several methods of joining components permanently or semi-permanently for example, by welding, riveting, using adhesives or screw fasteners. Fastening by screw is one of the most common methods used by industry [1].

In order to fasten one part to another, fastening bolts are often used in various industries. However, the design of bolted joints depends mostly on the experience of individual companies [2]. Experience and experiment show that, pre-loads produced in the field by current bolting practices tend to be lower than intended by the designer, many factors leading to this result [3].

The joints are extremely important part of any structure, they make complex structures and machines possible, bolted joints at least, also make disassembly and reassembly possible [4].

The mode and means of joining metals to metals is of paramount importance in industrial manufacture. The three things that have been responsible for a major revolution in fastener technology within the last few decades are speed, cost, and appearance of so many new materials. Speed and cost are interrelated and any technique that speeds up production saves man-hours or labour cost [5].

An optimum bolt design is one in which the desired mechanical properties are obtained with the minimum fastener weight and envelope. Light weight bolts extensively used in the aerospace industry, are used to increase product performance and conserve raw materials. In bolt design, weight reduction is achieved primarily by reducing the material content of the bolt head. The critical factors in bolt head design are those which affect the tensile fatigue or wrenching strength of the bolt [3].

The bolted joints are a fundamental element in most mechanical equipment. It is frequently the most highly stressed element in a structure, and the bolt itself is likely to be the component of smallest cross section [6].

In many bolted assemblies, the in-service durability is determined by the quality of the individual bolted joint. The assembly of bolted joints are basic mechanical engineering operation. Power tools for tightening threaded fasteners are well developed and are mainly air powered. Today the assembly operation is frequently the weakest line in producing a reliable joint. Some possible problem conditions which should be rejected. The following problems can be represented [7]:

- **bolt too hard:** The high yield point is caused by a heat treatment. This could lead to delayed fracture.
- **Bolt too soft:** The low yield point is caused by a heat treatment. This could lead to breakage in assembly or in service.
- **Fault in the male or female thread:** Causing high torque. May lead to inadequate pre-load.
- **Low friction:** Caused by wrong surface condition. May lead to over-tightening and breakage.

The combinations of these conditions can occur such as **soft bolt with low friction, soft bolt with high friction.**

In view of this, a research programme was undertaken at SPS Laboratories, Naas, Ireland, to develop better methods of tightening control and has resulted in a new closed-loop control system for vane-type air motors [8]. The demands for assurance of quality and reliability in engineering structure or components have steadily increased over the past two decades.

In mechanically fastened assemblies, reliability has frequently been assured by the simple expedient of over design which results in additional cost. Industries are coming under increasing pressure to reduce cost without compromising the reliability [9]. The reliability of the joint depends on the following three factors:

1. On the reliability of the fastener itself, that is, its geometry, correct material selection,and heat treatment.
2. On the reliability of the joint load analysis, that is, the accuracy of the calculated initial clamped load.
3. On the reliability with which the calculated joint clamping load is achieved by the assembly process.

Errors in any one of these factors can lead to the failure of the joint [10].

Bolted assemblies form a major part of most engineering structures. It has long been realized that the design and assembly of bolted joints must ensure that, the clamp load achieved on tightening the fastener is sufficient to withstand the effect of the static and dynamic loads, which may arise during service [11]. Increased safety requirement in the aerospace, automotive and other mechanical equipment manufacturing industries in the last decade have resulted in significant efforts to increase the reliability of bolted joints [10].

In establishing the design capability of a bolted joint, several questions need to be answered, such as:

(i) How tight should the bolt be ?
(ii) What level of external load will cause the joint to open ?
(iii) What load is felt by the bolt ?

(iv) What are the properties of the bolted joint under dynamic loads or fatigue conditions ?

With recent improvement in the control of fastener tightening, these questions merit detailed attention in order to obtain the maximum performance at minimum cost [12].

When the joints were fastened by the optimum clamping force after pre-setting, the fatigue limit was improved by about 10 percent as compared with that of the optimum clamping force. The reasons for the improvement of the fatigue limit of the bolts with pre-setting were not only the influences of the residual stress at the thread bottoms and the work-hardening of the bolts but also the effects of the optimum clamping force after pre-setting [13,14].

The problem of bolted joint design has been mainly with respect to its static stiffness properties. Thus, the designer should first define the required stiffness value for the joint. The calculated joint stiffness should be checked with the required joint stiffness. If the calculated value is not sufficient, the parameters influencing the joint stiffness should be changed. The parameters may be: The member height, the major diameter of the bolt and material. The static stiffness clearly improved with the increase in the member height and the major bolt diameter, and it failure with the increasing of the ratio of the member height to the major bolt diameter. Thus, to improve the calculated value of the static joint stiffness, may be either the major diameter increasing or the member height decreasing [15].

Finally, the only difference between a screw and a bolt is that the bolt needs a nut in order to be used as a fastener whereas a screw fits into a threaded hole [4].

1.2 Standardized screw threads

1.2.1 INTRODUCTION

The first serious attempt to rationalise screw thread production was undertaken by **Sir Joseph Whitworth** about 1841,the result of this study being the original`**Whitworth**' thread as the standard. Over the following twenty years it replaced the miscellaneous collection of threads in general use and with a fine thread series of the same form.

All these threads were based on a **55°** angle between the flanks with a specified rounding at crest and root, and differed only in the number of threads per inch. Which became the **BRITISH Standard Fine (BSF)**.

In **1864**, **Sellers** in America, quite independently proposed another national thread standard based on a **60°** angle between the flanks and with cut-off rather than rounded crests and roots (mainly because this was easier to produce by machining). The original **Sellers** thread was adopted in the America **(SAF)** and **(ASME)** standards.

In continental Europe, rotational threads were also developed with a variety of flank angles, but most again using **cut-off V** shapes rather than rounded crests and roots, these included the (German) **Loeuenhertz** thread with an angle of **53° 3'**, and the (Swiss) **Thury** thread with an angle of **47.5°**. Again in an attempt at standardisation the

SI (System International) thread was originated in 1898 based on a 60° angle between flanks with flat crests and rounded roots.

The basic disadvantage of this SI metric thread was that the small root radius led to inferior fatigue characteristics. Originally, international standardisation of screw thread merely led to the feed-in of further standard thread forms [4,5].

The various types of threads used on screws have been standardized, and it is important that the designer be aware of the various types available and what their important characteristics are, Power screw threads are of an entirely different type.

1.2.2 THE ISO METRIC SCREW THREADS

Standard proportions of threaded bolts are normally based on a shank diameter equal to the full or normal diameter of the thread. **ISO Metric screw threads** are designated by "M" followed by the two numbers separated by "X". The first number is the base thread diameter in milli-meters and the second number is the thread pitch in milli-meters.

In practice, the ISO Metric coarse thread provides a replacement for general engineering. The metric thread has a large root radius, and the ISO metric threads were based on a 60° angle between flanks with flat crests and rounded roots, as shown in **Figure (1.1)**.

In the tolerance system there are three classes of fits;close, medium and free. Tolerance is fully designated by letter/number symbols, a number and a capital letter specifying the tolerance grade and diameter of the nut and in lower case letter the tolerance grade and diameter of bolt. Most production fasteners are based on medium class thread (5) as follows:

Class	Bolts	Nuts
Close	4h	5H
Medium	6g	6H
Free	8g	7H

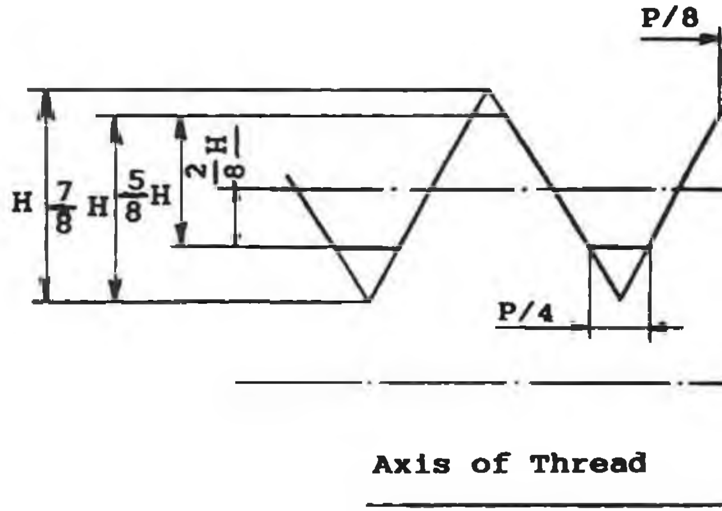
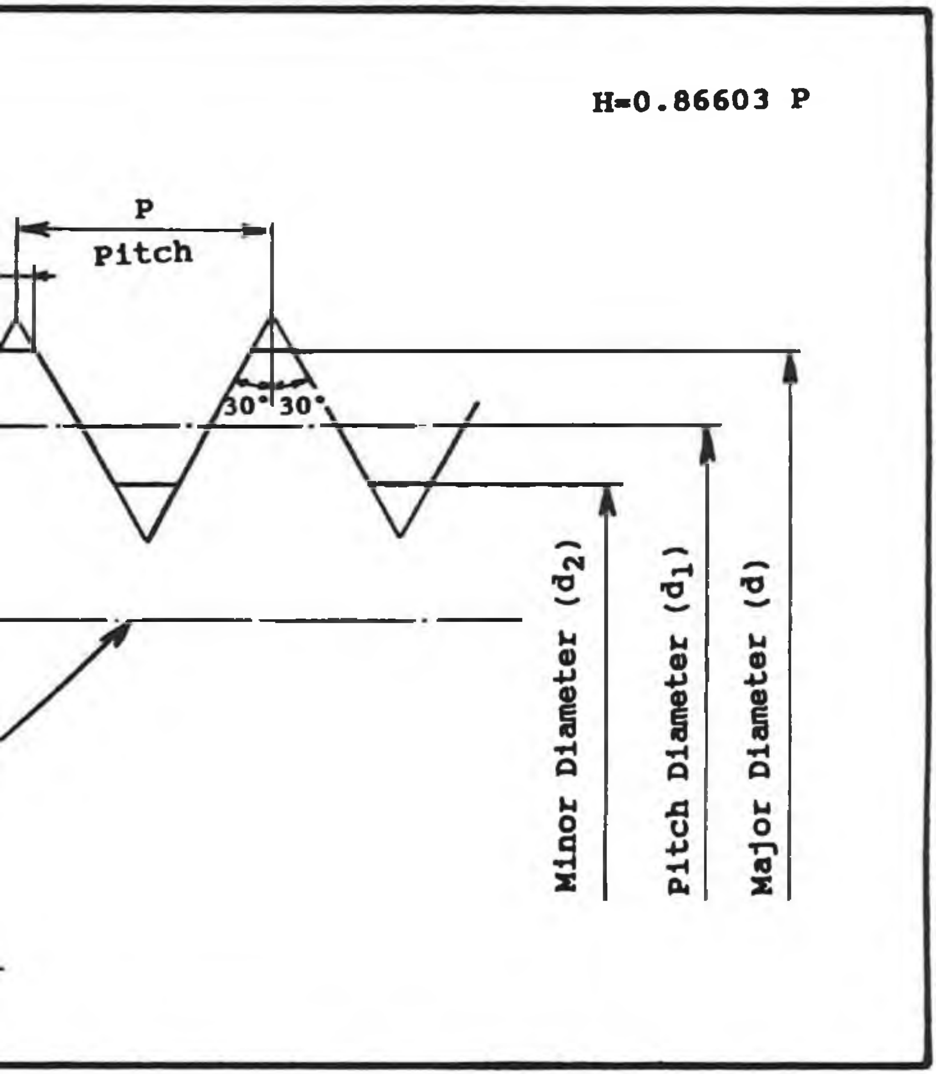


Fig.(1.1) ISO Metric Screw Thread.


H=0.86603 P
P
Pitch
30°
30°
Minor Diameter (d_2)
Pitch Diameter (d_1)
Major Diameter (d)

1.3 Mechanics of engineering material

1.3.1 INTRODUCTION

Deformation may occur in a material for a number of reasons, such as external applied load, change in temperature, tightening of bolts, irradiation effects, etc. [16]. Bending, twisting, compression, torsion and shear or combinations of these are common modes of deformation .

1.3.2 YIELD CRITERIA

Materials which exhibit `yielding' followed by some deformation prior to fracture as measured under simple tensile or compressive stress are termed ductile. This is a very important property as it provides a design reserve for materials if they should exceed the elastic range during service [16].

The yield point, indicating the onset of plasticity as readily seen on uniaxial stress-strain curve may not be obvious in complex stress states. A number of theoretical criteria for yielding [17] are expressed in terms of principal stresses. One of these theories has been used in this study, that is, shear or distortion strain energy theory, which is referred to as the MISES-HENCKY criterion.

1.3.3 SHEAR STRAIN ENERGY THEORY

This criterion states that yielding commence when the shear strain energy reaches the equivalent magnitude for Uniaxial yielding.

Consider an element of ductile material subjected to the principal stresses σ_1, σ_2 and σ_3. For Y to be the stress at yielding in simple tension the following equation is obtained [16,18]:

$$(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2=2Y^2. \qquad (1.1)$$

When the stress state consists of plane and shear stresses the following equation is obtained [19]:

$$(\sigma_x-\sigma_y)^2+(\sigma_y-\sigma_z)^2+(\sigma_z-\sigma_x)^2+6(\tau_{xy}^2+\tau_{yz}^2+\tau_{zx}^2)=6k^2; \qquad (1.2)$$

That is,

$$K=Y/\sqrt{3}. \qquad (1.3)$$

In the case of normal tightening of the fastener, the stresses in one direction need only be considered , so that the following equation is applicable :

$$Y=\sqrt{\sigma^2-3\tau^2}. \qquad (1.4)$$

This equation is called combined stress value according to MISES-HENCKY. It has been established by SIEBEL [20],

MORRISON [21], and **IVEY** [22], that for ductile materials, exhibiting yielding and subsequent plastic deformation, the **MISES-HENCKY** criterion correlated best with material behaviour.

1.3.4 PROOF STRESS

The typical load extension curve is shown in **Figure (1.2)** for certain material. For some material there may not be any upper and lower yield point and also in some cases no yield point exists.

In such cases a proof stress may be used to indicate the onset of plastic strain. However, as a comparison of relative properties with another similar material **UTS** is used. Proof stress involves a measure of the permanent deformation produced by a loading cycle [23], a **0.2%** proof stress value is obtained graphically, this value is located on the knee of the stress-strain curve by the intersection of a line drawn parallel to the elastic stress line at **0.2%** of strain as shown in **Figure (1.3), [24]**.

1.3.5 STRESS-STRAIN RELATIONS

A material is said to be elastic if it returns to its original, unloaded dimensions when load is removed. Within the elastic limits of material for which **Hooke`s** law is applicable.

Linear relations between the components of stress and the components of strain are known generally as Hooke`S law. Most applications of the mathematical theory of plasticity are based on the REUSS stress-strain relations [19].

A more detailed discussion about the theory of elasticity and the theory of plasticity of mechanics of engineering materials is given in Appendix A1.

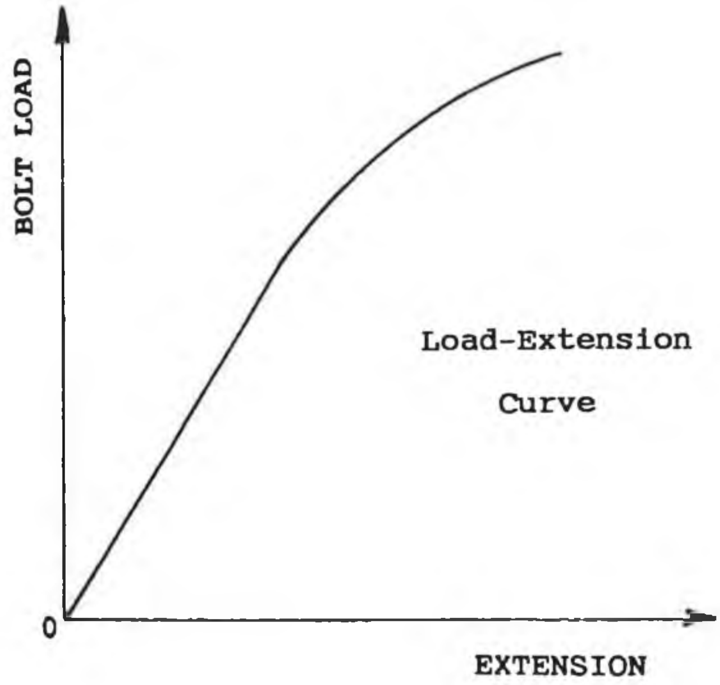


Fig.(1.2) Relationship between load and extension of the material of the bolt (uniaxial curve).

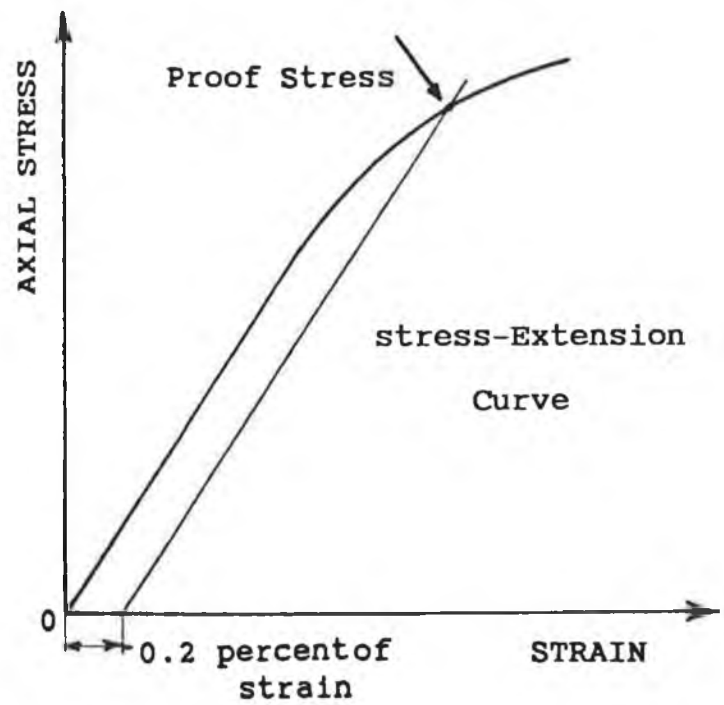


Fig.(1.3) Relationship between axial stress and strain of bolt material.

1.4 Background literature of bolted joint

1.4.1 INTRODUCTION

In order to fasten one part to another fastening bolts are often used in various kinds of industries [2]. To increase the reliability of bolted joints, three main factors should be considered:

(i) The analysis of joint loading (static and dynamic) and the effect on mechanically fastened members.

(ii) The selection and use of the fastener which must hold the joint together.

(iii) The amount of clamping force induced in the fastened joint during assembly.

The past decade has seen advances in all three of these factors. Various research and design groups have attempted to develop the relationship between the working loads imposed on the bolted assembly and the stresses felt by the bolt. Bolted joints are very important part of this design process [24].

1.4.2 ANALYSIS OF BOLTED JOINT

Normally, analysis of a bolted joint involves two plates clamped together by a bolt and a nut. An initial pre-load F_i is developed by tightening which induces a tensile loaded in the bolt and a compressive load on the clamped parts. All specialists in the field of the bolted joints agree that the value of the pre-load is one of the essential parameters of the joint, for example, carefully chosen pre-load will ensure that the shank of the bolt is not loaded in shear [25]. However, sometimes the bolt is designed for shear loading e.g. aircraft applications.

1.4.2.1 CLASSIC LINEAR CONCENTRIC BOLTED JOINT

The relationship between displacement and loading in elastic range can be represented as in **Figure (1.4)**. Under the initial pre-load F_i the bolt elongates by distance f_{bi} `line OB' and at the same time the clamped parts are compressed by distance $f_{ji}$ `line oj'. When the bolted joint is not subjected to the external load, these two lines are linear. The f_{bi} and f_{ji} are related to the tensile pre-load in the bolt, F_i, as follows:

$$f_{bi}=\delta_b . F_i . \qquad (1.5)$$

$$f_{ji}=\delta_j . F_i . \qquad (1.6)$$

The base line in **Figure (1.4)** represents the total elastic deformation of the bolt plus joint caused by the initial pre-load F_i. To calculate the spring constant for bolted joint **see section 1.4.3.5**.

1.4.2.2 CONCENTRIC EXTERNAL LOAD ON A CONCENTRIC JOINT

If the joint is now subjected to an external working load F_a, it will respond by:

- Further elongation of the bolt f_{ba}.
- Loss of compression of the clamped parts by the amount f_{ja}.

According to this model, the value of the pre-load has no-effect on the fatigue strength of the joint [2]. But the fatigue strength of threaded fasteners depends not only on the fastener material and pre-load, but also on the fastener diameter and the way it is manufactured [26].

Figure (1.5) shows that the application of an axial external load, F_a to tighten fastener results in a portion of F_a being felt by the fastener F_{ba}. This causes a further elongation of the fastener by an amount f_{ba},the remaining clamp load F_{kr} must then be sufficient to accommodate the axial and transverse forces on the plates during service [24]. That is, the external tensile load is divided into an additional bolt load F_{ba} and a load affecting the clamped parts F_{ja}. So that the following equations can be obtained:

$$F_a=F_{ba}+F_{ja}. \tag{1.7}$$

$$F_v=F_a+F_{kr}. \tag{1.8}$$

The simple joint diagram is shown in **Figure (1.5)** the linear line `**BCD**' indicates that the fastener material remains elastic. The external loading of the tightened fastener into the plastic range causes non-linearity in the joint diagram, given by the curve **AC** in **Figure (1.6)**, the remaining clamp load $F_{kr}$ being represented by the curve `EJ' in **Figure (1.6)**.

The ratio between the additional bolt load and external load is called load factor `$\phi$', and the ratio between the maximum and minimum clamp load is called tightening factor `α'.

1.4.2.3 CONCLUSION FROM THE CLASSIC LINEAR JOINT MODEL

The best relationship should be obtained between stiffnesses of the bolt and the clamped parts; hence, the bolt should have the lowest possible stiffness relative to the clamped parts. For this reason, the bolts are sometimes used with reduced shank diameter. However, bolts reduced shank always have reduced strength and hence not always desirable, since clamp load is the major influence on joint properties. Also, they are much more expensive and hence not often used.

It is also well known from that the fatigue strength of high grade fasteners can be increased with threads rolled after heat treatment. However, the fatigue strength is only relevant if the joint is subjected to cyclic loading, See **Figure (1.7)**.

The third conclusion from this **Figure** of bolted joints is perhaps the most important. The linear analysis indicated that the pre-load was a minor influence on the joint fatigue strength [6].

1.4.2.4 ECCENTRIC LOADING

Indeed, the geometry of bolted joint is more complex than simple concentric model. In particular, most joints have eccentricity and this is due to three axis not coinciding:

a. Axis of gyration of the clamped parts.
b. Axis of the bolt itself.
c. Axis of the applied external load.

Furthermore, there are two types of joints, open and close. Open joints are those in which the fastener is pre-loaded to a small fraction of the working load. In such joints the fastener is subjected to the full magnitude of the repeated dynamic load. Close joints are so called because the fastener pre-load generally exceeds the dynamic working load [26]. **Figure (1.8)** shows the diagram for an eccentrically clamped and loaded joint.In this case, the joint tends to open at the side closer to the applied load.

Figure (1.9) shows that the triangular joint diagram `SVP' changes with the assembly condition. If an eccentric external load F_a is applied the bolt extends under the partial load F_a. This causes the clamped parts to lose clamping force elastically until the point is reached when the joint begins to separate during this process. The joint behaves linearly and its characteristics are represented by the line `VH'.

At the point when the joint begins to open, the characteristics change from the linear line `VH' to a line asymptotic to line `SJ'. This means that the joint opening mechanism causes the bolt to feel more load at smaller external loads than in a classic linear joint model. The bolt also feels an increasing proportion of the external load as the external load increases.

In **Figure (1.9)** there are two diagrams, solid lines for a low pre-load F_{i1} and dotted lines for a high pre-load F_{i2} [6]. It is clear that the load experienced by the fastener, F_{ba1}, at high pre-load is much lower than that felt at low pre-load, F_{ba2}. If the external loads are dynamic the fatigue strength of the joint is improved by increasing the pre-load values. Higher pre-loads are achieved by using fasteners of higher ultimate tensile strength or by applying a more precise tightening method [24].

1.4.3 THEORETICAL DESIGN OF BOLTED JOINT

1.4.3.1 INTRODUCTION

The design of a bolted joint depends almost on experience. In reasonable design of a bolted joint, it is necessary to take account of distributions of the contact stresses and force ratios [2]. A mechanic with a typical set of wrenches will tighten a small bolt to higher initial stress than he will a larger one. For this reason, the design stress for bolts and screws should be a function of size when the computations consider only the external load [1].

In general, the large-diameter fasteners have only about one-third to one-half the fatigue endurance strength of smaller-diameter fasteners for given thread manufacturing techniques and joint types. However, fasteners with threads rolled after heat treatment have about three times the fatigue endurance of fasteners with cut threads in open joints and about 15% higher strength in closed joints [26]. But for alloy steel bolts this suggestion may not be applied.

1.4.3.2 THE INITIAL PRE-LOAD IN THE BOLT

The relationship between the applied torque and the initial pre-load is essentially linear as shown in Figure (1.10). This linearity is maintained up to and beyond the yield point of the bolt shank [8]. An approximate equation for the torque which is equal to thread torque plus collar torque is given by the following equation [27]:

$$T_{tot}=T_G+T_W. \tag{1.9}$$

Where:

$$T_G=F_i.C_1. \tag{1.10}$$

$$T_W=F_1.C_2. \tag{1.11}$$

$$C_1=d_1.(\tan\alpha'+\mu_G/\cos\Theta_n)/[2(1-\mu_G.\tan\alpha'/\cos\Theta_n)]. \tag{1.12}$$

$$C_2=\mu_W.R_c. \tag{1.13}$$

$$\tan(\alpha')=L/(\pi.d_m). \tag{1.14}$$

and

$$\tan(\Theta_n)=\tan(\Theta).\cos(\alpha'). \tag{1.15}$$

The magnitude of this torque may be either positive or negative. The tightening of a bolt is done by application of a rotation torque to its head, this applied wrenching torque causes the shank to rotate and the threads to tighten the bolt. Studies have shown that about **55** percent of the torque applied to the bolt head is transmitted to the shank and the screw thread, the remaining **45** percent is used to overcome the friction under the bolt head **[9]**.

It is recognized that the torque in the threaded part has little influence on the axial tension-elongation curve, and that the curve under external loading rapidly approaches the curve of the bolt in spite of the existence of the torque. Therefore, the tightening diagram in plastic region should be illustrated as shown in **Figure(1.11)**. In addition the tightening diagram in the case where the bolted joint is tightened in elastic region should be drawn as shown in **Figure (1.12)**. In the traditional tightening diagram of bolted joint, it is guessed that the threaded section torque induced by tightening is constant under external axial loading, and that the elongation curve of the bolt is drawn like the dotted line [28].

1.4.3.3 STRESS BEHAVIOUR DURING TIGHTENING

During tightening the fastener is subjected to a combined stress state comprising of an axial tensile stress due to the induced pre-load and shear stress due to torque applied to the bolt shank [11]. The uniaxial tensile stress is roughly constant across the bolt section while the shear stress increases from zero at the centre to maximum at the outer surface [9,11].

It is well known that a cylinder (like a bolt shank) subjected to combined tensile and torsion stresses, **Figure (1.13a)**, yield when the stresses get high enough. This yielding occurs first at the outer surface , the combination of tensile and shear stresses gives maximum stress when the combination is obtained according to the equation (1.4),[9].

That is: $$\sigma^2+3\tau^2=y^2. \qquad (1.16)$$

This expression is obtained from the **von Mises Yield Criteria** for two dimension stress [9].

Figure (1.13b) gives a Schematic diagram of the combined stress components in a bolt tightened below yield stress. These are represented as stress squared since the yield criterion is expressed in squared form. It can be said that the plastic region tightening and the elastic range tightening have more safety against fatigue because of higher pre-load; that is, lower stress amplitude inspite of the decrease of the fatigue strength of the bolt-nut joint by the increase of the mean stress [29].

The axial tensile stress component is dependent on the pre-load measured by the load cell and the tensile stress area of the fastener. The expression for direct tensile stress can be written as [30]:

$$\sigma=F_1/A. \tag{1.17}$$

The tensile stress area `A' is obtained from the following equation:

$$A=\pi.(d_1+d_2)^2/16. \tag{1.18}$$

Where,

d_1 is the pitch thread diameter.

d_2 is the minor thread diameter.

The shear stress is dependent on the thread torque `$T_G$', the ratio of the `T_G' to the wrenching torque `T_{tot}' will depend on the friction conditions at the mating surfaces [11]. Shear stresses may be calculated by using the theory of pure torsion for solid bars with circular cross section which gives [16,17]:

$$\tau=16.T_G/(\pi.d_2^{3}). \tag{1.19}$$

During tightening the tensile stress, the torsion stress and the resulting equivalent stress increase, but if the tightening process ends and the tightening torque is removed the following occur:

1. The torsion stress and also the combined stress decrease
2. The tensile stress remains almost unchanged after removal of the tightening torque. But this stress increases under the external load so that, the combined stress also increase and the torsion stress remains almost constant in this case [31] .

After the applied wrenching torque has been removed a relaxation of the material stresses occurs, which is characterised by a slight reduction in the clamp load. Hence, axial stress and a winding-back in the bolt shank leading to a reduction in torsional stress in the threads occurs. The current combined stress state Y^* is then given by the equation as follows:

$$\sigma_{com}^{*}=Y^{*}=\sqrt{\sigma^{*2}+3\tau^{*2}}. \tag{1.20}$$

In addition, the wrenching torque required to tighten fasteners to yield increase as lubrication conditions deteriorats [11,24].

In **Figure(1.14)** it is seen that the combined stress squared value is greater than the value of proof stress squared, and the material with the higher stress level may be considered to have entered the plastic region governed by the **Prandtl-Reuss Equations** [18]. The zone of material which becomes plastic, or the yield penetration from the minor diameter, may be observed from the **Figure (1.14)**.

The shear stress recovery may be large enough to cause the value of σ_{com}^{*} to decrease to a value less than the proof stress level. The magnitude of additional axial stress σ_{ba} required for the bolt to reach this subsequent yield may be represented by the following equation [24]:

$$\sigma_{ba}=\sqrt{\sigma_{0.2}^{2}-Y^{*2}}\ . \tag{1.21}$$

The value of σ_{ba} is represented in **Figure (1.15)**.

1.4.3.4 THE WRENCHING OFF-TORQUE

The torque required to loosen a fastener is dependent on the frictional conditions at the mating surfaces, that is, the required wrenching off-torque is the sum of the under-head torque, T_{w-off} and the thread torque, T_{G-off} [32].

$$T_{tot-off}=T_{G-off}+T_{w-off}. \tag{1.22}$$

1.4.3.5 <u>THE SPRING CONSTANT OF THE FASTENER AND JOINT</u>

To construct a joint diagram,it is necessary to determine the spring constants of both the bolt and the joint. In general the spring constant is defined as follows [33]:

$$K=E.A/L. \tag{1.23}$$

1. <u>THE SPRING CONSTANT FOR BOLT</u>

To calculate the spring constant of a bolt with different cross sections, if is necessary to calculate the spring constants of each section of the bolt, then the spring rates of each section are added:

$$1/K_b=1/K_1+1/K_2+\text{------------}+1/K_n; \tag{1.24}$$

Thus, for the bolt shown in **Figure (1.16)** the spring constant is calculated as follows:

$$1/K_b=(0.4d_2/A+L_1/A1+L_2/A_2+L_3/A_3+0.4d_2/A_m)/E_b. \tag{1.25}$$

This formula considers the elastic deformation of the head and the engaged thread with a length of **0.4d** each that is, the total spring rate of the bolt is equal to the sum of the reciprocal spring rate of each section of the bolt [33].

2. <u>THE SPRING CONSTANT FOR CLAMPED PARTS</u>

Calculation of the spring rate of the compressed joint parts is more difficult because it is not always obvious which parts of the joint are deformed and which are not. In general, the spring constant of clamped parts is given by the following equation [33]:

$$k_j = E_j . A_s / L_j . \quad (1.26)$$

Where A_s is the cross sectional area of a hollow cylinder which elastically behaves similar to the clamped plates. A more detailed discussion about the cross sectional area A_s is given in **Appendix A2**.

If the connected parts of the bolted joint are composed of two or more kinds of materials, for example, a gasket between connected parts, the spring constant for the connection is as follows [1].

$$1/K_j = 1/K_1 + 1/K_2 + \text{-------------------} + 1/K_n . \quad (1.27)$$

3. THE SPRING CONSTANT FOR BOLTED JOINT

The combined fastener and joint spring constant is dependent on K_j and K_b values, since the fastener and the joint are in parallel [33] in this case the following expression is used:

$$1/K = 1/K_b + 1/K_j ; \quad (1.28)$$

Or

$$K = K_b . K_j / (K_b + K_j) . \quad (1.29)$$

The factor which influences the alternating stress felt by a bolt in a concentrically loaded joint is the spring rate of the bolt [3].

1.4.3.6 THE RESILIENCE OF THE FASTENER AND JOINT

Resilience is a mechanical property of both the fastener and the clamped parts. It is the reciprocal of material stiffness and only applies to the elastic range of loading. Resilience is constant for the bolt and for the joint, and relates the amount of extension or compression to the applied load. The resilience can be derived from Hook`s law as follows [34]:

$$\delta=L/(A.E). \tag{1.30}$$

By comparing the value of δ and the value of K, the resilience is obtained by inverting the spring constant. So that, the resilience for the bolt is:

$$\delta_b=1/K_b. \tag{1.31}$$

And the resilience for clamped parts is given by:

$$\delta_j=1/K_j. \tag{1.32}$$

Since the fastener and the joint are in parallel, the combined fastener and joint resilience is written as in the following expression:

$$\delta=\delta_b+\delta_j. \tag{1.33}$$

1.4.3.7 THE TIGHTENING FACTOR

The tightening factor is defined as follows:

$$\alpha = \frac{\text{maximum clamp load}}{\text{minimum clamp load}}. \tag{1.34}$$

This equation represent the scatter between maximum and minimum clamp loads obtained on tightening to yield. The scatter increase as lubrication conditions detoriorate [24] where tightening factor varies between **1.25 and 3.0** depending on the tightening method **[33]**.

There are two advantages of tightening fasteners beyond the elastic region and into the plastic region; a higher pre-load and the diminished scatter [35]. **Figure (1.17)** shows the relationship between pre-load and tightening angle.

1.4.3.8 THE LOAD FACTOR

The load factor relates the amount of additional bolt load to the external loading. This is outlined by **JUNKER** in **VDI 2230** [34]. Thus, in general the load factor is written as follows:

$$\phi=F_{ba}/F_a. \tag{1.35}$$

The load factor indicate the proportion of working load, F_a, applied to the joint which is felt by the bolt [36]. To calculate the load factor for concentric bolted joint there are two situations which may be considered:

(i) The working load is constrained under the bolt head and nut as shown in **Figure (1.18a)**. In this case the load factor can be written in terms of the resilience for the bolt and the connected parts:

$$\phi_k=\delta_j/(\delta_b+\delta_j). \tag{1.36}$$

Or the load factor can be written by means of the spring constants for the bolt and the connected parts:

$$\phi_k=K_b/(K_b+K_j). \tag{1.37}$$

(ii) In most practical cases the working load will not be constrained and will be located somewhere between the bolt and nut bearing levels in distance $(n.L_j)$.In this case the load factor is given by:

$$\phi_n=n.\phi_k. \tag{1.38}$$

The result is that the parts outside the distance `$n.L_j$' are further compressed by the working load [36] as shown in

Figure (1.18b).

A more detailed discussion of the load factor for an eccentric bolted joint and the method to calculate this factor is given in **Appendix A3.**

1.4.3.9 SOME CAUSES FOR THE PRE-LOAD LOSS IN BOLTED JOINT

Loss of fastener pre-load is the cause of many bolt fatigue failures. The bolts in all joints lose a proportion of the pre-load due to relaxation of the bolted assembly. External loads cause only a small additional plastic elongation of the bolt above that resulting from relaxation or permanent set **[33]**. The pre-load loss due to permanent set may be given by the following equation:

$$DF_i = K.S. \qquad (1.39)$$

Where,

DF_i is the Pre-load loss due to permanent set.

K is the Spring constant for bolted joint.

S is the permanent set.

Permanent set which occurs during tightening can be compensated for by further tightening. Pre-load does not effect the alternating load felt by the bolt until the pre-load drops to the point where the external load causes the joint to open, at which time it feels the full alternating load. This latter condition would reduce the fatigue life of the bolt thread or head due to the increase in alternating stress.

One of the mechanisms that can lead to premature thread fatigue failures is pre-load loss due to permanent set. Permanent set may be caused by torsional and normal bearing stresses developed under the bolt head which exceed the compressive yield point of the joint material to produce embedding, or by plastic deformation of the bolt head [3].

More information on calculating the per-load loss due to permanent set is given in **Appendix A4** .

1.4.4 ADVANCES IN HIGH-PERFORMANCE FOR BOLTED JOINT

1.4.4.1 MEASUREMENT OF YIELD POINT

The yield point of materials is usually defined as the stress in a material at which there occurs a marked increase in strain with small or no increase in stress. In practice, this definition is often difficult to apply as, many materials have no clear `knee' or sharp transition from elastic conditions. Practical measurement of yield point is usually done by the **0.2%** offset method.

The usual laboratory technique for determining yield in tensile tests uses an autographic recorder plotting load versus extension, the knee is then applied by manual measurement. This involves increasing the load or the displacement at a constant rate and observing when the strain increases rapidly or the load increase stops; these may be regarded as slope methods [8].

1.4.4.2 METHODS OF TIGHTENING CONTROL

A brief review is made of the various methods which may be used for high quality installation of threaded fasteners. These methods are:

1. BOLT ELONGATION CONTROL

One of the most accurate methods is by measurement of bolt elongation. If the extension of the bolt could be measured directly, it would be an easy task to determine the clamp load. Since this is rarely possible, practical indirect procedures must be employed to control the clamp load under production conditions. However,these methods are time consuming or costly and are never used where time or cost are overriding factors, for example, automotive industry [8].

2. TORQUE AND ANGLE CONTROL

High accuracy methods which also permit low installation cycle times depend upon measurement of both torque and rotation angle of the fastener.

The most common method is the use of a control torque , which gives poor clamp load accuracy. Experimental studies confirm that the range of clamp load is rarely below ±20%, which means that the lowest clamp load is **66 percent** of the maximum with torque scatter of ±10% . The expected variation in clamp load is more of the order of ±25% [8].

The other control method which has been used in the construction industry is angle-controlled tightening. A predetermined tightening rotation is applied to the fastener to take it past the yield point. This gives both a high clamp load and relatively small scatter. While these advantages are very significant, there are two major disadvantages:

- In order to determine the setting of yield point, relatively large rotations must be chosen which can lead to over extension of the fastener.
- And the choice of rotation angle for a particular joint can involve extensive preparatory test work.

These two control methods are the most significant high production methods in use at the present time [8].

3. NEW JOINT CONTROL SYSTEM OR YIELD CONTROL

This work was based on studies of torque-rotation curve. It is further noted that the gradient of the torque-rotation curve had a characteristic shape with a peak followed by a steep drop in magnitude. Thus any characteristic change in the gradient of the torque-rotation curve will correspond to an equally characteristic point on the pre-load or clamp load-rotation curve. A suitable control algorithm was devised to

- Determine the torque gradient,
- Find and store the largest value for that particular tightening cycle.
- Stop and process when the current torque gradient drops significantly (50% of the maximum).

Evaluations of the new joint control system performance show that scatter in the clamp load is reduced to about ±8% while minimum clamp load is increased to about **160%** of that achieved with torque control as shown in **Figure (1.19)** [8].

It has the major advantages that the bolts are tightened only up to the yield point [10]. In **Figure (1.20)** the detection of torque-tension yield point in a bolt is shown by using torque-angle gradient.

1.4.4.3 HEAT TREATMENT FOR BOLTS

There are seven different heat treatment methods which may be used to enhance the properties of steel fasteners. These methods are Quenching, Annealing, Tempering, Stress Relieving, Carburising, Cyanide Hardening and Dry Cyanide. But the external threads for bolt are manufactured by machining.

A rolled thread is inherently stronger than a machined thread, especially under conditions of dynamic loading.

Other advantages offered by thread rolling are:

- Better surface finish which reduces the incidence of stress raises.
- Fastener production times.
- Suitability for many materials.

Threads may be rolled after heat treatment or before heat treatment. If heat treatment is applied after thread rolling then much of the advantage of rolling is lost because heat treatment will produce stress relief [5].

However, fasteners with threads rolled after heat treatment have about **three times** the fatigue endurance strength of fasteners with machined threads in open joints and about 15 percent higher strength in closed joints [26]. It is also higher than that of fasteners rolled before heat treatment. The fasteners with thread rolled after heat treatment are also found to withstand higher external loads before joint failure compared with those having thread rolled before heat treatment for the same lubrication. The thread rolling after heat treatment are more costly to produce since the process require higher forces during manufacture which result in increased tool wear and higher energy costs. Therefore, it is more economical to use fasteners with thread rolling before heat treatment when high clamp loads are not required [11,24].

A more detailed discussion about the heat treatment methods and the manufacture of bolts is given in **Appendix A5**.

1.4.4.4 RESIDUAL COMPRESSIVE STRESSES

When threads are rolled before heat treatment residual stress are relieved by the treatment which should reduce fatigue strength [26].

The stresses introduced during the manufacture of thresd rolled after heat treatment is called residual compressive stresses and act in the opposite direction to axial tension stresses. These stresses in the fastener are represented in **Figure (1.21)** for pure axial loads and in **Figure (1.22)** for the combined stress squared [37].

1.4.4.5 FATIGUE STRENGTH OF BOLTED JOINT

The characteristics of fatigue strength on joint are affected strongly by the load distribution on the screw thread connection.

In recent years, a new fastening method exceeding the yield strength of bolt material is adopted in order to increase the tightening performance for screw joints [38]. The loading capacity of high strength bolted joints is determined in most cases by the fatigue strength.

The fatigue strength of a bolted joint is strongly influenced by the bolt and nut and on the design factors and also on the pre-load introduced during assembly [31]. The fatigue strength of joints, is increased by increasing the pre-load in the fastener through improved control of the tightening process which involves taking fasteners to the yield point of the fastener material [6].

The fatigue properties of industrial grade fasteners are much lower than the fatigue strength of the base material [39]. The fatigue strength improvement resulting from high pre-load can be increased further by the use of high fatigue bolt materials [9]. Also the fatigue limits of the bolted joints can be improved when the bolt is tightened to the plastic region [35,40].

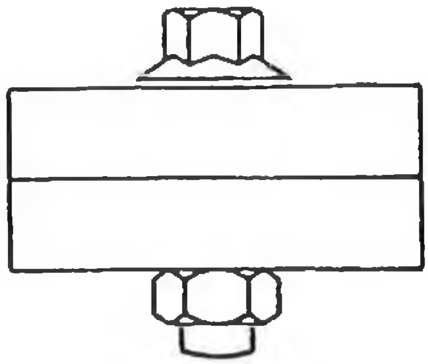
1.4.5 PREVIOUS CONCLUSION OF BOLTED JOINT

A number of experimental investigations have been undertaken to provide some results for bolted joints. Some of these results were provided by number of experimental works such as:

- The clamp load and extension at yield for a yield tightened fastener are dependent on the lubrication conditions on the contact surface [8].
- There are some benefits in tightening a joint to the bolt torque-tension yield point[9]:
  - The joint can withstand higher working loads before opening.

- Fatigue strength of the joint is increased to its maximum value because fatigue failure mainly occurs when the joint open.
- The level of external loading required to cause the joint to open and therefore joint failure are higher when fasteners are well lubricated.
- The additional fastener load due to external loading was found to be related to both the fastener type and lubrication conditions. As the lubrication conditions improved the additional fastener load decreased.
- Tightening factor and the coefficient of thread friction increased as the lubrication conditions deteriorated.
- The values of shear and tensile stress within the fasteners were influenced by lubrication conditions.
- For each lubrication conditions tested the fasteners with thread rolled after heat treatment gave higher clamp loads at yield compared to those having thread rolled before heat treatment [11,24].
- To produce high clamping forces under moderate wrenching torque the frictional effects on the mating surfaces should be low.
- Yield tightening achieved very constant value of clamp load under varying fastener friction conditions.
- To prevent self-loosening during service particularly for a joint under transverse loads, a reasonable level of friction is required.
- The most highly stressed joint can withstand the largest dynamic load [32].

- The two main causes of fastener failure are pre-load loss due to static and dynamic mechanisms. Static pre-load loss is caused by permanent set taking place at the joint interface and mating threads of the fastener pair [39].
- When the bolts were clamped in the plastic range, the fatigue limits of the bolted connections were improved, and the variations of the clamping force, axial load amplitude and force ratios were hardly recognized [40].
- The pre-load produced in a bolt by a given torque tends to decrease with repeated tightening even if the bolt is cleaned and re-lubricated every time the joint is disassembled [41].



Classic linear concentric bolted joint without external loading.

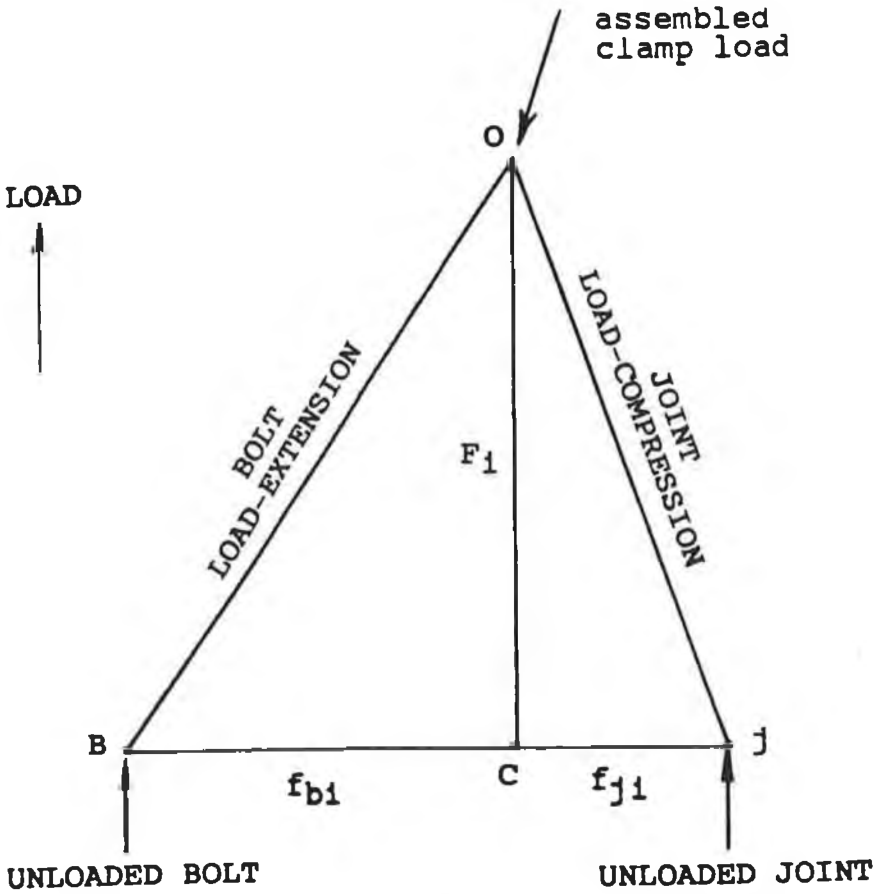
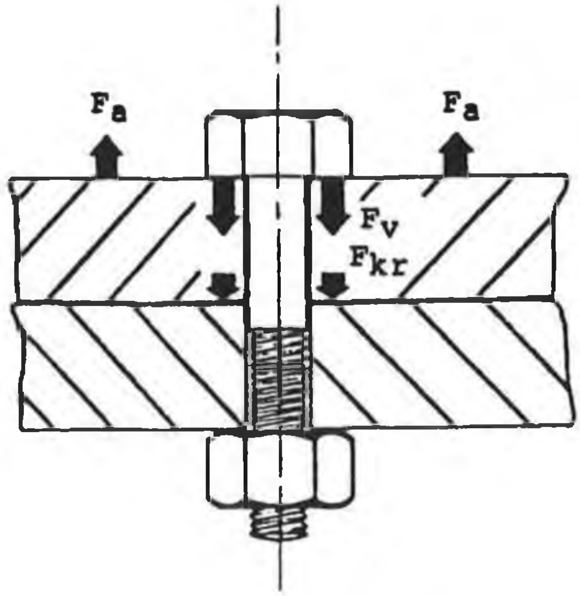


Fig.(1.4) Joint diagram for assembled symmetrical bolted joint.



Concentric external load on aconcentric joint.

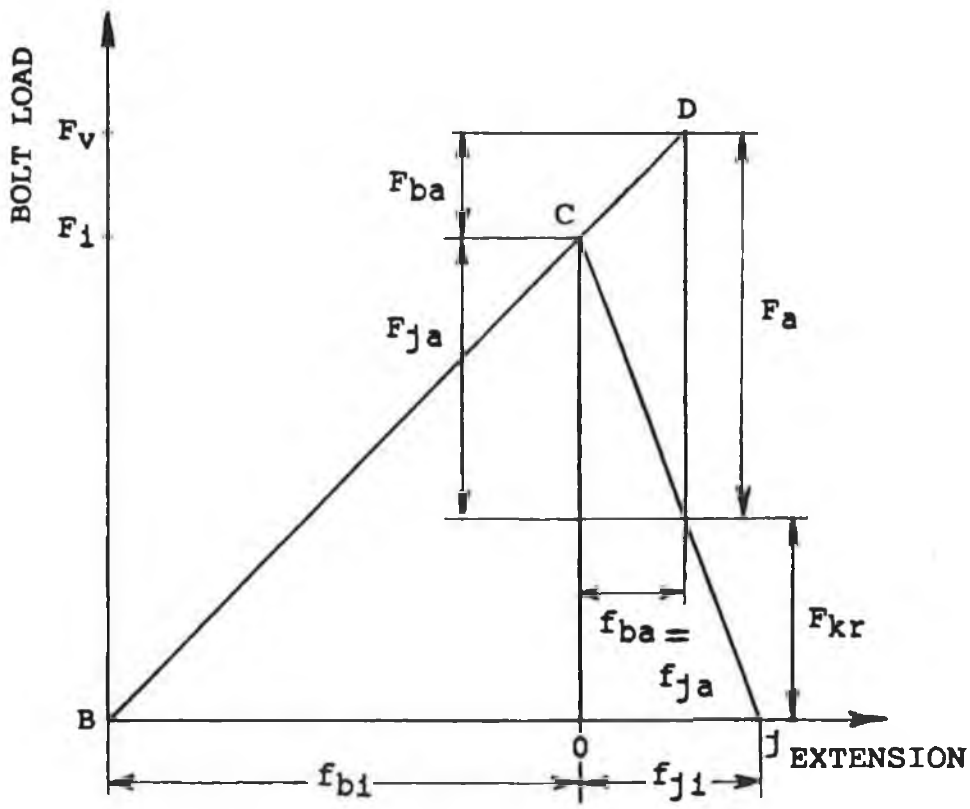


Fig.(1.5) Joint diagram for assembled symmetrical joint with external tensile load applied.

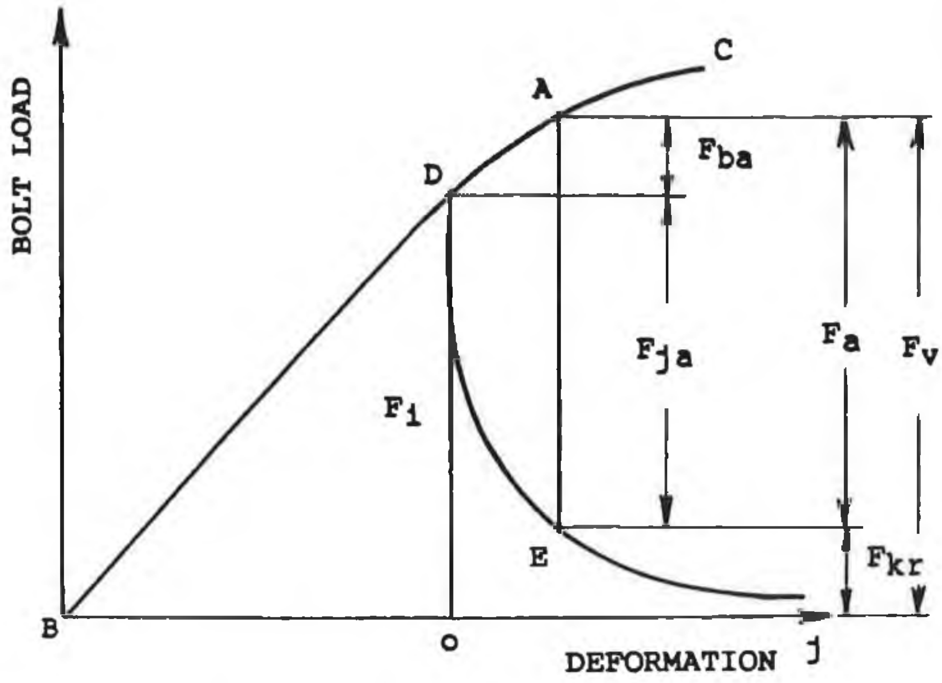


Fig.(1.6) Joint diagram for yield tightening showing bolt load - extension curve during tightening.

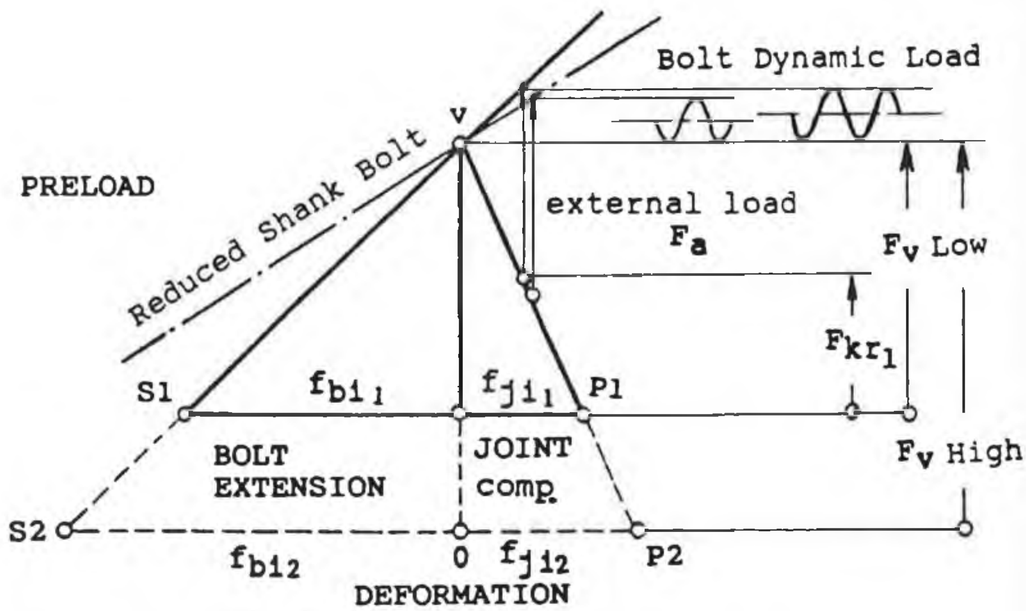
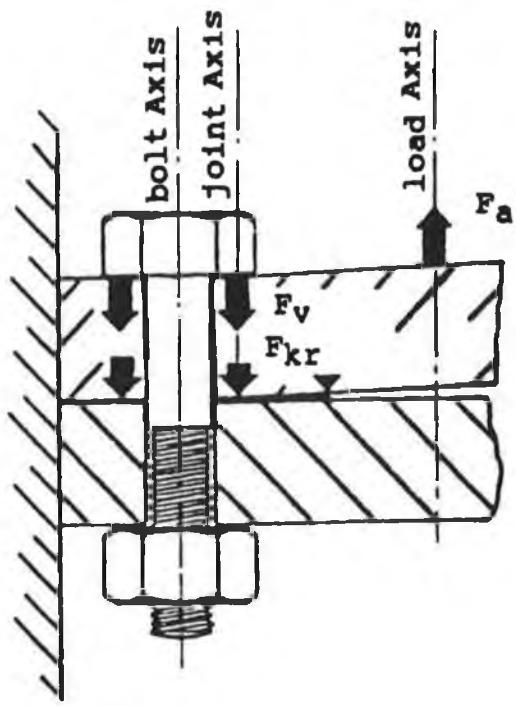


Fig.(1.7) Classic joint diagram.



Eccentric joint with external load applied.

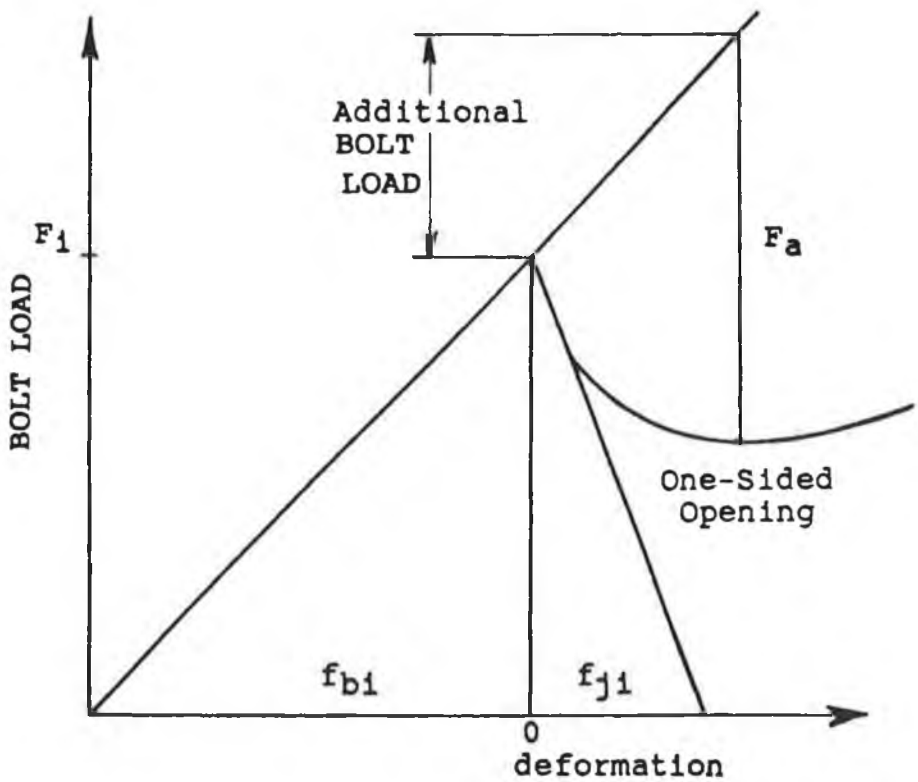


Fig.(1.8) Diagram for eccentric joint with external load applied.

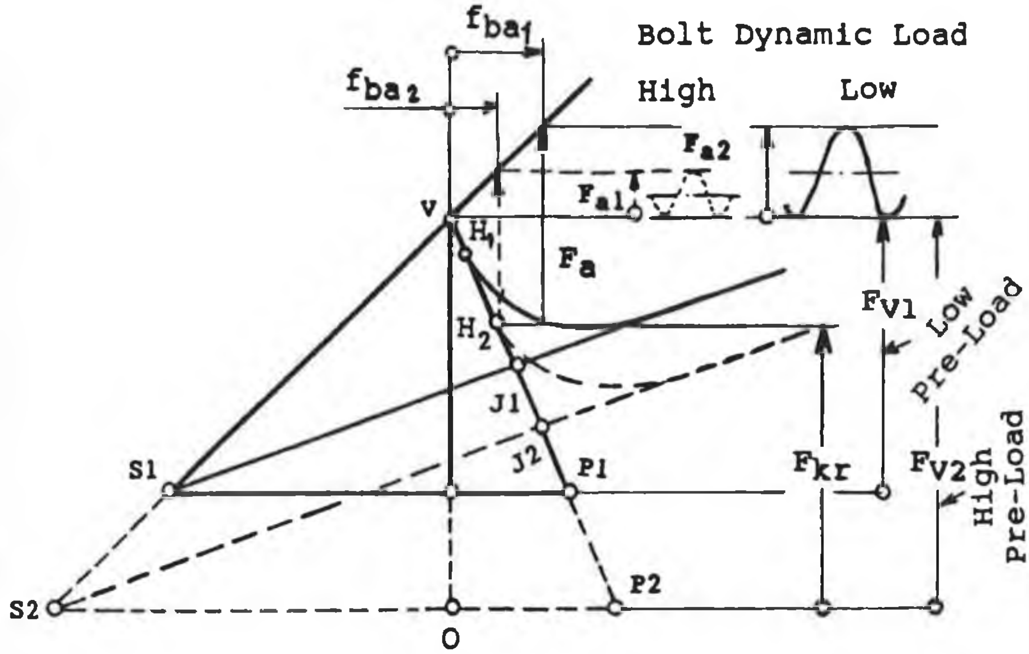


Fig.(1.9) New joint diagram for eccentrically clamped and load joint.

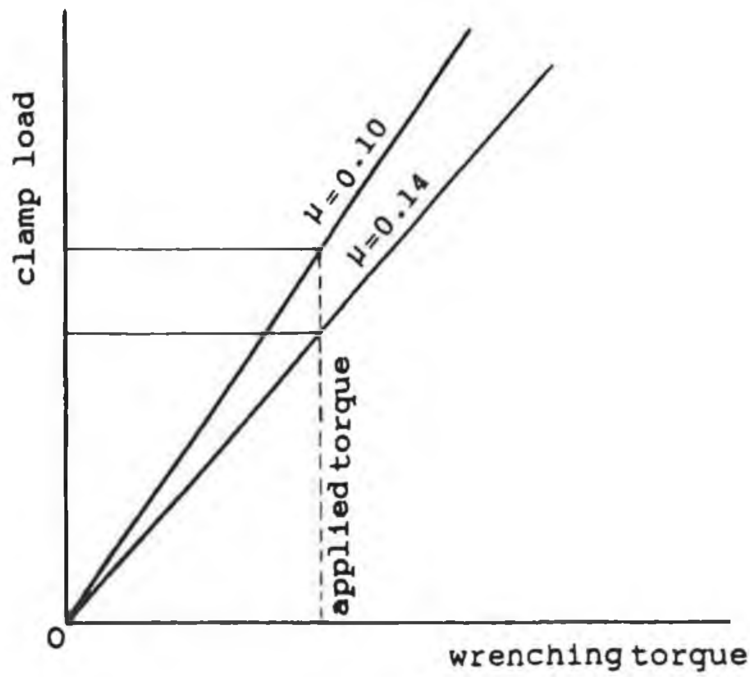


Fig.(1.10) Relationship between clamp load in bolt and wrenching torque.(Effect of friction on the clamp load).

