EFFECT OF CYCLIC BENDING ON THE NOTCH SENSITIVITY TO FRACTURE

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CERTIFICATE OF APPROVAL

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A Thesis

By

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ABSTRACT

An experimental investigation on the fatigue behavior of mild steel shaft is presented. A newly developed machine in Mechanical Engineering Department, BUET, was used to test the fatigue behavior of mild steel shaft. Specially, notch sensitivity of mild steel shaft with V-notches were examined in terms of fatigue life of the notched specimens and compared with those of plain mild steel specimens. The experiments were done at higher stress level.

Three notch angles were selected as 30°, 60° and 90°. Inclined loading was permitted in the machine with a view to introduce the combined effect of tensile stress and flexural stress on the fatigue life of the specimens. The effect of tensile stress over flexural stress was noted.

The effect of notch angle on fatigue life of mild steel specimens were investigated. Maximum stress levels were selected in the range of 60 percent to 85 percent of ultimate tensile strength of plain specimens. The effect of notch- root radius of V-notched specimens were also investigated. Two sets of V-notched specimens were selected to test the size effect of the specimens on the notch sensitivity to fracture. All the tests were done in the laboratory environment at ambient temperature.

List of Symbols

a	crack size, crack length
D	big diameter of shaft
d	diameter at the notch section
F_0	steady load on lower load arm
F_a	fluctuating load on lower load arm
K_t	stress concentration factor
l	moment arm, distance between specimen hole centre and notch root.
M	bending moment
N_f	fatigue life, life cycle
P_0	inclined steady force on the specimen
P_a	inclined fluctuating load on the specimen
R	flexure stress ratio (S_f / S_t)
R'	alternating stress ratio (S_a / S_m)
R''	Stress ratio (S_{min} / S_{max})
r	notch-root radius
S_{f0}	steady flexural stress component
S_{t0}	steady tensile stress component
S_{fa}	alternating flexural stress component
S_{ta}	alternating tensile stress component
S_f	total flexural stress
S_t	total tensile stress
S_a	total alternating stress amplitudes
S_m	total mean stress
α	notch angle
θ	loading angle, inclination of loading with axis of specimen
%EL	percent elongation
FFT	first Fourier transform
HR	Rockwell hardness
%RA	percent reduction of area
S_{max}	maximum stress
S_{min}	minimum stress
UTS	ultimate tensile strength

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Chapter 1
Introduction

1.1 General : In the last half of the 18th century, structural design was based almost entirely on the concepts of static strength of materials. At that time, there were relatively few sources of vibration or repeated stressing in comparison with those existing today. Sources of motive power were limited, speeds were low, and most structural parts were normally designed with large factor of safety. Under such circumstances design on the basis of static strength properties was quite satisfactory.

During the first half of the 19th century, development of the steam engine led to increasing sources of repeated stress on metal parts and structural elements. Shortly thereafter "unexplainable fractures", - particularly in locomotive axles, became great concern to engineers. Though the axles were made of ductile iron, they were found to crack in an apparently brittle manner after varying periods of service. The term "fatigue" was attached to such fractures which usually occurred only after a considerable length of service life.

The word "fatigue" refers to the behavior of materials under the action of repeated stresses or strains as distinguished from the behavior under static stresses or strains. The definition of fatigue as currently stated by ASTM as follows [20].

"The process of progressive localized permanent structural change occurring in a material subjected to conditions which produce fluctuating stresses and strains at some point or points and which may culminate in cracks or complete fractures after a sufficient number of fluctuations".

In modern times mechanization is increasing in almost every field. Modern buildings contain motors and sources of vibration which were absent a century ago. Factories and shops employ heavy machinery operating at high speeds. New means of rapid transportation are being developed with corresponding sources of vibration and dynamic stresses. Improvements in design require component parts to operate at higher stresses, both static and dynamic. Accordingly, design to resist fatigue failure under repeated stresses is becoming of increasing concern in many fields of engineering.

Failure of metals by fatigue results from loads which are varied or repeated and the maximum load required to failure in this way is much less than the static breaking load. In service, many components and structures are subjected to varying loads. Although the average stresses are often low, local concentrations of stress, which do not reduce the static strength much, may often lead to failure by fatigue. Indeed, a far greater number of failures in service are by fatigue and a relatively few by static loading.

The most striking characteristic of fatigue failures is the lack of deformation in the region of the fractures, even in materials like mild steel, which are quite ductile when broken by a static load. This is one of the dangers of fatigue for there is generally no prior indication of impending failure. Fatigue cracks are usually fine and difficult to detect and once they have grown to macroscopic size they may spread and cause complete fracture in a short time.

It is usually found that fatigue cracks originate at some surface discontinuity. This is because any change in section such as a hole, a change in shaft diameter, a groove, a key way or even a tool mark on the specimen gives rise to a concentration of stress. The static strength is little affected by such changes in section because the stress concentrations are relieved by plastic deformation. Under fatigue loading however much less plastic deformation occurs; consequently the range of stress remains considerably higher at stress concentrations than in the surrounding material, resulting in a reduced fatigue strength.

The magnitude of the stress at a stress concentration zone increases with the curvature of the surface. In other words the smaller the radius of a fillet or groove at a change of diameter in a shaft the greater is the stress concentration. It is therefore important in the design of parts to withstand dynamic stress to assess the stress concentration at each change in section and to ensure that an adequate radius of curvature is provided. Except in the few cases in which the variation of load in service is very small, it is unwise to design based on the static strength and to rely on a factor of safety as a safeguard against fatigue. This procedure, still widely adopted, leads to the use of excessively large factors of safety with a consequent increase in the weight of the whole component. By attention to detail in design and the adjustment of the dimensions in relation to the local concentrations of stress, the component can be made lighter without any danger of failure. Out of the many factors which affect fatigue resistance, local concentrations of stress is the most important. This alone renders design against fatigue more difficult than design for static stress. At the same time fatigue resistance is affected by the size of a component, by the relative magnitude of static and fluctuating loads and by the number of load reversals.

1.2 Scope of the thesis : Fatigue covers an exceedingly wide field. It may cause failure in simple plain test specimens, in parts containing some form of discontinuity where stress concentration is present. It is influenced by the loading pattern that is imposed, which can vary in the frequency and magnitude of loads that are applied. Small wonder is that this wide range of variables coupled with the occasional disastrous failure in service have inspired a vast amount of researches and investigations into fatigue behavior and much knowledge has been gained though much remains to be learnt.

Broadly two separate and distinct fields of knowledge are being acquired-firstly theories of fatigue are being studied and secondly experimental data are being accumulated on the fatigue strength of particular specimens . A precise association between theory and practice has generally eluded a solution and until this is achieved both fields of research must necessarily be pursued. In fact, there seems little likelihood that this link can ever be made for fatigue, like other properties of materials can only be assessed by the practical test. However, there is a half way stage in linking theory with practice, which can be regarded as an engineer's solution for application to design problems.

To obtain a quantitative measure of resistance to fatigue it is necessary to carry out tests under controlled conditions and for this purpose a wide variety of fatigue testing machines are available. Many different methods of fatigue testing can be adopted, from laboratory tests on smooth specimens under the simplest stress conditions, to test on full scale components and structures under conditions simulating those occurring in practice. Tests on laboratory specimens are used primarily for determining the influence on fatigue resistance of such factors as alloy content, heat treatment or surface finish because the results can be obtained quickly and economically. Such tests can be made on smooth or notched specimens if necessary at a high or low temperature or under corrosive conditions. To provide data for design, fatigue tests on actual parts are usually more valuable. For this purpose, special testing facilities are sometimes required.

The usual objective in designing structures and structural elements to resist repeated stress is to make certain that repeated loads do not cause failure during the useful life of the structure. For this, the following types of data are likely to be required :

1. The effect of combined steady and alternating stress on the fatigue behavior of materials.
2. The effect of combined shear, bending or direct stresses on fatigue behavior.
3. The effect of stress gradients.
4. The effect of stress raisers, such as notches, fillets, threads, holes, riveted joints, welds, etc.
5. The effect of surface finish on fatigue behavior.
6. The effect of high and low temperature on fatigue strength.
7. The effect of size of structural elements.
8. The extent of permanent damage resulting from any given number of stress cycles.
9. The extent of variation in fatigue strength to be expected in any class of materials.

The fatigue properties of mild steels have been investigated very extensively particularly under rotating bending or repeated axial loading tests of small un-notched specimens by the various researchers. Many tests have sought particularly to establish values of fatigue limit under these conditions. There are fewer data for high stress, short lifetime conditions. There are relatively few data under conditions other than fully reversed loading.

1.3 : Objective of the thesis : This research was carried out by using a newly designed machine in which inclined loading was permitted. That means, direct tensile stress and flexural stress can be applied simultaneously to the test specimens. The following were the specific aims of this research work.

1. The effect of stress raisers such as different type of notches on fatigue life of mild steel shaft.
2. The effect of V-notch angle on life cycle of notched mild steel shaft.
3. The effect of combined bending and direct stress on fatigue life.

4. The effect of combined steady and alternating stress on the fatigue life of notched mild steel specimen.
5. The effect of combined steady and alternating stress on life cycle of plain mild steel specimen.
6. The effect of notch-root radius on failure cycle of 90°V-notched specimen.

Chapter 2
Literature survey

2.1. Historical back ground: This chapter includes a brief historical survey of fatigue of metals and a short account of the recent developments towards the understanding of the mechanism of fatigue.

Mechanical failures have caused many injuries and much financial loss. Failure due to repeated loading has accounted a significant number of these mechanical failures. Most of these are unexpected fractures. This challenges the engineers to improve design decisions involving fatigue.

Many approaches to fatigue design exist. They can be simple, inexpensive approaches or they may be extremely sophisticated and expensive. A more complete analysis may initially be more expensive but in the long run it may be the least expensive. Thus an important question in fatigue design is how complete the analysis should be.

Fatigue of materials is still only partly understood. To gain some general understanding, review of fatigue is necessary. This review shows a few basic ideas and indicates very briefly how they were developed by the efforts of many researchers.

The first major impact of failures due to repeated stresses involved the railway industry in the 1840s. It was observed that railroad axles failed regularly at shoulders [1]. The word fatigue was introduced in the 1840s and 1850s to describe failures occurring from repeated stresses. This word has continued as the normal description of fracture due to repeated stresses.

In Germany, during the 1850s and 1860s, August Wohler performed many laboratory fatigue tests under repeated stresses. These experiments were related to the railway axle failures and are considered to be the first systematic investigation of fatigue. Thus, Wohler is called the father of systematic fatigue testing. He showed from stress versus life diagrams how fatigue life decreased with higher stress amplitudes and, below a certain stress amplitude, the test specimens did not fracture. Thus Wohler introduced the concept of the stress-life diagram and the fatigue limit. He pointed out that, for fatigue, the range of stresses is more important than the maximum stress [2].

During the 1870s and 1890s, some other researchers expanded Wohler's classical work. Gerber along with others investigated the influence of mean stress, and Goodman proposed a simplified theory concerning mean stresses.

In the 1900s the optical microscope was used to pursue the study of fatigue mechanisms. Localized slip lines and slip bands leading to the formation of micro cracks were observed. In the 1920s Gough and associates contributed much to the understanding of fatigue mechanisms. In 1920 Griffith [3] published the results of his theoretical calculations and experiments on brittle fracture using glass. He found the strength of glass depended on the size of microscopic cracks. If 'S' is the nominal stress at fracture and 'a' is the crack size at fracture the relation is $S\sqrt{a}$ = constant. By this classical pioneering work on the importance of cracks, Griffith became the father of fracture mechanics. In 1929-1930 Haigh [4] presented his rational explanation of the difference in the response of high tensile strength steel and mild steel to fatigue when notches are present. He used concepts of notch strain analysis and self stresses that were later fully developed by others.

During the 1930s an important practical advance was achieved by the introduction of shot peening in the automobile industry, where fatigue failures of springs and axles had been common; they then became rare. Almen [5] explained correctly the spectacular improvements by compressive stresses produced in the surface layers of peenned parts and promoted the use of penning and other processes that produce beneficial self stresses. Horger [6] showed that surface rolling can prevent the growth of cracks. In 1937, Neuber [7] introduced stress gradient effects at notches and the elementary block concept which considers that the average stress over a small volume at the root of the notch is more important than the peak stress at the notch.

During world war II, the deliberate use of compressive residual stresses became common in the design of aircraft engines and armored vehicles. Many brittle fractures in welded tankers and in ships motivated substantial efforts and thinking concerning pre-existing defects in the form of cracks and the influence of stress concentrations.

In 1945, Miner [8] formulated a linear cumulative fatigue damage criterion, suggested by Palmgren [9] in 1924. This linear fatigue damage criterion is now recognized as the Palmgren-Miner law. It has been used extensively in fatigue design and, despite its many shortcomings, still remains an important tool in fatigue life predictions.

Electron microscopy opened new horizons to better understanding of basic fatigue mechanisms. Irwin [10] introduced the stress intensity factor , which has been accepted as the basis of linear elastic fracture mechanics and of fatigue crack growth life predictions. Paris [11] in the early 1960s showed that fatigue crack growth rate could best be described using the stress intensity factor range. In the late 1960s the catastrophic crashes of F-III aircraft were attributed to brittle fracture of members containing pre-existing flaws. These failures along with fatigue problems in other US Air Force planes laid the ground work for the requirements of using fracture mechanics concepts in the B-1 bomber developments program of the 1970s. This program included fatigue crack growth life considerations based on a pre-established detectable initial crack size.

In July, 1974, the US Air Force issued a rule which defines damage tolerance requirements for the design of new military aircraft. The use of fracture mechanics as a tool for fatigue was thus thoroughly established through practice and through regulations.

2.2 Present state of art for testing notched bar : From literature review, it was found that many attempts have been made to test the notched bar/sheet under cyclic loading. But, in most of the cases uniaxial cyclic loading was applied to test the specimen like tension - compression test and plain bending test. In some cases biaxial tension-torsion tests on round bar have been done.

J. Botsis and C. Huang [12] performed experimental studies on damage evolution. To facilitate experimental observations, the crack is made by inducing a 60° V- notch on to the mid-span of the specimen edge. Sheet type amorphous polystyrene was used as test specimen.

Tension fatigue experiments were conducted on an Instron Testing system in laboratory environment at ambient temperature. All experiments were performed under load controlled mode with sinusoidal wave form.

X.C. Yin and X.H. Liu [13] performed the test using three-points bending specimens and compact specimens. An extensometer fixture surrounding the compact specimen was used for displacement measurements. Stable and local unstable crack growth were observed.

J.Faleskog[14] tested large surface cracked plate specimens and on small compact tension specimens. Some of the surface cracked specimens were subjected to combined tension and bending in such a way that the loading was strongly non-proportional.

