# STUDY OF JOINTING BOLTS UNDER TENSILE LOADING


MD. ASADUL HAQUE





DEPARTMENT OF MECHANICAL ENGINEERING

BANGLADESH UNIVERSITY OF ENGINEERING & TECHNOLOGY

DHAKA-1000, BANGLADESH

# STUDY OF JOINTING BOLTS UNDER TENSILE LOADING.

BY
MD. ASADUL HAQUE
B.SC.ENGG.(MECH.)

A THESIS
SUBMITTED TO THE DEPARTMENT OF MECHANICAL ENGINEERING IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR
THE DEGREE OF MASTER OF SCIENCE IN MECHANICAL ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING
BANGLADESH UNIVERSITY OF ENGINEERING & TECHNOLOGY
(BUET)
DHAKA,BANGLADESH

JUNE 2001

## CERTIFICATE OF APPROVAL

The board of examiners hereby recommends to the Department of Mechanical Engineering, BUET, Dhaka the acceptance of this thesis," **STUDY OF JOINTING BOLTS UNDER TENSILE LOADING"** submitted by Md. Asadul Haque , in partial fulfilment of the degree of Master of science in Mechanical Engineering.

17/7/2010

| | | |
|---|---|---|
| Chairman(supervisor) | : | Dr.Abu Rayhan Md. Ali<br>Associate Professor<br>Department of Mechanical Engineering<br>BUET, Dhaka, Bangladesh. |
| Member(EX-officio) | : | Dr.Md. Abdur Rashid Sarkar<br>Professor & Head<br>Department of Mechanical Engineering<br>BUET, Dhaka, Bangladesh. |
| Member | : | Dr. Dipak Kanti Das<br>Professor<br>Department of Mechanical Engineering<br>BUET, Dhaka, Bangladesh. |
| Member | : | Dr. Muhammad Mahbubul Alam<br>Professor<br>Department of Mechanical Engineering<br>BUET, Dhaka, Bangladesh. |
| Member(External) | : | Dr. Abdul Muqtadir<br>Professor<br>Department of civil Engineering<br>BUET, Dhaka, Bangladesh. |

## CERTIFICATE OF RESEARCH

This is to certify that the work presented in this dissertation is the outcome of the investigation carried out by the author under the supervisor of Dr. Abu Rayhan Md. Ali, Associate professor, Department of Mechanical Engineering, BUET, Dhaka, Bangladesh and it has not been submitted anywhere of the award of any degree or diploma.

Supervisor:
Dr. Abu Rayhan Md. Ali

Author:
Md. Asadul Haque

# ACKNOWLEDGEMENT

The author would like to take this opportunity of expressing his heartfelt gratitude and indebtedness to his thesis Supervisor Dr. Abu Rayhan Md. Ali, Associate professor Department of Mechanical Engineering, BUET, Dhaka , Bangladesh for his valuable and continuous guidance, suggestions in the course of the present work.

The author feels grateful to Dr.Md. Abdur Rashid Sarkar, Professor & Head of the Department of Mechanical Engineering, BUET, for his valuable suggestions from time to time.

The author is also grateful to all of the technical staff of applied machanics laboratory for their relentless effort and rendering assistance to carry out detail experiment.

Finally the author also thanks to his colleagues, office staff, family and relatives, who shared the troubles with him.

# ABSTRACT

In this modern era a lot of development has taken place in jointing technology but still the bolted joint is the most important jointing method. Bolted joints are mostly used in motor transport, aircraft and other machine tool industries. The effectiveness of a bolted joint is usually measured by its ability to maintain an adequate level of clamping force on the jointing surface. It has been found that friction condition between the bolt under head and the matting surface has greatly influenced the magnitude of the clamping force and wrenching torque. In the present study detail experimental investigation, regarding the behavior of jointing bolt under different lubricating condition as well as under external load was carried out. Further more, the relationship between the wrenching torque and the torque transmitted to the shank of the bolt was also established. It has been observed that the better the lubricating condition between the bolt under head and the mating surfaces, the better the torque transmitted to the shank of the bolt and hence better the pre load developed in the bolt body .

It has also been seen that in all the cases, the maximum load carrying capacity of a tightened bolt, when external tensile load was applied, remained the same. Moreover with the application of external tensile load, up to the ultimate load, the shank torque reduces to the residual torque of the bolt material. In order to do the aforementioned experimental investigation a test rig and a 30 kn capacity of compressive type load cell were designed, fabricated and calibrated. During experiment, attention was particularly given to find out the pre-load developed, torque transmission and the effect of external tensile load on shank torque. Finally it has been concluded that friction conditions between two matting surfaces not have any effect on the maximum load carrying capacity of the bolt.

## CONTENTS

# LIST OF FIGURES

# NOMENCLATURE

| | |
|---|---|
| **A** | cross sectional area of the bolt. |
| **a** | radius of the bolt. |
| **c** | elastic core radius |
| **E** | modulus of elasticity |
| **F** | axial load |
| $F_i$ | initial tightening load |
| G | Shear modulus |
| **J** | polar moment of inertia |
| $J_2$ | second invariant |
| **L** | length. |
| **P** | external tensile load |
| $P_b$ | partial tensile load taken by the bolt. |
| $P_m$ | partial tensile load taken by the member |
| **r** | radius |
| **T** | torque |
| $T_h$ | bolt under head torque |
| $T_w$ | wrenching torque |
| $T_t$ | thread torque |
| $T_s$ | shank torque |
| Y | yield strength |

***Greek***

| | |
|---|---|
| $\varepsilon$ | axial strain |
| $\gamma$ | shear strain |
| $\theta$ | total angle of twist |
| $\sigma$ | axial stress |
| $\tau$ | shear stress |
| $\Phi$ | angle twist |
| $\varphi_c$ | critical angle of twist |
| $\delta$ | diflection of the bolt |

*Subscript*

| | |
|---|---|
| **O** | Initial value |
| **e** | in elastic region |
| **p** | in plastic region |
| **u** | ultimate load |
| **oct** | octahedral |
| **Y,y** | yield value |

# CHAPTER - ONE
# INTRODUCTION

## 1.1 GENARAL

Despite the many advance in joint technology in recent years, bolted joints are still the most frequently used method of clamping assemblies together. This is still the case for many components used within the motor, aerospace, and machine tool industry. Consequently, factors affecting the design and integrity of bolted joints are of considerable industrial interest. The effectiveness of a bolted joint is usually measured by its ability to maintain an adequate level of clamping force on the joint surfaces during service. It has been shown [1-4] that uniformity in the magnitude of the clamping force exerted by each fastener of the joint is best achieved by tightening each fastener to its yield point. It has also been established [5-7] that the friction conditions at the fastener threads and under head contact surface greatly influence the magnitude of the joint clamp load. The friction conditions also influence the magnitude of the wrenching torque, Tw, require to achieve a given value of clamp force. As the friction conditions deteriorate, a larger wrenching torque is required to achieve a given level of clamp load. In addition, a greater proportion of the wrenching torque is required to overcome friction, thereby setting up increased torsional stresses within the fastener.

The practice of tightening critical bolts in automotive applications to clamping loads which are at or beyond the torque tension yield point is becoming more widespread, especially in Europe and Japan. The benefits of doing this result form the closer control of initial clamp load in the joint, which in turn results from a process which is controlled by bolt strength properties (yield tightening) rather than friction (torque tightening). The strength of bolts is much easier to control than their friction properties, so that the scatter in clamp load from yield or beyond yield tightening methods is typically less than one quarter of that in conventional torque tightening.

The design and assembly of bolted joints must assure that the joint remains tightly clamped, and the fastener is capable of withstanding the static and dynamic loads that are applied. In order to achieve this the designer must appreciate that the service performance of a joint not only depends on the fastener and the structure being clamped, but also the tightening process.

In establishing the design capability of a bolted joint, several points have to be considered :

a. How tight should the bolt be, and what assurance is there that the assembly process can consistently achieve this level of tightening ?

b. What load is felt by the bolt when the service stresses are applied to the fastened assembly ?

c. What level of external load will cause the joint to open ?

## 1.2 JUSTIFICATION

Bolted assemblies form a major part of most engineering structures, in particular, the machine tool and transport industries have continuously sought to improve the characteristics of bolted joints, so that higher speeds and forces can be accommodated. It has long been realised that the design and assembly of bolted joints must ensure that the clamp load achieve on tightening the fastener should be sufficient to withstand the effects of the static and dynamic loads which may arise during service. The designer of a bolted joint is concerned with ensuring the minimum clamping load or pre-load which is necessary to provide adequate in service durability. If the scatter in the tightening process is large then the range of clamp loads which has to be accommodated in the design will also be large, but must never be allowed to decrease below the minimum required for durability. Where higher clamp loads are likely because of large scatter, a large size bolt must be used, which usually means the joint must also be larger to accommodate the bigger bolt.

The demands for assurance of quality and reliability in engineering structures 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. This results in additional cost, either in component cost or in running costs. However, pressure on costs has also been increasing, and there is need to eliminate over design while, at the same time, maintaining reliability. This can often be done by detailed analysis of working loads, component stress distribution, and material capabilities.

It is well known that the best way of tightened a bolt is yield tightened [6]. In this tightened, a bolt can withstand maximum load. Further more, to save the weight, now a days, the bolt are being tightened to the plastic range. So it has become inevitable to know the performance of a bolt, when a bolt tightened up to or beyond yield and within the plastic range. By proper designing and proper selection of a bolt, it can be used most economically, by which weight of the motor transport, air craft and machinery's can be reduced. And at the same time power/weight ratio can be increased. With the increase of power/weight ratio, fuel consumption will decrease. Again, best possible design of a bolt will ensure safety of the equipment, and decrease the cost of production of the bolt. So it has become very much essential to carry out detail study of a bolt under different operating condition.

## 1.3 OBJECTIVES

The main objectives of this investigation are as follows:

a. To design and fabricate a test set-up for testing a bolt under different condition.

b. To design and fabricate a load cell.

c. To observe the influence of wrenching torque in developing pre load in the tightening bolt.

d. To observe the torque transmission, during tightening a bolt from bolt head to the bolt shank under different lubricating condition.

e. To examine the maximum load carrying capacity of a well tightened bolt under external tensile loading (i.e. bolt is tightened up to or beyond yield point).

f. To examine the effect of external loading on shank torque of a well tightened bolt (i.e. tightened up to or beyond yield point).

## 1.4 LAYOUT OF THE THESIS

For the convenience of presentation, the total contents of this research paper are divided into five chapters. Chapter-one contains introduction, justification and objectives of this investigation. Chapter- two contains literature review, behaviour of a material under different loading condition, failure criteria of a materials, influence of loading condition during tightening a bolt, influence of lubricating condition. Chapter- three contains design of a experimental set up and instrumentation, calibration of load cell, calibration of torque load cell, and shear strain gauge and experimental procedure. Chapter-four contains results and discussion of the whole experiments. Chapter-five contains conclusion and recommendation on the whole experiments.

# CHAPTER – 2

# LITERATURE REVIEW AND BACK GROUND THEORY

## 2.1 GENERAL

In this chapter the mechanism of tightening a jointing bolt and the stress distribution have been analyzed. It has been observed that during tightening a bolt, certain portion of the torque is transmitted to the bolt shank and considerable portion is used to overcome the friction under bolt head. This tightening torque will develop pre-load on the bolt and this pre-load will be dependent upon the lubricating condition of the mating surface. On the other hand when a jointing bolt is tightened, it will experience with tensile loading as well as torsion loading. Tensile stresses are roughly constant across the bolt section but the torsion stresses increases from zero at the central to a maximum at the outer surface. When the stresses get high enough, the bolt starts yielding. This yielding occurs first at the outer surface where the combination of tensile and shear stresses are greatest

## 2.2 PREVIOUS WORK

The designer of a bolted joint is concerned with ensuring the minimum clamping load or pre-load which is necessary to provide adequate in-service durability. If the scatter in the tightening process is large then the range of clamp loads, which has to be accommodated in the design will also be large, but must never be allowed to decrease below the minimum required value for durability. Where higher clamp loads are likely because of large scatter, a larger size bolt must be used, which usually means the joint must also be larger to accommodate the bigger bolt. Chapman *et al* [8] has carried out a number of investigations on jointing bolts and the stress distribution in the bolt under different condition. Measurement were made of static and dynamic loading of the joint. Monaghan [9], Hagiwara *et al.*[10] and Hariri[11] have carried out experimental investigations on jointing bolts. It was seen that during tightening a bolt, developed pre-load depends upon the frictional condition between the two mating surface. Monaghan and Duff [12] has shown that better consisting of clamping load can be achieved in the tightening bolt using yield tightening technique rather than more traditional torque control technique. Newnham and Curley [13] also carried out, a number of investigations and concluded that in combined loading shear stress has greater influence than normal stress in failure. Monaghan [9] has established that the lubrication condition dramatically effect the maximum clamp load achieved on the joint and torque distribution on the fastener.

## 2.3 BACKGROUND THEORY

### 2.3.1 Uni-Axial Tensile Behaviour of a Bolt

The most important mechanical property of a material is the tensile stress i.e. load per unit area. With the increase of tensile load bolt will be elongating and it will maintain a relationship. Initially the relation between stress and strain is essentially linear. This linear part of the curve is called the proportional limit. It is in this range that the linear theory of elasticity, using Hook's law is valid. Upon further increase of the load, the strain no longer increases linearly with stress, but the material still remains elastic; i.e. upon removal of the load the specimen returns to its original length. This condition will prevail until some point, called the elastic limit, or yield point. In most materials there is very little difference between the proportional limit and the elastic limit. For some materials the yield point is defined by using a fixed value of permanent strain. Beyond the elastic limit, permanent deformation, called plastic deformation, takes place. As the load is increased beyond the elastic limit, the strain increases at a greater rate. However, the specimen will not deform further unless the load is increased. This called work hardening, or strain hardening. The stress required for further plastic flow is called flow stress.

Finally a point is reached, where the load is a maximum. Beyond this point, called the point of maximum load, or point of instability, the specimen necks down. The stress at the maximum load point it called the tensile strength, or ultimate stress.

If at any point between the elastic limit and the maximum load point, the load is removed, unloading will take place along a line parallel to the elastic. Part of the strain is thus recovered and part remains permanently. The total strain can therefore be considered as being made up of two part $\varepsilon^e$ the elastic component, and $\varepsilon^p$, the plastic component:

$$\varepsilon = \varepsilon^e + \varepsilon^p \qquad (2.1)$$

### 2.3.2 Mechanics of Bolted Joint

When a connection is desired which can be dismantled without destructive methods and which is strong enough to resist both external tensile loads and shear loads, or a combination of both, then the simple bolted joint using hardened washers is a good solution. Bolted joints have clearance between the bolt and the hole. Manufacturing tolerances will allow certain bolts to carry an unpredictable shear of the load.

A portion of a bolted connection is shown in figure 2.1. The bolt has been pre-loaded to an initial tensile load $F_i$, then the external tensile load P was applied. The effect of the pre-load is to place the bolted member components in compression and the bolt under tension.

The spring constant, or stiffness constant, of an elastic member such as a bolt, is the force required to cause said displacement of the bolt. The deflection of a bolt in simple tension or compression was found to be

$$\delta = \frac{Fl}{AE} \tag{2.2}$$

Where, δ = deflection of the bolt , F= force applied , A= minimum cross section area of the bolt, E= modulus of elasticity of the bolt material. $l$ = length of the bolt

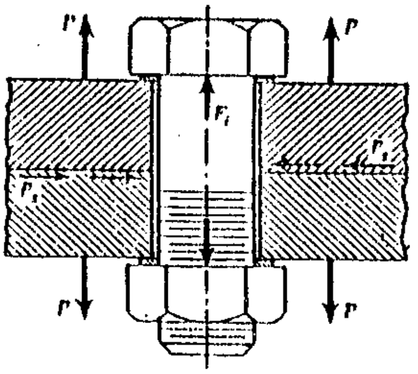

Fig – 2.1 Bolted connection

Therefore the stiffness constant is

$$K = \frac{F}{\delta} = \frac{AE}{l} \tag{2.3}$$

Where grip $l$ is the total thickness of the parts which have been fastened together.

It is visualized from figure 2.1 that when the external load P is applied to the pre-loaded assembly, there is a change in the deformation of the bolt and also in the deformation of the connected members. The bolt, initially in tension, gets longer. This increase in deformation of the bolt is

$$\Delta\delta_b = \frac{P_b}{k_b} \qquad (2.4)$$

Where $P_b$, is the portion of the external load P taken by the bolt and $K_b$ is the spring constant of the bolt. The connected members have initial compression due to the pre-load. When the external load P is applied, this compression will decrease. The decrease in deformation of the members is

$$\Delta\delta_m = \frac{P_m}{k_m} \qquad (2.5)$$

Where, $P_m$ is the Portion of the external load P taken by the member and $K_m$ is the spring constant of the member.

On the assumption that the members have not been separated, the increase in deformation of the bolt must equal the decrease in deformation of the members, and consequently.

$$\frac{P_b}{k_b} = \frac{P_m}{k_m} \qquad (2.6)$$

Since $P = P_b + P_m$, so,

$$P_b = \frac{k_b P}{k_b + k_m} \tag{2.7}$$

Therefore the resultant load on the bolt is

$$F_b = P_b + F_i = \frac{k_b P}{k_b + k_m} + F_i \tag{2.8}$$

In the same manner, the resultant load in the connected members is found to be

$$F_m = \frac{k_m P}{k_b + k_m} - F_i \tag{2.9}$$

Equations (2.8) and (2.9) hold only as long as some of the initial compression remains in the members. If the external force is large enough to remove this compression completely, the members will separate and the entire load will be carried by the bolt.

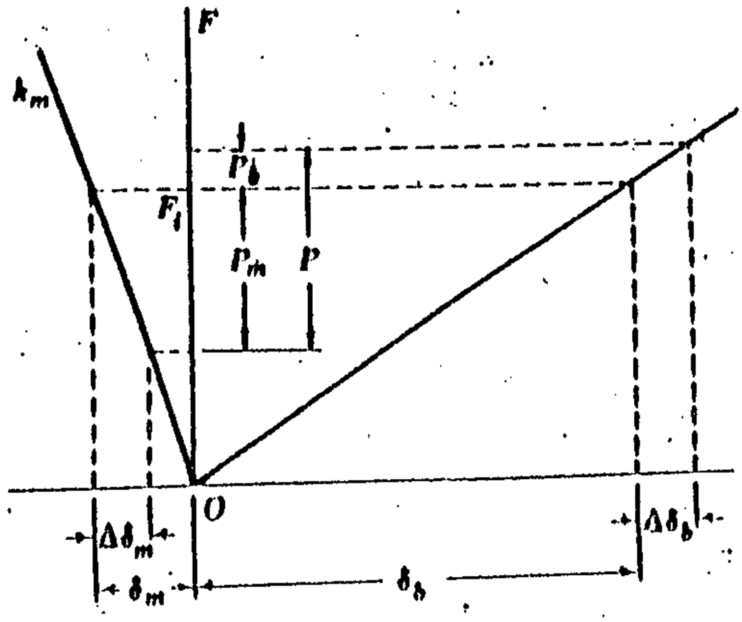


Fig – 2.2 Force deflection of a bolt under loading [15]

Figure-2.2 is a plot of the force deflection characteristics of a bolted joint. The line $K_m$ is the stiffness of the members; any force, such as the pre- load $F_i$, will cause a compressive deformation $\delta_m$ in the members. The same force will cause a tensile deformation $\delta_b$ in the bolt. When an external load is applied, $\delta_m$ is reduced by the amount $\Delta\delta_m$ and $\delta_b$ is increased by the same amount $\Delta\delta_b = \Delta\delta_m$. Thus the load on the bolt increases and the load in the members decreases.

### 2.3.3 Elastic Analysis of Jointing Bolt

Figure 2.3 represents the forces on a jointing bolts and its elongation with respect to the applied external tensile load. When a bolt is tightened, the load on the bolt increases, and the deformation of the bolt increases. Within the elastic range, Hook's law applies, and the force – deformation curve in figure-2.3 for the bolt is a straight line, represented by $OAM$ and the connected members deforms (in compression), and if they too are elastic, their force-deformation curve is straight line, represented by $CA$ in the same figure-2.3. The more rigid a member, the steeper is its F-δ curve, because it takes a larger force to produce a particular deformation. Usually the connected members are more rigid than the bolt, as shown in figure. 2.3 with α > Ø. The slope of $CDA$ is negative and represents a compressive deformation.

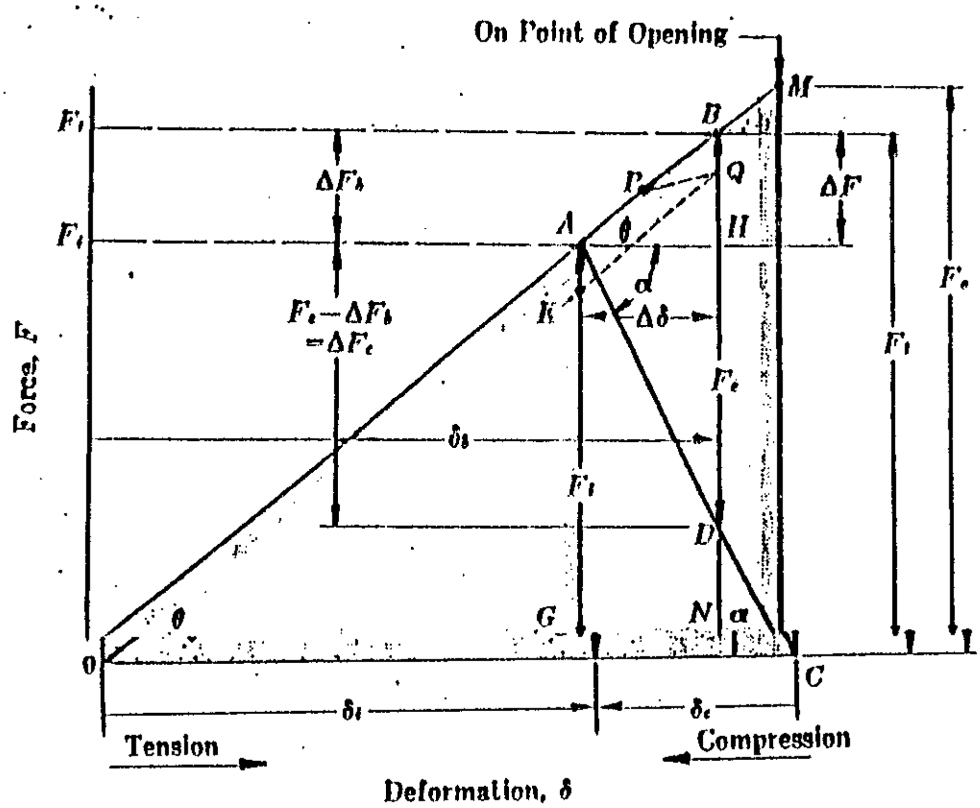


Fig 2.3 Forces on a jointing bolt [15]

When stop tightening at a point $A$, the load on the bolt as well as on the connected part is $F_i$, which is the initial tightening load. The initial elongation of the bolt is $\delta_i$ , and the corresponding compressive deformation of the connected parts is $\delta_c$. To get the external load that would cause a joint to open, assume that the bolts do not bend, which is equivalent to assuming that the head and flange do not bend, and let an external load $F_e$ be applied. The bolt elongates more $\Delta\delta$, say to $B$, (figure. 2.3) and the deformation of the connected parts *decreases* the same amount by $\Delta\delta$. The load on the bolt increases an amount $\Delta F_b$; the load on the connected parts decreases a greater amount $\Delta F_c$ if they are more rigid. For elastic deformations, the bolt elongation continues along the line $O$ $M$, and the compressive deformation decreases along $AC$. The joint will be on the point of opening when the deformation of the connected parts becomes zero, at $C$, because if the bolt is stretched any further, the connected parts can no longer expand to maintain the surfaces in contact. At the instant marked by $C$, the total elongation of the bolt is represented by the distance $OC$, and the total load on the bolt is $CM$ $= F_o$, the limiting load for opening of the joint, which is also the external load at this limiting condition.

2.3.4. **Tightening of a Bolt**

The tightening of a bolt is done by the application of a wrenching torque to its head as shown in figure 2.4. This applied wrenching torque causes the shank to rotate and the threads to tighten the bolt, it also overcomes the friction under the head of the bolt. Studies [14-16] have shown that a certain portion of the wrenching torque in only transmitted to the shank of the bolt and the screw thread. A considerable portion of wrenching torque is used to overcome the friction under bolt head.