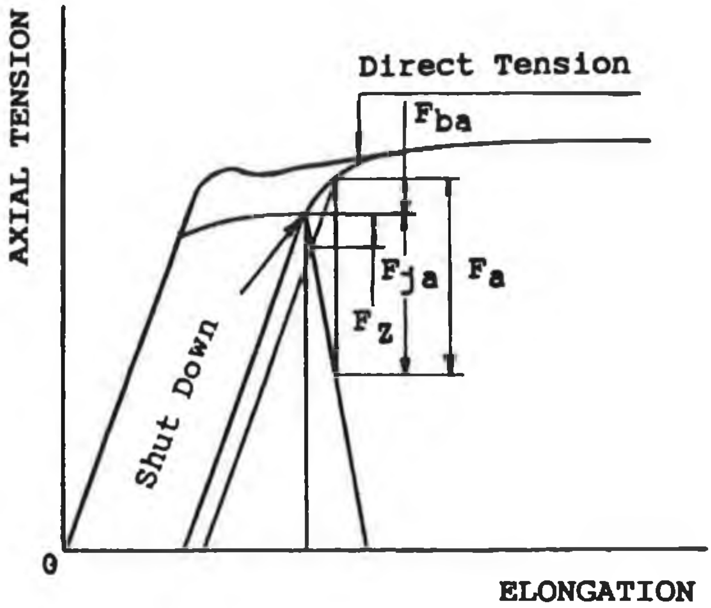


Fig.(1.11) Tightening diagram in plastic region.

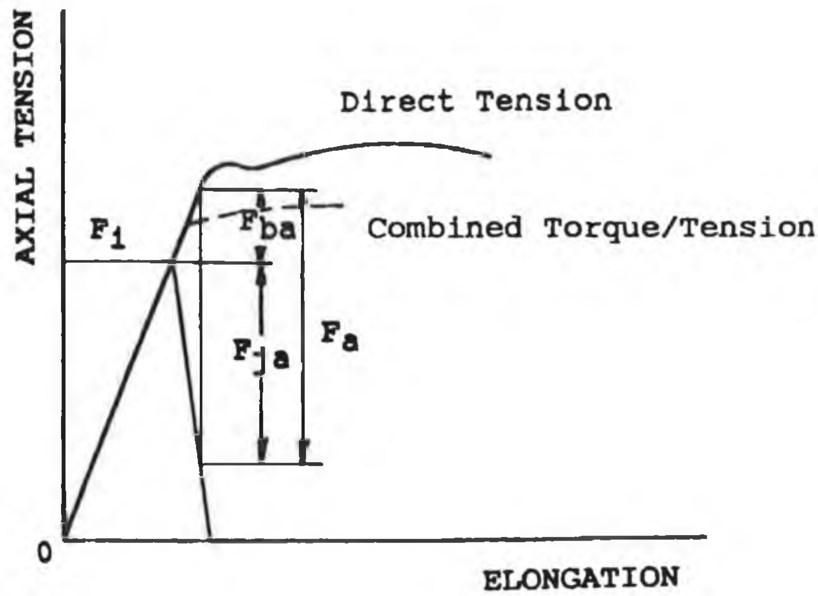


Fig.(1.12) Tightening diagram in elastic region.

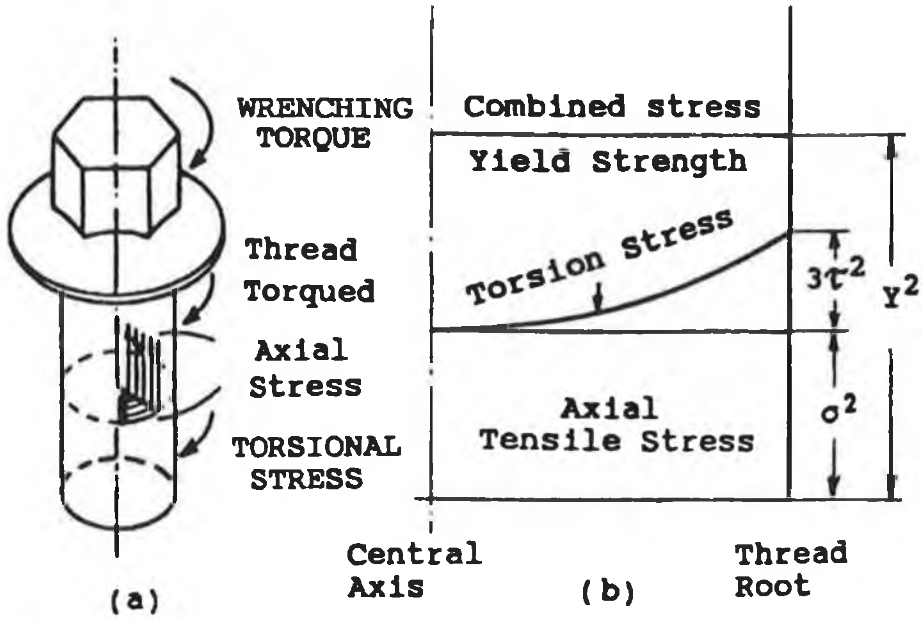


Fig.(1.13) Representation of the stress components in a bolt tightened below the yield point with external torque still applied.

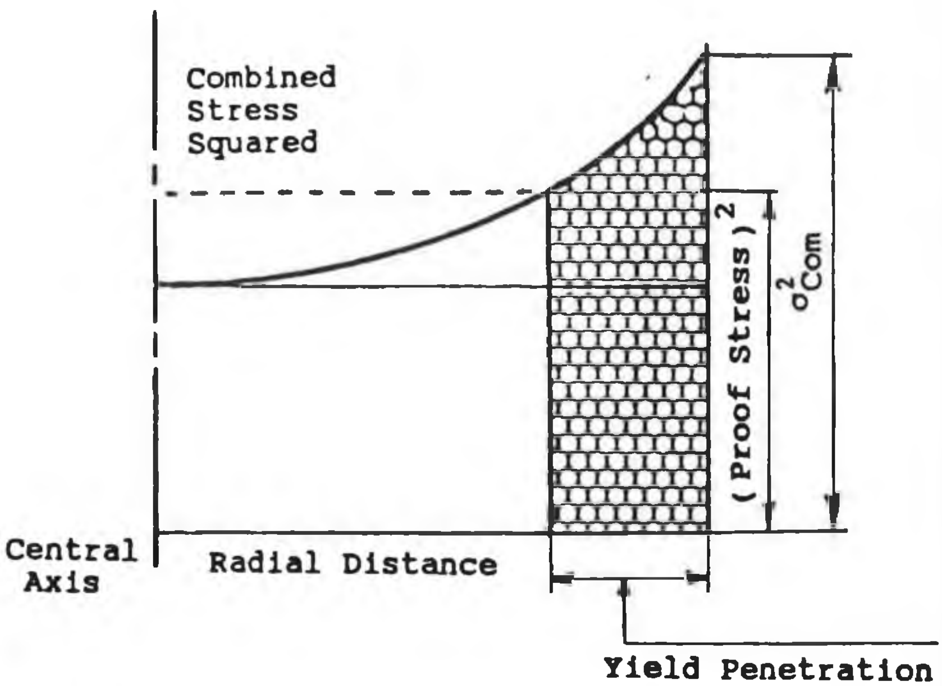


Fig.(1.14) The yield penetration from the minor diameter.

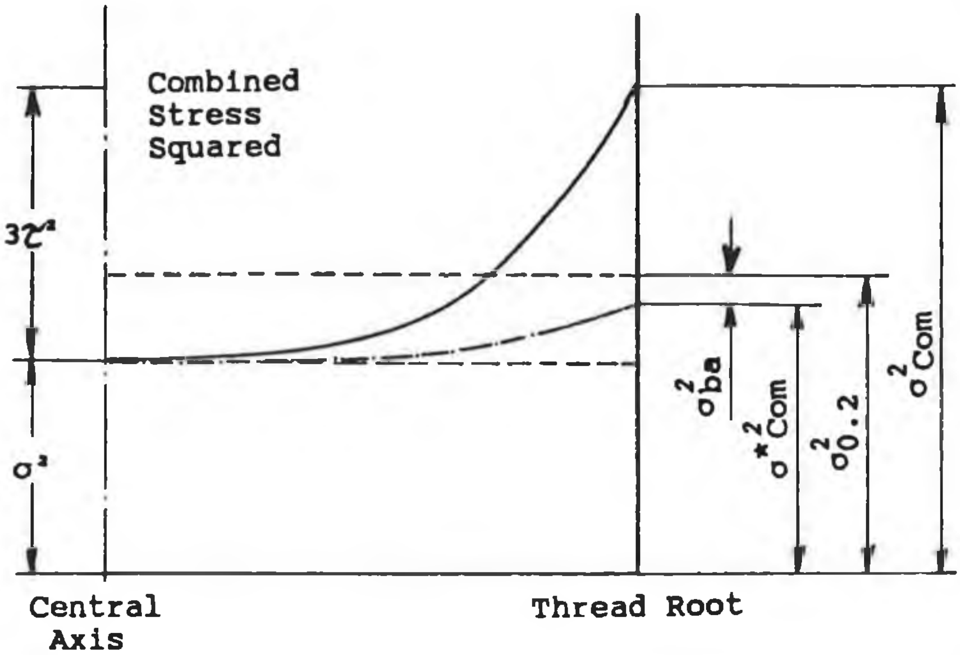


Fig.(1.15) Combined stress squared in bolt thread from torsional and tensile stress.

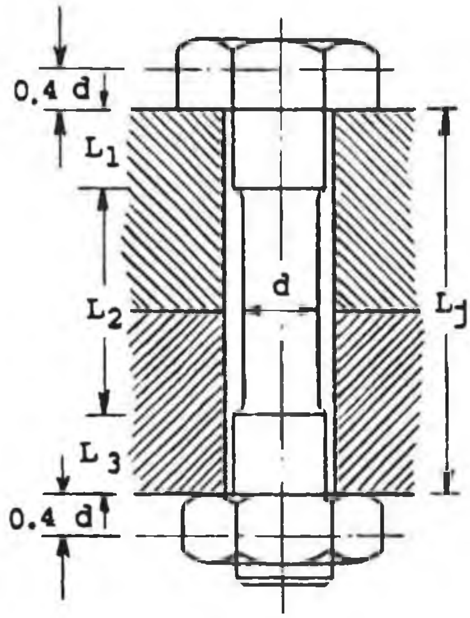


Fig.(1.16) Analysis of bolt lengths contributing to the bolt spring rate.

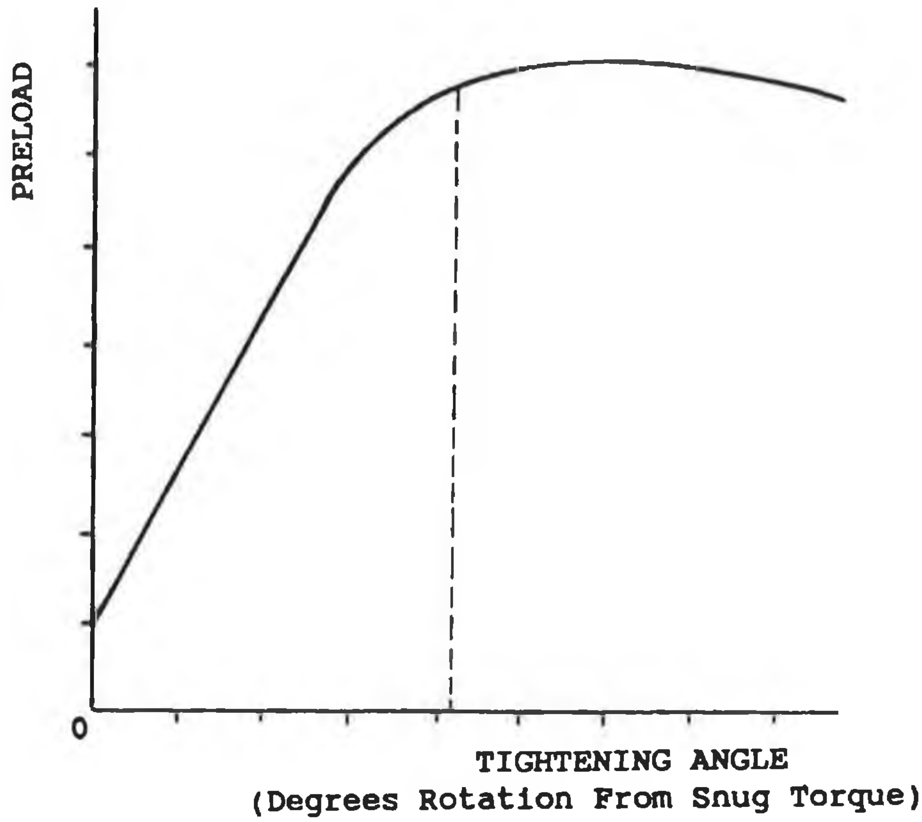


Fig.(1.17) Relationship between preload and tightening angle.

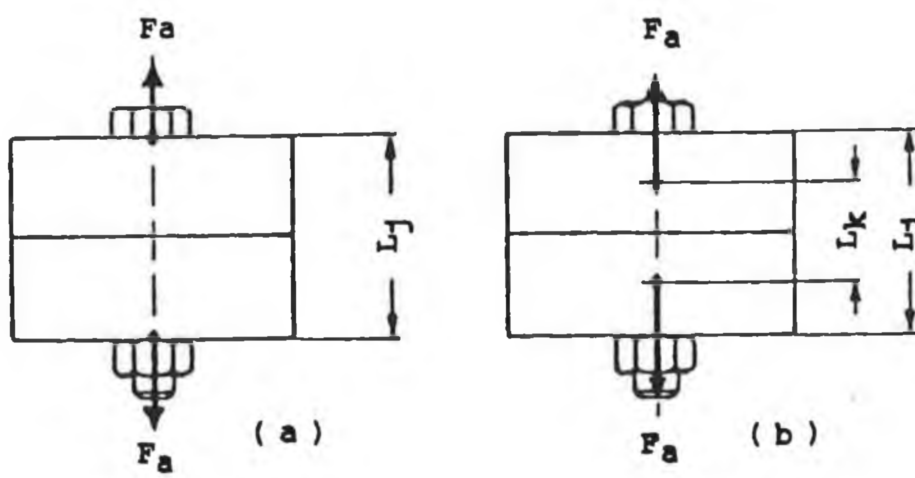


Fig.(1.18a) The working load introduction under bolt head form concentric bolted joint.

Fig.(1.18b) The working load introduction in levels for concentric bolted joint.

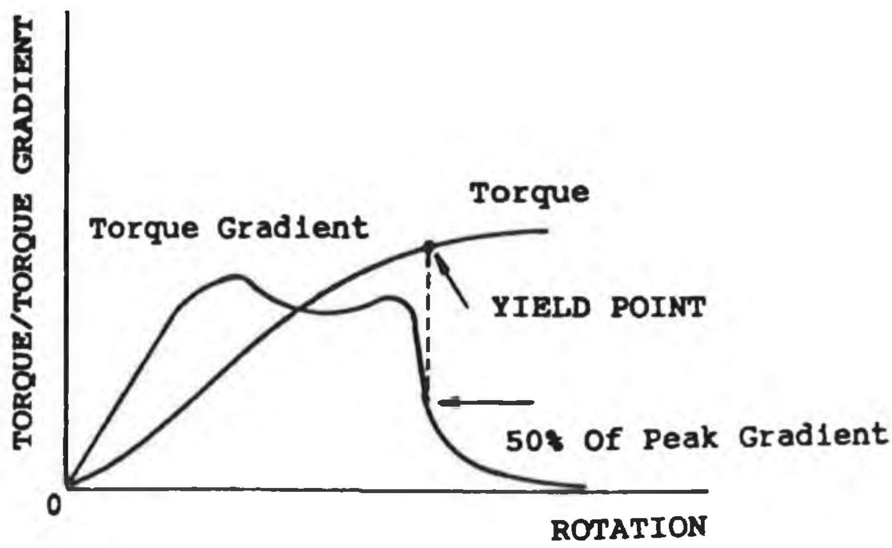


Fig.(1.19) Observed torque and torque gradient signals as functions of the rotation angle for a typical bolted joint.

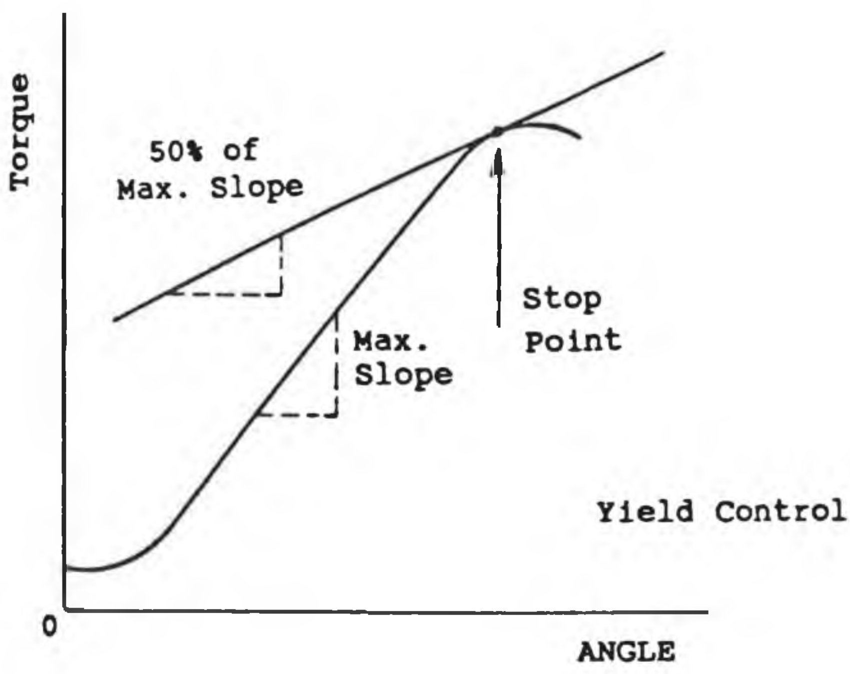


Fig.(1.20) Detection of torque-tension yield point in a bolt by using torque-angle gradient.

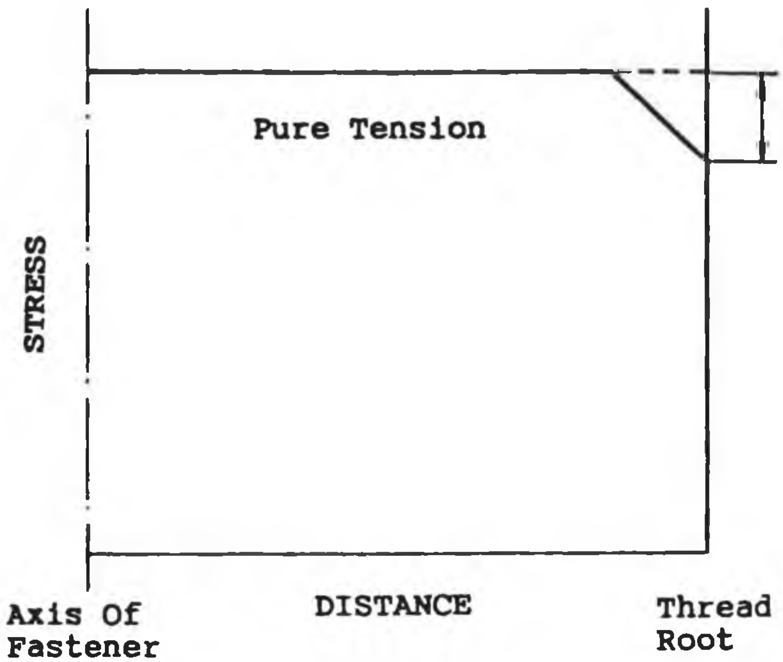


Fig.(1.21) Compressive residual stresses for pure axial loads in the fastener.

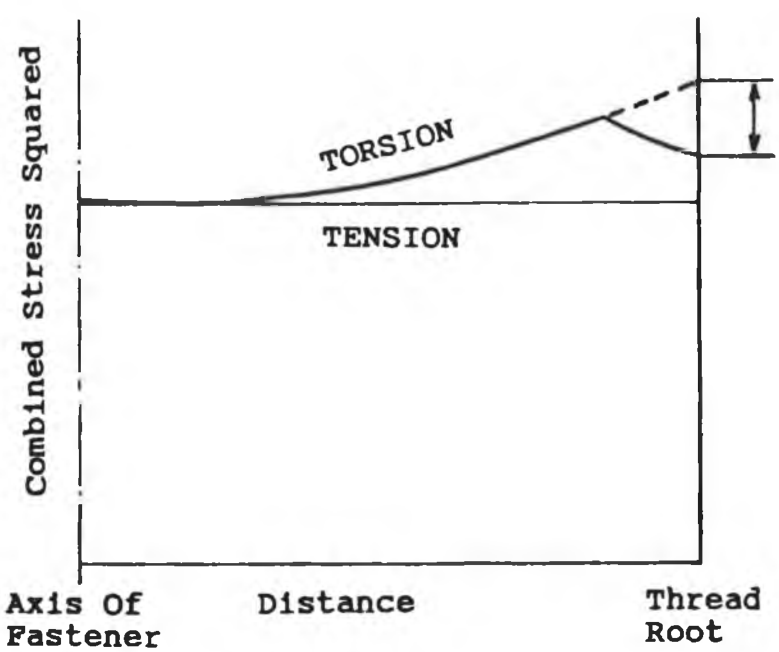


Fig.(1.22) Compressive residual stresses for the combined stress squared in the fastener.

1.5 Plan and aim of the present work

1.5.1 INTRODUCTION

When a fastener is tightened properly, it does not become loose under normal service due to vibration but when a fastener is over or under tightened it may break during tightening or fail while in service. It is therefore very important to apply the tightening torque accurately.

If a fastener is torqued too low so that the mean tension is less than the peak load on the parts then failure eventually will occur in the fastener or the parts. If a fastener is torqued too high the fastener will either break immediately or fatigue rapidly in use and break. So that the fastener should be torqued correctly [42].

1.5.2 PLAN OF THE CURRENT WORK

The investigations undertaken in this thesis were in collaboration with SPS Research Laboratories, Nass, Ireland. This company manufactures high performance fasteners and wished to investigate the effect of combined stresses (tension and torsion) on the yield stress of the fastener material by means of:

- The response of fasteners under uniaxial loading.
- The response of fasteners during tightening.
- The response of fasteners under combined torque and axial load.

The present work is carried out by using bolts having thread rolled after heat treatment. The fasteners selected for testing were **M12X1.75X150 MM**, **I.S.O.**, **Grade 10.9**. The thread rolled after heat treatment fasteners were selected because previous studies found that this kind of fasteners have better fatigue life than the fasteners having thread rolled before heat treatment.

The present investigation was scheduled according to the following plan of work:

1. calibration of the equipment.
2. The mechanical response of the fasteners under uniaxial (tensile) loading when tightening is not involved in order to establish the yield strength of the bolt material.
3. Calibration of electronic torque wrench, to determine the relationship between torque and the angle of twist in the fastener.
4. Effect of combined torque and axial load on the fastener. This test was carried out as follows:

4.a At first, the fastener is subjected to a torque within elastic range, then it is subjected to different magnitudes of axial load until yielding occurs, when the torque is still held.

4.b At first, axial loading is applied to the fastener within elastic range,then it was subjected to different magnitudes of torque, when the load is still held.

5. At first, the bolt is loaded under uniaxial load to the plastic yielding, then unloaded and subsequently subjected to combined axial load and torque.

6. Experiments in stages 4 and 5 were repeated after attaching strain gauge to the bolt shank to determine the torque.

All these tests are carried out for fasteners having thread rolled after heat treatment, and by using flat washers, in order to restrict loss of tightening of the joint when the joint is unloading resulting from vibration and corrosion. An electronic hand torque wrench was used to tighten the fasteners.

For this investigation, it is required that concentric loads are applied to the fastener on concentric bolted joint.The concentric loading consisted of direct increasing external tensile load which was applied by a hydraulic pump. This work was undertaken to investigate the effect of pure torsion, axial load and the combination of torsion and axial load on the tightening of bolted joint, by determining:

- the load and extensions generated in the fastener due to the application of external loading on the tightened fasteners.

- the shear stress, axial stress and combined stress states in the fastener, which may be used to establish the yield point location of the fastener material from **von Mises `Hencky'** yield criterion.
- the torque transmitted to the bolt shank due to the application of external load after applying pure torsion.

1.5.3 AIM OF THE CURRENT WORK

Finally, the principal aim of the present work is to establish the mechanism in which yielding takes place under combined torque and axial load in a bolted joint. And the response of the fasteners to combined torque and axial load.

However, the main investigation was to determine the following questions:

1. Is the bolt material strain hardening ?.
2. Where does the torsional and tensile energy go ?.
3. How is yielding taking place under combined torque and axial load ?.
4. Does the bolt un-rotate when axial load is applied after initial torque tightening ?.
5. Why does the torque curve drop as soon as axial load is applied ?.

1.6 Outline of the chapters in the thesis

The research programme undertaken in this thesis was as follows. **The chapter one** provides an introduction of bolted joint, it focuses on the background literature and analysis of bolted joint. **Chapter two** describe in detailed the experimental equipment, design and materials. **Chapter three** presents to the theoretical analysis of the response of the bolt. **Chapter four** gives a detailed description of the experimental procedure and calibration of the equipment. **Chapter five** presents the experimental results. **Chapter six** provides the analysis and discussion of the results. **The final chapter** provides the conclusions of the research investigation and suggestions for further work and recommendations.

CHAPTER TWO

EXPERIMENTAL EQUIPMENT & DESIGN AND MATERIALS

CHAPTER TWO

EXPERIMENTAL EQUIPMENT & DESIGN AND MATERIALS

2.1 Introduction

The fundamental unit of fastener testing equipment was the instrumented joint apparatus. This joint was assembled to allow the fastener M12X1.75X150 mm, I.S.O., Grade 10.9, to be testing as follows:

- Load the bolt in tension to detect the yield point.
- Tightening the bolt.
- Apply an external tensile load to the joint until the bolt yields in tension, after initial torque tightening.

The instrumentation of the bolt testing rig was used for the experiments instead of using a conventional tensile test machine; thus avoiding the need for any special test joint. The output of the L.V.D.T. and load cell were connected to the amplifiers, and the outputs from the amplifiers were channelled through the input of a X-Y recorder.

2.2 Main equipment

2.2.1 ARRANGEMENT OF TEST JOINT

The geometry of the joint allowed a length of clamped components of the test joint consist of parts as shown in **Figure (2.1)** an enumerated as follows:

- Top and bottom uniaxial joint plates.
- Load cell.
- Hydraulic cylinder.
- Locking nut.
- Uniaxial loading plate.
- Flat load washer.
- Nut.

see **Plate (2.1)**.

2.2.1.1 TOP AND BOTTOM UNIAXIAL JOINT PLATES

The geometry of the top and bottom uniaxial joint plates were cylindrical to locate the load cell centrally. A suitable material was required for these plates to prevent embedding. A round Tool Steel, **Impax(20)** was selected, typical analysis of this material is given as follows:
Composition:

C/0.36% Mn/o.7% Si/o.3% Ni/0.7% Cr/1.8% Mo/0.2%

Heat treatment was applied to this material by the supplier to increase hardness to **32-35 Rc**, so that it was unnecessary to make further heat treatment. These plates were supplied by **S.P.S.** Laboratories Ireland. Details of the uniaxial joint plates are shown in **Figures (2.2)** and **(2.3)**.

2.2.1.2 LOAD CELL

The load cell was required to withstand loads of up to 100 kN. The force of the load cell used in this work measures dynamic and static force up to 100 kN. Because of its robust design the load cell can be centred on the inner or outer casing diameter. To adapt the mounting height the measuring surfaces can be ground for easier mounting. The load cell consisted of a metal framed structure with four strain gauges mounted in a Wheatstone bridge circuit arrangement, which consisted of one 120 ohm strain gauge in each arm of the bridge . The ohmmeter reading across points A,D is equal 90 ohm , while the ohmmeter reading between the points A,C is equal to 120 ohm .

Figure (2.4) shows the dimensions and the wiring diagram of load cell. The load cell was also supplied by S.P.S. Laboratories, Ireland.

2.2.1.3 HYDRAULIC CYLINDER

A hydraulic cylinder was used to apply external tensile load by a hand operated hydraulic pump. It can withstand high pressure of up to 700 bar and is made by Enerpac Hydraulic Cylinder Model RWH-120 BGOC, U.S.A. .It was also supplied by S.P.S. Laboratories, Ireland.

2.2.1.4 LOCKING NUT

To prevent the separation between the joint and the hydraulic cylinder, a locking nut was required to prevent self-loosening, a suitable material, Round Tool Steel, **Impax(20)**, was selected. It was supplied by **SPS**, Laboratories, Ireland.

Details of the locking nut geometry are shown in **Figure (2.5)**.

2.2.1.5 UNIAXIAL LOADING PLATE

The uniaxial loading plate was a cylindrical part. The joint consisted of a load cell between two uniaxial joint plates with the hydraulic cylinder to apply tensile loads. A suitable high strength material, Round Tool Steel, **Impax(20)**, was selected to prevent embedding. By heat treatment the hardness was increased to **32-34 Rc**. These uniaxial loading plates were supplied by **S.P.S.**, Laboratories, Ireland.

Details of the plate geometry are shown in **Figure (2.6)**.

2.2.1.6 FLAT LOAD WASHER

The load washer was required to reduce any possibility of embedding and to ensure evenly distributed loading. The flat washer, pre-hardened to **45 Rc** was supplied by **S.P.S.**, Laboratories, Ireland, and were thick enough to allow for re-grinding of the surfaces if necessary. Details of the flat load washer are given in **Figure (2.7)**.

2.2.1.7 NUT

The nuts used in the experiments were M12 with a thread pitch 1.75 mm. they were pre-hardened to 35 Rc to enable high tensile load of the fastener without thread deformation. In order to reduce the number of the nuts used in the testing a nut was machined from a bar of a Tool Steel Impax(20). This nut was also supplied by S.P.S. Laboratories, Ireland. See Figure (2.8).

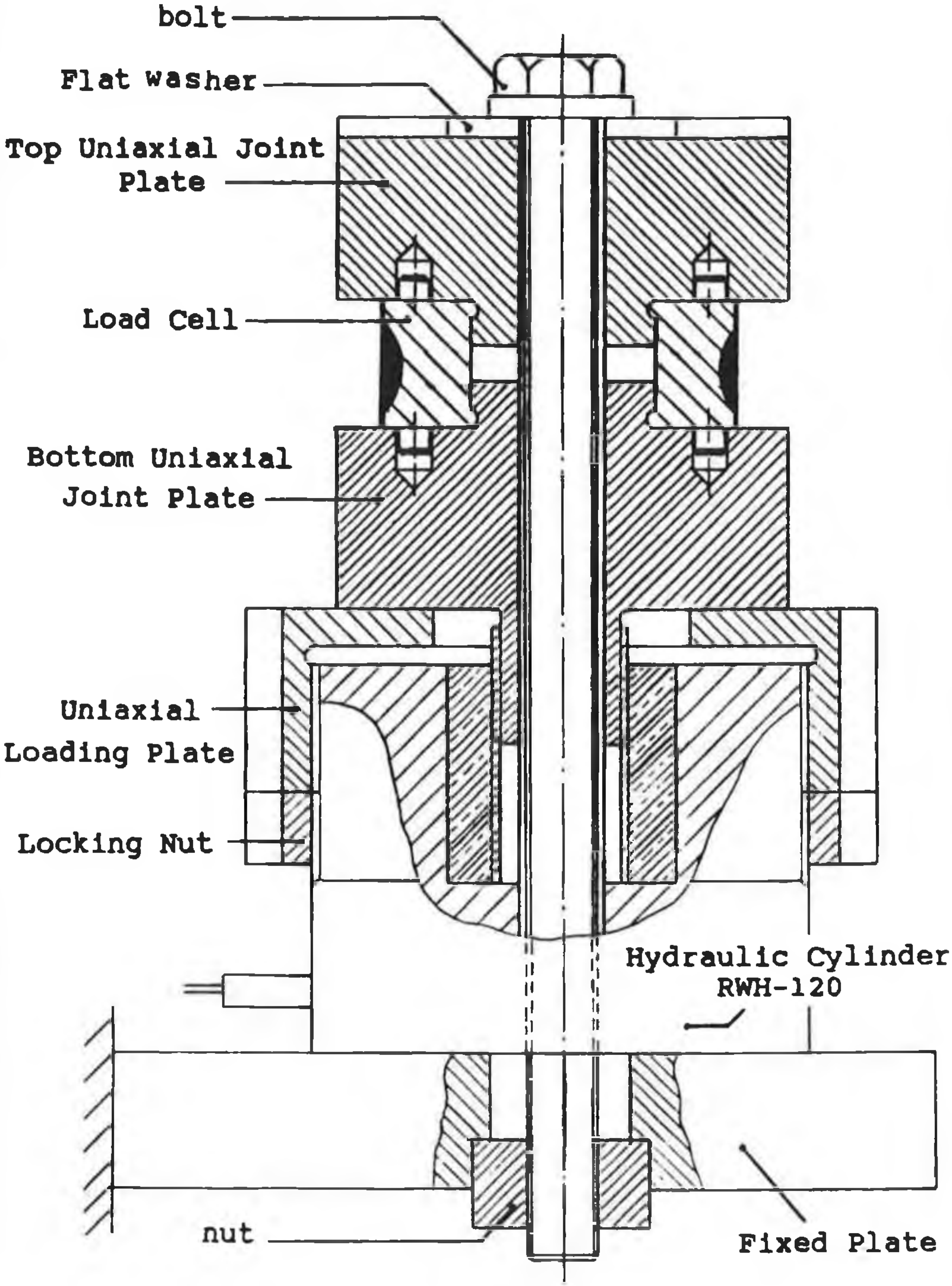
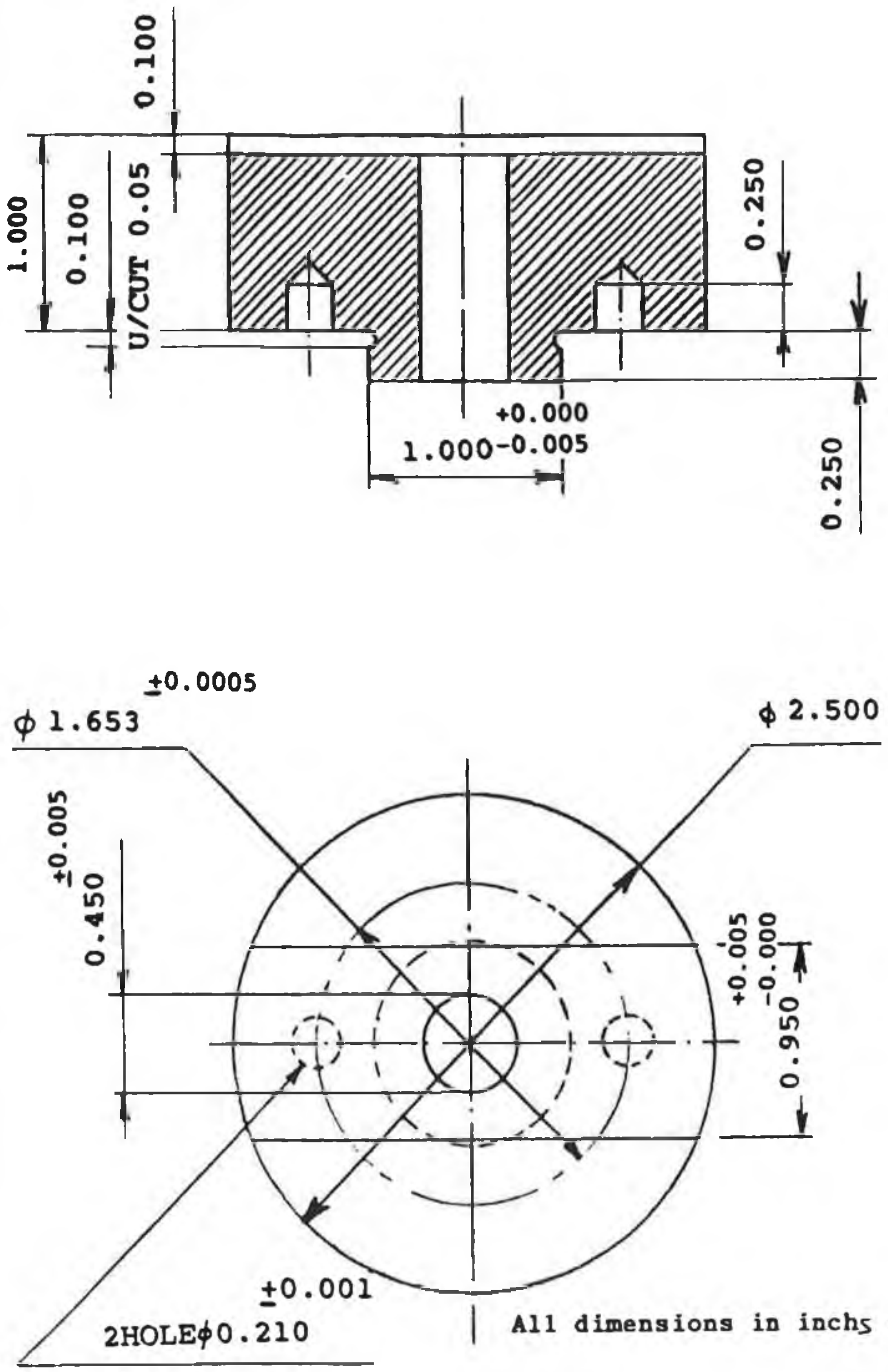


Fig.(2.1) The components of the test joint.



All dimensions in inchs ±0.005

Tool Steel Impax(20).
Heat Treatment 32-34 Rc.

Fig.(2.2) Top uniaxial joint plate geometry.

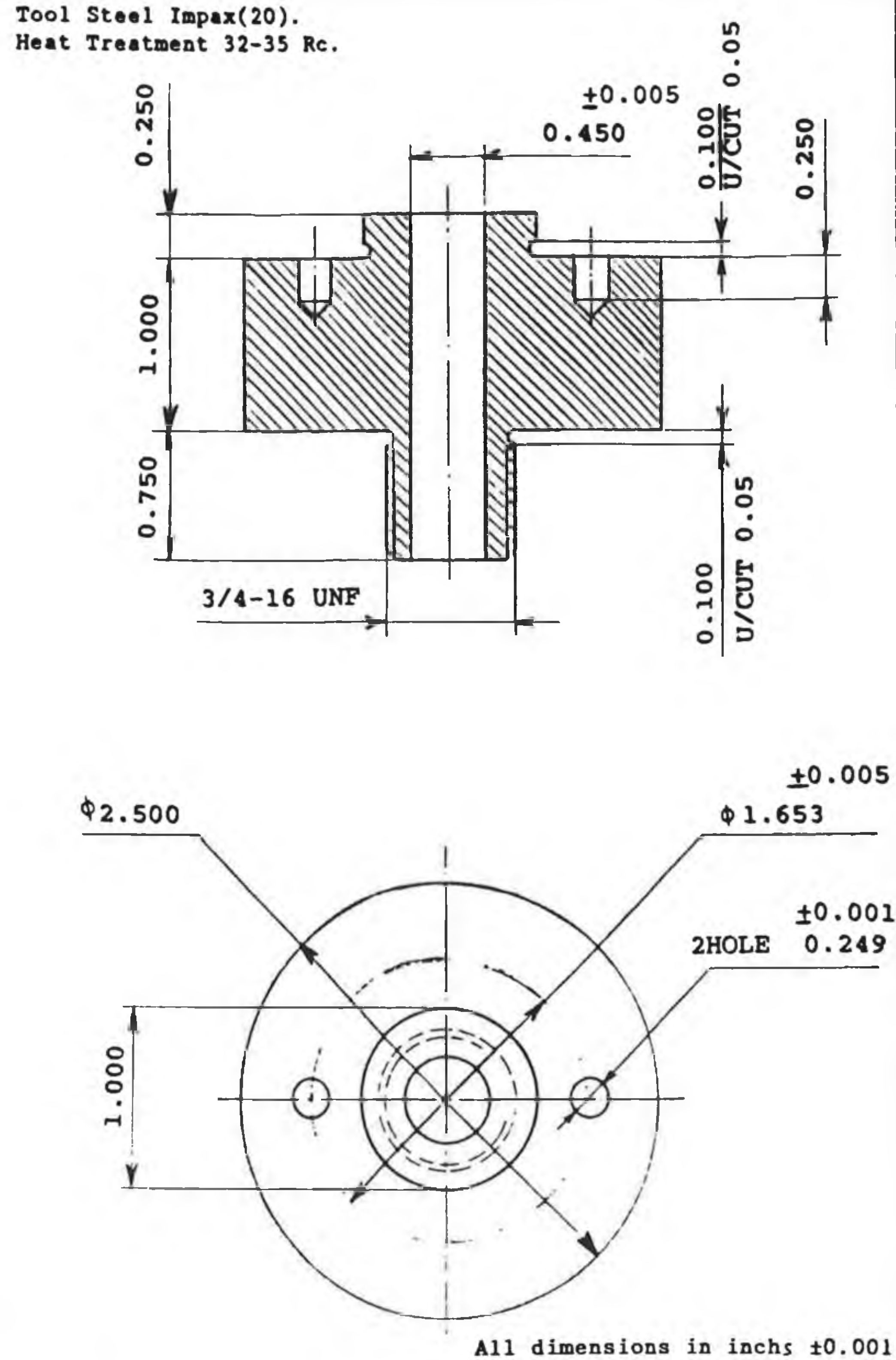
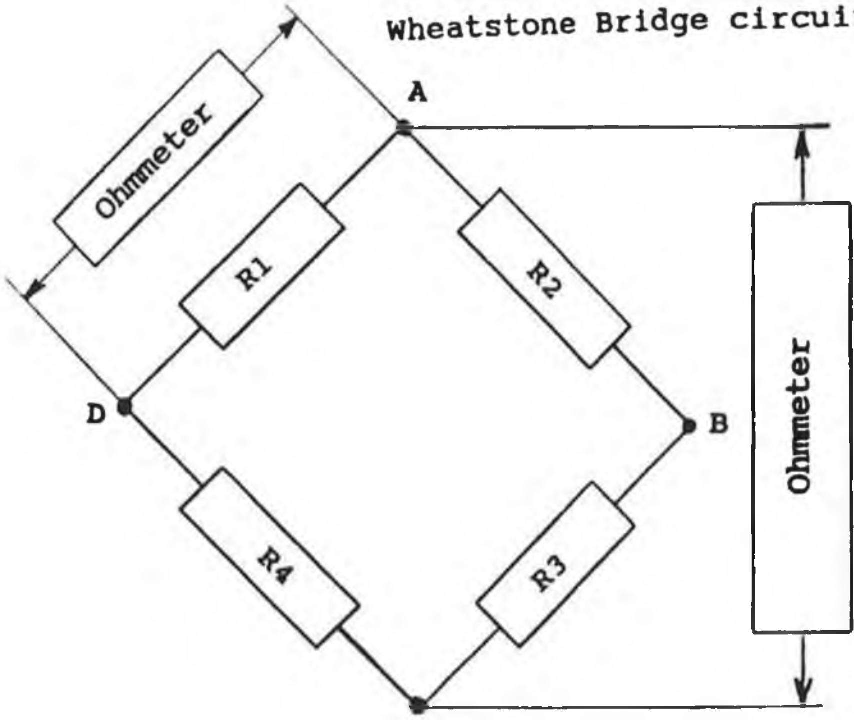


Fig.(2.3) Bottom uniaxial joint plate geometry.

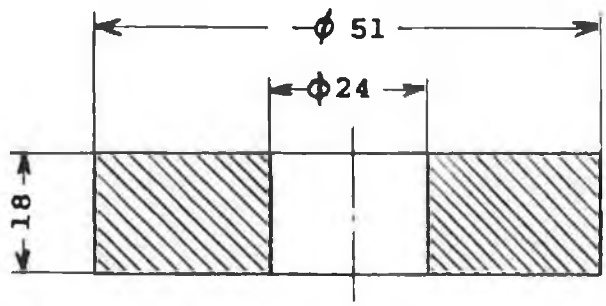
Wheatstone Bridge circuit .



R1,R2,R3,R4 Strain gauge resistance.

A,B, C,D Poles of circuit .

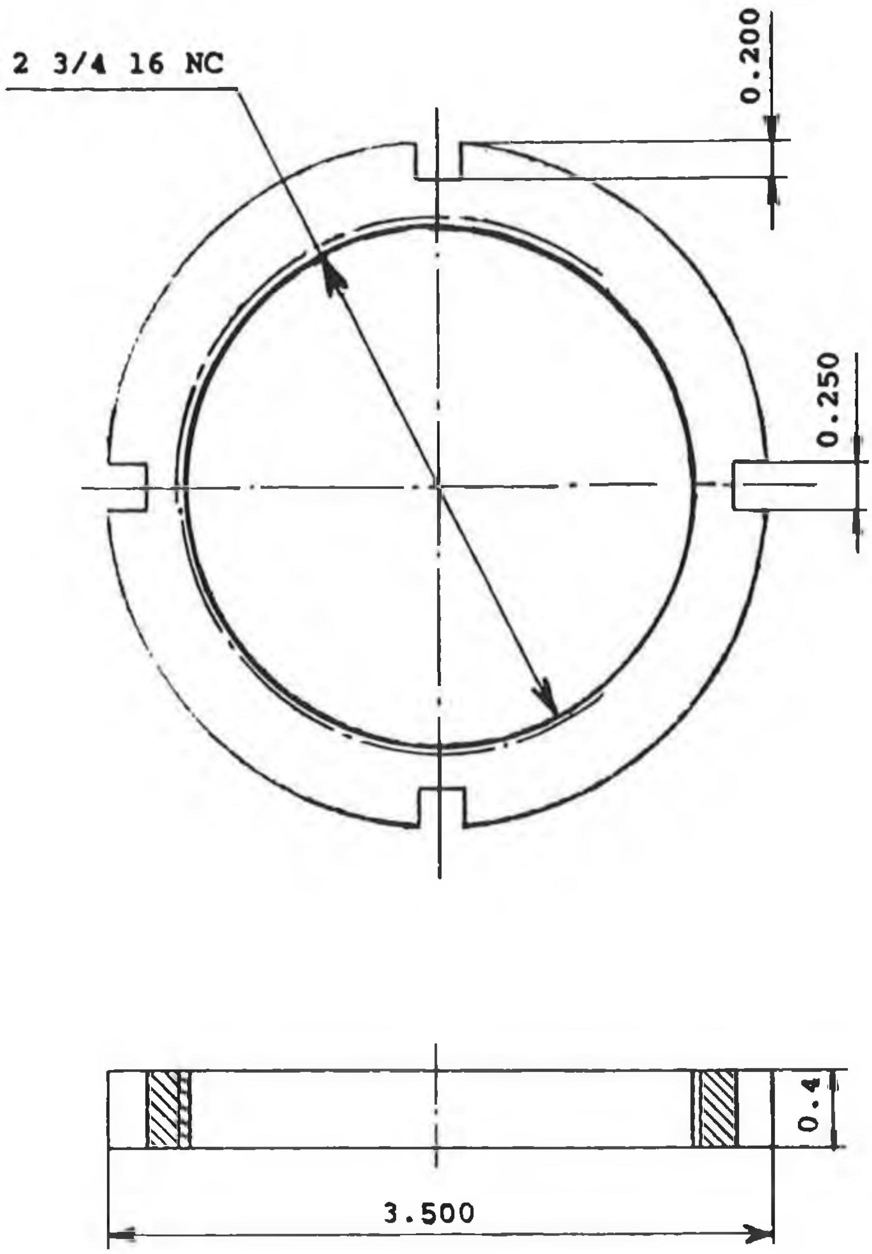
load cell



All dimensions in mm

Fig.(2.4) Dimensions and wiring diagram of load cell .

All dimensions in inchs



Tool Steel Impax(20).
Heat Treatment 32-35 Rc.

Fig.(2.5) Locking Nut.

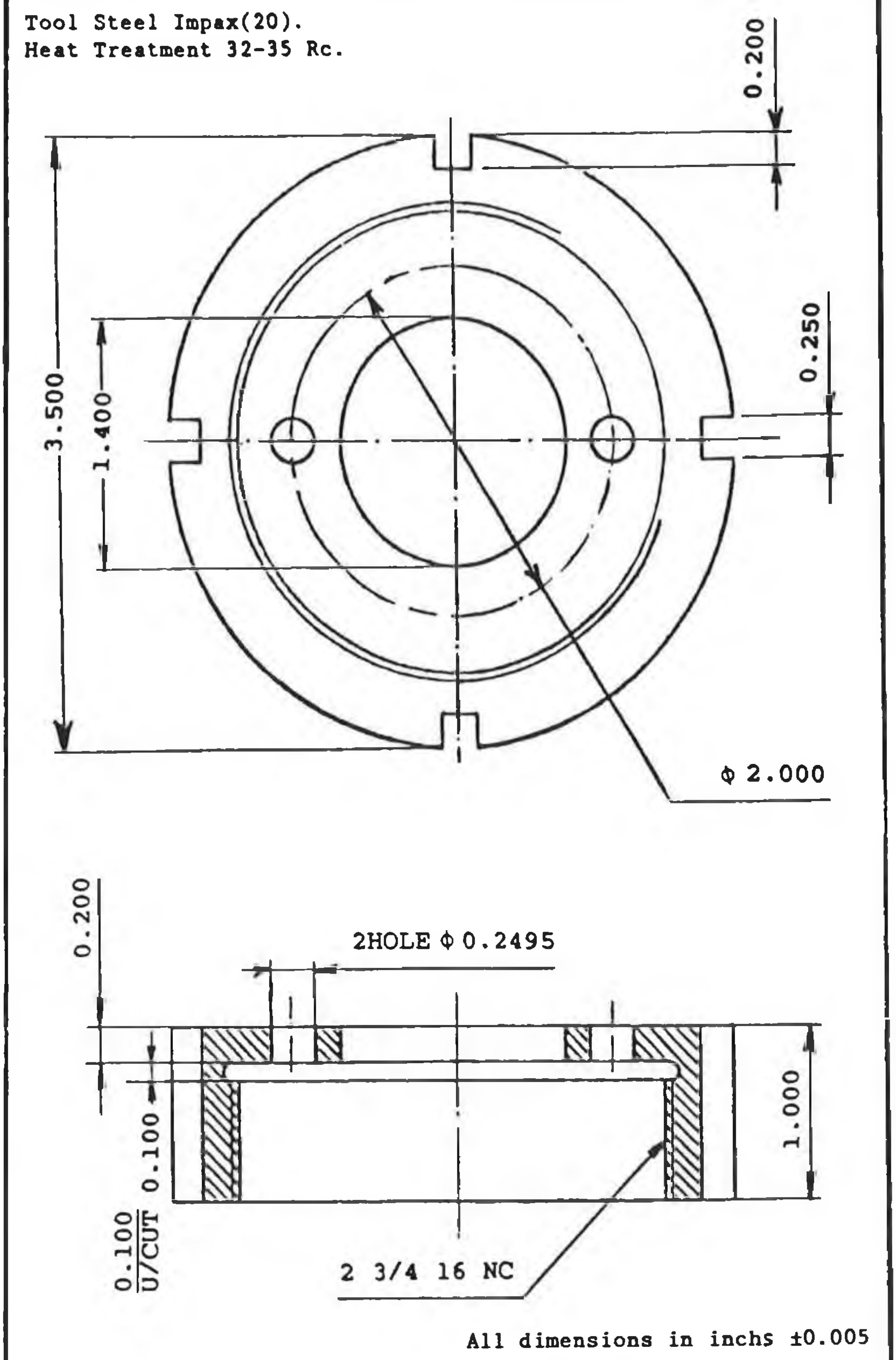


Fig.(2.6) Uniaxial loading plate geometry.

All dimensions in mm

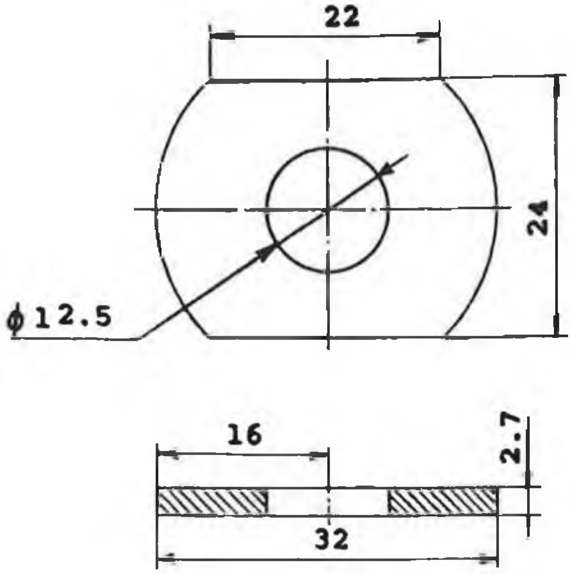
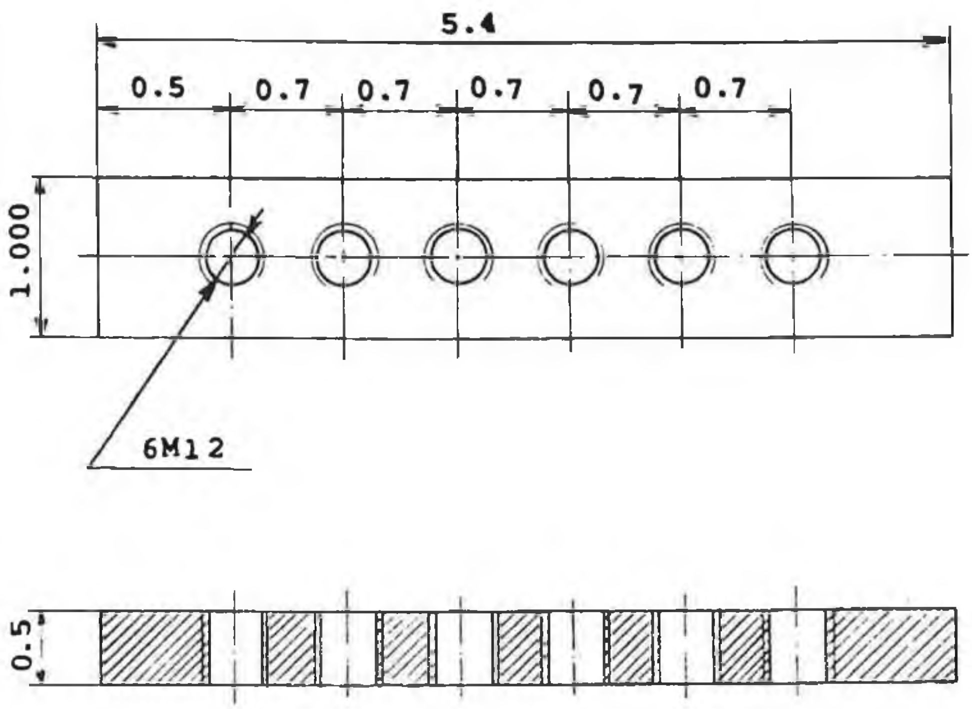


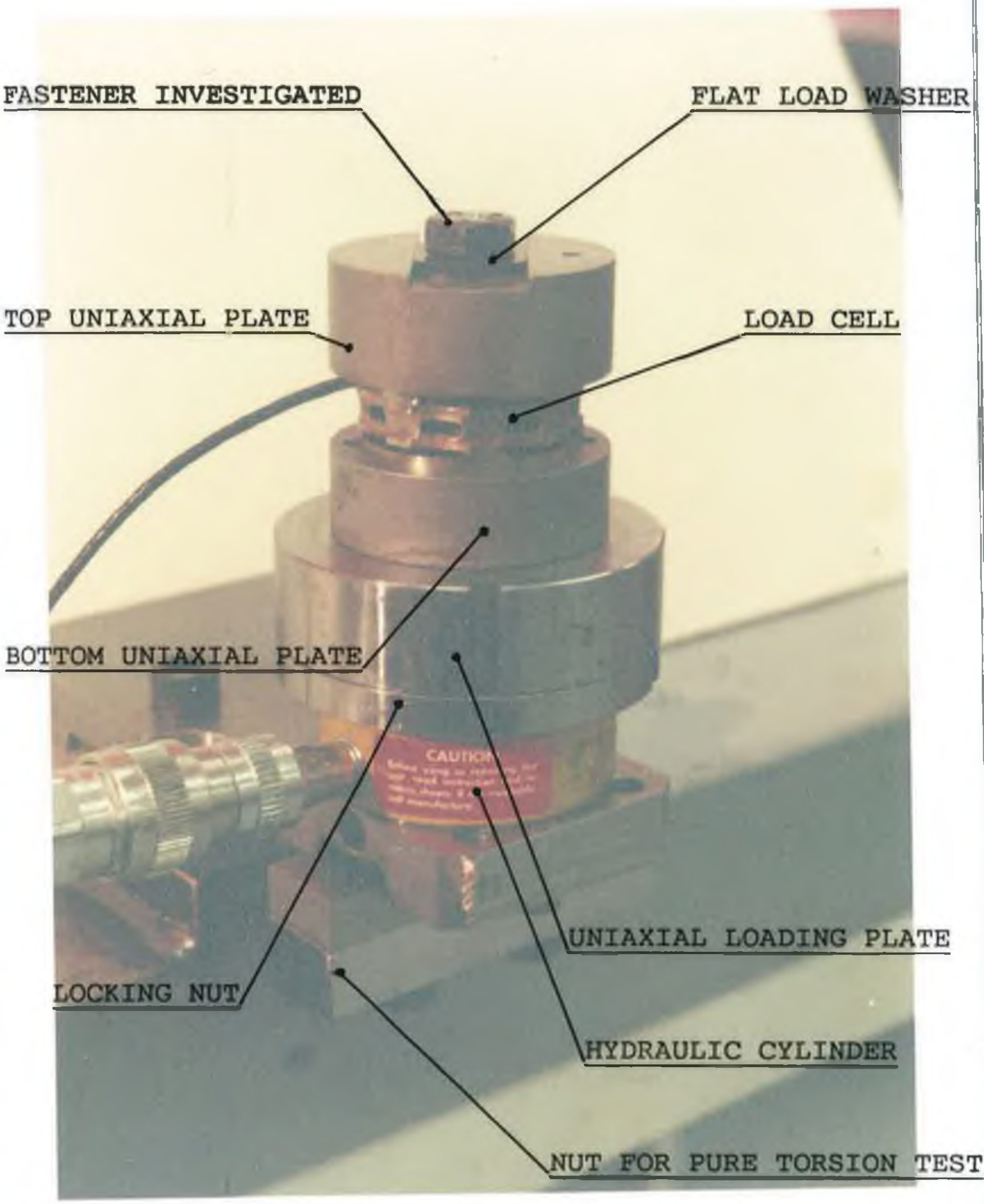
Fig.(2.7) load washer.



All dimensions in inchs

Fig.(2.8) nut.

Plate (2.1) ARRANGEMENT OF TEST JOINT



2.2.2 ARRANGEMENT OF TENSILE RIG

Concentric tensile loads were applied to the concentric joint and fastener by using the hydraulic pump which was connected to the hydraulic cylinder by a high pressure cable, Enerpac K7A, U.S.A. . The test-rig is shown in **Figure (2.9)** and consisted of:

- Fixed plate.
- hand hydraulic pump.
- Instrument of joint with fastener as explained in **section (2.2.1)**.
- The L.V.D.T. .

2.2.2.1 FIXED PLATE

The fixed plate was required to fix the components of the joint. The material chosen was mild steel and this plate was supplied by **S.P.S.**, Laboratories, Ireland. Details of the fixed plate is shown in **Figure (2.10)**.

2.2.2.2 HYDRAULIC CYLINDER PUMP

The hydraulic cylinder pump was used to apply high pressure of up to **700 bar** to obtain external tensile load for testing the fasteners. The model of hydraulic pump was Enerpac PH-39, U.S.A., High pressure. It was supplied by S.P.S. Laboratories, Ireland., see **Plate (2.2)**.

2.2.2.3 THE L.V.D.T.

The **L.V.D.T.** was required to obtain fastener extensions which was fitted to the fastener by using a specially designed extensometer. The main advantages of the Linearly Variable Differential Transformers (**L.V.D.T.**) are:

- High accuracy and linearity.
- No friction need be introduced into the system being measured as the internal mechanism of the **L.V.D.T.** is a Non-contact sensor. A series of Precision Transducers from R.D.P. offers a unique combination of quality, performance, and reliability to satisfy the measurement needs of discerning industrial users.

The **L.V.D.T.** was of type GTX2500 to give maximum displacement equal to **±2.5 mm**. This kind of **L.V.D.T.**has a non-rotating spring-loaded armature running in precision linear ball bearings. This L.V.D.T. was purchased from R.D.P. Electronic Ltd. Wolverhampton, England. The dimensions of the L.V.D.T. and some information are given in **Figure(2.11)**.

Full details of the L.V.D.T.type GTX2500 are given in the **R.D.P.** Electronics Ltd. Catalogue.