Hiizu Hyakutake, Terutoshi Hagio and Toshihiro Yamamoto [15] carried out tests on notched specimens of a glass-fiber for a wide range of notch-root radius and stress amplitudes using plane bending fatigue machine. The validity of the fracture criterion on the severity near the notch-root for the fatigue failure of notched FRP (fiber reinforced plastics) plates have been investigated.

Kazuaki Shiozawa, Seiichi Nishino and Keiichi Handa [16] conducted tests under the stress ratios (R") of o and -1 using specimens of 0.37% carbon steel coated with TiN by PVD (physical vapor deposition) and CVD (chemical vapor deposition) in order to study the influence of applied stress ratio on the fatigue strength using rotating bending fatigue testing machine.

Masahiro Jono and Atsushi Sugeta [17] performed load-controlled fatigue crack growth tests on two kinds of structural steels under constant-amplitude and repeated two-and three-step loading in the post yield region. The effects of load variation on the elastic plastic fatigue crack growth behavior were investigated.

There are some important fatigue design criteria on which fracture mechanics are established.

2.3 Fatigue design criteria : Criteria for fatigue design have evolved from so called infinite life to damage tolerance. Each of the successively developed criteria still has its place depending on the application.

2.3.1: Infinite-life design : Unlimited safety is the oldest criterion. It requires design stresses to be safely below the fatigue limit. For parts subject to many millions of almost uniform cycles like engine valve springs, this is still a good design criterion.

2.3.2: Safe-life design : The practice of designing for a finite life is known as safe-life design. Infinite life design was appropriate for the railroad axles that Wohler investigated but automobile designers learned to use parts that, if tested at the maximum expected stress or load, would last only some hundred thousands of cycles instead of many millions. The maximum load or stress in a suspension spring or in a reverse gear may never occur during the life of a car. Designing for finite life under such loads is quite satisfactory. It is used in many other industries, for instance in pressure vessel design and in jet engine design. The safe-life design must of course include a margin for the scatter of fatigue results and for other unknown factors. The calculation may be based on stress-life relation, strain-life relations or crack growth relations. Ball bearings and roller bearings are examples of safe life design. The ratings for such bearings are often given in terms of a reference load that 90 percent of all bearings are expected to withstand for a given lifetime for instance 3000 hours at 500 rpm or 90 million revolutions. For different loads or lives or for different probabilities of failure the bearing manufactures list conversion formulas. They do not list any load for infinite life nor for zero probability of failure at any life. The margin for safety in safe-life design may be taken in terms of life, in terms of load or by specifying that both margins must be satisfied.

2.3.3 Fail-Safe design : Fail-safe fatigue design criteria were developed by aircraft engineers. They could not tolerate the added weight required by large safety factors nor the danger to lives implied by small safety factors. Fail-safe design recognizes that fatigue cracks may occur in the structure so that cracks will not lead to failure of the structure before they are detected and repaired. This philosophy originally applied mainly to air frames (wings, fuselages, control surfaces). It is now also used in other applications. A landing gear is not fail-safe but is designed for a safe-life.

2.3.4 Damage tolerant design : This philosophy is a refinement of the fail-safe philosophy. It assumes that cracks will exist caused either by processing or by fatigue. Fracture mechanics analyses and tests to check whether such cracks will grow large enough to produce failure before they are sure to be detected by periodic inspection. This Philosophy looks for materials with slow crack growth and high fracture toughness. Damage tolerant design has been specified by the US Air Force in some contracts. In pressure vessel design "leak before burst" is an expression of this Philosophy.

Chapter 3

Experimental Set-up and Test Procedure

Introduction : To obtain a quantitative measure of resistance to fatigue it is necessary to carry out tests under controlled conditions and for this purpose a wide variety of fatigue testing machines are available. To provide data for design, fatigue tests on actual parts are usually more valuable. For this purpose special testing facilities are sometimes required.

During a fatigue test the stress cycle is usually maintained constant so that the applied stress condition can be written $S_m \pm S_a$, where S_m is the static or mean stress, and S_a is the alternating stress. The positive sign is used to denote a tensile stress and the negative sign a compressive stress. Some of the possible combinations of S_m and S_a are illustrated in Fig. 3a to 3d. When S_m=0 (fig 3a), the maximum tensile stress is equal to the maximum compressive stress and this is called a completely reversed stress. When $S_m = S_a$ (Fig. 3b), the minimum stress of the cycle is zero and this is termed as pulsating or repeated tensile stress. Any other combination is known as a fluctuating stress which may be a fluctuating tensile stress (Fig. 3a), a fluctuating compressive stress or may fluctuate from a tensile to a compressive value (Fig. 3d).

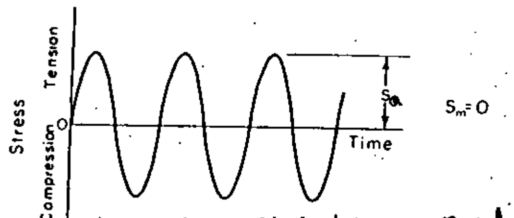


Fig 3a: completely reversed stress.

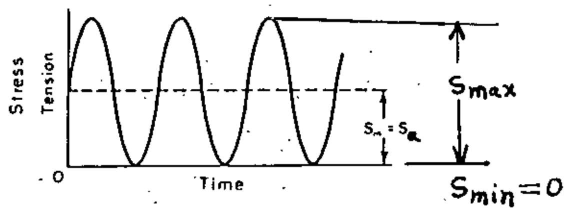


Fig. 3b: Repeated tensile stress.

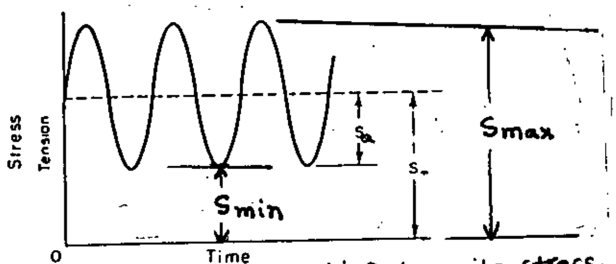


Fig. 3c : Fluctuating tensile stress.

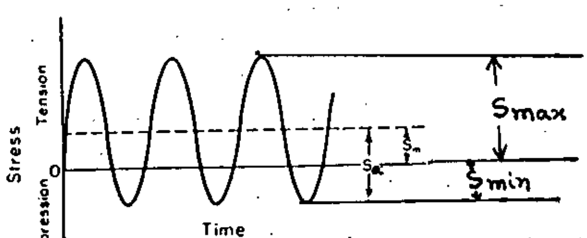


Fig. 3d: Fluctuating from a tensile to a compressive stress.

In some applications, components are subjected to bending stress in addition to tensile stress. The fluctuation of these two types of stress may cause different mode of stress and the ratio of these two stresses should affect failure of components differently. Therefore a different term " flexural stress ratio" is used during the investigations, which is defined as follows.

Flexure stress ratio (R) - The flexural stress ratio 'R' is defined as the ratio of total flexure stress (S_f) to total tensile stress (S_t) in case of inclined loading, that is, $R = S_f/S_t$.

Some other notations and definitions used are as follows.

Steady tensile stress component (S_{t0}): It is the tensile stress component of steady inclined load on the specimen.

Steady flexural stress component (S_{f0}): It is the flexural stress component of steady inclined load on the specimen.

Alternating tensile stress component (S_{ta}): It is the tensile stress component of fluctuating inclined load on the specimen.

Alternating flexural stress component (S_{fa}): It is the flexural stress component of fluctuating inclined load on the specimen.

Total flexural stress (S_f): It is the algebraic sum of steady flexural stress component and alternating flexural stress component, that is, $S_f = S_{f0} \pm S_{fa}$. In the total flexural stress calculation maximum value of S_f, that is, $S_f = S_{f0} + S_{fa}$ is used.

Total tensile stress (S_t): It is the algebraic sum of steady tensile stress component and alternating tensile stress component, that is, $S_t = S_{t0} \pm S_{ta}$. In the calculation of maximum tensile stress, $S_t = S_{t0} + S_{ta}$ is used.

Total mean stress (S_m): It is the algebraic sum of steady tensile stress component and steady flexural stress component, that is, $S_m = S_{t0} \pm S_{f0}$. In total mean stress calculation $S_m = S_{t0} + S_{f0}$ is used which is the stress in the outer fibre at the notch section.

Total alternating stress (S_a): It is the algebraic sum of alternating tensile stress component and alternating flexural stress component, that is, $S_a = S_{ta} \pm S_{fa}$. In total alternating stress calculation, $S_a = S_{ta} + S_{fa}$ is used which is the maximum alternating stress in the outer fibre at the notch section.

Maximum stress (S_{max}): It is the sum of the total mean stress and total alternating stress, that is, $S_{max} = S_m + S_a$.

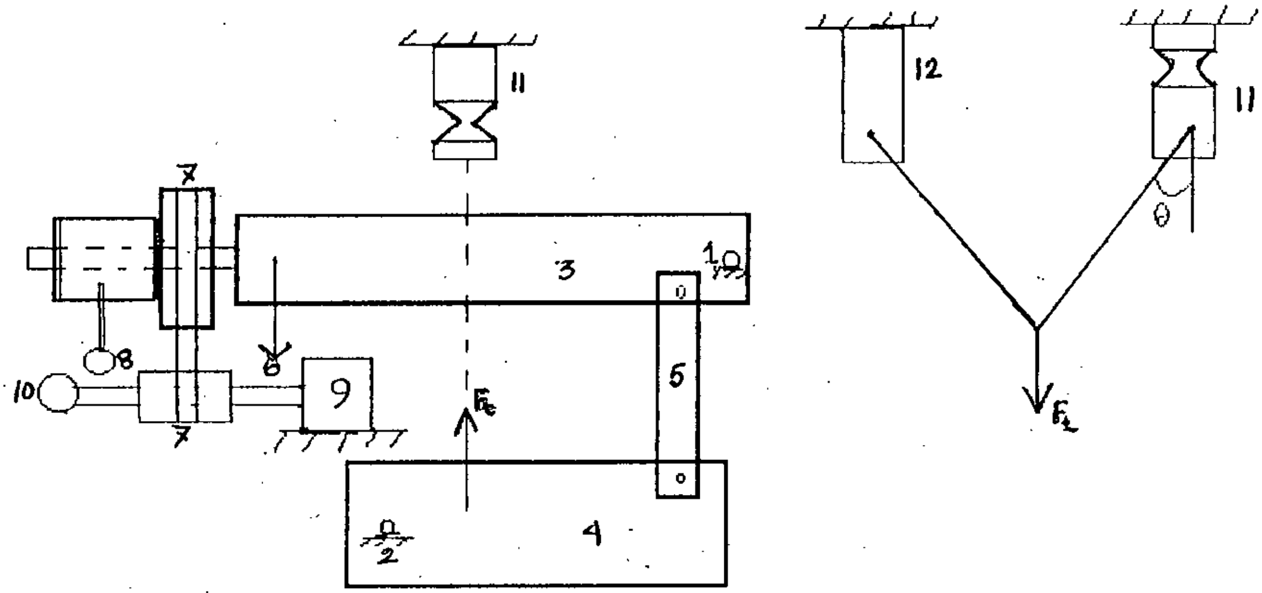
Minimum stress (S_{min}): It is the difference between total mean stress and total alternating stress, that is, $S_{min} = S_m - S_a$.

Alternating stress ratio (R'): It is the ratio of total alternating stress to total mean stress, that is, $R' = S_a / S_m$.

Stress ratio (R''): It is the ratio of minimum stress to maximum stress, that is, $R'' = S_{min} / S_{max}$.

3.1 Experimental set-up : Fatigue testing machines can be classified by the type of straining action that is applied to the specimen, caused by direct or axial stress, bending, torsion, and combined stress. In direct stress machines the specimen is subjected to tension-compression stresses. Bending fatigue machines are generally of two types, one in which the specimen is rotated and the other in which it is repeatedly bent to and fro. The latter type is generally known as reversed bending machine. In reversed bending machine bending moment is applied to the specimen in such a way that tensile stress and compressive stress are produced at the test section repeatedly. But the bending fatigue machine in which all the experiments have been done was such that inclined loading to the specimen can be applied varying the loading angle from 0° to 45°. To do the experiment there should be a constant tension in the specimen. In the experiments the specimen is bent repeatedly to and fro in the tension-tension fluctuating loading condition. No compressive stress was developed at the test section of the specimen.

Fig. 3e and 3e' are the schematic diagram to show the arrangement of the components of the testing machine and test section.



1. Pinned support of upper load arm
2. Pinned support of lower load arm
3. Upper load arm
4. lower load arm
5. Load transfer assembly
6 Dead weight
7. Pulley
8. Centrifugal mass
9. Motor
10. RPM Counter
11. Test specimen
12. Dummy specimen

Fig. 3e: Schematic diagram to show the arrangement of the components of the testing machine.