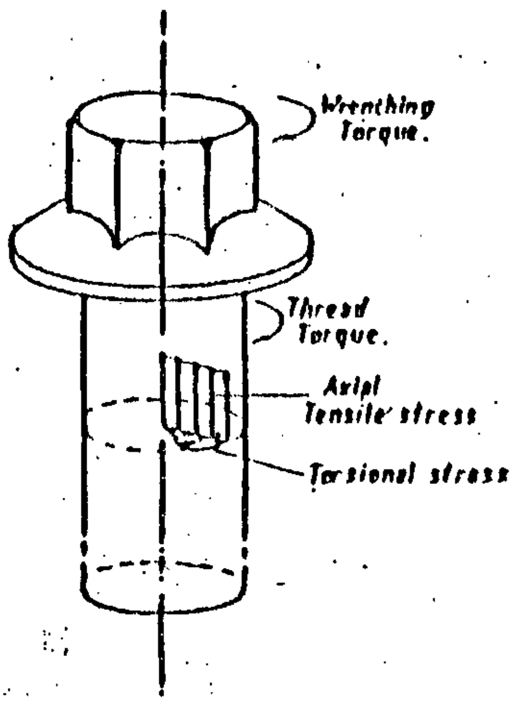


Figure-2.4 Tightening of a bolt [8]

It is found that the clamping force, normally called pre-load, developed by the bolt is usually proportional to the wrenching torque applied to the bolt head. The relationship between torque and clamp load is maintained over a range of torque and clamp loads. It produces combined stresses in the shank and free threads of the bolts i.e. (i) Tensile stress (Clamp load/area) (ii) Torsional or shear stresses (proportional to torque). Figure 2.5 is the representation of a tightened bolt, subjected to tensile stress as well as shear stress.

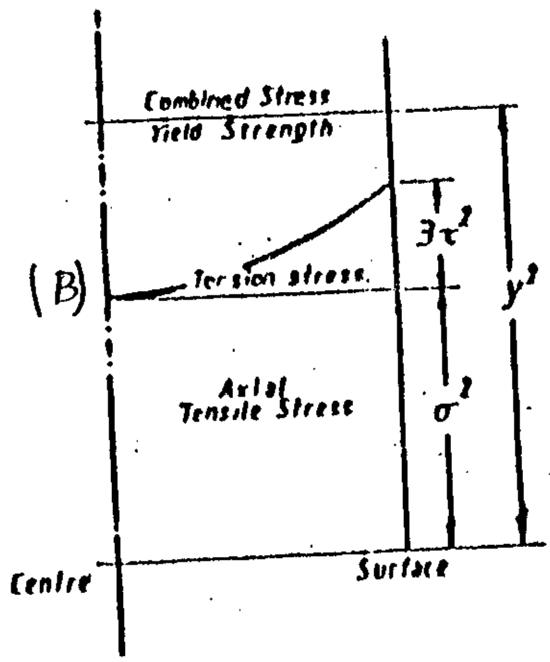


Fig – 2.5 Representation of the stress components in a bolt tightened below the yield point, with external torque still applied [21]

The tensile stresses are roughly constant across the bolt section. The torsion stresses increase from zero at the centreline to a maximum at the outer surface. It is well known that a tightened bolt subjected to combined tensile and torsion stresses, yields when the stresses get high enough. This yielding occurs first at the outer surface, where the combination of tensile and shear stresses is the greatest [8]. It occurs when the combination [21] :

$$\sigma^2 + 3\tau^2 = Y^2 \qquad (2.10)$$

(tensile stress)$^2$ + 3 (shear stress)$^2$ = (tensile yield stress of the material)$^2$

This is the von Misses yield criterion (proposed by Von Mises, 1913)[17], and has been supported by many investigations since that time.

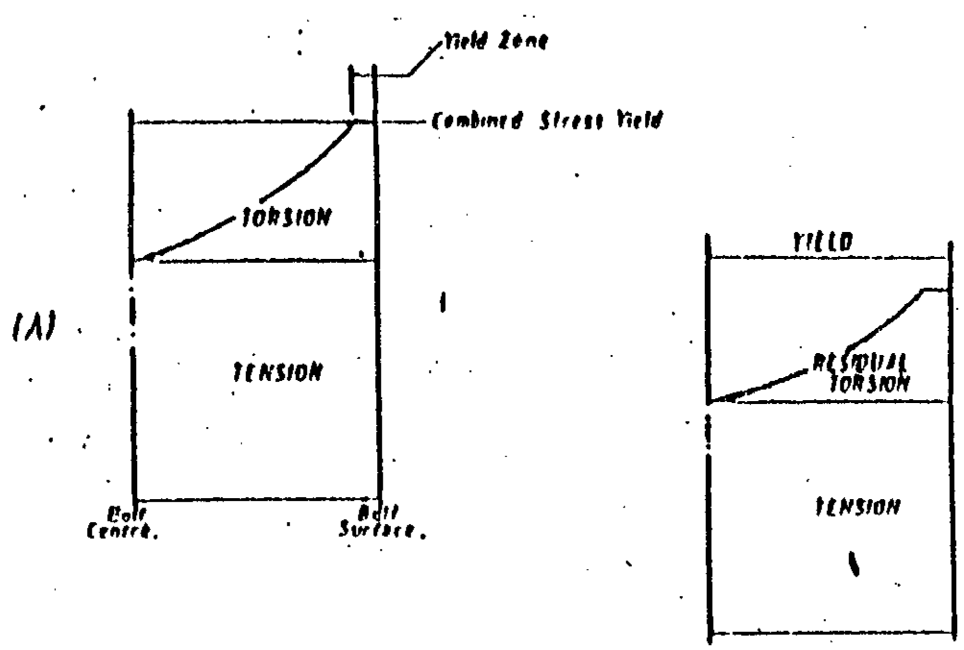


Fig-2.6 Representation of the effect of combined tension and torsional stresses in a bolt tightened to yield.

### 2.3.5 Bolt Under Pure Torsion

When a bolt of radius $a$ subjected to a twisting moment $T$, so long as the bolt is elastic, the shear stress acting over any cross section is proportional to the radial distance $r$ from the central axis [22]. The applied torque $T$ is the resultant moment of the stress distribution about this axis. If the angle of twist per unit length of the bar is denoted by $\theta$ , the elastic shear stress may be written as

$$\tau = Gr\theta = \frac{2Tr}{\pi a^4} \qquad (2.11)$$

Since the shear stress has its greatest value at $r = a$, the bolt begins to yield at this radius. When the torque is increased to Te, the corresponding twist being $\theta e$. Setting $\tau = k$ at $r = a$, It is found

$$T_e = \frac{1}{2}\pi k a^3 \qquad \theta_e = \frac{k}{Ga} \qquad (2.12)$$

where, Te is the critical torque applied for yielding.

If the torque is increased further, a plastic annulus forms near the boundary, leaving a central zone of elastic material with in a radius C (Fig. 2.7a). The stress distribution in the elastic region is linear, with the shear stress reaching the value k at $r = c$. For a non hardening material, the shear stress has the constant value k through out the plastic region, and stress distribution becomes

$$\tau = k\frac{r}{c} \qquad 0 \le r \le c \qquad (2.13)$$

$$\tau = k \qquad c \le r \le a \qquad (2.14)$$

Since the shear stress within the elastic zone is also equal to $Gr\theta$, so $\theta = k/Gc$ The twisting moment then becomes equal to

$$T = 2\pi \int_0^a \tau r^2 dr = \frac{2}{3}\pi k\left(a^3 - \frac{1}{4}c^3\right) = \frac{1}{3}T_e\left\{4 - \left(\frac{\theta e}{\theta}\right)^3\right\} \qquad (2.15)$$

As the elastic/plastic torsion continues, the torque rapidly approaches the fully plastic value $\frac{2}{3}\pi ka^3$, Since $\theta$ tends to infinity as $c$ tends to zero, an elastic core of material must exist for all finite values of the angle of twist. In the case of an annealed material,

there is no well-defined yield point, and the elastic plastic boundary is therefore absent. Since the engineering shear strain at

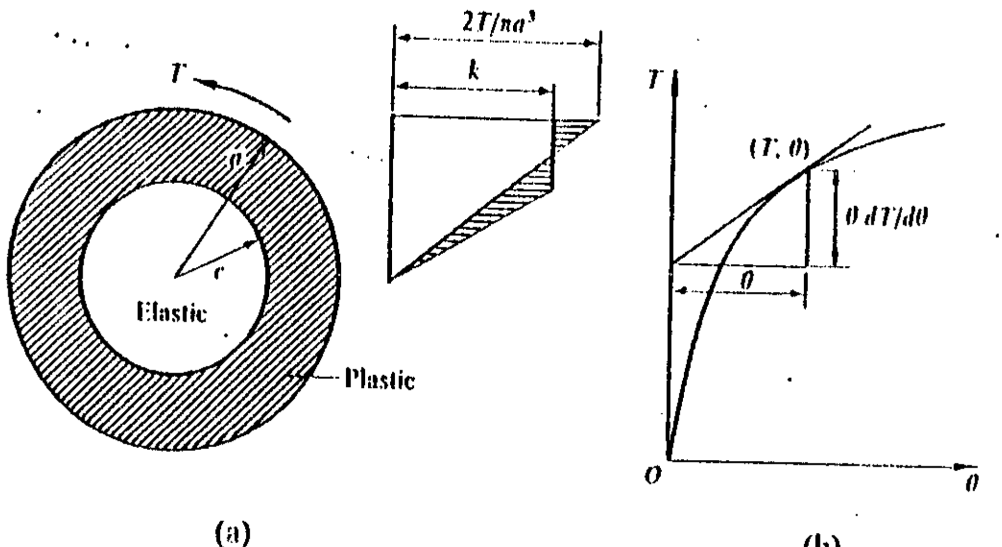


Figure 2.7 Torsion of a solid cylindrical bar (a) Plastic annulus and stress distribution for H=0 (b) Torque-angle of twist relationship .[ 22]

Any radius $r$ is $y=r\theta$, the torque may be expressed as

$$T = 2\pi \int_0^a \tau r^2 dr^2 = \frac{2\pi}{\theta^3} \int_0^a \tau \gamma^2 d\gamma \qquad (2.16)$$

When the shear stress-strain curve of the material is given, the torque can be calculated from above, using the known ($\tau$ , $\gamma$ ) relationship. Conversely, if the torque–twist relationship for a circular rod is experimentally determined, the shear stress-strain curve can be easily derived from it. The differentiation of the above equation with respect to $\theta$ gives

$$\frac{d}{d\theta^3}\left(T\theta^3\right) = 2\pi a^3 \theta^2 \tau_0 \qquad (2.17)$$

Where $\tau_o$ is the value of $\tau$ at $r = a$ Where the shear strain is $\gamma_o = a\ \theta$. The relationship between $\tau_o$ and $\gamma_o$ is therefore given by

$$\tau_0 = \frac{1}{2\pi a^3}\left(\theta\ \frac{dT}{d\theta^3} + 3T\right) \qquad \gamma_0 = a\theta \qquad (2.18)$$

The geometrical significance of the first term in the bracket is indicated in figure 2.7b. Since $dT/d\theta$ must be obtained graphically from the measured ($T$, $\theta$) curve, the computation based on (2.18) is not very accurate for the initial part of the curve. The accuracy may, however, be improved by rewriting the shear stress as

$$\tau_0 = \frac{1}{2\pi a^3}\left\{\theta^2 \frac{d}{d\theta}\left(\frac{T}{\theta}\right) + 4T\right\} \qquad (2.19)$$

The ratio $T/\theta$ is constant in the elastic range, and decreases slowly over the initial part of the plastic range. The contribution of the first term in the bracket is not therefore small over this part. For the latter part of the curve, where the strain hardening is small, the formula (2.18 ) should give more satisfactory results.

### 2.3.6 Maximum Wrenching Torque of a Bolt

As it is known that a high pre-load is very desirable in important bolted joint and consider means of assuring that the pre-load is actually developed when the parts are assembled [18-20]. If the over all length of the bolt can actually be measured with a micrometer when it is assembled, the bolt elongation due to the pre-load Fi can be computed using the formula $\delta = F_i l/AE$. Then simply tighten the nut until the bolt elongates through the distance $\delta$. This assures that the desired pre-load has been attained.

The elongation of a screw cannot usually be measured because the threaded end is often in a blind hole. It is also impractical in many cases to measure bolt elongation. In such cases the wrench torque required to develop the specified pre-load must be estimated. The torque wrench has a built-in dial which indicates the proper torque. With impact wrenching the air pressure is adjusted so that the wrench stalls when the proper torque is obtained, or in some wrenches, the air automatically shuts off at the desired torque.

The turn-of-the-nut method requires that one compute the fractional number of turns necessary to develop the required pre-load from the snug-tight condition. For heavy hexagon structural bolts the turn-of-the-nut specification states that the nut should be turned a minimum of $180^0$ from the snug-tight condition under optimum conditions. Note that this is also about the correct rotation for the wheel nuts of a passenger car.

### 2.3.7 Removal of Wrenching Torque of a Bolt

When the application of the wrenching torque ceases, the bolt shank tends to unwind. This effect is opposed by the under head torque, $T_h$, which changes its direction of action to prevent the rotation of the bolt shank. This unwinding of the bolt ceases when equilibrium is established between the under head and the thread torque, figure – 2.8. At this point the magnitude of each of the torque components will reduce compared to their values one tightening to yield. Consequently, because of the reduction in thread torques from $T_i$, to a lower value, $T_i$, the residual shear stress at the threaded section of the fastener will have reduced. A slight reduction in the clamp force will also have occurred due to unwinding, However, this is not significant and the axial stress, $\delta_a$, is assumed to maintain the value sustained on initial tightening to yield. Due to the reduction in the stress components, and particularly the shear stress, the overall stress state of the fastener will now be lower than that sustained during yield tightening. Consequently the fastener, and hence the joint, is now capable of carrying additional loading before the fastener is once again stressed to its yield point.

As in tightening, the magnitude of the torque required to loosen a fastener is dependent upon the frictional conditions on the mating surfaces. The torque required to loosen a fastener, $T_{w-off}$, is generally less than the tightening torque due to the assistance provided by the helix angle, figure – 2.8. As a result, the frictional conditions on the under head surface generally offers the greatest resistance to loosening, except it the case of an un-lubricated thread region.

It has been stated earlier that the maximum clamping force on a joint is achieved under conditions of low friction at the contacting surfaces. In service however, the clamping force can be reduced significantly by the effects of the applied loads on the joint. If this should occur then it is the frictional resistance on the contact surfaces which must oppose the self-loosening tendencies of the fastener.

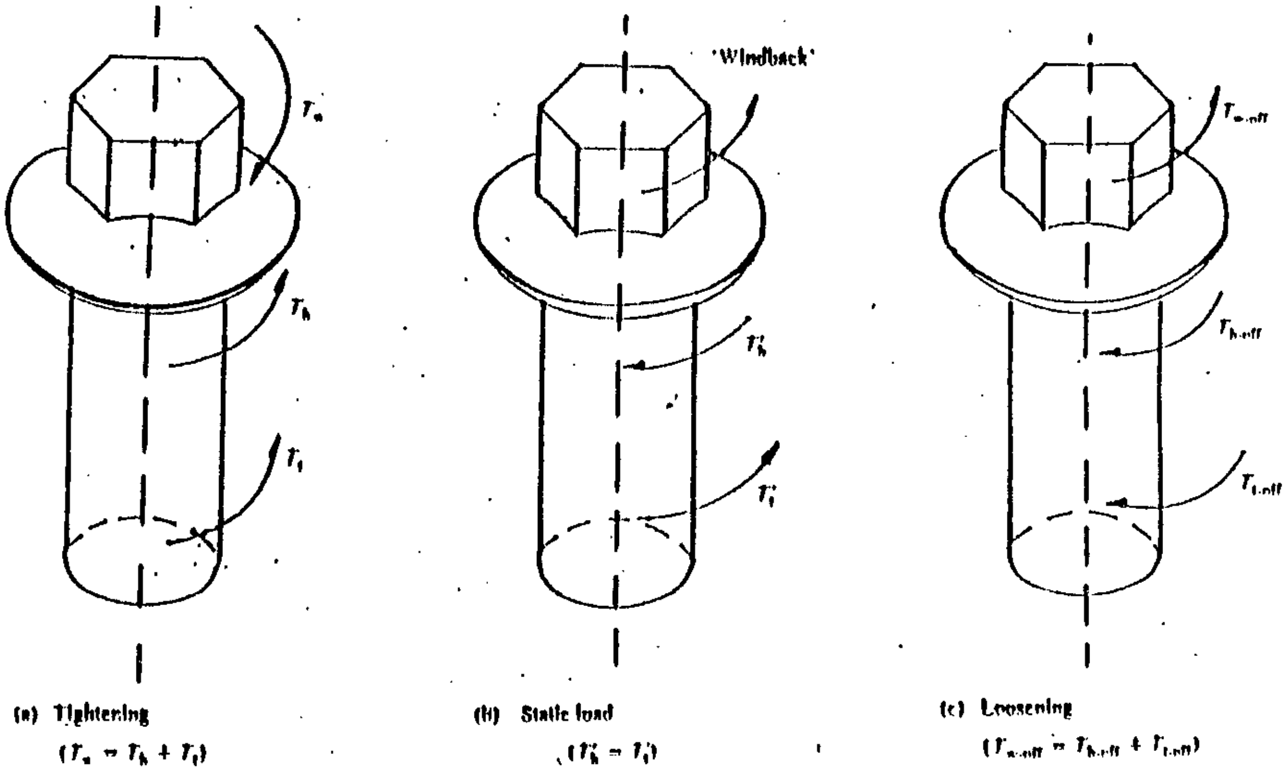


Fig.2.8 Torque conditions during (a) tightening (b) static loading (c) loosening [9]

# CHAPTER- 3
# EXPERIMENTAL SET UP AND PROCEDURE

## 3.1 INTRODUCTION

In the present study emphasis has been given to observe the performance characteristics of a well tightened bolt under external loading. The effect of other parameters on the mechanical properties of the bolt itself was also studied in the present investigation. In order to carry out detail experimental investigation of the jointing bolt, a test rig along with the auxiliary components/parts were designed. The details of the test rig are given below.

## 3.2 DESCRIPTION OF THE TEST RIG

Figure-3.1 represents the experimental set up, which has been designed for carrying out the experiment on a jointing bolt. It consists of a load cell, base plate, hexagonal nut, base stand, dial gauge, torque load cell and attachment for applying external load etc. Before use, it was fixed on a working table with a nut bolt arrangement. In this set up, a testing bolt was fixed against the load cell with the help of two base plates and one hexagonal nut and then placed on base stand. A magnetic dial gauge was placed under the base stand in such a way that its floating shaft touches the free end of the testing bolt as shown in figure-3.1. Torque load cell was set at the top of the testing bolt head to measure the applied wrenching torque at the bolt head. This torque load cell was connected with the power supply through adopter and multimeter, A torque wrench was used to apply torque to the torque load cell. Before starting the experiment the compressive load cell and torque load cell were calibrated. This experimental set up has the following provision.

a. Different levels of tightening torque can be applied at the bolt head.

b. Both the wrenching torque and shank torque can be measured.

c. Pre-load / axial load developed in the bolt due to tightening can be measured.

d. Due to tightening, elongation of the tightened bolt can be measured.

e. Different sizes of the bolt can be tested with this set up.

f. After tightening a bolt to the different level of torque, the test rig can be dismantled in such a way that it can be set into a universal testing machine and external tensile load can be applied to that part of the test set up.

## 3.3 DETAIL OF THE COMPONENTS

### 3.3.1 Load Cell

Figure-3.2 represents the schematic diagram of the load cell. It was designed to measure the pre-load developed in the bolt due to tightening. The load cell consists of metallic body and electrical resistance strain gauge. During designing the load cell, following points were considered, material of the load cell, geometry of the load cell, capacity of the load cell, and type of loading (tensile/compressive). The load cell designed in this experiment is a circular hollow steel pipe whose height was 100 mm, external diameter 85 mm and thickness 1 mm. As the expected maximum pre-load developed in the bolt before fracture was about 24 kN compressive load, hence a compressive type load cell was designed with a capacity of 30 kn. Here FLA-6-11 type of strain gauge was used in the load cell. As this circular shaped and compressive type load cell was not available in the concerned dept. so it became essential to fabricate this type of load cell.

### 3.3.2 Torque Load Cell

Figure-3.3 represents the schematic diagram of the torque load cell. It was used to measure the wrenching torque applied at the bolt head. It has socket end and plug end. Socket end was used with the bolt head and plug end with the torque wrench. It is a strain gauge based torque load cell. Its input excitation voltage was 6v DC supply. The -LT-10KA type torque load cell was used in this experiment, whose capacity was 98 N-m, sensitivity 2.0 mv/V, diameter 34 mm and total height 44 mm. It was manufactured by - TOKYO KANKYU JO CO(LTD),JAPAN. Though the torque load cell was factory calibrated, but as it was purchased long time back, so it became necessary to recalibrate it to know the exact response against the applied torque. So it was calibrated during the present investigation.

### 3.3.3 Base Plate

Figure-3.4 represents the schematic diagram of a base plate. It was used in this experiment to transmit the developed pre-load in the bolt to the load cell. The base plate was made of mild steel. The diameter of the base plate was 100 mm and thickness 12 mm. It has three holes along the central cord. One at the centre of the base plate and the other two at the opposite of the centre hole. The location and dimensions of the holes are shown in figure-3.4. The testing bolt can pass through the centre hole and a separate attachment can be fixed with the other two holes as shown in figure. It has a circular groove of diameter-80 mm on its surface. The depth and width of the groove are 2 mm & 3 mm respectively. This groove was used for the proper seating of the load cell. Total two base plates were used in this experiment, one at the top of load cell and another at the bottom of the load cell. Position of the base plates with respect to the load cell has been shown in figure-3.1.

### 3.3.4 Hexagonal Nut

Figure-3.5 represents the schematic diagram of a hexagonal nut. In this experiment it was used to tighten the testing bolt against the load cell with the help of two base plates. Hexagonal nut was so positioned that during dismantling it can hold the bolt in tightened condition against the load cell.

### 3.3.5 Base Stand

Figure-3.6 shows the base stand of the test rig. It has the upper plate supported by four legs on the lower plate. Detail dimension has been shown in Fig-3.6. Generally lower plate was fixed with the working table with nut bolt arrangement and upper plate hold the portion of the test rig. The portion of the test rig which contains testing bolt, load cell, two base plates and a hexagonal nut has shown in figure 3.14.

## 3.4 TESTING BOLT

Figure-3.7 represents the details of the bolt, which has been tested during the present investigation. The bolt material was mild steel, which was 134 mm long, nominal diameter 11.2 mm, length of the thread 50 mm. The type of the bolt thread was BSW and TPI was 12. In this experiment a number of bolts have been tested. All the bolts have been collected from the local market and these were thread rolling before heat treatment. Detail dimensions (in millimeter) of the bolt has been shown in figure -3.7.

## 3.5 SPECIAL ATTACHMENT FOR TENSILE TEST

Figure-3.8 represents the schematic diagram of a special attachment for tensile test of a virgin bolt. In this experiment a virgin bolt has been tested under pure tension. In order to perform the uni-axial tension test of the virgin bolt this special attachment was designed . Detail diagram and dimensions of the various parts of the attachment are shown in the Figure 3.8.

## 3.6 SPECIAL ATTACHMENT FOR APPLYING EXTERNAL LOAD

Figure-3.9 represents the schematic diagram of a special attachment for applying external load in this experiment. It consists of three bolts and one thick plate. Bolt `A' and bolt `B' were fixed with the base plate, placed top and bottom of the load cell and bolt `C' has been used for insertion into the universal machine. Two sets of attachment were used in the experiment. One at the top of the portion of the test rig and other at the bottom of the portion of the test rig.

## 3.7 AUXILIARY EQUIPMENT

### 3.7.1 Universal Testing Machine

A 100 ton capacity universal testing machine situated at the applied mechanics laboratory, was used to apply the external tensile load after tightening the bolt. Using this machine, the load cell was also calibrated. It is hydraulically operated and can be used for tensile loading as well as compressive loading.