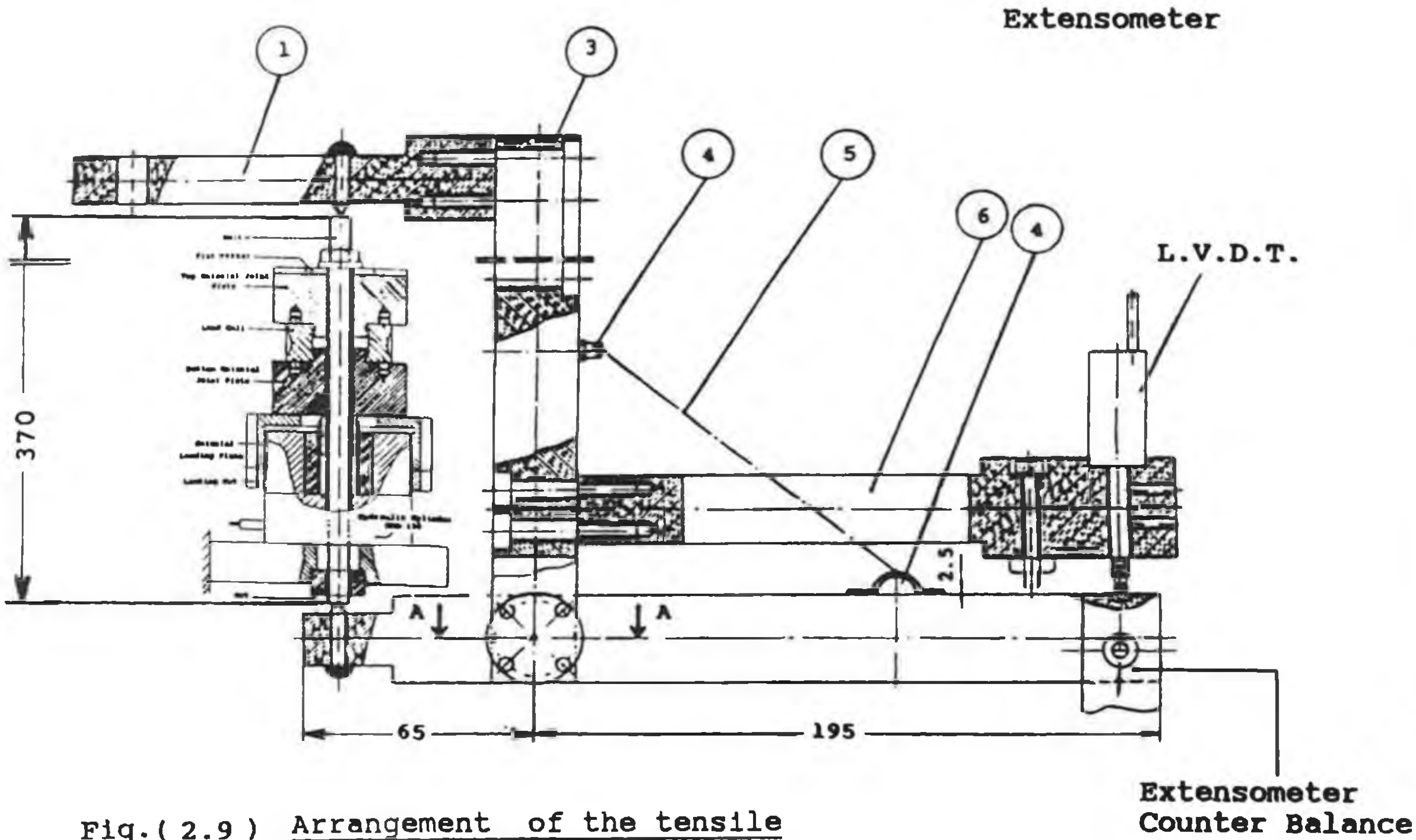
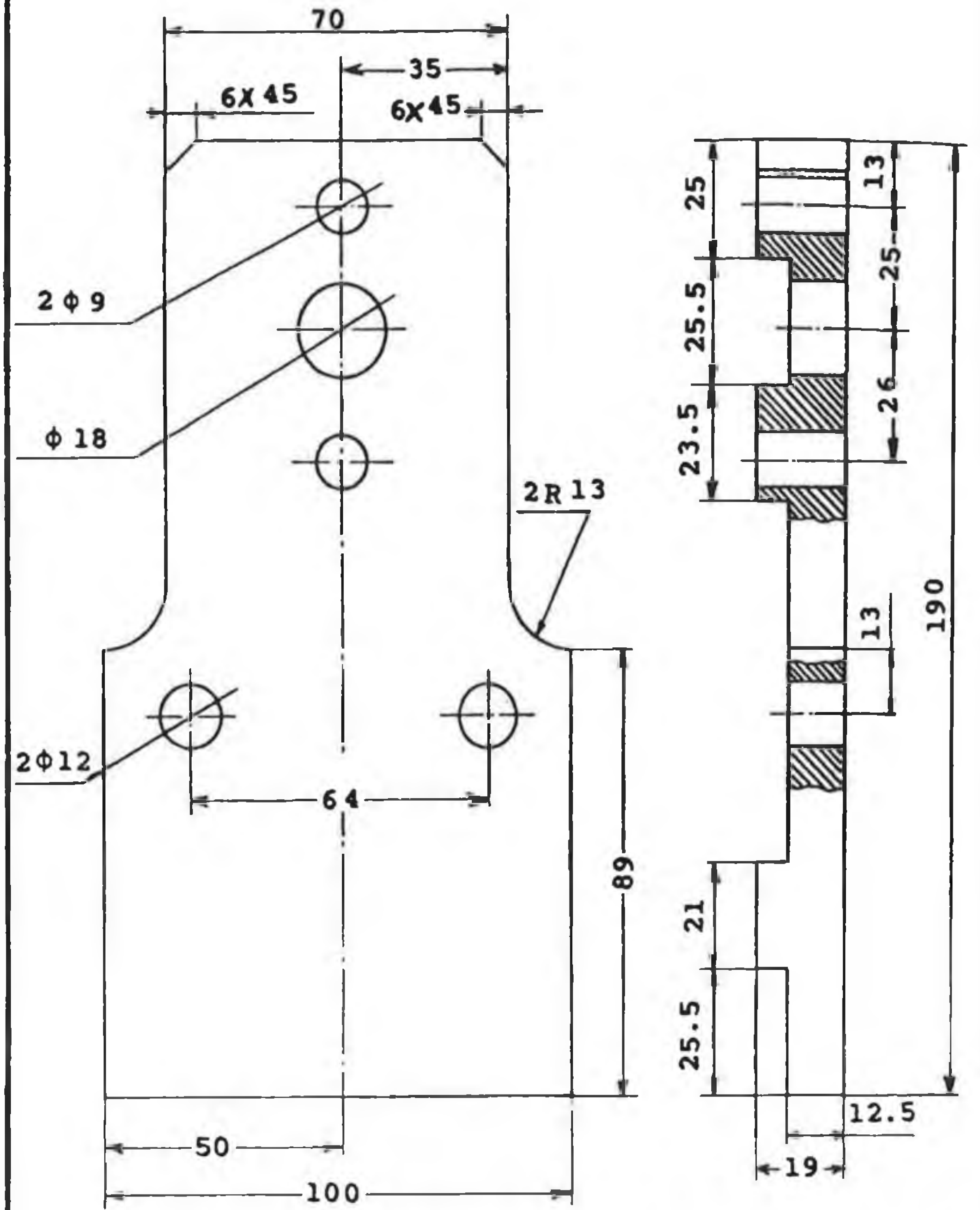


Fig.(2.9) Arrangement of the tensile test rig with extensometer and L.V.D.T. .



All diamensions in mm

Mild steel

Fig.(2.10) Fixed Plate.

TECHNICAL DATA

- Linear stroke ±2.5 mm .
- O/P Voltage 160 mV/V.
- Current L.V.D.T. 16mA(6mA).
- LINEARITY 0.25% Of full stroke.
- Operation temperature -40 C to +100 C.
- Weight (less cable) 18 gm.
- Spring force 118 gm.

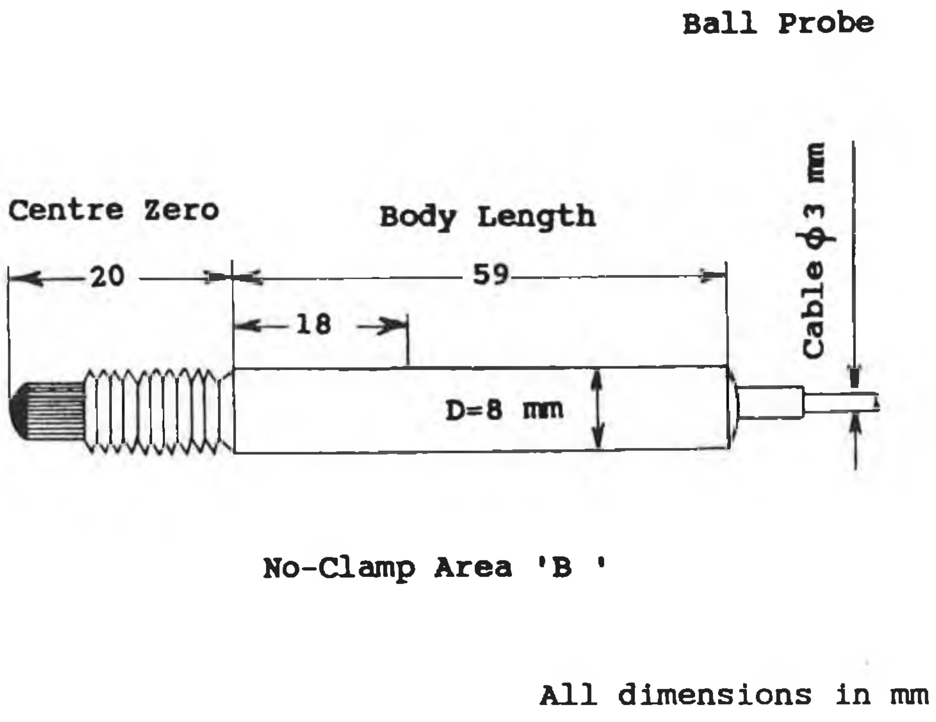
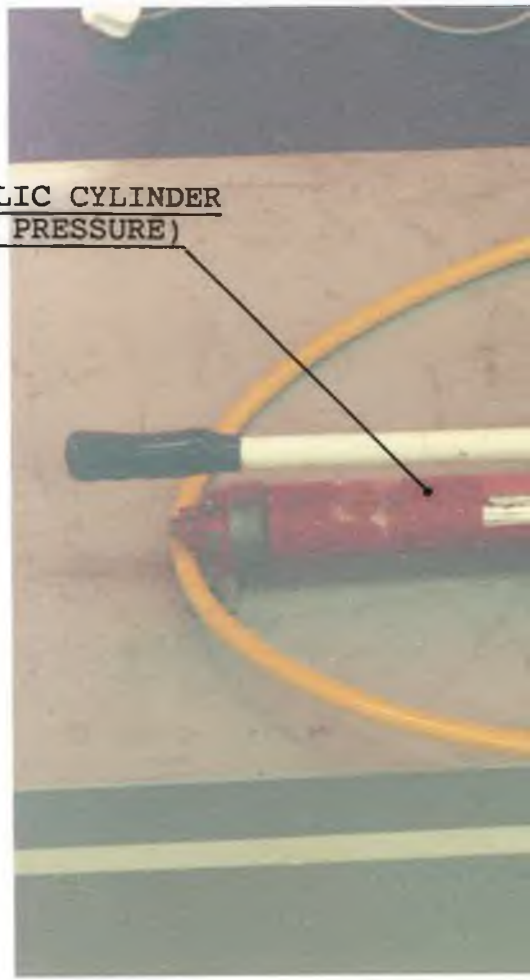
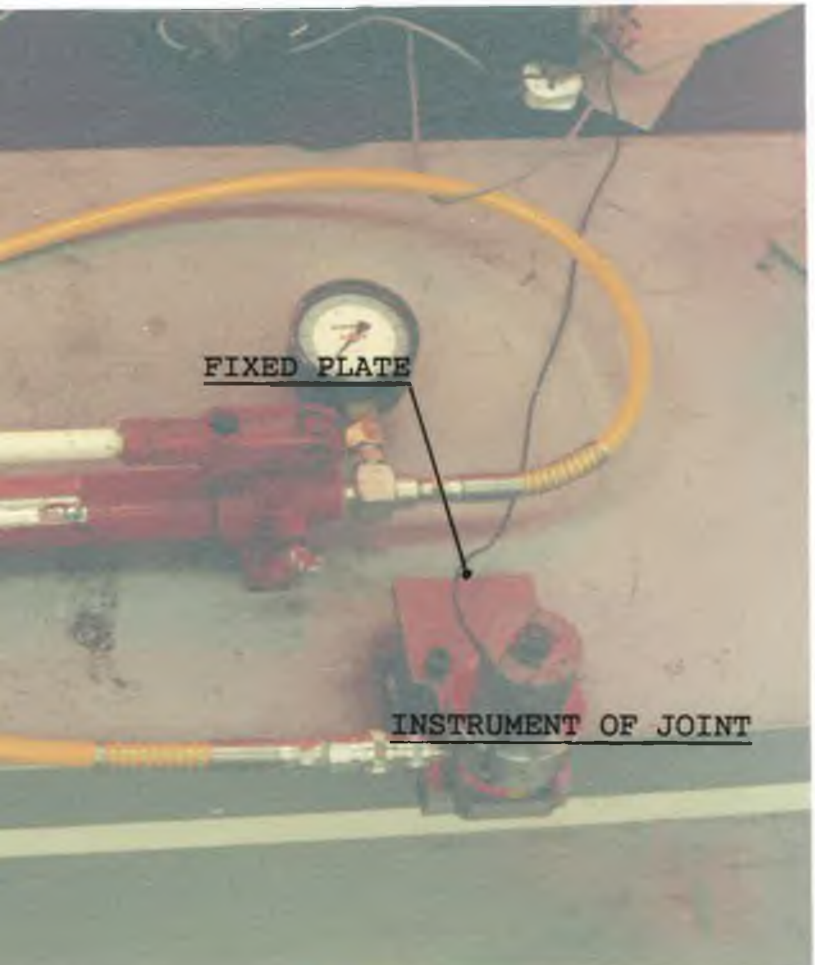


Fig.(2.11) Dimensions of L.V.D.T. Model GTX2500 .

Plate (2.2) **HAND HYDRAULIC PUMP**




FIXED PLATE
INSTRUMENT OF JOINT

2.2.3 THE ELECTRONIC TORQUE WRENCH

2.2.3.1 TIGHTENING METHOD

The quality of the fastener itself is only one of the major parameters in the bolted joint, other factors affecting the reliability of the bolted joint are:

- Determination of the design clamping load required to withstand external load influences.
- Accuracy of the tightening method.

There are several tightening methods to obtain optimum levels of pre-load, these methods include torque control, angle control and the joint control.

2.2.3.2 joint control system

The joint control system (JCS) is employed in the SPS Electronic 200 N.M Torque Wrench to detect the yield point of the fastener material. This is achieved by measuring and controlling the torque and the fastener rotation. The joint control system calculates the shut-off point based on the changes in the tightening torque gradient. It minimizes the effect of frictional conditions and the joint stiffness variations that are major factors in other modes.

The maximum gradient is stored as a reference value, all gradient values established after these are compared with the maximum stored value and used to establish a yield the maximum stored value and used to establish a yield point. When the gradient drops to **50%** of the maximum stored value, shut-off point is achieved and the wrench registers yield [6]. Before tightening, a minimum level of torque called `Snug Torque' is applied. This snug torque ensures that the tightening operation has truly started and that all gaps and non-linearities in the joint are eliminated before the control is activated.

2.2.3.3 <u>OTHER CONTROL SYSTEM METHODS</u>

The **SPS** Electronic Torque Wrench has four separate tightening levels or methods, one of these is selected before operation.

1. <u>TORQUE CONTROL</u>

The torque control method indicates when a preset torque level is reached, this method of tightening is greatly influenced by friction conditions of the fastener and has been improved by using the **(JCS)** mode [6,39].

2. <u>ANGLE CONTROL</u>

The angle control method indicates when a preset angle of relative rotation occurs between the fastener and the nut. An angle reference arm is attached to the wrench to give a reference position for angle measurement [6]. This method of operation overcomes the dominating influence of friction by taking the fastener into its plastic range. Again the tightening operation has been improved by using **(JCS)** mode.

3. OFF-TORQUE

The off-torque measurement records the peak value of off-torque during untightening of the fastener, the magnitude is displayed on the digital readout of the wrench.

2.2.3.4 THE SPECIFICATIONS OF THE ELECTRONIC TORQUE WRENCH

The **SPS** electronic **200 N.M** hand torque wrench is used to tighten fasteners. This is achieved by measuring and controlling the torque and the fastener rotation, see **Plate (2.3)**. The general specifications for operating methods of the SPS electronic **200 N.M** hand torque wrench are as follows, see the **Figure (2.12)**.

A. MEASUREMENT AND DISPLAY

A.1 THE TORQUE

- Torque units : Nm or Lb.ft.
- Maximum torque : 170 Nm or 125 Lb.ft.
- Torque accuracy : ±1% full scall .
- Torque resolution : ±1% Nm or +1% Lb.ft.

A.2 THE ANGLE

- Angle units : degree.
- Maximum angle : 510 degree.
- Maximum turning angle : 210 degree per ratchet sweep.
- Angle accuracy : ±1% of reading.
- Angle resolution : 1 degree.

B. <u>PHYSICAL</u>

- Length : 665 mm or 26 inches.
- Weight : 2.3 Kg or 5 Lb.
- Square drive : 12.5 mm or 0.5 inches.

C. <u>BATTERY</u>

- Operating time : 8 hours.
- Recharge time : 10 hours.
- Automatic power down : Turns wrench off after one minute of inactivity.
- Operating temperature range : 0 - 50 $^{\circ}$C or 32 - 122 $^{\circ}$F.
- Battery charger input voltage : 110/220 VAC (selectable).

Full details of the wrench operation and other information are given in the manual of the SPS Electronic Torque Wrench.

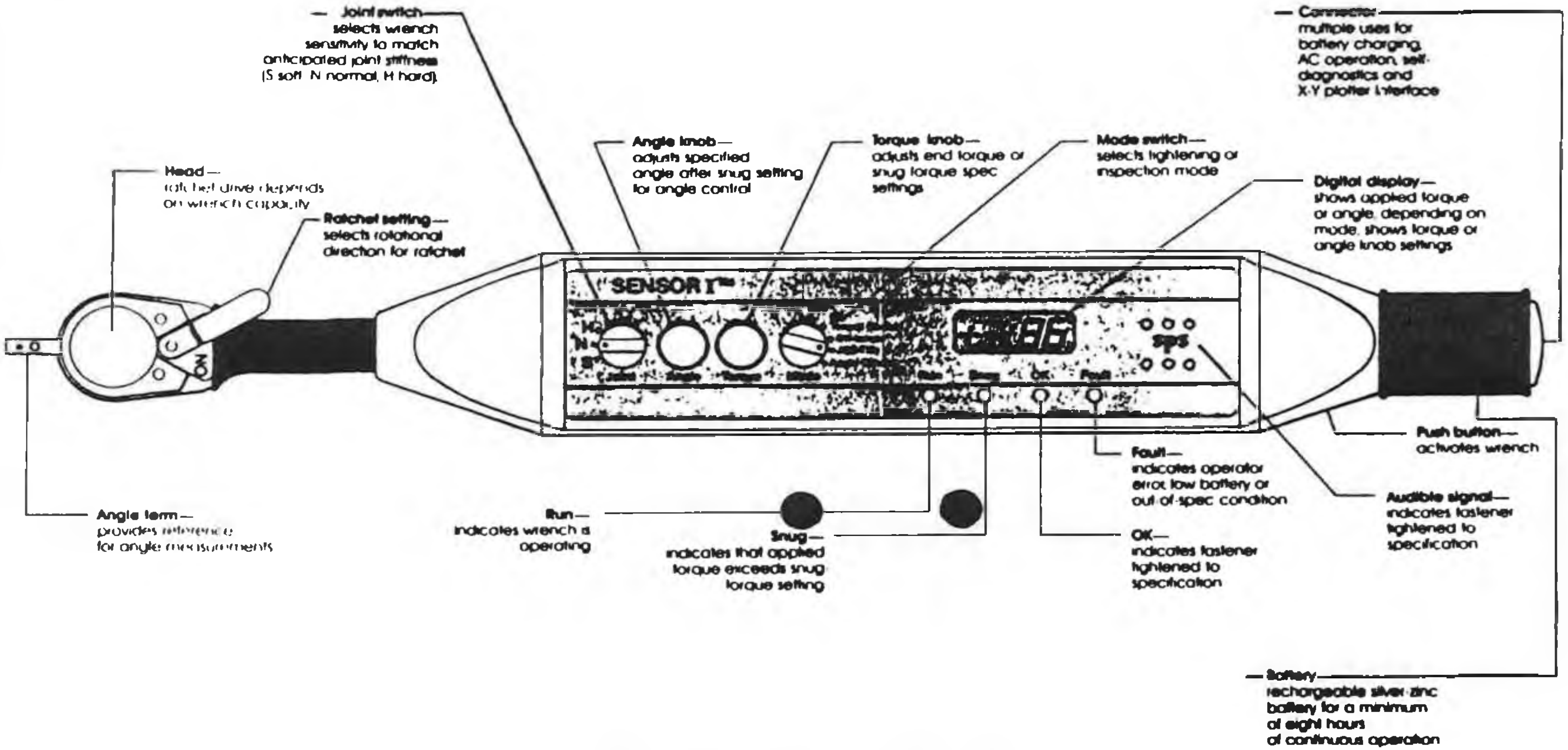
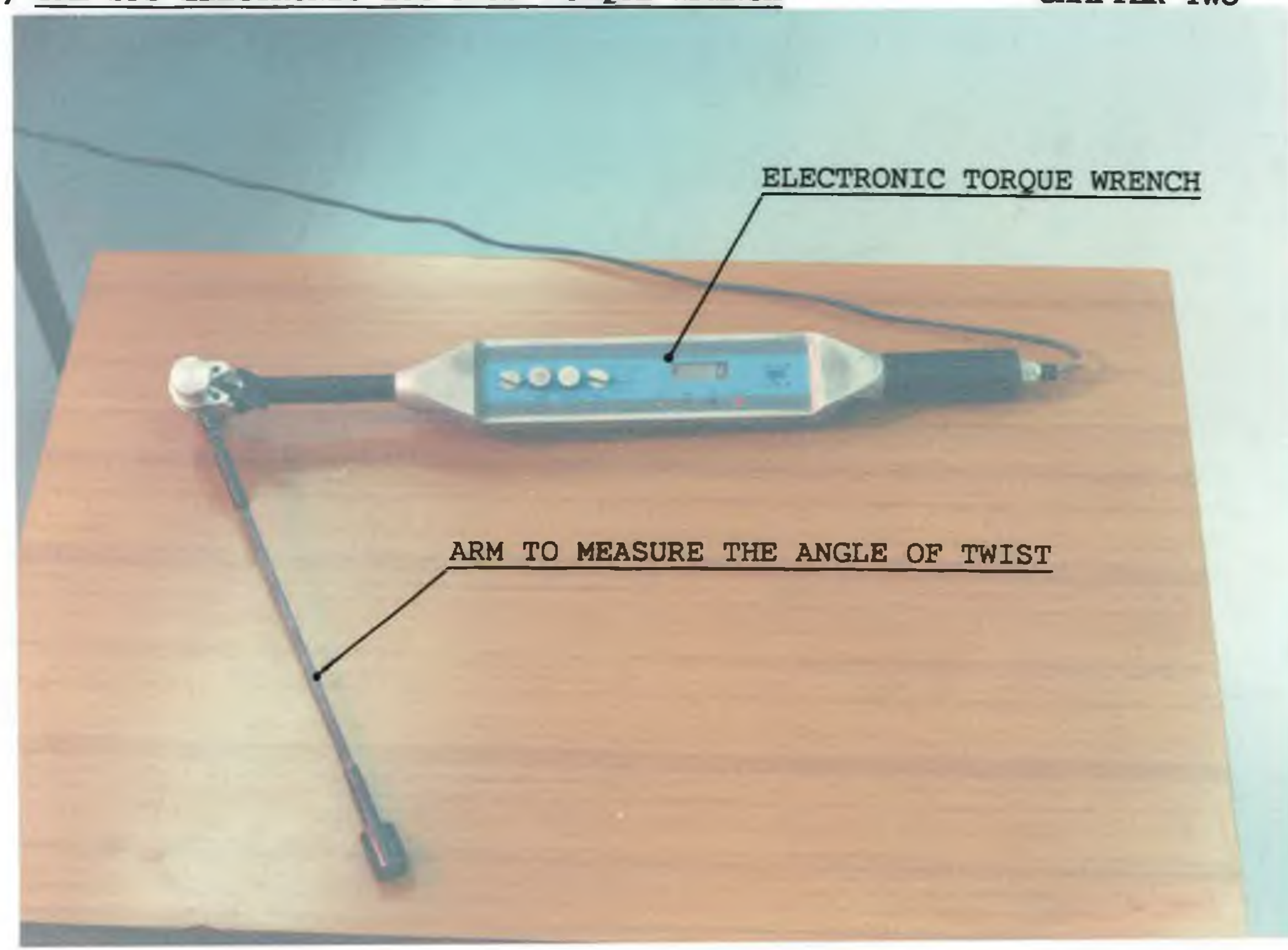


Fig.(2.12) S.P.S. Sensor 1 200 NM
The electronic torqe wrench

Plate (2.3) THE SPS ELECTRONIC 200 HAND TORQUE WRENCH



2.2.4 INSTRUMENTATION

The signal produced by the load cell within the joint was amplified by a Digital Transducer Indicator series **E307-2** to measure the load, and transfer to a **X-Y** recorder.

The bolt extension under loading was measured with the aid of a Linearly Variable Differential Transformer `LVDT'. The signal was amplified by a Digital Transducer Indicators series **E307** to measure the bolt extension and transfer to a **X-Y** recorder, as shown in **Plate (2.4)**.

The Transducer Indicators and L.V.D.T. were purchased from R.D.P. Electronics Ltd., Wolverhampton, England. Each instrument supplied had a digital output with optional output suitable for **X-Y** recorders. The **X-Y** recorder used was a Linseis Model **LY 17100**, and allowed permanent records of both the clamp load and extension to be made.

Figure(2.13) shows a schematic diagram of the tensile rig and instrumentation, see **Plate (2.5)**.

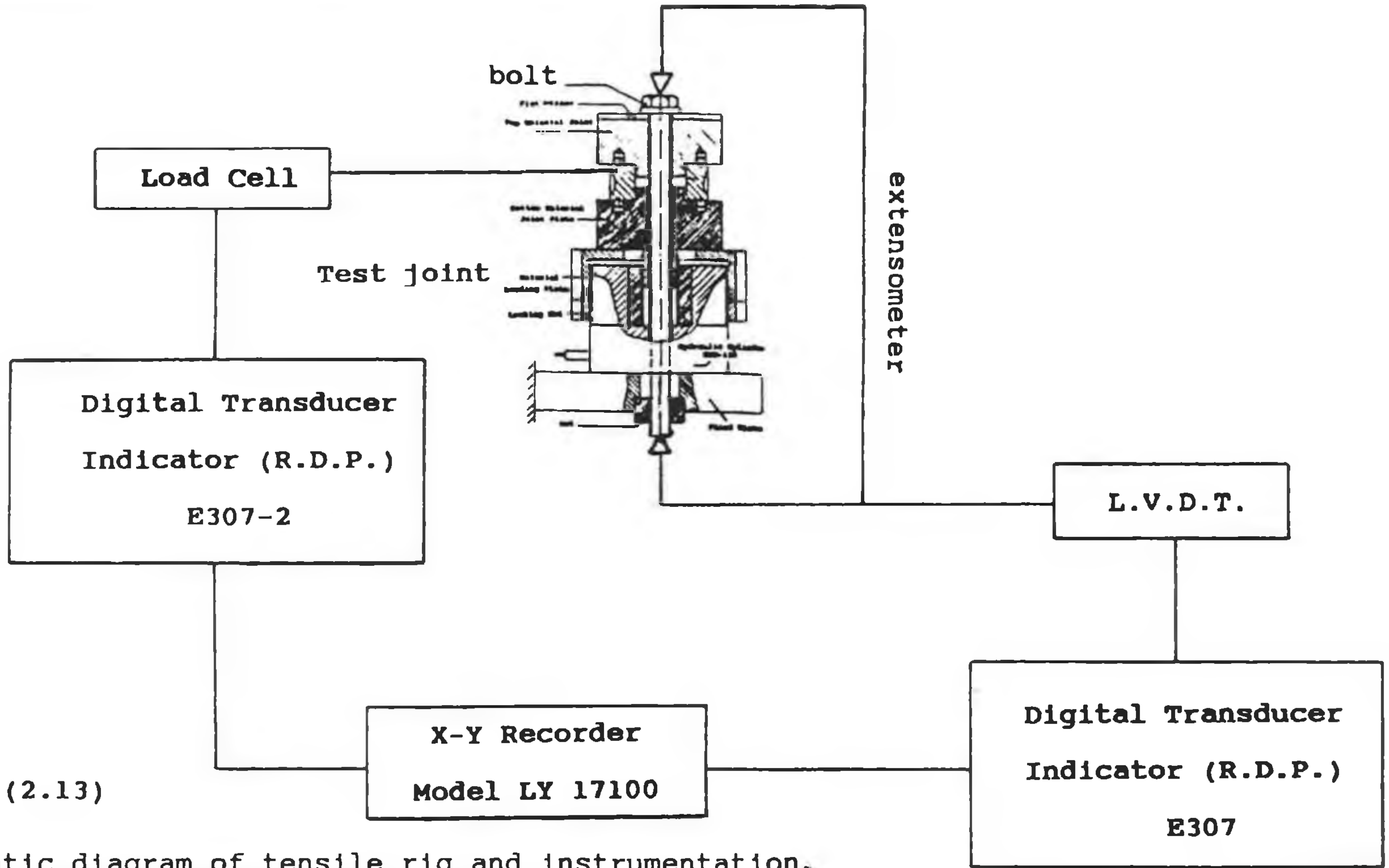


Fig.(2.13)

Schematic diagram of tensile rig and instrumentation.

Plate (2.4) DIGITAL TRANSDUCER INDICATOR PASSED TO X-Y RECORDER

DIGITAL TRANSDUCER INDICATOR
(E307 AND E307-2)

X-Y RECORDER

L.V.D.T.

LOAD CELL

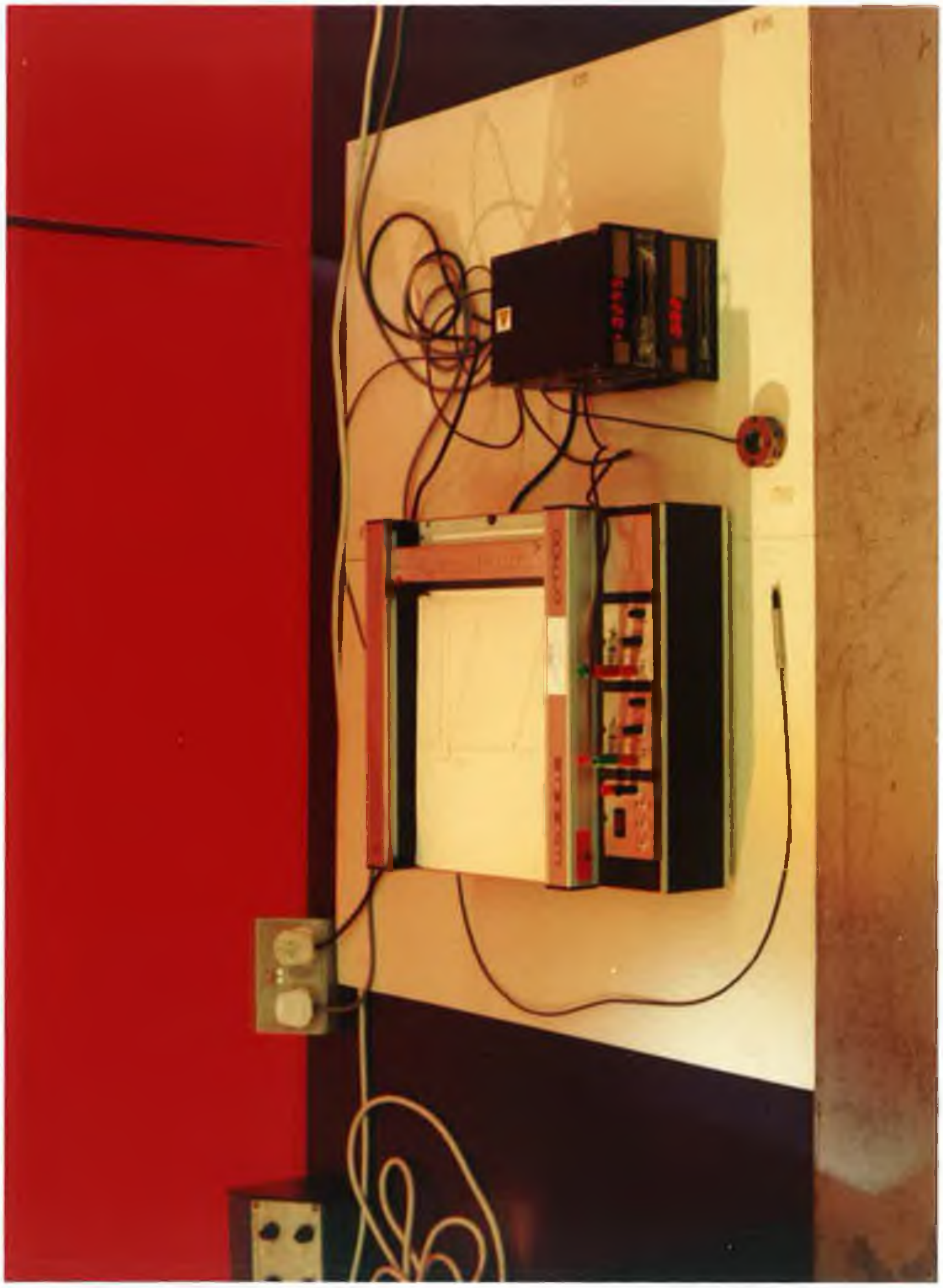


Plate (2.5) Arrangement of tensile rig with instrumentation

AMPLIFIER MODEL E307 & E307-2

X-Y RECORDER

BLOCK TO FIX THE SPANNER

EXTENSOMETER WITH L.V.D.T.

INSTRUMENT OF JOINT

HAND HYDRAULIC CYLINDER

2.3 Additional equipment

2.3.1 SPANNER AND BLOCK

During the experiments, it was impossible to use the electronic torque wrench to apply torque to the fastener, due to the extensometer was attached to the bolt. Therefore, a spanner was used. In order to hold the torque constant during the experiments, a block was necessary for this purpose.

2.3.2 STRAIN GAUGE

2.3.2.1 MEASUREMENT OF TORQUE USING STRAIN GAUGES

The measurement of the shear strain on the bolt surface permitted to determine the torsional stress and the corresponding torque in the bolt. The procedure relies on the principle that when a torsional shear stress is applied to the body of a bolt the shear strain can be measured by attaching a ` V ' shaped strain gauge on the surface at 45° angle to either side of the longitudinal axis of the bolt.

The response takes place in a push-pull fashion in a two-active-arm bridge to determine the shear strain. Special strain gauges are produced for this purpose and a typical one used in these tests is called torque gauge, type ` EA-06-062TV-350 '. The resistance of this strain gauge is equal to 350 Ohm at 24°C. The strain gauges were purchased from Welwyn Strain Measurement Ltd., (WSM), England [43].

2.3.2.2 STRAIN GAUGE BRIDGE

Because the changes in the resistance of a strain gauge is a small and the base resistance is high, it is normal practice to use bridges to measure the changes accurately. The strain gauge which was used in the experiments, had a Wheatstone bridge circuit arrangement. It consisted of one torque strain gauge set which was used as two active gauges and two other resistances each having the same resistance of **350 Ohm.** Using a two-active-arm bridge methode, gives an increase in the resistance of one arm and a decrease in the adjacent arm. Both effects unbalancing the bridge indicates that the torsional strain was applied to the bolt. The circuit described consisted of four-arm bridge which is called full bridge for measuring shear strain **[44,45].**

2.3.3 STRAIN GAUGE AMPLIFIER

The signal produced by the strain gauge was amplified by a Strain Gauge Amplifier series **2300** system to measure the torsional stress and transfer to a **X-Y** recorder.

The strain gauge amplifier is a purpose designed hybrid, low noise, low drift, linear dc amplifier. It is specifically configured for resistive bridge measurement at high speed and low gains. **Figure (2.14)** shows a schematic diagram of the strain gauge circuit with the strain gauge amplifier and recorder.

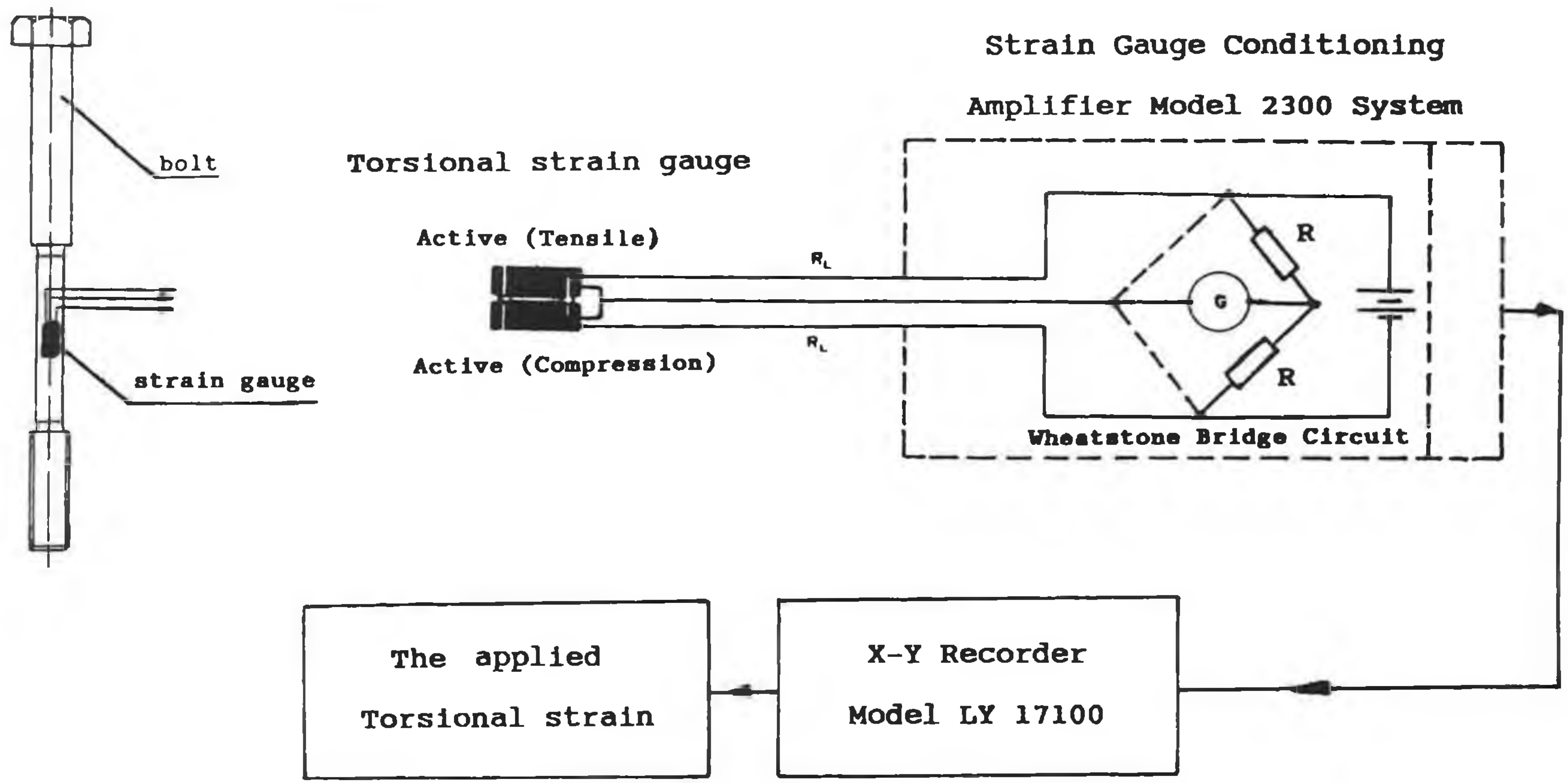


Fig.(2.14) Schematic diagram of the strain gauge circuit with a strain gauge amplifier and recorder.

2.4 Design of the instrumentation

2.4.1 DESIGN OF THE EXTENSOMETER

An extensometer was designed to measure the changes in the bolt length by using L.V.D.T. which was fitted to the designed extensometer.

In **Figures(2.15) and (2.16)** is shown the assembly drawing of the designed extensometer which consists of a lever mechanism with extension spring and the L.V.D.T. connected to the fastener. There are two advantages of the extensometer designed inhouse:

- The extensometer might be used for different lengths of the clamped parts, the maximum length of the clamped parts which can be used in this extensometer is approximately **370 mm** .
- It is capable to detecting small displacements, because the readings of the displacement is given by the output of a L.V.D.T. are the magnified displacements of the fastener. This magnification factor was equal to 3.

The extensometer was made of aluminium alloys in the workshop of the school of mechanical & manufacturing engineering in DCU, Dublin. Since the material was not a significant factor in the design of the extensometer. Aluminium alloy was chosen to obtain lighter weight. Full details of the extensometer are shown in **Figures (2.17) up** to **Figure (2.27)**, also see **Plate (2.6)**.

2.4.2 DESIGN OF THE SPECIAL NUT

It was very difficult to apply pure torsion to a bolt head without any initial load in the bolt, by using the instrumentation of tensile test rig, see **Figure (2.1)** because the initial load in the fastener is generated by rotating the nut, when the torque was applied.

The idea was to use the instrumentation of tensile test rig, to apply pure torsion to the fastener, instead of using a torsion test machine which would not be suitable for the test joint. It was, therefore, decided to use the instrumentation of tensile test rig, for pure torsion test, with some modification.

One of the best solution was to make a simple design such as a nut, see **Figure (2.28)**. This special nut was designed to use in the tensile rig to apply pure torsion. This nut was prevented from rotation due to the torque applied to the bolt head. This special nut was made in a robust manner so that there was negligible deflection when axial load was applied after the application of pure torsion.

2.4.3 SIMPLE DESIGN FOR TORSION TEST

To make sure that no rotation in the threaded section of the bolt takes place during the torsion test, a simple device was required. As shown in **Figure (2.29)**, the test specimen was made without any thread in the fixed end, see Plate (2.7).

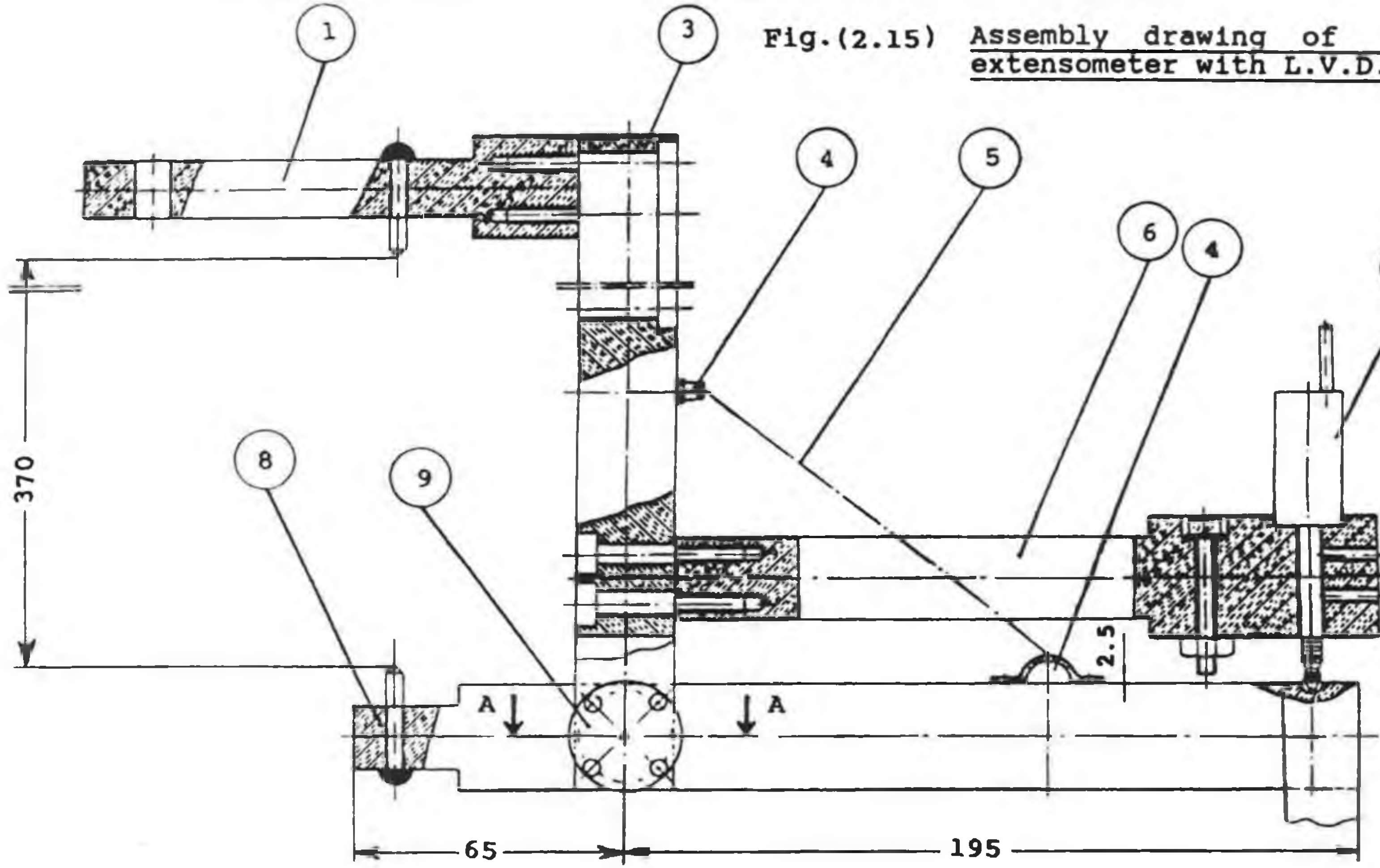


Fig.(2.15) Assembly drawing of the extensometer with L.V.D.T..

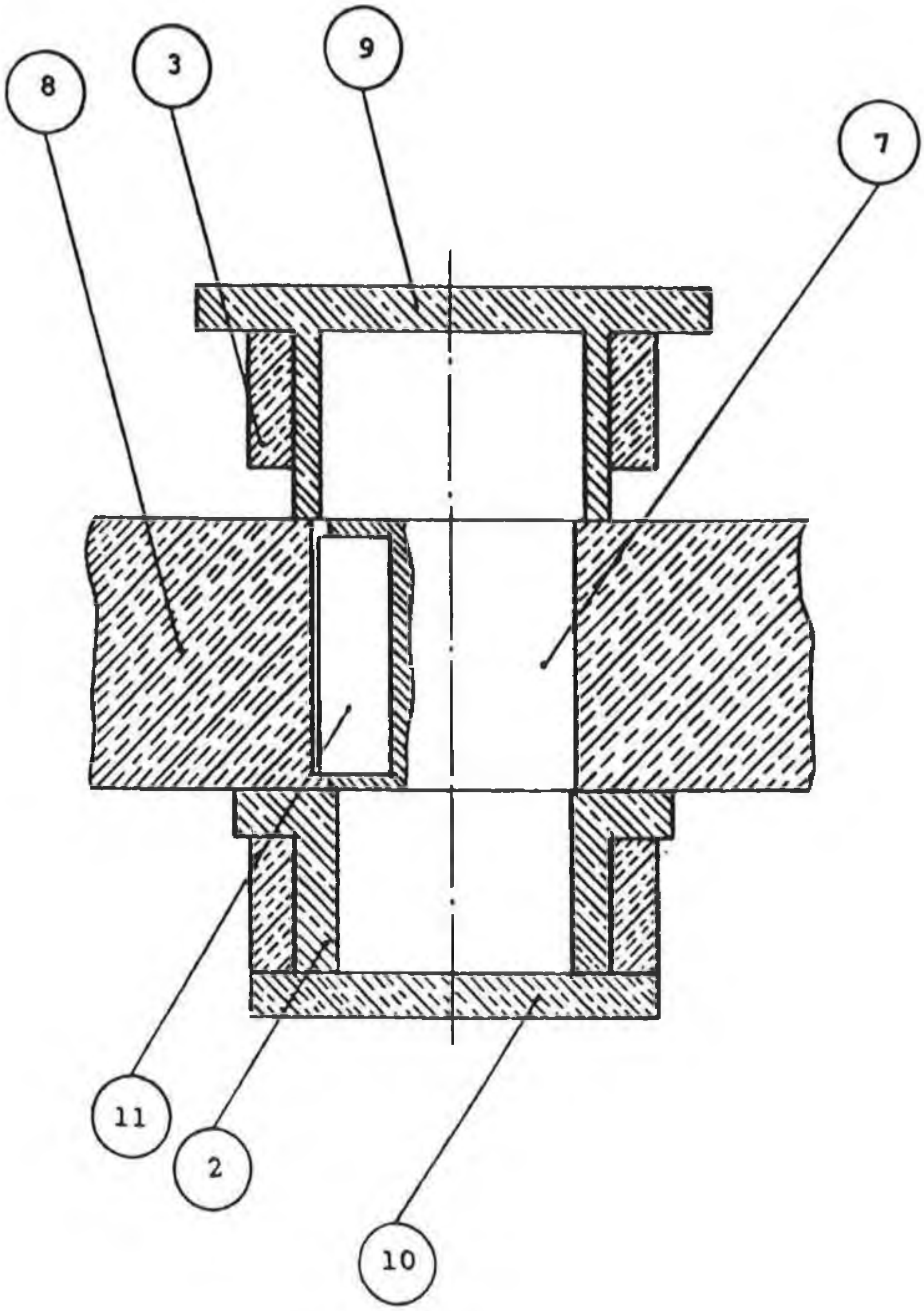
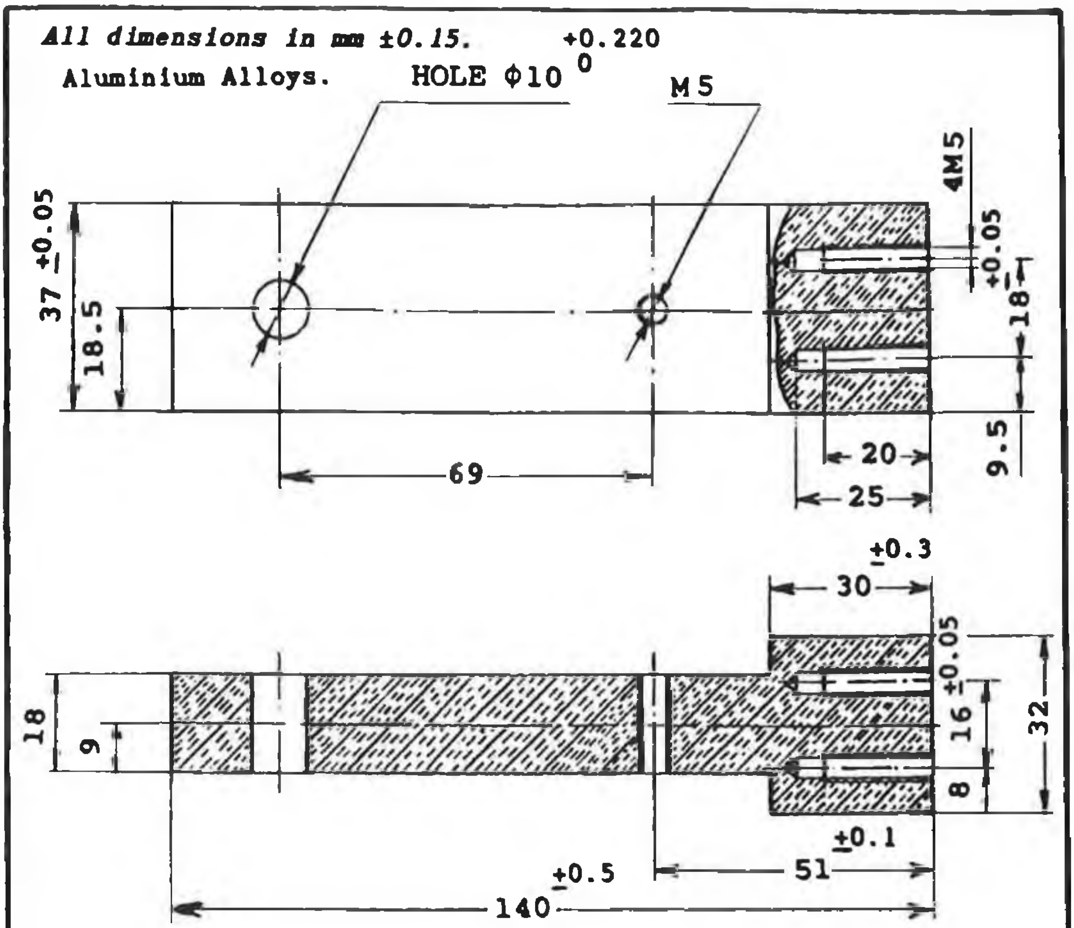
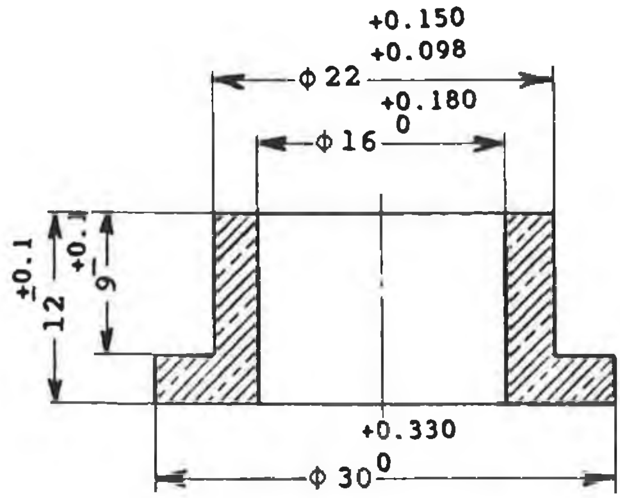


Fig.(2.16) Section of the assembly drawing of the extensometer at A-A .



Drawing NO. 1

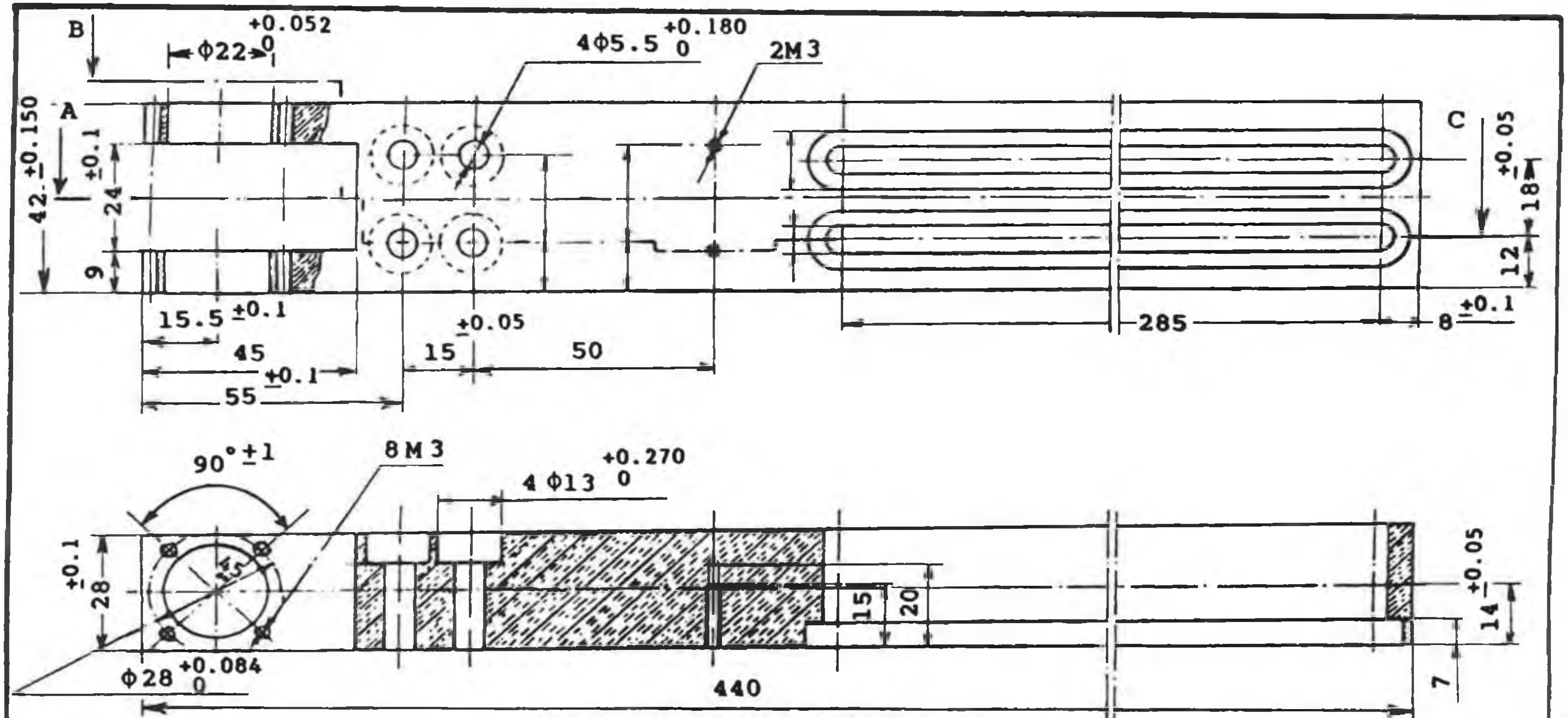
Fig.(2.17)



Drawing No. 2

All dimensions in mm
Brass.

Fig.(2.18)



Section A-C AND B-C

drawing No. 3

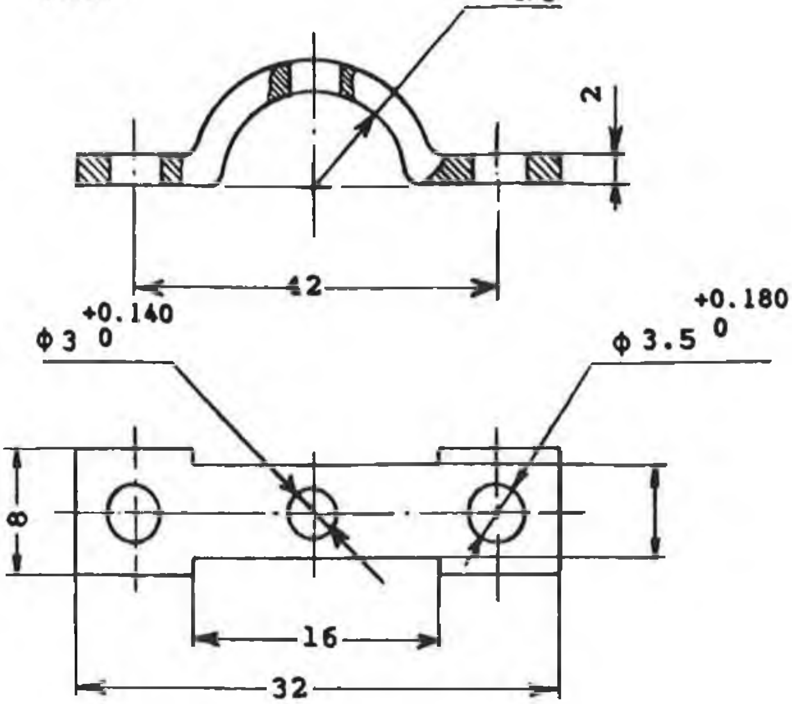
Fig.(2.19)

All dimensions in mm ±0.20.

Aluminium Alloys.

All dimensions in mm ±0.10.

Mild Steel.

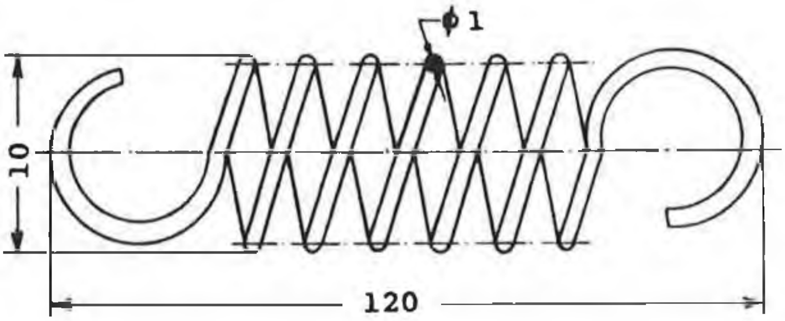


Drawing No. 4

Fig.(2.20)

Force =10 N .

Diameter of wire =1 mm .

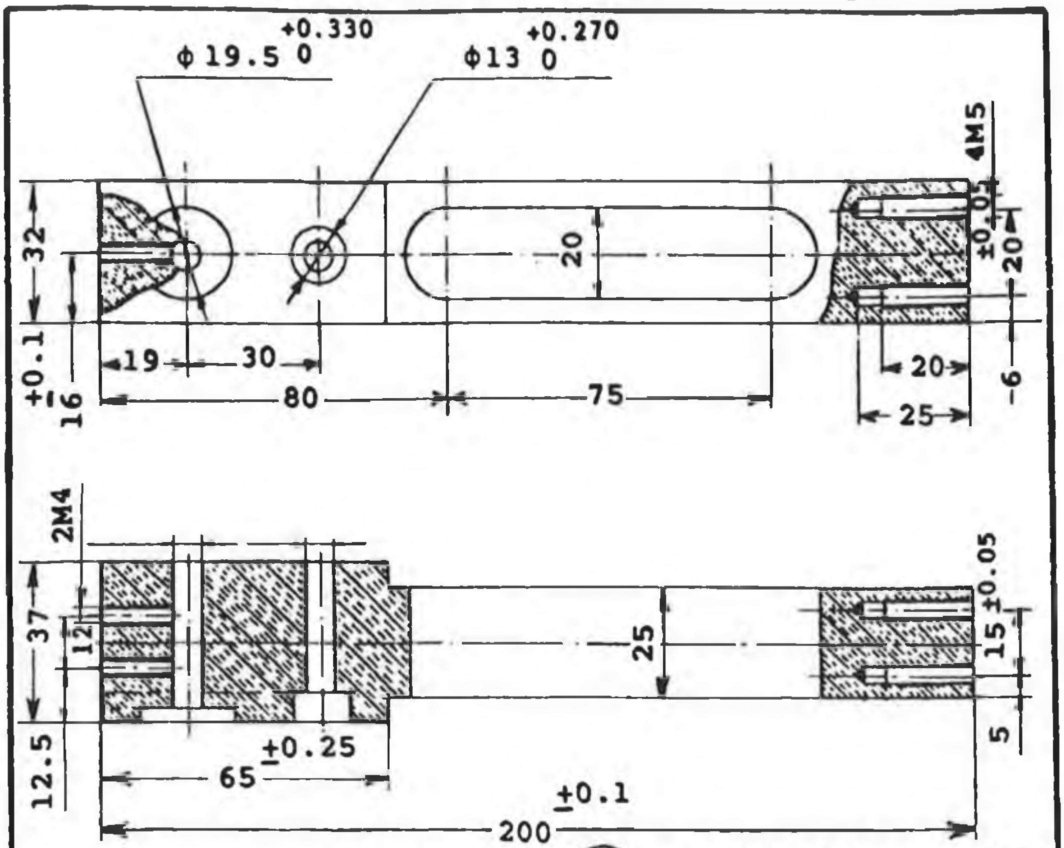


Drawing No. 5

Fig.(2.21)

All dimensions in mm

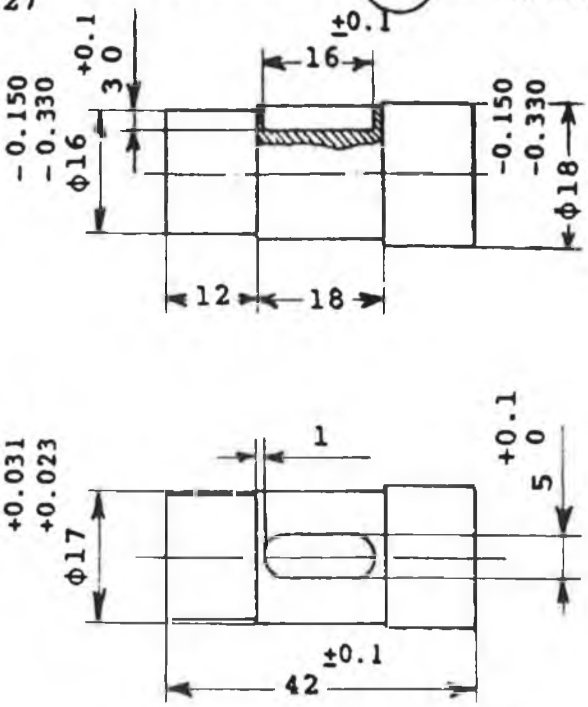
Steel.



Drawing No. 6

All dimensions in mm ±0.20.
Aluminium Alloys.