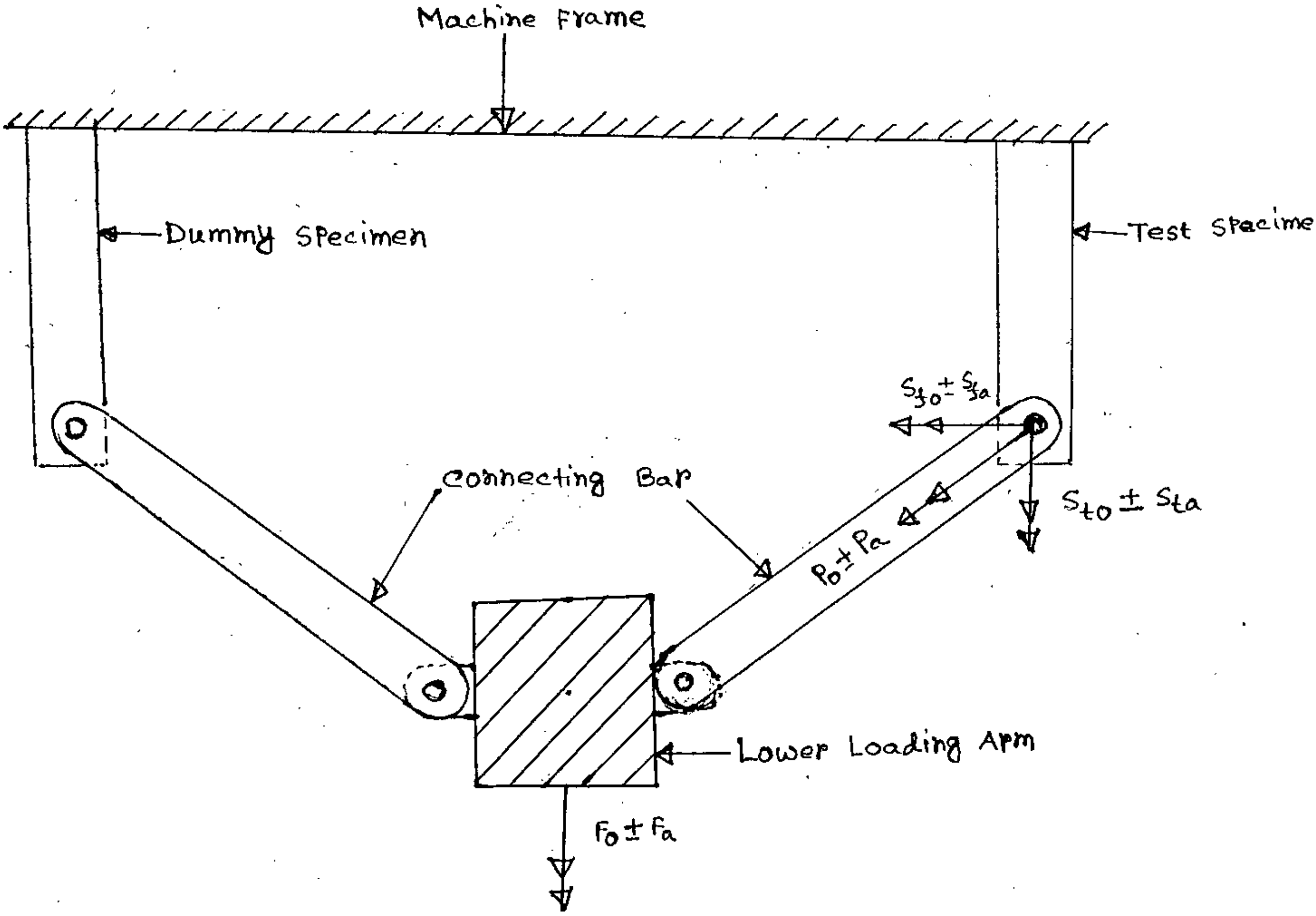


Fig 3e': Test Section

The experimental set up comprises of the following.

1. Upper load arm.
2. Lower load arm with loading assembly.
3. Motor-pulley arrangement.
4. Measuring instruments.

Upper load arm : The upper load arm is a hollow box beam of rectangular cross section hinged at one end. At the other end of this arm there is a shaft welded to it. A pulley with a bush is fitted on the shaft with two bearings such that the pulley with the bush can rotate. Another shaft is fitted with the bush at one end and there is arrangement to fix weight (centrifugal mass) at the other end of this shaft.

When the pulley and bush is rotated, the rotating mass produces a centrifugal force, which causes an up and down motion of the upper load arm. When this up and down motion is restricted through some connecting member, it creates equivalent load on connecting member. There is a provision to change the load on the specimen by fixing dead weight in the upper load arm at some point of the upper load arm. This dead load can be varied .Centrifugal force can be adjusted by changing the centrifugal mass or by changing the radius of rotation of the mass.

Lower load arm with loading assembly : The lower load arm is pinned at one end while the other end is connected to the upper load arm. Load from upper arm is transmitted to lower arm through a connecting member made of mild steel sheet in the form of U-shaped channel. When the centrifugal mass rotates, the upper load arm is subjected to an up and down motion which is transmitted to the lower load arm through the connection. The load on lower arm is transmitted to the specimen through another set of connecting bars which connect the lower arm to the test specimen. Force due to dead weight and centrifugal action are magnified through lever action of the upper load arm and passes to the lower load arm through the connecting member. Load transmitted to the lower arm is again magnified due to lever action. This magnified load is transmitted to the test specimen from a point of the lower load arm through connecting bars. Test specimen can be set rigidly with the support of the machine in any desired direction. To run the machine, an initial tension is applied to the test specimen. When the machine runs, the centrifugal force along with dead weight on upper load arm, mass of the load arms and loading assembly make a constant amplitude fluctuating or alternating load on the specimen. By changing the centrifugal mass and radius of rotating mass, dead weight, loading angle - total mean stress (S_m), total alternating stress (S_a), total flexural stress and total tensile stress can be changed.

Motor pulley arrangement : An AC motor of 1.0 hp, is installed at the base of the machine for rotating the centrifugal mass . Power is transmitted from the motor shaft to the rotating mass through V-belts and pulley arrangement. The rpm of the rotating mass was kept constant. By changing the size of the pulley, the rpm of the rotating mass can also be adjusted as required.

Measuring Instruments : To measure the load and the cycle of the loading the following instruments were used during the experiment.

1. RPM Counter
2. Strain gage
3. Strain meter
4. FFT analyzer
5. Bevel Protector

3.2 Calibration of the testing machine: There are no international standards of accuracy or methods of calibration for fatigue testing machines, and, in this respect fatigue testing methods fall short of static methods. This is unsatisfactory because the dynamic measurement and errors are more likely to be made. Therefore care was taken to measure the actual dynamic load applied on the test specimen.

The following is the procedure followed in calculating the dynamic load. The connecting bar through which the test specimen is loaded is such that its deflection is wholly elastic when the maximum stress of the fatigue testing machine is applied. Strain gages were used to measure the extension of this connecting bar under load. A strain meter was used to measure the strain of the bar when the bar was in tension. The extension of the bar and the corresponding applied load was measured by using an universal testing machine before fitting the bar in the testing machine as the connecting bar. During trial test run of the machine, it was observed that due to the high frequency of loading, the indicator of strain meter failed to indicate the peak of the loading. Therefore , it was concluded that strain meter can not be used to indicate the peak load applied to the test specimen. FFT analyzer was used to measure the voltage variation of the strain gage. With FFT analyzer, the peaks of voltage variation during fluctuating load can easily be measured if it is correctly calibrated. The strain gage on the connecting bar was calibrated using the universal testing machine.

Load-strain relation : By using universal testing machine, the relation between load and strain of connecting bar as in Fig. 3f was obtained.

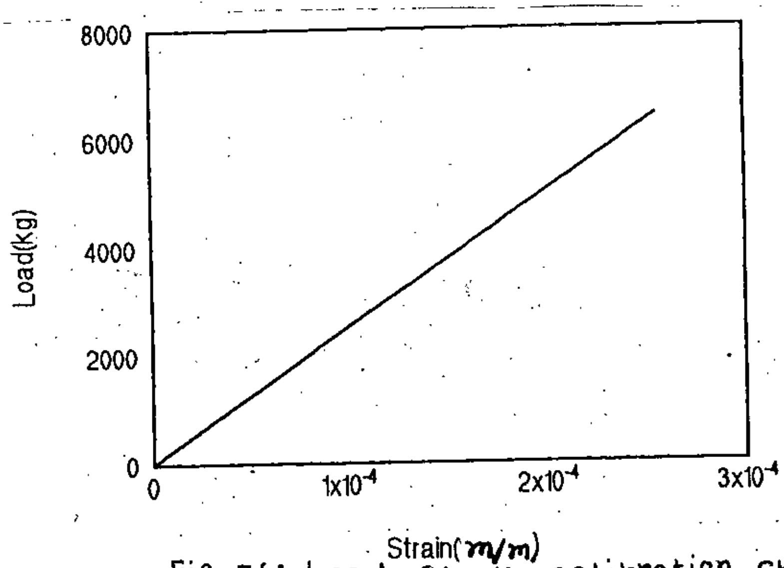


Fig. 3f: Load-Strain calibration curve.

Volt - Strain relation : As the change of voltage with load was very small, it was very difficult to correlate voltage change and strain of connecting bar directly by using universal testing machine. To solve the problem, the same strain gage which was used to measure the dynamic strain of connecting bar was fastened to another piece of material from which correlation between volt and strain can be easily found out. The change of mili volt with strain change was found by using strain meter and FFT analyzer. Again the volt-strain shows a straight-line relation (Fig. 3g).

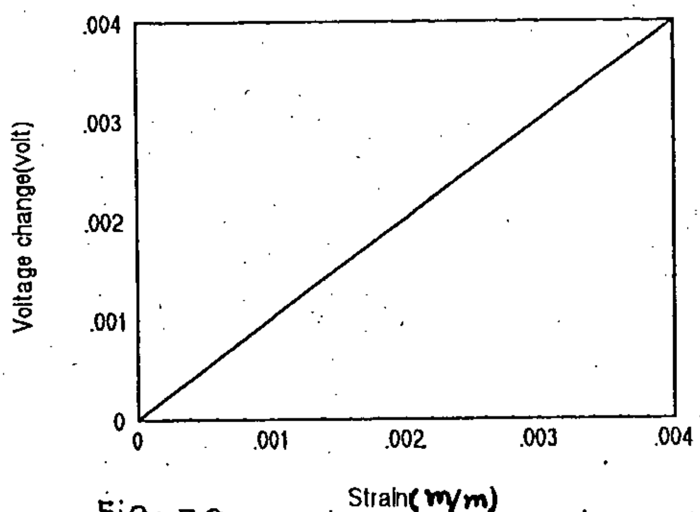


Fig. 3g: Volt - strain calibration curve.

Load-volt relation : From the above two relations it is clear that change of voltage with load is also a straight line just like as load with strain (Fig. 3h).

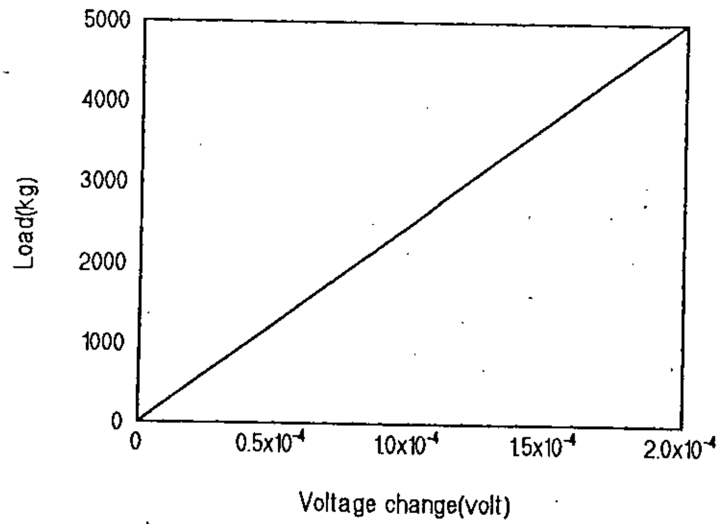


Fig. 3h: Load- volt calibration curve

Using FFT analyzer, the wave form of the dynamic strain was displayed when the fatigue machine was operated. Then the wave form of dynamic strain was printed in heat sensitive paper for the analysis. The dynamic load can then be calculated using the result of a prior static calibration. The advantage of the strain gage method is that the response of the strain gauge is independent of frequency and an accuracy of ± 1% of the maximum load range can be achieved. The following are some example of voltage wave form generated by FFT analyzer to measure the dynamic load.

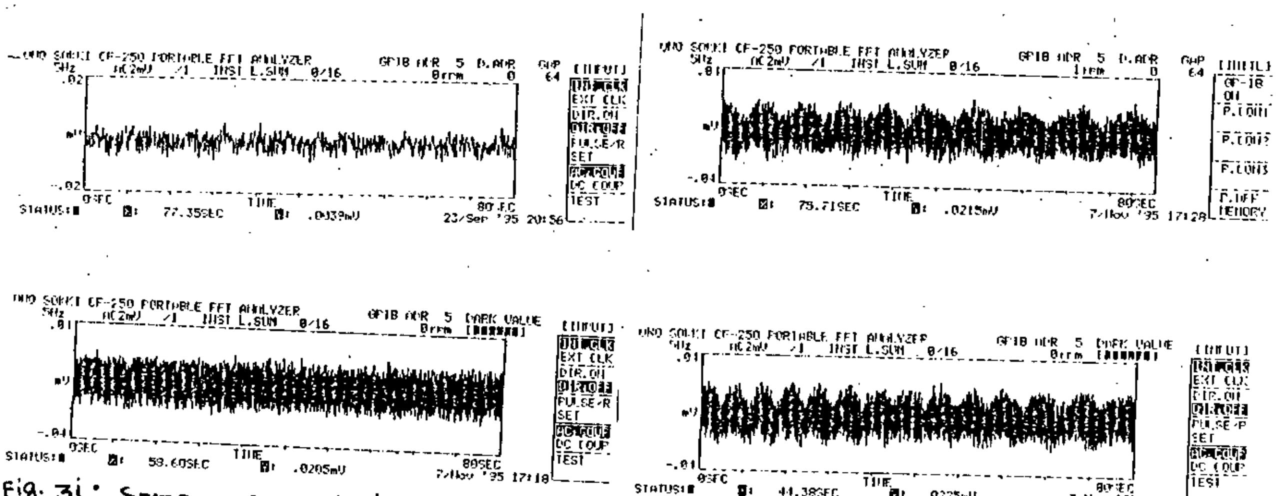


Fig. 3i: Some representative voltage wave form generated by FFT analyzer to measure the dynamic load.

3.4 Test specimens : The following are the test specimens with dimensions which were used for fatigue testing.

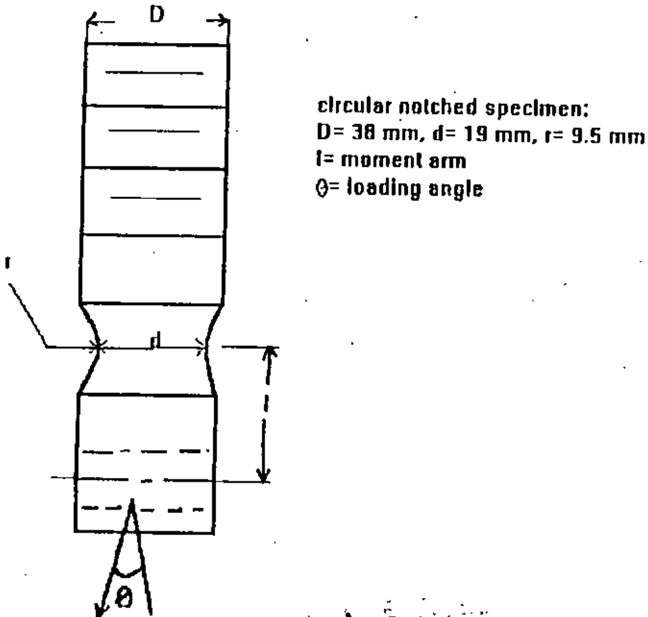


Fig. 3j: circular notched specimen

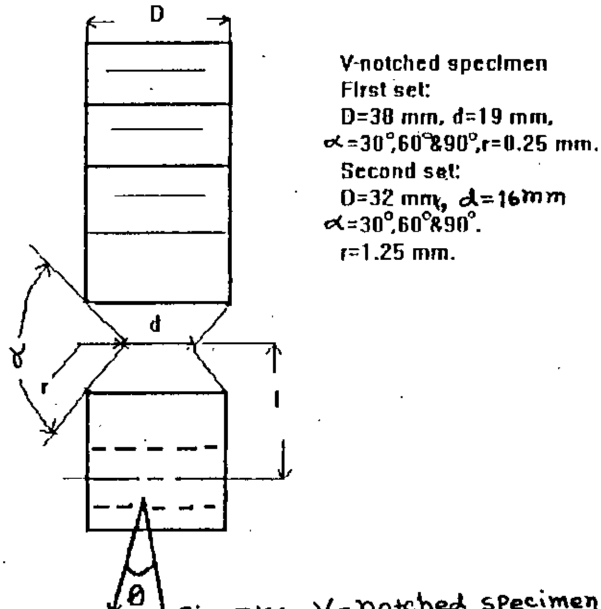


Fig. 3K: V-notched specimen.