### 3.7.2 Torque Testing Machine

The universal torque testing machine situated at mechanics laboratory was used to calibrate torque load cell and shear strain gauge fixed on the surface of the bolt shank. It is mechanically operated and can be used to measure angle of twist with respect to applied torque.

### 3.7.3 Torque Wrench

It is a mechanical torque wrench used with the torque load cell. In this experiment, it was used to apply torque to the torque load cell but no reading were recorded from the torque wrench.

### 3.7.4 Data Acquisition Equipment

**a. Strain Meter**

It was used to measure the strain of the strain gauge fixed on the load cell as well as shear strain gauge on the bolt surface during calibration and experiment. It has a graduation scale from 0 to 30,000 micro strain. As it is known that the load is proportional to the strain, so from the strain meter reading, applied load on the load cell was calculated.

**b. Multimeter**

It was used with the torque load cell to measure the applied torque. There were four electrical wire coming out from the torque load cell. Of which no. 1 & 2 were connected with the adapter for excitation voltage & no. 3 & 4 were connected with the multimeter. When torque applied to the torque load cell, the multimeter displayed the mv reading of the corresponding torque.

**c. Extensometer**

Figure-3.10 represents the schematic diagram of a extensometer. It was used to measure the elongation of the tightened bolt in this experiment. It consists of dial gauge. magnetic stand, extension bar and attachment for fixing in different position. It was placed at the free end of the bolt in such a way that floating shaft of the dial gauge placed below the bolt. With the elongation of the bolt, floating shaft moves down and gives deflection of the pointer of the dial gauge. With this extensometer, minimum 0.01 mm and maximum 25 mm elongation of the bolt can be measured.

## 3.8 EXPERIMENTAL PROCEDURE

### 3.8.1 Fixing of The Shear Strain Gauge on the Bolt Shank

In order to measure the percentage of applied wrenching torque transmitted to the shank of the bolt, a number of bolts were tested, fitted with the shear strain gauge on the shank of the bolt under different lubricating condition. In this experiment, CEA-06-187 UV-120 type of shear strain gauges was used. Before fixing the shear strain gauge on the surface of the bolt shank, it was cleaned from dirt, paint and rust with emery paper and then washed with a solvent to remove metal particle and grease. The clean surface of the bolt has been treated with a basic solution to give surface proper chemical affinity to the adhesive. The adhesive has been applied on the bolt surface as well as under surface of the strain gauge. The strain gauge has been placed on the bolt surface and pressed down gently with the thumb. The strain gauge has been bonded with the testing bolt and covered it with PVF sheet supplied with the adhesive. The strain gauge fixed with the testing bolt has not given the direct value of shank torque, so it was calibrated .

### 3.8.2 Calibration of the Axial Load Cell

Figure-3.11 represents the applied compressive load vs strain meter reading of the strain gauge fixed on the load cell. During calibration, universal testing machine and strain meter were used. Applied known compressive load was recorded from the universal testing machine and its corresponding strain of the load cell was recorded from the strain meter. From the figure it is seen that the applied load and strain meter reading (in micro strain) maintains a linear relationship. The equation of the load vs strain from the figure is as follows:

**Compressive load (N) = 46.9 (strain meter reading in micro strain).**

### 3.8.3 Calibration of the Torque Load Cell

Figure-3.12 represents the variation of the torque applied to the torque load cell with the out put (mv) reading obtained from the multimeter. The applied torque is a known torque, which was applied with the help of the torque testing machine available at the mechanics lab and its corresponding mv recorded by the multimeter. From the figure it is seen that applied wrenching torque and multimeter reading maintain a linear relationship. The equation of this relationship is as follows:

**Applied torque (N-m)=3.56* multimeter reading (mV).**

### 3.8.4 Calibration of the Shear Strain Gauge

Figure-3.13 represents the applied wrenching torque to the bolt head vs strain meter reading of the shear strain gauge fixed on the bolt surface. The known torque was applied with the help of torque testing machine available at the machines lab and its corresponding strain meter reading recorded in micro strain. From the figure it is seen that the applied torque and strain meter reading (in micro strain) maintain a linear relationship and is as follows:

**Applied torque (N-m) = 0.0484 * (Micro strain reading).**

### 3.8.5 Instrumental Set Up

Figure–3.1 represents the instrumental set up for the experiment. Before starting the experiment, the load cell was placed on the upper plate of the base stand with the help of base plate, the testing bolt was fastened with a hexagonal nut against the load cell. The torque load cell was connected with the multimeter and the excitation voltage supplied to the torque load cell was 6 volts. The strain gauge connection of the torque load cell was made with the strain meter as per figure- 3.1. The dial gauge was placed below the testing bolt, so that the floating shaft of the dial gauge touched the free end of the testing bolt. With the extension of the bolt the pointer of the dial gauge deflected and reading was recorded. The wrenching torque was applied with the help of torque wrench at the bolt head. The bolt was tightened and in the bolt axial pre-load as well as torsional load were developed. From which normal and shear stresses were calculated.

### 3.8.6 Uni Axial Tension Test of a Virgin Bolt

The testing bolt was fixed in the universal testing machine and the extensometer was set with the testing bolt. Before starting the experiment all the physical measurements of the bolt were taken. The gauge length and diameter of the bolt were 120 mm and 11.2 mm respectively.

### 3.8.7 Testing of a Bolt Without Lubrication

The testing bolt was fixed in the experimental setup/test rig, with the help of base plate and hexagonal nut against the load cell .The extensometer was placed under the upper plate of the base stand, so that its floating shaft touched the free end of the testing bolt. All the necessary electrical connections were given. Then wrenching torque was applied at the bolt head through torque load cell with the help of torque wrench. Pre-load developed in the load cell was measured by the strain meter. The wrenching torque was applied by the torque wrench through torque load cell. Torque load cell reading was recorded by the multimeter and the elongation of the bolt was measured by the extensometer. In this experiment all the readings were taken at a regular interval of time.

### 3.8.8 Testing of Bolt Under Lubricating Condition

In this experiment two different types of lubricants were used which were, Viscosity grade-15 named as VG-15 and Viscosity grade-46 named asVG-46.

### (a) Testing of Bolt With Lubrication by VG-15

Test procedure for the testing of a bolt with lubrication by VG-15 was same as the testing of bolt without lubrication. But in this case VG-15 lubricant was used in between the bolt under head and base plate surface. Here applied wrenching torque, pre-load developed in the bolt and bolt elongation were recorded by the, multimeter, strain meter and extensometer respectively.

### (b) Testing of Bolt With Lubrication by VG-46

Test procedure for the testing of a bolt with lubrication by VG-46 was same as the testing of bolt without lubrication. But in this case VG-46 lubricant was used in between the bolt under head and base plate surface. Here applied wrenching torque, pre-load developed in the bolt and bolt elongation were recorded by the multimeter, strain meter and extensometer respectively.

## 3.8.9 Testing of the Tightened Bolt Followed by External Tensile Load

Figure-3.14 represents the details of the test set up which was used to apply the external tensile load (using the universal testing machine) after tightening the bolt. In this arrangement the portion of the test setup was dismantled from the test rig and fixed with a special attachment (figure-3.8),and then put into universal testing machine for applying external tensile load. The external tensile load was recorded from the universal testing machine and elongation of the bolt was measured with the help of extensometer directly.

# CHAPTER- 4

# RESULTS AND DISCUSSION

## 4.1 INTRODUCTION

In the present study detail experimental investigations on the behavior of well tightened jointing bolts under different lubricating conditions as well as under external tensile loading were done. Here bolts were tightened up to combined yield and with in the plastic range. Further more, effect of external tensile load on the shank torque was also examined. In order to do this investigation a number of preliminary tests were also carried out. The following sections contain the detail analysis of the results of the present investigation.

## 4.2 Uni Axial Tension Test of the Virgin Bolt

Figure-4.1 represents the uni axial tensile load vs elongation curve of the virgin bolt (i.e bolt is neither tested nor used before) . From figure it is observed that the average yield load measured at zero percent (0%) offset (i.e. at proportional limit) was found 23.52 kN and its corresponding yield stress calculated was 327 MPa based on the cross sectional area of minor diameter of the bolt. From the figure it is also seen that the average ultimate load of the bolt was 34 kN and the percentage of elongation found to be 17.5% based on the total length of the bolt. Total elongation of the bolt was measured 21 mm.

## 4.3 Effect of Lubrication on Pre-Load Developed

Figure-4.2 represents the applied wrenching torque at the bolt head vs pre-load developed in the bolt under different lubricating condition. It is also seen from the figure that applied wrenching torque and developed pre-load ( i.e tensile load developed on the bolt due to applied wrenching torque called pre-load) maintains a linear relationship up to certain range of applied torque. After that linear relationship is not maintained because, at that stage the bolt starts yielding due to the combined loading. From the figure it is also seen that for the different lubricating condition, clamping load thus developed also varies. But in all the cases, the wrenching torque and clamping load maintained a linear relationship. It has been seen from the figure that for 88(N-m) of applied wrenching torque, pre-load developed for without lubrication, with lubrication by VG-15 and lubrication by VG-46 are 17 kN, 20.5 kN and 24.7 kN respectively. It has also been calculated that when lubricant VG-15 was used , pre-load increased by 22% and when lubricant VG-46 was used pre-load increase by 47% compared to those without lubricant. Hence it can be concluded that pre-load development will be higher when higher viscosity grade lubricant is used. From figure.-4.2, it can be concluded that the friction between the bolt under head and mating surface, has great influence in developing pre-load due to applied wrenching torque. When friction condition deteriorate, a larger wrenching torque is required to achieve a given level of pre-load. So with the improvement of the friction co-efficient between the surface, pre-load developed in the bolt due to the applied wrenching torque can be increased.

## 4.4 Torque Transmission From the Bolt Head to the Shank

Figures-4.3 to 4.8 represent the relationship between the wrenching torque applied at the bolt head to the torque transmission at the shank of the bolt under different lubrication condition. All the figures show that wrenching torque and shank torque maintains a linear relationship. This represents the torque transmission from the bolt head to the shank of the bolt. Whatever torque applied at the bolt head, same can not be transmitted to the shank of bolt. Some portion of torque has been utilized to over come the friction under the bolt head. From the figures it is also seen that, with the change of lubricant quality torque transmission rate is also changed. Higher the viscosity grade lubricant, higher the torque transmission rate. It is seen from figure-4.3 that, without lubrication only

52% of the applied torque can be transmitted to the shank, with lubrication by VG-15, 62% ~ 63% of the applied torque can be transmitted to the shank of the bolt (Figure-4.5 & 4.6) and with better lubricant (i.e. higher viscosity grade), like VG-46 the applied wrenching torque transmission to the shank of the bolt can be increased up to 67% ~ 68% (Figure–4.7 & 4.8). Hence it can be concluded that the applied wrenching torque transmission to the shank of the bolt depends upon the frictional condition between the bolt under head and mating surface. By proper lubrication this transmission rate can be increased by an appreciable amount.

## 4.5 Tightening of the Bolt Into the Plastic Region

Figure-4.9 represents the pre-load developed due to torque applied at the bolt head vs elongation of the bolt. The solid line shown in this figure i.e. curve-1 represents the uni-axial tensile load vs elongation of the virgin bolt. Here the bolts were tightened for three different lubricating condition in the plastic region of the bolt material and up to the fracture. The curve-2 of this figure, represents the pre-load developed due to torque applied at the bolt head vs elongation of the bolt without lubrication, the yield points and ultimate load of this curve are 15.68 kN and 21.6 kN respectively. The curve-3 represents the pre-load developed vs elongation of the bolt with lubrication by VG-15, the yield points and ultimate load of this curve are 17.64 kN and 23.03 kN respectively .The Curve-4 represents the pre-load developed vs elongation of the bolt with lubrication by VG-46, the yield points and ultimate load are 19 kN and 24.73 kN respectively. It is also observed from the figure that yield point of curve-2 is lower than that of curve-3 and much lower than that of curve-4. Thus from the above discussion it can be concluded that higher the viscosity grade lubricant, higher the pre load developed. The total elongation of the bolt due to tightening without lubricant 16 mm, with lubricant by VG-15; 18 mm , with lubricant VG-46; 19 mm and for virgin bolt 21 mm respectively. However the total elongation of the bolt due to tightening in the plastic region decreases than that of virgin bolt.

## 4.6 Effect of External Load on Tightened Bolt

Figure 4.10- 4.15 represents the effects of external tensile load on different level of tightened bolt up to yield point and within the plastic range under different lubricating condition. In figure 4.10 the bolt was tightened up to point "A"(i.e. when yielding just to start) and after that the bolt was subjected to simply external tensile load. From the figure it is observed that, when further tensile load was applied the bolt material behaves nearly elastically and this curve nearly matches with the uni-axial tensile load vs elongation curve of the virgin material. Figure 4.11 shows the behavior of the bolt when it was initially tightened up to point "B", in this case the plastic deformation of the tightened bolt was about 3.5 mm ( i.e. elongation of the bolt from "A" to "B") After that in a similar way external tensile load was applied and the bolt behaves nearly elastically like that of the virgin bolt. From the figure it is seen that after reaching point 'C' again the curve nearly collapses with the uni-axial tensile load vs elongation curve of the virgin bolt. Figure-4.12 shows the behavior of a bolt when it was initially tightened up to point "D" without lubrication. In this case plastic deformation of the bolt was 7.5 mm (i.e. elongation of bolt from A to D), after that in a similar way external tensile load was applied and the bolt behaves nearly elastically up to point 'E' like that of the virgin bolt. From figure it is seen that after reaching point 'E' again the curve matches very closely with the uni-axial tensile load vs elongation curve of the virgin bolt. Figure 4.13 represents the comparison of figure 4.11 and 4.12 . Figure 4.14 and 4.15 show similar results, like those of figures. 4.10 and 4.11, but in these case lubricant were used. In figure 4.14 lubricating oil used was VG -15 and in figure 4.15 lubricant oil used was VG-46. In both the figures similar observations were found in maximum load carrying capacity. Thus from the above discussion, it is observed that whether the bolt is

tightened up to the combined yield point or beyond that (i.e. with in plastic region) the maximum tensile load carrying capacity of the bolt remain unaffected. It was further observed that even after tightening the bolt with the plastic range the bolt behaves elastically during the application of external tensile load. Moreover it was also found that the lubricating condition does not have any effect on the maximum load carrying capacity of the bolt.

## 4.7 Effects of External Tensile Load on the Shank Torque

Figure 4.16 - 4.18 represent the effects of external tensile load on the shank torque of the bolts. In this arrangement the shear strain gauge was fixed on the bolt shank and proper electrical connection was given with the strain meter. When external tensile load was applied with the help of universal testing machine , the strain meter reading also starts changing. Recording of the external tensile loading and strain meter reading were done simultaneously. The same experiment were done under different lubricating condition, which has been represented by the figure 4.16- 4.18. Figure 4.16 displays that the effects of external load on the shank torque with out lubricant condition. Here the bolt was initially tightened with in plastic region whose corresponding shank torque in the bolt was 30 N-m and pre-load developed 17.64 kN. It is observed from the figure that with the application of external load the shank torque starts decreasing. When the applied external load was increased up to 30.38 kN the shank torque reduces to 14 N-m at 4 mm elongation of the bolt i.e. the shank torque reduces about 53%. Furthermore it was observed from figure 4.16 that when the external tensile load reaches nearly equal to its ultimate load, the reduction in the shank torque ceases and the available shank torque remains as the residual torque of the bolt. From the figure it is seen that when the external tensile load reaches nearly equal to ultimate load, then the shank torque reduces to 10.75 N-m at 12 mm elongation of the bolt i.e. the maximum reduction in the shank torque was 65%. Similarly from figure 4.17 the bolt was initially tightened using lubricant VG-15 in plastic region whose

corresponding shank torque in bolt was 35 N-m and the pre-load developed 21 kN. Then the applied external load was increased up to 32 kN, the shank torque reduces to 16 N-m at 4.5 mm elongation of the bolt i.e. the shank torque reduces to 55%. Furthermore it was observe from the figure that when external tensile load reaches nearly equal to its ultimate load, the reduction of the shank torque ceases and the available torque in the shank remains as the residual torque of the bolt. From the figure it is observed that when the external tensile load reaches nearly equal to the ultimate load, the residual shank torque reduces to 11.5 N-m. at 10 mm elongation of the bolt i.e. shank torque reduces to 67%. From Figure 4.18 it is observe that the initially the bolt was tightened with in plastic region using lubricant VG-46, whose corresponding shank torque in the bolt was 38 N-m and corresponding pre- load developed 23.5 kN when the applied external load was increase up to 32.5 kN the shank torque reduces to 16 N-m at 6 mm elongation of the bolt i.e. shank torque reduces 58%. Furthermore it was observe from the figure that when external tensile load reaches nearly equal to its ultimate load the reduction in the shank torque ceases and the available torque in the shank of the bolt remains as the residual torque of the bolt. From the figure it is also seen that when the external tensile load reaches nearly equal to ultimate load ,residual shank torque reduces to 11.5 N-m at 10 mm elongation of the bolt i.e. shank torque reduces about 70%. Thus from the above discussion it can be concluded that the total combined stress carrying capacity of a material remains same and hence after reaching combined yield the point, increase in tensile load causes reduction in the shank torque.

# CHAPTER-5
# CONCLUSION AND RECOMMENDATION

## 5.1 CONCLUSION

### 5.1.1 GENERAL

The behavior of the jointing bolts under different lubricating conditions with in plastic region has been studied in the present investigation .It has been observed that the lubricating condition and the method of tightening the bolt severely influence the behavior in the jointing bolt. From the detail investigation, conclusion has been drawn which has been summarized in the following section.

### 5.1.2 Conclusion

The conclusions that have been arrived at the present study are presented below:

i). Friction condition between the bolt under head and mating surface has a great influence in torque transmission.

ii). By using higher viscosity grade lubricant, higher rate of torque transmission is possible and hence higher pre-load will be developed in the bolt.

iii). The relationship between the applied wrenching torque and the developed pre-load for various lubricating condition has been obtained, as shown in figure - 4.2.

iv). It has been observed that pre-load developed for the lubricantVG-15 was 22% higher than that of the tightened bolt without lubrication. Similarly when higher viscosity grade lubricant VG-46 was used, pre-load developed was 47% higher than that of the bolt with out lubricating condition.

v). Torque transmission to the shank of the bolt for without lubrication condition was 52%, with lubrication by VG-15 and VG-46 were 62% and 68% respectively. Thus it can be concluded that the higher the viscosity grade lubricant used, the higher the rate of torque transmission to the shank of the bolt and hence higher the pre-load developed.

vi). It was observed that when the bolt was tightened up to the combined yield point or beyond that (i.e. with in plastic region), the maximum tensile load carrying capacity of the bolt remains unaffected.

vii). It was observed that even after tightening the bolt within plastic region the bolt behaves elastically during the application of external tensile load.

viii). It was further observed that the lubrication condition does not have any affects on the maximum load carrying capacity of the bolt.

ix). The application of working load to a bolted joint results in additional tensile stress thus the torsional stress remaining after tightening has no effect on the bolt strength or elasticity under these circumstances.

x). When the bolt is tightened up to yield point and subsequently external load is applied, its maximum tensile load carrying capacity remains the same like that of a virgin bolt.

## 5.2 RECOMMENDATION

### 5.2.1 General

In the present investigation, maximum emphasis has been given on the frictional condition between the bolt under head and mating surface. This frictional condition varies from test piece to test piece, but in this experiment frictional condition has considered remain fixed. Hence to obtained a best possible result from the experiment due consideration will be given on the recommended points.

### 5.2.2 Recommendation

To obtained a best possible result from the experiment special consideration will be given on the following points:

i). It has been considered in this experiment that all the members of the test rig are perfectly rigid, but ideally it is not so possible, hence due consideration will be given to obtain highest possible rigidity of the test rig.

ii). Although it has been considered that the testing bolt is completely homogeneous throughout its length and cross section but ideally it is not so possible. Hence non-homogeneous material may have some effects on its results. This point has to be considered to obtain a best possible result.

iii). Axial tensile load and torsional load have some effects on the thread of the bolt which will be considered to obtain a best possible result.

iv). Sudden change of dimension and type of TPI of the bolt have got some stress concentration effects, to obtain best possible result due consideration will be given on the stress concentration factor.

v). When external tensile load was applied to the tightened bolt, a special arrangement has to be made to measure the elongation of the bolt accurately to obtain best possible result, which was not possible in this experiment.

vi). The transmission of the wrenching torque to the shank of the bolt was calculated by the calibration method, it would have been much better if the shank torque was obtained by fixing the shear strain gauge on the bolt shank every time. In that case, although the experiment becomes very expensive but very accurate result can be obtained.

vii). Lubricating condition has a great influence on torque transmission, hence bolt under head and mating surface may have with solid lubricant, so that maximum torque can be transmitted.

viii). Yield control technique is the best method of tightening bolt. So bolt has to be designed, based on yield control tightening method.

ix). Torsion doesn't have any effect on maximum load carrying capacity, so bolt has to be designed, based on uni-axial tensile behaviour of the bolt material.