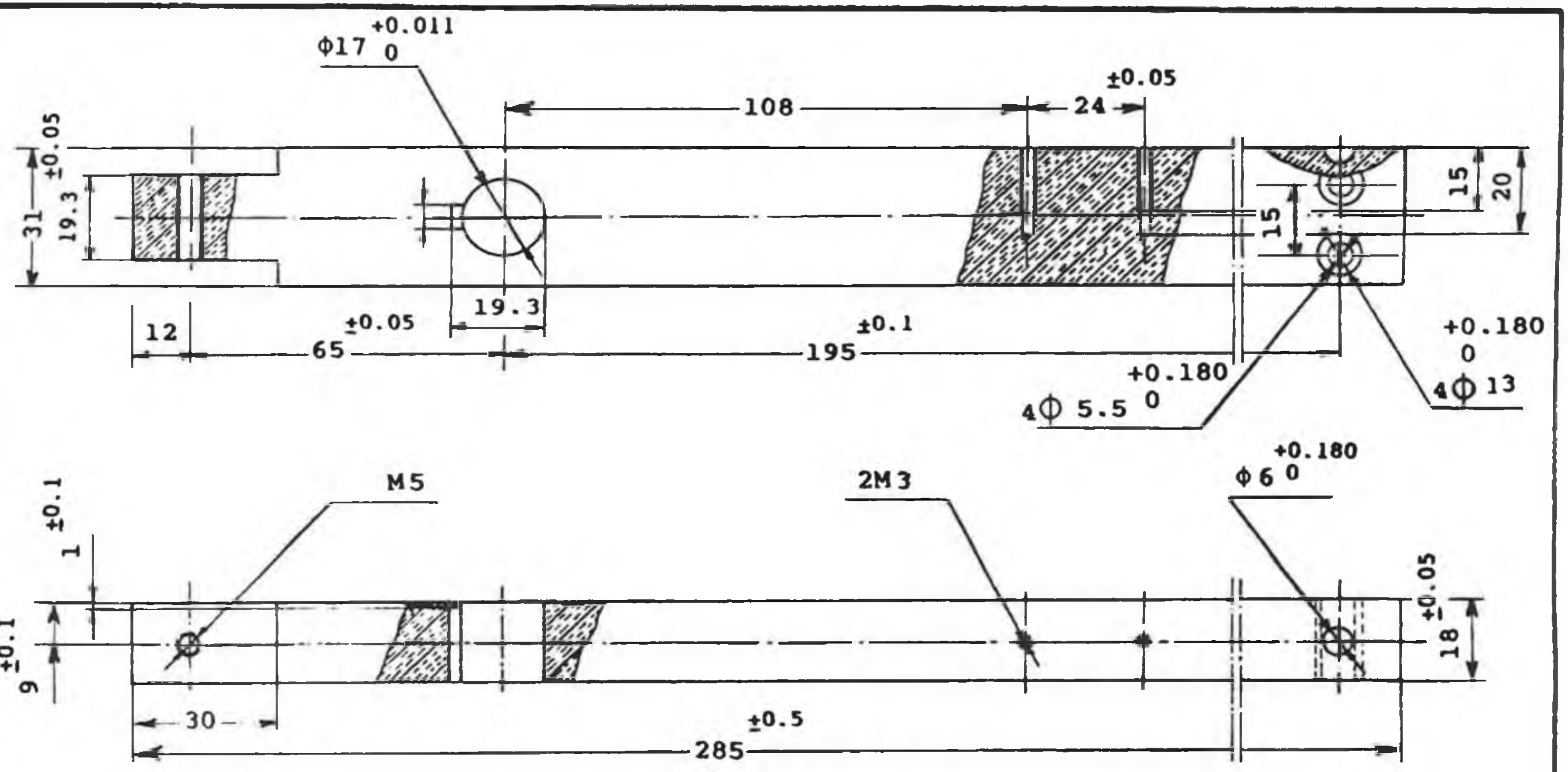
Fig.(2.22)



Drawing No. 7

Fig.(2.23)

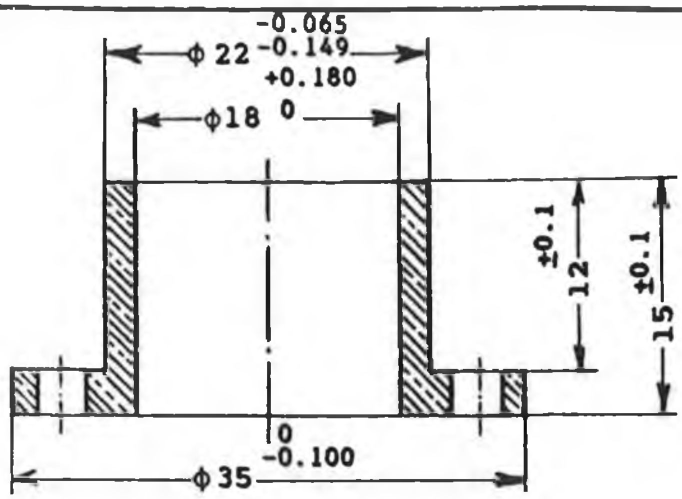
All dimensions in mm ±0.15.
Mild Steel.



Drawing No. 8

Fig.(2.24)

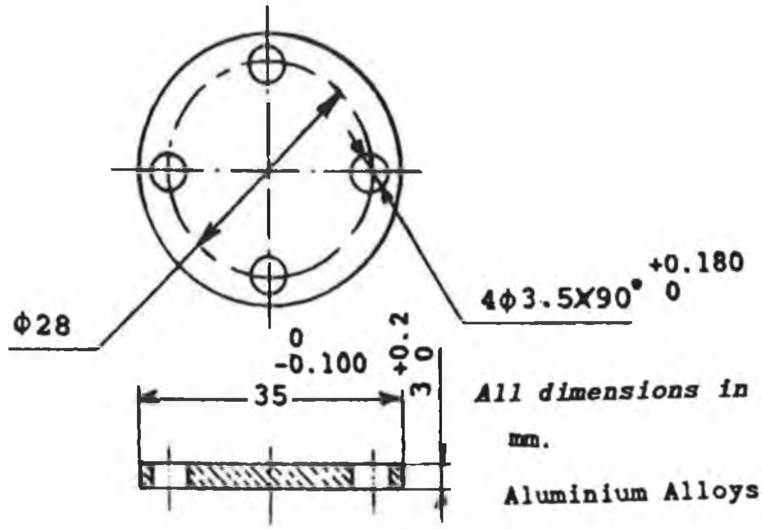
All dimensions in mm ±0.20 .
Aluminium Alloys.



All diamensions in mm
Brass.

Drawing No. 9

Fig.(2.25)

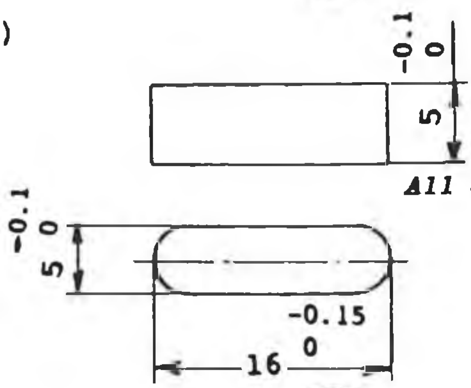


All dimensions in mm.

Aluminium Alloys.

Drawing No. 10

Fig.(2.26)



All dimensions in mm
Mild Steel.

Fig.(2.27) Drawing No. 11

All dimensions in mm ±0.15.

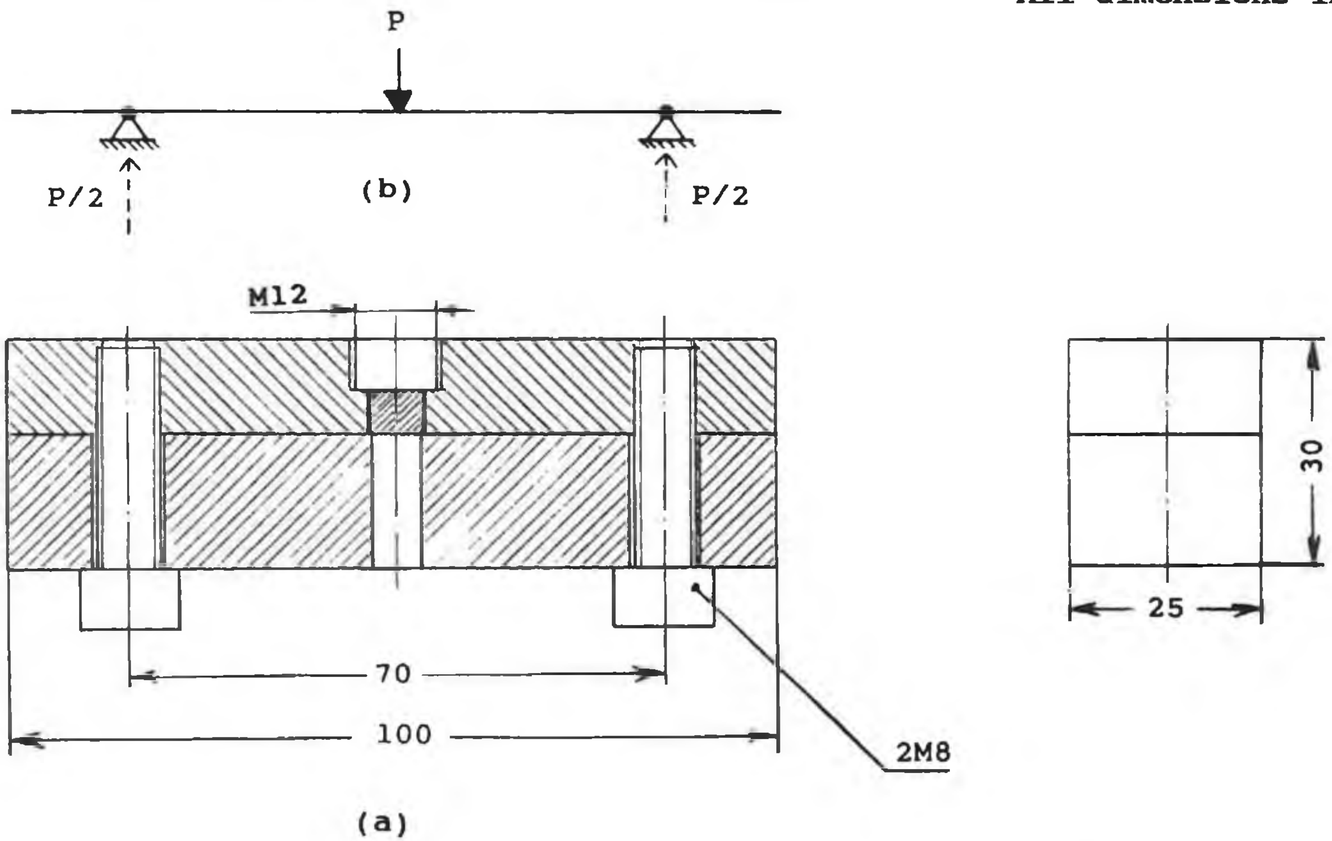


Fig.(2.28) A specially designed nut to apply pure torsion to the fastener.

All dimensions in mm ±0.10.

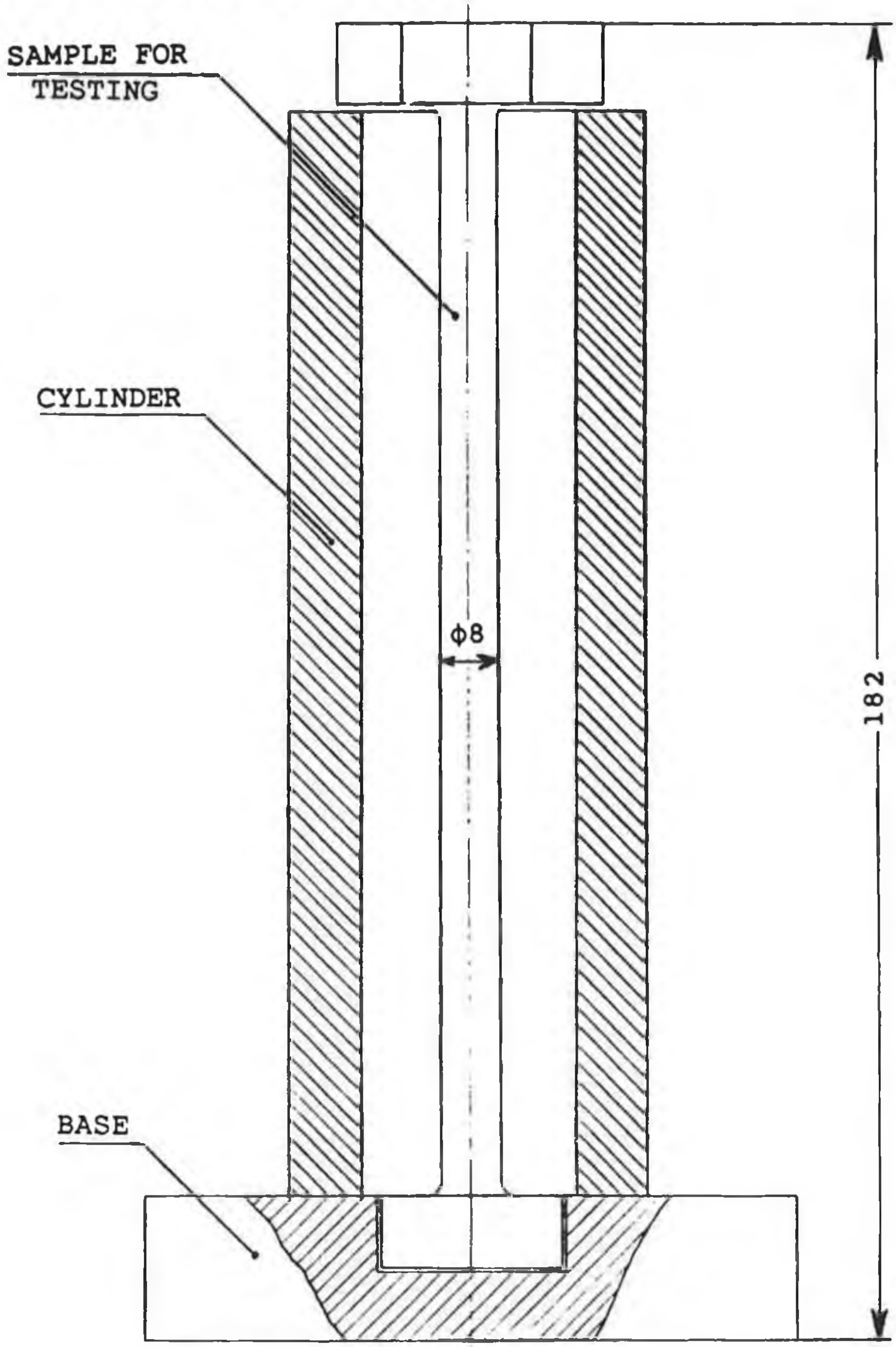


Fig.(2.29) Simple design (Assembly drawing) for pure torsion test.

Plate (2.6) DESIGNED EXTENSOMETER

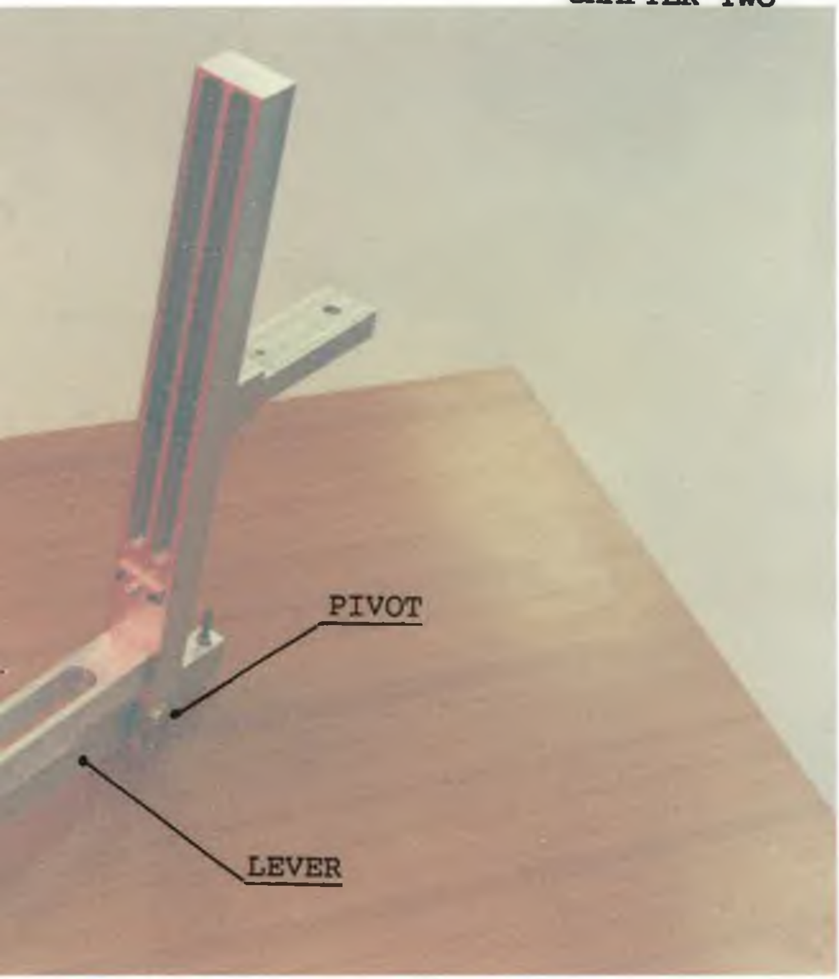
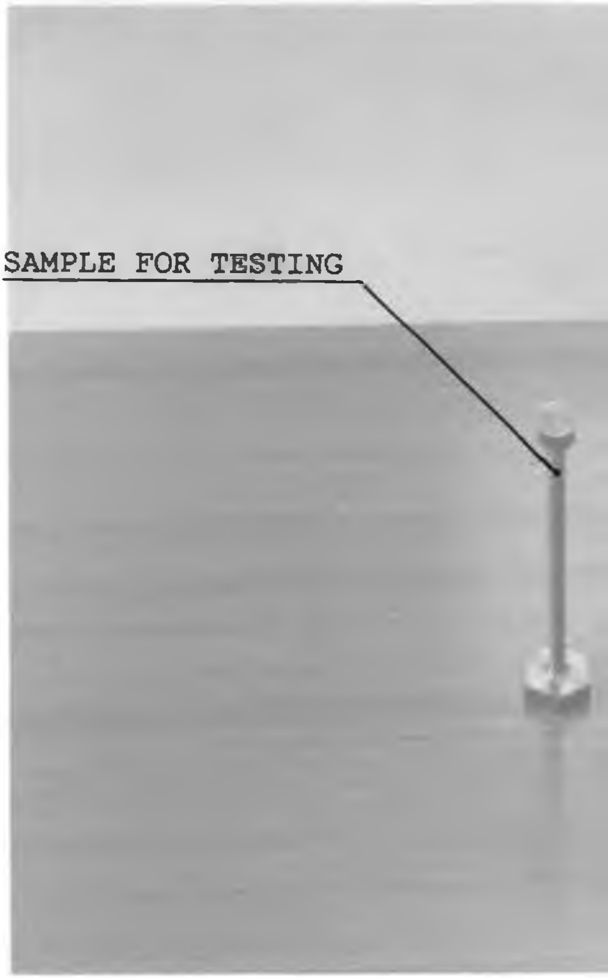
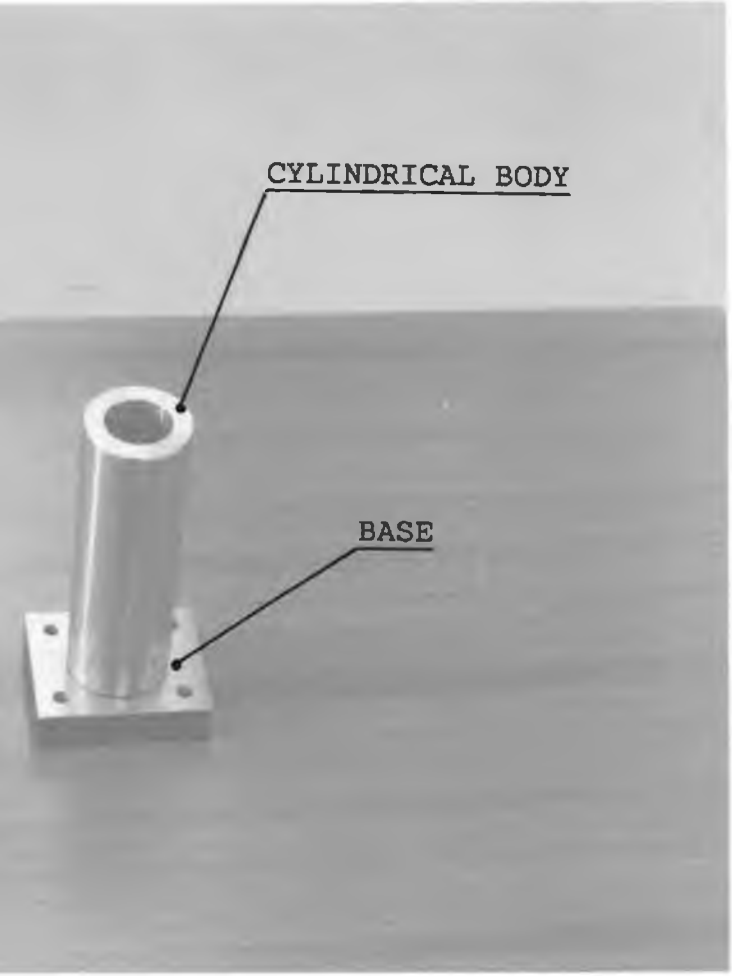

PIVOT
LEVER

Plate (2.7) SAMPLE FOR PURE TORSION TEST




CYLINDRICAL BODY
BASE

2.5 Materials

2.5.1 FASTENERS INVESTIGATED

The fasteners used in the tests were **M12X1.75X150 MM** manufactured by thread rolled after heat treatment **I.S.O.,** Grade 10.9, but after reducing the bolt shank. Thus it was decided to machine the bolt to a size as shown in **Figure** (2.30). These kinds of fasteners have a clean metal surface in the thread regions due to the thread rolling operation. The length of the clamped parts was sufficient for a long bolt to be installed.

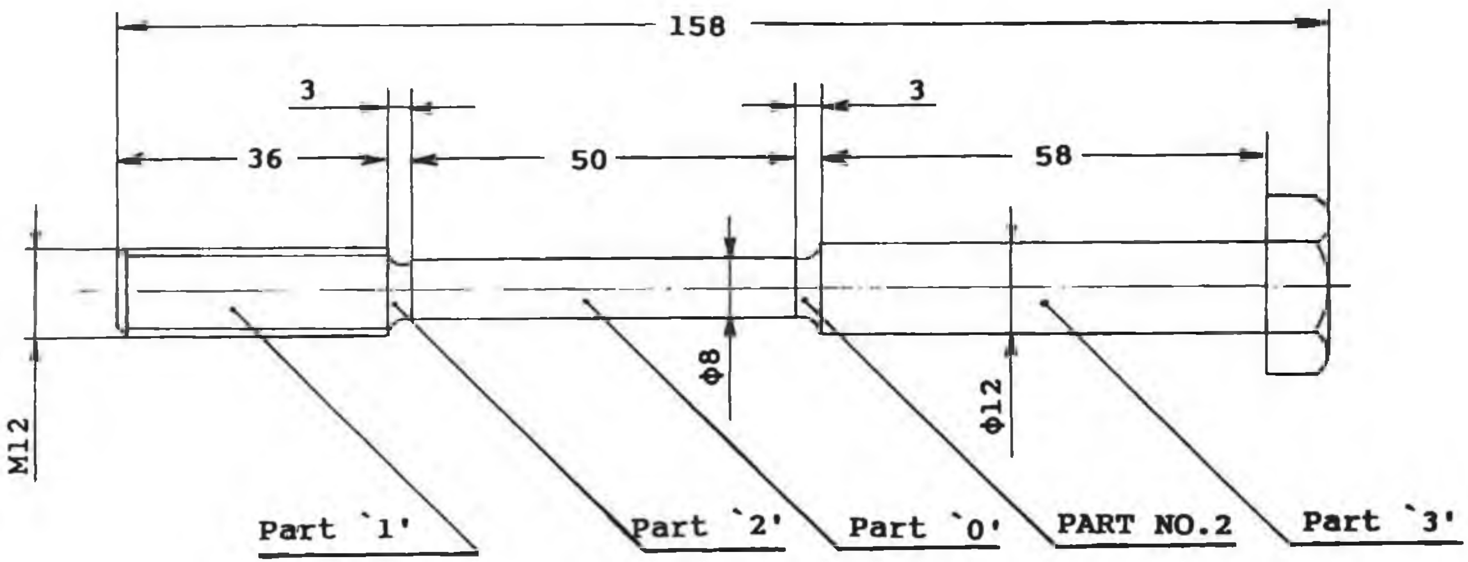
2.5.2 FASTENERS DETAILS

The fasteners were supplied by S.P.S. Laboratories, Ireland. Details of the dimensions of the fastener are given in **Figure** (2.31), see plate (2.8), [46].

- Size M12-1.75-150 MM.
- Grade 10.9 .
- Nominal ultimate tensile strength 1000 N/mm^2.
- Nominal yield stress 900 N/mm^2.
- Helix Angle 2.9354°.
- Thread angle 30°.
- Mean Under Head Radius 6.25 mm.
- Thread Pitch 1.75 mm.
- Pitch Diameter 10.8634 mm.
- Minor Diameter 9.8531 mm.
- Nominal Stress Area 84.268 mm.

Fig.(2.30) dimensions of fasteners used in the tests.

M12X1.75X150 Grade 10.9



All dimensions in mm ±0.20.

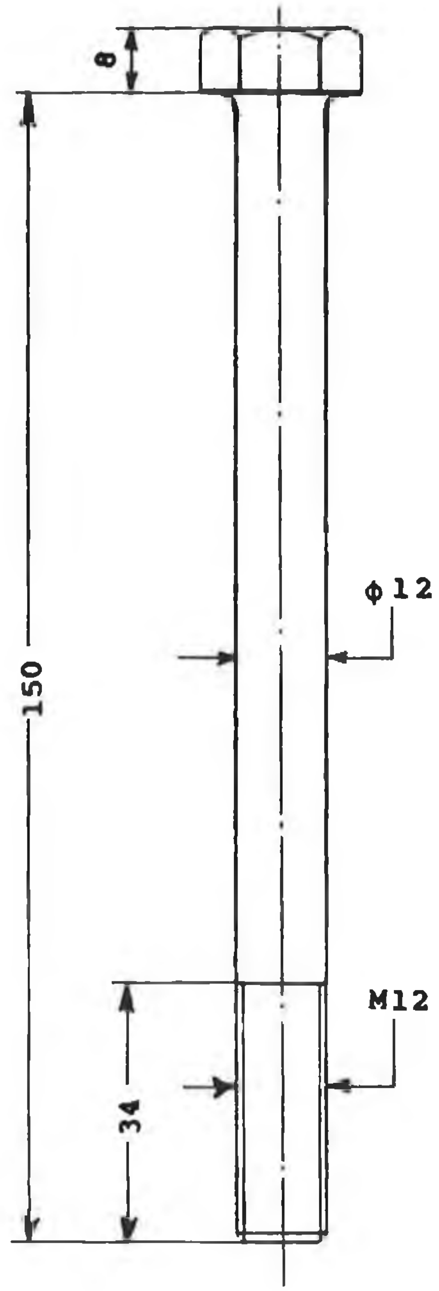
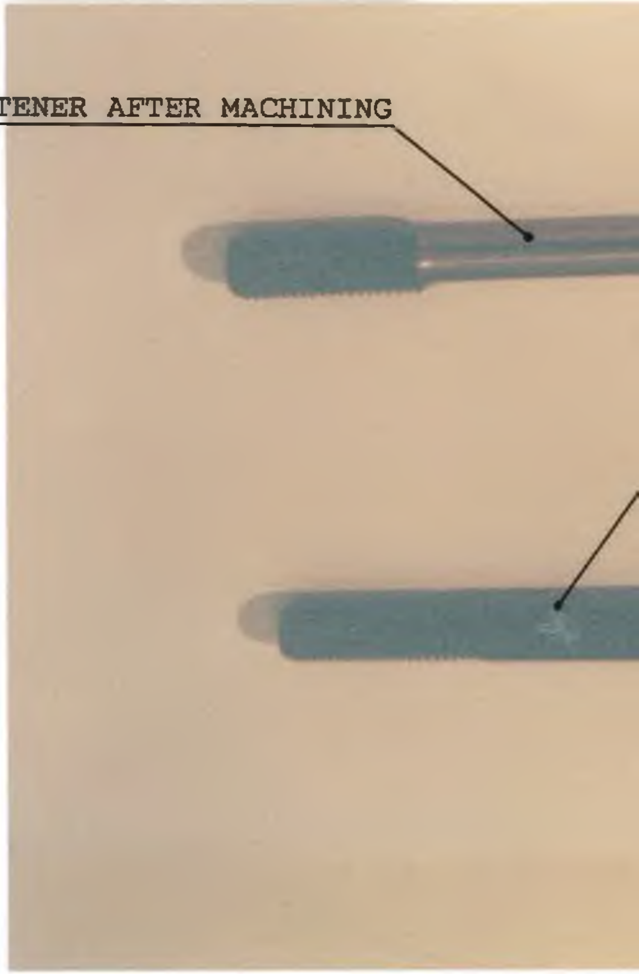
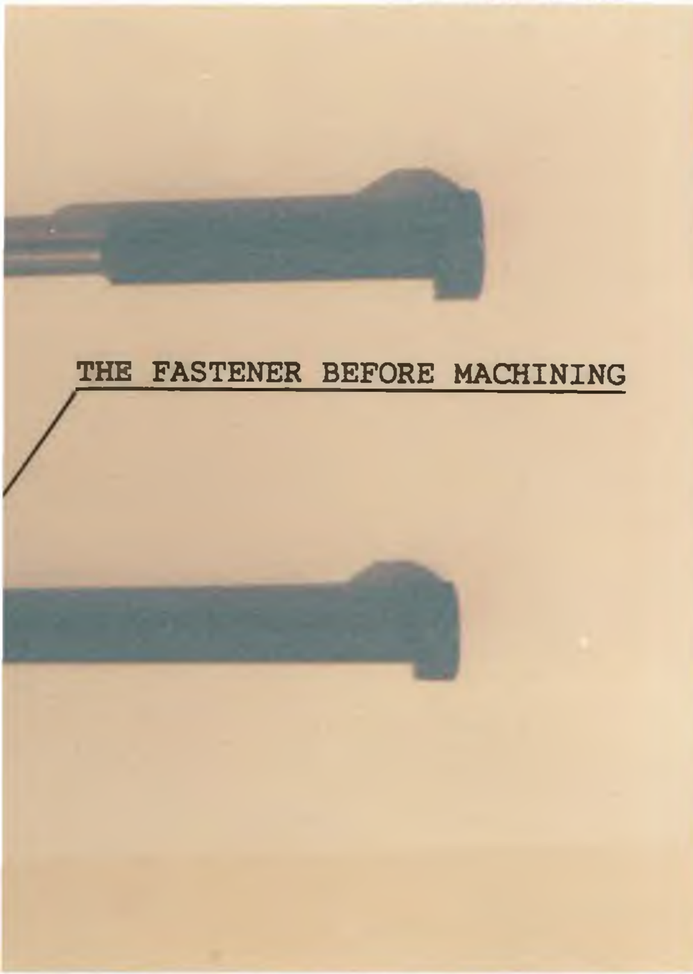


Fig.(2.31) Dimensions of the fastener size M12X1.75X150 Grade 10.9

Plate (2.8) FASTENER INVESTIGATED




THE FASTENER BEFORE MACHINING

CHAPTER THREE

THEORETICAL ANALYSIS OF THE RESPONSE OF THE BOLT

CHAPTER THREE

THEORETICAL ANALYSIS OF THE RESPONSE OF THE BOLT

3.1 Introduction

In this study an attempt has been made to theoretically explain the elastic-plastic behaviour of the bolt subjected to torsional and tensile loads applied sequentially during the experimental tests. In this chapter a theoretical investigation is presented of cases when the fasteners respond to combined torque and axial load.

3.2 Principal aims of the theoretical work

One of the reasons for this investigation is to establish the mechanism in which yielding takes place under combined torque and axial load in a bolted joint. Published literature shows some conflicting evidence on the load-extension characteristics under pure axial load and combined axial load and torque.

The experimental results carried out by SPS laboratories, Ireland [49], show that when a bolt is loaded under uniaxial load to plastic yielding and then unloaded and subsequently subjected to combined axial load and torque, there is no appreciable reduction in uniaxial tensile strength. This suggests that the torsional stresses remaining from the tightening operation disappears as further external tensile load is applied. This also suggests that the explanation for reaching the full value of uniaxial yield strength is not the elastic springback of the joint at the end of the tightening process. However, these results differ from experimental results reported by Wilhelm Schneider [31] which shows that under combined torsion and tensile loading the uniaxial tensile yield point was not fully reached.

The aim of this investigation is to confirm the finding of one of the two investigations and also to establish the dissipation mode of torsional and tensile energy in bolted joint during tightening.

The investigation, consisting of two phases, was scheduled according to the following:

Phase one: The fastener is subjected to an axial tensile load after an initial torque is applied.

Phase two: The fastener is subjected to a torque after an initial axial tensile load is applied.

3.3 Theoretical procedure

3.3.1 FASTENER INITIALLY SUBJECTED TO A TORQUE

Firstly, the bolt was assumed to be subjected to a certain amount of torque, say less than or equal to the yield torque and while this torque was held constant the bolt was assumed to be loaded in tension to different magnitudes of axial load.

3.3.2 FASTENER INITIALLY SUBJECTED TO AN AXIAL LOAD

For this theoretical investigation it is assumed that the bolt is initially subjected to a certain level of axial load, say less than the yield stress in tension, and then torque of different magnitudes were assumed to be applied to the bolt.

3.4 Theoretical analysis

3.4.1 RESPONSE OF THE BOLT EXPLAINED ACCORDING TO THE YIELD CRITERIA

3.4.1.1 FASTENER INITIALLY SUBJECTED TO A TORQUE

According to the yield criteria the plastic zone of the material can not be extended to the core of the fastener for any torque less than the yield torque. When the initial torque is equal to the yield torque then only the material at outside diameter of the bolt can be yielded. This means, any increment of axial stress will lead to the reduction in the applied torque to maintain the combined stress to the level of the yield stress in tension following the von Mises` criterion given as follows:

$$(\sigma/\sigma_y)^2+(\tau.\sqrt{3}/\sigma_y)^2=1. \tag{3.1}$$

This equation plots as part of an ellipse in the rectangular axes of σ/σ_y and τ/σ_y , see **Figure (3.1)** . According to this Figure, plastic yielding takes place in pure torsion when no axial stress is applied, see point `**a**'. When an axial stress is applied, the torsional stress starts to decrease by following the curve `**ad**'. The reduction of shear stress is represented by the distance `**ac**' and at the same time the increment in the axial stress is represented by the distance `**cb**'. According to the **von Mises yield criterion** [23] the material always follows the curve `**ab**', and the stress state can not be outside the yield locus `**ad**'. At any value of torsional and axial stress, the combined stress should be equal to or less than the yield stress in tension. This theory will dictate that the material is yielding only at the outside diameter during this combined loading.

3.4.1.2 FASTENER INITIALLY SUBJECTED TO AN AXIAL LOAD

According to the yield criteria, the plastic zone in the bolt subjected to an initial axial stress and then subsequently to a torque can not extend to the core of the fastener. Only the material at outside diameter of bolt can yield. It thus follows that any increment in the torque causes reduction in the axial stress to keep the combined stress equal to the yield stress in tension. Equation (3.1) again plots as an ellipse in the rectangular axes of σ/σ_y and τ/σ_y as shown in **Figure (3.2)**. This **Figure** is showing that the material yields at axial stress equal to σ_y in the absence of any torque. This corresponds to point `c'. As soon as any torque is applied, the axial stress should start to decrease and the stress state would follow the curve `ce' because the material can not carry stresses more than the yield stress in tension.

Figure (3.2) is also showing that for an initial axial stress (σ/σ_y =0.72) no reduction in the axial stress will take place until the subsequently applied torque gives rise to torsional stress equal to τ/τ_y=0.4.

3.4.2 EXTENT OF THE REDUCTION OF TORQUE DUE TO ELONGATION OF BOLT

It is necessary to calculate the reduction of torque as a result of the extension of the bolt due to the application of an axial stress. These calculations were made to establish the effect of the elongation of the bolt on the magnitude of the initial torque. Suppose the bolt is twisted by the angle θ_0 so that the shear strain is given by,

$$\gamma_0=R_0.\theta_0/L_0. \tag{3.2}$$

Due to the axial stress, the bolt is extended which reduces the shear strain to γ given by,

$$\gamma=R_0.\theta_0/L. \tag{3.3}$$

Then the reduction of shear strain can be obtained as:

$$d\gamma=\gamma_0-\gamma. \tag{3.4}$$

Therefore, the reduction of shear stress due to the extention in the bolt is given by,

$$d \ =G.d\gamma. \tag{3.5}$$

Assuming that the bolt is twisted to the yield point so that,

$$\gamma_0= {}_y/G. \tag{3.6}$$

For the axial stress in the elastic range the Hooke's law can be applied, thus,

$$\sigma_x=E.\epsilon. \tag{3.7}$$

Where

$$\epsilon=\ln(L/L_0). \tag{3.8}$$

Based on the above equations it can be shown that the reduction of shear stress due to the extension of the bolt is negligibly small. This reduction was about 2 to 2.5 MPa when the axial load increases from zero to σ_y, so that, this reduction can be neglected by comparing with the other reduction of torque.

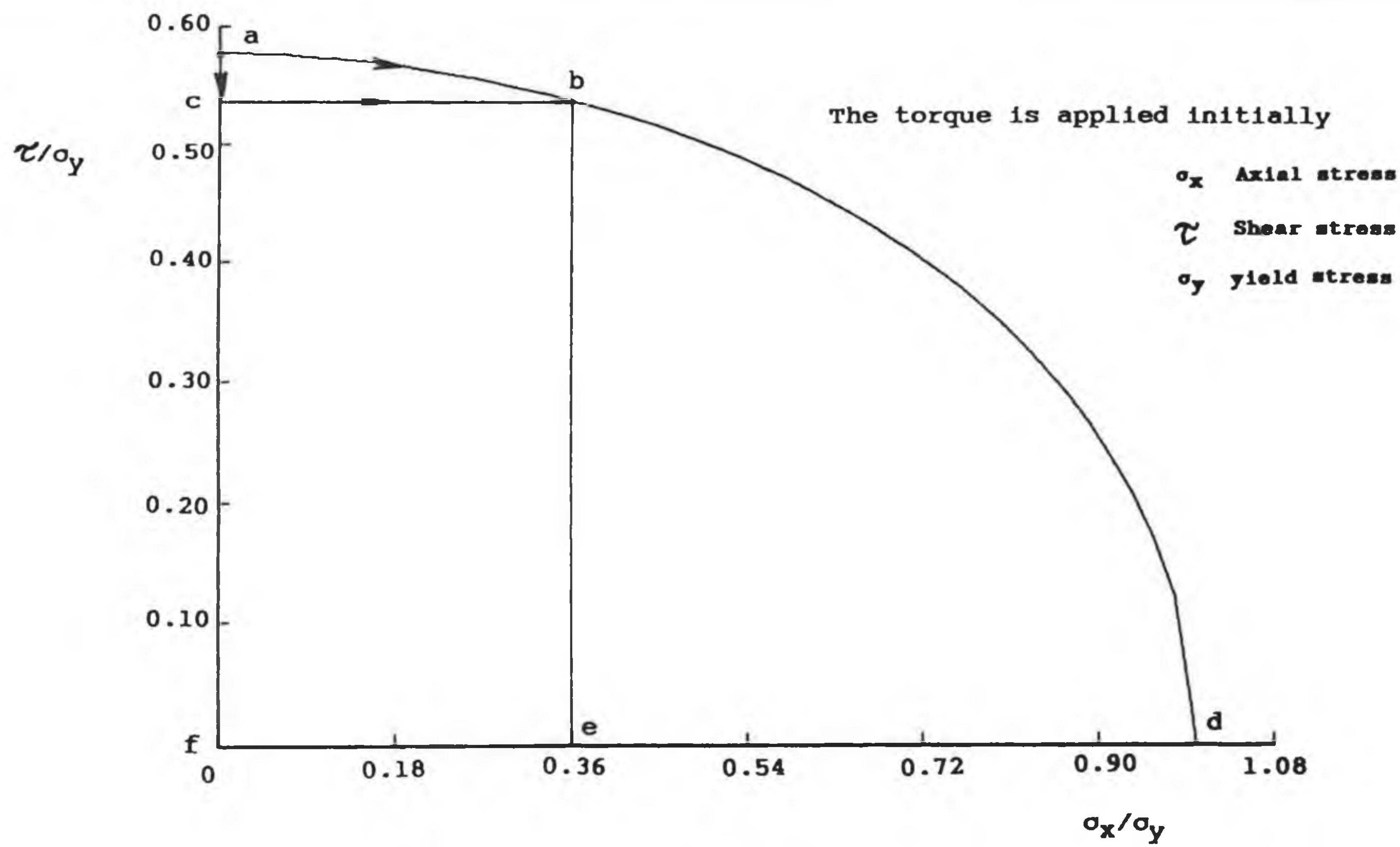


Fig.(3.1) VON MISES` YIELD CRITERION

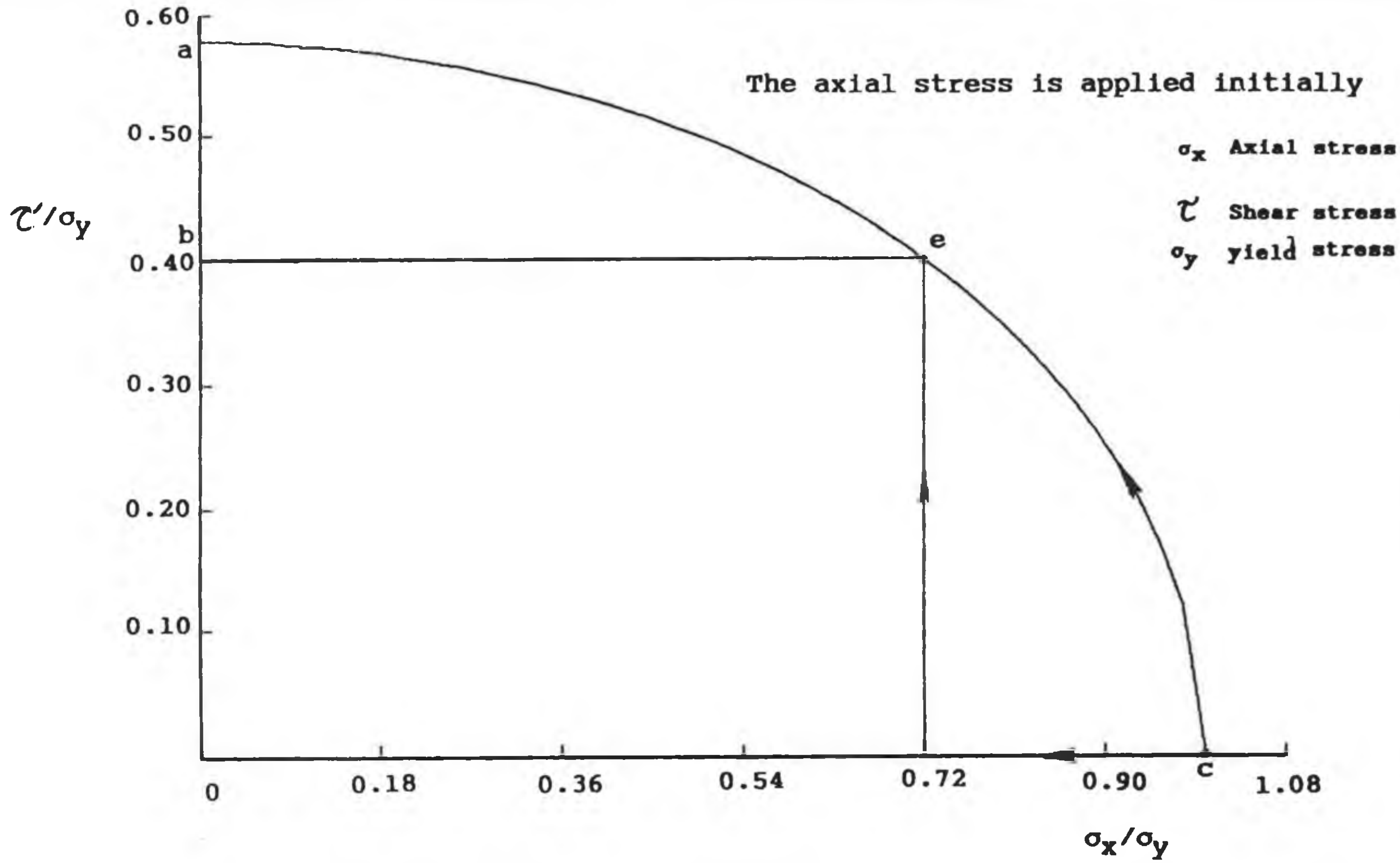


Fig.(3.2) VON MISES` YIELD CRITERION

3.4.3 THEORETICAL ANALYSIS ASSUMING PLASTIC ZONE EXTENDS TOWARDS THE CORE OF THE BOLT

3.4.3.1 FASTENER INITIALLY SUBJECTED TO A TORQUE

1. TORSIONAL STRESS BEHAVIOUR

It is clear that, according to the yield criteria methode that no plastic deformation will take place when the combination of axial and torsional stresses is still less than the yield stress in tension of the material of the bolt. When the combined stress in the fastener becomes equal to the yield stress in tension, the material of the bolt starts to yield at the outside diameter. The magnitude of the torque is constant in the fastener when the combined stress is still less than or equal to the yield stress in tension. The objective now is to establish theoretically what may happen when the combined stress exceeds the yield stress and plastic zone is allowed to develop.

Suppose the mechanical properties of the material of the bolt in shear corresponds to the shear stress strain diagram as shown in **Figure (3.3a)**. It is clear from this figure that for elastic-plastic condition the shear stress would vary linearly from 0 to τ_y for $0 \le \rho \le R$ and the stress would be constant at τ_y for $\rho \ge R$. We also assume that the idealized stress-strain diagram of the material is elastic perfectly plastic as shown in **Figure (3.3b)** . In this case the mechanical properties of the material of the bolt corresponds to elastic linear work-hardening, but the magnitude of the plastic modulus is negligibly small in comparison with the elastic modulus.

Figure (3.4) is showing the distribution of the combined stress squared along the radius of bolt. Since according to the yield criteria the material can not exceed the yield stress in tension, the stress profile can not follow the curve `**dac**' as shown in **Figure (3.4)**. For material properties corresponding to the diagrams in **Figure (3.3)** the stress distribution should corresponded to the curve `**da**' from 0 to σ_y for $0 \leq \rho \leq R$ and then the stress is constant at σ_y for $\rho \geq R$ upto the outside radius R_0 , thus following line `**ab**'.

This would mean that the area `**abc**' is excess so that the plastic zone in the bolt would extends from the outside diameter towards the core of the fastener. This would occur when the level of stress induced in the fastener due to the combined stresses is greater than the yield stress. The depth of plastic zone will depend on the magnitude of the axial stress applied to the fastener after the initial torque as indicated in **Figure (3.5)**. It is obvious that the material can not follow the dotted curve resulting from the combined stresses.

This reduction in the torque may be explained as follows: when the combination of tensile and torsion stresses induced in the fastener exceeds the yield stress in tension, the material can not carry the excess stress due to the torque. Hence, the torque reduces corresponding to the reduction in the torsional stress.

According to this suggestion, the level of torsional stress in the bolt would decrease as the tensile load is increased making the combined stress in the fastener greater than the yield stress in tension.

2. <u>CALCULATE THE REDUCTION OF TORQUE</u>

From Figure (3.4) it may be deduced that the elastic shear stress distribution correspond to the torque `$T_1$' and the shear stress distribution in the plastic zone correspond to the torque `T_2'. The total torque giving rise to this stress distributions in the fastener may be given as follows:

$$T` = T_1 + T_2. \tag{3.9}$$

The reduction in torque due to the applied axial load can thus be written as follows:

$$DT = T - T`. \tag{3.10}$$

This reduced torque in the fastener is calculated according to the cross-sectional area of the bolt over which the elastic and plastic shear stresses act. Based on the stress-strain diagram as shown in **Figure (3.3)** the torque ` $T_1$` ' is given by the following equation:

$$T_1=\tau_r.I_r/R. \tag{3.11}$$

And the torque `T_2' is represented as follows:

$$T_2=2\pi.\tau_r(R_0{}^3-R^3)/3. \tag{3.12}$$

where

I_r=Polar moment of inertia of the cross-sectional area.

$$I_r=\pi.R^4/2. \tag{3.13}$$

τ_r=Shear stress at radius ,r, given by:

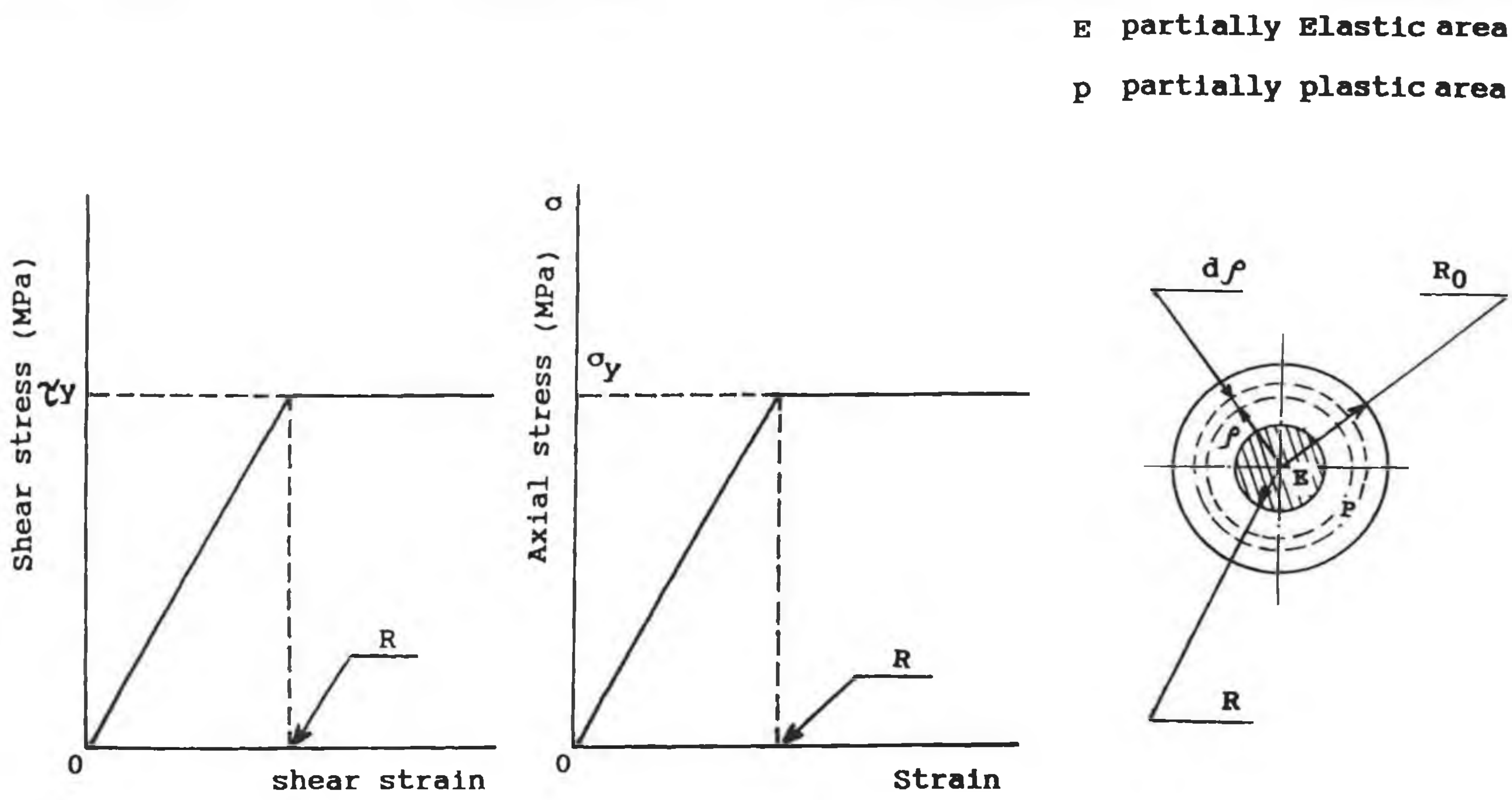
$$\tau_r=T.R/I_r. \tag{3.14}$$

By combining equations (3.9), (3.11), (3.12), (3.13) and (3.14) we obtain the expression for the torque ` T` 'remaining in the fastener. Thus,

$$T`=\pi.\tau_r[R^3+4(R_0{}^3-R^3)/3]/2. \tag{3.15}$$

It is clear from the above that as the magnitude of the applied axial load is increased the magnitude of the initially applied torque is reduced. Eventually, when the applied axial load becomes equal to the yield stress in tension, only a small elastic torque would be present. Further increase in the tensile load will lead to failure of the bolt through necking and fracture. The elastic torque still being present until fracture takes place. Results were calculated on the basis of the above equations for different magnitudes of initial torque.

Figure (3.6) is showing these results illustrating the relationship between the initial torque and subsequently applied axial stress in the fastener, assuming plastic zone extends towards the core of the bolt. It clearly demonstrates the manner in which the torque is reduced as soon as the combined stress exceeds the yield stress in tension due to increasing axial load.



(a) (b) (c)

Fig.(3.3) Mechanical properties of material in the bolt (Corresponding stress-distribution).

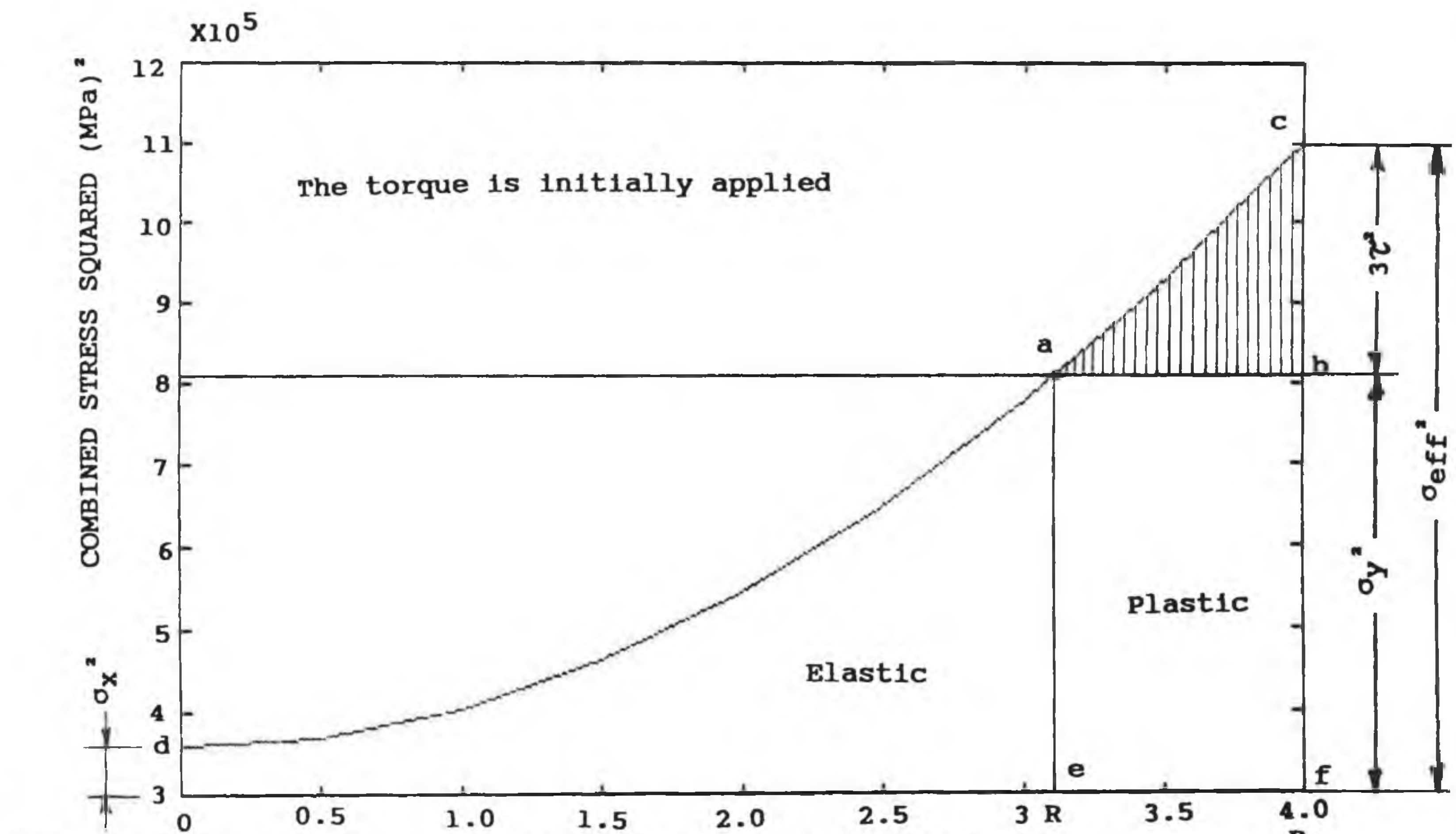


Fig. (3.4) Relationship between combined stress squared and radius of bolt body in the fastener.

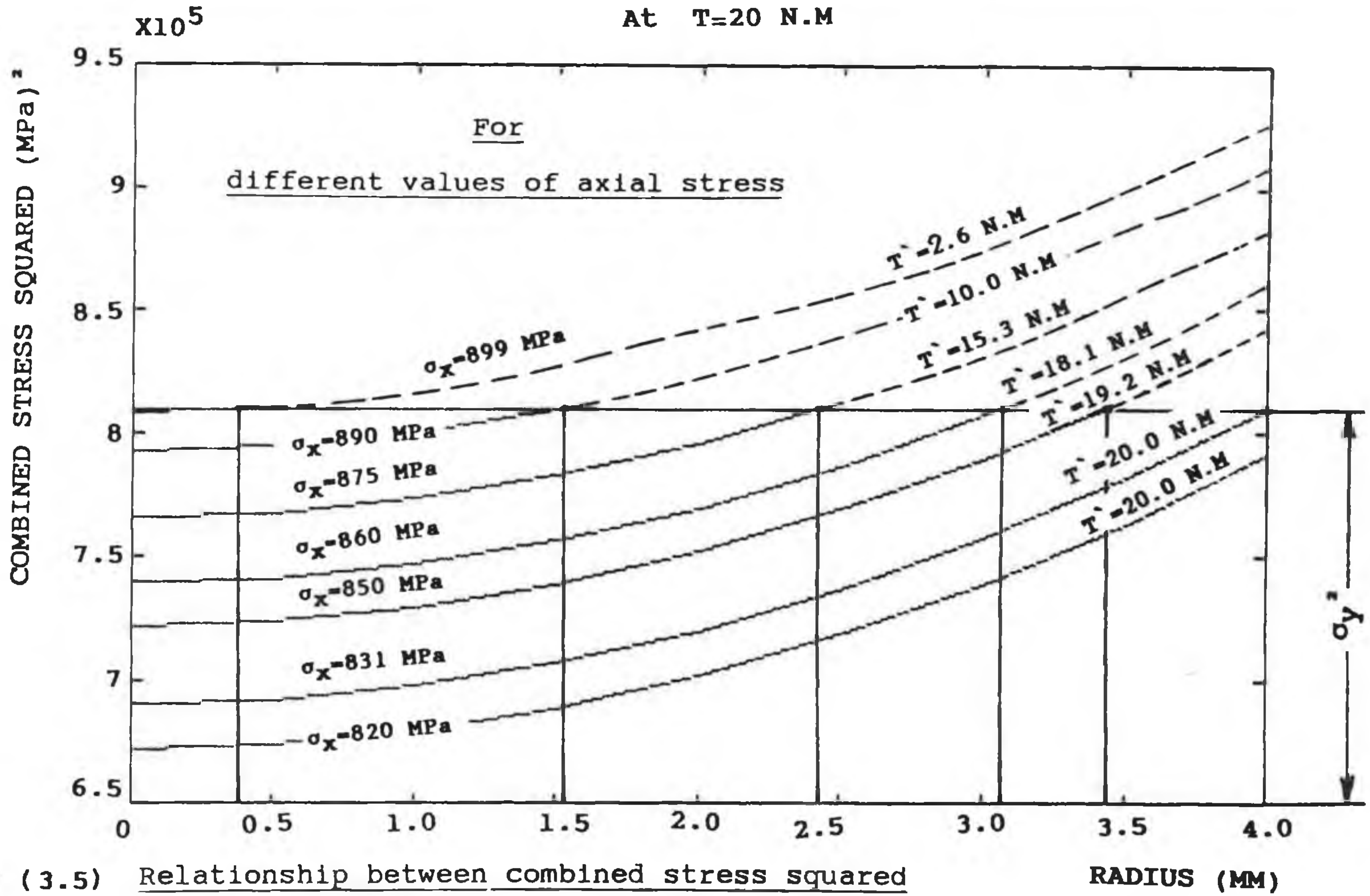


Fig. (3.5) Relationship between combined stress squared and radius of bolt body in the fastener.

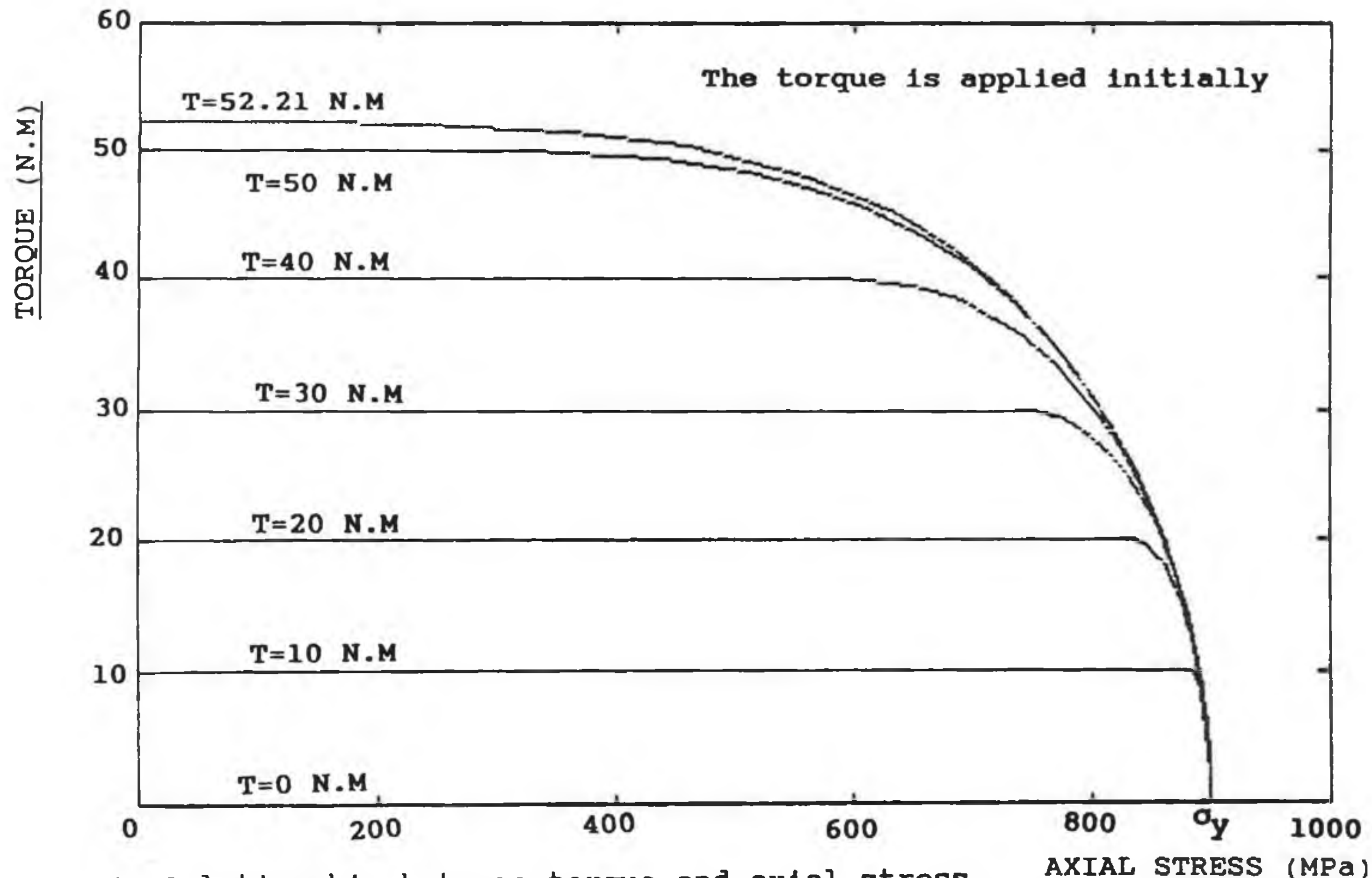


Fig.(3.6) Relationship between torque and axial stress in the fastener.