One end of the specimen was threaded to fasten it to the support of the machine (Fig. 3j and 3k). There was a hole at the other end of the specimen for loading by bolting it to the connecting bar assembly. V-notch or circular notch was located at a certain distance from hole centre. Notch angles were selected as 30°,60° and 90°.

All the specimens were made on turning and drilling machines. No special finishing was used at test section. Notch-root radius was varied between 0.25 mm to 2.00 mm arbitrarily. The values of k_t were ranged from 2.07 to 3.63. The k_t values of specimens were found by using following equation [18].

$$K_t = K_0 - (2/\pi)\,(K_0 - 1)\tan^{-1}\{A\,(\rho/d)^B\}$$

where, K_t = stress concentration factor of general notches,

K_0 = stress concentration factor of notch in semi-infinite plate,

d = diameter at the notch section,

ρ = notch-root radius,

α = flank angle of V-notch,

A,B = coefficients,

Material and properties of specimen : The composition and mechanical properties of the specimens have been found experimentally and presented in the table A - 4. The material was collected from the local market commonly known as bright steel.

3.5 Test Procedure : At the beginning of experiments, the initial tension was measured from the strain in the connecting bar from which total mean stress (Sm) at the test section was calculated. During the running of the machine the dynamic strain was measured in terms of voltage wave form by using FFT analyzer. It was found that the wave form was almost same throughout the experiment (up to the time of fracture of a specimen). From the calculation of dynamic load, total alternating stress (S_a) was calculated at the test section. For every setting of specimen, loading angle was known and moment arm was also known. So, from the known dimension of specimen the value of total flexural stress and that of total tensile stress were calculated. When the loading condition was such that the stress in the specimen was not uniform, the result was presented in terms of nominal stress, which is the stress calculated by simple theory without taking into account the variations in stress conditions caused by geometrical discontinuities such as holes, grooves and fillets. The experiments were carried out in the higher stress level taking both notched and plain specimen. Initially two types of notch were selected to check the sensitivity. One was V-notch and another was circular notch. Both specimens were of mild steel shaft of D = 38 mm, d = 19 mm. The circular notched shaft had a notch of radius of 9.5 mm. Three angles of V-notch were taken as 30°, 60° and 90°. In every case notch-root radius was selected arbitrarily as 0.25 mm. One of the objectives of the experiments was to compare the results of notched mild steel shaft with those of plain mild steel shaft. From a trial run, a shaft of d = 19 mm was selected as a plain specimen. But the specimen did not fail even after 0.5 million cycles. As the frequency of loading was low, a long time was needed to fail such specimen. So a new set of specimens were chosen for the test. The dimensions of the specimen were, D = 32 mm d = 16 mm. Three notch angles, 30°, 60° and 90° were used again. Three maximum stress levels were selected, S_{max} = 503.51 MPa, 470.40 MPa and 435.40 MPa. The levels of S_{max} were selected in the range of 70 percent to 80 percent of ultimate tensile strength. This new set of specimens were also used to test the effect of size of the specimen on fatigue life.

Chapter 4
Results and discussions

Sensitivity test of circular notched and V-notched specimens :

The first objective of the experiments was to find out the effect of stress raisers such as different types of notch on fatigue life of mild steel shaft. For that, two types of notch were selected. One was circular notch and the other was V-notch. Three notch angles of V-notch were chosen - 30°,60° and 90°. The dimensions of circular notch were, D=38 mm, d=19 mm, and r=9.5 mm. V-notched specimens had the dimensions, D=38 mm, d=19 mm $\alpha=30^\circ$, 60° and 90°, and r=0.25 mm. All the dimensions except notch angle and notch root radius were selected considering the loading capacity of the testing machine. Notch angle and notch-root radius of V-notched specimens were selected arbitrarily. The effect of different types of notch on fatigue life were measured by the notch sensitivity of the specimens. That notch which failed at small number of repetition of stress was considered more sensitive. Initially four specimens were tested, taking S_{max} of 488.18 MPa which was about 79 percent of ultimate tensile strength. For every specimen, total mean stress, total alternating stress, total flexural stress and total tensile stress were kept constant. Circular notched specimen was failed at maximum number of stress cycles (92005 cycles), whereas 90° V-notched mild steel shaft was failed at a minimum number of stress cycles (22370 cycles). 60° and 30° V-notched specimen were failed at 27010 and 33007 cycles respectively. From the experimental results it can be concluded that 90° V-notch is the most sensitive and circular notch is the least sensitive notch to fracture at maximum stress level of 488.18 MPa for inclined loading under the same loading conditions. As one of the objectives of the investigation was to find out the more sensitive notch, circular notch was discarded due to its low notch sensitivity in comparison with 30°, 60° and 90° V-notches. For the explanation of why a circular notch is less sensitive, the effect of notch is to be analysed. Notch has two effects on the properties of material :

1). It concentrates the stress at the periphery of the notch;
2) It reduces the elongation of the specimen - an effect due to the shoulders of the notch.

The concentration of the stress tends to reduce the strength of the material while the reduction in the elongation increases its strength [19]. Consequently a specimen which is ductile will show an increase in strength since the elongation will be less; and a specimen of brittle material will show a reduction both in strength and in elongation.

In fatigue failure, the specimens failed almost in brittle mode. So, for circular notched specimen and V-notched specimens, stress concentration factor is the dominating one than reduction in elongation. The stress concentration factor of V-notched specimens were reasonably higher than circular notched specimen. So logically V-notched should be more sensitive than circular notch as supported by the experimental results.

The effect of V-notch angles and the size of the specimens on the fatigue life: Figure-1 shows the effect of notch angle on fatigue life of notched mild steel shaft for the maximum stress of 488.18 MPa 454.79 MPa and 391.58 MPa with $D = 38$ mm $d = 19$ mm and $r = 0.25$ mm . The curves show that notch sensitivity increases with increase of notch angle in the range of 30° to 90°.

A new set of specimens were chosen with different diameters and notch root radius to test the notch sensitivity with notch angle of 30°, 60° and 90°, as before. The dimensions of the specimen were, $D = 32$ mm, $d = 16$ mm and r=1.25 mm. As the experiments were done at higher stress level, maximum stress levels were selected in the range of 71% to 82% of ultimate tensile strength. Three S_{max} levels were chosen - 503.51 MPa (82% of UTS) 470.40 MPa (76% of UTS) and 435.40 MPa (71% of UTS). For each S_{max} level, three specimens were tested. Every sub set of specimens (three specimens) were tested keeping the total mean stress, total alternating stress, total flexural stress and total tensile stress constant.

Fig. 2 shows the variation of fatigue life with notch angle. This figure is similar in nature with figure 1. In this figure, it is also noted that 90° V-notch is the most sensitive and 30° V-notch is the least sensitive notch to fracture. Comparing Fig. 1 with Fig. 2, an interesting and remarkable conclusion can be drawn. Though one of the maximum stress level selected in Fig. 1 was 391.58 MPa and that in Fig. 2 was 435.40 MPa, specimens in Fig. 1 was failed at a lower number of stress cycles than the one in Fig. 2.

It can thus be concluded that small diameter specimen is less sensitive to fracture than big diameter specimen subjected to flexural stress. If two specimens of different diameters are tested in bending at the same nominal stress at the outer surface of notch root, they will have different stress gradients. The consequence of this is that a larger volume of material is subjected to a stress nearly as high as that of the outer surface stress for the specimen of larger diameter [43]. For this, a higher stress is produced in the big diameter specimen throughout the notch section in comparison to small diameter specimen. This causes the big diameter specimen to fail earlier than smaller one.

Fig. 3 shows the variation of fatigue life with different maximum stress for three different notch angles - 30°, 60° and 90°. For each type of notch angle the value of S_{max} = 391.58 MPa, 454.79 MPa and 488.18 MPa $R' = 0.47$, 0.46 and 0.41; $R = 7.46$, 9.96 and 11.19 and S_{min} = 142.94, 161.49 and 205.10 MPa were selected. It is observed from Fig. 3 that the curves for different notch angles show almost the same pattern. But curve for 60°-V lies above 90°-V curve and 30°-V curve lies above 60°-V curve.
It is further proved that for any value of S_{max}, R, R', R'' and for same type of loading condition 90° V- is the most sensitive notch than 60°-V or 30°V-notch. From Fig. 3, it is also observed that for each notch angle, fatigue life increases at a higher rate in the lower range of maximum stress level than upper range of maximum stress level, provided that, in every case, notch-root radius is kept constant to $r = 0.25$ mm.

Fig. 4 is the S_{min} versus N_f plot for different V-notched specimens, 90°-V shows the maximum notch sensitivity to fracture again.

Fig. 5 and Fig. 3 are similar. The difference is that, in Fig. 5, the values of S_{max}, R, R' and R'' are different from the previous one (Fig. 3). Another difference is that, in this experiment the used specimens were of $D = 32$ mm, $d = 16$ mm and r=1.25 mm. These two figures show almost similar nature with only difference in the decrease rate of sensitivity. From Fig. 5 it is clear that the increase rate of life cycle of different type of notch with decrease of maximum stress is much faster than that of Fig. 3. This also indicates that small specimen shows less notch sensitivity to fracture than big one.

The results for specimens of D = 32 mm, d = 16 mm, r = 1.25 mm are shown in Figs. 4 and 6. In Fig. 6, the notch sensitivity to fracture is seen to be less than Fig. 4 for the different values of minimum stress. This fact also indicates that small diameter specimen is less sensitive to fracture than that of big diameter specimen.

The effect of combined bending and direct stress on fatigue life :

As 90° V-notch was found to be the most sensitive to fracture, the remaining tests were carried out using V-notch angle of 90°. The effect of notch-root radius on fatigue life was examined taking the notch-root radius in the range of 1.25 mm to 2.00 mm. All other notched specimens had root radius of 1.25 mm.

Fig. 7 shows the variation of survival life with varying R. It shows that higher the value of R, higher is the fatigue life of 90° V-notch. The rate of increase of life increases with the increase of the value of R. For Fig. 7, 8 and 9 maximum stress of 503.51 MPa and alternating stress ratio of 0.4611 were used.

From Fig. 8, it is seen that the increase of total flexural stress increases the life, that is the notch sensitivity decreases with increase of total flexural stress. The rate of increase of life increases with increase of total flexural stress.

Fig. 9 shows that the higher the tensile stress, the lower is the life of the specimen that is, the notch sensitivity increases with increase of total tensile stress. The curve also shows that the steepness of the curve is greater in the higher range of tensile stress than in it's lower range. This means, the higher the tensile stress, the higher is the sensitivity of notch to fracture.

Keeping the maximum stress constant at 503.51 MPa the alternating stress ratio was changed to R=0.4215 in Fig. 10 to 12. These figures are almost similar in nature to those in Fig.. 7, 8 and 9. The only difference is that the rate of increase of life is almost constant for the increase of flexural stress and decrease of tensile stress (Fig. 11 and 12).

Fig. 13 shows the effect of R on N_f for S_{max} of 435.40 MPa and R' = 0.693. It is clear form the curve that at lower value of R, the rate of increase of life cycle is greater than that at higher value of R. This indicates that notch sensitivity decreases at a higher rate in the lower range of R whereas in higher range of R, notch sensitivity decreases at a slower rate.

From Fig. 14, it is found that, the more the flexural stress the less is the sensitivity of notch. But the most noticeable fact is that at higher flexural stress level, notch sensitivity decreases with increase of flexural stress at a rate which is much smaller than that at lower flexural stress level. Fig. 15 shows the variation of N_f with S_t. This curve shows that, the more the total tensile stress the more is the sensitivity of the notch.

From Fig. 9 and 12, it is seen that notch sensitivity increases with increase of tensile stress almost at constant rate. However, Fig. 15 shows that notch sensitivity decreases at a higher rate at higher tensile stress level, whereas, notch sensitivity decreases at a lower rate at lower tensile stress level. From the comparison of Fig. 7 to 15 an important conclusion can be made. The conclusion is that, in both lower and higher S_{max} level, for any value of R, total tensile stress effect is damaging than total flexural stress to fracture. It can also be concluded that at higher maximum stress level, tensile stress is more damaging than flexural stress.

The following is the explanation for above observation. Tensile stress is distributed uniformly throughout the notched section. But due to stress gradient flexural stress shows its maximum value at the root of the notch and decreases towards the centre of the specimen. For flexural stress only peak value at the outer fibre of notch is taken. The average value of flexural stress throughout the notched section is much smaller than peak value. That is why tensile stress has a more damaging effect than flexural stress.

The effect of total mean stress and total alternating stress on life cycle : To test the effect of total mean stress and total alternative stress on fatigue life of mild steel specimen, three specimens of 90° V of 1.25 mm root radius were tested at constant maximum stress of 470.40 MPa. The values of R' were taken as 0.62, 0.52 and 0.42 and the values of R″ were 0.235, 0.317 and 0.40 respectively.

Variation in life with R' is shown in Fig. 16. It is seen that notch sensitivity increases with increase of R'. S_a versus N_f and S_m versus N_f plots are shown in the same graph (Fig. 17). Fig. 17 clearly indicates that total alternating stress (S_a) has the damaging effect than total mean stress (S_m) for the fatigue failure. That means, life decreases with increase of alternating stress whereas life increases with increase of mean stress. Fig. 18 shows the variation of fatigue life at different minimum stress. It is observed from Fig. 18, that life increases with the increase of S_{min}. Fig. 19 is R'' versus N_f plot. This is similar to S_{min} versus N_f curve (Fig. 18). From Fig. 16 and 17 it can be concluded that for constant S_{max} and constant R, total alternating stress is more damaging than total mean stress to fracture whatever the value of S_{min} and R'' may be.