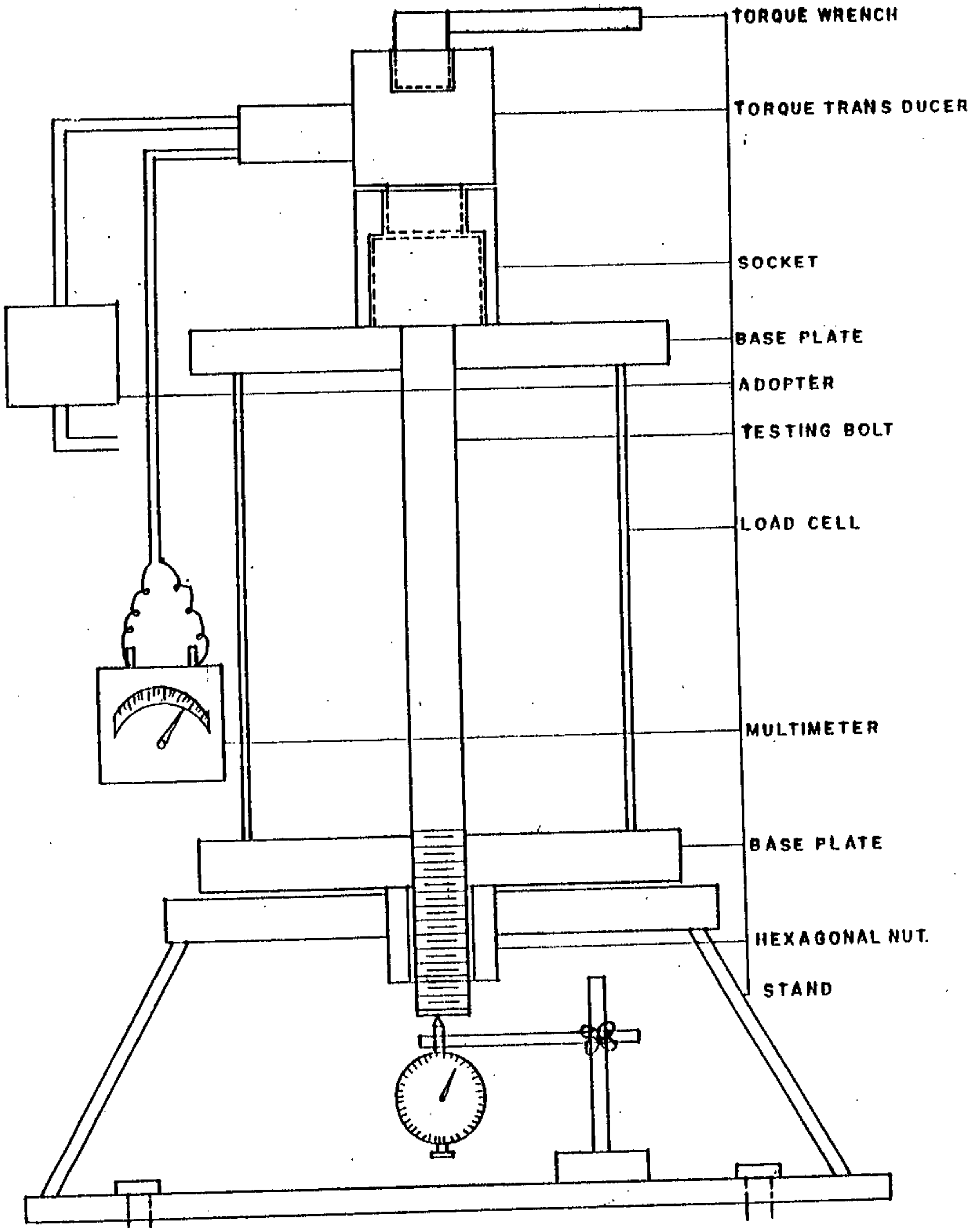


**Figure: 3.1 Schematic of the test rig .**

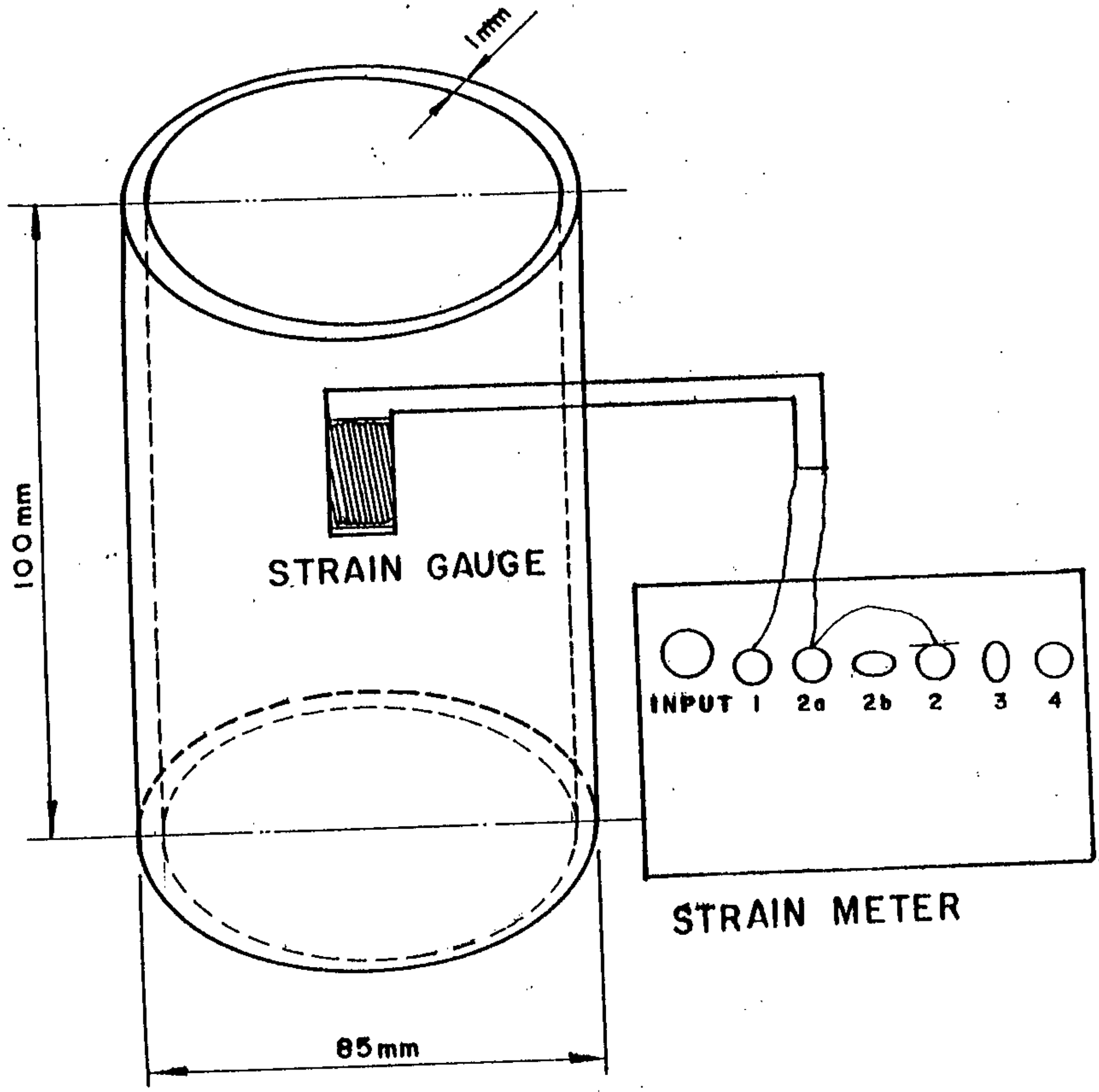


Figure: 3.2 Schematic of the load cell .

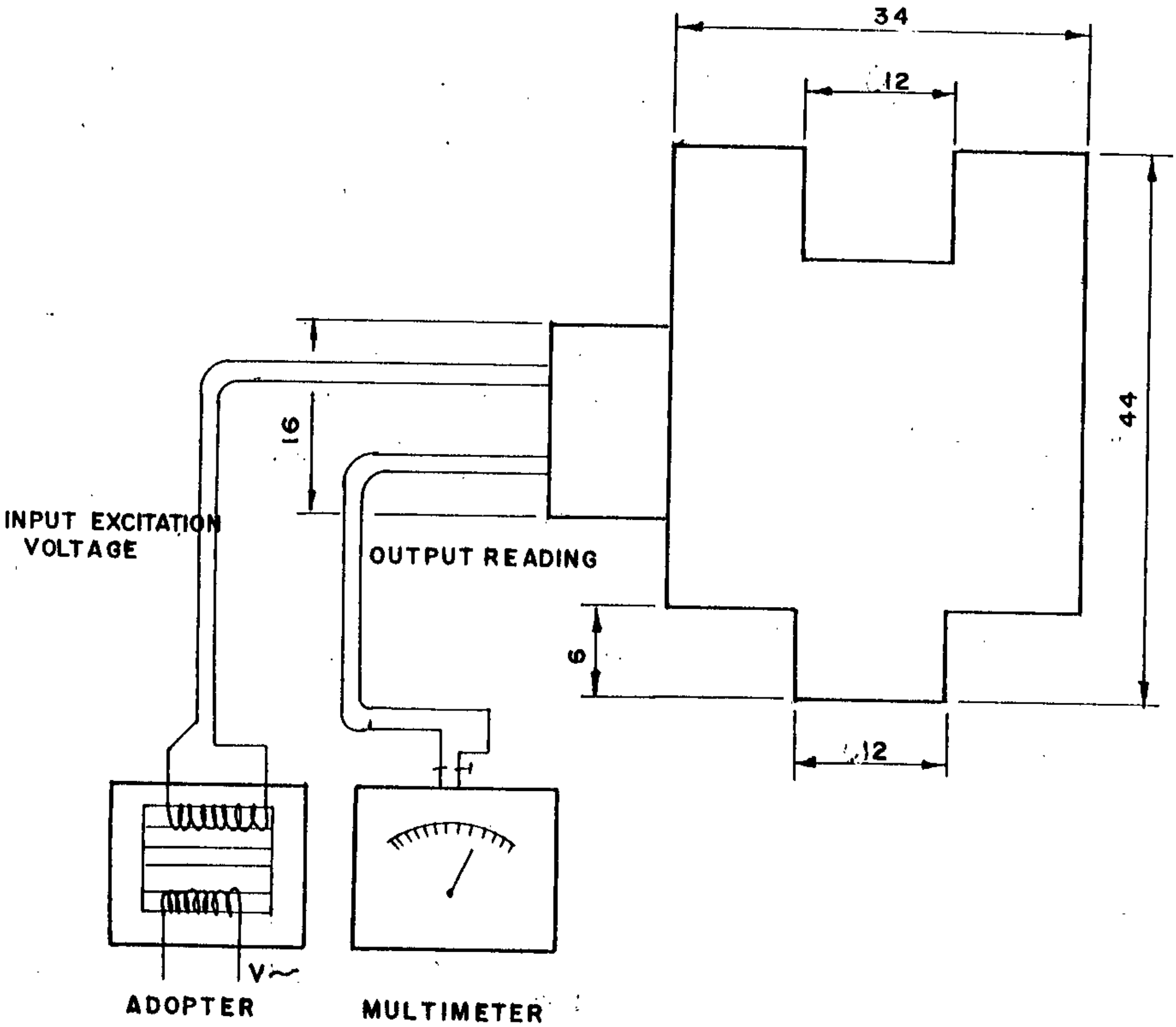


Figure: 3.3 Schematic of the torque load cell.