3.4.3.2 FASTENER INITIALLY SUBJECTED TO AN AXIAL LOAD

In this section we will attempt to establish the effect of torque stress distribution in the bolt initially subjected to an axial load.

1. TENSILE STRESS BEHAVIOUR

When the bolt is initially subjected to an axial stress and then the torque is applied, no platic deformation will take place until the combination of tensile and torsion stresses becomes equal to the yield stress in tension of the material of the bolt. As soon as the combined stress induced in the fastener reaches the yield stress, the outside diameter of the bolt starts to yield. However, for the condition when the combined stress is greater than the yield stress then yielding will follow the mechanical properties of the material in tension and torsion corresponding to the diagrams in **Figure (3.3)**.

Figure (3.7) is showing the distribution of combined stress (squared) over the cross-section of the bolt. The bolt is subjected to an initial axial stress `$\sigma_x$' and then a torque is applied to give rise to shear stress `τ' at the outer radius. According to the yield stress the annular area between points `jh' can not withstand the combined stress. The shear stress field over this area manifests itself as reduction of direct stress over the entire cross-section and the combined stress (squared) field follows the path `bkh' rather than `aji' for the elastic-perfectly plastic bolt material. The material within the radius R is still elastic but for $\rho > R$ upto the outside radius `R_0 ', the material is plastic being stressed at σ_y. This suggests that, for a given torque a plastic outer zone will develop as well as the reduction in the initial axial stress.

The depth of this plastic penetration depends on the magnitude of the applied torque to the fastener after the initial axial load is applied. It is clear from Figure (3.8) that the stress field in the material can not follow the dotted curves due to the fact that the combined stresses are greater than the yield stress. As a result the plastic zone of the material extends towards the central core and the axial stress disappears.

2. CALCULATE THE REDUCTION OF AXIAL STRESS

Referring to Figure (3.7) the reduction of axial stress may be calculated by considering the combined stress (squared) field `$R'jiR_0$' over the annular section between the radii R` and R_0 and subtracting plastic stress field `$R`jhR_0$' over the same area. Thus the reduction of axial stress `$D\sigma_x$' may be obtained by equating the `$D\sigma_x^2$' over the entire cross-section with the subtracted results explained above:

$$\pi . R_0^2 . D\sigma_x^2 = \int_{R'}^{R_0} 2\pi . \rho d\rho (\sigma_x^2 + 3\tau_r^2) - \sigma_y^2 . \pi (R_0^2 - R'^2), \qquad (3.16)$$

Where $$3\tau_r^2=3\tau_{out}^2.\rho^2/R_0^2. \tag{3.17}$$

After integrating the reduction in axial stress is given by,

$$D\sigma_x^2=(1-R'^2/R_0^2).[1.5\tau_{out}^2(1+R'^2/R_0^2)+(\sigma_x^2-\sigma_y^2)]. \tag{3.18}$$

Where

$$R'=R_0.\sqrt{((\sigma_y^2-\sigma_x^2)/3)}/\tau_{out}. \tag{3.19}$$

Referring to **Figure (3.7)** the remaining axial stress is then given by the following equation:

$$\sigma_x'=\sqrt{\sigma_x^2-D\sigma_x^2}. \tag{3.20}$$

In the following equations **R** represents the radius of the bolt at which the material is yielding. This radius can be calculated from the following equation:

$$R=R_0.\sqrt{((\sigma_y^2-\sigma_x'^2)/3)}/\tau_{out}. \tag{3.21}$$

According to these calculations the stress field given by the area **`gajig'** should be equal to that given by the area **`bkhfb'** which are calculated using the following equations:

The area **`gajig'** is equal to **(A1)**:

$$A1=3\pi.\tau_{out}^2.R_0^2/2. \tag{3.22}$$

And the area `**bkhfb**' is equal to (**A2**):

$$A2=1.5\pi.\tau_{out}^{2}.R^{4}/R_{0}^{2}+\pi(R_{0}^{2}-R^{2}).(\sigma_{y}^{2}-\sigma_{x}'^{2}). \qquad (3.23)$$

It is also clear from the above that for any torque applied to the fastener, the axial stress is reduced. For a small initial axial stress this may mean total unloading. However for a high initial axial stress the applied torque to the fastener may not reduce the axial stress to zero. Thus, a certain level of axial stress may be left in the fastener.

Figure (3.9) is showing the relationship between the initial axial stress and the subsequently applied torque in the fastener for different levels of axial stresses, assuming plastic zone extends towards the core of the bolt. This **Figure** clearly illustrates the manner in which the initially applied axial stress may be decreased as soon as the combined stress induced in the fastener exceeds the yield stress in tension.

At σ_x=500 MPa

The axial stress is applied initially

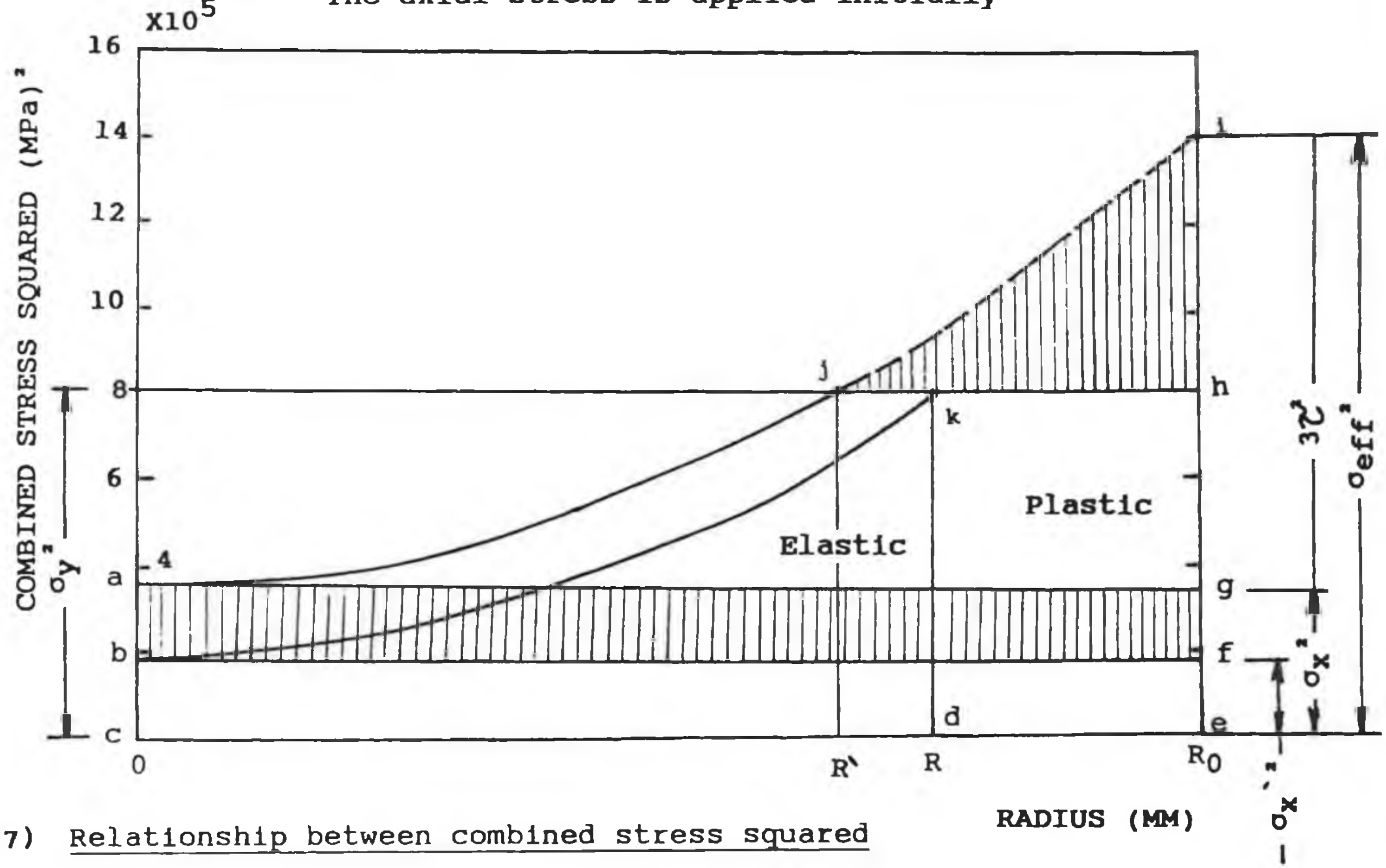


Fig.(3.7) Relationship between combined stress squared and radius of bolt body in the fastener.

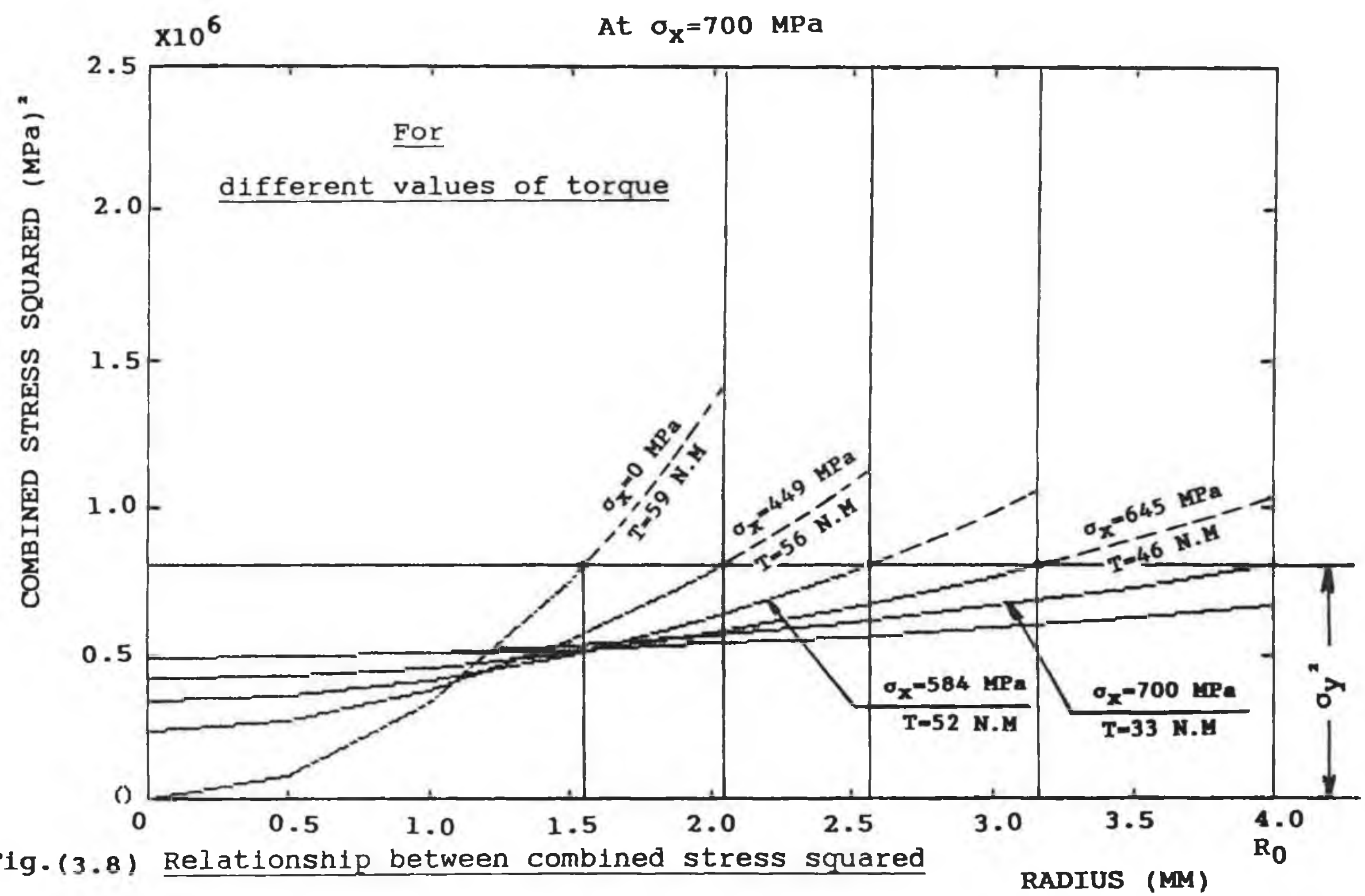


Fig.(3.8) Relationship between combined stress squared and radius of bolt body in the fastener.

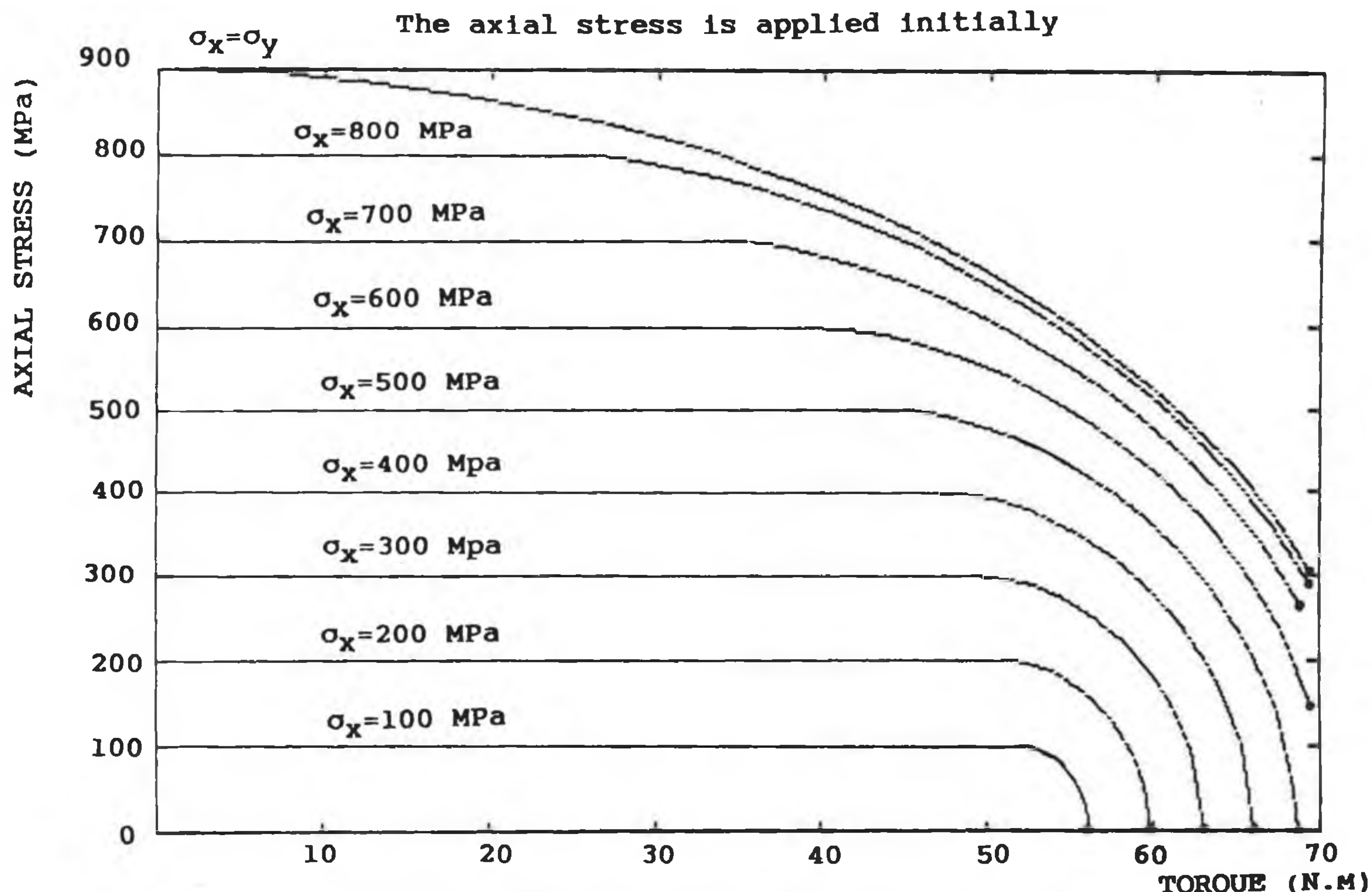


Fig.(3.9) Relationship between torque and axial stress in the fastener.

CHAPTER FOUR

EXPERIMENTAL PROCEDURE

CHAPTER FOUR

EXPERIMENTAL ROCEDURE

4.1 Introduction

This chapter presents the experimental procedure and calibration of the test equipment. Before commencing any test all interfaces of the bolted joint were firmly fixed to prevent interface rotation.

4.2 Calibration of the test equipment

It was necessary to calibrate the load cell, L.V.D.T., amplifier with plotter, and the spanner before the experiments, to make the equipment suitable for the experiments.

4.2.1 CALIBRATION OF THE L.V.D.T. WITH AMPLIFIER

The L.V.D.T. was calibrated with the amplifier to obtain the elongations of the bolt directly from the reading of the amplifier. The suitable input voltage range for the amplifier was 2V to 5V , which was chosen from 8 options.

4.2.2 CALIBRATION OF THE LOAD CELL WITH AMPLIFIER

Calibration of the load cell was very important in order to establish the relationship between the input and the output of the load cell, by determining the relationship between the load and the voltage output of the load cell. The calibration chart was constructed from experiments carried out by using a tensile test machine.

The measurements show a linear relationship between the input and output of the load cell, as shown in **Figure (4.1)**. The sensitivity of the load cell was calculated at **5V** excitation of the amplifier, it was found to be equal to **47 μV/V/kN.**

Each amplifier supplied excitation and had a digital output with optional output suitable for a **X-Y recorder.**

4.2.3 CALIBRATION OF THE ELECTRONIC TORQUE WRENCH

During the experiments, it was not possible to use the electronic torque wrench to apply torque to the fastener and hence, a spanner was used.

It was therefore, necessary to calibrate the spanner before the experiments. This calibration was done to establish a relation between the applied external torque and the angle of twist in the fastener. The **SPS** electronic torque wrench was used in this calibration. No lubrication was required during the tests.

The measurements show a linear relation between the applied torque and the angle of twist in the fastener, within the elastic range of the bolt material , as shown in **Figure (4.2)**.

The calibration of the torque wrench was repeated again on the new sample under similar conditions. The results from this test was found to be very close to the previous results. based on this, it seemed that the specially designed nut, see **Figure (2.29)**, was very suitable for the pure torsion test.

4.2.4 CALIBRATION OF STRAIN GAUGE

In order to establish the relationship between the output of the torsional strain gauge attached to the body of the bolt and the input torque to the bolt, the calibration of strain gauge system was made. The assembled strain gauge was bonded to the body of the bolt with a thin layer of suitable adhesive.

The signal produced by the strain gauge was amplified by a Strain Gauge Conditioning Amplifier **series 2300** system to measure the torque and transfer to a **X-Y recorder**.

It was not possible to calibrate the strain gauge under combined axial tensile and torsional stresses. The bolted joint apparatus used in the experiments, was suitable for calibration of the applied torque only. Therefore, the strain gauge was calibrated in two stages:

- The signal from strain gauge was calibrated by applying a known torque `**in elastic range**' to the bolt head without any axial load ` **pure torsion** ', using the SPS torque wrench.

- A second calibration was also made by applying a known axial load without torsion in **`elastic range'** , using the hydraulic pressure.

This second calibration caused only negligible response of the strain gauge. This confirmed that the strain gauge was attached correctly.

Figure (4.3) illustrate the relationship between the applied torque to the bolt head and the output of the strain gauge as voltage output. This relationship represents the calibration chart from the experiments. The measurement show a linear relationship between the input torque and output response of the strain gauge, as shown in **Figure (4.3)**.

4.3 Experimental procedure

The following procedures were followed before carrying out any test.

4.3.1 PROCEDURE FOR TENSILE TEST

The extensometer was attached to the bolt first and then the hydraulic pressure was applied to increase the uniaxial load on the bolt until yield occurs. After that the hydraulic pressure is decreased to zero. The extensions of the fastener due to loading were plotted on a **X-Y** plotter. No lubrication was required during these tests, because no tightening was involved. These tests were repeated five times.

4.3.2 PROCEDURE FOR PURE TORSION TEST

It was easy to detect, if there was any tension in the fastener during the torsion test. This was determined from the reading of the amplifier. During the application of pure torsion there was no change in the reading of the amplifier, even though the load cell was still attached to the connected parts. Thus, no initial load was observed in the fastener.

At first, a small torque called `**Snug Torque**' is applied to ensure that the pure torsion operation has truly started, and that all gaps and looseness in the joint are eliminated before the pure torsion is applied. The bolt was then subjected to different values of torque, within the elastic range, and the corresponding angle of twist was measured.

The measurements show a linear relation between the applied torque and the angle of twist in the fastene, within the elastic range of the bolt material , as shown in **Figure (4.2)**.

4.3.3 PROCEDURE FOR COMBINED TORQUE AND AXIAL LOAD TEST

4.3.3.1 THE BOLT INITIALLY SUBJECTED TO A TORQUE

In order to measure any change in the torque initially applied to the bolt during the application of axial load, torsion strain gauge was attached to the body of the bolt. No lubrication was required during these tests, since the pure torque was applied to the fastener. A number of bolts were tested as follows: After installation of the bolt into the test apparatus the bolt was initially subjected to a certain amount of torque within elastic range. While preventing the bolt from untwisting, the extensometer was attached to the bolt to record the changes in length of the bolt. After that, the external tensile load was applied gradually and during this time the torque in the bolt was recorded simultaneously with the applied tensile load. The hydraulic pressure was increased to apply an external tensile load on the bolt until the bolt yielded.

4.3.3.2 THE BOLT INITIALLY SUBJECTED TO AN AXIAL LAOD

It was not essential to use any lubricant during these tests because the torque in the bolt was measured by means of the strain gauge.

A series of tests were performed to establish the effect of the application of torsional stress in the bolt initially subjected an axial load. A number of bolts were tested using the loading apparatus. In each case the extensometer was re-attached to the fastener to measure the changes in the length of the bolt. The bolt was initially subjected to a certain level of axial load within elastic range by means of the hydraulic device. Subsequently, torques of different magnitudes were applied to the bolt and the axial load in the bolt was recorded simultaneously with the bolt torque. The axial load decreases only by a small amount as the torque is increased gradually.

4.3.3.3 EFFECT OF COMBINED STRESS ON THE YIELD POINT TEST

After installation of the bolt into the joint, the loading sequence was applied to the bolt. A number of bolts were tested as follows:

1. The extensometer is attached to the bolt to measure the extension in the bolt. The zero reading was adjusted to the zero external tensile load. The pressure in the hydraulic cylinder was increased until the tensile yield was reached. The bolt was strained plastically upto a load, Y_1 , before the hydraulic pressure was released and the measured load returned to zero (unloading).

2. The extensometer was taken off and the bolt was initially subjected to a certain amount of torque in the elastic range. This torque was measured by means of the strain gauge. This applied torque was held constant. The extensometer was re-attached to the fastener and the pressure in the hydraulic cylinder was increased to simulate an external tensile load until the bolt had yielded. The yield stress, Y_2, was measured at 0.05 mm strain offset from the elastic line. The bolt was

strained plastically further upto a load ,Y_3, before the hydraulic pressure was released and the measured load returned to zero `unloading'.

3. The extensometer was re-attached to the fastener and the hydraulic pressure was increased again until the bolt was yielded for the third time. The yield stress, Y_4, was measured at o.o5 mm strain offset from the the elastic line.

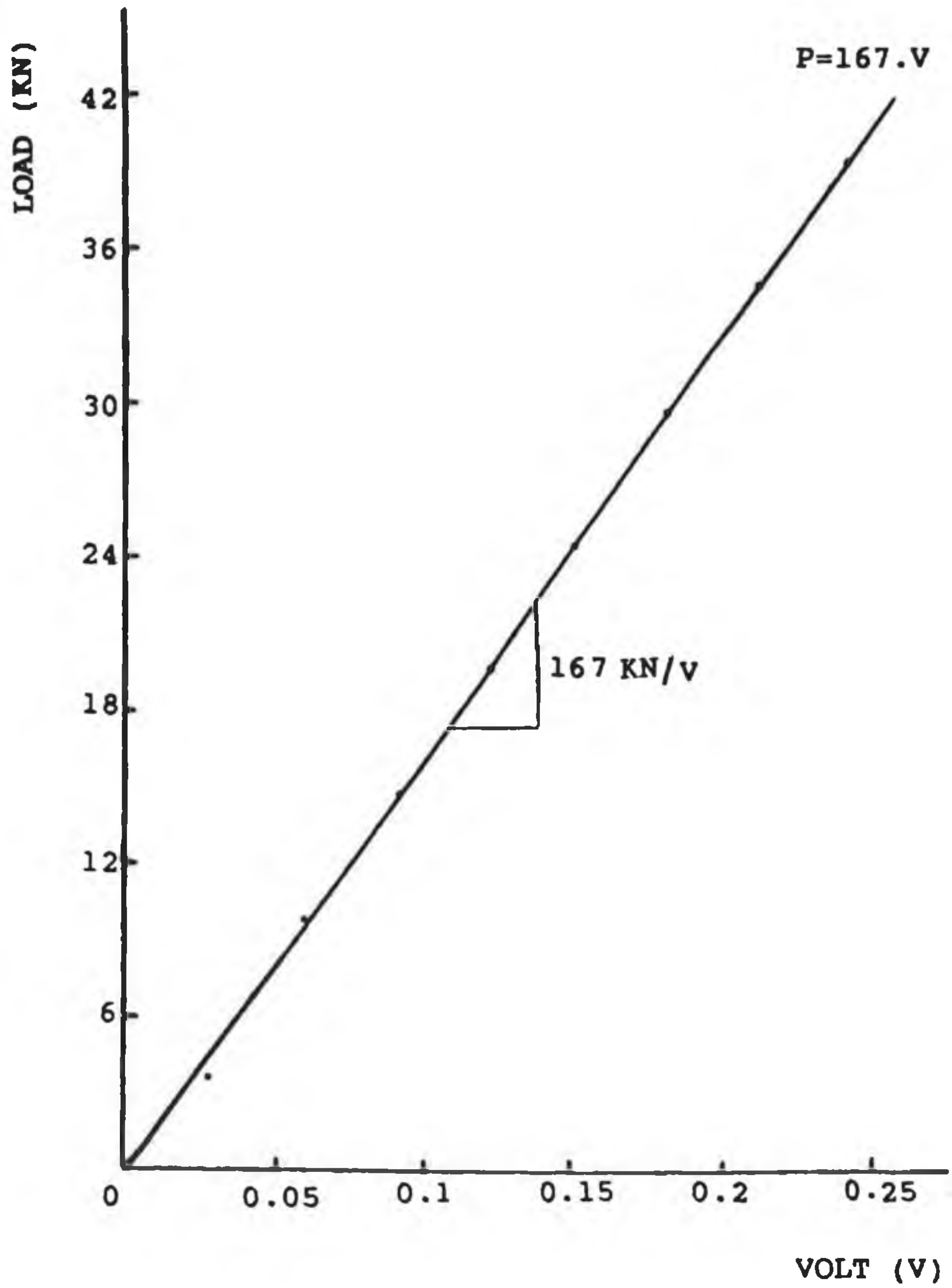


Fig.(4.1) Calibration chart of load cell.

CALIBRATION OF WRENCH TORGUE

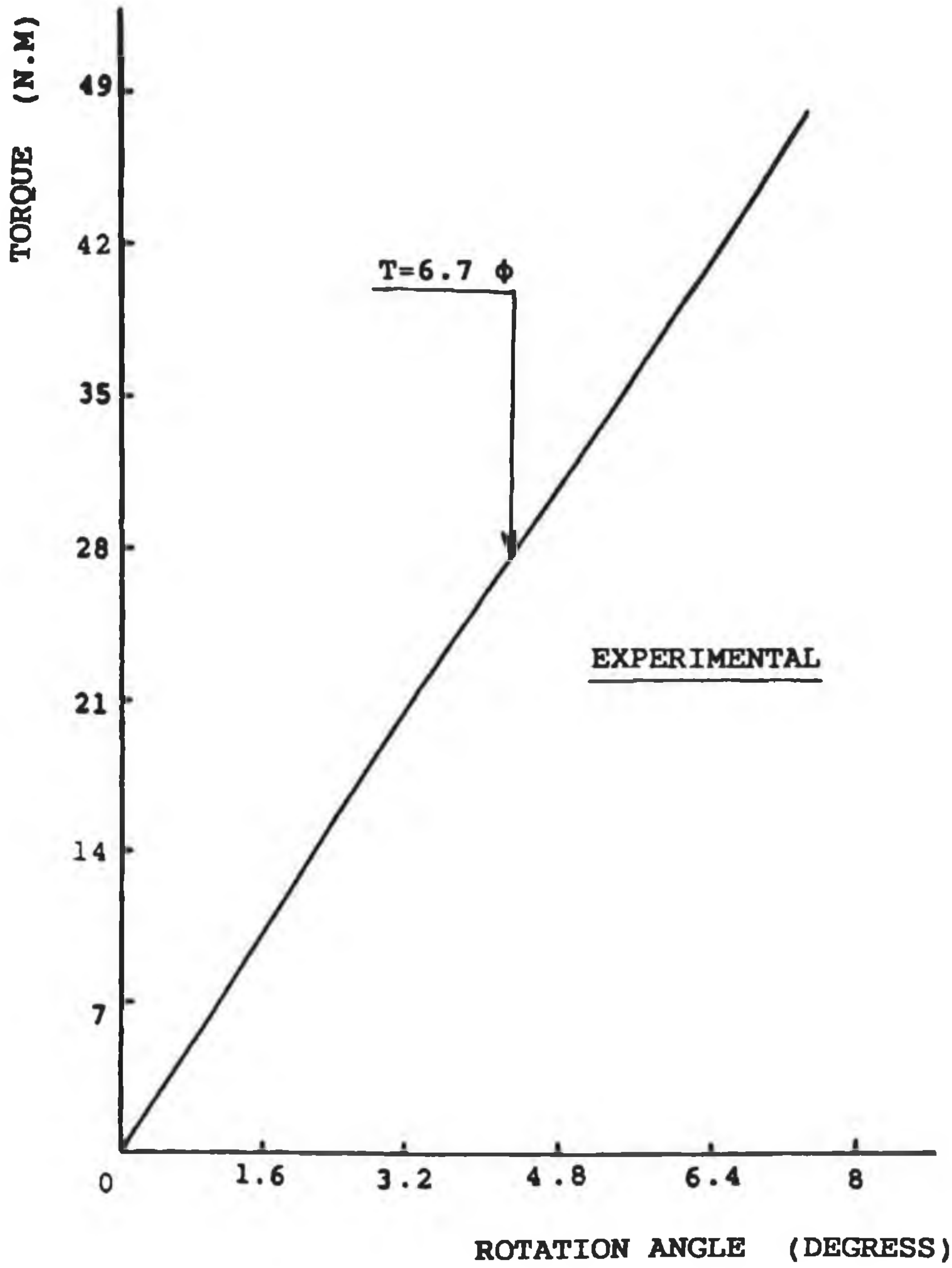


Fig.(4.2) Relationship between torque and rotation angle in the fastener in pure torsion in elastic range.

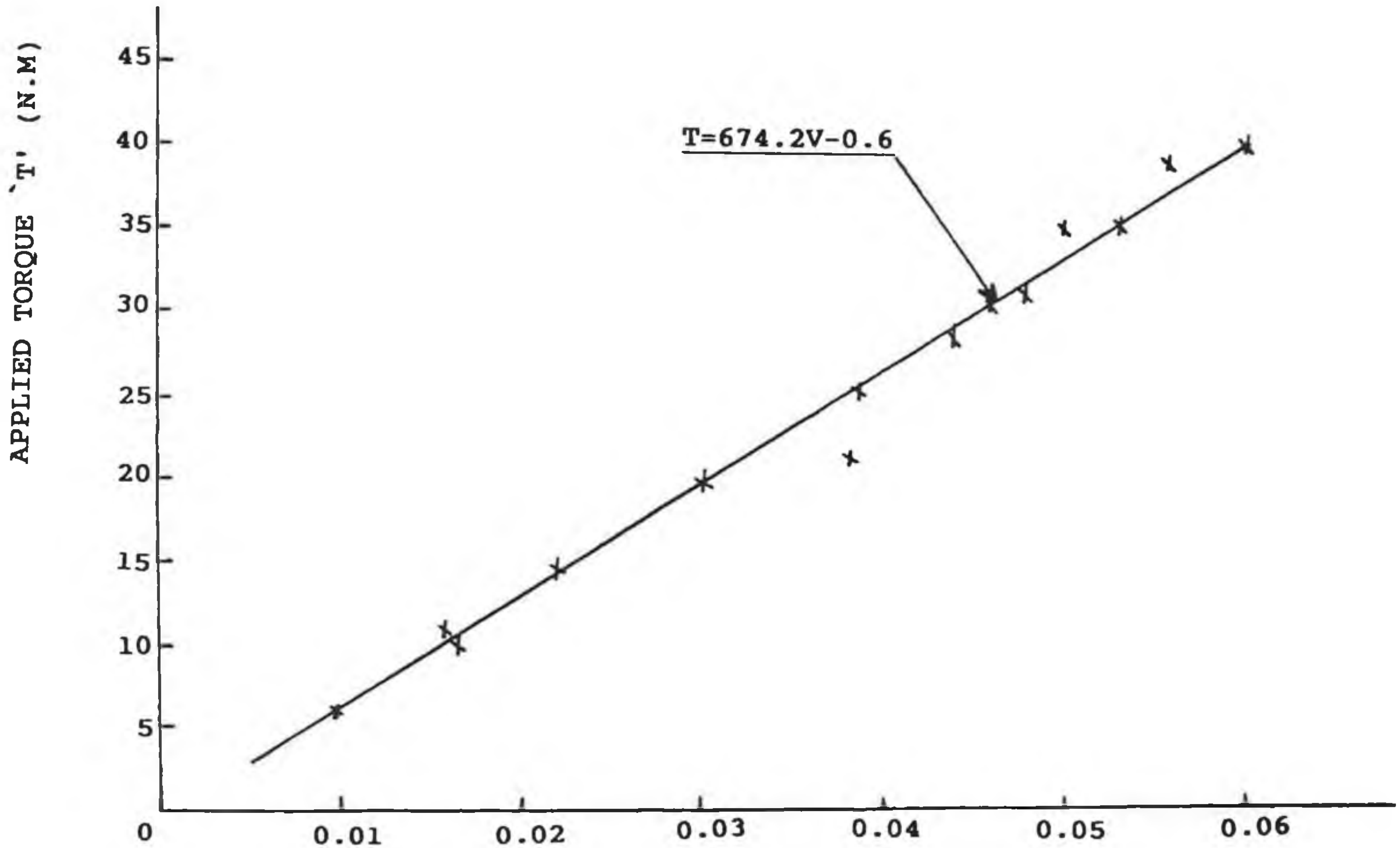


Fig.(4.3) Relationship between the applied torque to the bolt body and the output of strain gauge.

CHAPTER FIVE

RESULTS OF THE EXPERIMENTS

CHAPTER FIVE

RESULTS OF THE EXPERIMENTS

5.1 Introduction

In chapter three, detailed theoretical investigation was made to establish the mechanism in which yielding may take place under combined torque and axial load in a bolt. In this chapter experimental results are presented which were obtained according to the experimental procedures explained in detailed in the previous chapter.

5.2 Principal aims of the experimental work

The aim of these experimental investigation is to establish the following:

1. The response mechanism of the fasteners under uniaxial tensile load when tightening is not involved.
2. Any variation in the yield stress of the bolt under pure axial load when the bolt is subjected to an initial torque and then to the axial load.
3. The mode in which the initially applied torque in the bolt is reduced due to subsequent application of axial stresses.
4. The mode of reduction in initially applied axial load due to the subsequent application of the torque.
5. The mechanism in which the torsional and tensile strain energy is dissipated in the bolt.

In order to establish the above aims this investigation, consisting of five phases, was scheduled according to the following:

Phase one: The fastener is subjected to an axial load to the plastic yielding and then unloaded.

Phase two: The fastener is subjected to a pure torque within elastic range, and then the torque was released.

Phase three: The fastener is initially subjected to a torque before axial tensile load is applied.

Phase four: The fastener is initially subjected to an axial tensile load before the torque is applied.

Phase five: The fastener is initially subjected to an axial load to the plastic yielding and then unloaded. The same bolts is then reloaded initially with a torque and subsequently subjected to an axial load to yield.

The apparatus described in chapter two allowed to apply uniaxial stresses, pure torsion and the combination of tensile and torsional stresses in the bolt. In order to record the torsional stresses induced in the body of the bolt due to the applied torque, strain gauges were attached to the body of the bolt, see Figure (2.14).

5.3 Experimental work

5.3.1 RESULTS OF UNIAXIAL TENSILE TEST

To determine the mechanical properties of the fastener material, a tensile test was performed.

5.3.1.1 ANALYSIS OF RESULTS

Figure (5.1) shows the manner in which the yield load, which is equal to **45.0 kN**, has been determined based on results from a number of tests. the standard deviation for the yield load was calculated as **1.077 kN**. This deviation was due to the difference in the tolerance of the diameters of the test bolts after machining. This tolerance was between **+0.300 mm** and **-0.150 mm**. Some error was also introduced due to slight fluctuation in the load signal recording due to hand pumping of the hydraulic cylinder.

As shown in **Figure (5.1)**, initially the relation between the load and total extension in the fastener, is essentially linear, and the material is still elastic in the sense that deformations are completely recovered, when the load is removed. When the material reaches the yield point, then Hooke's law does not apply, and permanent deformation occurs. The yield point has been determined by drawing a parallel line at a distance equivalent to **0.05 mm** extension from the elastic line [9]. The slope of the load-extension diagram represents the spring constant of the bolt and from the experiment it is found to be equal to

114.6 kN/mm.

Stress rather than force is the more significant parameter in the study of bolt material. **Figure (5.2)** shows the relationship between the true stress and natural strain. This relationship represents results under uniaxial test for the fastener. The slope of a straight line from the origin to the yield point on a stress-strain diagram represents the modulus of elasticity. The yield stress in tension was **900 MPa**, which corresponds to the tensile strength for the bolt material **I.S.O.**, Grade **10.9**, as suggested in **I.S.O. 898/1 1978 [46]**. This indicates the superior strength properties of fastener due to the heat treatment. The curve in **Figure (5.2)** corresponds to the elastic-work hardening responses, that is, the bolt material has strain hardening but only marginally.

The nominal stress at maximum tensile load is known as the tensile strength of the material. Owing to the large reduction in area produced by the necking process, the actual stress at fracture is often greater than the tensile strength see **Plate (5.1)**. Further loading of the bolt caused fracture at the smaller diameter as shown in **Plate (5.2)**. Machining down the diameter of the bolt as shown in **Figure (2.30)** helped to avoid fracture in the threaded section.

Figure (5.3) is showing the relationship between bolt load and extension determined theoretically in each section of the bolt. As shown in this **Figure** at certain value of load the section `0' of the bolt will extend most. Experimentally it was **52.3** percent of total extension in the fastener, while the threaded section has extension equal to **16.5** percent of total extension, see **section 5.3.1.2** .

Figure (5.4) is showing the relation between the axial stress in section **`0'** of the bolt and total extension in the fastener. Experimentally, it is clear from this **Figure** that the yielding is taking place in section **`0'** of the bolt at total extension in the fastener equal to **0.41 mm.**

After each fastener was loaded to its yield point, it was unloaded and loaded a second time. It was found that its yield stress is increased only marginally. As shown in **Figure (5.5)** after loading for the first time, the bolt was unloaded, during unloading the material followed approximately a straight-line. When the load was applied to the bolt for a second time, the material again followed approximately Hooke's law until yielding, the new yield point was **45.72 kN,** has been determined based on results from a number of tests. the standard deviation was **0.822 kN.** Upon reloading a still higher yield point was obtained, the yield load this time was **46.8 kN,** and the standard deviation was **0.48 kN.** It clear from this **Figure** that the total permanent extension in the fastener due to these tests was approximately 2.4 mm.

5.3.1.2 CALCULATION OF MODULUS OF ELASTICITY IN TENSION

In tensile test, the bolt was subjected to the axial tensile load which was effected under bolt head and nut. The equations of equilibrium of forces are the same through each section of the bolt that is,

$$P=P_1=P_2=P_3=P_0. \tag{5.1}$$

Also, the overall change in the length of the bolt is equal to the sum of extensions in each section of bolt [52] that is:

$$\delta=\delta_1+\delta_2+\delta_3+\delta_0. \tag{5.2}$$

Since, it is a simple uniaxial stress system in elastic range, the following equations can be obtained:

$$\sigma_1=E.\epsilon_1 \quad , \quad \sigma_2=E.\epsilon_2 \quad , \quad \sigma_3=E.\epsilon_3 \quad \text{and} \quad \sigma_0=E.\epsilon_0 \tag{5.3}$$

$$\sigma_1=P/A_1 \quad , \quad \sigma_2=P/A_2 \quad , \quad \sigma_3=P/A_3 \quad \text{and} \quad \sigma_0=P/A_0 \tag{5.4}$$

And

$$\epsilon_1=\delta_1/L_1 \quad , \quad \epsilon_2=\delta_2/L_2 \quad , \quad \epsilon_3=\delta_3/L_3 \quad \text{and} \quad \epsilon_0=\delta_0/L_0 \tag{5.5}$$

The equations (5.3) , (5.4) and (5.5) can be re-expressed as follows:

$$\delta_1=P.L_1/E.A_1 \quad , \quad \delta_2=P.L_2/E.A_2 \tag{5.6}$$

And

$$\delta_3=P.L_3/E.A_3 \quad , \quad \delta_0=P.L_0/E.A_0 \tag{5.7}$$

After substituting the values of δ_1, δ_2, δ_3 and δ_0 into the equation **(5.2)**, the modulus of elasticity in tension can be calculated by the following equation:

$$E=N\ (P/\delta) \tag{5.8}$$

Where

$$N=L_1/A_1+L_2/A_2+L_3/A_3+L_0/A_0 \tag{5.9}$$

The constant **N** is dependent on the dimensions of the test bolt. After calculating, the magnitude of the constant **N** was ` **1.902** ', which after substituting in equation **(5.8)** gives:

$$E=1.902\ (P/\delta). \tag{5.10}$$

where, **P** is the bolt load in **kN**, and **δ** is the total extension in the fastener in **mm**, the modulus of elasticity in tension is given in **GPa**. Experimentally, **E** seems to be equal **218.0 GPa**. for most steels **E** is between **200 and 210 GPa** [52,53], the difference between the magnitude of modulus of elasticity could be due to experimental error. When the reduced section of the bolt started to yield, the other sections of the bolt remained still elastic. Then the axial stresses in each section of the bolt, can be calculated by using the following equations:

$$\sigma_1=P_y/A_1 \quad , \quad \sigma_2=P_y/A_2 \quad , \quad \sigma_3=P_y/A_3 \quad \text{and} \quad \sigma_0=P_y/A_0 \qquad (5.11)$$

where P_y is the yield load in section `0' of the bolt. the stress in threaded section of the bolt was calculated to be **65** percent of the yield stress in section `0' of the bolt, see point `1' in **Figure (5.2)**.

It was possible also to calculate the yield load in each section of the bolt, by using the following equations:

$$P_{y1}=\sigma_y \cdot A_1 \quad , \quad P_{y2}=\sigma_y \cdot A_2 \qquad (5.12)$$

and

$$P_{y3}=\sigma_y \cdot A_3 \quad , \quad P_{y0}=\sigma_y \cdot A_0 \qquad (5.13)$$

And also, the extensions for each section of the bolt can be calculated by using the equations **(5.6) and (5.7)**. at certain magnitude of axial load within elastic range, say at **40 kN**, the extensions were obtained as follows:

$\delta_1=0.058$ mm , $\delta_2=0.015$ mm , $\delta_3=0.093$mm , $\delta_0=0.183$ mm

As shown in **Figure (5.3)**, this calculation follows that,

$$\delta_0>\delta_3>\delta_1>\delta_2. \qquad (5.14)$$

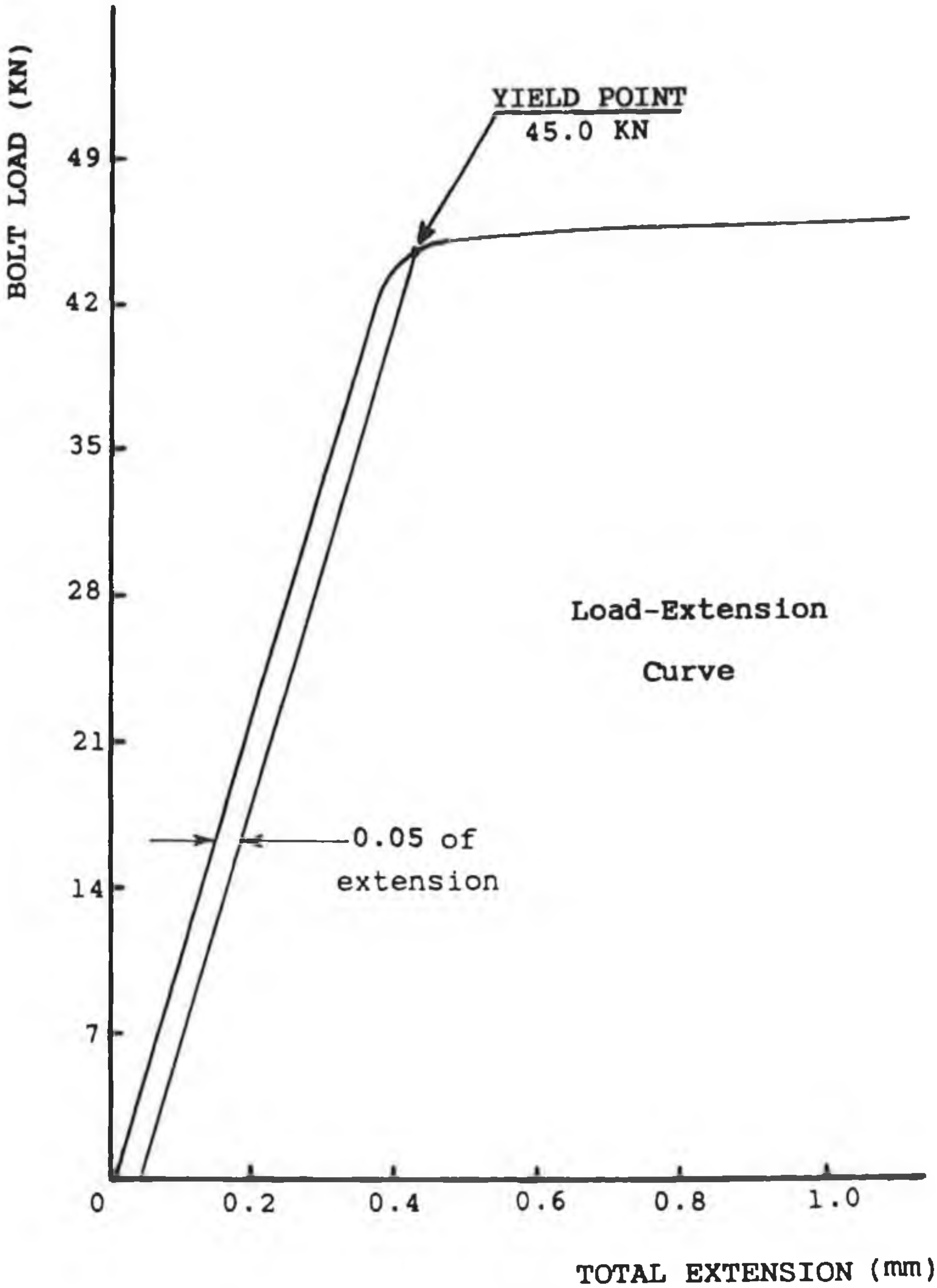


Fig.(5.1) Relationship between load and extension in the fastener (Uniaxial curve).

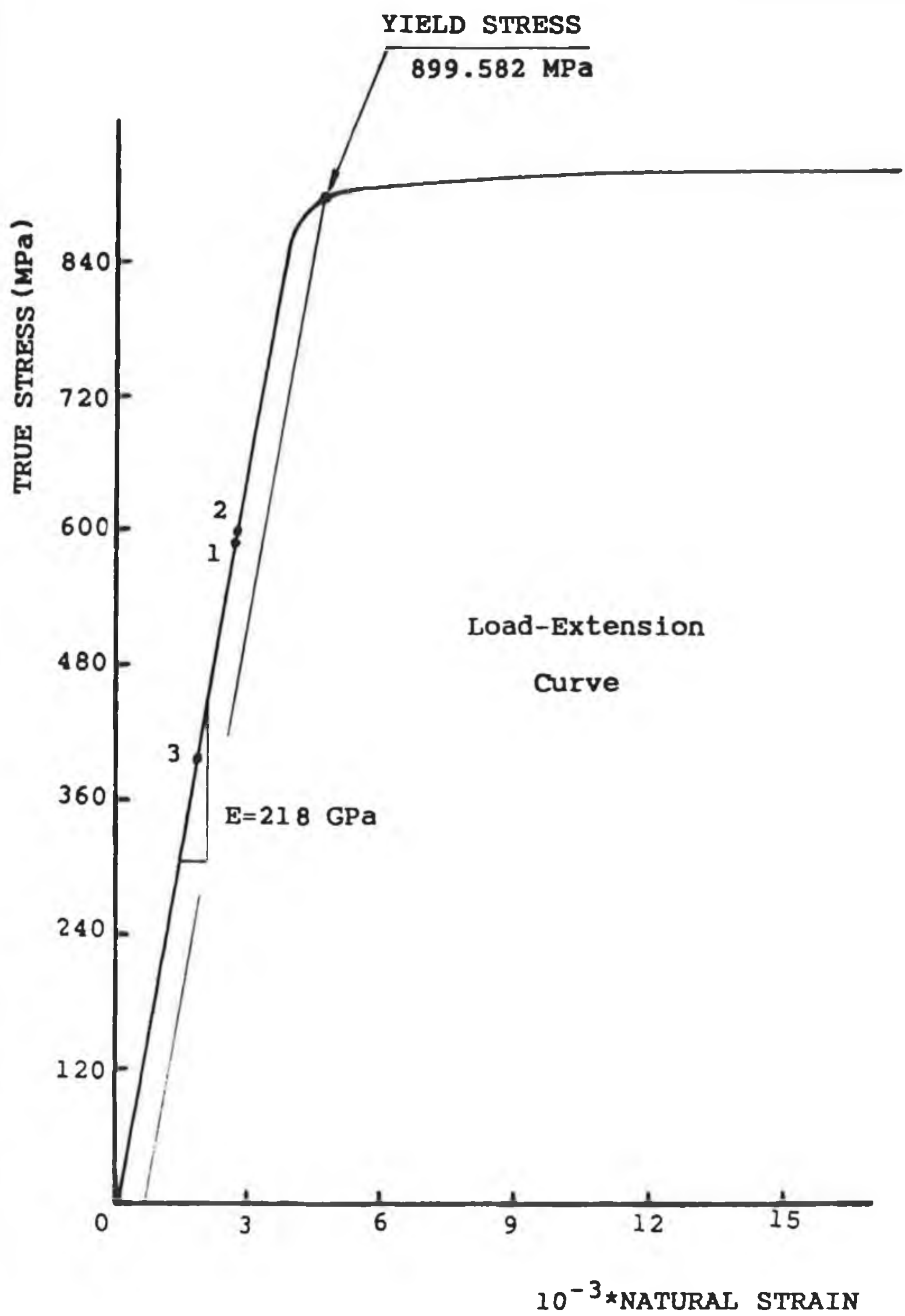


Fig.(5.2) Relationship between true stress and natural strain in the fastener (Uniaxial curve).

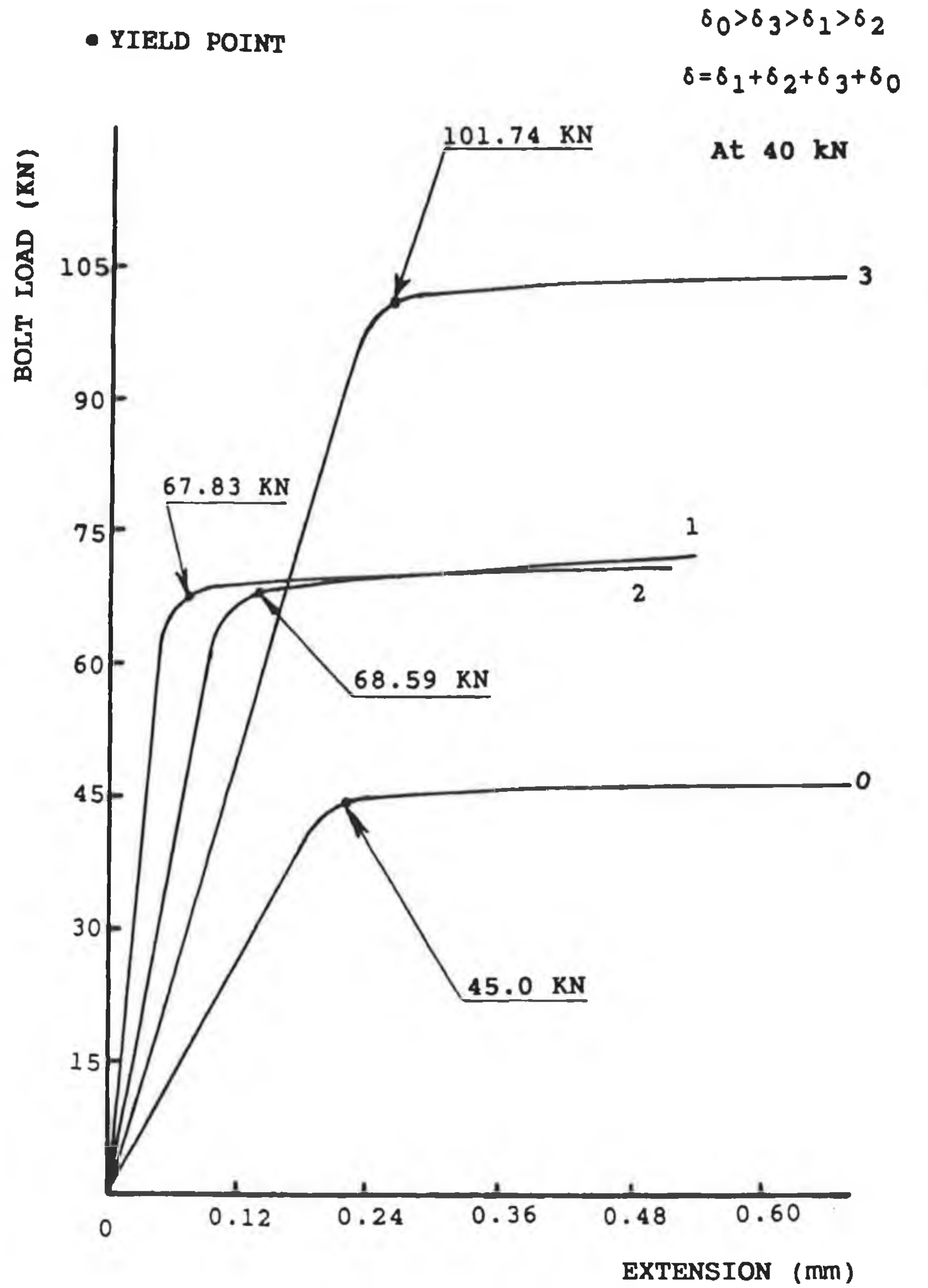


Fig.(5.3) Relationship between load and extension in the fastener for each section of the bolt

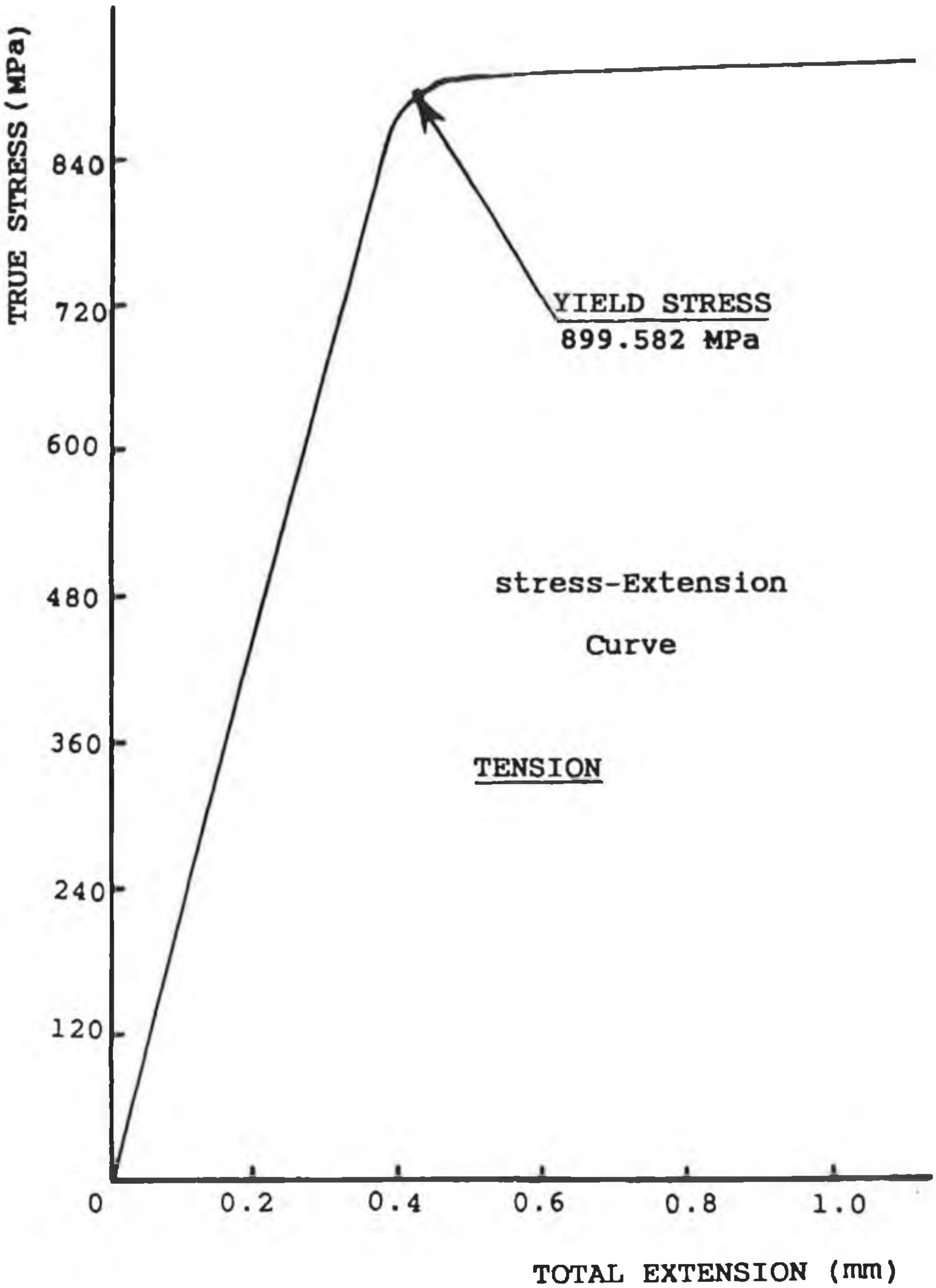


Fig.(5.4) Tensile stress against extension of the test bolt under service load.

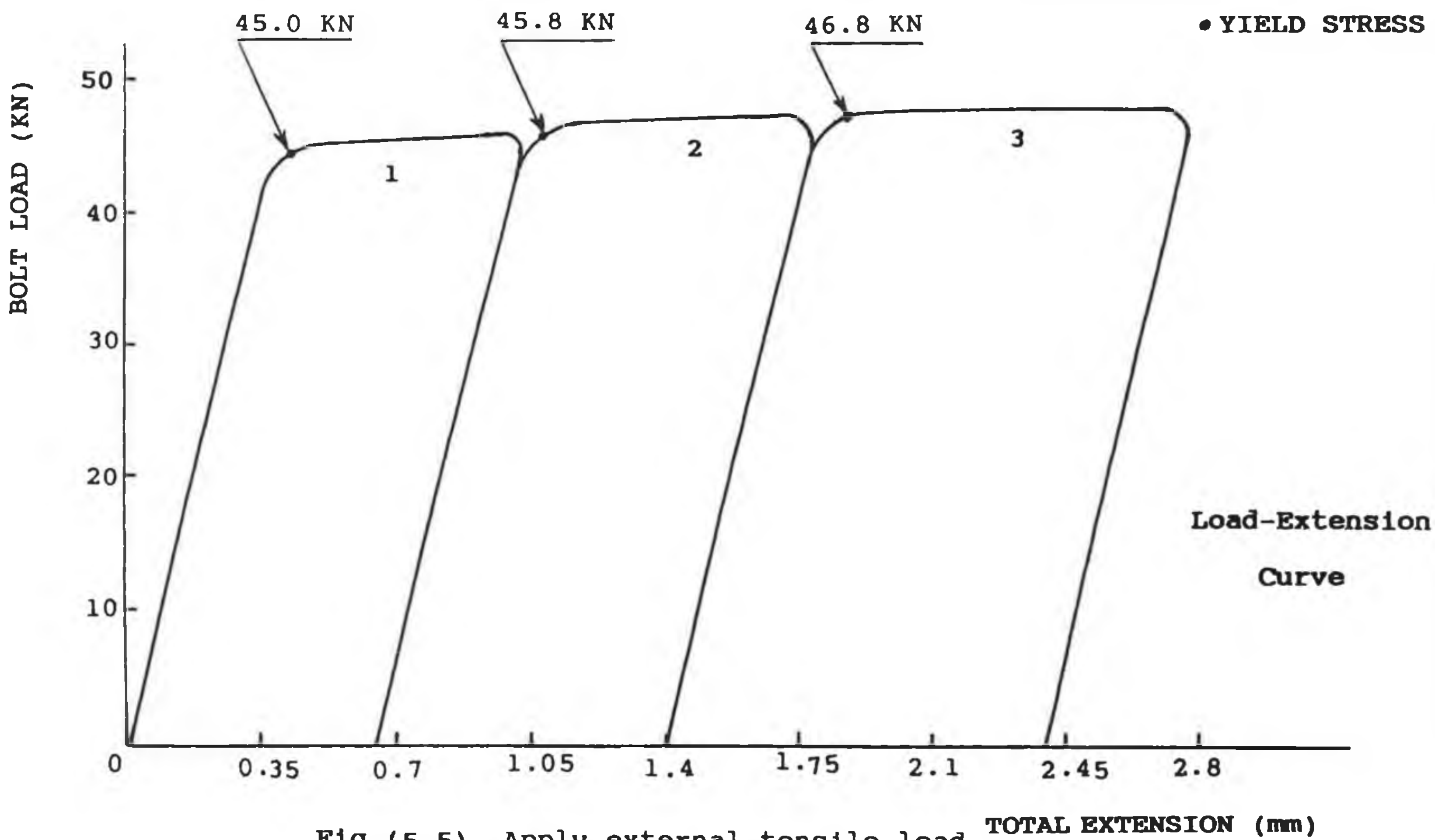


Fig.(5.5) Apply external tensile load three times to the fastener after unloading.