Fig. 20 to 23 are for plain specimens of d = 16 mm. For these experiments a higher value of S_{max} of 523.53 MPa was used (85 percent of UTS). The values of S_{min} were between 13.51 and 140.77 MPa; R'' between 0.026 and 0.269; R' between 0.58 and 0.95 were selected. R was kept constant at 6.3976. Fig. 20 is R versus N_f curve which is similar to that in Fig. 16. Fig. 16 is for notched specimens whereas Fig. 20 is for un-notched specimens. From Fig. 16 and 20, it is seen that, life of plain specimen increases at a greater rate than that of notched specimen. Fig. 21 is similar to Fig. 17. The difference is that both the curves S_a versus N_f and S_m versus N_f in Fig. 21 show the flat nature in comparison to that in Fig. 17. Fig. 22 is almost similar to Fig. 18. Fig. 22 shows slightly different nature than that in Fig. 18. Life increases at a higher rate in the higher range of minimum stress value than that in lower range of minimum stress value. Fig. 23 has a similar pattern to that in Fig. 19. Curve in Fig. 19 is more steeper than that in Fig. 23 showing the long life of plain specimen in comparison with 90° V-notched specimen. From Fig. 20 and 21, it is seen that for constant maximum stress and constant R, for any value of R'', total alternating stress has a damaging effect total mean stress. The nature of different curves are almost similar in both cases (notched and un-notched). Only the slopes of curves are different. The slope of curves for plain specimens are more flat than that of notched specimens.

The effect of notch-root radius on fatigue life : Four specimens were tested to find the effect of notch-root radius on fatigue life of 90°V-notched mild steel shaft. The specimens had the dimension of D=32 mm, d=16 mm and r=1.25, 1.5, 1.75 and 2.0 mm. For the test, maximum stress level was selected as 435.40 MPa. The value of R, R′ and R″ were kept constant at 5.98, 0.693 and 0.181 respectively. That means, all these four specimens were tested at the same loading conditions.

Fig. 24 represents the relation between the fatigue life and the notch-root radius. Up to a certain value of root radius, N_f increases at a constant rate maintaining a straight line relationship between r and N_f, But above that value of r, fatigue life increases at a higher rate than before. From the curve it is found that notch sensitivity decreases with the increase of notch root radius up to a certain value of root radius almost at a constant rate. But, thereafter, notch sensitivity decreases at a slower rate with increase of root radius.

CHAPTER 5
CONCLUSIONS AND RECOMMENDATIONS

CONCLUSIONS :

From the experimental evidences, following conclusions are drawn.

1. For constant maximum stress and alternating stress ratio, total tensile stress is damaging than total flexural stress to fracture for inclined loading.
2. In higher stress level, total tensile stress is more damaging than total flexural stress.
3. V-notch is more sensitive than circular notch to fracture.
4. 90° V-notch is the most sensitive and 30° V-notch is the least sensitive notch in the range of notch angle from 30° to 90° for every test value of maximum stress and for inclined loading condition.
5. For constant maximum stress and flexural stress ratio, total alternating stress is damaging than total mean stress to fracture.
6. Notch sensitivity decreases with increase of notch root radius. Up to a certain root radius, sensitivity decreases at higher rate, but after that level notch sensitivity decreases at slower rate.
7. For the test range of stress level, sensitivity of different notches decrease with decrease of stress level almost in the same pattern .
8. For same D/d, large diameter notched specimens are more sensitive to fracture than those of small diameter notched specimens.

RECOMMENDATIONS :

Following recommendations are made for further investigation.

1. It is recommended to select the notch angle from 20° to 150°. Because the value of k_t is almost same in the range of notch angle from 20° to 90°. Above 90° there is a remarkable change in k_t and these may cause different results. [18]
2. The notch-root radius is recommended to select in the range from 0 mm to 4 mm to observe the variation of fatigue life with different notch-root radius.
3. For same D/d, with different D and d , experiments can be done to test the effect of size of the specimens on fatigue life.
4. The frequency of the machine is very low (f = 4 Hz). This can be increased to test the fatigue limit of plain as well as notched specimens for the inclined loading.
5. Provision has to be made to stop the machine automatically after failure the specimen .
6. More data points are recommended, to plot graphs, to obtain the actual nature of the curves.

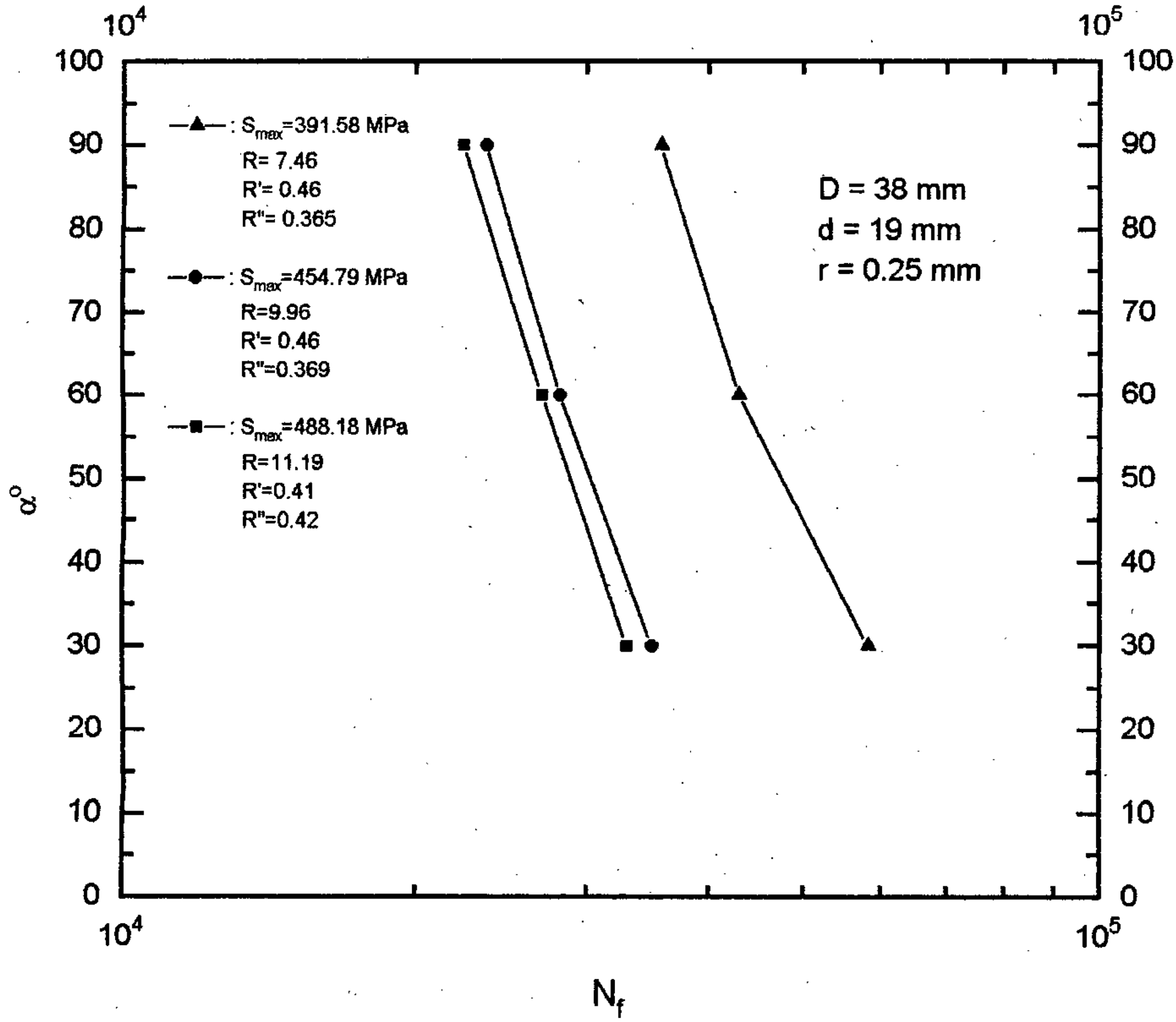


Figure 1 : Effect of notch angle on fatigue life of notched mild steel shaft for D=38mm, d=19mm and r=0.25mm for different maximum stress.

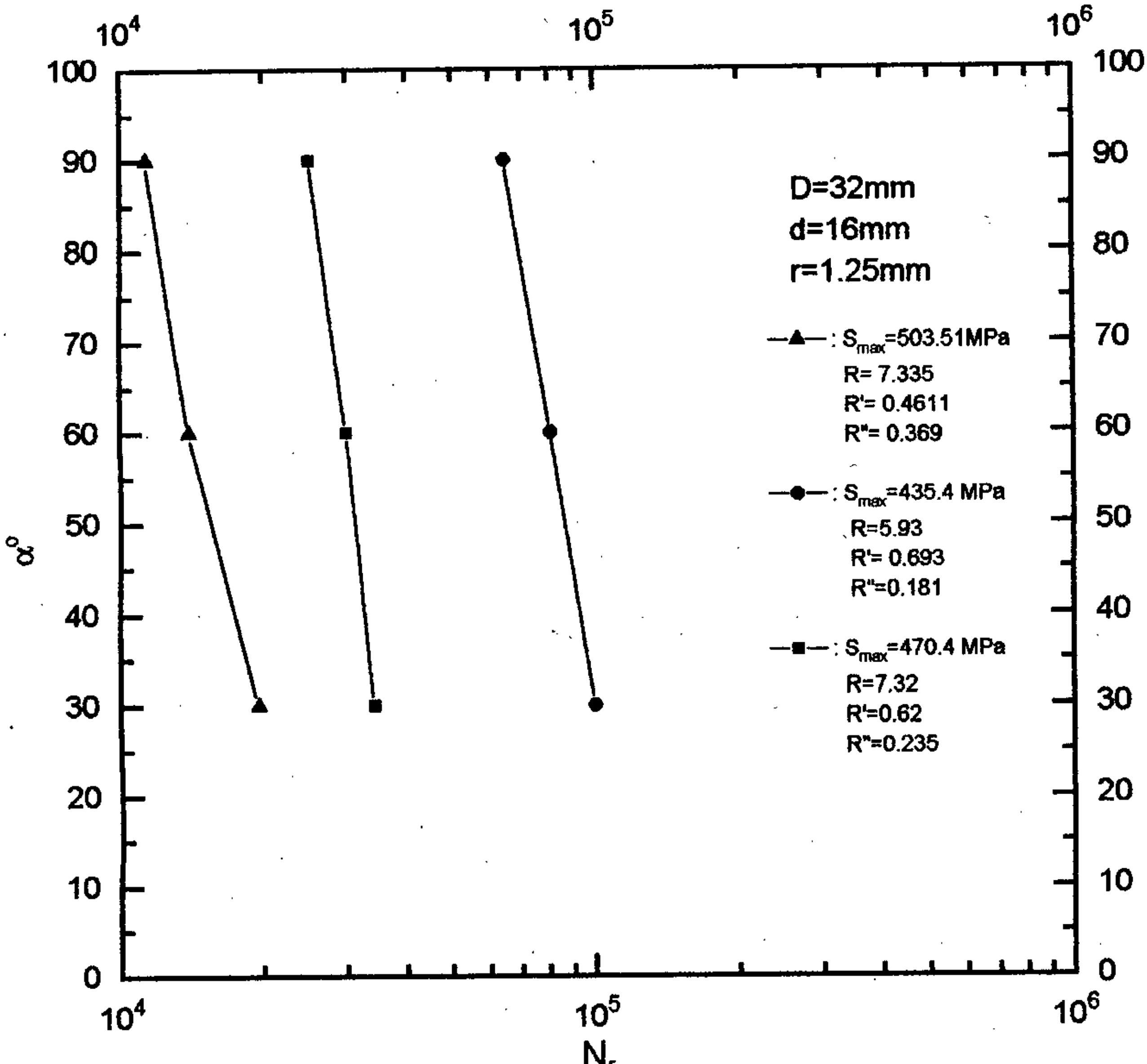


Figure 2 : Effect of notch angle on fatigue life of notched mild steel shaft for D=32mm, d=16mm and r=1.25mm for different maximum stress.

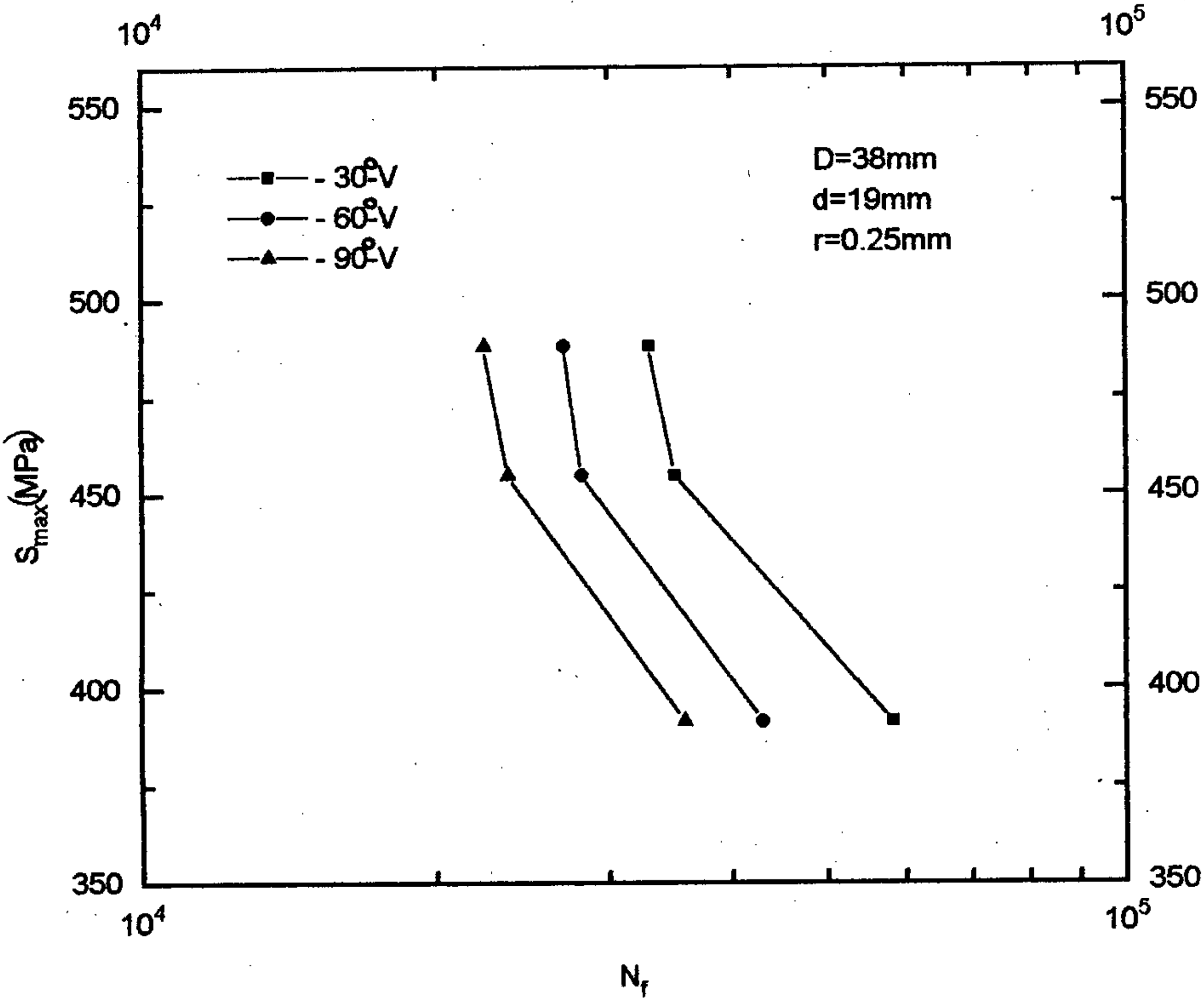


Figure 3 : Effect of maximum stress on fatigue life of notched mild steel shaft for D=38mm, d=19mm and r=0.25mm.