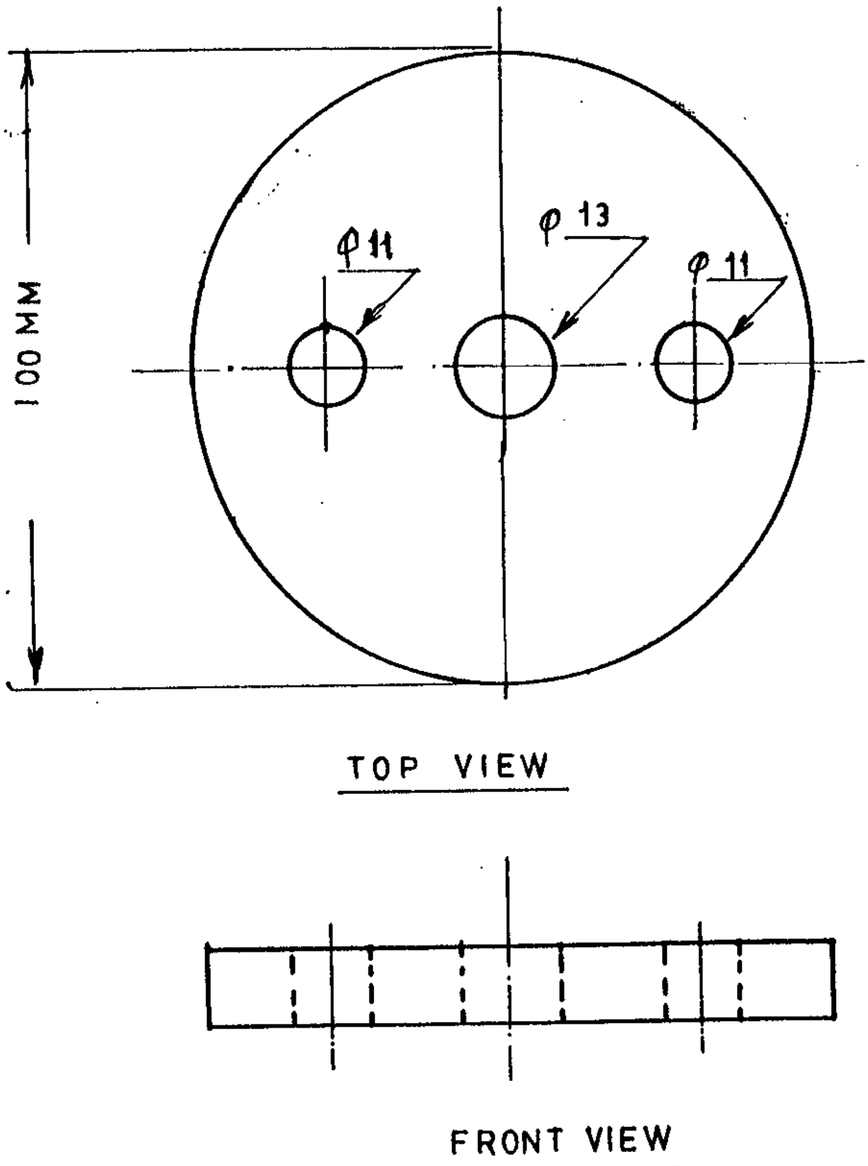


**Figure: 3.4 Schematic of a base plate.**

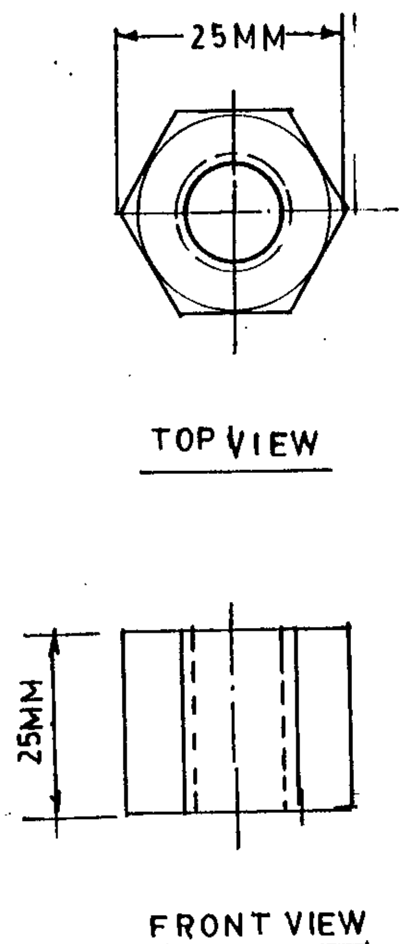


**Figure: 3.5 Schematic of a hexagonal nut .**

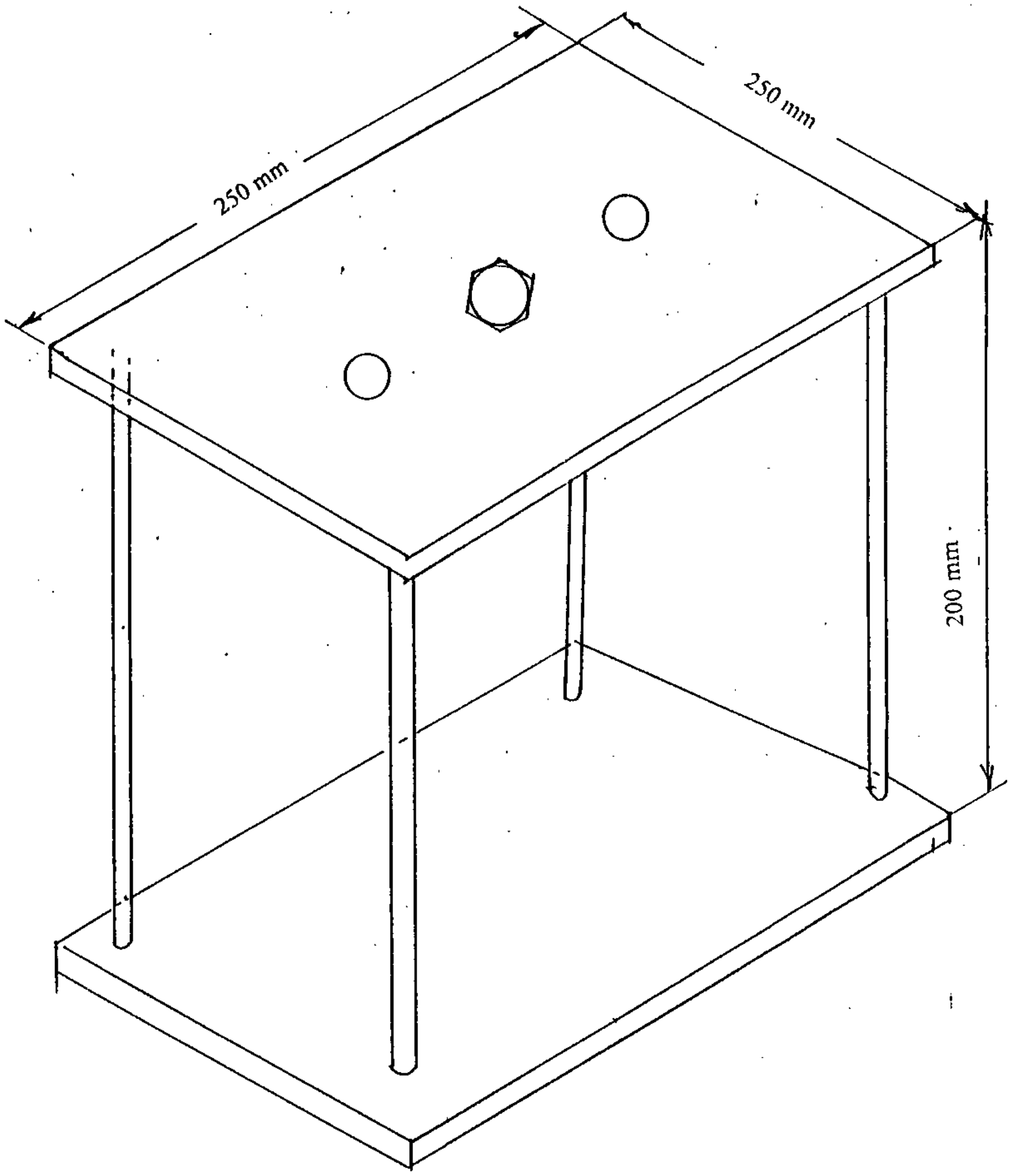


**Figure: 3.6 Schematic of the base stand .**

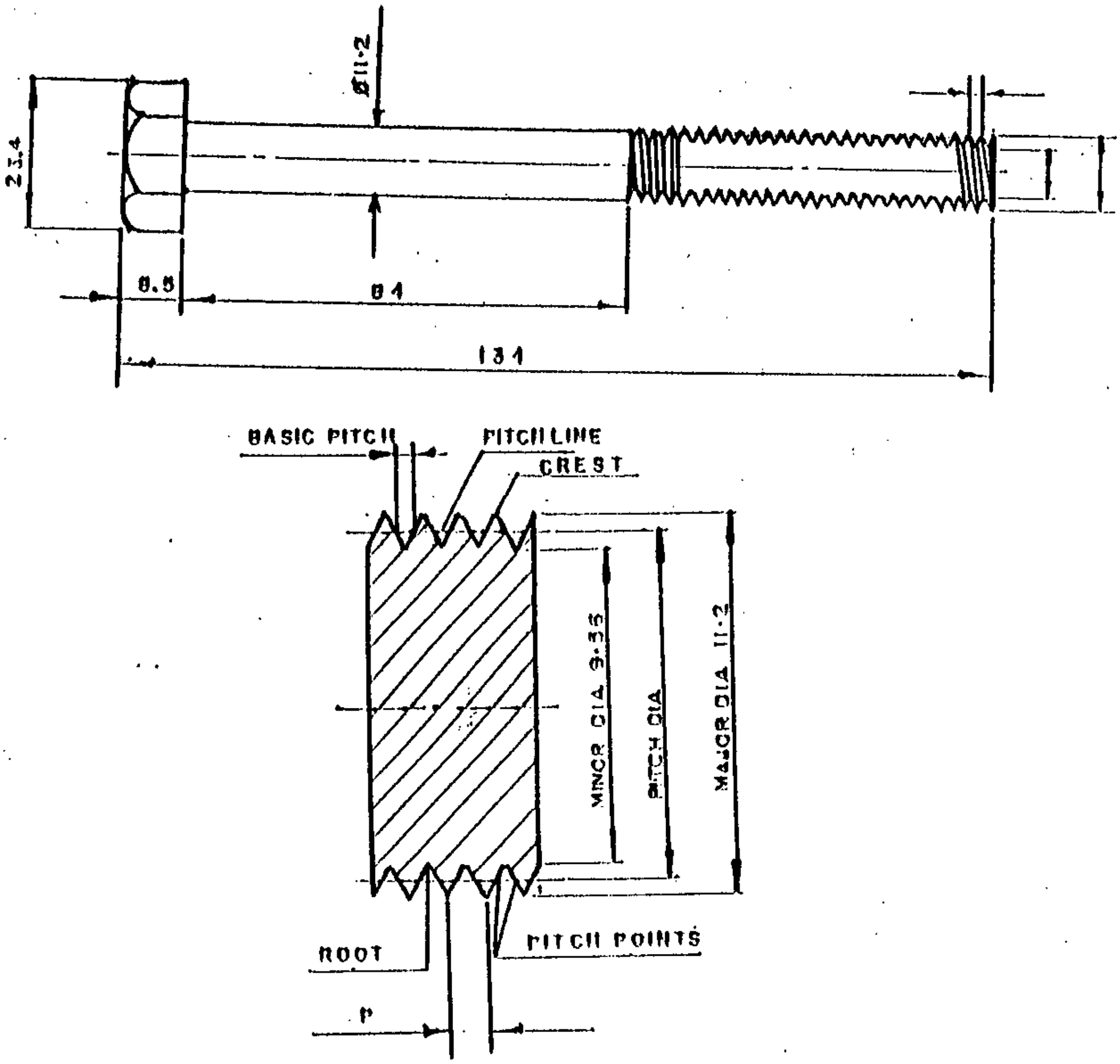


**Figure: 3.7 Schematic of the testing bolt.**

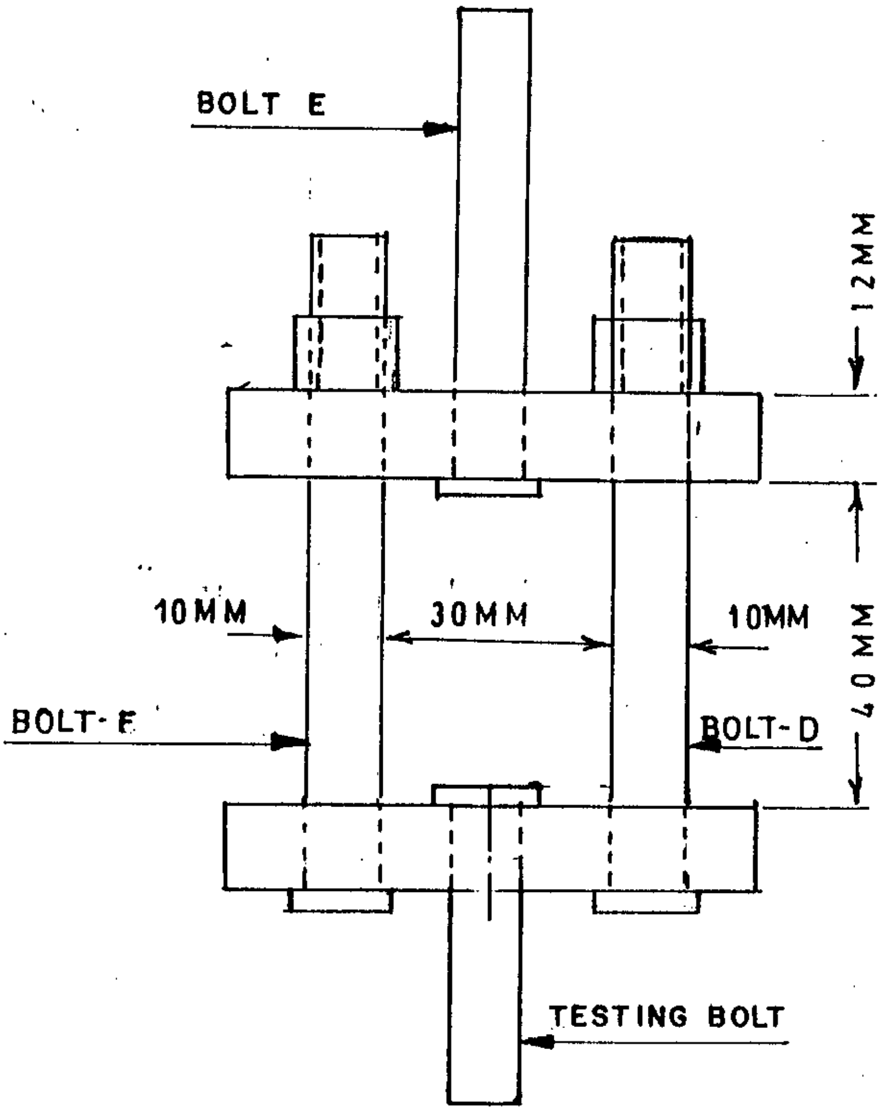


**Figure: 3.8** **The special attachment used for uni-axial tension test of the virgin bolt.**

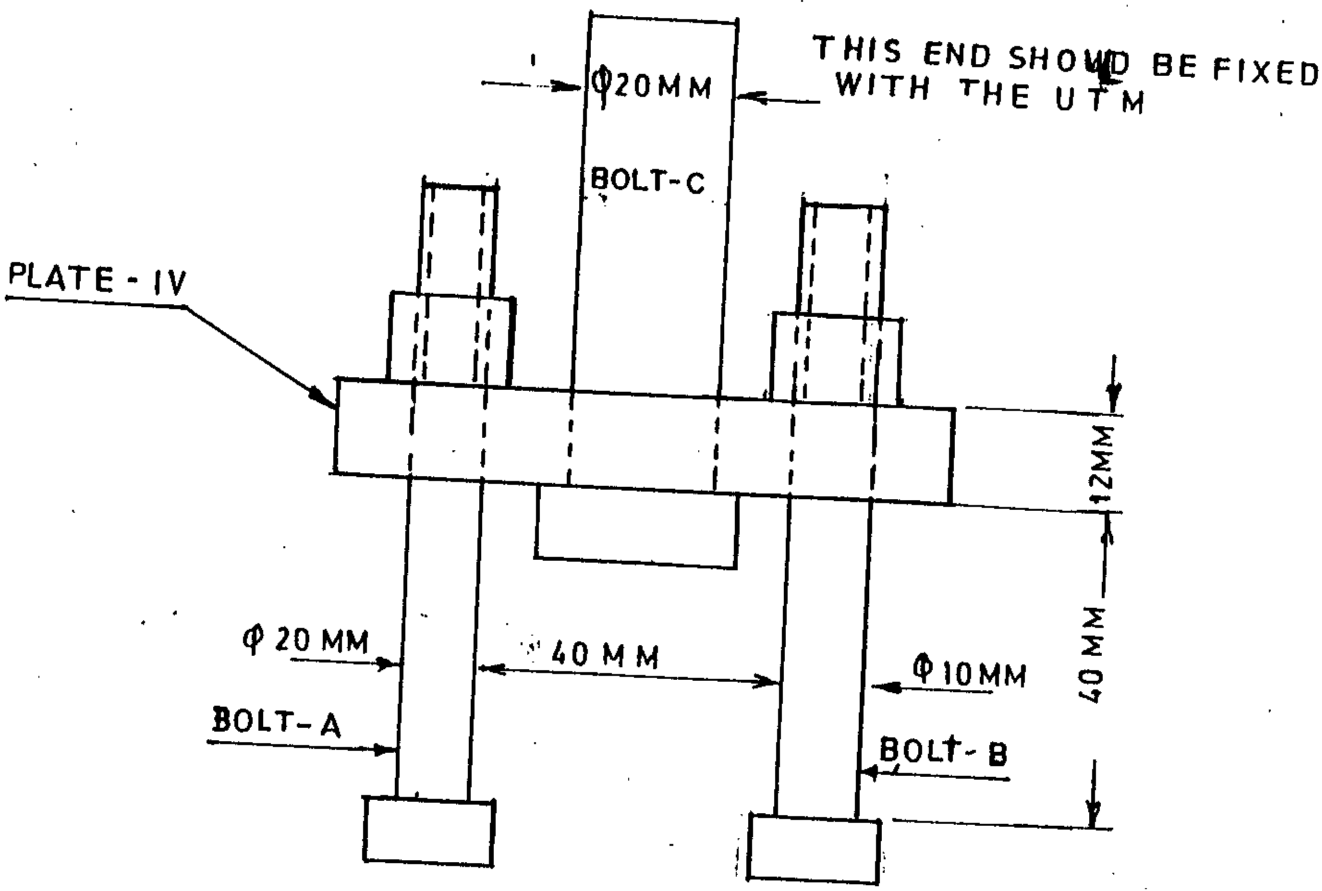


**Figure: 3.9** **The Special attachment for applying external tensile load on the portion of the test rig.**

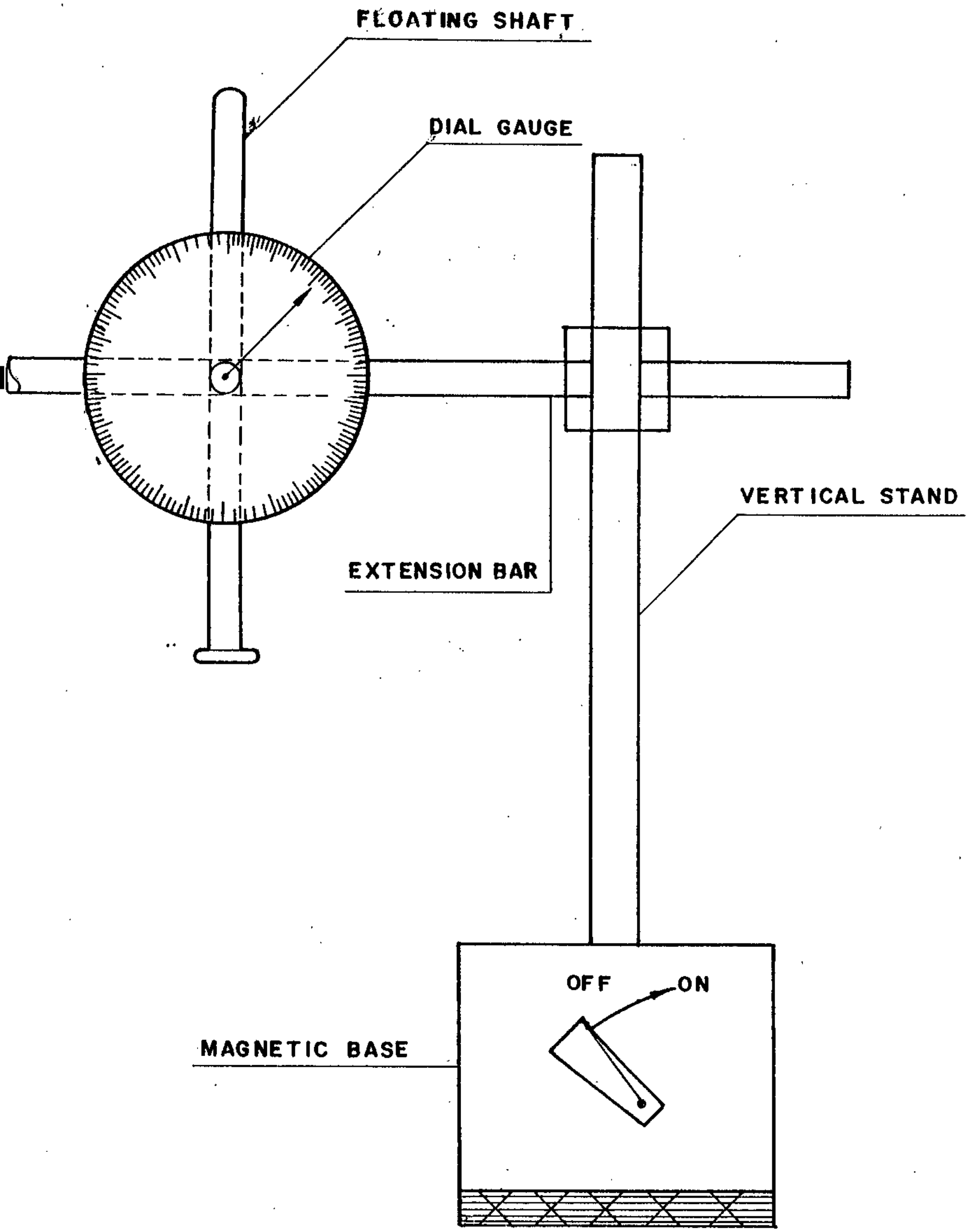


Figure: 3.10 Schematic of an extensometer .

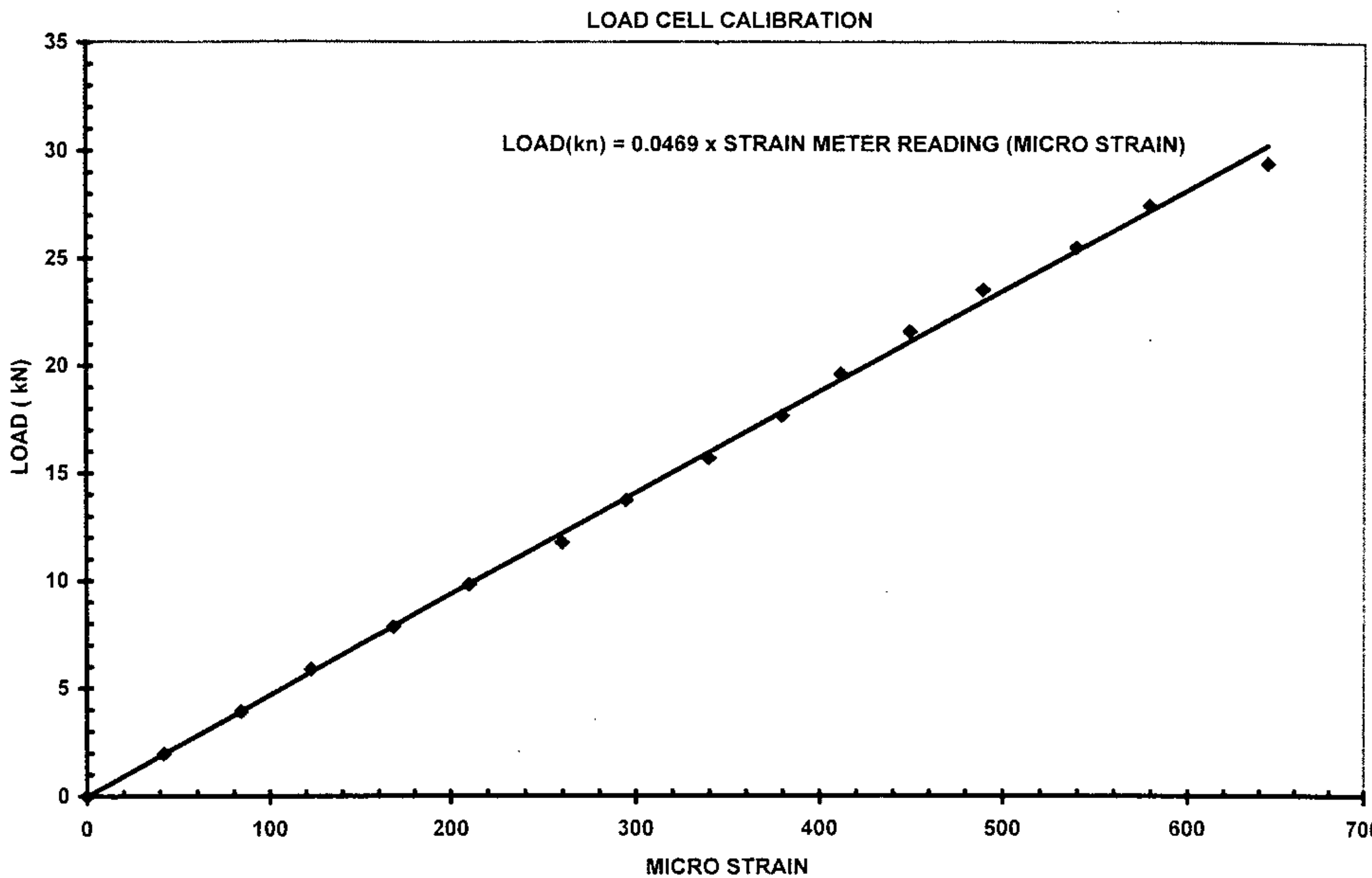


FIGURE-3.11 The relationship between the uni-axial compressive load imposed on the load cell and micro strain reading recorded by the strain meter.

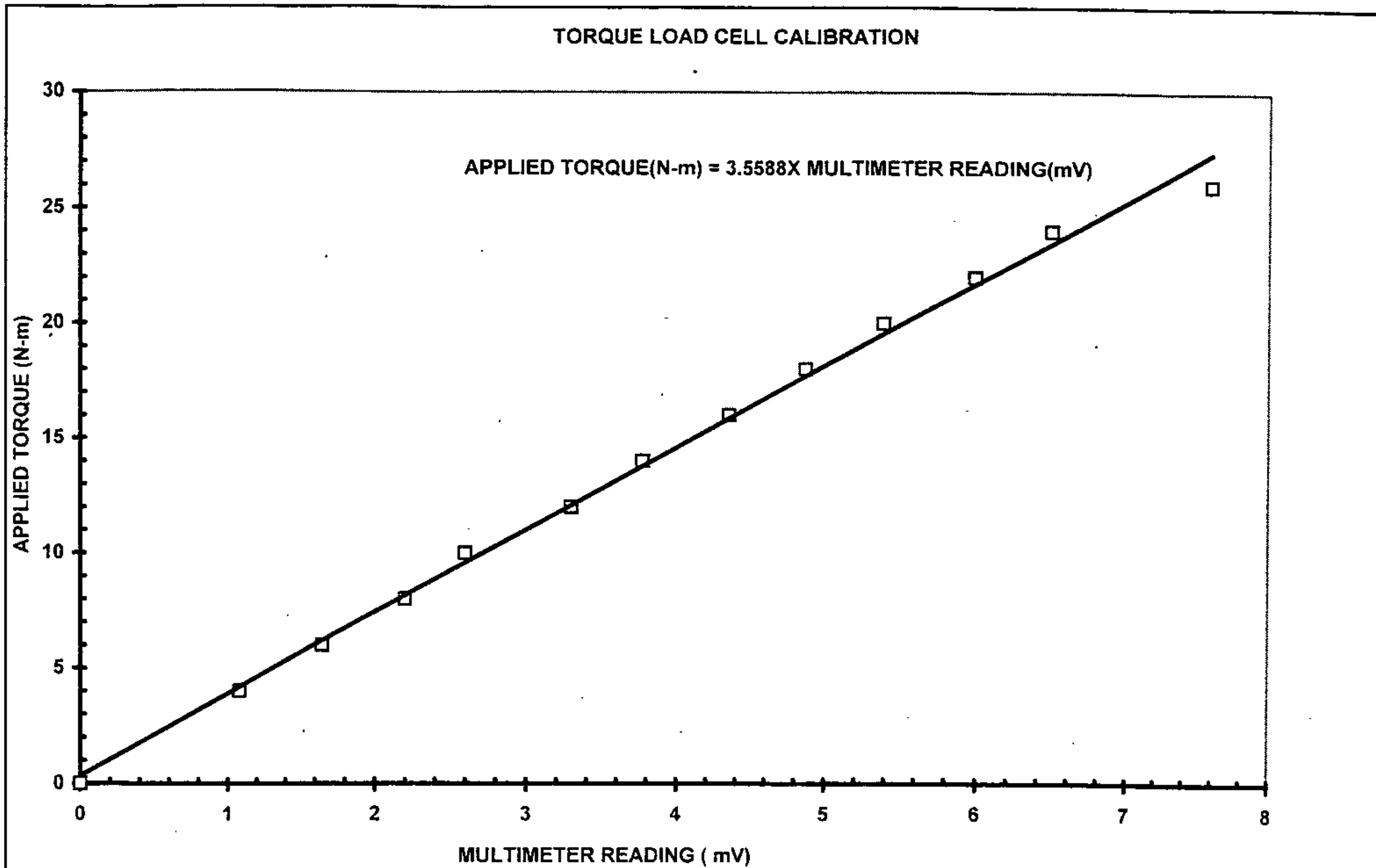


FIGURE-3.12 The relationship between the wrenching torque (N-m)applied to the torque transducer & its corresponding mV reading recorded by multimeter.

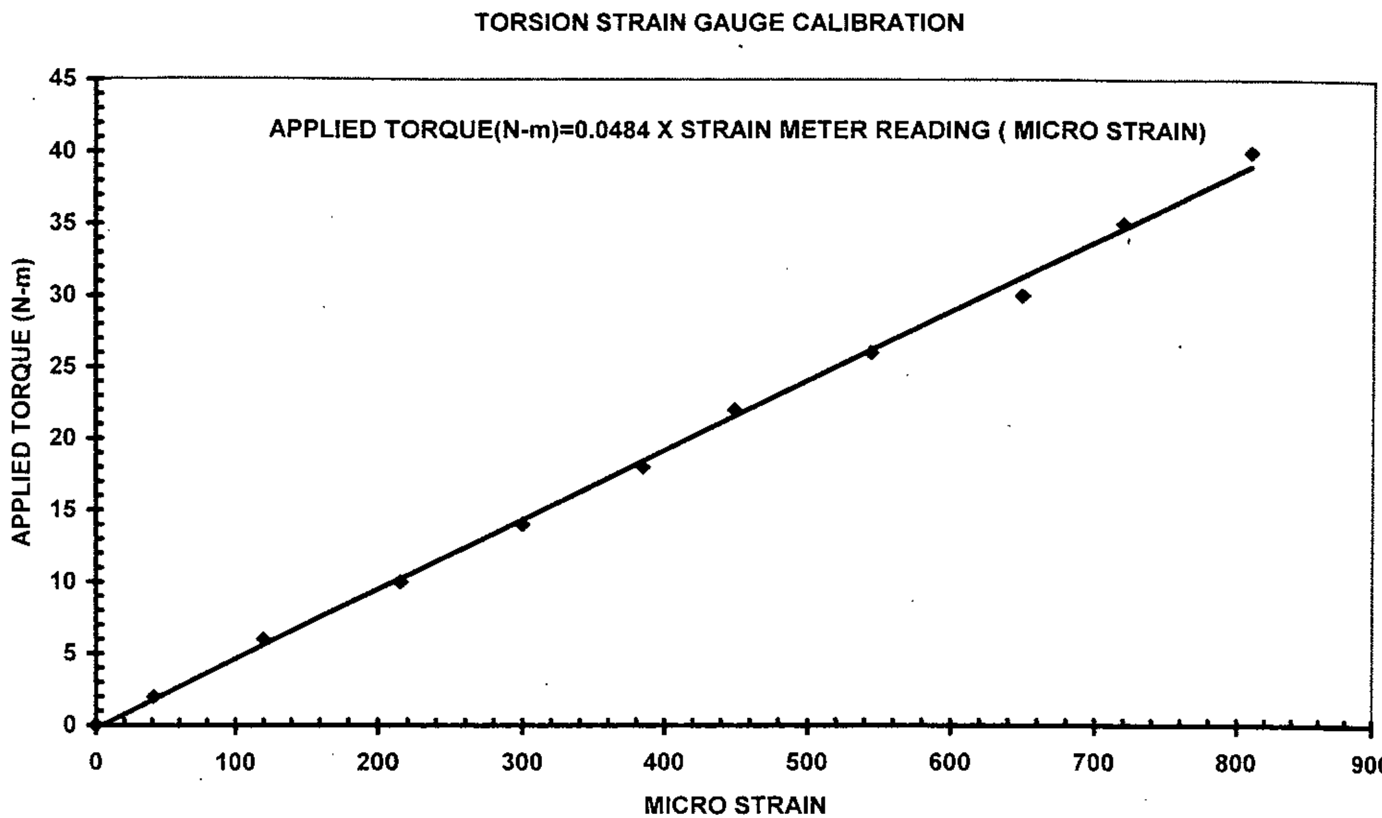


FIGURE-3.13 The relationship between the wrenching torque (N-m)applied at the bolt head and its corresponding micro strain reading of the torsion shear strain gauge fitted with the bolt shank recorded by the strain meter.

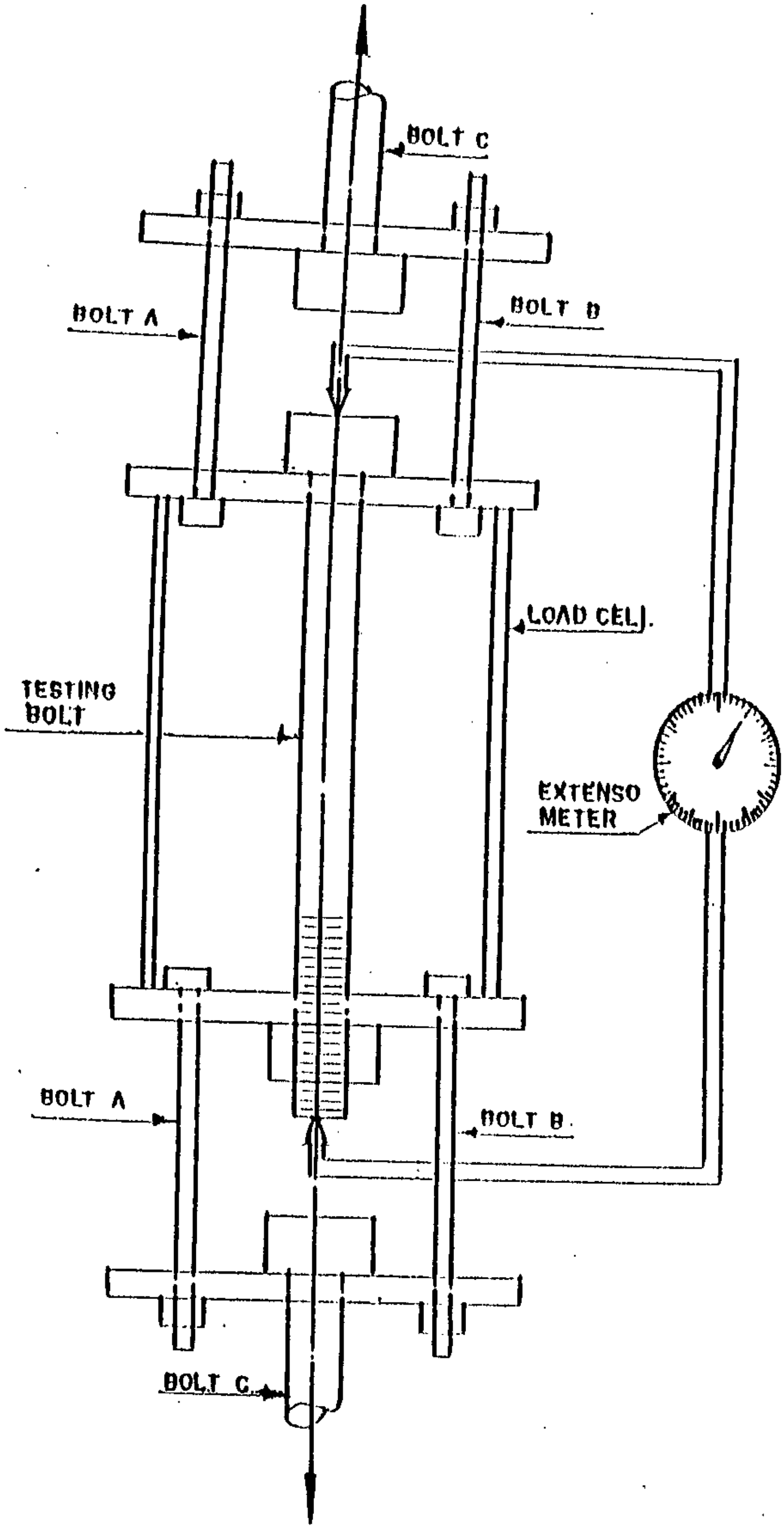


**Figure: 3.14** **Details of the test set-up which was used to apply the external tensile load (using the universal testing machine) after tightening the bolt.**

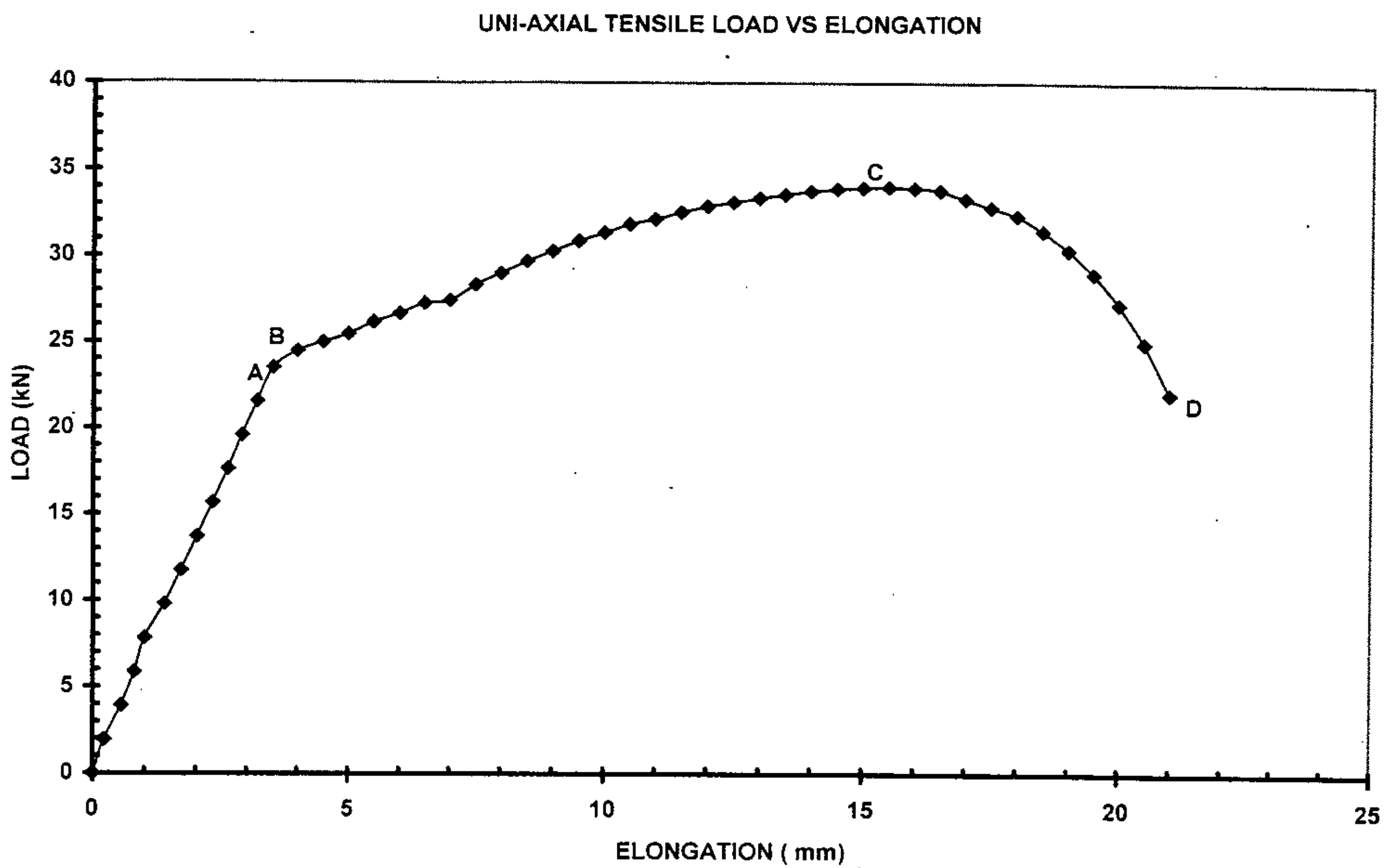


FIGURE-4.1 The uni axial tensile load applied on the bolt by universal testing machine vs its corresponding elongation of the bolt measured by the extensometer.