Plate (5.1) UNIAXIAL TENSILE TEST

CHAPTER FIVE

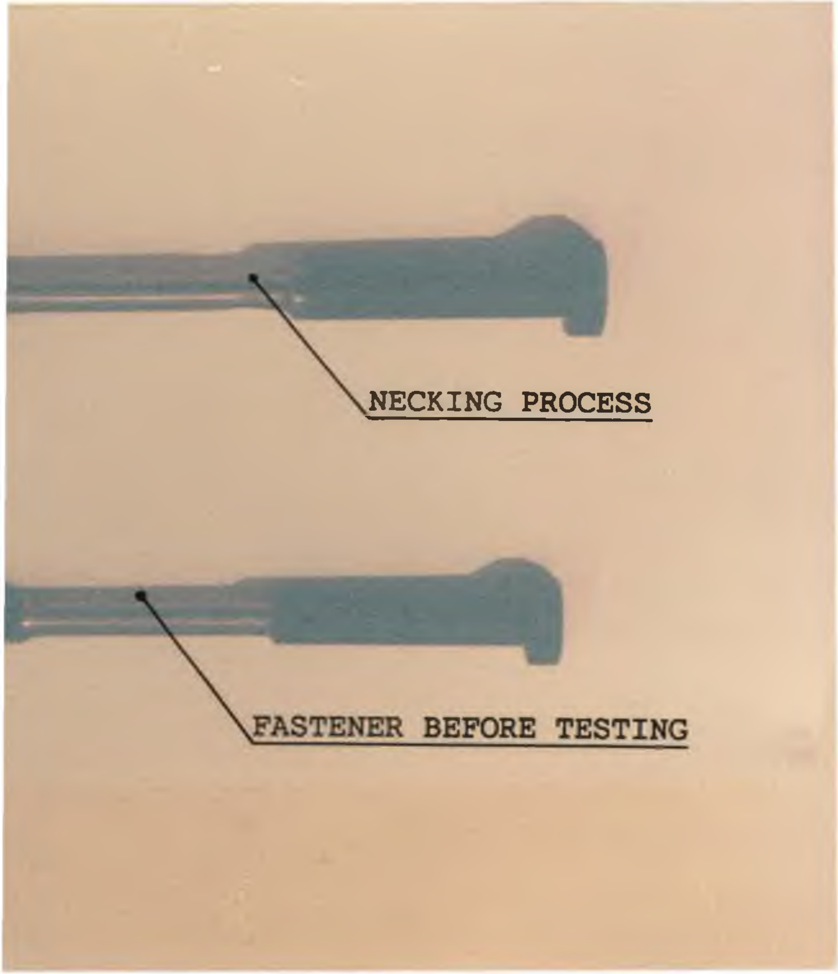
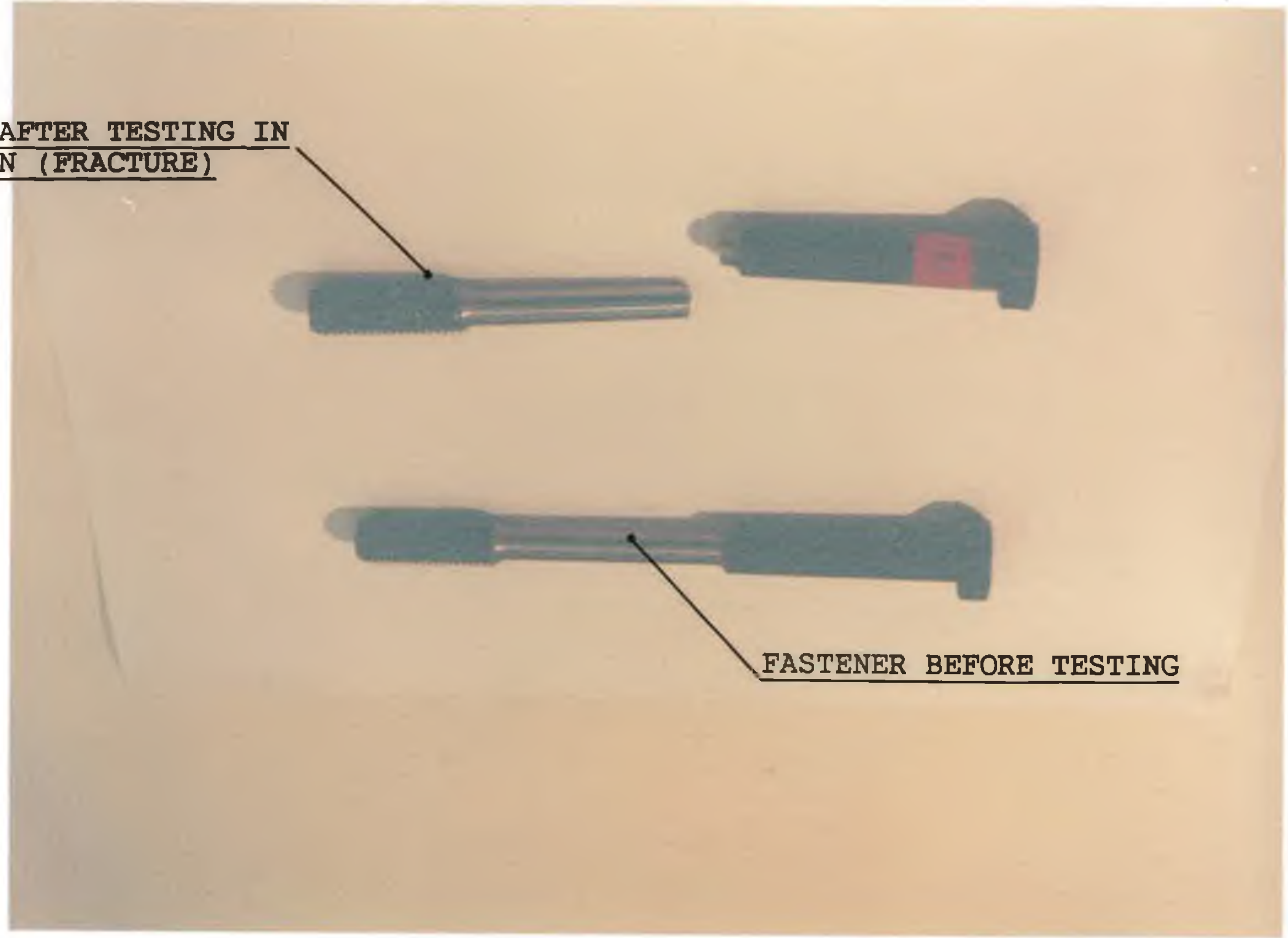


Plate (5.2) MAXIMUM LOAD WAS APPLIED TO THE FASTENER



5.3.2 RESULTS OF PURE TORSION TEST

This section presents the results of detailed investigation of the behavior of the bolt under pure torsion. Normally, the initial load in the bolt is applied by tightening the nut. This initial load in the bolt is proportional to the wrenching torque applied to the bolt head. The principal aim of the torsion test is to apply pure torsion to the bolt head without any initial load thus, preventing any extension in the fastener until external load is applied. The pure torsion was applied to the bolt by means of a torque wrench which caused the shank to rotate in relation to the threaded section which is prevented from rotation.

5.3.2.1 ANALYSIS OF RESULTS

The slope of the torque-angle of twist diagram is called the torsional spring constant, experimentally it seems to be equal to **6.7 N.M/Degree.** **Figure (5.6)** shows a comparison between the theoretical and experimental results of calibration of the torque wrench. The discrepancy between these results could be due to the variation in strength of the bolt material, tolerance of bolt shank after machining, and also due to the fact that the torque was applied manually. The reading of the angle of rotation due to the applied torque is given as integral value of the angle, so that at low level of torque `say less than **15 N.M'** the measurement of the angle of twist was not so accurate. The standard deviation between the theoretical and experimental results was found to be between 0 and 0.35 Degrees.

Figure (5.7) is showing the relation between the torque and the angle of twist in each section of the bolt. Within the elastic range, this relation is linear. From the torque-angle of twist diagram the following equation can be written:

$$\Phi_0>\Phi_1>\Phi_3>\Phi_2. \tag{5.15}$$

It is clear from the above that the maximum angle of twist is in the section `0' and the minimum angle of twist is in the section `2' of the fastener. The angle of twist in the threaded section of the bolt is approximately 15.4 percent of the total angle of twist in the fastener.

Figure (5.8) is showing the shear stress and shear strain diagram at the outside diameter of the reduced section of the bolt within elastic range.

The slope of a straight line represents the modulus of rigidity . From the experiment it is clear that the maximum elastic torque capacity of the bolt material is 52.21 N.M , and the shear yield stress is to be taken from`von Mises Expression' for yield criteria [52] as 0.577 times the yield stress in simple tension. Hence, the yield stress in torsion is 519.374 MPa for the bolt material. It is clear that the section `0' of the bolt will start yielding before any other sections. When the bolt starts to yield in torsion it is calculated that the shear stress in the threaded section is approximately **54** percent of the yield stress in torsion. The magnitude of the torque for the threaded section to start to yield is **98 N.M** when the bolt is subjected to pure torsion.

Since the ultimate torque is **4/3** times the yield torque, only **33.33** percent of the torque capacity remains after yield-point shearing stress is reached at the extreme fibres of a bolt [53]. Experimentally the ultimate torque was detected by SPS torque wrench as **69.5 N.M**, after the ultimate torque, the bolt was fractured in torsion. At this torque the bolt section of diameter **8 mm** is twisted so that a core only of **6 mm** diameter remains elastic as shown in Figure (5.9).

Plate (5.3) shows three samples of bolts which were tested in pure torsion. These bolts were marked by straight black line on the bolt shank **`at 8 mm diameter'** before testing. One of them was subjected to elastic torque, and the other two bolts were subjected to the torque beyond the plastic range but at different values of torque. This plate shows the difference between the elastic and inelastic twists, illustrating the residual rotation of the bolt.

5.3.2.2 CALCULATION OF MODULUS OF RIGIDITY

Under pure torsion, the applied external torque in each section of the fastener is equal, that is:

$$T=T_1=T_2=T_3=T_0. \tag{5.16}$$

But the total angle of twist in the bolt is made up of those in each section . In this case the following equation can be written:

$$\phi=\phi_1+\phi_2+\phi_3+\phi_0; \tag{5.17}$$

Since, it is a pure torsional stress system in elastic range, the following equation can be obtained:

$$\phi_1=T.L_1/G.I_{p1} \quad , \quad \phi_2=T.L_2/G.I_{p2} \tag{5.18}$$

And

$$\phi_3=T.L_3/G.I_{p3} \quad , \quad \phi_0=T.L_0/G.I_{p0} \tag{5.19}$$

By calculating the angle of twist in each section of the bolt, and substituting each value of ϕ in equation (5.17), the modulus of elasticity in shear can be found as:

$$G=N0.(T/\phi). \tag{5.20}$$

Where

$$N_0=L_1/I_{p1}+L_2/I_{p2}+L_3/I_{p3}+L_0/I_{p0} \tag{5.21}$$

The constant 'N_0' is dependent on the dimensions of the bolt. After calculating, the magnitude of the constant N_0 was ` 0.189 ', which after substituting in equation (5.20) gives:

$$G=0.189\ (T/\phi). \qquad (5.22)$$

As shown in Figure (5.7), this calculation follows the equation (5.15).

Where, the total angle of rotation is measured in radian, torque in N.M, dimensions in mm and the modulus of elasticity in shear is measured in GPa. The magnitude of the modulus of elasticity in shear was obtained experimentally to be 72.57 GPa, it is very close to the theoretical value, which is between 77 and 83 GPa for structural steel [50,51]. The modulus of rigidity and Young's modulus are related by the equation:

$$G=E/[2(1+\nu)]. \qquad (5.23)$$

Where ν is the Poisson's ratio. Provided that the load on the bolt material is retained within the elastic range, the ratio of the lateral and longitudinal strain is always constant, this ratio is termed Poisson's ratio:

$$\nu=\epsilon_{lat}/\epsilon_l; \qquad (5.24)$$

Where,

$$\epsilon_{lat}=\delta_d/d. \quad \text{and} \quad \epsilon_l=\delta_l/L. \qquad (5.25)$$

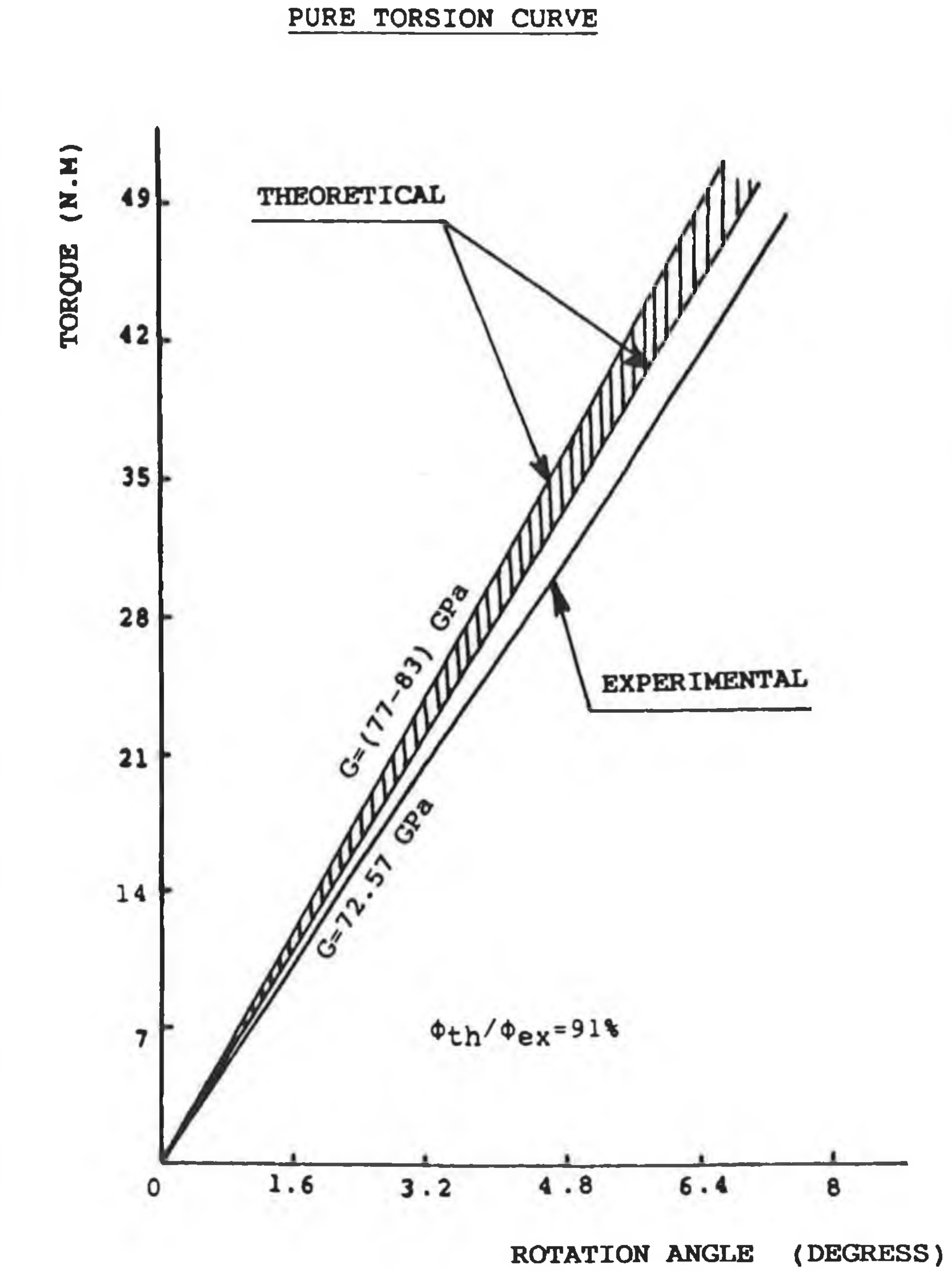


Fig. (5.6) Comparison between torque and rotation angle in the fastener theoretical and experimental (within elastic range).

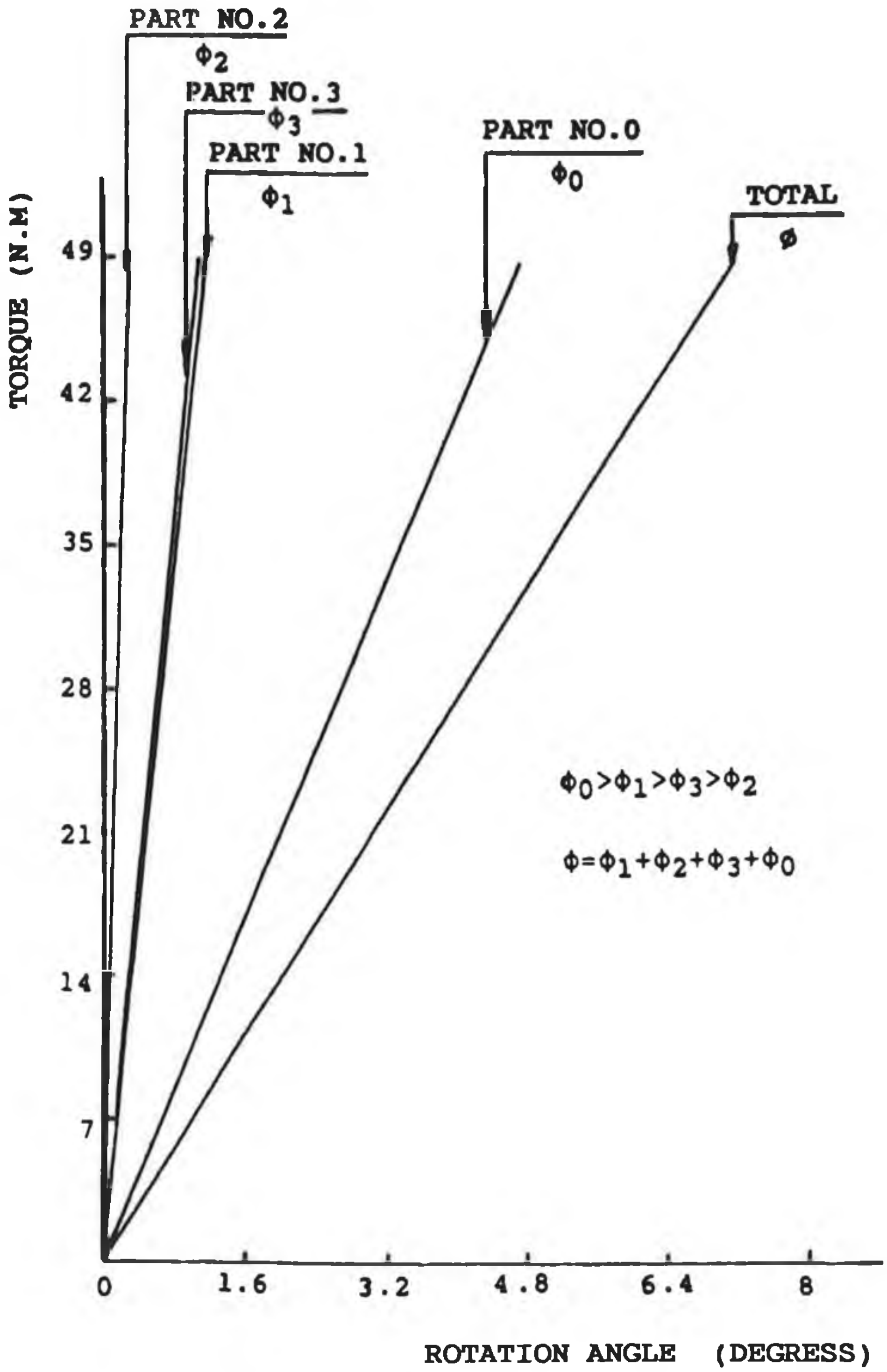


Fig.(5.7) Torque versus rotation angle for each part of bolt in pure torsion (elastic range).

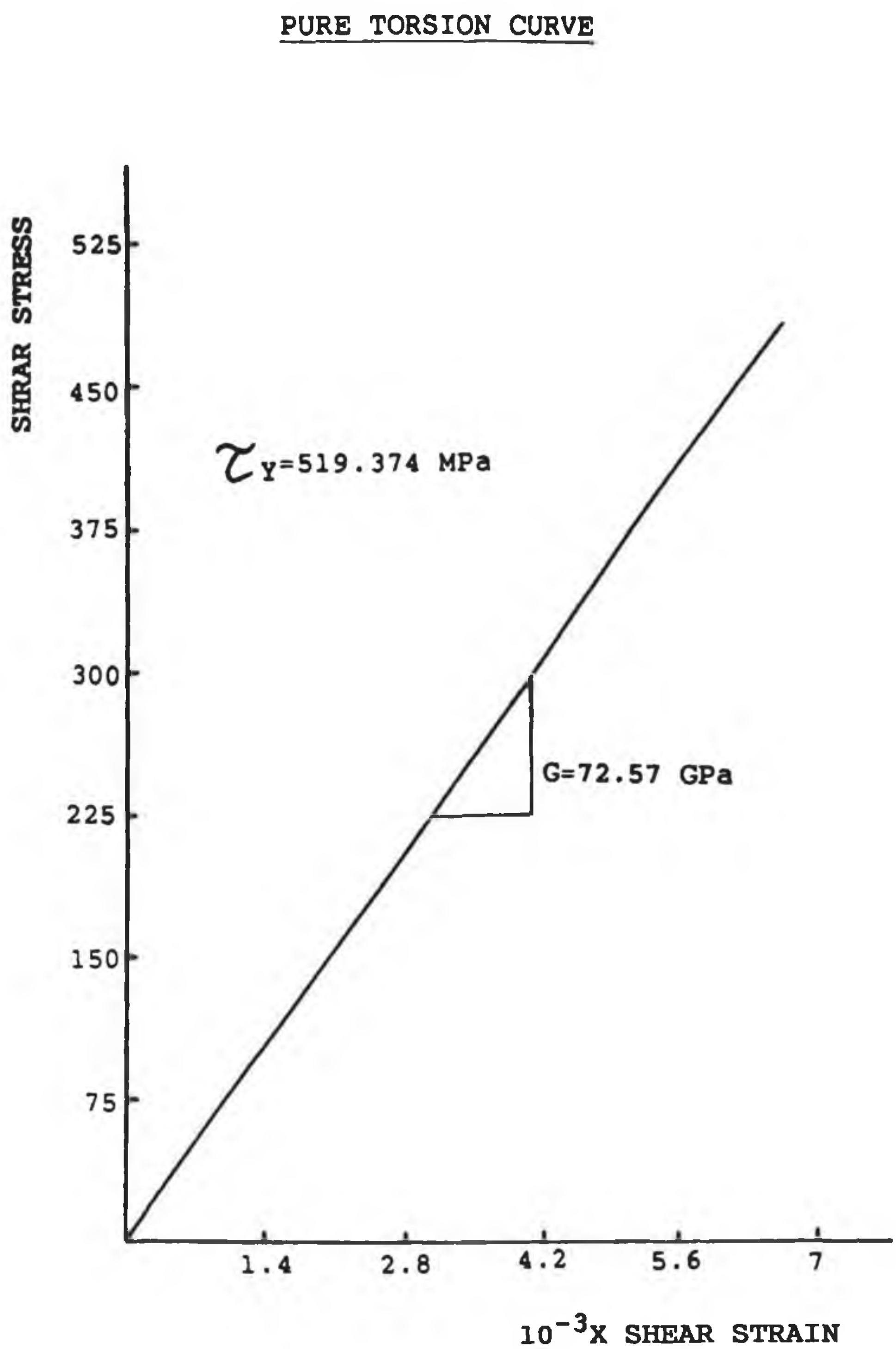


Fig.(5.8) Shear stress versus shear strain in elastic range.

$y = d/d_e \cdot \tau_{max}$, $d=8$ mm , $d_e=6$ mm , $d/d_e=4/3$

e Elastic

p partially plastic

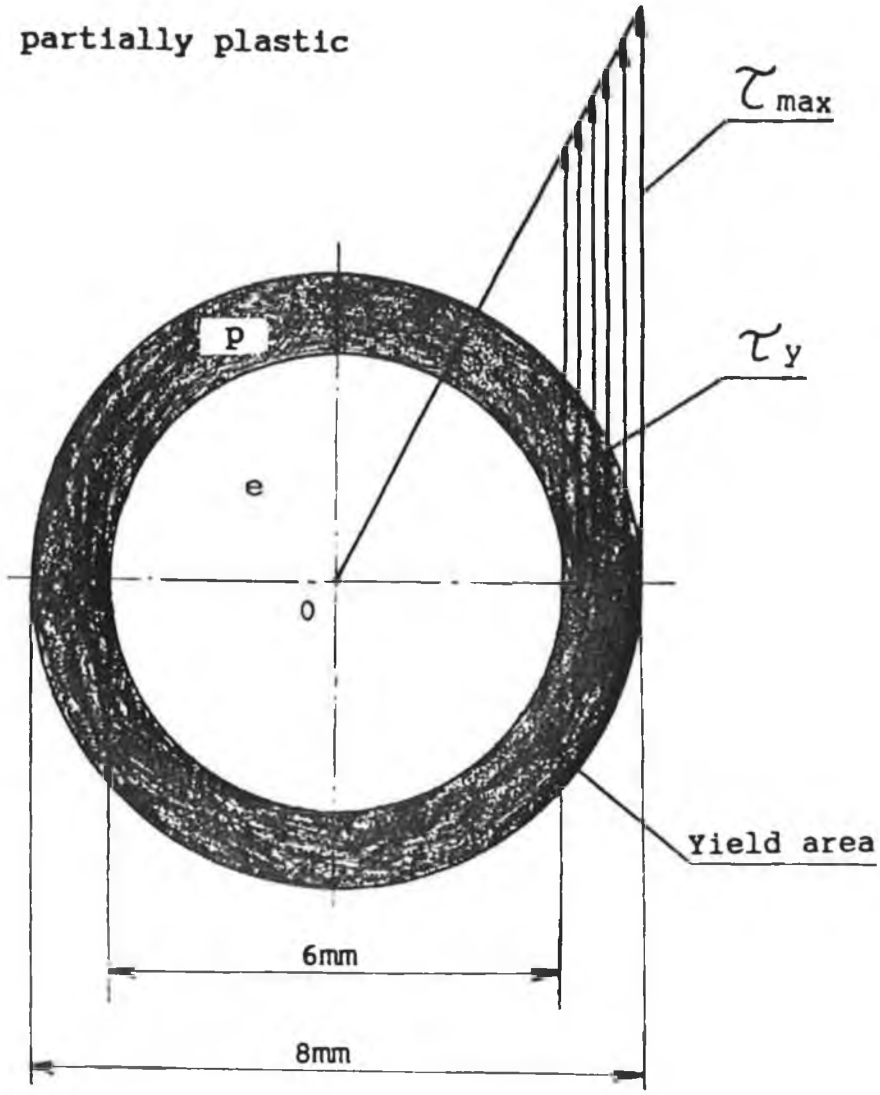
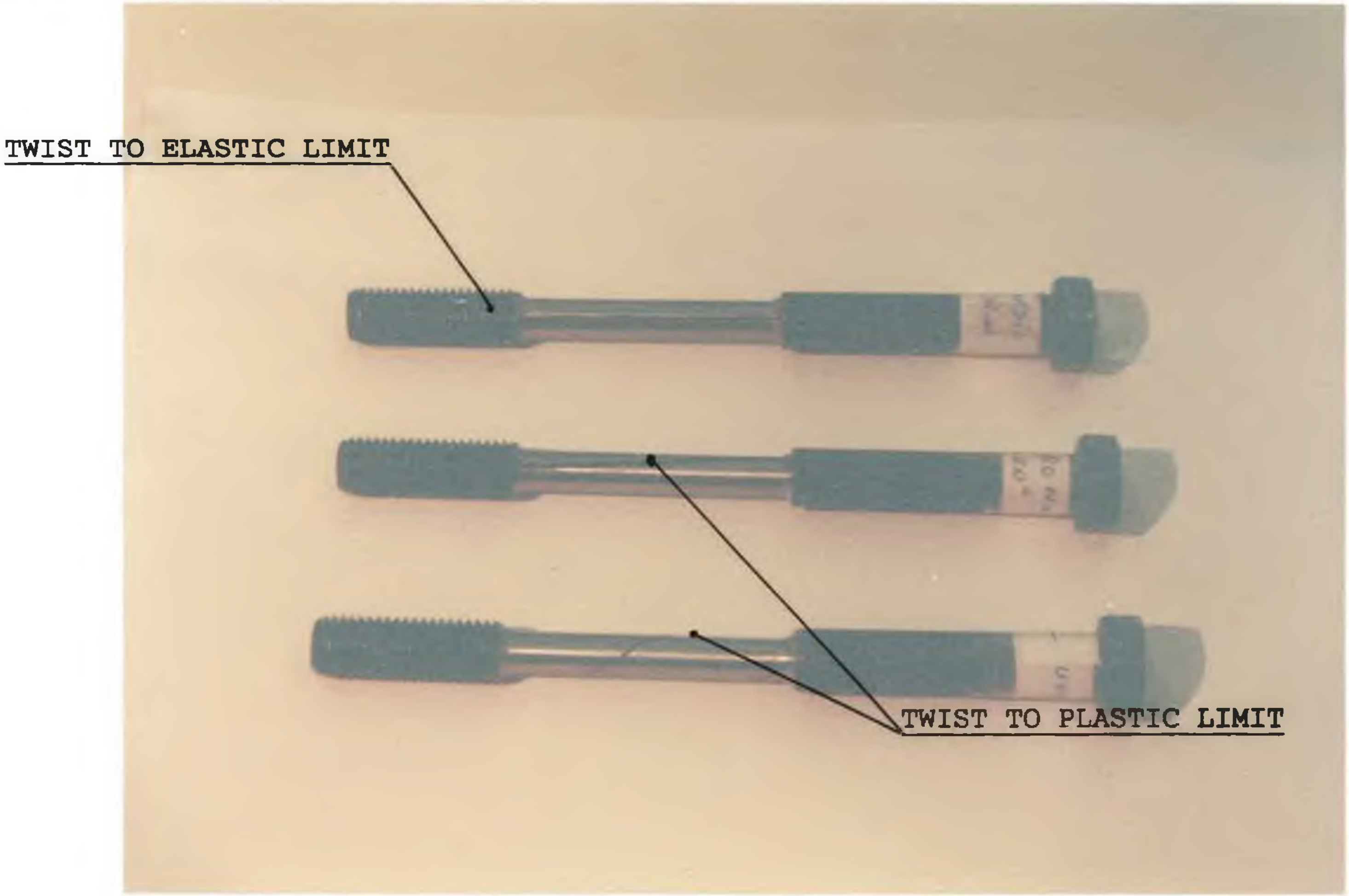


Fig.(5.9) Elastic-Plastic stress distribution in the fastener

Plate (5.3) BOLTS ARE TESTED IN PURE TORSION



5.3.3 RESULTS OF COMBINED TORQUE AND AXIAL LOAD TEST

5.3.3.1 THE BOLT INITIALLY SUBJECTED TO A TORQUE

In this section we will attempt to establish experimentally the effect of applying axial stress in the bolt initially subjected to a torque.

Figure (5.10) upto Figure (5.16) illustrate the relationship between the torque and tension forces in the bolt as the external tensile load is applied. The magnitudes of the applied torque were between **22.4 to 42.7 N.M** in elastic range, which were measured by strain gauge.

These figures are showing that as the bolt extends under the external tensile load the initially applied torsional stress remains unchanged until the combination of the axial and torsional stresses reaches the yield stress in tension of the material of the bolt. When the combined stress in the bolt becomes equal to the yield stress in tension the material starts to yield at the outside diameter and to satisfy the yield criteria the torsional stress decreases rapidly. Finally only a small torque would be present in the bolt. The torque remaining in the bolt after the axial load was removed is the residual torque. It is clear from these figures that the torque starts to reduce before the material of the bolt starts to yield in tension. This is due to fact that the combined stress in the bolt becomes equal to the yield stress in tension well before the applied tensile load causes yielding on its own.

The experimental results show that at the point of tensile yield the torque was reduced to between **30 to 50** percent of the total applied initial torque to the bolt. The residual torque remaining in the bolt after the release of the axial load were between **21 to 43** percent of the total applied initial torque.

5.3.3.2 THE BOLT INITIALLY SUBJECTED TO AN AXIAL LOAD

In this section we will attempt to establish experimentally the effect of the application of torsional stress after the bolt was initially subjected to an axial load.

Figures (5.17) illustrate the relationship between the initially applied axial load and subsequent torque applied to the bolt. It is seen that when the bolt is initially subjected to the axial load and then to the torque in elastic range, no deformation will take place until the combination of tensile and torsion stresses becomes equal to the yield stress in tension of the material of the bolt. However, when the combined stress is greater than the yield stress the yield criteria becomes applicable and the axial load starts to reduce. However, the level of reduction of axial load due to the application of torque is much lower than the level of the reduction of the initially applied torque due to subsequently applied axial load as shown in the previous section.

5.3.3.3 EFFECT OF COMBINED STRESS ON THE YIELD POINT

This section presents the results of detailed investigation of the behaviour of the bolt under pure axial load, when the bolt is initially subjected to an initial torque and then to an axial load.

Figures (5.18) and (5.19) illustrate the relationship between the axial load and total extension in the fastener. This relation represents results under successive loading sequence. After the bolt was subjected to uniaxial load to plastic yielding, it was unloaded, see **Curve** \`1' in **Figures (5.18)** and **(5.19)**, and subsequently subjected to combined axial load and torque, see **Curve** \`2' in **Figures (5.18)** and **(5.19)**. The magnitudes of the applied torque were between **27 to 28.8 N.M** within elastic range, which were measured by strain gauge. It is clear from these **Figures** that there is a small reduction in the yield load when a pre-torque is applied. This reduction however is very small, so that the reduction in the yield load can be neglected by comparing with the original magnitude of the yield load, see **curve** \`2' in **Figures (5.18) and (5.19)**. When the load was applied to the bolt for a third time, the material again followed approximately Hooke's law until yielding, the new yield point was increased only marginally see **curve** \`3' in **Figures (5.18) and (5.19)**.

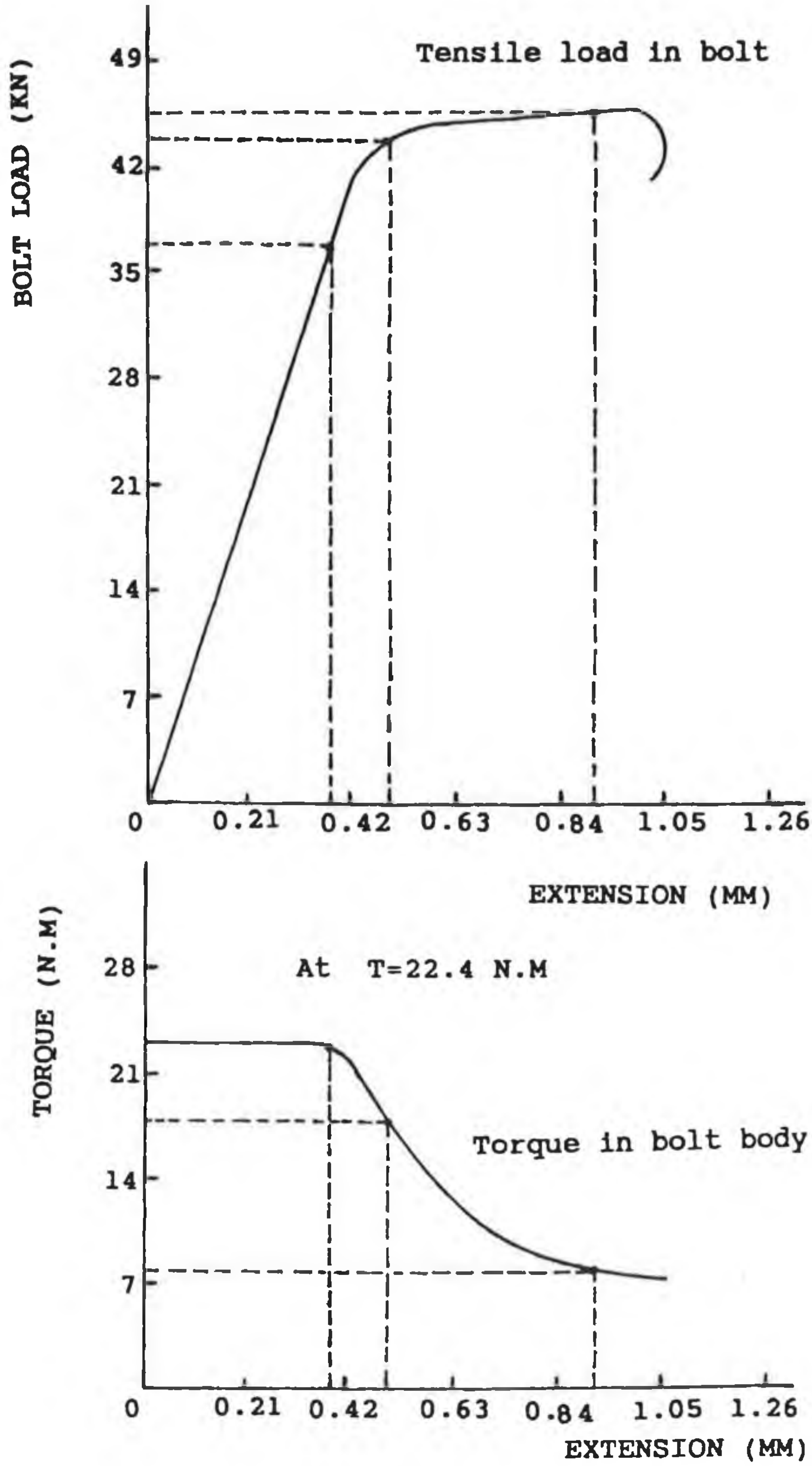


Fig.(5.10) Torque and tension forces in the bolt body as the external load is applied.

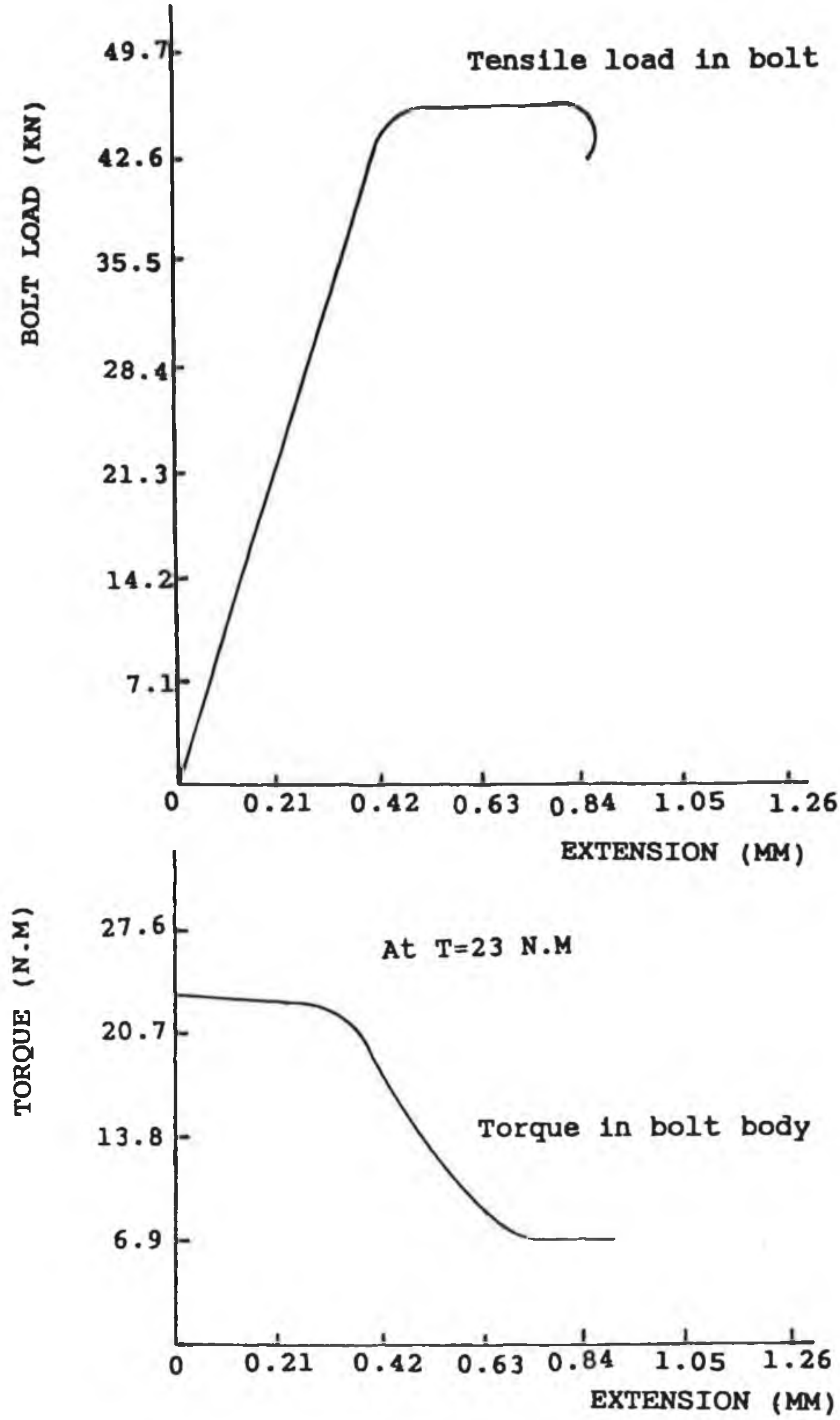


Fig.(5.11) Torque and tension forces in the bolt body as the external load is applied.

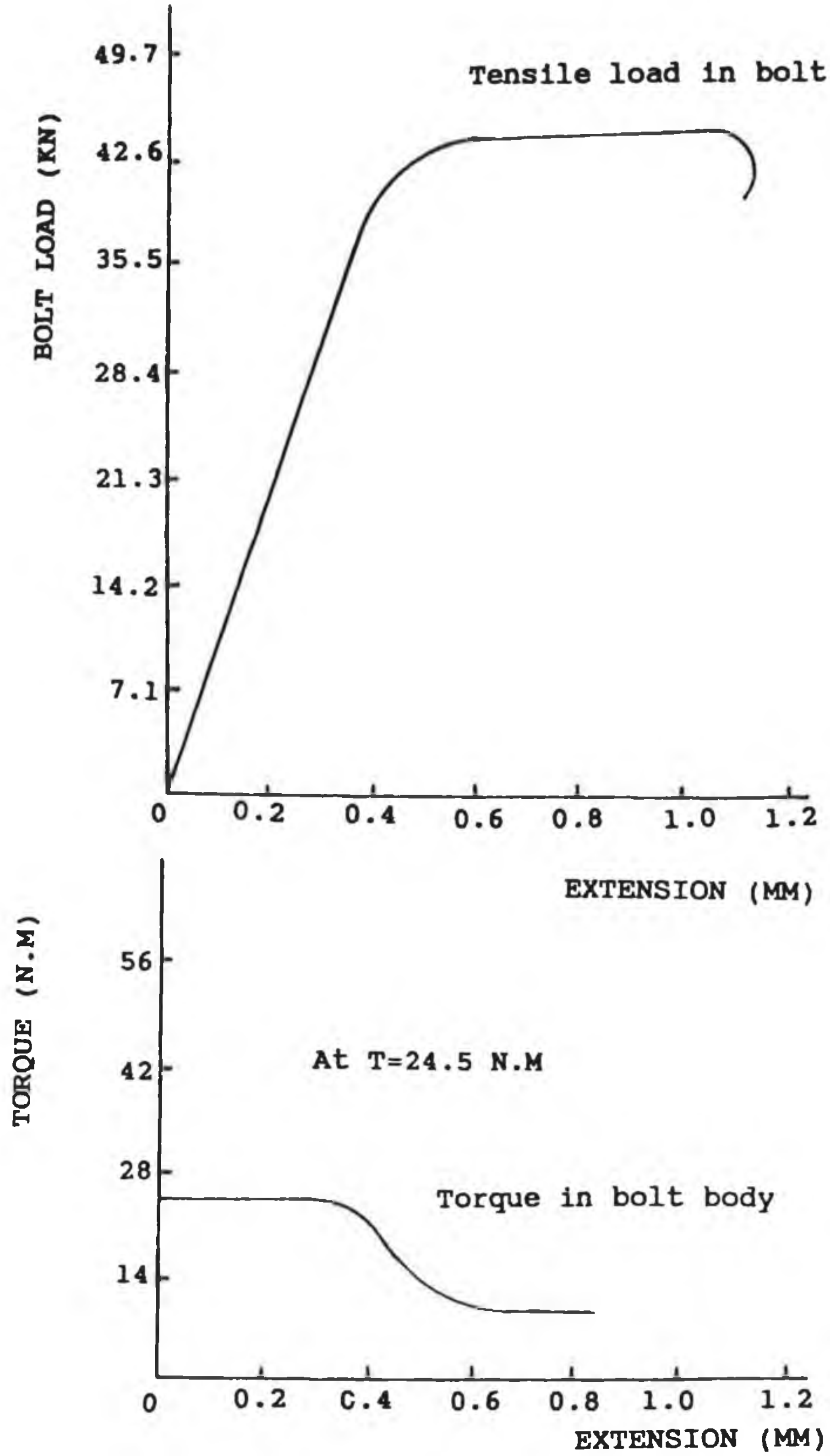


Fig.(5.12) Torque and tension forces in the bolt body as the external load is applied.

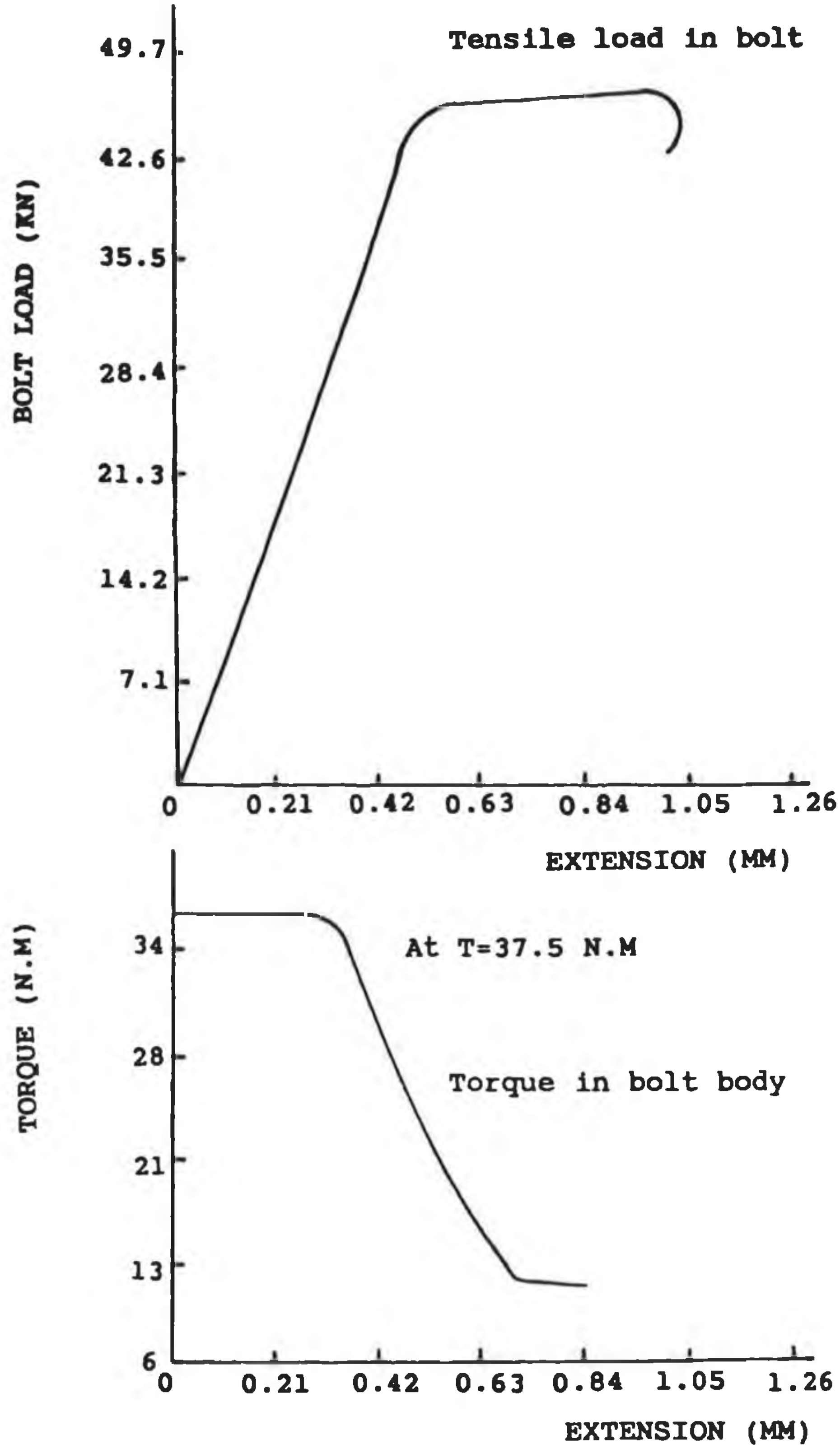


Fig. (5.13) Torque and tension forces in the bolt body as the external load is applied.

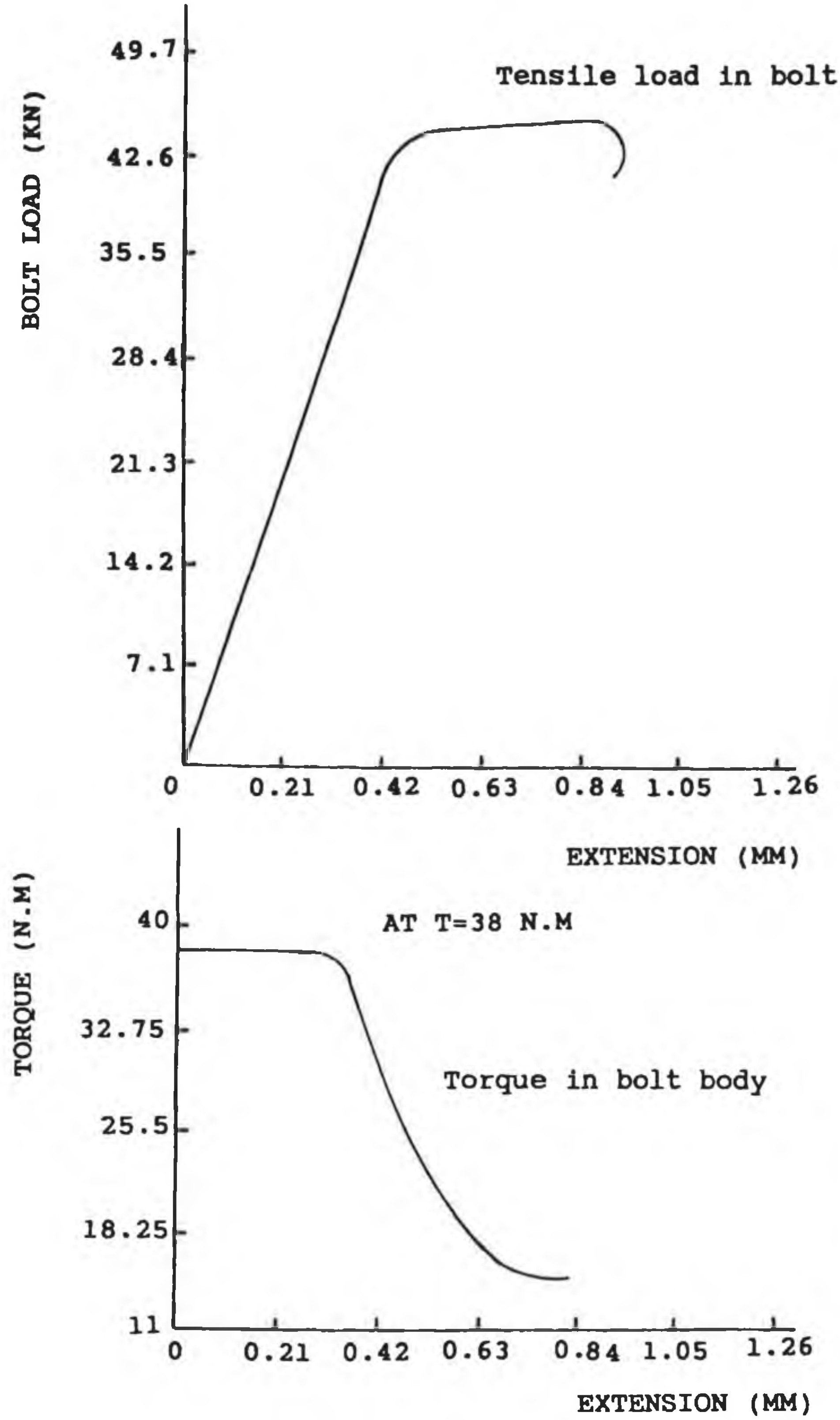


Fig.(5.14) Torque and tension forces in the bolt body as the external load is applied.

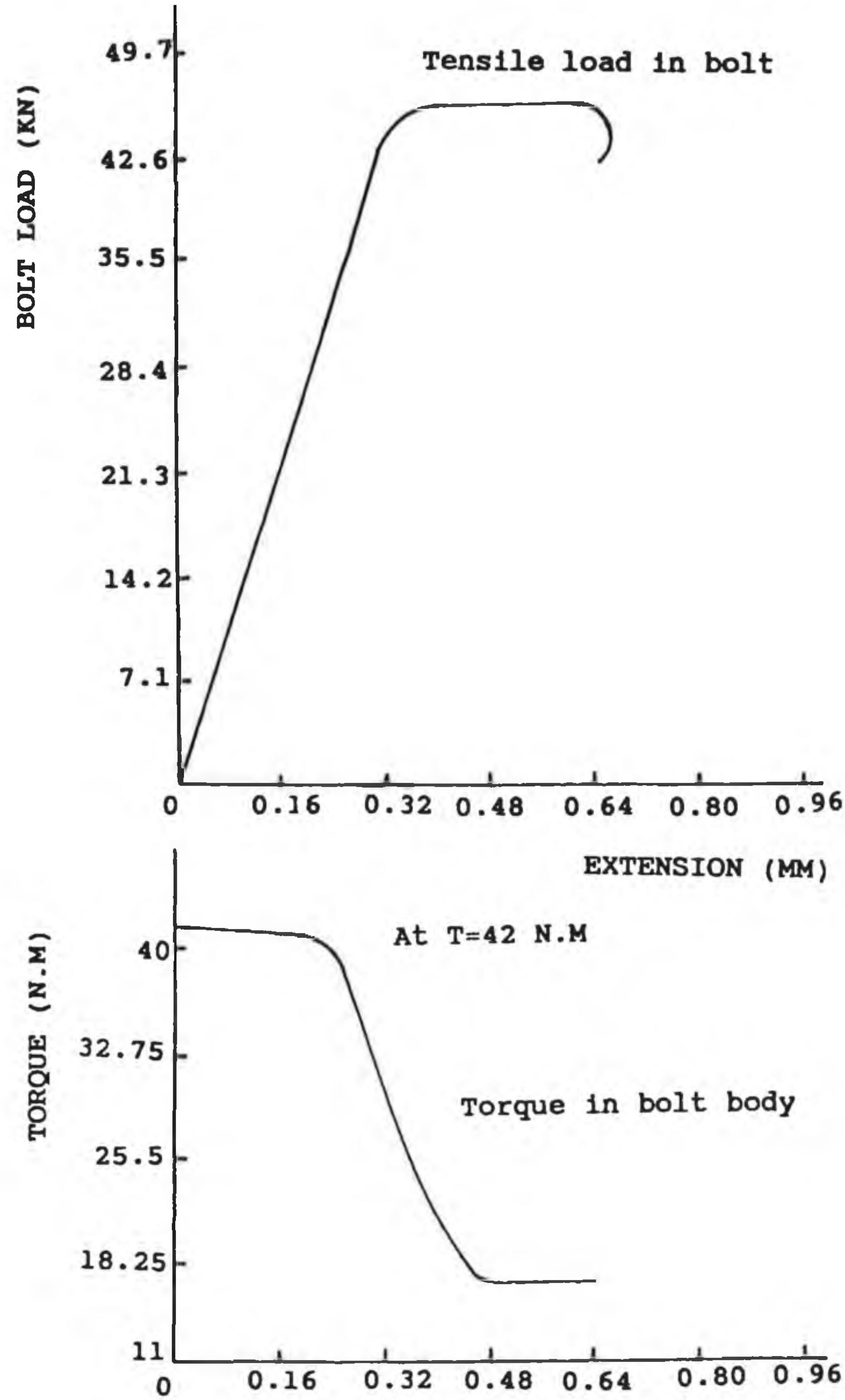


Fig.(5.15) Torque and tension forces in the bolt body as the external load is applied.

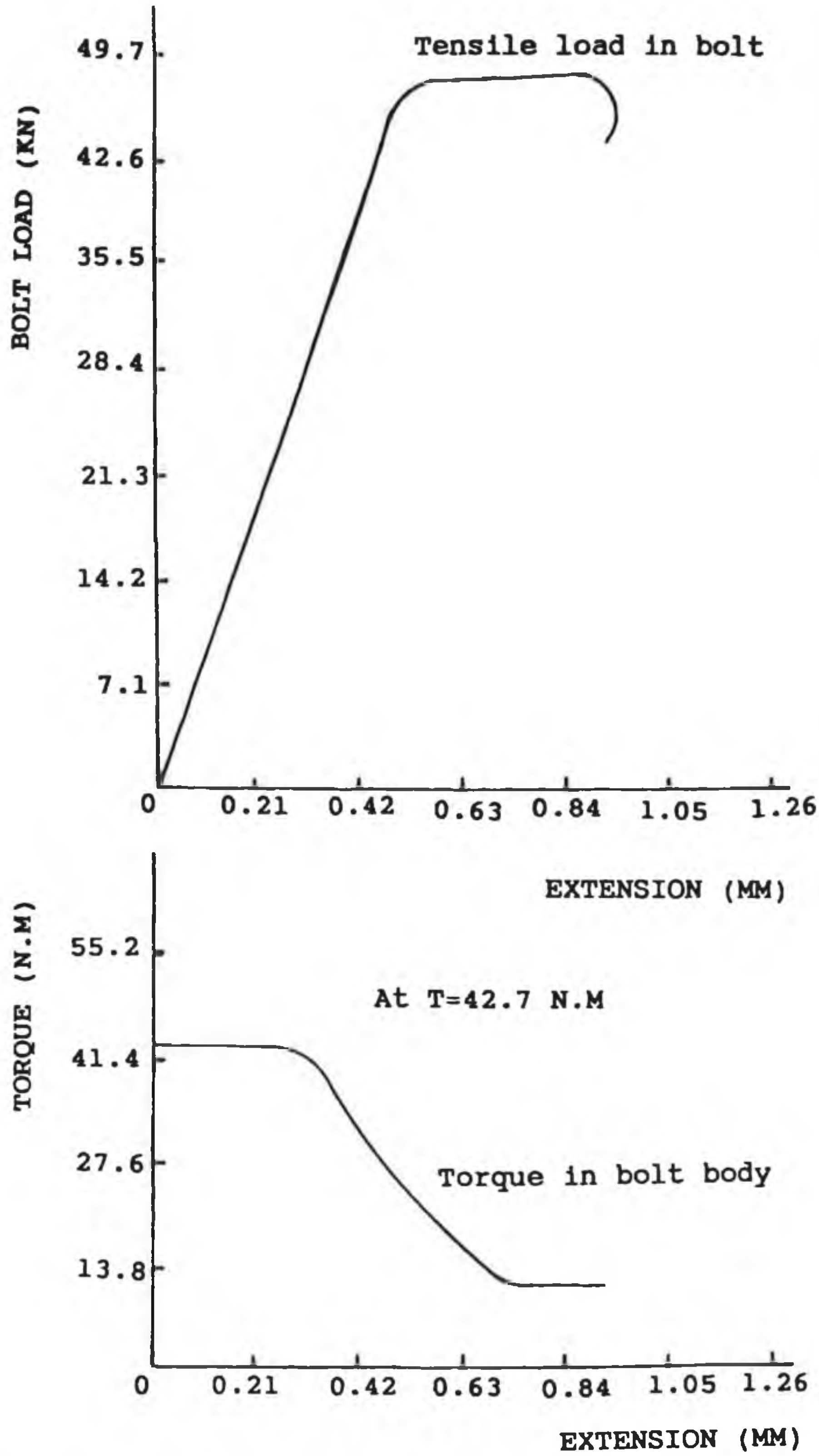


Fig.(5.16) Torque and tension forces in the bolt body as the external load is applied.

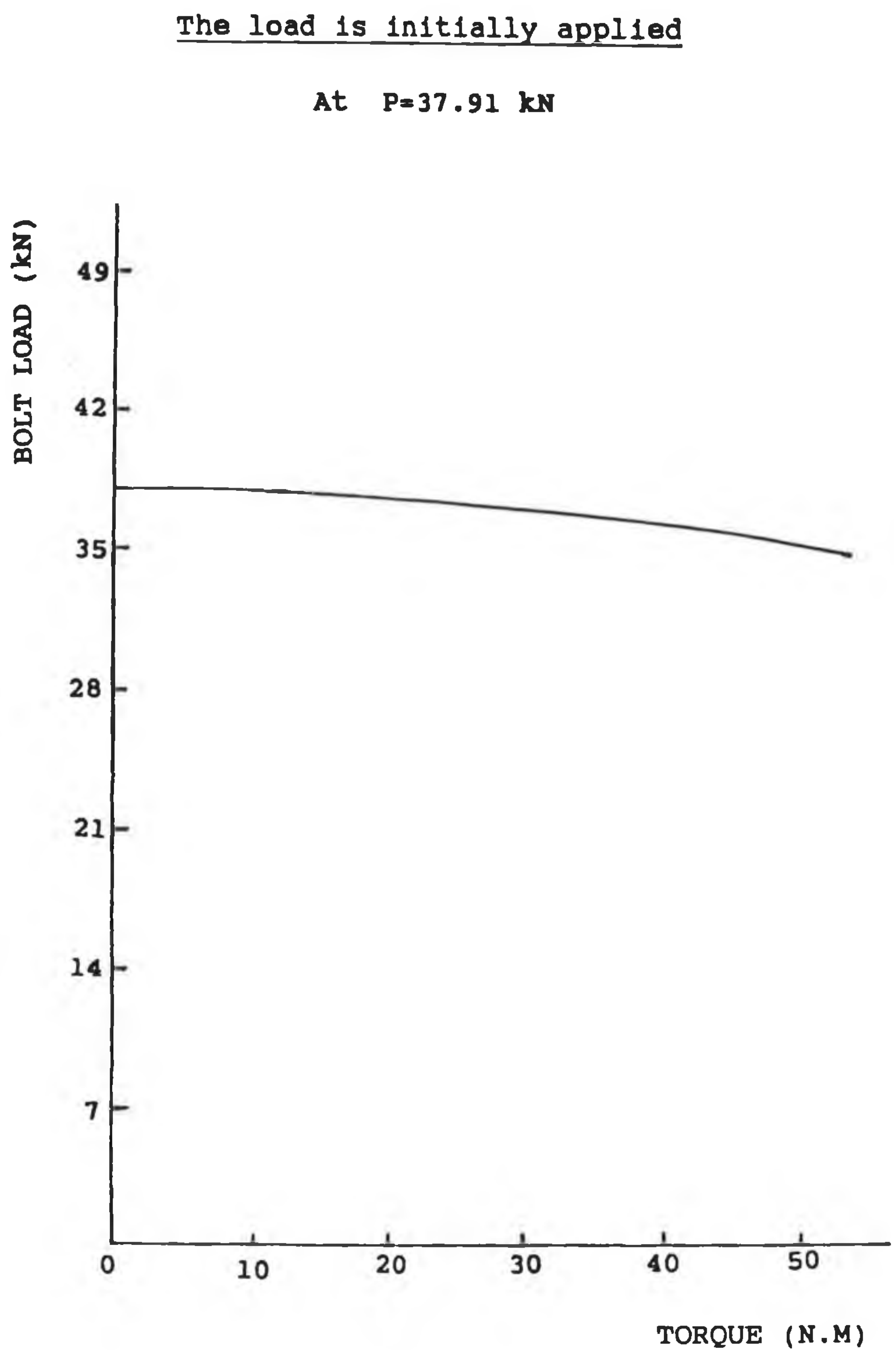


Fig. (5.17) Relationship between the initial axial load and the applied torque in the bolt body.