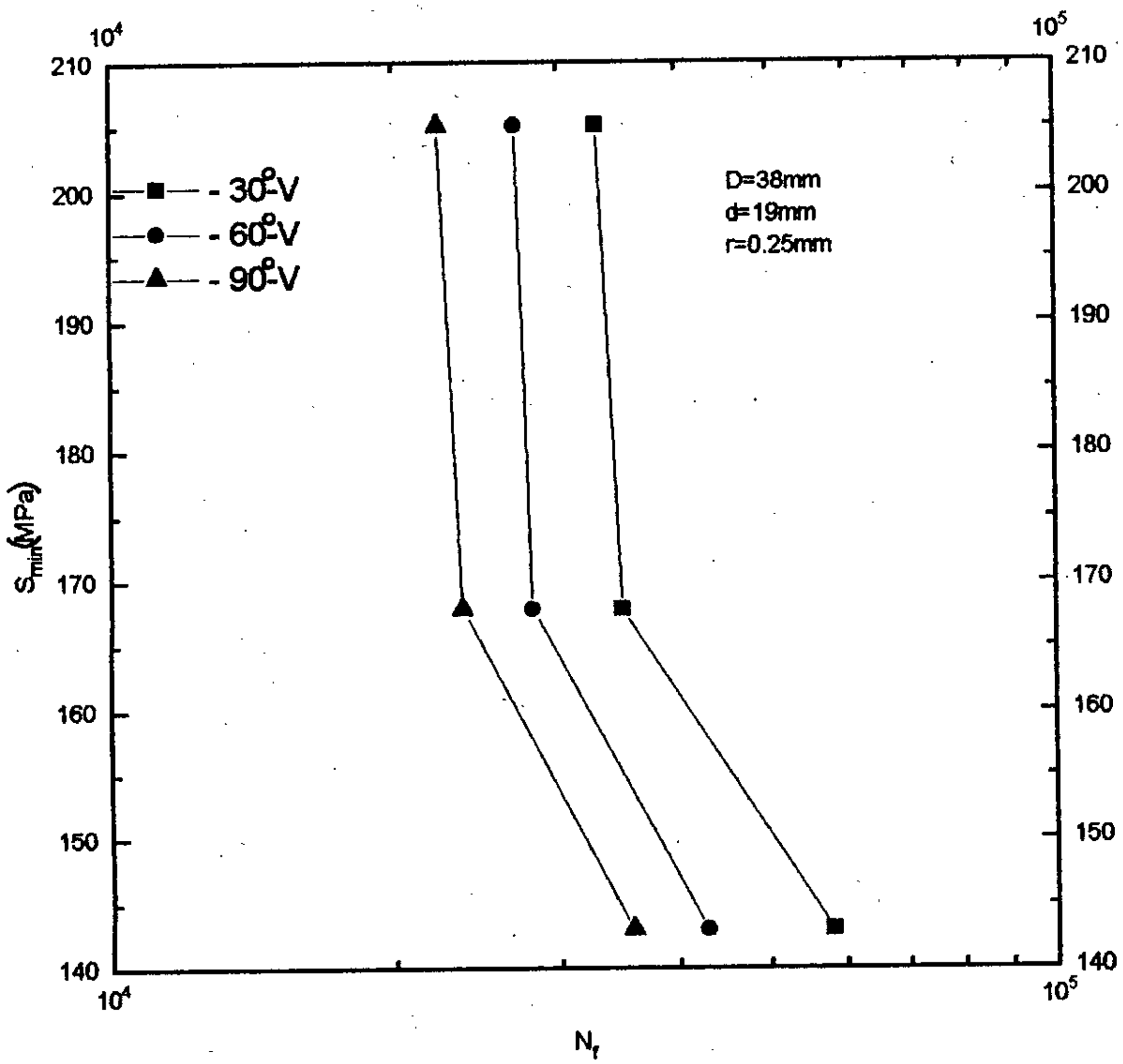


Figure 4 : Effect of minimum stress on fatigue life of notched mild steel shaft for D=38mm, d=19mm and r=0.25mm.

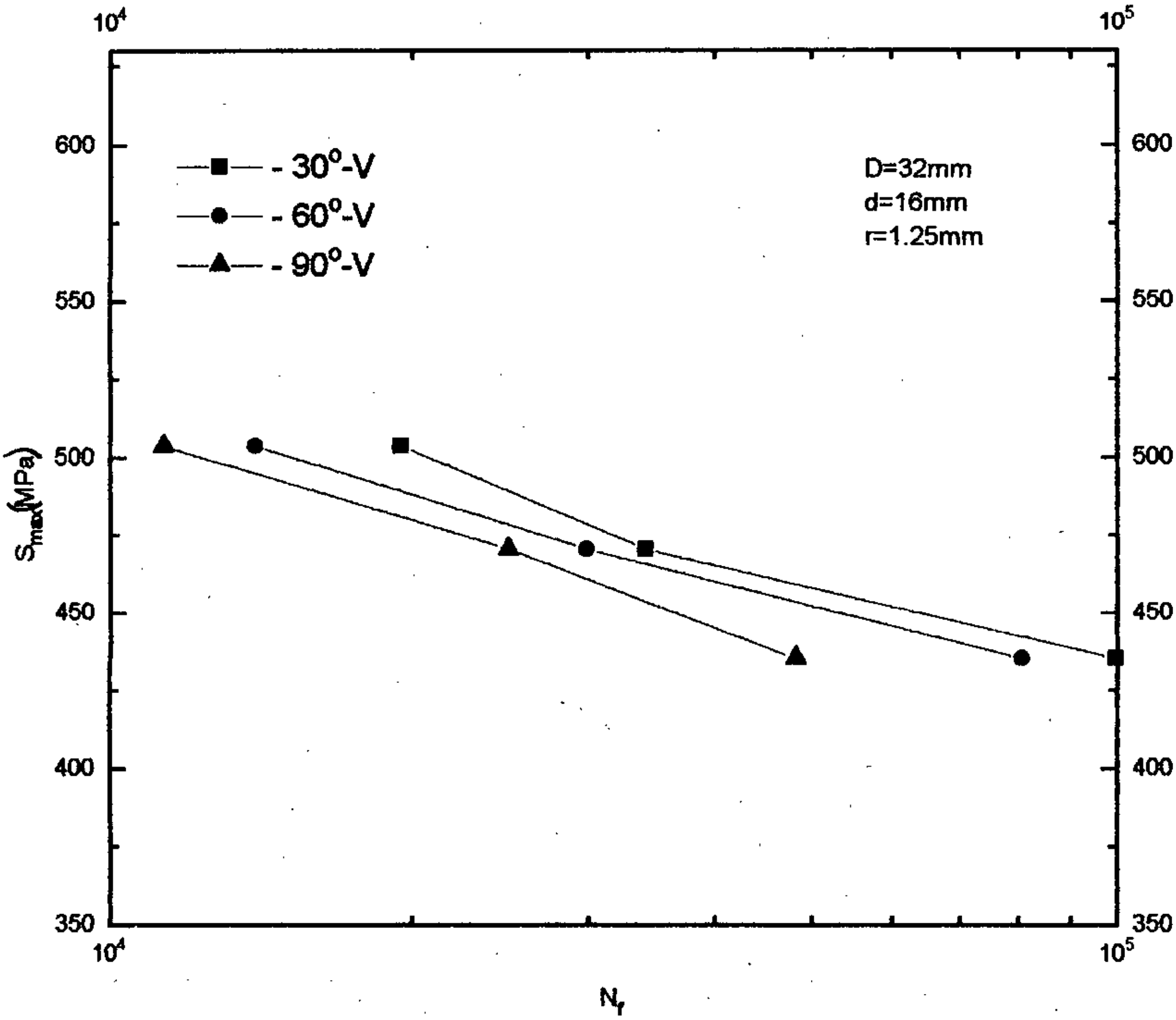


Figure 5 : Effect of maximum stress on fatigue life of notched mild steel shaft for D=32mm, d=16mm and r= 1.25mm.

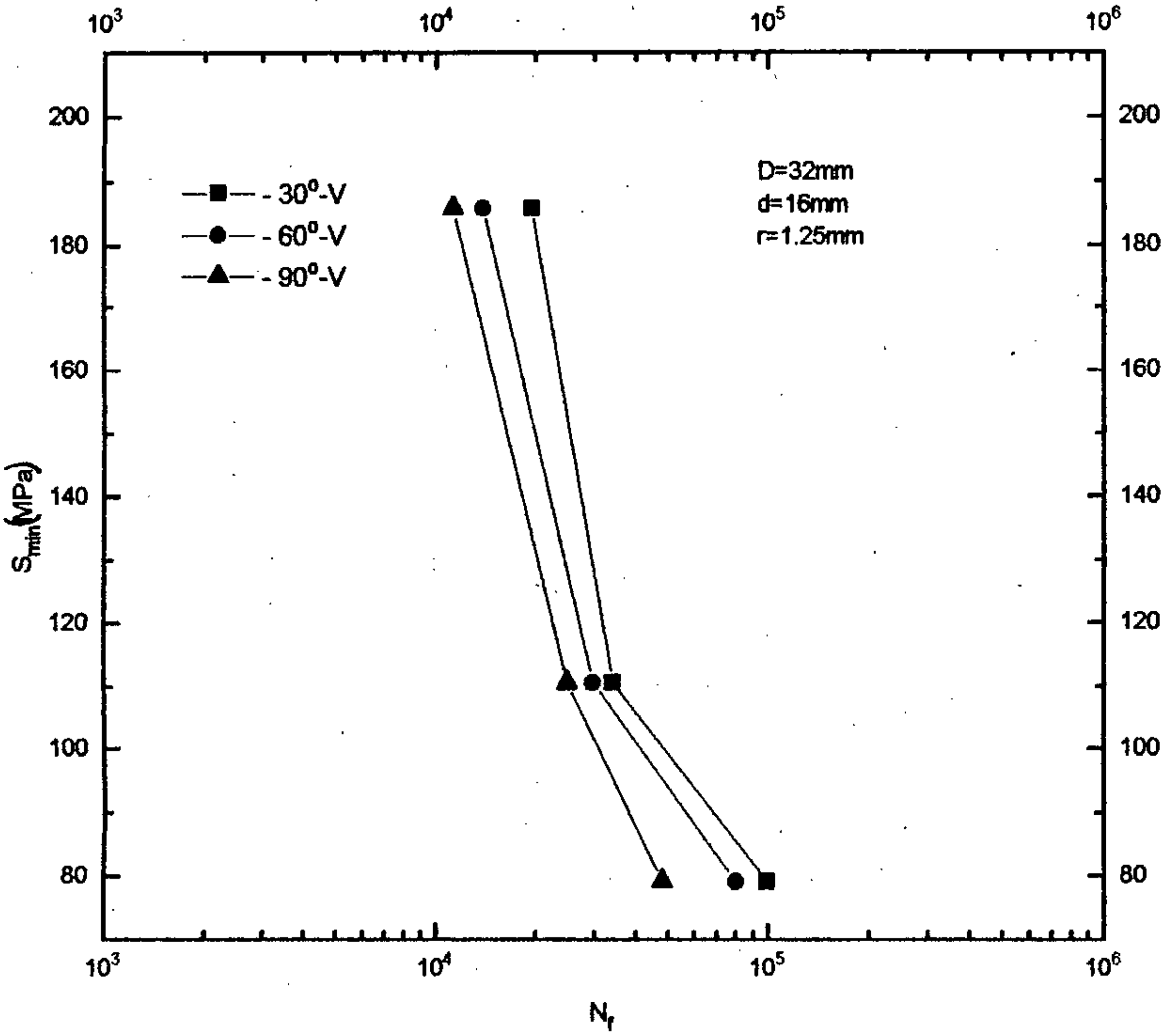


Figure 6 : Effect of minimum stress on fatigue life of notched mild steel shaft for D=32mm, d=16mm and r=1.25mm.