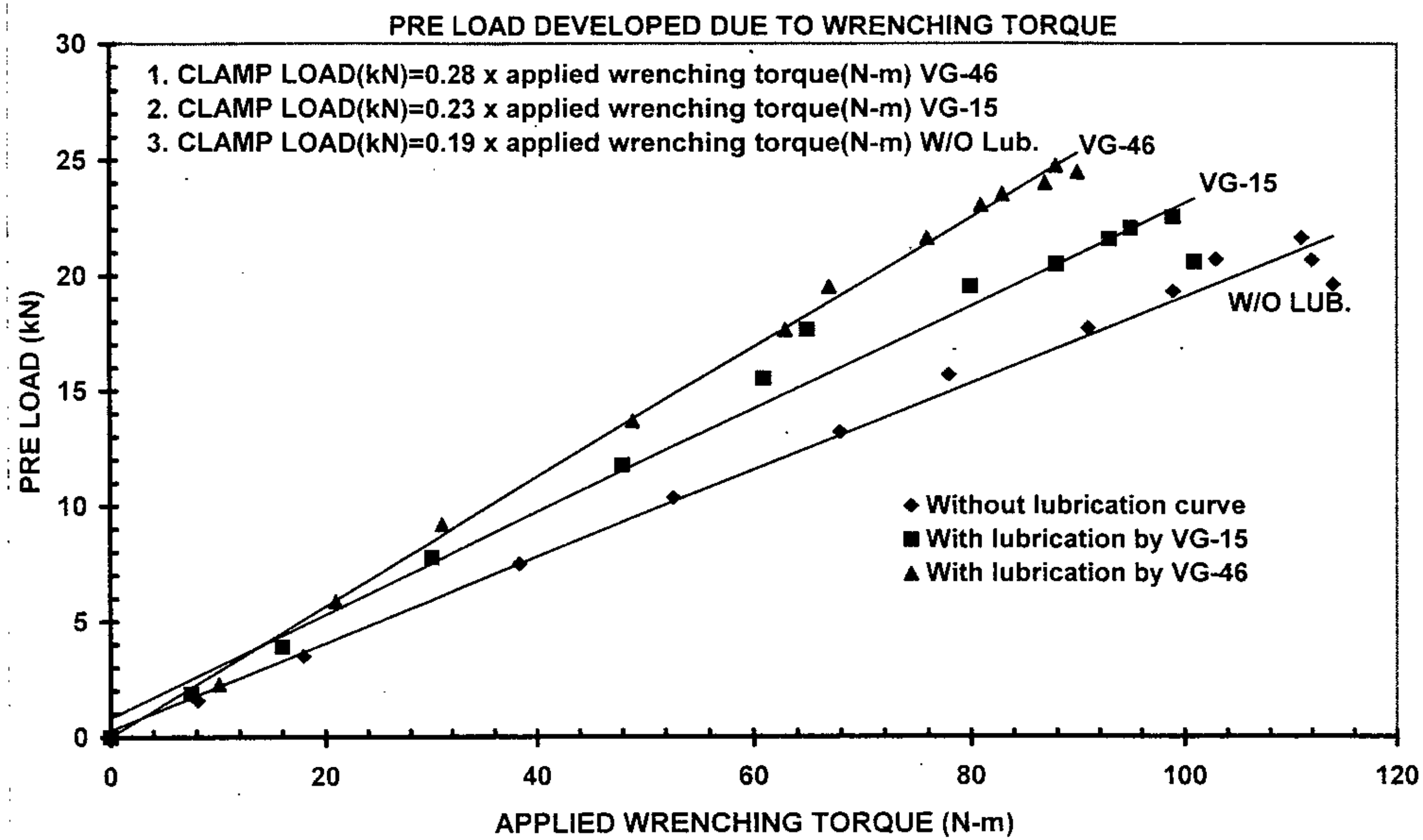


FIGURE-4.2 The relationship between the wrenching torque (N-m) applied at the bolt head and pre- load developed in the bolt due to applied wrenching torque under different lubricating condition.

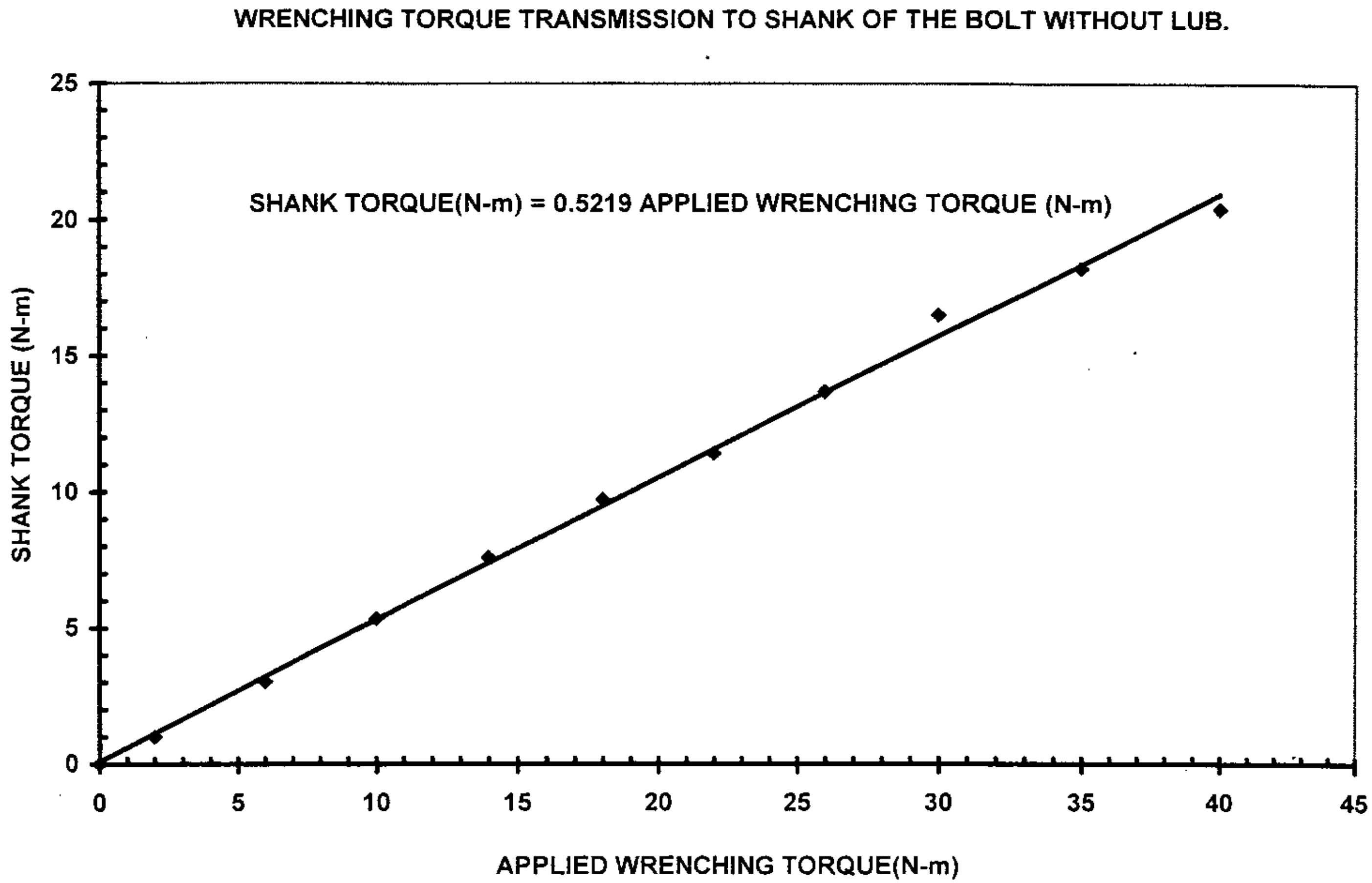


FIGURE-4.3 The relationship between the wrenching torque(N-m) applied at the bolt head & the amount of torque (N-m) transmitted to the shank of the bolt without lubrication.

WRENCHING TORQUE TRANSMISSION TO SHANK OF THE BOLT W/O LUB.

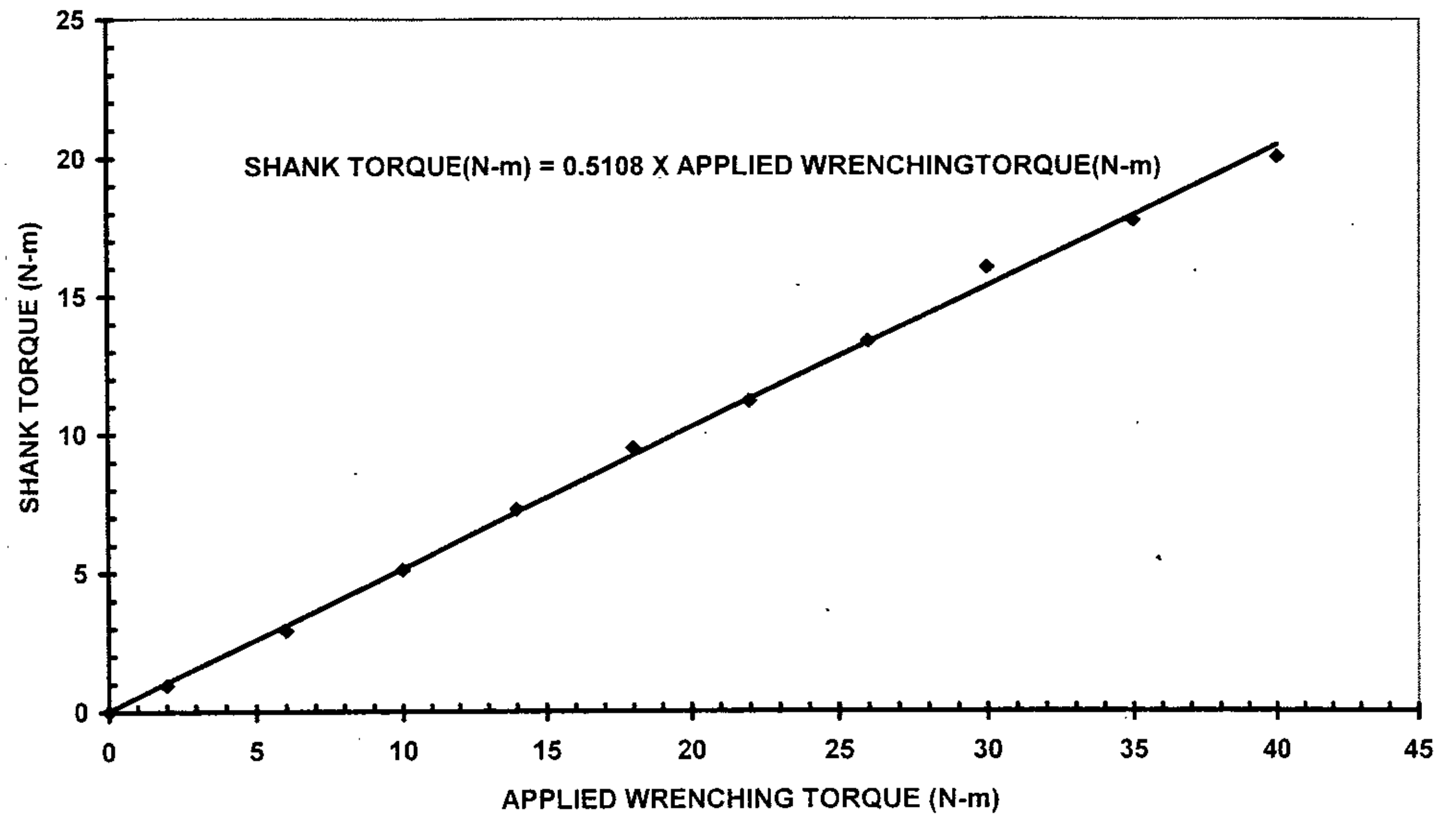


FIGURE-4.4 The relationship between the wrenching torque(N-m) applied at the bolt head & the amount of torque (N-m) transmitted to the shank of the bolt without lubrication.

WRENCHING TORQUE TRANSMISSION TO SHANK OF THE BOLT WITH LUB VG-15

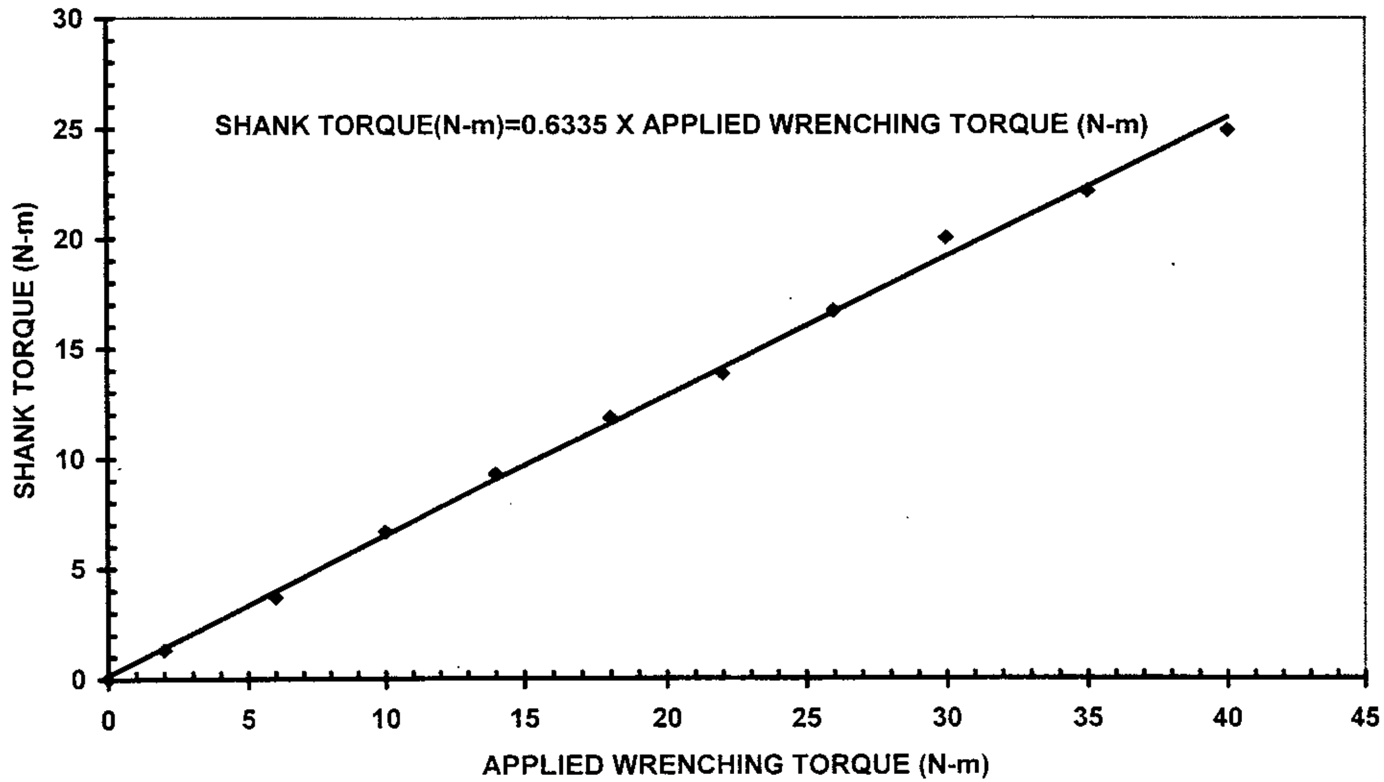


FIGURE-4.5 The relationship between the wrenching torque(N-m) applied at the bolt head and the amount of torque (N-m) transmitted to the shank of the bolt with lubrication by VG-15

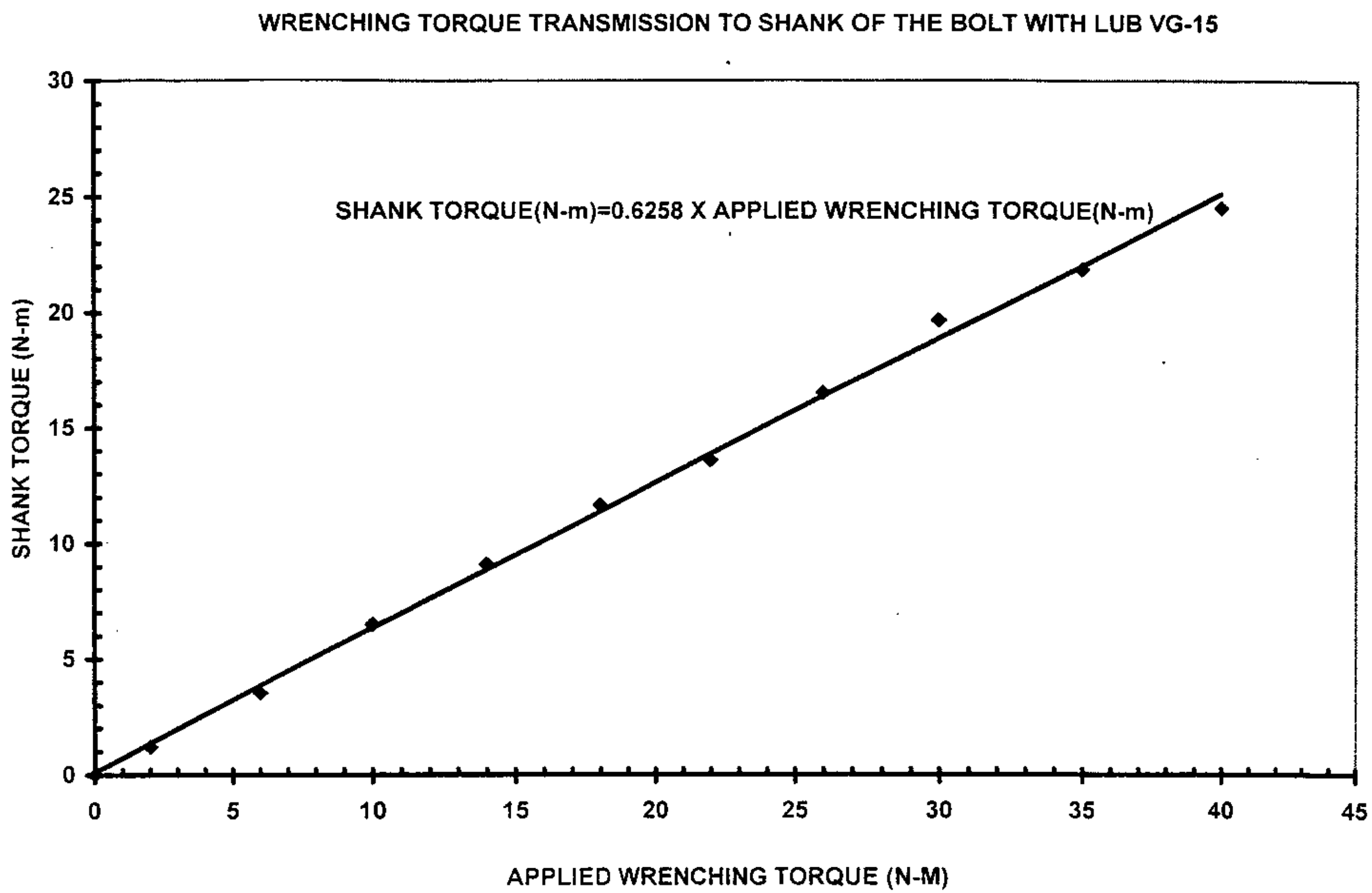


FIGURE-4.6 The relationship between the wrenching torque(N-m) applied at the bolt head and the amount of torque (N-m) transmitted to the shank of the bolt with lubrication by VG-15

95652

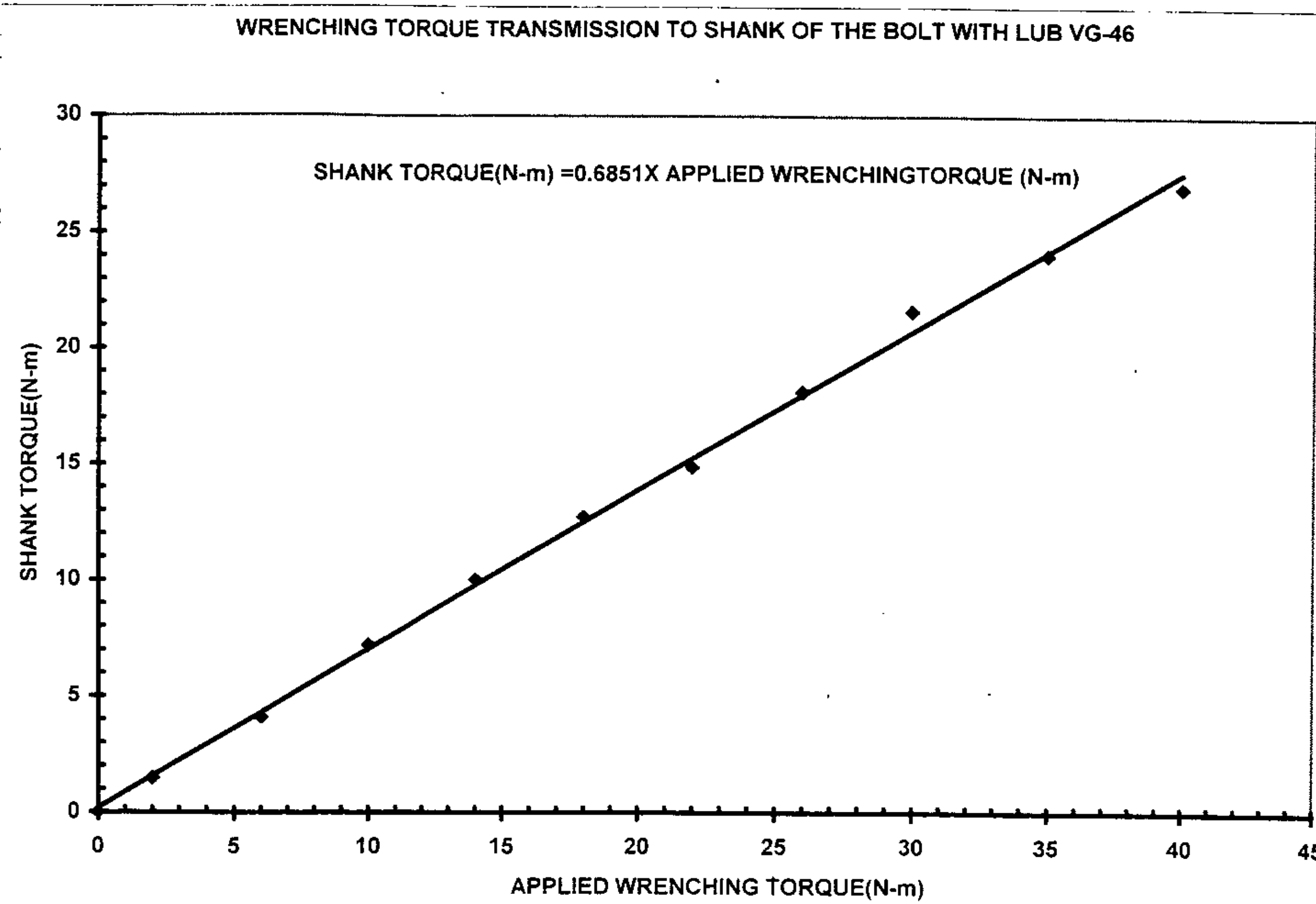


FIGURE-4.7 The relationship between the wrenching torque(N-m) applied at the bolt head and the amount of torque (N-m) transmitted to the shank of the bolt with lubrication by VG-46