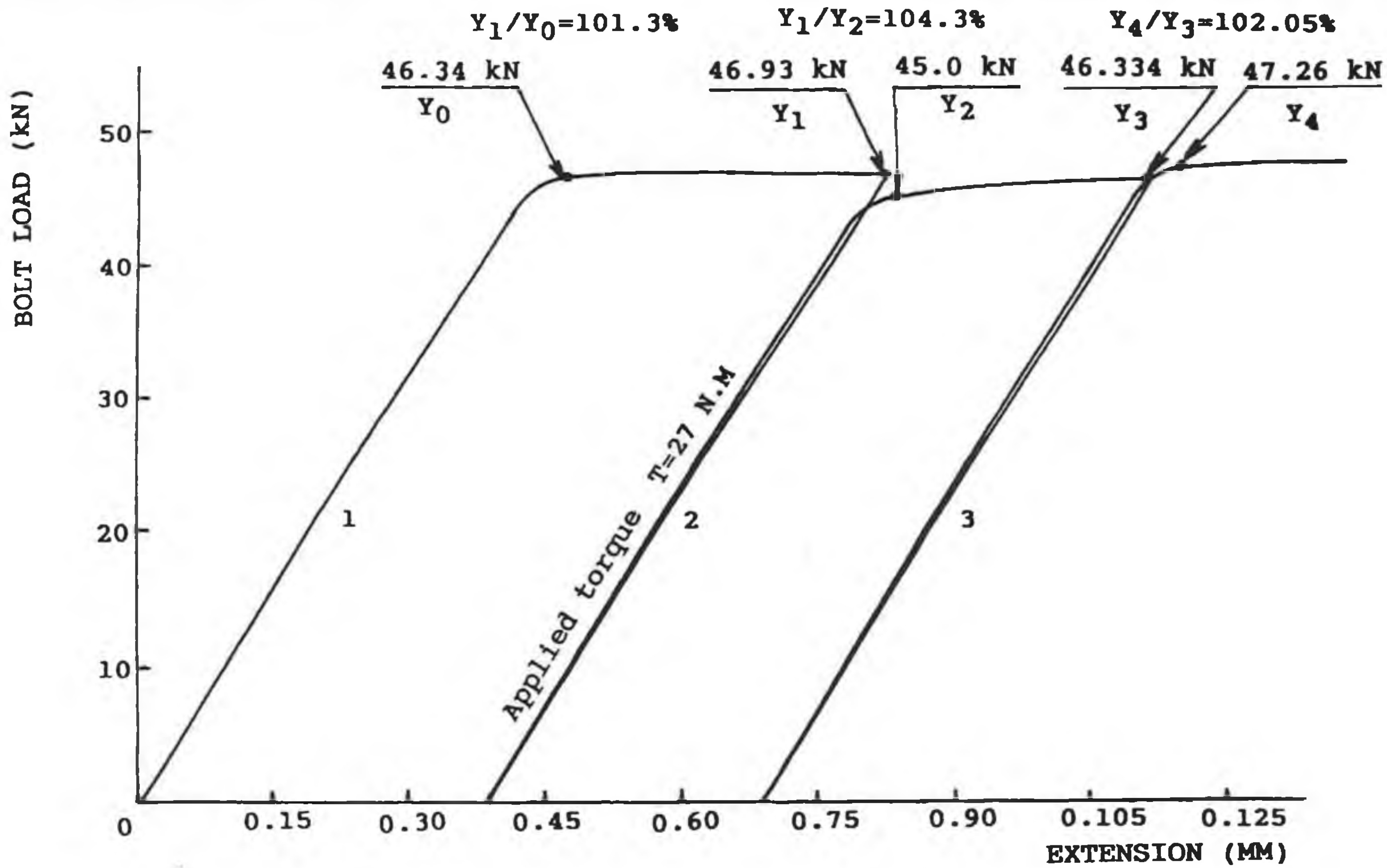


Fig. (5.18) Loading sequence, (1) Tensile load only, (2) Torque-tension, and (3) Tensile load only.

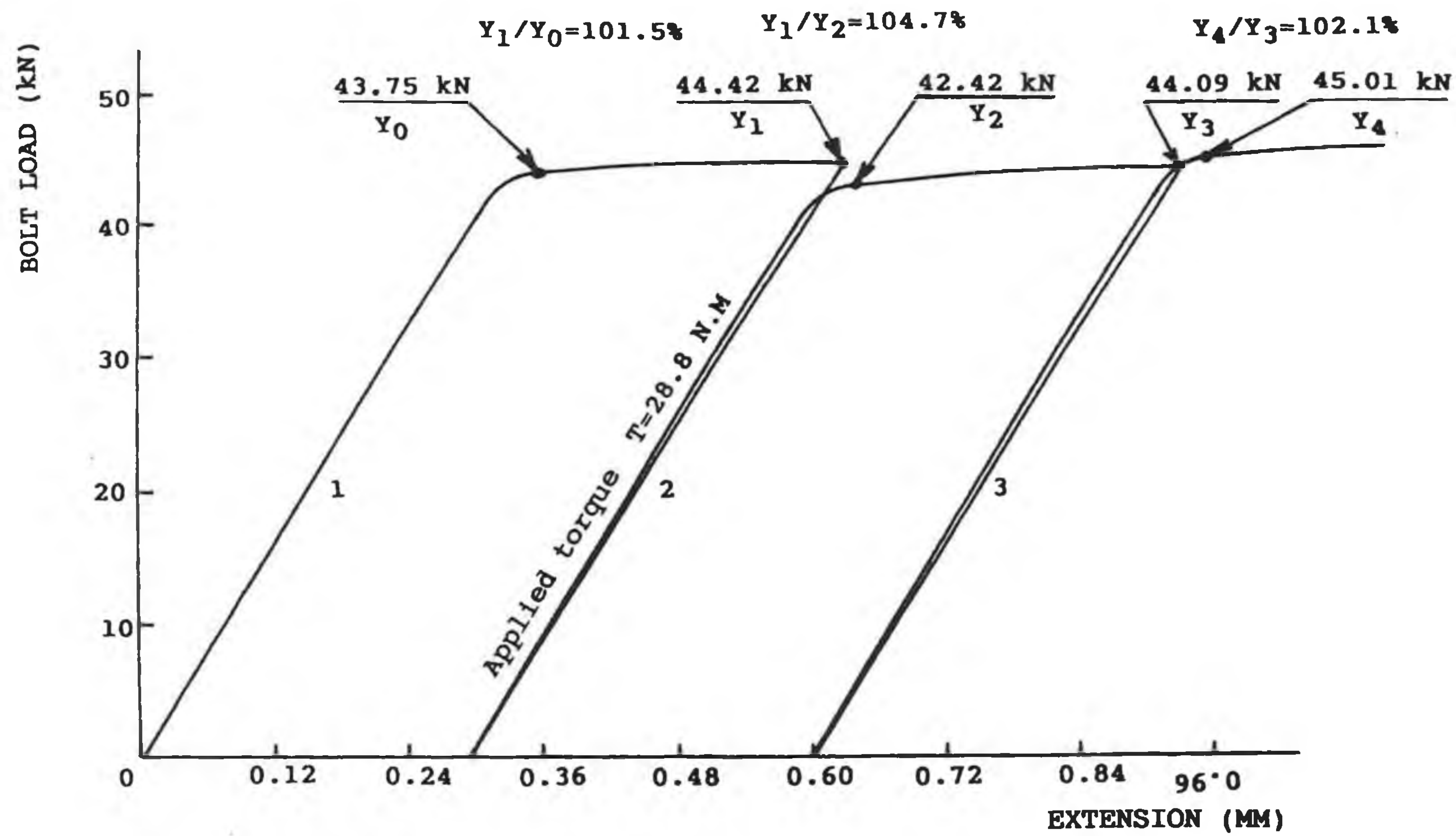


Fig. (5.19) Loading sequence, (1) Tensile load only, (2) Torque-tension, and (3) Tensile load only.

5.3.3.4 EFFECT OF APPLIED TORQUE IN PLASTIC RANGE

This test was initially made without strain gauges.

1. THE BOLT INITIALLY SUBJECTED TO A TORQUE

Figure (5.20) illustrates the relationship between the load and total extension in the fastener, when the bolt is initially subjected to a certain amount of torque within plastic range, and then it was loaded until yield occurs. It is clear from this **Figure** that it was not possible to detect the yield point in tension of the material of the bolt. This is due to fact that the material of the bolt has yielded in torsion before the axial load was applied. The plastic zone in the bolt extends from the outside diameter towards the core of the fastener.

2. THE BOLT INITIALLY SUBJECTED TO AN AXIAL LOAD

Figure (5.21) illustrates the relationship between initially applied axial load and total extension in the bolt then the torque was applied. It is seen that when the bolt is initially subjected to an axial load within elastic range and then subjected to the torque within elastic range, no permanent deformation will take place, and the material will still be elastic. In the sense that deformation is completely recovered, see **Curve** \`1' in **Figure (5.21)**. When the combination of tensile and torsional stress becomes greater than the yield stress in tension , then the axial load starts to reduce. **Curve** \`2' in **Figure (5.21)** is showing the combined stress exceeds the yield stress in tension due to increasing the applied torque into plastic range, then permanent deformation occurs. This is due to the fact that the material of the bolt has yielded in torsion and the plastic zone of the material of the bolt extends from the outside diameter towards the core of the bolt. Further applied torque on the bolt caused fracture at the smaller diameter. It is clear from that the elastic recovery is not sufficient to return the bolt length to its original value before the torque was applied, permanent extension of the bolt takes place.

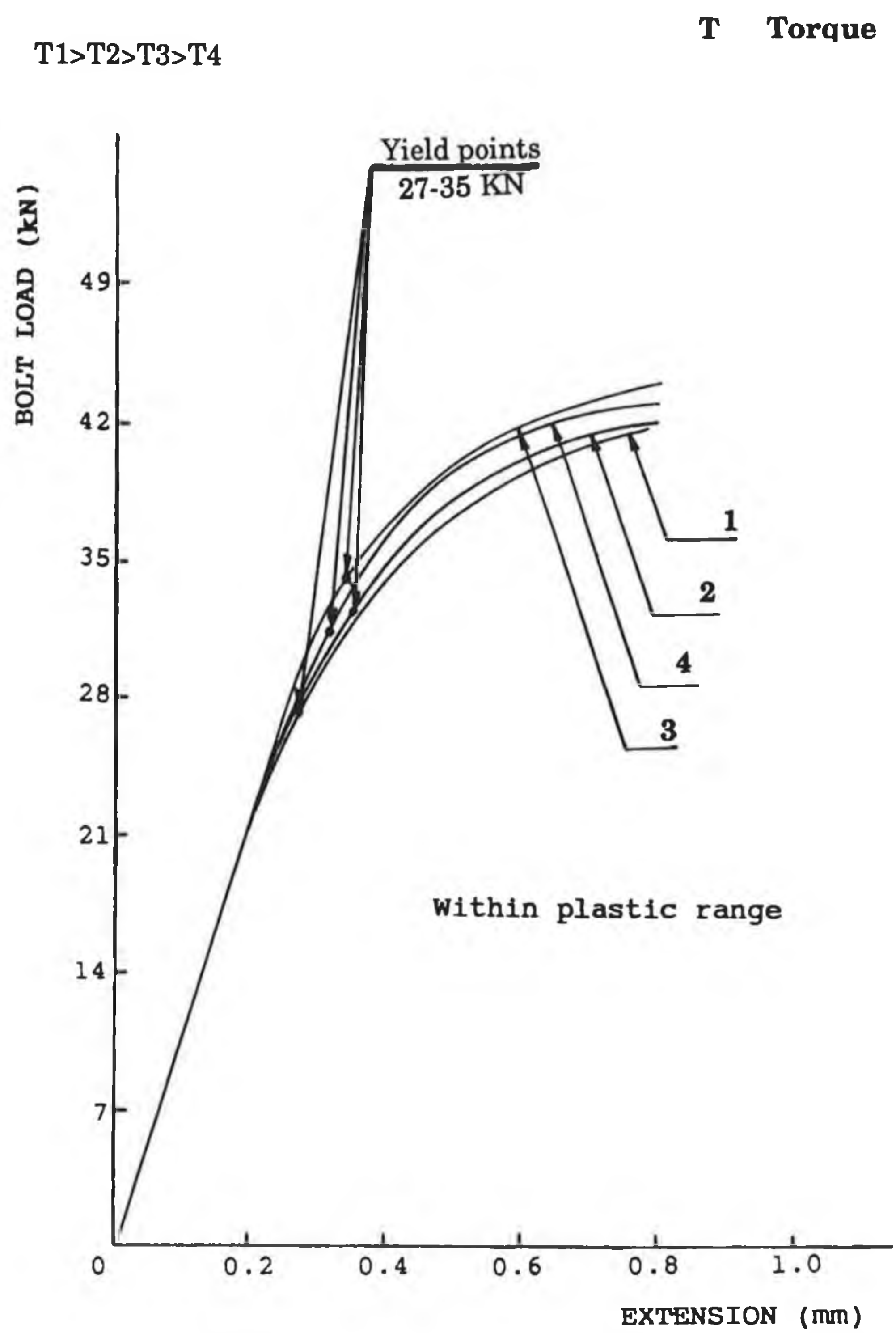


Fig.(5.20) Effect of applied torque within plastic range on the yield point of the material of the bolt.

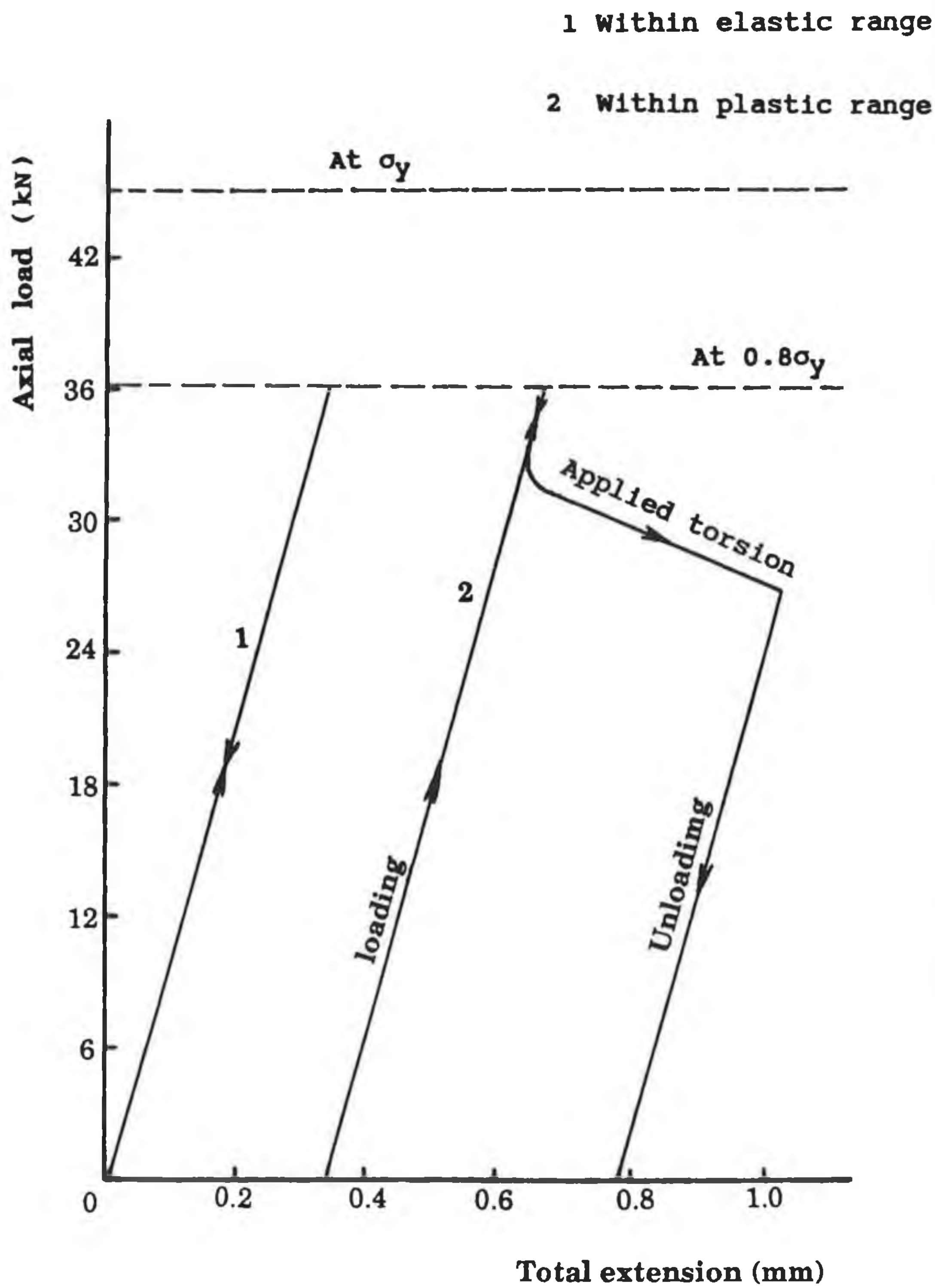


Fig.(5.21) Effect of applied torque within plastic range on the bolt initially subjected to an axial load.

CHAPTER SIX

DISCUSSION OF THE RESULTS

CHAPTER SIX

DISCUSSION OF THE RESULTS

6.1 Introduction

The theme of the present work was mainly confined to establish the dissipation modes of torsional and tensile energy in a bolted joint. And also to establish the mechanism in which yielding may take place under combined torque and axial load in a bolted joint. To achieve this aim a number of preliminary tests were carried out without using strain gauge. Results from these preliminary tests greatly helped to optimise the test conditions to be used in the main investigation. Following these initial tests, strain gauges were attached to the body of the bolt, in order to record the torsional stresses due to the applied torque. Various aspects of these tests are discussed in the following chapter.

6.2 Calculation of stresses in the body of the bolt

In the analysis of results, it was necessary to calculate the stresses induced in the body of the bolt. The pure torsion was applied to the bolt by means of the wrenching torque to the bolt head. In the absence of any friction under the bolt head, the maximum shear stresses were calculated using the theory of torsion for circular shafts. Thus the shear stress at the outside diameter of the body of the bolt is given by,

$$\tau = 16.T/(\pi.d^3). \quad (6.1)$$

where d is the diameter of a bolt (8 mm).

The uniaxial stresses were calculated from the bolt data using the following formula:

$$\sigma_x = P/A. \quad (6.2)$$

Where P is the uniaxial tensile load and A is the cross-sectional area in the body of the bolt ($A=\pi.d^2/4$).

The combined stress in the body of the bolt were calculated using the von Mises yield criteria, that is,

$$\sigma_{eff} = \sqrt{\sigma_x^2 + 3\tau^2}. \quad (6.3)$$

where σ_x is the applied tensile stress and is the torsional shear stress.

6.3 Effect of axial load on the bolt initially subjected to a torque

The results obtained in the experimental work show that when the combination of tensile and torsional stresses induced in the bolt exceeds the yield stress in tension, the material can not carry excess stress due to the initially applied torque. Hence, the torque reduces corresponding to the reduction in the torsional stress, see

Figure (5.10) upto Figure (5.16).

These results are summarised in **Figure (6.1)** which is illustrating the relationship between the shear stress and axial stress in the bolt during the application of axial stress for various magnitudes of initially applied torque. This **Figure** clearly demonstrates the manner in which the initially applied torque is reduced as the combined stress exceeds the yield stress in tension due to the increase in the axial load.

In order to remove any uncertainty in the exactness of the magnitude of the yield stress in tension ± **10** percent variation was introduced to form a `yield band' where σ_y was obtained experimentally at **0.05** strain offset from the elastic gradient.

This **Figure** is also showing the comparison between these results and the yield locus according to **von Mises yield criterion.** Such yield criterion dictates that once the yield locus (band) is reached, any subsequent increment of axial stress will lead to the reduction in the applied torque to maintain the combined stress to the level of the yield stress in tension. The experimental results show that the combined stress induced in the bolt is between **0.98 and 1.14** of the yield stress in tension. This suggests that the reduction of the initial torque due to the application of axial load may take place according to the yield criteria. so that the material is yielding only at the outside diameter due to the combined loading under initial torque and subsequent tension. However, the fact that the experimental curves overshoot the yield locus leads to the suggestion that perhaps some plastic deformation takes place at the outer region of the bolt cross-section. It is also possible that the slight strain hardening properties would cause the overshoot. the material may not truly conform to the von Mises y.c. .

The experimental results also show that a small amount of torque is still left in the fastener after the axial load is released. The explanation for this could be the fact that due to torsional loading and subsequent unloading there will be some residual stresses left in the bolt.

Figure (6.2) is showing the comparison between the experimental and theoretical results (dotted curve) calculated according to the assumption that the plastic zone in the bolt extends from the outside diameter towards the core of the fastener. For different magnitudes of the initial torque the agreement between the theory and experiment appears to be very close. This perhaps substantiates the assumption that the plastic zone in the bolt body may be extended from the outside diameter to the core of the bolt as initially applied torque gets unloaded.

In Figure (6.3) the experimental results are compared with the theoretical curves obtained according to yield criteria and plastic zone theory. This shows that most of the experimental curves fall in between the two theoretical limits curves which suggest that the unloading of torque takes place in such a way that some plastic zone is formed in the bolt. A third theoretical limit curve is drawn with 1.1 σ_y for the von Mises yield criteria. It shows that most experimental curves are contained by this limit curve as well.

Figures (6.4) and (6.5) are showing distribution of the combined stress squared along the radius of bolt subjected to initial torques of 22.4 and 42.7 N.M respectively. As shown in these figures the plastic zone in the bolt may be extended to the core only by a small extent when the initial torque is small until the material yields completely due to axial load. For a large initial torque the plastic zone may extend substantially towards the core before the material yields completely due to the axial load.

Based on these figures the reduction in the torque due to the application of an axial load may be explained as follows. When the initially applied torque is well below the yield torque of the bolt the combination of tensile and torsion stresses induced in the bolt does not reach the effective yield stress until an axial load of much higher magnitude is applied. Thus the unloading of the torque does not start until towards the end of the axial loading towards yielding. However, when the magnitude of the initially applied torque is very close to the yield torque, see Figure (6.5), the unloading of the torque starts with only moderate magnitudes of the axial load as the plastic zone in the bolt extends from the outside diameter towards the core of the fastener. It is also clear from these figures that the residual torque left in the fastener after the application of axial load to yielding under combined stress is greater when the initially applied torque is greater.

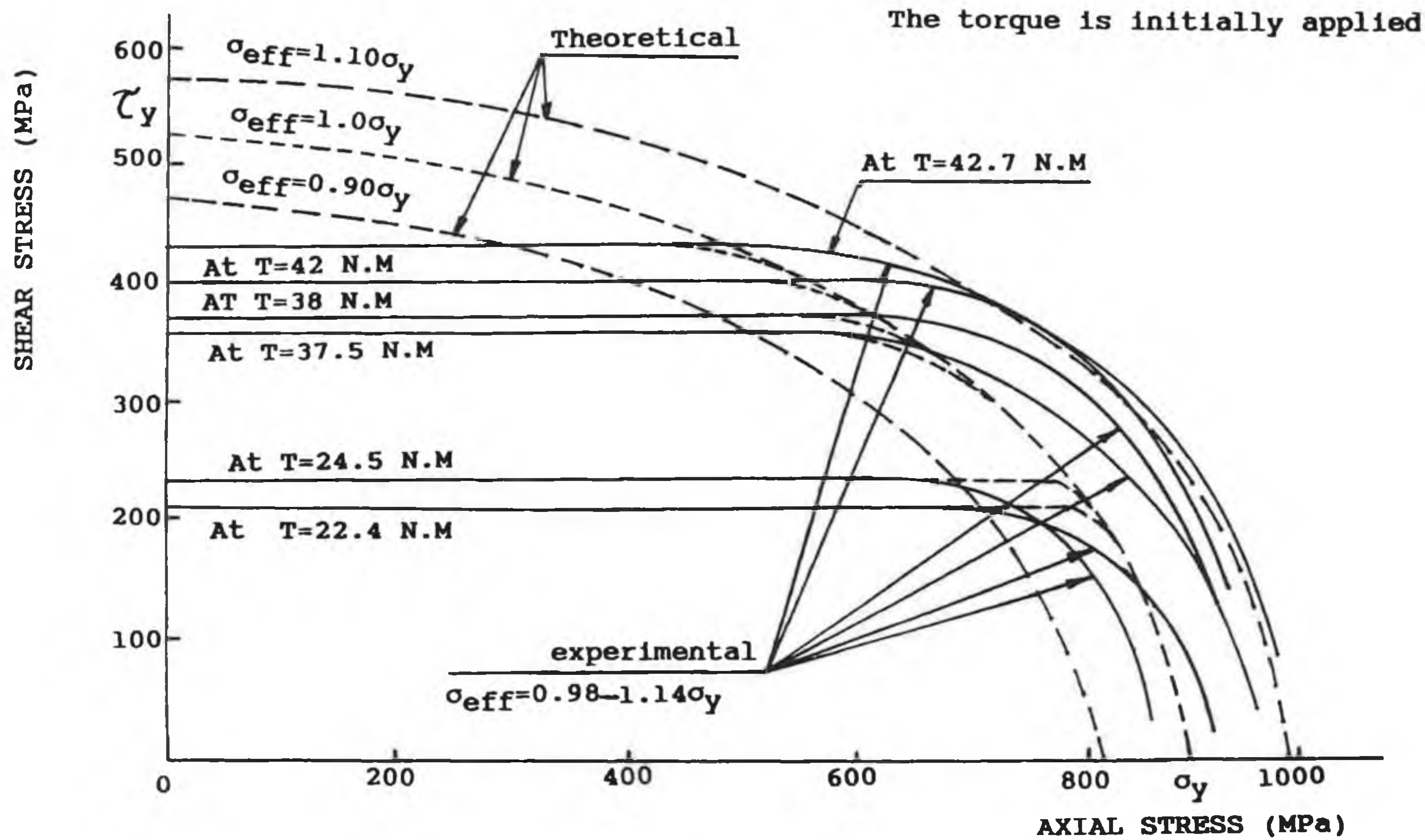


Fig.(6.1) Comparison between the experimental and theoretical results according to the yield criteria.

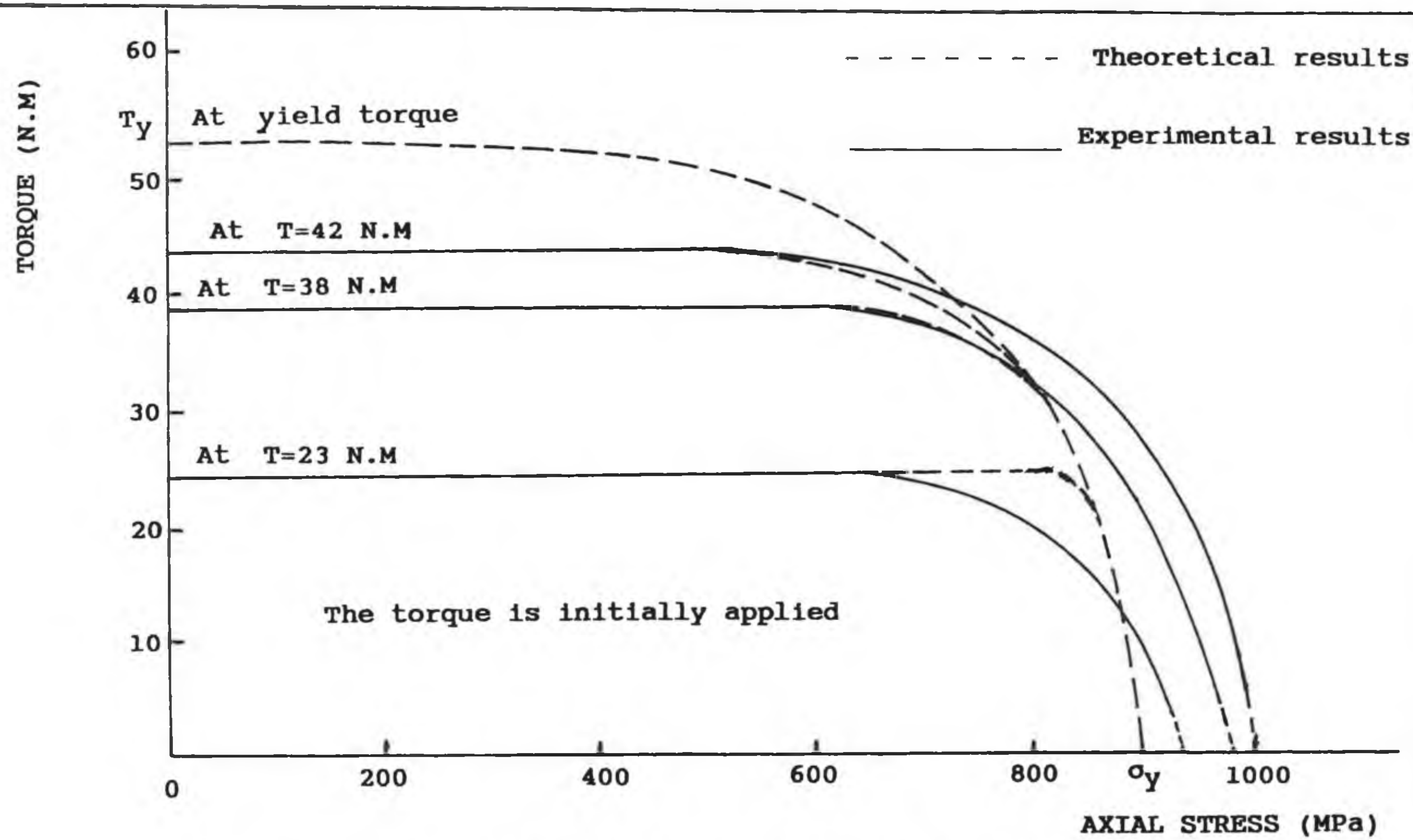


Fig.(6.2) <u>Comparison between the experimental and theoretical results according to the plastic zone theory.</u>

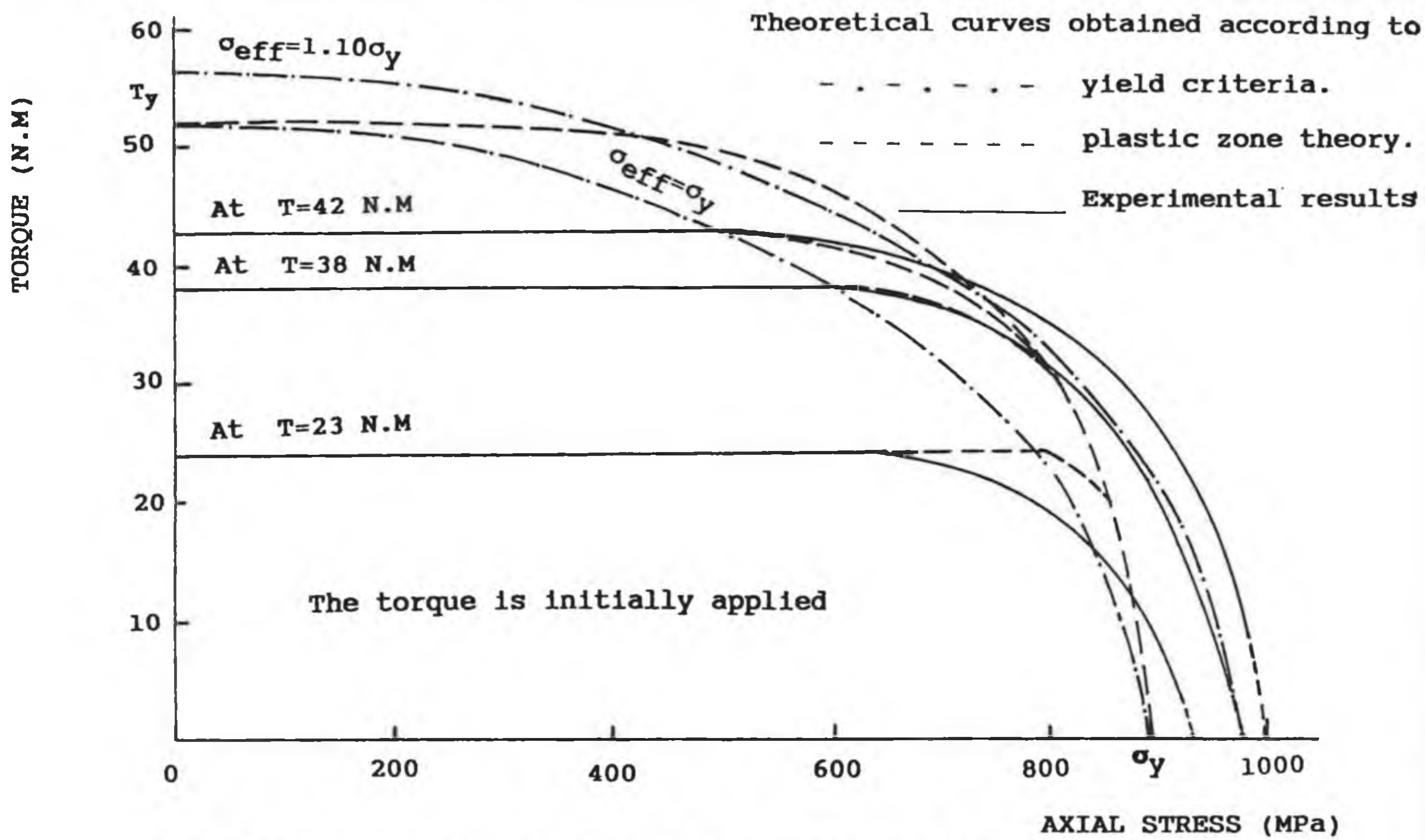


Fig.(6.3) Comparison between the experimental results and theoretical results.

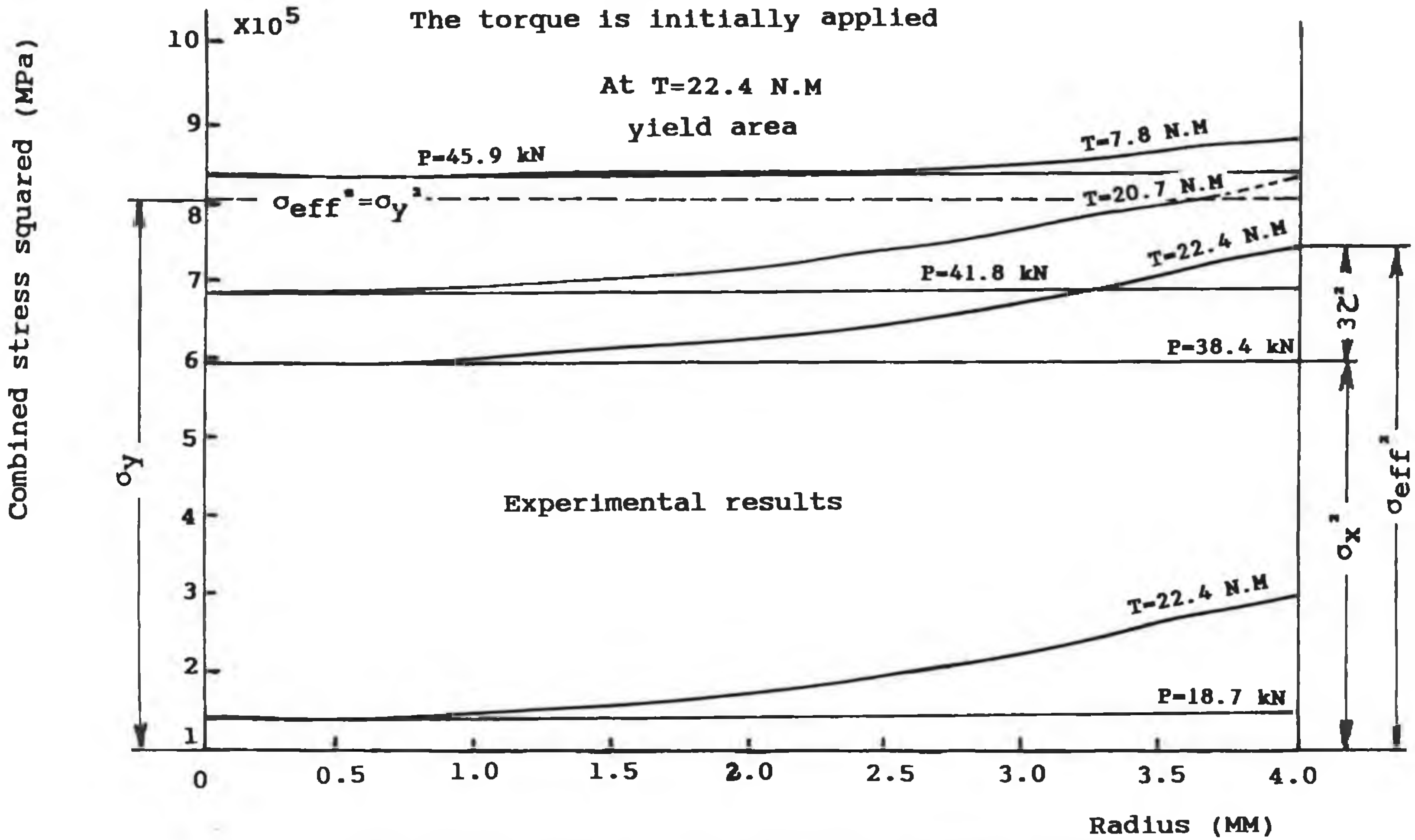


Fig.(6.4) Distribution of the combined stress squared along the radius of bolt initially subjected to the torque

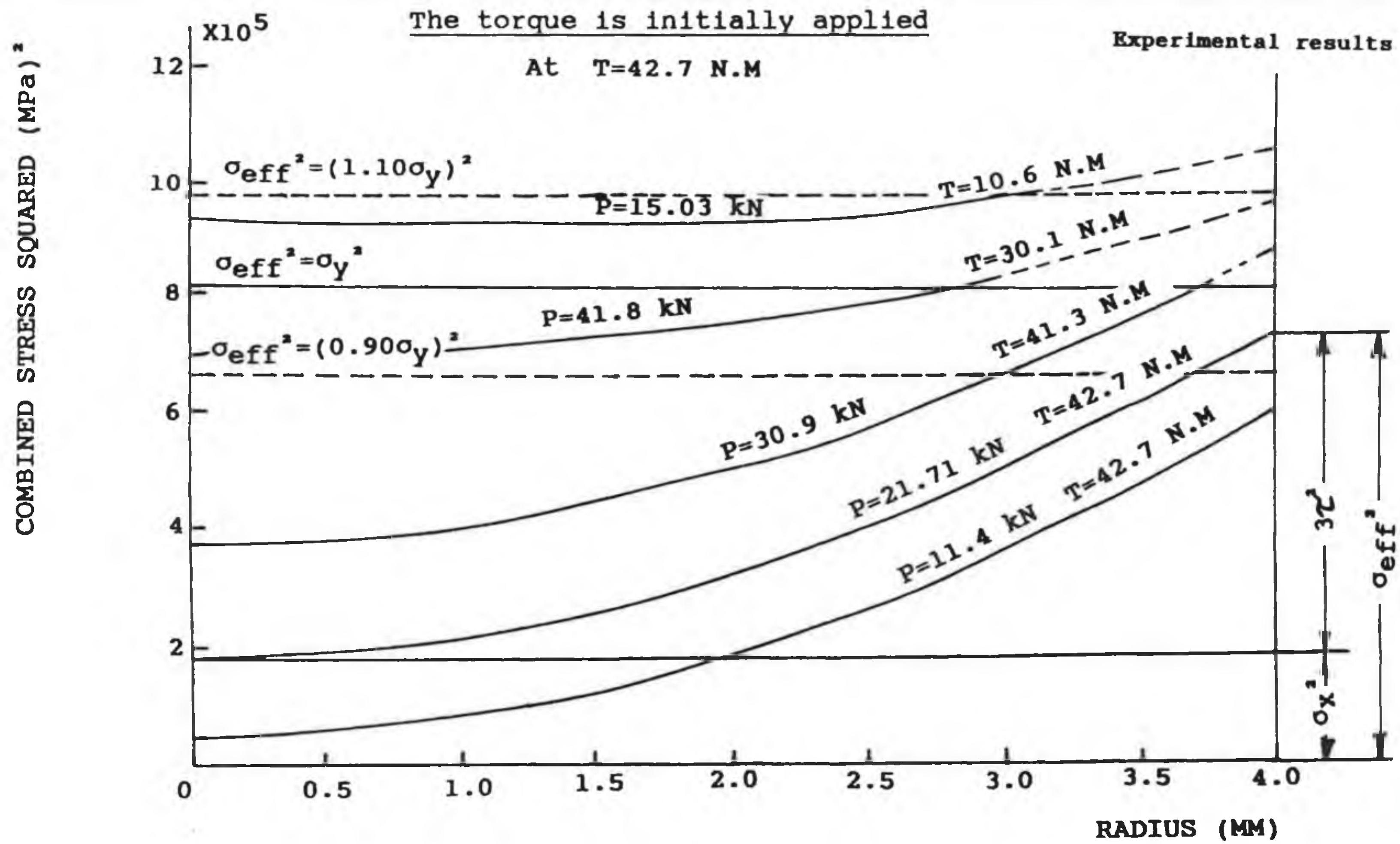


Fig.(6.5) Distribution of the combined stress squared along the radius of bolt initially subjected to the torque.

6.4 Effect of torque on the bolt initially subjected to an axial load

The results obtained in the experimental work show that when the combined stress is greater than the yield stress in tension the yield criteria becomes applicable and the axial load starts to reduce, see **Figure (5.17)**.

Figure (6.6) is showing these results illustrating the relationship between the torque and the axial stress in the bolt when the axial stress is initially applied. This **Figure** shows that the comparison between the experimental and the theoretical results (dotted curves) according to assumption that the plastic zone in the bolt extends from the outside diameter towards the core of the bolt. It is clear from this **Figure** that the manner in which the axial load is reduced as the combined stress exceeds the yield stress in tension at the outer radius of the bolt due to the increase in the torque in relation to the yield locus, according to von Mises yield criteria when the combination of torsional and tensile stresses at the outer diameter of the bolt reaches the yield stress in tension, the material at the outer surface may follow the yield ctiteria. It thus follows that any increment in the torque would cause reduction in the axial stress to keep the combined stress at about the yield stress in tension at the outer radius of the bolt. According to the yield criteria when the material of the bolt starts to yield under increasing torsional stress, the axial stress should decrease. The experimental results, However, show that the reduction of the axial load due to the applied torque is only marginal compared to the reduction of the torque due to the applied axial load as shown in **Figure (6.1)**. This is due to the fact that only outer surface of the bolt has reached the yield stress. For much higher torque, the plastic zone will progress towards the core of the bolt with more reduction in the axial load. Eventually, when the entire cross-section becomes plastic in torsion most of the initially applied axial load should disappear. This plastic torque is also shown in this figure and also the suggested reduction curve for the initially applied axial stress.

Figure (6.7) is showing the distribution of the combined stress squared along the radius of the bolt initially subjected to an axial load. It is clear from this **Figure** that no plastic deformation will take place until the combination of the tensile and torsional stresses becomes equal to the yield stress in tension of the bolt material. As soon as the combined stress induced in the bolt reaches the yield stress, the outside diameter of the bolt starts to yield. However, when the combined stress is greater than the yield stress in tension, the reduction of axial load takes place and for a given torque a plastic outer zone will develop as well as the reduction in the initial axial stress when eventually this torque reaches 'τ_p', the plastic torque of the bolt the axial stress should reduce to zero.

6.5 Effect of combined stress on the yield point

The results obtained in the experimental work show that there are three segments of this test, namely, the points at which the loads `$Y_1$' , `Y_2' and the initial torque `T' were measured (see Figures (5.18) and (5.19)). The ratio between Y_1 and Y_2 for both loading cases compares well. The average ratio between Y_1 and Y_2 for a number of tests was 104.5 percent. It is clear from these figures that there is a small reduction in the yield load when a pre-torque is applied. This reduction however is very small. The experimental results show that no significant reduction in the uniaxial tensile load takes place whatever the magnitude of the initially applied torque (within elastic range), so that the reduction in the yield load can be neglected by comparing with the original magnitude of yield load. The ratio ,Y_1/Y_2, was determined to indicate that, for all magnitudes of initial torque, approximately the full value of the uniaxial yield load of the bolt was reached.

Figures (5.18) and (5.19) show the points at which the loads `$Y_3$', `Y_4' were measured. The ratio between Y_4 and Y_3 for both loading cases compares well. The average ratio between Y_4 and Y_3 for a number of tests was 102.05 percent. It is found that the yield stress is increased only marginally. The ratio Y_4/Y_3 was determined to confirm that upon reloading a still higher yield point was obtained. Therefore, the initial torsional stresses induced in the fastener will not have any effect on the yield load.

From the above results, it appears that these results are in agreement with the experimental results obtained by SPS Laboratories, Ireland, [49].

This also suggests that the torsional stresses remaining in the fastener from the tightening operation disappeared as the external tensile load is applied subsequently. This would follow that the torsional stress induced in the bolt body during tightening does not effect on the yield load of the material of the bolt. However, these results differ from experimental results reported by Schneider, [31], Which shows that the uniaxial tensile yield point was reduced substantially. This reduction may be due to fact that the fasteners have already been tightened as far as the yield point.

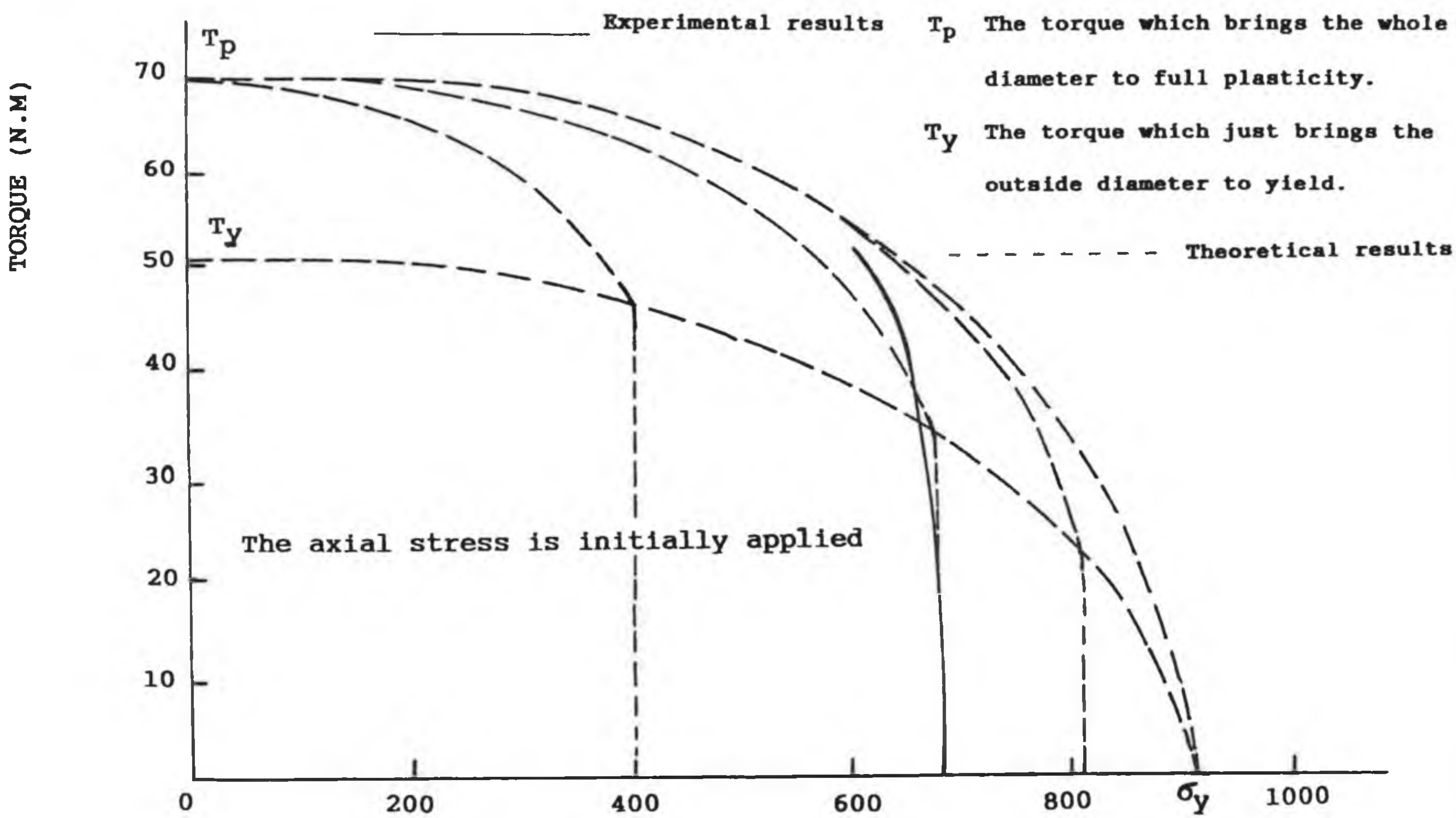


Fig.(6.6) Comparison between the experimental and theoretical results according to the plastic zone theory.

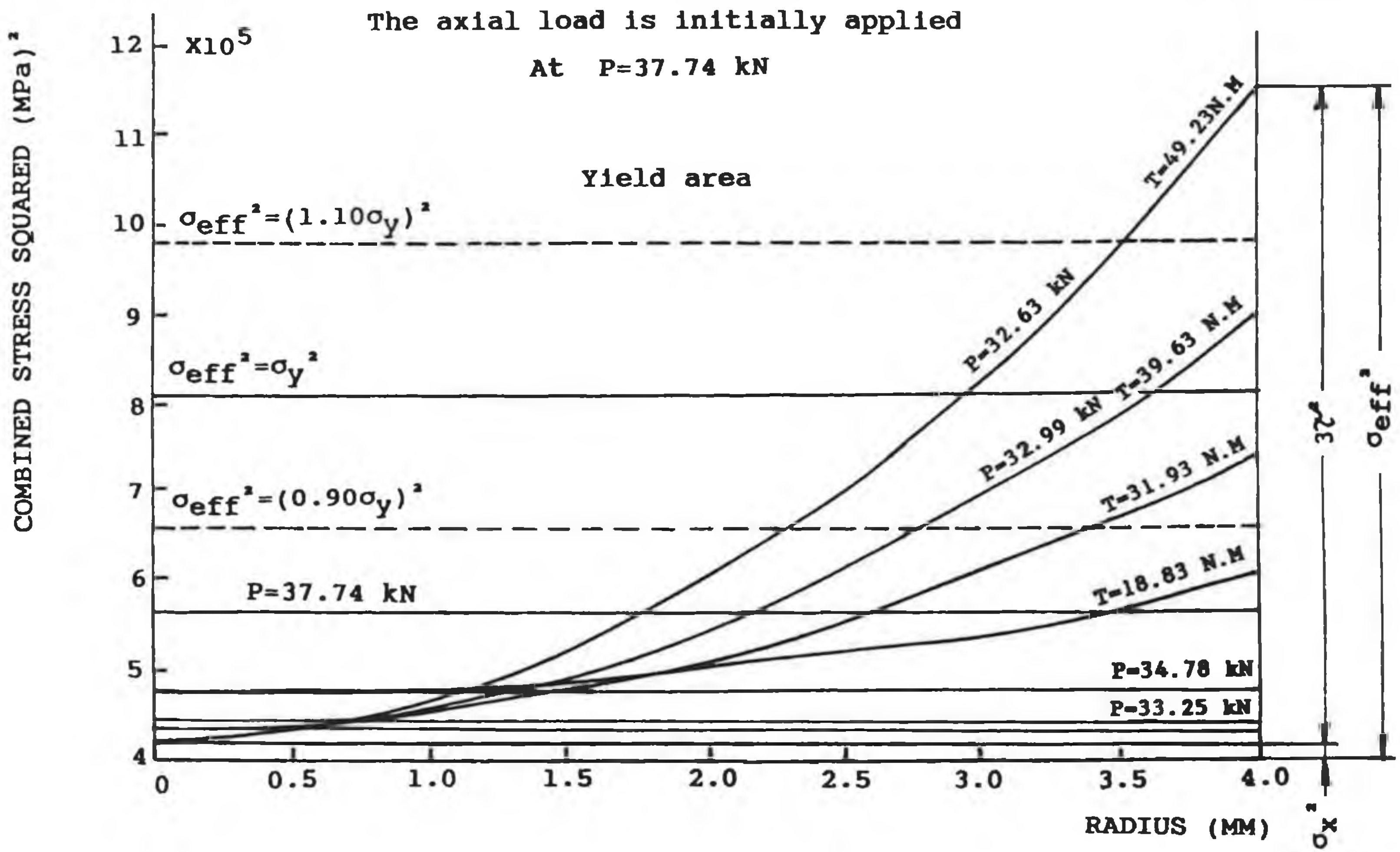


Fig.(6.7) Distribution of the combined stress squared along the radius of bolt initially subjected to axial load.

6.6 General discussion of the results

6.6.1 INTRODUCTION

The results obtained in the experimental work were established for a bolt body, subjected to combined axial load and pure torque. These results show the dissipation modes of torsional and tensile energy in the body of the bolt initially subjected to an axial load or pure torque and subsequently subjected to the other. These results also show that there is no appreciable reduction in the uniaxial tensile strength when a bolt is subjected to uniaxial load to plastic yielding and then unloaded and subsequently subjected to combined axial load and torque. It is obvious from these results that these observations would apply equally to a bolt in a joint.

6.6.2 DISCUSSION OF RESULTS RELATED TO THE NORMAL BOLT

6.6.2.1 STRESS BEHAVIOUR DURING TIGHTENING OF THE BOLT

During tightening,the fastener is subjected to a combined stress state comprising of an axial stress due to the pre-load and shear stress due to torque applied to the bolt head.

In the normal bolt, the axial tensile stress is dependent on the pre-load `$F_i$' which is developed by the extent of tightening measured by the load cell and the tensile stress area `A' of the fastener.

The shear stress is dependent on the thread torque `$T_G$', and the ratio of the torque `T_G' to the wrenching torque `T'.The wrenching torque will depend on the friction at the mating surfaces.

The combination of the tensile and torsional stresses can be obtained according to the **von Mises yield criteria** for two dimensional stress.

For a normal bolt the situation can be explained by the behaviour of the bolt shank which can be represented as a circular shaft at minor thread diameter `d_2'.

The tightening of a bolt is achieved by application of a rotation torque to its head. This applied wrenching torque causes the bolt shank to rotate and the thread to tighten the bolt. Before the wrenching torque `T_1' to the bolt head is removed the following equation can be written, see **Figure (6.8a)**.

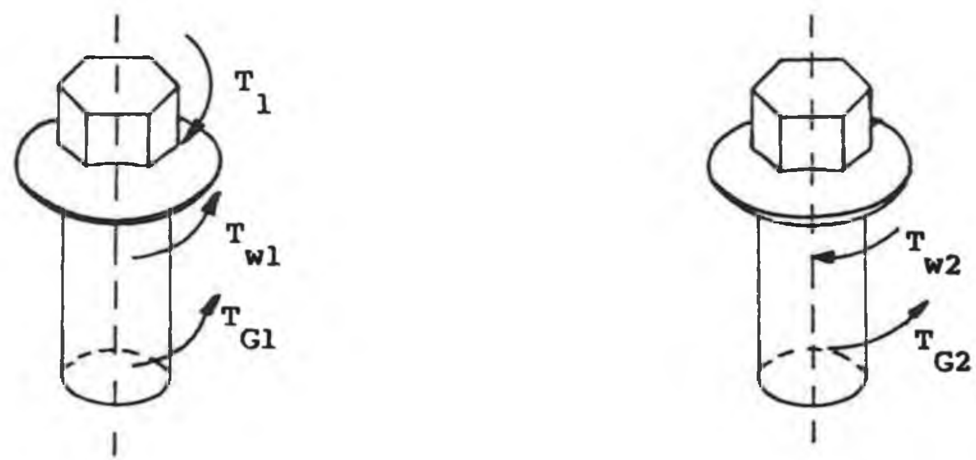
$$T_1 = T_{G1} + T_{W1}. \qquad (6.4)$$

Where`$T_{G1}$'is the thread torque and`T_{W1}' is the under-head torque. After the applied wrenching torque has been removed a relaxation of the material stresses occurs, and a winding-back in the body of the bolt lead to a reduction in torsional stress on the thread. On removal of the wrenching torque the fastener shank unwinds slightly . This effect is opposed by the under-head torque which changes direction to oppose the rotation of the fastener shank. The unwinding of the fastener shank stops when the equilibrium is established between the under-head torque and thread torque. So that after the wrenching torque has been removed the following equation can be obtained:

$$T_{G2} \approx T_{w2}. \tag{6.5}$$

Where `$T_{G2}$'is the thread torque and`T_{w2}'is the under-head torque after the applied wrenching torque has been removed. It can be seen that the direction of the under-head torque `$T_{w2}$' has reversed compared with its direction during tightening `T_{w1}'. It always acts to oppose the fastener head rotation. The residual torque on the threads `$T_{G2}$' acts in the same direction as during tightening `T_{G1}' [32], see Figure (6.8b).

(a) (b)



(a) Before the wrenching torque has been removed

(b) After the wrenching torque has been removed

Fig.(6.8) Torque conditions during (a) Tightening (b) Static loading.

6.6.2.2 DISCUSSION OF THE RESULTS ACCORDING TO YIELD CRITERIA

The results obtained in the experimental evaluations show that when an axial load is applied to the fastener initially subjected to a torque, this torque is reduced after the axial stress reaches certain magnitude. This may happen if the combined stress in the fastener follows the yield criteria. This is due to fact that the combined stress induced in the fastener becomes equal to the yield stress in tension well before the applied tensile load causes yielding on its own. In otherwords, any increment of axial stress will lead to the reduction in the torque to maintain the combined stress to the level of the yield stress in tension. This behaviour was seen in all the bolts tested, whatever the magnitude of initial applied torque in the fastener. However, the reduction in the torque was not as much as should have been if yield criteria was strictly governing the behaviour.

Attempts were made to establish whether any unwinding takes during the application of the axial load thus, causing a reduction in the initially applied torque.

In such event the torque should start to reduce as soon as any axial load is applied to the fastener. Experimentally it was observed that no reduction in torque starts before the combined stress becomes equal to the yield stress in tension. The torsional stress decreases rapidly as further axial load is applied. This confirms that the bolt does not un-rotate as the axial load is applied to the fastener after applying an initial torque.

The experimental results also show that when the bolt is initially subjected to an axial load and then to the torque, the plastic zone of the material can not extend to the core of the bolt until the applied torque causes the combined stress in the fastener to become equal to the yield stress in tension well before the applied torque causes yielded on its own. However, when the combined stress becomes greater than the yield stress in tension the yield criteria becomes applicable and the axial load starts to reduce. It is clear from this that any increment in the torque would cause reduction in the axial stress to maintain the combined stress at about the yield stress in tension. This behaviour was seen in all bolts tested, whatever the magnitude of initial applied axial load to the fastener.

6.6.2.3 DISCUSSION OF DISSIPATION OF TORSIONAL AND TENSILE STRAIN ENERGY IN THE BOLT

During tightening, the fastener is subjected to combined stress states comprising an axial tensile stress, due to the induced pre-load, and shear stress due to the torque transmitted to the body of the bolt. When the tightening of the bolt stops and the torque wrench is removed the shank torque decreases causing a winding-back of the bolt shank and a significant reduction in the torsional stress on the threads due to a reduction in the torque.

Figure (6.9) is showing the distribution of the combined stress squared along the radius of the bolt, when the bolt is subjected to an axial load after initial tightening torque. This stress is also represented in **Figure (6.10)** according to yield criteria. This **Figure** illustrate the relationship between the shear stress and axial stress in the fastener. It is clear from this **Figure** that the magnitude of initial pre-load which is developed in the fastener due to tightening operation is `$\sigma_{x1}$' and the torsional stress remaining in the fastener after wrenching torque has been removed is `τ_0', and the combined stress induced in the fastener due to tightening is represented in this **Figure** (see point `a'). It is shown that the combined stresses induced in the fastener is still less than the yield stress in tension. As the external tensile load is applied to the fastener, the initial pre-load increases to `σ_{x2}' and the torsional stress on the threads remains unchanged (see point `b'), until the combination of the torsional and tensile stress reaches the yield stress in tension of the material of the bolt (see point `c'). At this point the applied axial load is increased from `$\sigma_{x2}$' to `σ_{x3}', and the material starts to yield at outside diameter of bolt. Any further increase in the level of applied axial load will cause the torsional stress to reduce. When the applied axial load reaches the magnitude

`$\sigma_{x4}$' the torsional stress is reduced to `τ_1'.

It is clear from this **Figure** that the combined stress induced at the outer surface of the fastener becomes equal to the yield stress in tension before the applied axial load reaches to the yield load. Therefore, the increment of the applied axial load from `$\sigma_{x3}$' to `σ_{x4}' causes reduction in the torsional stress from `$\tau_0$' `τ_1 '.

Referring to **Figure** (6.10) the increment of the applied axial load `**D**$\sigma_x$' may be linked to partially plastic loading of the outer surface. The residual stresses are thus obtained by subtracting the torsional stress remaining in the fastener `τ_1' (after wrenching torque has been removed) from the initially applied elastic stress `$\tau_0$'. It is clear from this that the material starts to yield at the outside diameter of the bolt **(point C)**, then for any increment of the applied axial load causes the torque to reduce from `T_1' to `T_2. The magnitude of plastic torque depends on the magnitude of the applied axial load.

It clearly demonstrates that the torsional energy may be partially or totally disappeared in the fastener due to a relaxation of the material stresses in plastic form.

Figure (6.11) is showing distribution of the combined stress squared along the radius of bolt, when the bolt is initially subjected to a certain amount of axial load `σ_{x1}'

within the elastic range, then the torque `T_1' of different magnitudes are applied to the fastener.

Figure (6.12) illustrates the relationship between the shear stress and axial stress in the fastener initially subjected to an axial load. It is clear from these **Figure** that the combined stress induced in the fastener is still less than the yield stress in tension see point `a'. Increasing the torque to the magnitude `T_2' to give rise to shear stress `$\tau_2$', the combined stress reaches the yield stress in tension of the material of the bolt at point `b', then the material can not withstand any excess stress beyond the yield stress. Hence, the axial load starts to reduce and the material will follow the **Curve** `bc'. It is clear from this **Figure** that the combined stress becomes equal to the yield stress in tension before the applied torsional stress reaches to the yield stress in torsion. That means the applied torque is still within the elastic range and at the same time the material starts to yield in tension at the outer surface. When the applied torque reaches to the magnitude `$T_3$' to give rise to shear stress `τ_3', the axial stress is reduced to magnitude `$\sigma_{x2}$'. Any increment of the applied shear stress `$D\tau$' causes the reduction in the axial load by `$D\sigma_x$'.

It clearly demonstrates that the tensile energy may be partially or totally disappeared in the fastener due to a relaxation of the material stresses in plastic form.

Finally, based on the above observations it may be said that the torsional and tensile energy in a bolted joint is dissipated partially or totally in the form of plastic work of the material of the bolt at the outer surface.