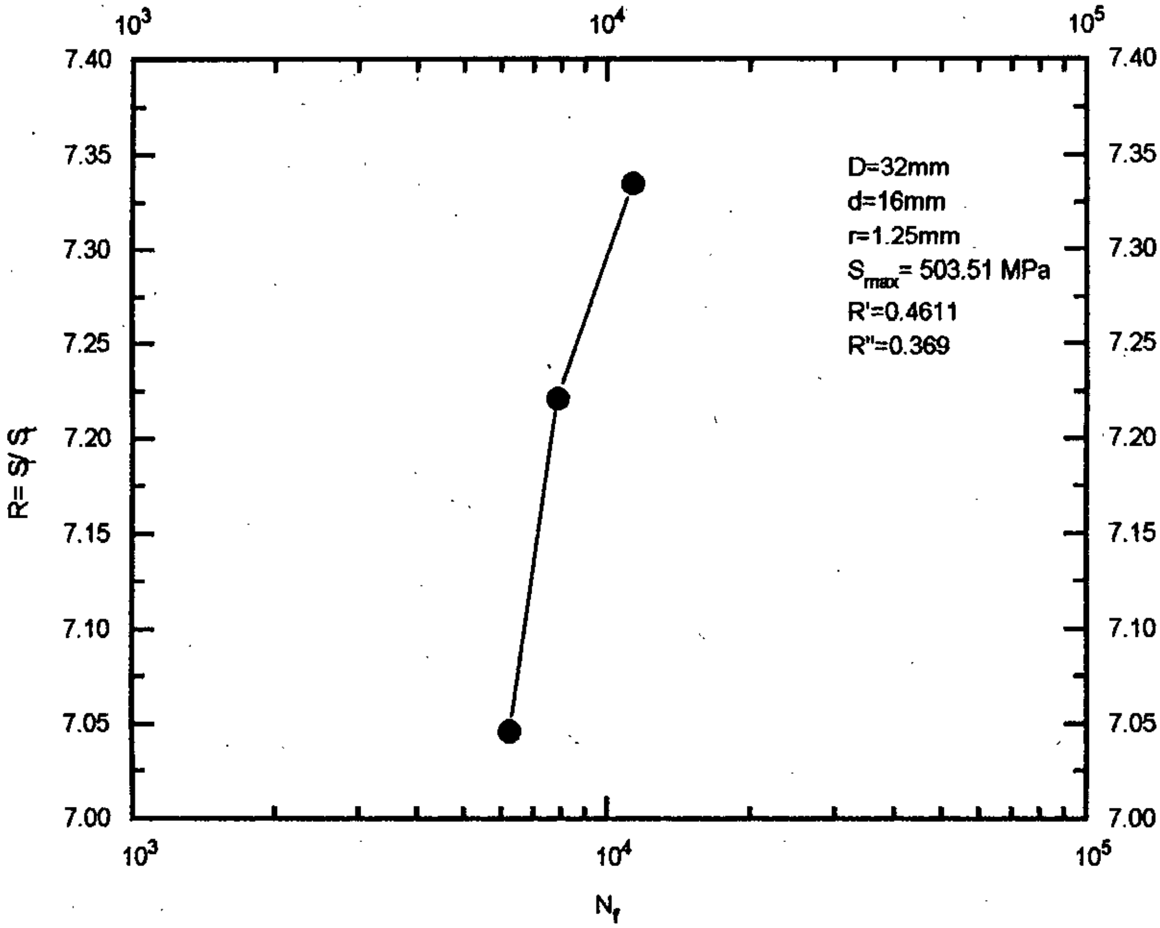


Figure 7 : Variation of fatigue life of 90° V-notched mild steel shaft at different flexural stress ratio for the max^{m} sterss of 503.51 MPa and alternating stress ratio of 0.4611.

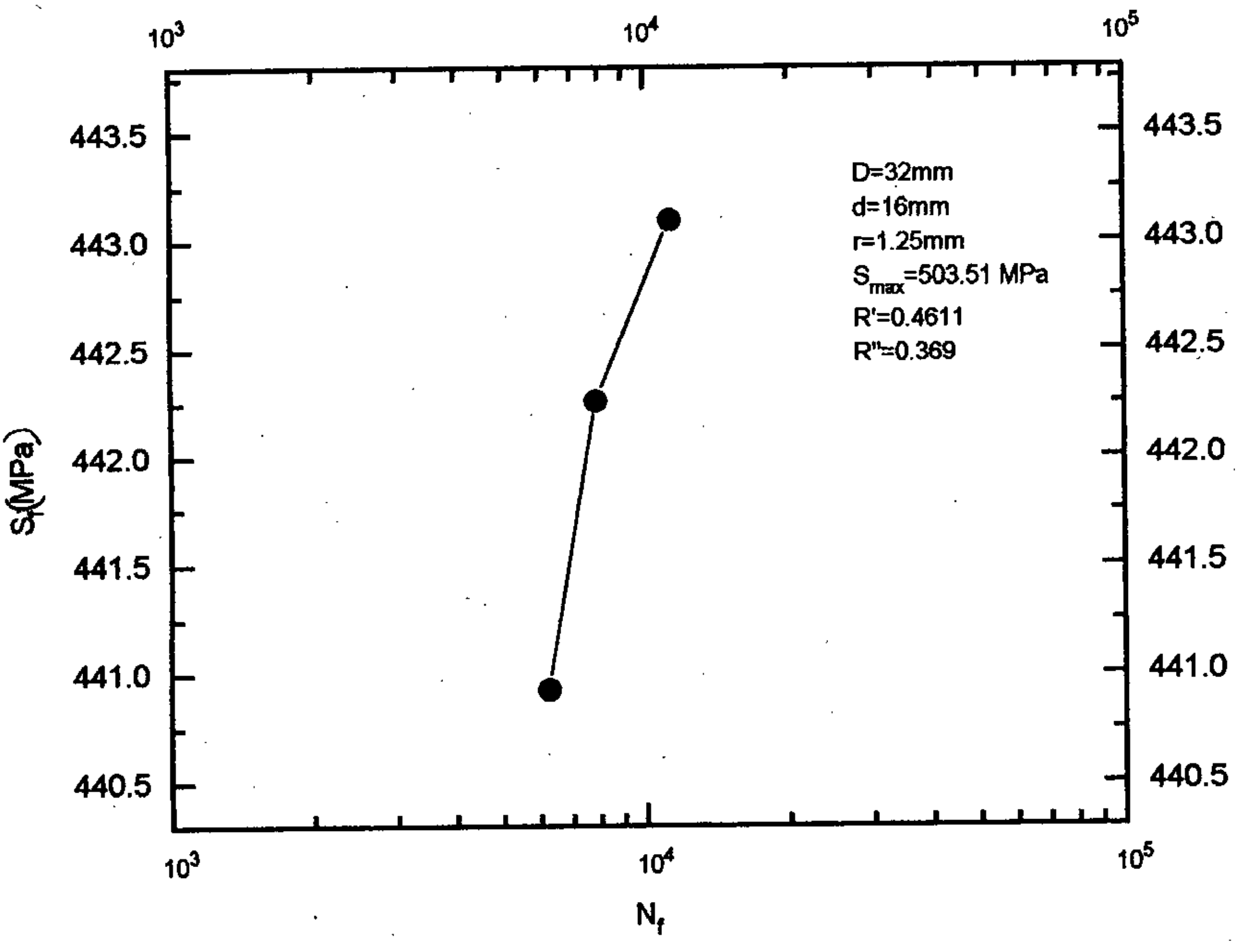


Figure 8 : Variation of fatigue life of 90° V-notched mild steel shaft at different flexural stress for the max^m stress of 503.51 MPa and alternating stress of 0.4611.

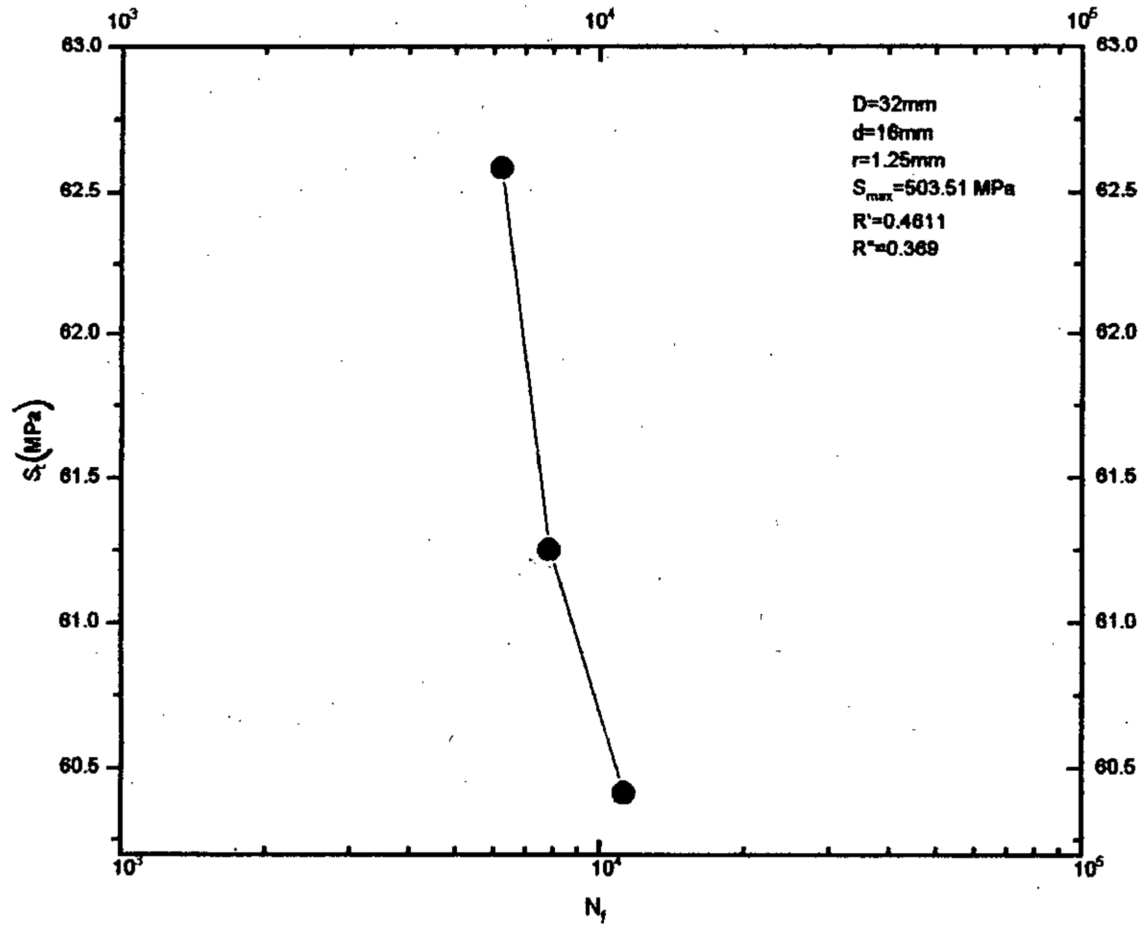


Figure 9 : Variation of fatigue life of 90° V-notched mild steel shaft at different tensile stress for max^m stress of 503.51 MPa and alternating stress ratio of 0.4611.

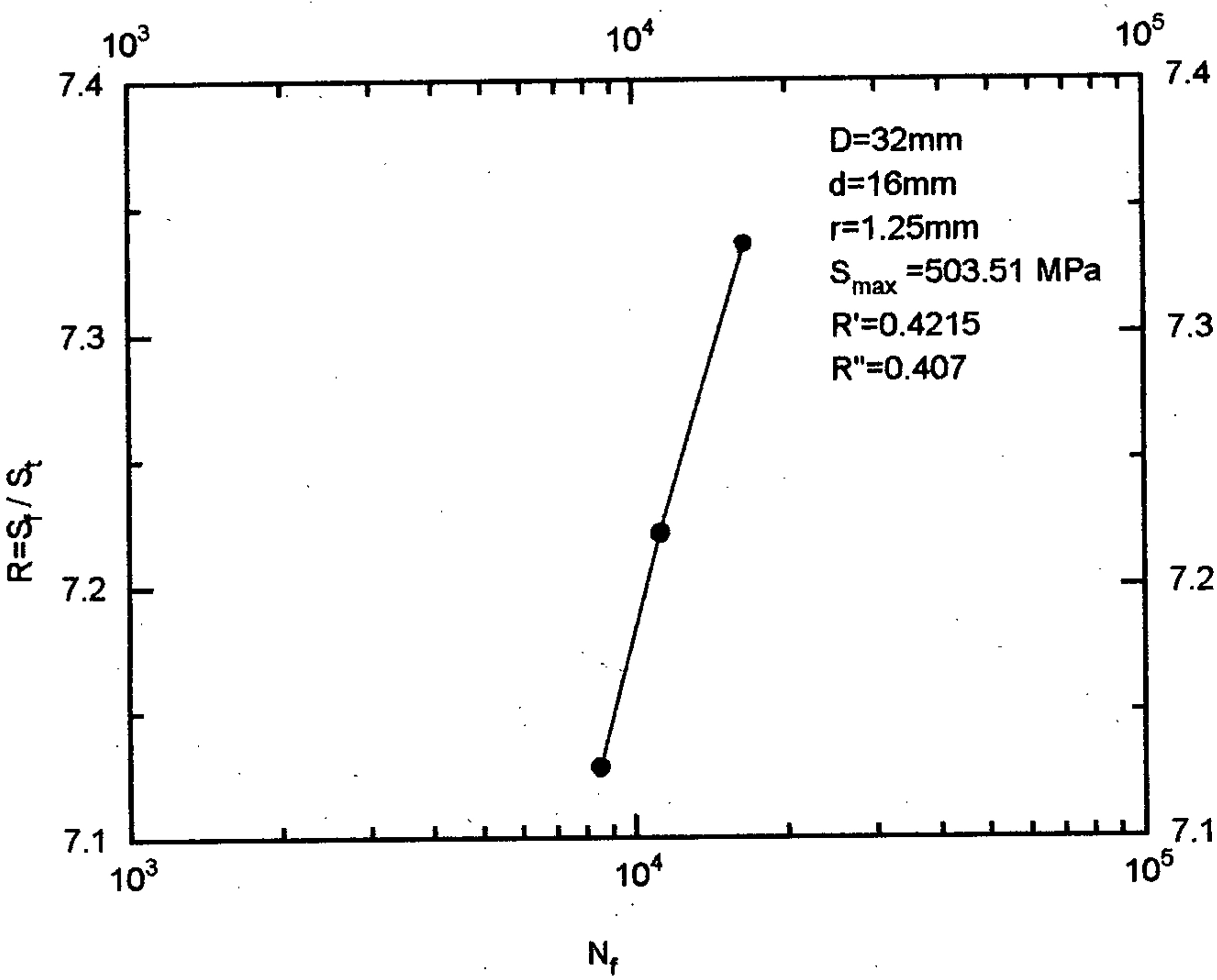


Figure 10 : Variation of fatigue life of 90 ° V-notched mild steel shaft at different flextural stress ratio for the maximum stress of 503.51 MPa and alternating stress ratio of 0.4215.