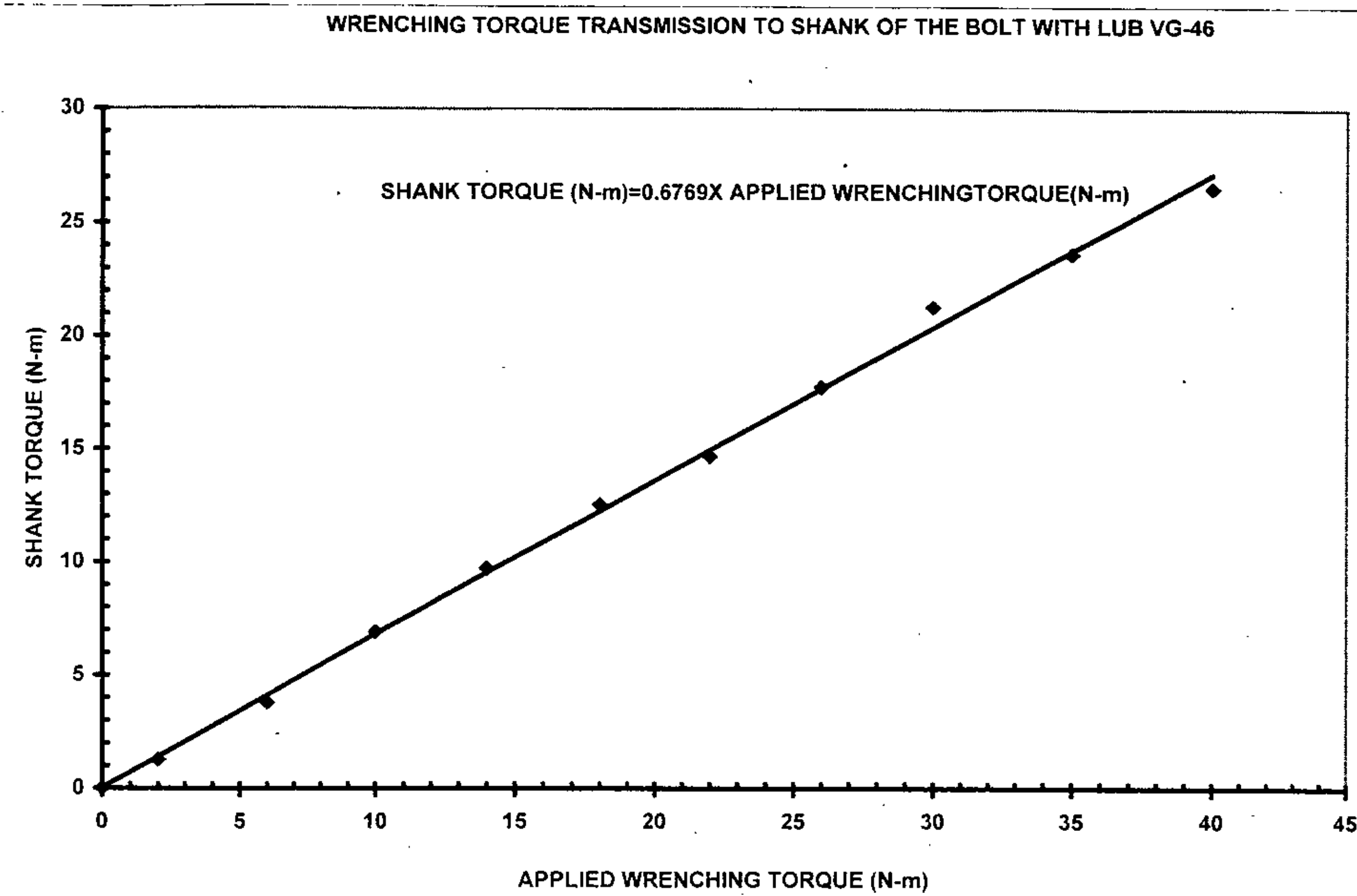


FIGURE-4.8 The relationship between the wrenching torque(N-m) applied at the bolt head and the amount of torque (N-m) transmitted to the shank of the bolt with lubrication by VG-46

COMPARISION OF PRE-LOAD VS ELONGATION OF A TIGHTENED BOLT UNDER DIFFERENT LUB CONDITION WITH THAT OF UNI-AXIAL TENSILE LOAD CURVE.

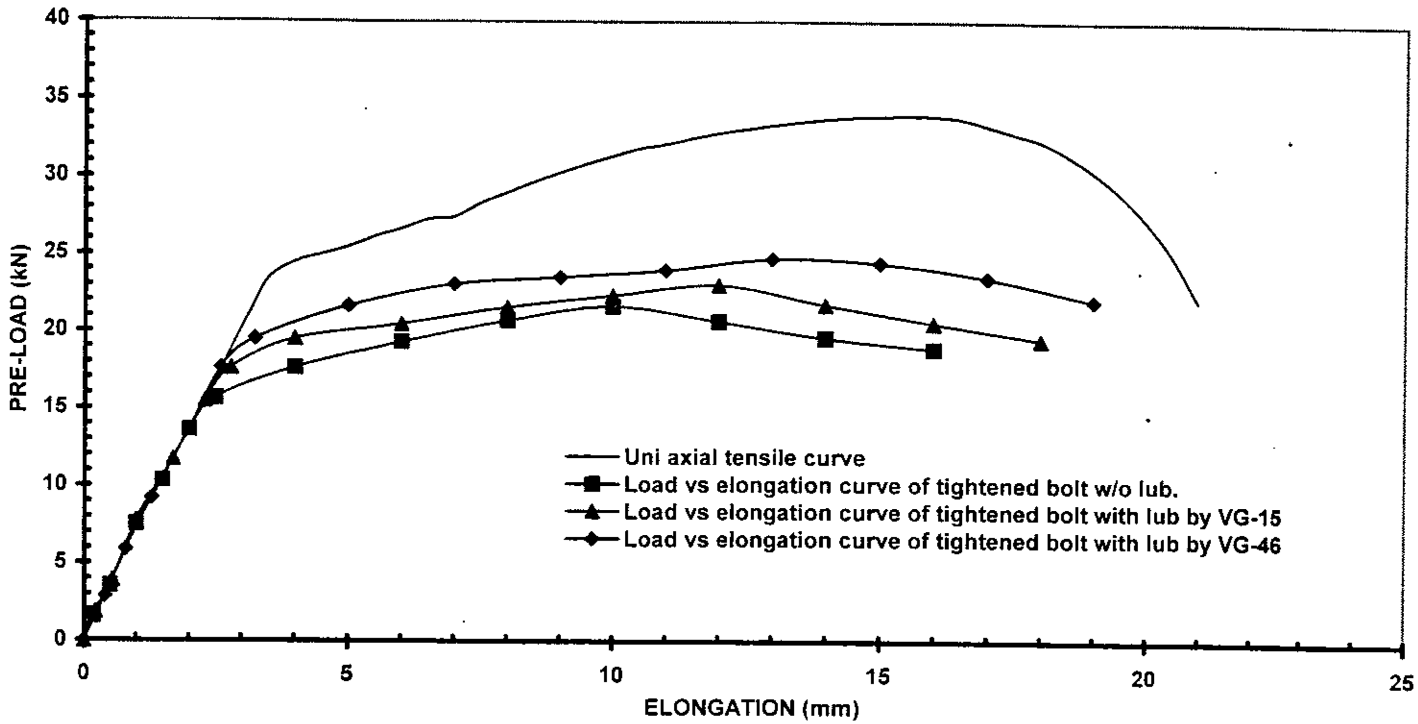


FIGURE-4.9 (a)Curve-1,The uni-axial tensile load vs elongation on the bolt curve, (b) Curve-2, Pre-load developed on the bolt head vs elongation of the bolt w/o lub. (c) Curve-3, Pre-load developed on the bolt head vs elongation of the bolt w/lub by VG-15 (d) Curve-4,Pre-load developed on the bolt head vs elongation of the bolt w/lub by VG-46.

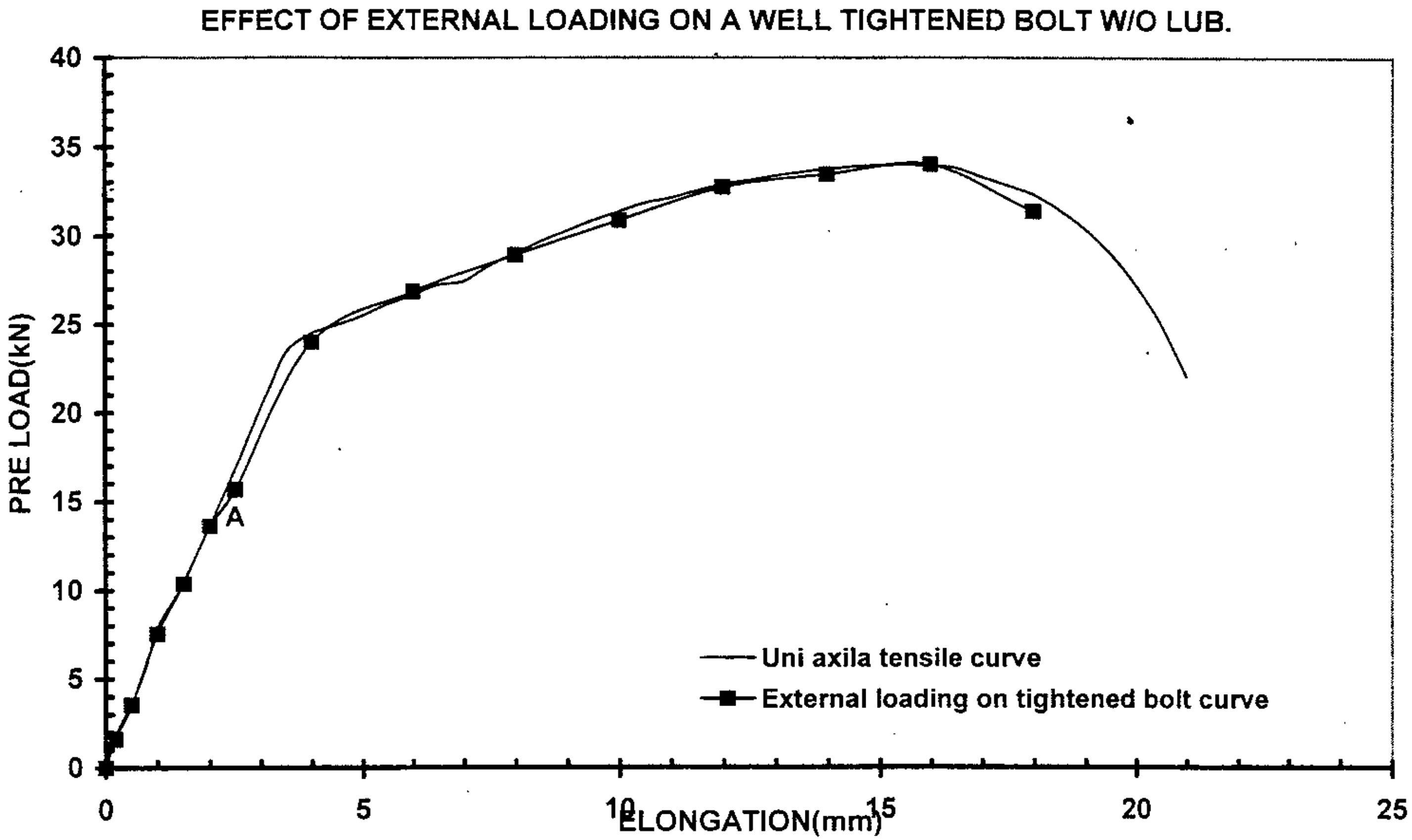


FIGURE-4.10 Comparision of maximum load carrying capacity of a well tightened bolt up to combined yield point and then applied simply external tensile load with that of a virgin bolt.

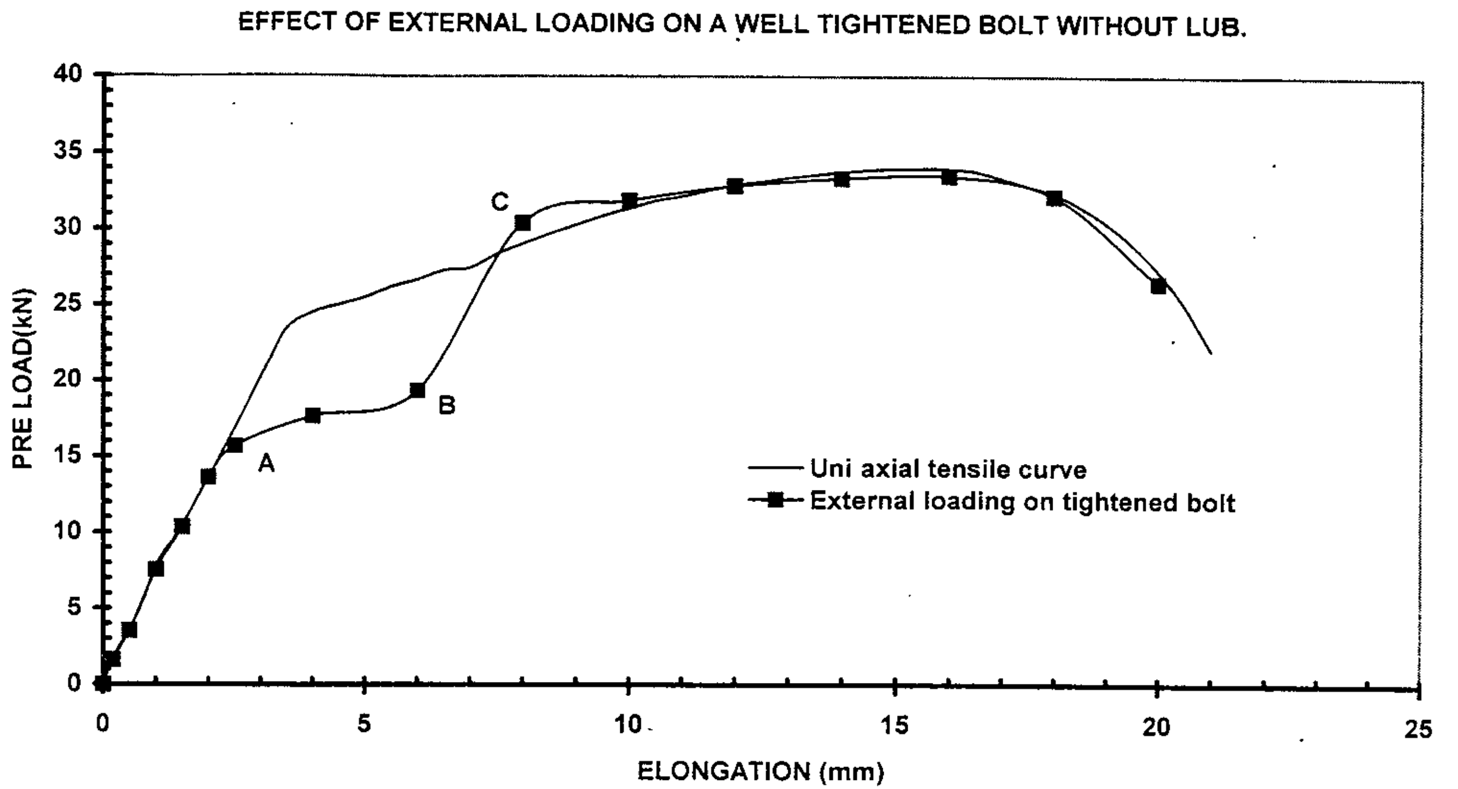


FIGURE-4.11 Comparision of maximum load carrying capacity of a well tightened bolt without lubrication followed by external loading when plastic deformation of 6 mm with that of a virgin bolt.

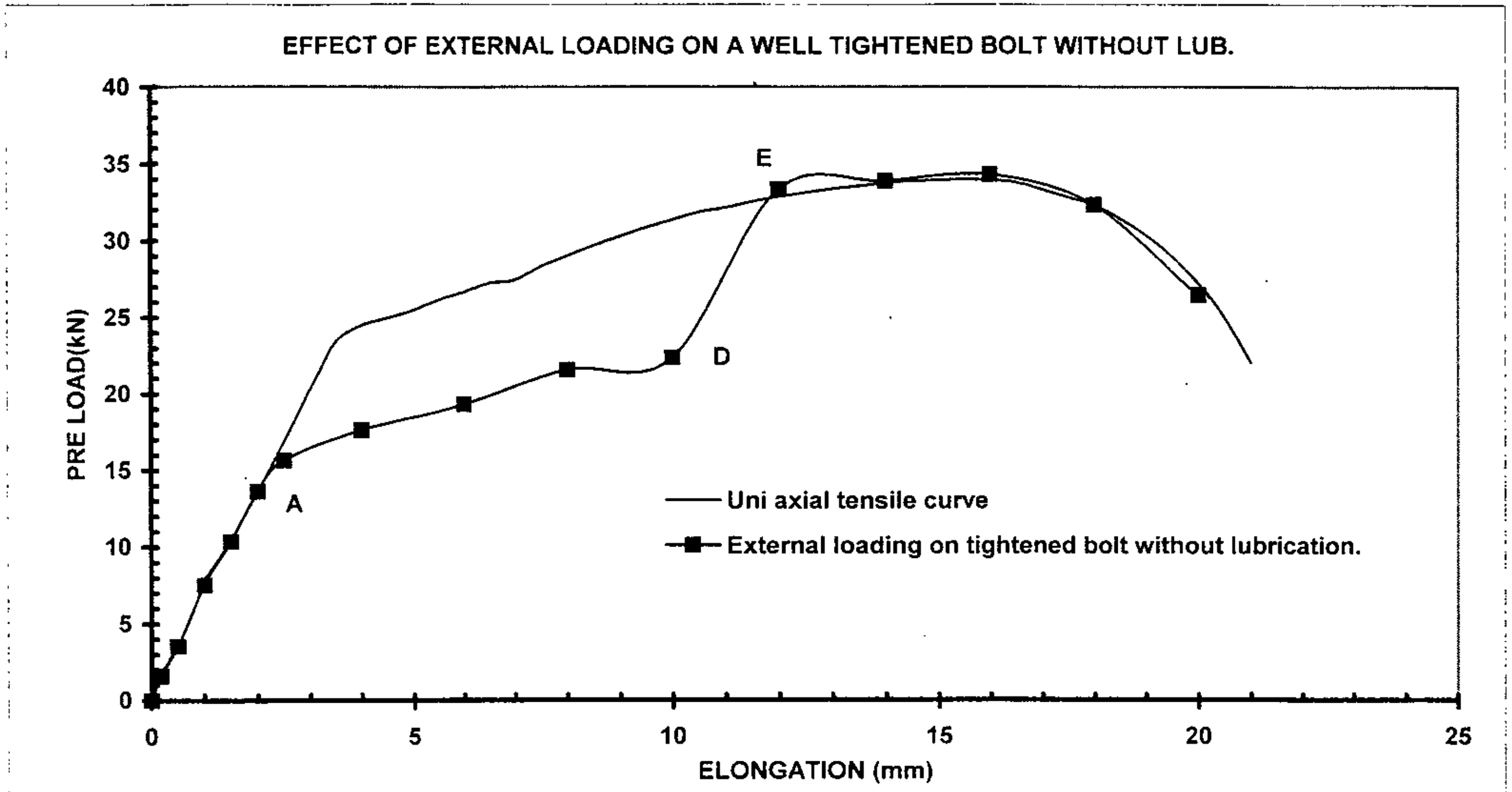


FIGURE-4.12 Comparision of maximum load carrying capacity of a well tightened bolt without lubrication followed by external loading when plastic deformation of 10 mm with that of a virgin bolt.

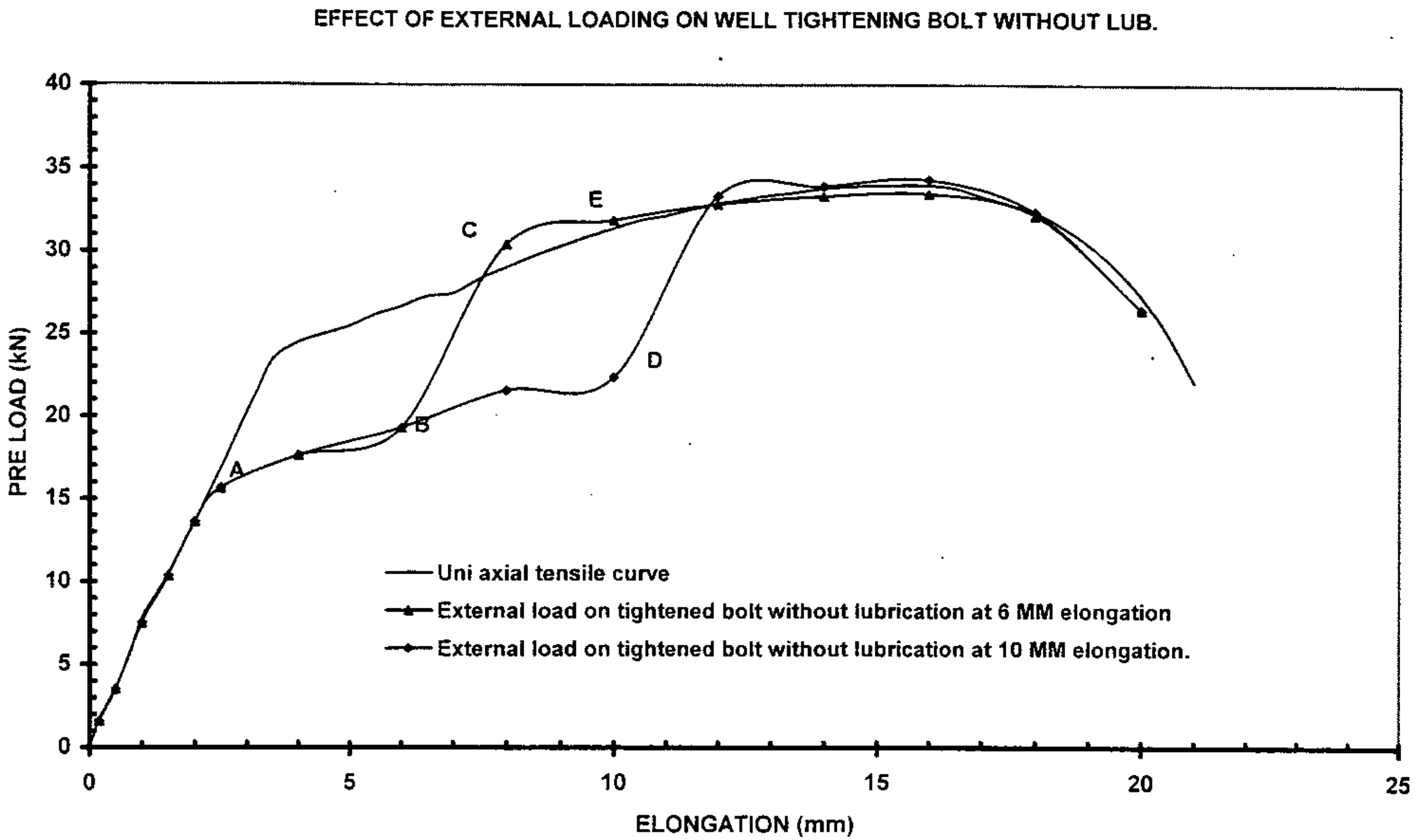


FIGURE-4.13 Comparision of maximum load carrying capacity of a well tightened bolt without lubrication followed by external loading at different level of plastic deformation with that of a virgin bolt.