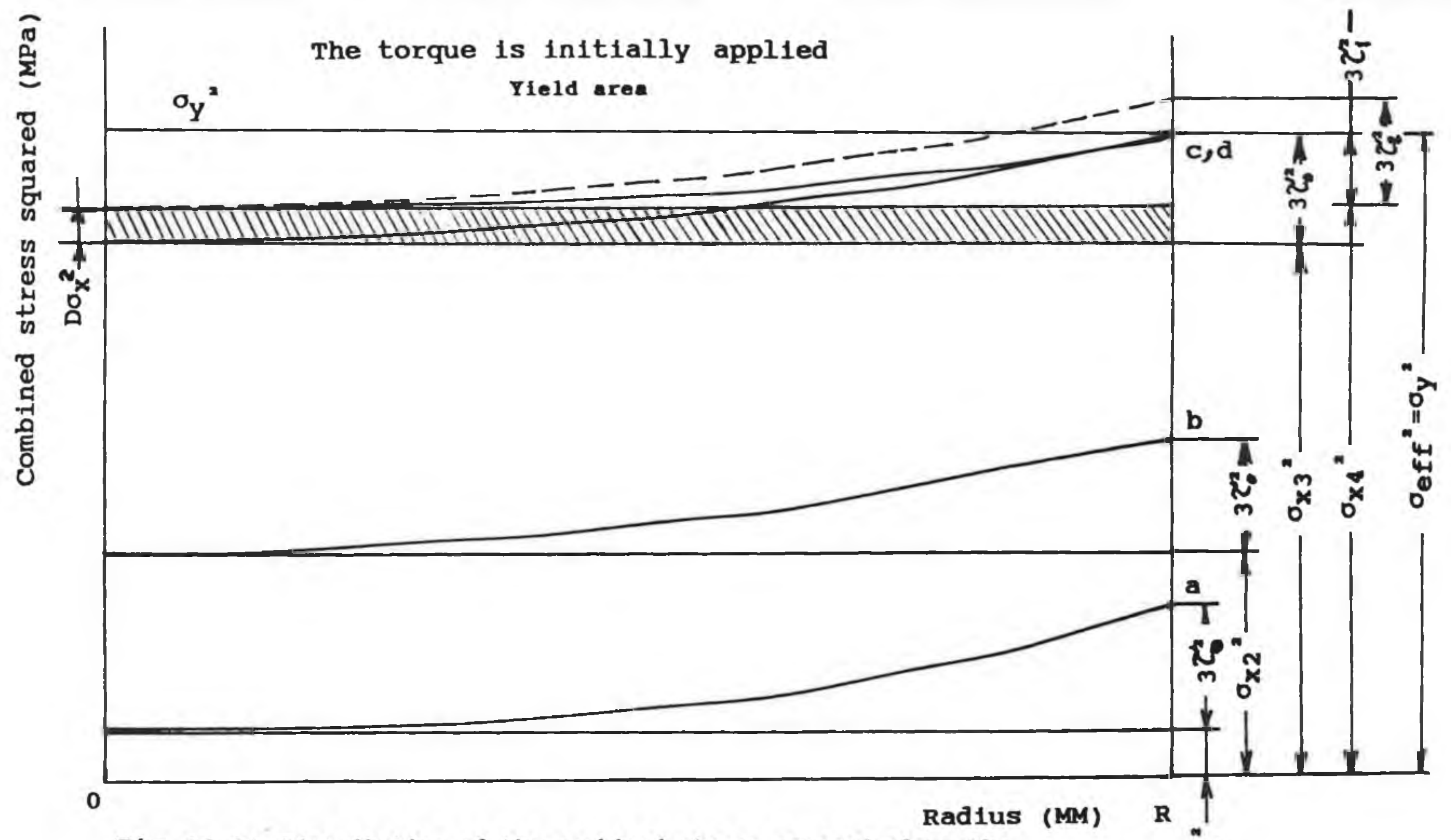


Fig.(6.9) Distribution of the combined stress squared along the radius of the bolt initially subjected to a torque.

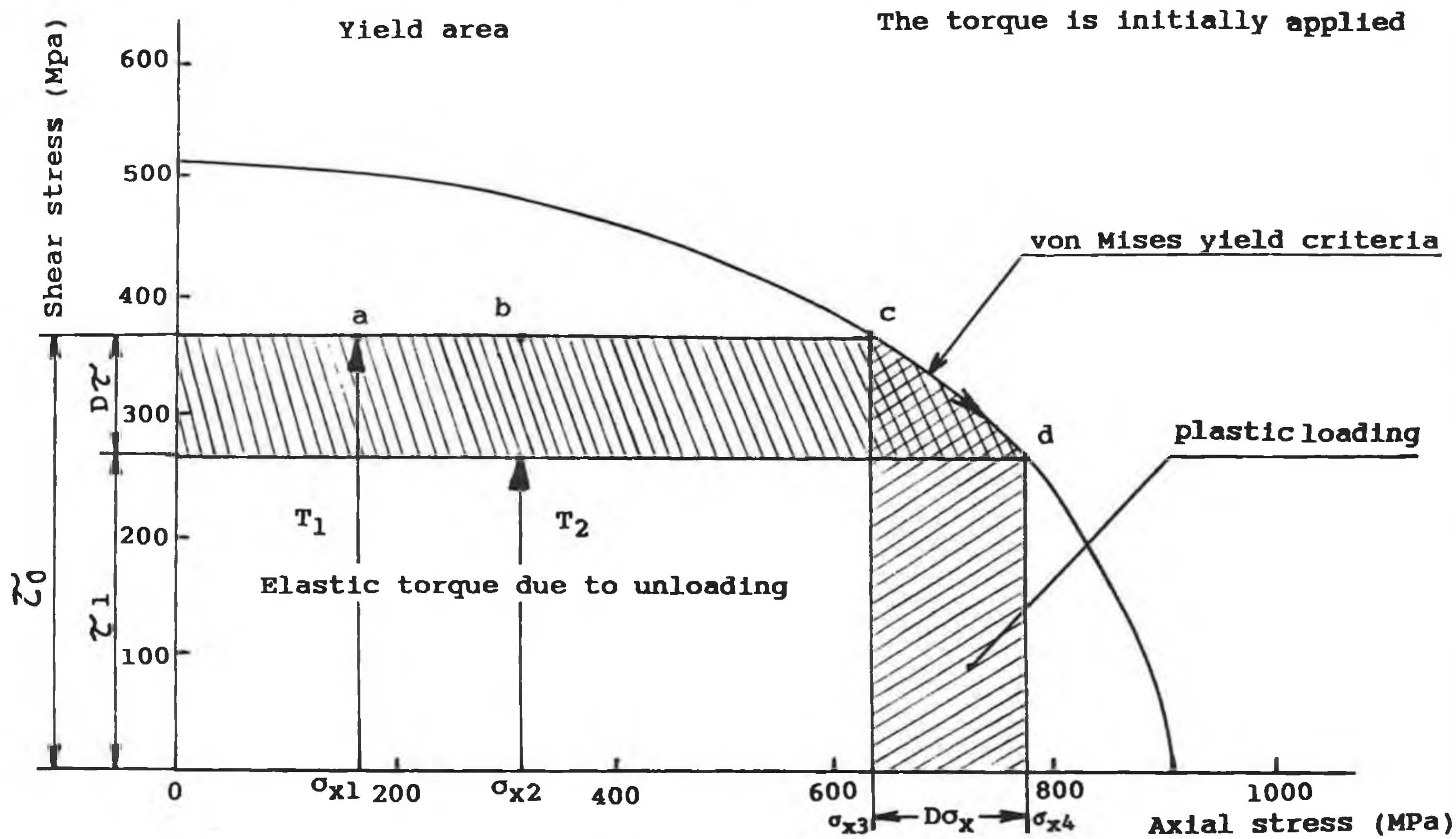


Fig.(6.10) Dissipation of torsional strain energy according von Mises yield criteria.

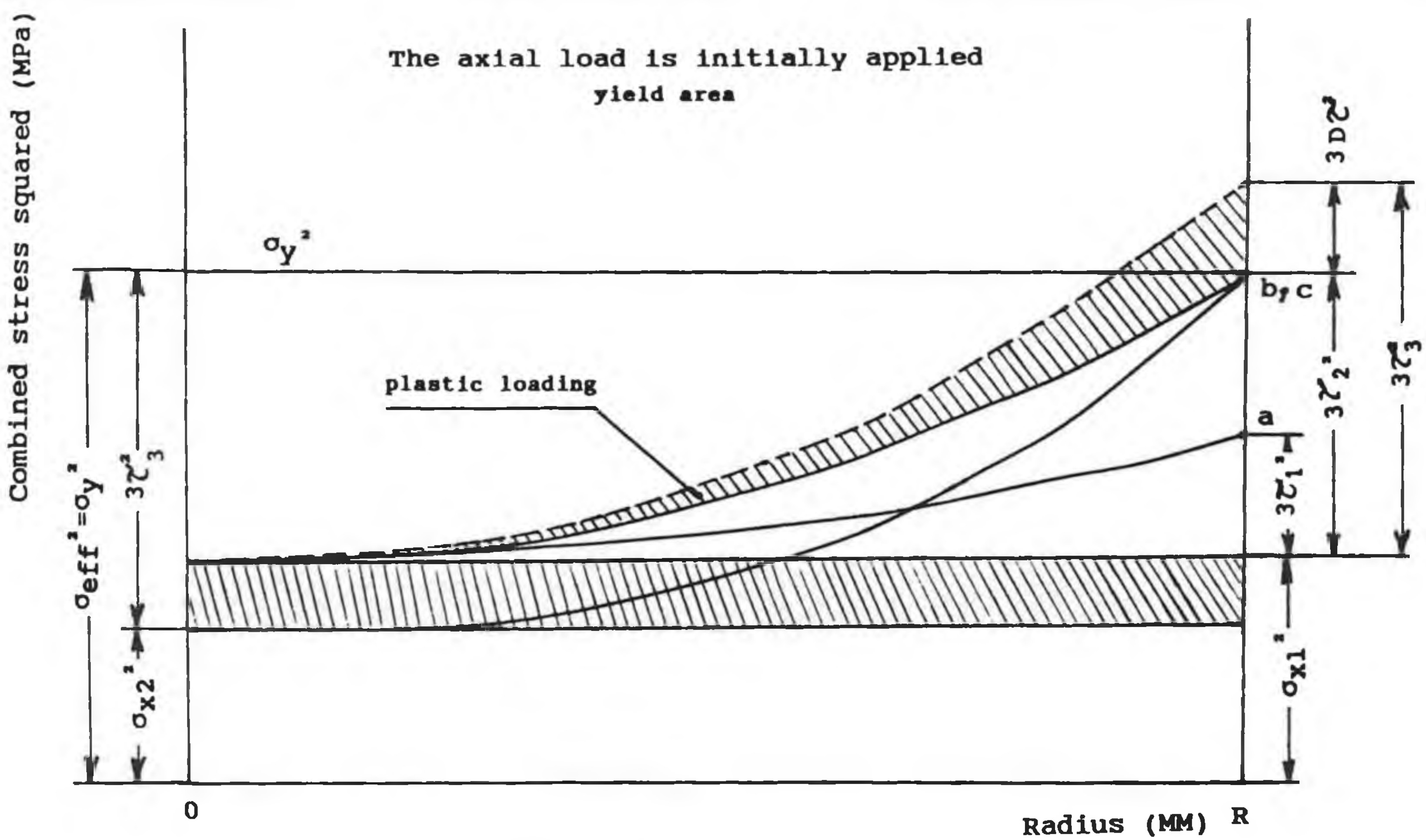


Fig.(6.11) <u>Distribution of the combined stress squared along the radius of the bolt initially subjected to an axial load.</u>

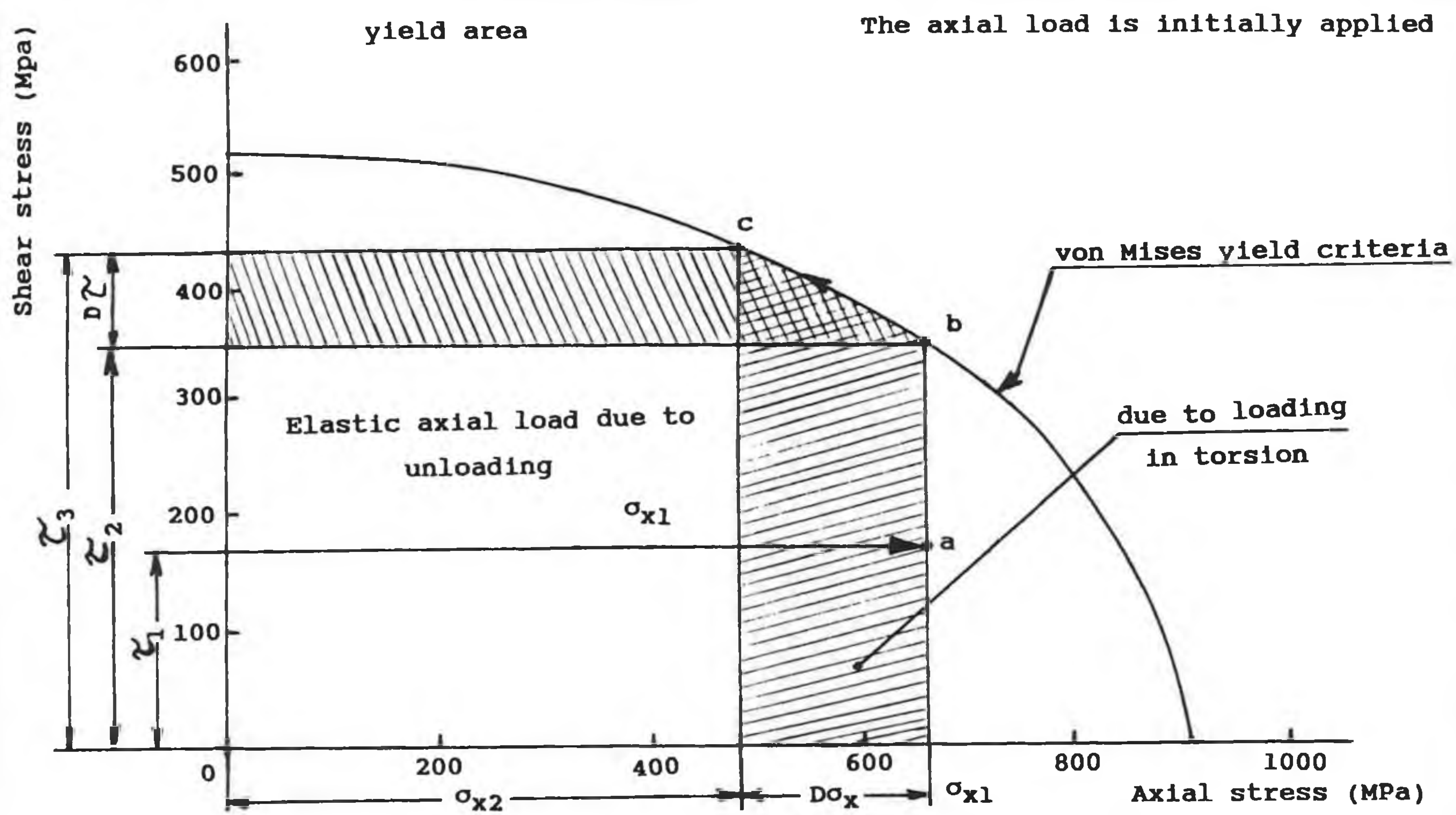


Fig.(6.12) Dissipation of tensile strain energy according von Mises yield criteria.

CHAPTER SEVEN

CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK

CHAPTER SEVEN

CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK

The following conclusions were drawn from the experimental and theoretical work carried out in the present study regarding the tightening of bolted joints.

7.1 General conclusion

7.1.1 Tensile test

1. The material of the bolt is elastic and marginally work hardening.
2. The yield stress in tension is 900 MPa at 5 percent proof strain.
3. The stress in the threaded section was calculated to be 65 percent of the yield stress.
4. The reduced section of the bolt has extension equal to 52.3 percent of total extension in the fastener, while the threaded section has extension equal to 16.5 percent of total extension.
5. Youngs modulus `E' was found to be 218.0 GPa. This compares favourably with accepted theoretical value of (190-210) MPa [46,47].

7.1.2 Pure torsion test

1. The maximum angle of twist is in the reduced section of the fastener which is about 65.5 percent of the total angle of twist. The minimum angle of twist is in the threaded section of the fastener which is approximately 15.4 percent of the total angle of twist in the fastener.
2. The modulus of elasticity in shear was obtained experimentally to be 72.57 GPa, which is very close to the generally accepted value of between 77 and 83 GPa [48,49].
3. The maximum elastic torque capacity of the bolt is 52.21 N.M.
4. The shear yield stress was determined from **von Mises yield criteria** [46] as 0.577 times the yield stress in tension. Hence, the yield stress in torsion is 519.615 MPa.
5. When the bolt starts to yield in torsion, it is calculated that the shear stress in the threaded section is approximately 54 percent of the yield stress in torsion.
6. The magnitude of the torque for the threaded section to start to yield is 98 N.M.
7. The ultimate torque was detected in the reduced section of the fastener using the torque wrench as 69.5 N.M. At this torque the bolt section of diameter 8 mm is plastically deformed so that a core of only 6 mm diameter remains elastic.

7.2 Specific conclusions on combined torque and axial load test

Two theoretical approaches were used to explain the experimentally observed behaviour of the bolt subjected initially to a torque or axial load and then subsequently to increasing axial load or torque respectively.

(i) Unloading of the torque or axial load commences only when the combined stress at the outer surface of the bolt reaches the uniaxial yield stress of the material. Furthermore as the axial load or torque are increased further, the Unloading of the torque or axial load continues without any progression of the plastic zone towards the inner core of the bolt.

(ii) Unloading of the torque or axial load would take place through progressive increase in the width of the plastic zone at the outer surface of the bolt once the combined yield stress has reached the magnitude of the uniaxial yield stress in tension.

7.2.1 THE FASTENER INITIALLY SUBJECTED TO A TORQUE

In this case, the experimental results show a trend which suggests that the unloading of torque initially applied torque would take place according to mixed mode response. The experimental torque reduction curves lie within the envelop contained by the limit curves following the two theoretical approaches. Thus some plastic zone would form near the outer surface of the bolt.

The reduction in torque due to axial elongation of the bolt is negligibly small compared to the reductions according to the above approaches.

The torsional energy may be partially or totally dissipated due to the plastic work done in the bolt.

7.2.2 THE FASTENER INITIALLY SUBJECTED TO AN AXIAL LOAD

In this case, the experimental results show that the reduction in the initially applied axial load is very small even after the combined stress due to the axial load and torque has reached the uniaxial yield stress. This is due to the fact that when any additional torque is applied after the combined stress reaches the yield locus, off-loading of the axial stress and formation of plastic zone continues simultaneously. such action continues until the outer plastic zone progresses towards the central core and makes the bolt cross-section totally plastic. At this point all of the initially applied axial load would disappear. Thus, experimentally observed results clearly show that off-loading of the axial load is not governed by the yield criteria relevant to the outer surface of the bolt but by the criteria for yielding of the cross-section of the bolt in pure torsion.

7.2.3 EFFECT OF INITIALLY TORQUE ON THE SUBSEQUENT YIELD STRESS IN AXIAL LOAD

experimental results demonstrate that when a bolt is subjected to an initial torque (with or without accompanying axial stress), and then subjected to increasing axial load, there would be no significant fall in the yield stress of the material. This would be true, irrespective of the magnitude of the initially applied torque. Theoretical approaches also suggest the some conclusion. The torque disappears and the torsional energy is dissipated in plastic strain in plastic zone at the outer surface of the bolt.

The loss of torque does not take place due to any unwinding of the bolt during the application of the axial load.

7.3 Suggestions for further work

Several aspects have been considered in the present work which affect the response of the bolt under loading. Some of these have been investigated thoroughly, but others have not been investigated due to the time available and the scope of the present work.

1. It is of interest to study the case when the axial load and torque are applied to the fastener simultanceously and alternately. Under these conditions it is suggested to establish:

- The mechanism in which yielding takes place under combined stresses.
- Effect of torque on the axial stress and effect of axial stress on the torque during combined stressing.

2. study the case when the torsional and tensile strain gauges are attached to each section of the bolt, in order to establish the torsional and tensile stress behaviour in each section of the bolt under combined stresses.

3. To make the experimental work more accurate by using a purpose built device to apply uniaxial tensile load and torque independently and/or simultaneously instead of using the hydraulic cylinder and wrench torque.

4. It is of interest to establish the precise mechanism of off-loading the torque or axial stress by investigating the plastic zone formation in the bolt by micro-structural investigation or other means.

5. It would be of interest to provide experimental and/or theoretical explanation of the precise manner in which the torsional and tensile energy is dissipated.

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APPENDIX A1

Mechanics of engineering materials

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MECHANICS OF ENGINEERING MATERIALS

A1.1 INTRODUCTION

Deformation may occur in a material for a number of reasons, such as external applied load, change in temperature, tightening of bolts, irradiation effects, etc. [16]. Bending, twisting, compression, torsion and shear or combinations of these are common modes of deformation .

A1.2 THE THEORY OF ELASTICITY

A1.2.1 STRESS-STRAIN RELATIONS

A material is said to be elastic if it returns to its original, unloaded dimensions when load is removed. Within the elastic limits of material for which **Hooke`s law** is applicable, the following equation can be used [23].

$$E=\sigma/\epsilon. \qquad \textbf{(A1.1)}$$

Consider the member shown in **Figure (A1.1)** subjected to axial tensile loading **F**. If the resulting extension of the member is **DL** and its unloaded length is L_0, the following equations can be written [16,18]:

- The engineering strain $e=DL/L_0$. **(A1.2)**

- The nominal stress $\sigma_0=F/A_0$. (A1.3)

- The natural strain $\epsilon=\ln(L_1/L_0)$. (A1.4)

- The true stress $\sigma=F/A_1$. (A1.5)

That is $\epsilon=\ln(1+e)$. (A1.6)

And $\sigma=\sigma_0(1+e)$. (A1.7)

The length L_0 is not the length of the sample but it is the original gauge length. The nominal stress is defined as the load divided by the original cross-sectional area of the member. A nominal stress-strain curve is therefore simply a load extension diagram for a member with unit original cross-sectional area and unit gauge length. The true stress-strain curve is a more informative diagram for plasticity purposes [18].

A1.2.2 ELASTICITY EQUATIONS FOR STRESS STATES

Linear relations between the components of stress and the components of strain are known generally as Hooke`S law. When an element of material is subjected to triaxial stresses σ_x , σ_y and σ_z as shown in **Figure (A1.2)** uniformly distributed over the sides, the total strain is obtained by superimposing each influence from the **X,Y** and **Z** directions:

$$\epsilon_x=[\sigma_x-\nu(\sigma_y+\sigma_z)]/E. \quad \text{(A1.8)}$$

$$\epsilon_y=[\sigma_y-\nu(\sigma_z+\sigma_x)]/E. \quad (A1.9)$$

$$\epsilon_z=[\sigma_z-\nu(\sigma_x+\sigma_y)]/E. \quad (A1.10)$$

Where ν is called the Poissons ratio, relating lateral strain to direct strain.

It can be shown that the constants of elasticity **E, G** and are related as [**47**].

$$G=E/2(1+\nu). \quad (A1.11)$$

A1.3 THE THEORY OF PLASTICITY

A1.3.1 THE PLASTIC BEHAVIOUR OF MATERIALS

Most applications of the mathematical theory of plasticity are based on the REUSS stress-strain relations [**19,48**]. The REUSS relations suggest that each increment of strain can be resolved into additive elastic and plastic parts:

(i) The elastic part being governed by the elastic equations.

(ii) The plastic part being described by the **LEVY-MISES** relations.

The onset of plastic yield takes place sharply either from zero strain, in which case the property of material is rigid-plastic see **Figure (A1.3)**, or from an elastic strain, in which case the property of material is elastic-plastic, see **Figure (A1.4)**. Work-hardening materials have also been divided into rigid-work hardening or elastic-work hardening see **Figure (A1.5)** and **Figure (A1.6)**.

For isotropic materials the yield stress is independent of the direction of straining. The plastic stress-strain relations are used only when all of the material has reached yield. By drawing a line parallel to the elastic line at a stated **0.2%** strain on the stress-strain curve. a proof stress value may be established [18].

A1.3.2 THE PRANDTL-REUSS EQUATIONS

For the stress-strain relations for an elastic-perfectly plastic material, **REUSS** assumed that the plastic strain increment is at any instant proportional to the instantaneous stress deviation and the shear stresses as follows:

$$d\epsilon_x^p/\sigma'_x = d\epsilon_y^p/\sigma'_y = d\epsilon_z^p/\sigma'_z = d\lambda. \qquad \textbf{(A1.12)}$$

$$d\gamma_{yz}^p/\tau_{yz} = d\gamma_{zx}^p/\tau_{zx} = d\gamma_{xy}^p/\tau_{xy} = d\lambda. \qquad \textbf{(A1.13)}$$

Or

$$d\epsilon_{ij}^p = \sigma'_{ij}.d\lambda. \qquad \textbf{(A1.14)}$$

The total strain increment is the sum of the elastic strain increment and plastic strain increment so that,

$$d\epsilon_{ij} = d\epsilon_{ij}^{p} + d\epsilon_{ij}^{e}. \qquad (A1.15)$$

Considering principal stress directions we have,

$$(d\epsilon_1^{p} - d\epsilon_2^{p})/(\sigma_1 - \sigma_2) = (d\epsilon_2^{p} - d\epsilon_3^{p})/(\sigma_2 - \sigma_3);$$

$$= (d\epsilon_3^{p} - d\epsilon_1^{p})/(\sigma_3 - \sigma_1) = d\lambda. \qquad (A1.16)$$

Since the principal stresses are equal to hydrostatic stress plus deviatoric stress then the following equations can be written:

$$\sigma'_x = \sigma_1 - \sigma_h. \qquad (A1.17)$$

$$\sigma'_y = \sigma_2 - \sigma_h. \qquad (A1.18)$$

$$\sigma'_z = \sigma_3 - \sigma_h. \qquad (A1.19)$$

$$\sigma_h = (\sigma_1 + \sigma_2 + \sigma_3)/3. \qquad (A1.20)$$

Also plastic straining causes no change of volume so that

$$d\epsilon_1^{p} + d\epsilon_2^{p} + d\epsilon_2^{p} = d\epsilon_x^{p} + d\epsilon_y^{p} + d\epsilon_z^{p} = 0; \qquad (A1.21)$$

or

$$d\epsilon_{ij}^{p} = 0. \qquad (A1.22)$$

The **PRANDTL-REUSS** equation (A1.13) can be written as three equations in X , Y and Z direction for example, in the X direction we get:

$$d\epsilon_x{}^p=2.d\lambda[\sigma_x-(\sigma_y+\sigma_z)/2]/3; \qquad \textbf{(A1.23)}$$

The **PRANDTL-REUSS** equation may be then be written as:

$$d\epsilon_{ij}=\sigma'_{ij}.d\lambda+d\sigma'_{ij}/(2G). \qquad \textbf{(A1.24)}$$

And

$$d\epsilon_{ii}=(1+2\nu).d\sigma_{ii}/E; \qquad \textbf{(A1.25)}$$

These equations for an elastic-plastic solid are usually difficult to calculate by hand in a real problem.

Stress-strain relations for such a material were proposed by **LEVY** and **VON MISES [18]**. According to **FORD [19]**, the incremental strain may be written in the form of **REUSS** equations in shear as follows:

$$d\gamma=\tau.d\lambda+d\tau/(2G). \qquad \textbf{(A1.26)}$$

This equation becomes equal to zero in the plastic range since the shear stress is constant **[24]**.

A1.3.3 <u>THE THEORY OF HOHENEMSER'S EXPERIMENT</u>

HOHENEMSER`s experiment endeavoured to investigate the stress-strain relationship in complex stress state. However, the theory developed is useful in explaining how shear stress responds in the plastic range to the combination with tensile stress which are applied in bolted joint. In this experiment outlined by FORD [19] and HILL [48] a hollow cylindrical tube was twisted to the point of yielding, then holding this angle of twist constant, the tube was extended longitudinally. In this case **PRANDTL REUSS** relations are expressed as follows:

$$d\epsilon=2.d\lambda.\sigma/3+d\sigma/E; \qquad \textbf{(A1.27)}$$

During external loading the shear strain remains constant that is:

$$\gamma=\text{Constant} \quad \text{And} \quad d\gamma=0$$

Thus, from equation (A1.26) the following equation is obtained,

$$d\lambda=-d\tau/(2.G.\tau); \qquad \textbf{(A1.28)}$$

By substituting equation (A1.28) in equation (1.26) the following equation is obtained,

$$d\epsilon=-\sigma.d\tau/(3.G.\gamma)+d\sigma/E; \qquad \textbf{(A1.29)}$$

In the case of normal tightening of the fastener, the stresses in one direction need only be considered, so that the following equation is applicable:

$$Y=\sqrt{\sigma^2+3\tau^2} \ . \tag{A1.30}$$

Also, by substituting equation (A1.30) in equation (A1.29) the following equation is obtained,

$$d\tau/\tau=-\sigma.d\sigma/(Y^2-\sigma^2); \tag{A1.31}$$

After integrating equation (A1.31) we can obtain the following equation:

$$\epsilon=Y.\{(1-2\nu)S+(1+\nu)\ln[(1+S)/(1-S)]\}/(3E). \tag{A1.32}$$

Where:

$$S=\sigma/Y \qquad (\ S=0 \quad \text{when} \quad \epsilon=0\ )$$

Before the tension is applied, the shear stress is a maximum so that,

$$\sigma=0 \qquad , \qquad \tau=Y/\sqrt{3}$$

Which is the yield stress in pure shear.

From equation (A1.32) the strain is infinite when S=1, σ=Y. When ϵ=Y/3E,

$$(1-2\nu)S+(1+\nu).\ln[(1+S)/(1-S)]=1 \tag{A1.33}$$

And S=0.99. Thus for quite small strains σ is already practically equal to Y. Therefore $\tilde{\tau}$' rapidly approaches zero.

Where:

- L_0 Is the original gauge length.
- L_1 Is the current length.
- A_0 Is the original area.
- A_1 Is the current area.
- E Is the modulus of elasticity in tension.
- G Is the modulus of elasticity in torsion.
- Is the Poisson`S ratio.
- σ_h Is the hydrostatic stress.
- $d\lambda$ Is the proportionality factor which relates the incremental plane strain $d\epsilon pij$.
- σ_{hij} Is the deviatoric stress component.
- $\sigma_x,\sigma_y,\sigma_z$ Is the axial stress in the directions X,Y and Z.
- $\epsilon_x,\epsilon_y,\epsilon_z$ Is the strain in the directions X,Y and Z.
- $\sigma_x`,\sigma_y`,\sigma_z`$ Is the hydrostatic stresses in the direction X,Y and Z.
- $\sigma_1,\sigma_2,\sigma_3$ Is the principal stresses.

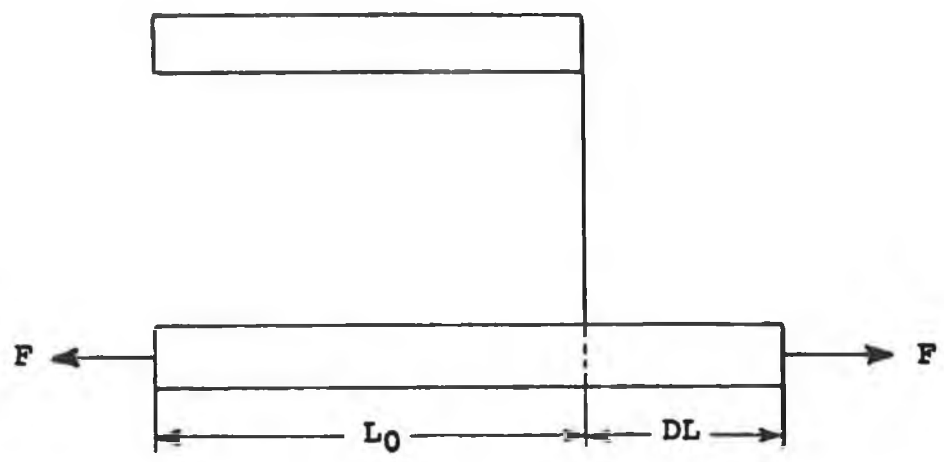

Fig.(A1.1) Bar is subjected to axial tensile loading.

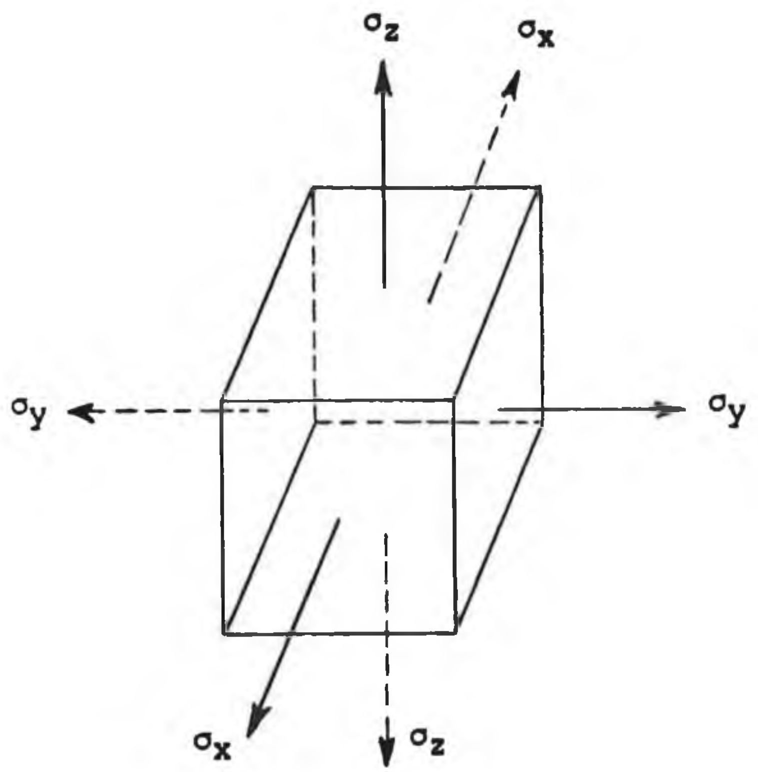

Fig.(A1.2) An element of material is subjected to triaxial stresses.

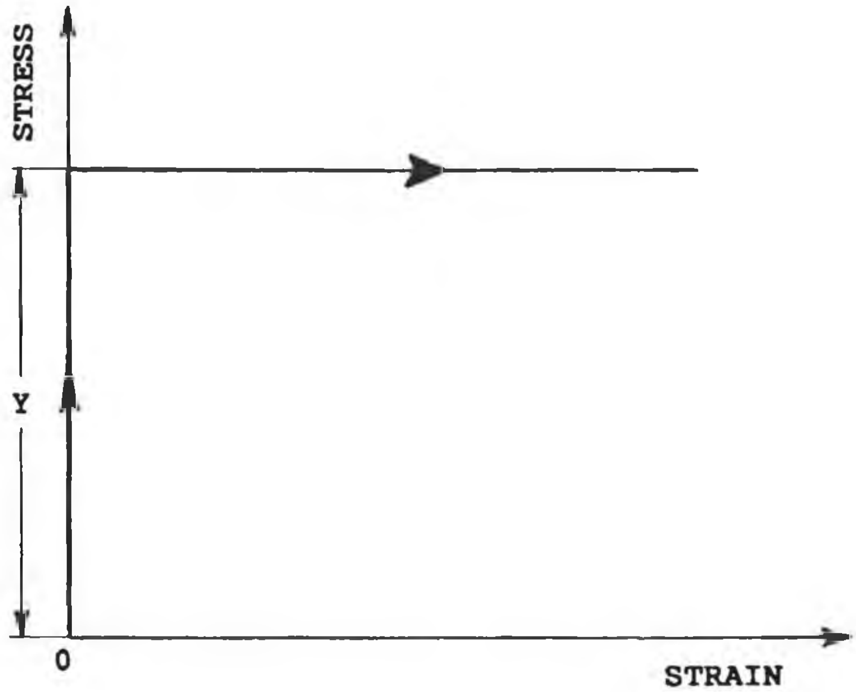

Fig.(A1.3) Relationship between stress and strain to Rigid-Plastic material.

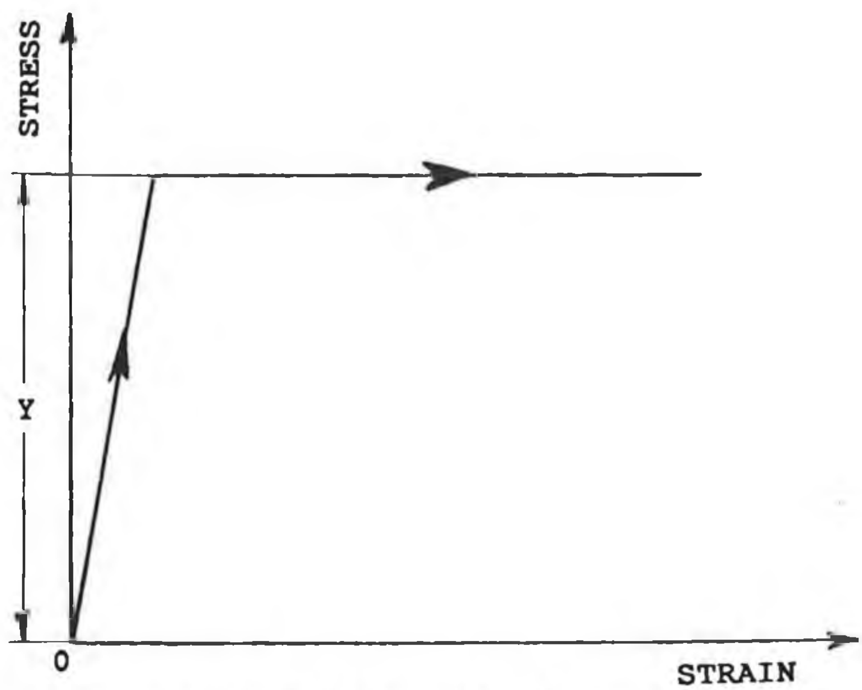

Fig.(A1.4) Relationship between stress and strain to Elastic-Plasti material.

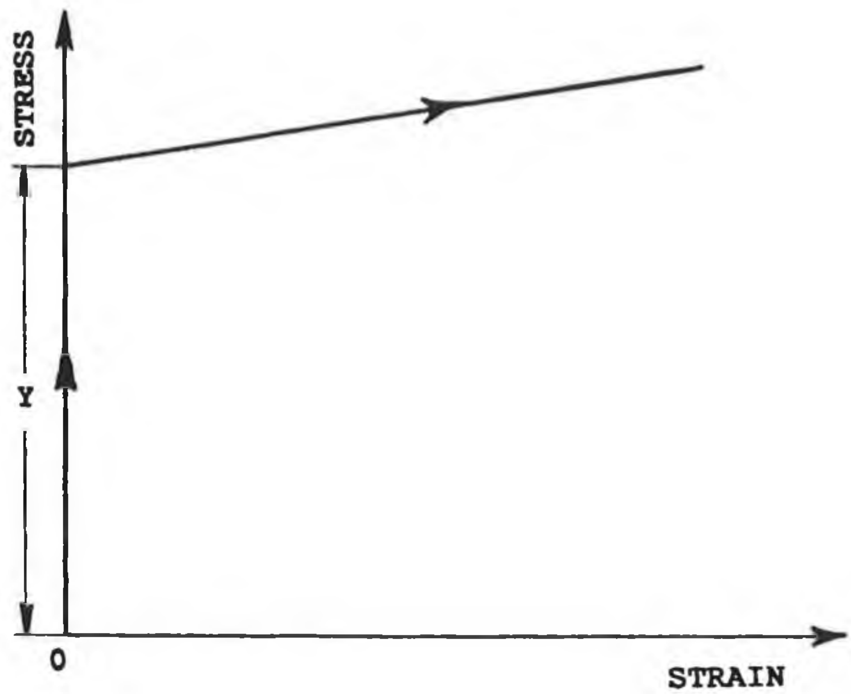

Fig. (A1.5) Relationship between stress and strain to Rigid-Work Hardening material.

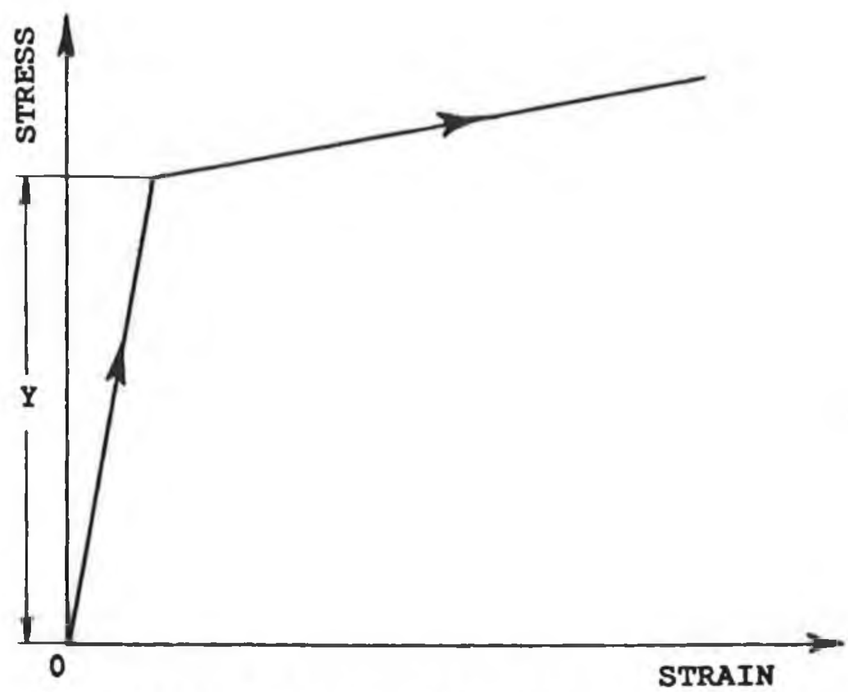

Fig. (A1.6) Relationship between stress and strain to Elastic-Work Hardening material.

APPENDIX A2

The spring constant for clamped parts

APPENDIX A2

THE SPRING CONSTANT FOR CLAMPED PARTS

The resilience factor for a prismatic part is as follows:

$$\delta_J = L/E.A. \qquad \textbf{(A2.1)}$$

Where, **A** is the cross section area in **mm²**, **E** is the modulus of elasticity in tension in **GPa**, and **L** is the length of clamped parts in **mm**.

For clamped parts of large cylindrical volume complex equations are applicable as shown in **Figure (A2.1)**, subsequently the elastic resilience of clamped parts can be calculated according to the equation (A2.1) by replacing:

$$A=A_S \quad , \quad L=L_J \quad \text{and} \quad E=E_J$$

Then, the following equation can be obtained:

$$\delta_J=L_J/E_J.A_S. \qquad \textbf{(A2.2)}$$

The results of comparative investigation have shown that all proposals made in the reference literature accurately follow up to plate thickness of 8d, where d is the nominal diameter of bolt.

This describes the cross sectional area of a hollow cylinder which elastically behaves similar to the plates that are under compression by bolt head and nut [34,36].

To calculate the area (A_S) there are two cases:

1. When $D_J <= D_H$

   Then, the following equation can be used:

$$A_S = \pi.(D_J^2 - D_h^2)/4. \quad \text{(A2.3)}$$

2. When $D_J > D_H$ Then, there are two cases:

2.a When $3D_H > D_J > D_H$

   Then, the following equation can be used:

$$As = \pi.(D_H^2 - D_h^2)/4 + \pi.(D_J/D_H - 1).(D_H.L_J/5 + L_J^2/100)/8. \quad \text{(A2.4)}$$

2.b When $D_J >= 3D_H$

   Then, the following equation can be used:

$$A_S = \pi.[(D_H + 0.1L_J)^2 - D_h^2]/4. \quad \text{(A2.5)}$$

The resilience of clamped parts is valid for concentrically clamped parts,for eccentric bolted joint,that is, bolts in an eccentric location and with eccentrically applied external load ,in this case the the resilience of clamped parts increases, see **Figure (A2.1)**.

Where:

D_j is the outside diameter of cylinder or bushing.

D_H is the head bolt or washer diameter.

D_h is the hole diameter in the clamped parts.

D_S is the diameter of a substitute cylinder to be determined.

L_j is the original joint length or length of joint.

A_S is the area of a substitute cylinder to be determined or substitutional area of a hollow cylinder with the same resilience as clamped plates.

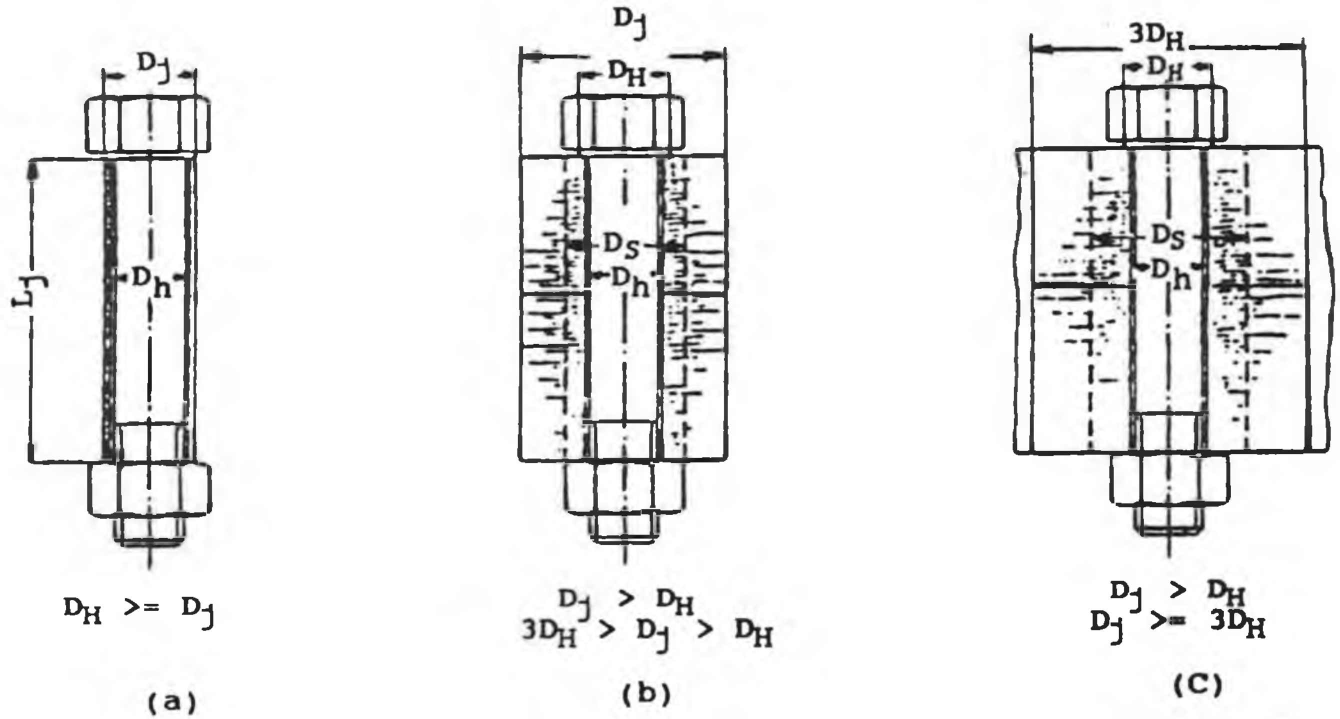

Fig. (A2.1) Substitutional compression Solid for calculation of the resilience of clamped sleeves and plats.

APPENDIX A3

The load factor for eccentric bolted joint

APPENDIX A3

THE LOAD FACTOR FOR ECCENTRIC BOLTED JOINT

The most complicated case of load introduction are the eccentric ones. The most joints are loaded in this way, the resilience of the clamped parts increases.

Further, when the load is applied eccentrically to the bolt axis and joint centre line, the resilience factor for eccentric position of the bolt axis is δ_j^{*} and for an additionally eccentrically applied working load δ_j^{**} [36]. To calculate the load factor there are two cases:

1. in the case of load introduction under bolt head and nut the load factor is given by the following equation [34,36], see in **Figure (A3.1)**.

$$\Phi_{ek}=\delta_j^{**}/(\delta_b+\delta_j^{*}). \qquad \textbf{(A3.1)}$$

2. For the most general case of load introduction which is both eccentrically applied and in levels of a distance $(n.L_j)$ the term Φ_{en} can also be derived by the following equation, see **Figure (A3.2)**.

$$\Phi_{en}=n.\delta_j^{**}/(\delta_b+\delta_j^{*}). \qquad \textbf{(A3.2)}$$

For bolts in an eccentric location and with an eccentrically applied working load, the resilience of the clamped parts increases, the calculations are made under the following simplifying conditions:

- The clamped parts are represented as a prismatic solid.
- The clamped parts in the interface section, the interfacial pressure is greater than zero at the bending tension side.
- All cross-sections of this prismatic solid remain plain under load and there are linear stress distribution.

Under these conditions these are valid for deformation in the bolt axis [34,36]. To calculate the magnitude of δ_j^* and δ_j^{**}, there are two cases:

1. For eccentric load in the distance `S' from the axis of gyration and with load introduction in the eccentrically located bolt axis `a=s', then the following equation can be used:

$$\delta_j^* = \delta_j . (1+S^2/R_k^2). \qquad \textbf{(A3.3)}$$

2. For eccentric load in the distance `S' from the axis of gyration and with a load introduction in the distance `a' from the axis of gyration, then as a general case the following equation can be obtained:

$$\delta_j^{**} = \delta_j . (1+S.a/R_k^2). \qquad \textbf{(A3.4)}$$

Where:

- a is the eccentricity of load.
- A_j is the cross section of the substitutional bending solid.
- I is the moment of inertia of interface.
- L_j is the length of clamped parts.
- S is the eccentricity of bolt axis.
- δ_j is the resilience of clamped parts for concentric bolt `S=0' and also with concentric load introduction `a=0'
- δ_b is the resilience of bolt `S=0 , a=0'.
- R_k is the radius of inertia which is given by the following equation:

$$R_k=\sqrt{(I/A_j)}. \tag{A3.5}$$

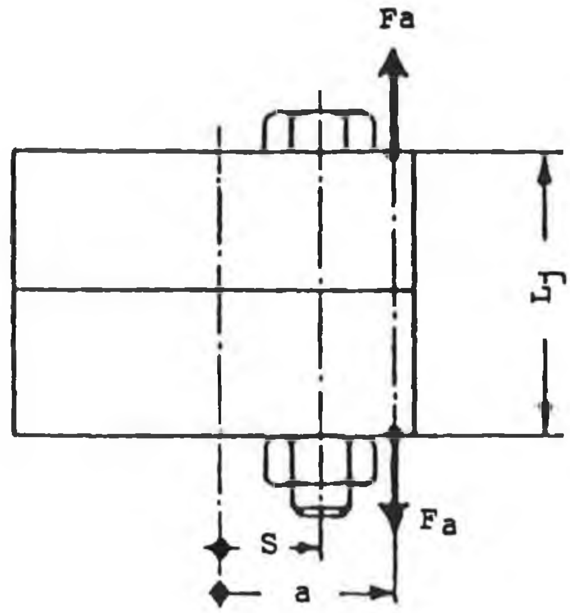

Fig. (A3.1) The working load introduction under bolt head for eccentric bolted joint.

$L_k = n.L_j$

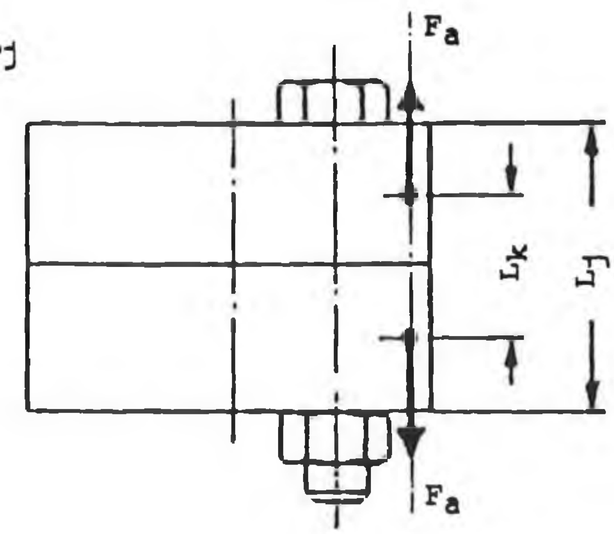

Fig. (A3.2) The working load introduction in levels for eccentric bolted joint.

APPENDIX A4

Pre-load loss due to permanent set

APPENDIX A4

PRE-LOAD LOSS DUE TO PERMANENT SET

The bolts in all joints lose pre-load from relaxation of the bolt, joint and any gasket, if permanent set occurs some of the pre-load may be lost and premature failure may occur. The external loads cause only a small additional plastic elongation of the bolt above that resulting from relaxation or permanent set. Permanent set can occur from elongating the bolt beyond its yield point during tightening.

The diagram in Figure **(A4.1)** is for a bolted joint tightened only into the elastic region, in this case the preload DF_i is directly proportional to the amount of permanent set `**S**' and the total joint spring rate `**K**' that is:

$$DF_i = S.K, \tag{A4.1}$$

In addition,

$$S_b=F_i/K_b. \tag{A4.2}$$

$$S_j=F_i/K_j. \tag{A4.3}$$

From equations (A4.1), (A4.2) and (A4.3) the following equation can be obtained:

$$S=S_b+S_j \qquad \textbf{(A4.4)}$$

When tightening the bolt beyond the yield strength, an additional amount of plastic bolt elongation is created by an additional external load causing further plastic elongation ($S^`$)which creates a per-load loss ($DF_i^`$). **Figure (A4.2)** shows that the total amount of pre-load loss, in the elastic-plastic joint is greater than in the elastic joint due to the plastic bolt elongations.

The minimum remaining pre-load ($F_i^`$) in the joint is greater in the elastic-plastic joint because of the high tightening load, so that, the following equation can be obtained:

$$DF_{itot}=DF_i+DF_i' \qquad \textbf{(A4.5)}$$

For example, if three bolted joints are subjected to the same pre-load `$F_i$',the same external load `F_a' and the same permanent set `S', these bolted joints are tightening below yield point. Increasing the flexibility of both the bolt and the clamped parts reduces the pre-load loss caused by permanent set [33,35].

By comparing between these cases the results gave a spring bolt and medium stiff clamped part is the best design for minimizing pre-load loss as well as dynamic bolt loads. The joint conditions required to minimize permanent set are:

- Small number of interface.
- Perpendicular parts with no bending.
- Low bearing stresses.
- Sufficient length of engaged thread.
- Heavy nut type.
- Smooth surface finish.
- No gasket or no plastics within joint interface.
- No spring washers.
- Sufficient nut hardness.

Also for the joint designs to minimize pre-load loss due due to permanent set are :

- Use of high strength bolt.
- Long bolts should be greater than or equal to 5d.
- Reduced both shank diameter.
- Longer joint length.
- Elastic bolt head and/or elastic nut.
- Materials with low modulus of elasticity.
- Hardened washer.

With the development of the pre-load loss equation, it is possible to predict the amount of permanent set and also the pre-load loss by using the nomographs which are given for industrial or aircraft fastener [33].

All specialists in the field of bolted joints agree that the value of the pre-load is one of the essential parameters of the joint. A carefully chosen pre-load will:

- Ensure that the shank of the bolt is not loaded in shear.
- Ensure that any sealing is maintained.
- Control any self-slacking due to the application of dynamic shear loads.
- Allow the mechanical characteristics of the bolts to be used more effectively,thus allowing their diameter and number to be reduced leading to a reduction in procurement, machining and assembly costs.
- The effect of an external loading on the bolt in a ratio which is a function of the apparent stiffnesses of the assembled parts and of the bolt [25].

Where:

F_i	Is the initial pre-load.
K_b	Is the spring constant for clamped parts.
K_j	Is the spring constant for bolt.
K	Is the spring constant for a bolted joint.
S_b	Is the permanent set in the bolt.
S_j	Is the permanent set in the clamped parts.
S	Is the permanent set in a bolted joint due to elastic deformation.
$S^`$	Is the permanent set in a bolted joint due to plastic deformation.

DF_i Is the pre-load loss due to permanent set from elastic deformation in a bolted joint.

$DF_i`$ Is the pre-load loss due to permanent set from plastic deformation in a bolted joint.

DF_{itot} Is the total pre-load.

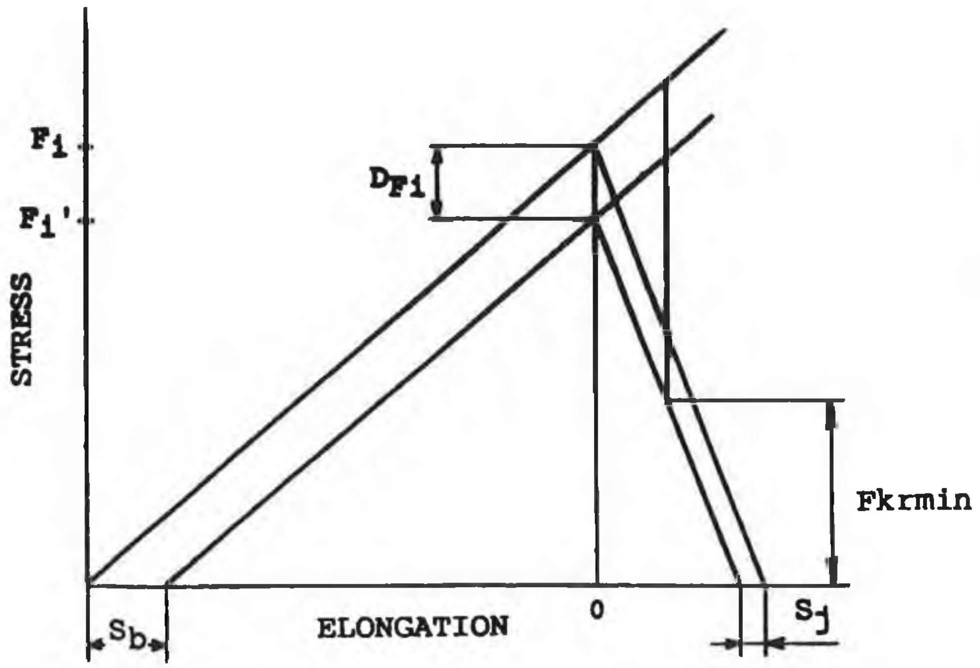

Fig. (A4.1) Digram pre-load loss and remaining load in elastic joint.

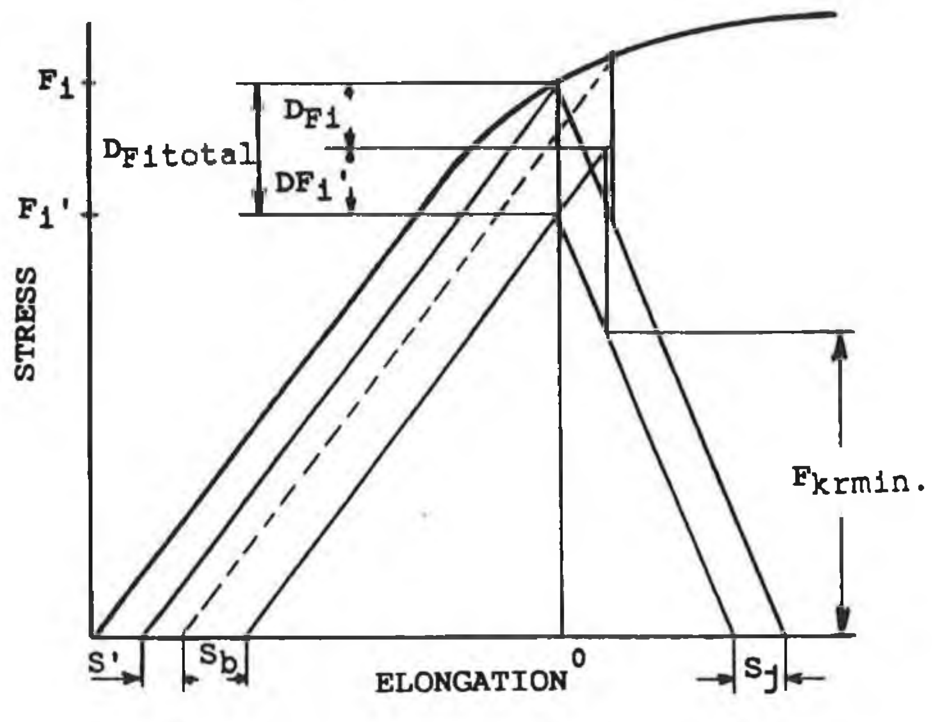

Fig. (A4.2) Digram pre-load loss and remainingload in elastic-plastic joint.

APPENDIX A5

Heat treatment methods and manufacture of bolts

APPENDIX A5

HEAT TREATMENT METHODS AND MANUFACTURE OF BOLTS

A5.1 MANUFACTURE OF BOLTS

External threads are formed by machining, thread cutting, thread grinding or rolling. Nut blanks are made by cold forming, hot forming, cold punching or milling from bar stock.

Operations which may be used in the production of the bolts, screws and nuts include drilling, sawing, slotting, shaping, trimming, milling, grinding, turning, pointing, polishing and plating, as well as heat treatment. Relieved body bolts are used for locating or centering members where shear and bending forces are present . In such cases the minimum shank area is determined by tensile strength requirements,the shoulder areas which then act as guide surfaces should be separated from both the head and thread [5].

A5.2 HEAT TREATMENT METHODS FOR BOLTS

Any of the following seven different heat treatment methods may be used to enhance the properties of steel fasteners. These methods are:

A5.2.1 QUENCHING

To refine the structure of steel and to harden it to a consistent quality, quenching is normally applied to bolts, screw, pins and washers,and to a lesser extent to medium carbon nuts.

A5.2.2 <u>ANNEALING</u>

To soften the steel,primarily applied to studs which are to be cold-hardened but also to special bolts to improve their ductility. Various degrees of softness can be produced by annealing.

A5.2.3 <u>TEMPERING</u>

To control the properties of the steel and also to a black corrosion resistance surface ` by cooling in soluble oil'.

A5.2.4 <u>STRESS RELIEVING</u>

To relieve high stresses generated during cold forming, a grain, a black corrosion-resistance coating.

A5.2.5 <u>CARBURISING OR CASE HARDENING</u>

To produce a hard wear-resistant surface with a softer interior , commonly applied to bolts, pins and nuts where resistance to wear is important.

A5.2.6 <u>CYANIDE HARDENING</u>

Producing an extremely hard surface of shallow depth for high wear resistance, etc. on bolts, pins and nuts.

A5.2.7 DRY CYANIDING

An alternative case hardening process for bolts, pins, etc. [5].

A5.3 ROLLED VERSUS CUT THREADS

Thread cutting or grooving by machine produces cuts across the grain lines of the metal resulting in a more uniform distribution of the stress when the thread are loaded in tension or compression.

Thread rolling produces plastic deformation of the metal with grain generally following the contour of the thread. There is also a compacting of the grain at the root of the thread resulting in increased strength at critical area because of compression face stressing. A rolled thread therefore, is inherently stronger than a cut thread especially under condition of dynamic loading.

The chief limitation with thread rolling is that the material to be formed must be sufficiently ductile to permit plastic deformation without cracking, incipient failure or embrittlement. Short threads are generally formed by`plunge rolling',longer threads are formed by `through-rolling'.

Threads may be rolled after heat treatment when full benefits of compressive re-stressing are realised, if heat treatment is applied after thread rolling, then much of the advantage of rolling is lost because heat treatment will produce stress relief[5]. Where threads are rolled before heat treatment residual stresses are relieved by the treatment which should reduce fatigue strength to the level obtained with cut or ground threads. Threads can be formed by rolling, cutting or grinding.

Rolling is more accurate than cutting or grinding because it is done by hardened dies with radius tolerances of 0.005 to 0.008 inches. Rolled threads offer the best resistance to fatigue because the rolling process produces continuous grain structure at the thread root [26].