EFFECT OF EXTERNAL LOADING ON WELL TIGHTENED BOLT WITH LUB BY VG-15

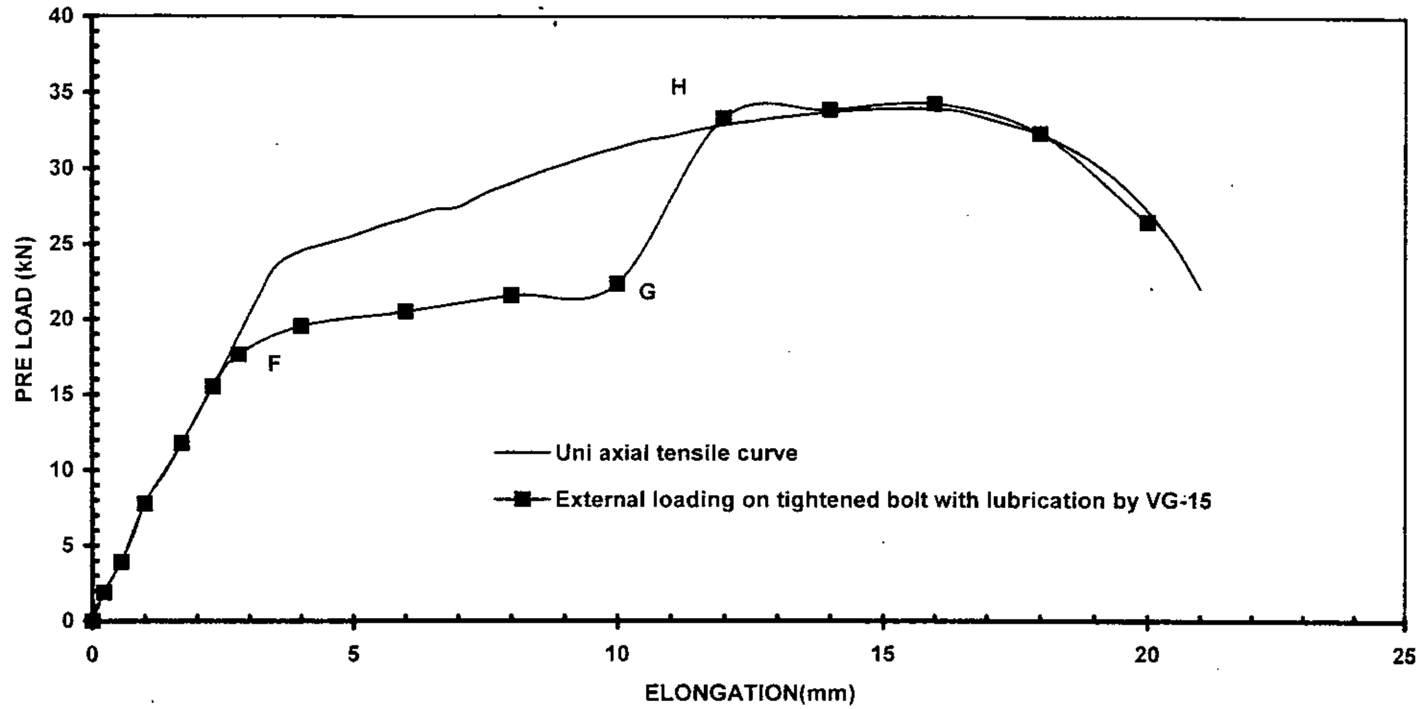


FIGURE-4.14 Comparision of maximum load carrying capacity of a well tightened bolt with lubrication by VG-15 followed by external loading when plastic deformation of 10 mm with that of a virgin bolt.

EFFECT OF EXTERNAL LOADING ON A WELL TIGHTENED BOLT WITH LUB BY VG-46

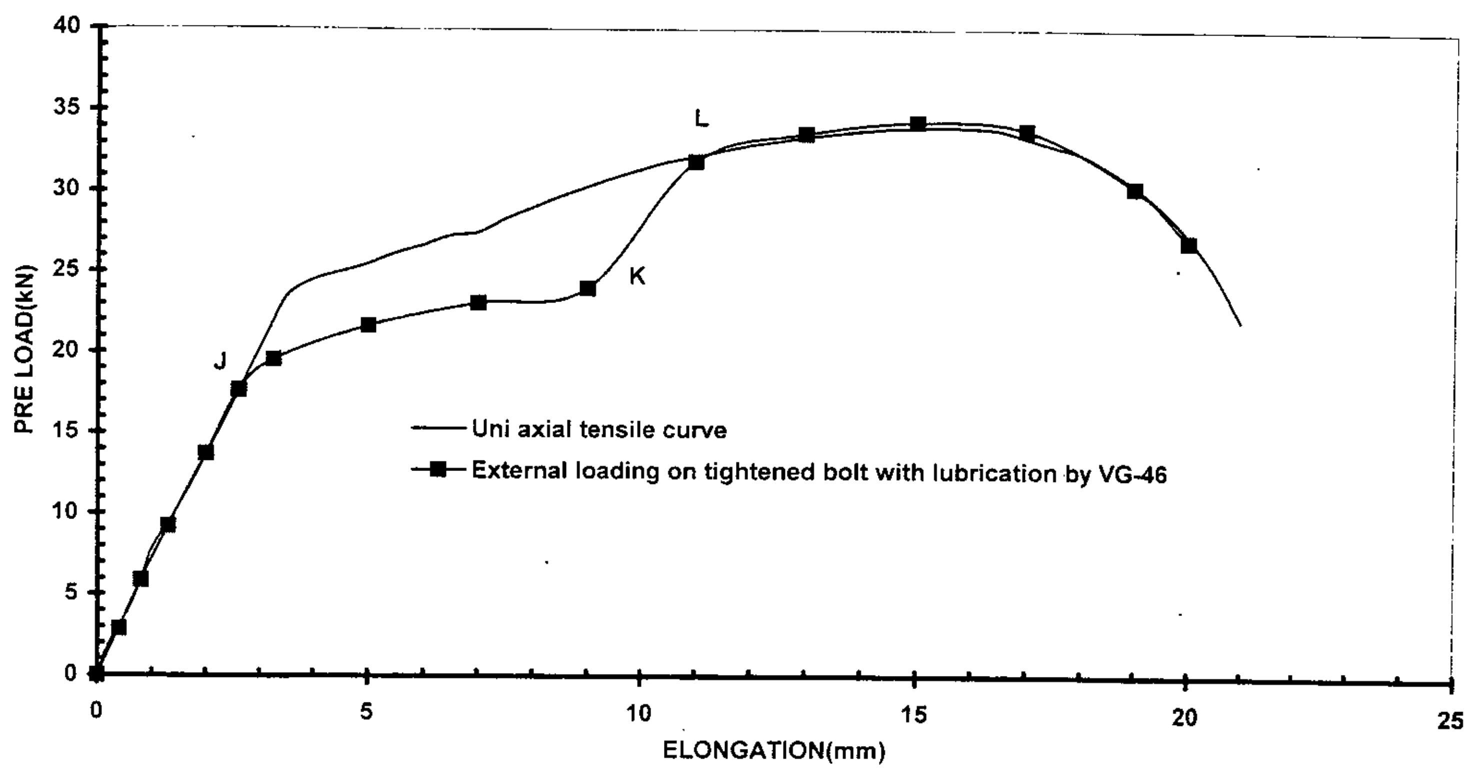


FIGURE-4.15 Comparision of maximum load carrying capacity of a well tightened bolt with lubrication by VG-46 followed by external tensile loading when plastic deformation of 9 mm with that of a virgin bolt.

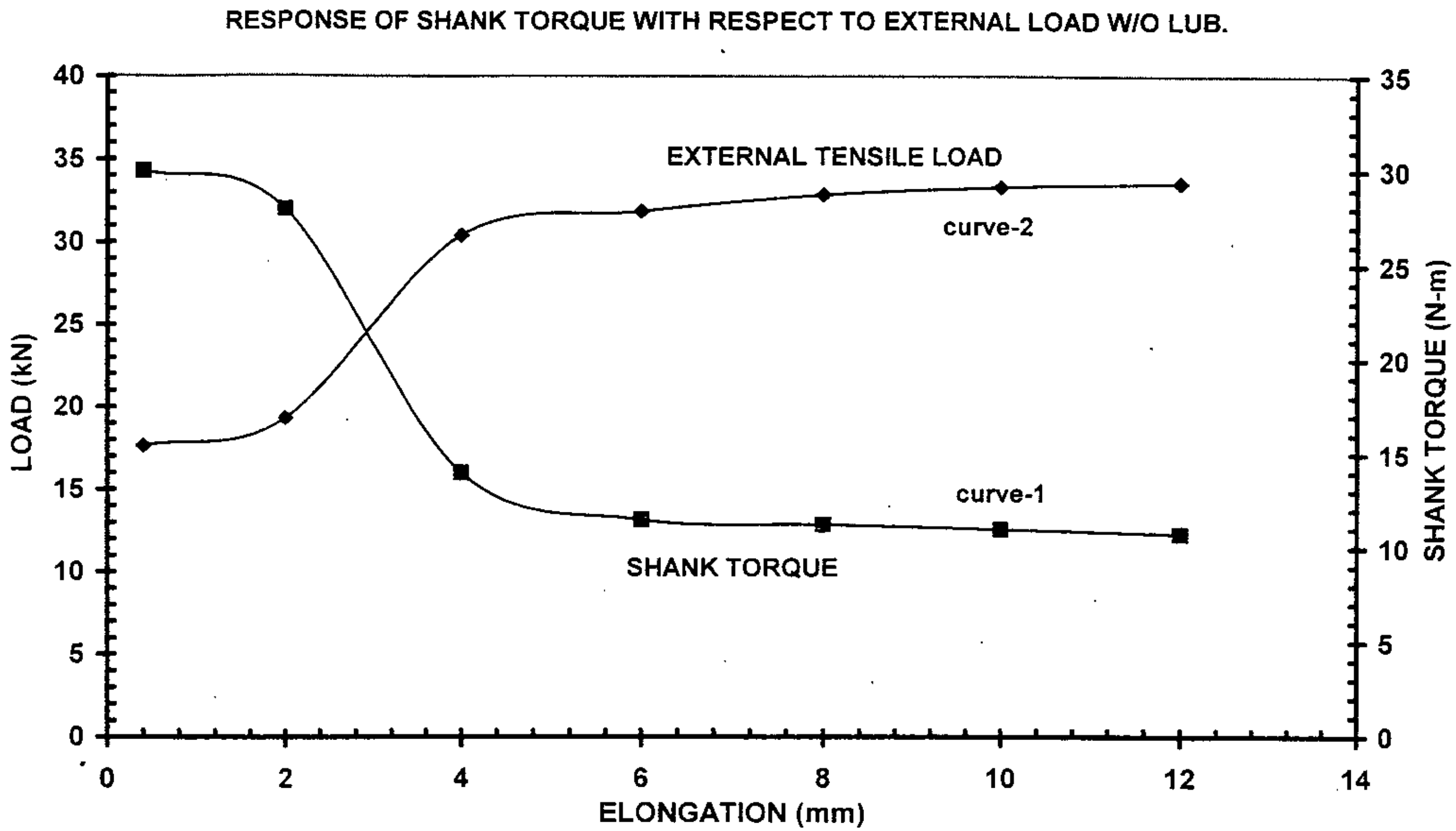


FIGURE:4.16 (a) curve-1 Represents the state of the shank torque of the bolt with respect to the elongation of the bolt when external tensile load was applied (b) Curve-2, Represents the external load applied on the bolt vs elongation of the bolt without lubrication condition.

RESPONSE OF SHANK TORQUE WITH RESPECT TO EXTERNAL LOAD WITH LUB VG-15

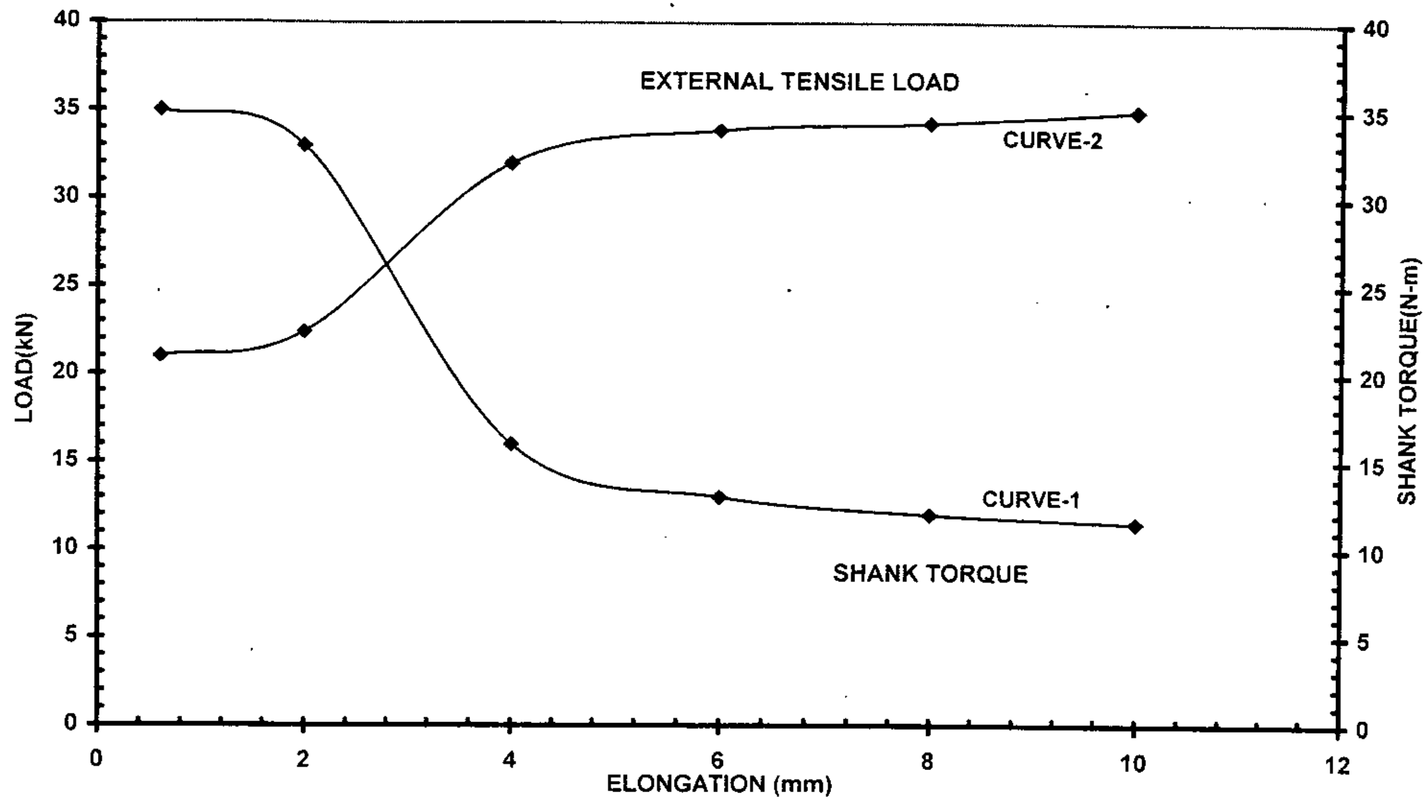


FIGURE-4.17 (a)Curve-1, Represents the state of the shank torque of the bolt with respect to the elongation of the bolt when external tensile load was applied.(b) Curve-2,Represents the external load applied on the bolt vs elongation of the bolt under lubrication condition by VG-15 .

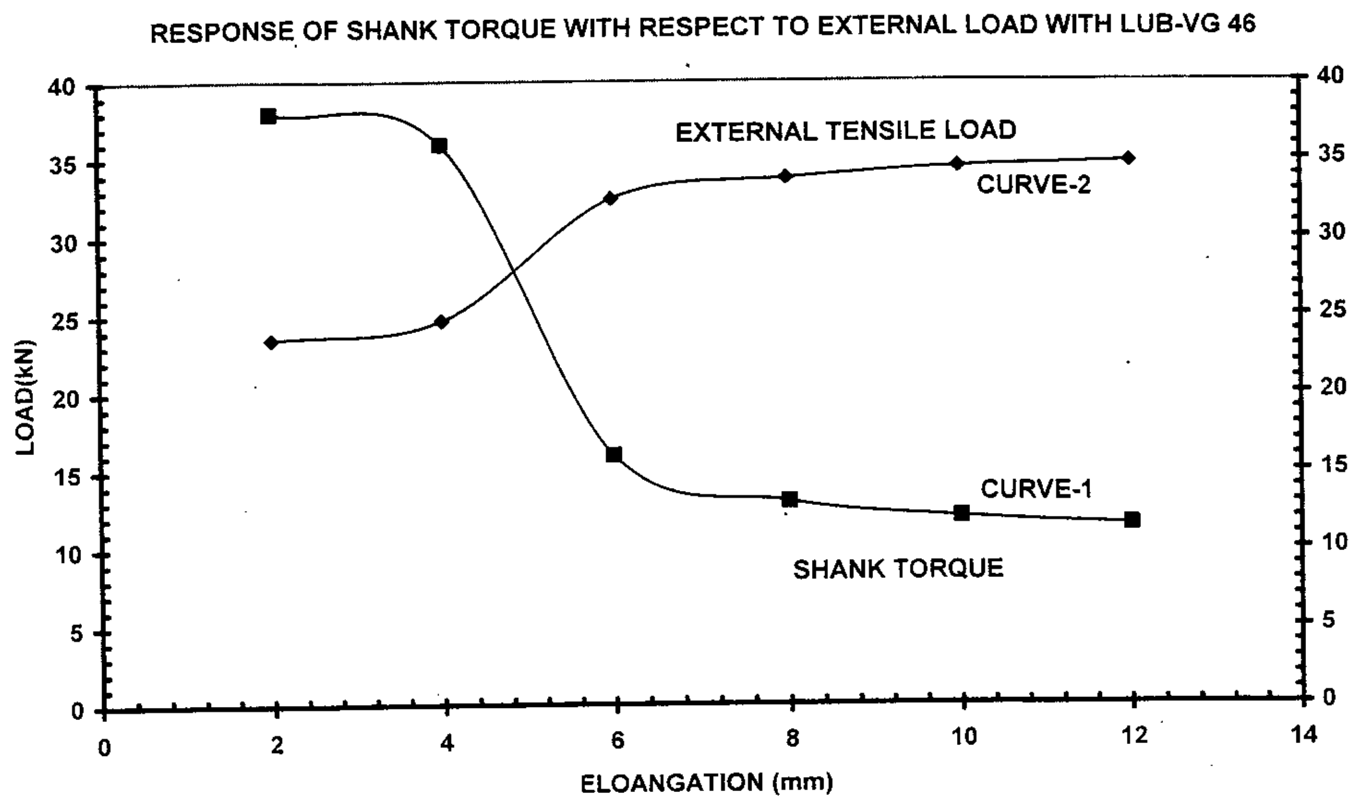


FIGURE-4.18 (a)Curve-1, Represents the state of the shank torque of the bolt with respect to the elongation of the bolt when external tensile load was applied.(b) Curve-2, Represents the external load applied on the bolt vs elongation of the bolt under lubrication condition by VG-46 .

**REFERENCES:**

[1] Finkelston, R.J. and Wallace, P.W.,“ Advances in High Performance Mechanical Fastening”, 1980, SAE Paper No 800451.

[2] Junker , G.H. and Wallce, P.W.,“ The Bolted Joint Economy of Design Through Improved Analysis and Assembly”, Proc. *Instn, Mech Engrs,* 1984, 1,255-266.

[3] Duff, B. and Monaghan, J.M., “ Behaviour of Fastener Tightened to Yield ”,Proc IMC-1, trinity college dublin, 1984, pp. 119-143.

[4] Morrison, J. L. M. and Shepherd, W.M.,“An Experimental Investigation of Plastic Stress-Strain Relations”, *Proc, Inst, Mech. Eng.,* Vol. 163, 1950, p-1.

[5] Monaghan, J.M, and Duff, B., “The Influence of Friction on the Stress Distribution within Yield Tightened Fastener ”, Proc. IMC-2 University of Ulster, 1985, pp. 113-135.

[6] Monaghan, J.M and Duff,B., “The Effect of External Loading on A Yield Tightened Joint”, Int, J. Mach , tools manufacture , 1987, 27, 443-455.

[7] Boys, J.T. and Wallace, P.W., “ Design and Performance of an Automatic Control System for Fastener Tightening”, *Proc Inst. Mechanical Engioneers*, Vol. 191, 1977, pp. 371-380.

[8] Chapman ,I., Newnham ,J.,and Wallace, P.,“ The Tightening of Bolts to Yield and The Performance Under Load ”,ASME Journal of Vibration, Acoustics, Stress, and Reliability in Design ”,Vol.108,pp.213-221,1986,

[9] Monaghan, J.M., “The Influence of Lubrication on the Design of Yield Tightened Joints”, Journal of Strain Analysis, vol. 26, No 2, 1991.

[10] Hagiwara, M., Ohashi, N., and Yoshimoto, I.,“ Characteristics of Bolted Joints in Plastic Region Tightening ”, Bull Japan Soc. of prec. Engg.,Vol.20,no.4,1986.

[11] Hariri, B. H., “M. Engg. Thesis”, Dublin City University ,Ireland, 1990.

[12] Monaghan, J. and Duff, B.,"The Effects of External Loading on a Yield Tightened Joint ", April , Int. J. Mach Tools Manufact 7 Apr. 1986.

[13] Newnham ,J. and Curley, L.,"Bolted Joints Under Tensile Service Loads: The Effect of Torsion in the Bolt on In-Service Clamping Loads ", ASME journal of Transactions , Vol. 115 , p-36, 1993.

[14] Morrison, J.L.M. and Shepherd, W.M., " An Experimental Investigation into Plastic Stress-Strain Relations ", *Proc L., Mechnical Engineers.* Vol. 163, pp.1-9,1950,

[15] Ford, H., "*Advanced Mechanics of Materials*, Longmans ", London, 1963.

[16] Meyer,G. and Strelow, D.,"How to Calculate Pre-load loss Due to Permanent Set in Bolted Joints.", *Assembly Engineering*, Mar 1972.

[17] Von Mises, R., "Mechanik Der Festen Koerper im Plastisch Deformable Zustant, Goettinger Nachar ", Math.- Phys., KI., pp. 582-592, 1913.

[18] Newnham, J. Curly, L., and Boos, H.P.," The response to External Loads of Bolted Joints Tightened to Yield.", VDI – Berichtsheft No. 766, Verein Deutscher Ingeniere Conference "Schraubenverbindungen.", Essen, Oct. 1989

[19] VDI 2230, 1986., "Systematic Calculation of High Duty Bolted Joints", Beuth Verlag GmbH, Berlin, 1986.

[20] Boys, J.T. and WALLACE, P.W.," Design and Performance of An Autometic Control System for Fastener Tightening ",*Proc. Inst. Mech Engng* 198B, (14) 255 (1984)

[21] Tresca, H., "Sur, I' Ecoulement Des Crops Solids Soumis a de Fortes Pression ",compt. Rend.59, pp.754, 1864.

[22] J. Chakrabarty.,"Theory of Plasticity.", McGraw-Hill,Inc.,1987.

## APPENDIX-A
## PROPERTIES OF THE LUBRICANT USED

---

In this experiments two types of lubricant have used. These are viscosity grade-15 named VG-15 viscosity grade-46 named VG-46. The characteristics of the lubricant are as follows :-

| Sr. no. | characteristics | VG-15 | VG-46 |
|---|---|---|---|
| 1. | Density at $15^0$ c | 0.869-0.875 | 0.879-0.880 |
| 2. | Viscosity at $40^0$ c | (12-16.5) cst | (41.4-50.6)cst |
| 3. | Viscosity at $100^0$ c | 3.32 cst | 6.9 cst |
| 4. | Viscosity Index | 60 | 95 |
| 5. | Flash point(min) | $163^0$c | $225^0$c |
| 6. | Pour point | $-32^0$c | $-30^0$c |
| 7. | Neutralization value(max) | 0.20 mg ko H/gm | 0.20 mg ko H/gm |
| 8. | Water content | Absent | Absent |
| 9. | Aniline point | 87 | 100 |