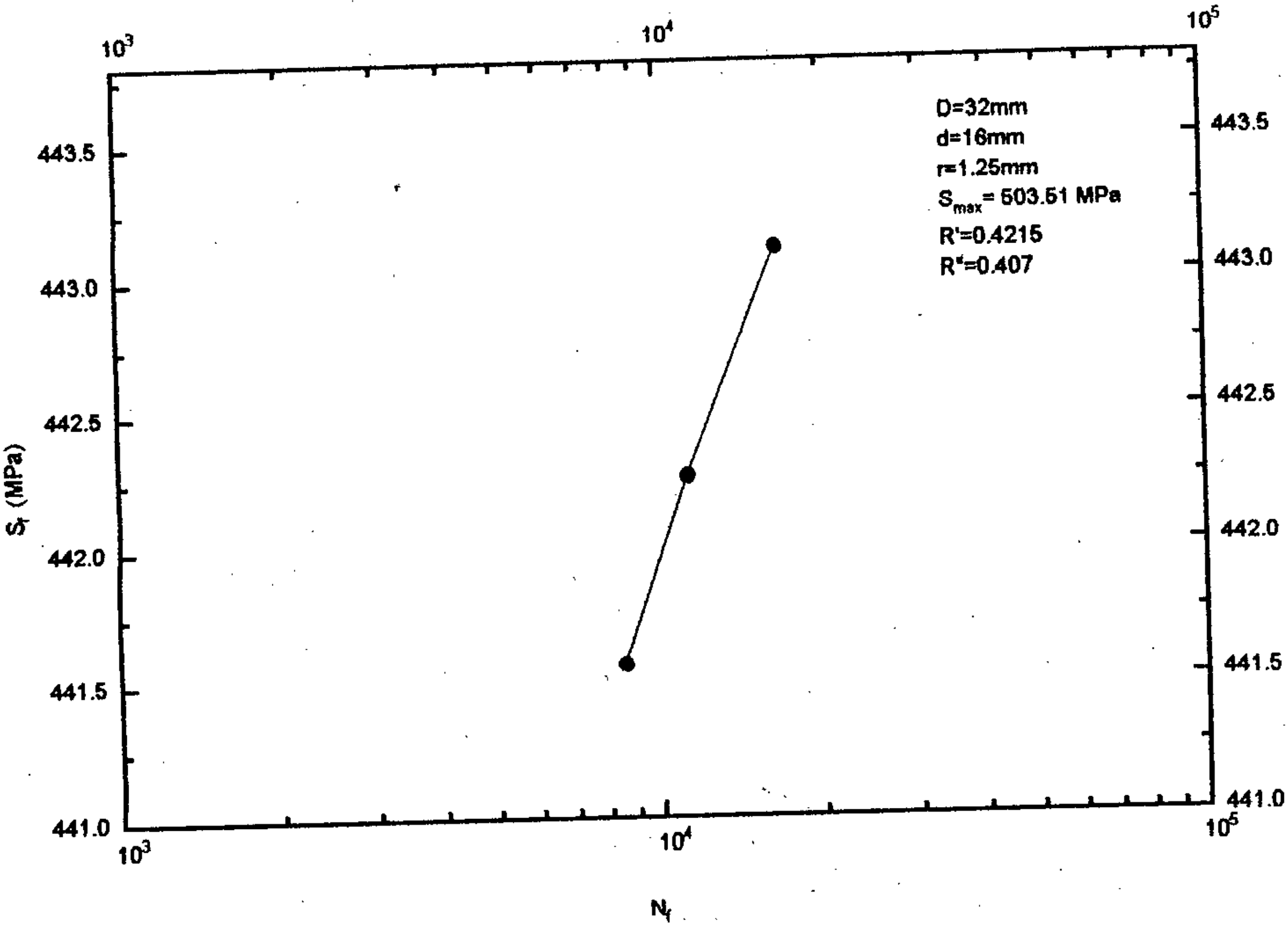


Figure 11: Variation of Fatigue life of 90° V- notched mild steel shaft at different flextural stress for the maximum stress of 503.51 MPa and alternating stress ratio of 0.4215.

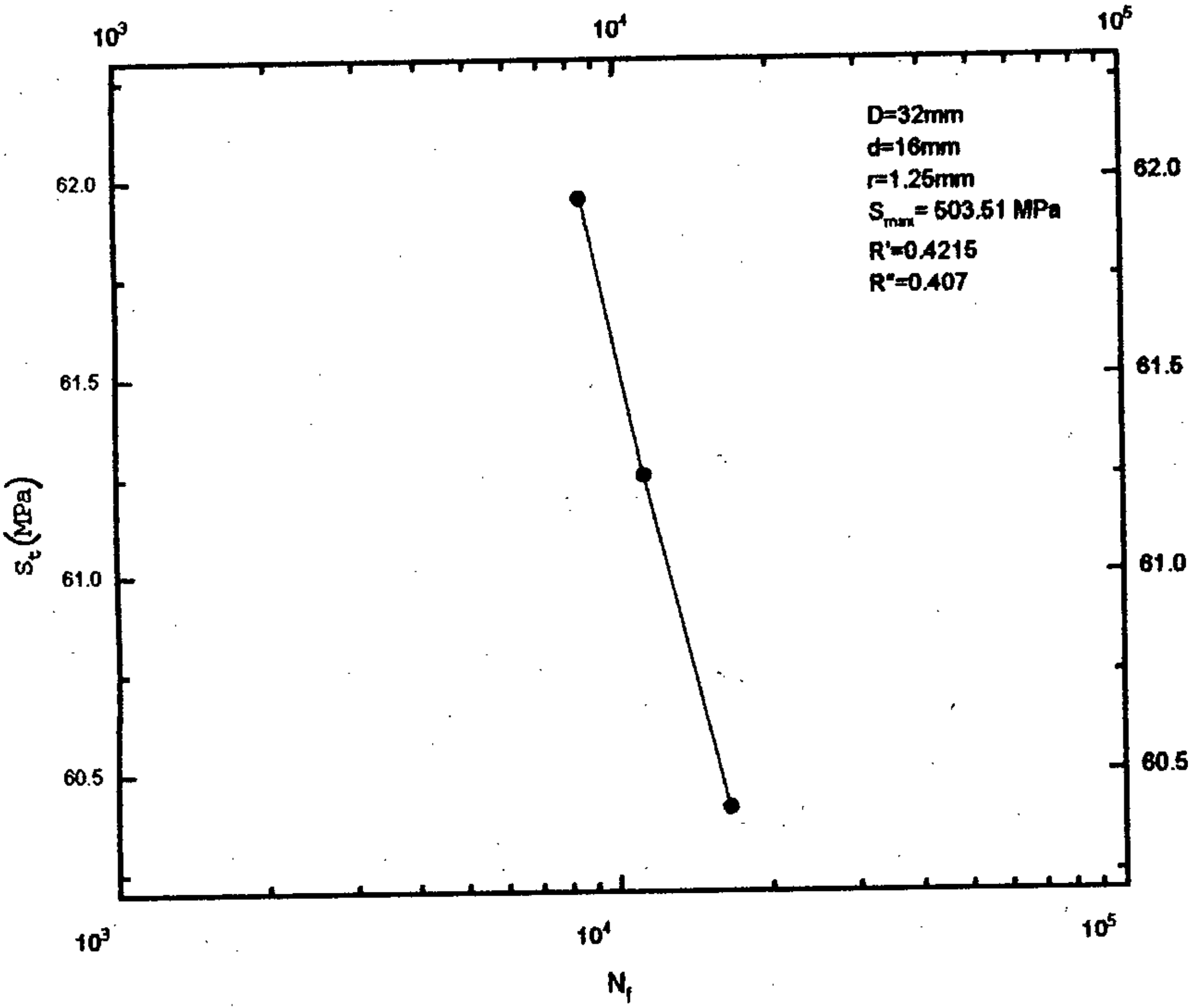


Figure 12 : Variation of fatigue life of 90° V- notched mild steel shaft at different tensile stress for the maximum stress of 503.51 MPa and alternating stress ratio of 0.4215 .

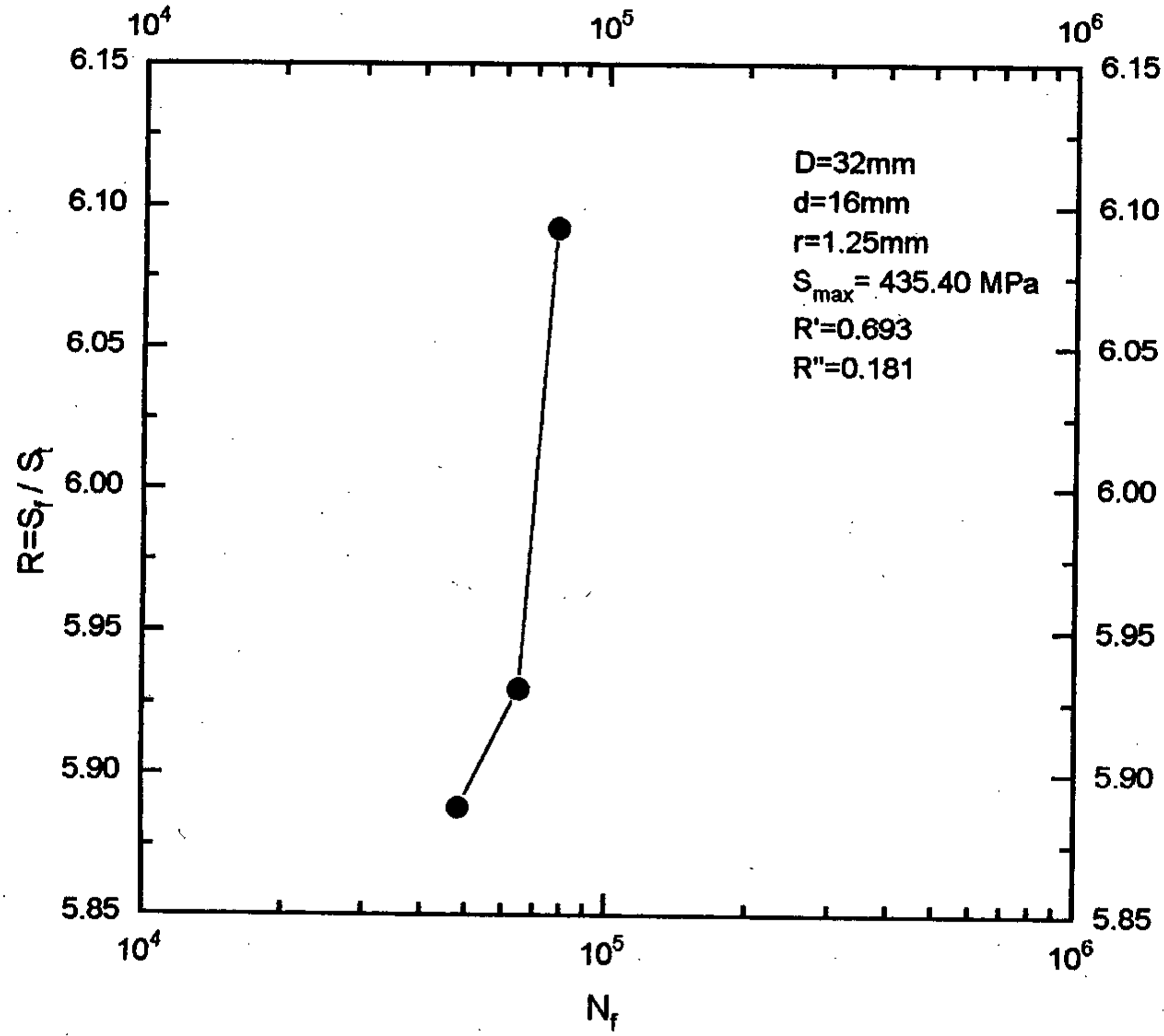


Figure 13 : Variation of fatigue life of 90° V-notched mild steel shaft at different flexural stress ratio for the max^{m} stress of 435.40 MPa.

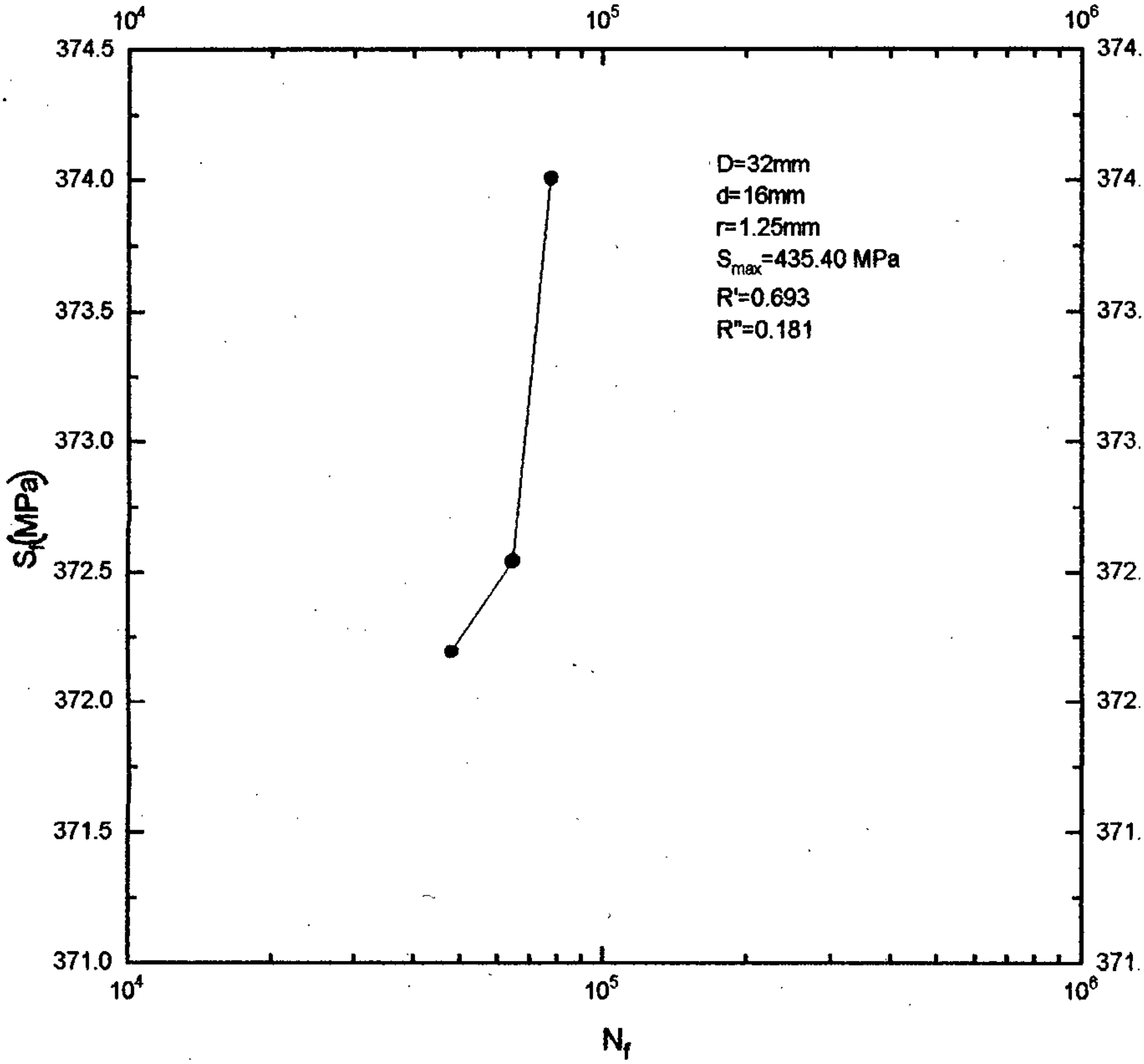


Figure 14 : Variation of fatigue life of 90° V-notched mild steel shaft at different flexural stress for the maximum stress of 435.40 Mpa.

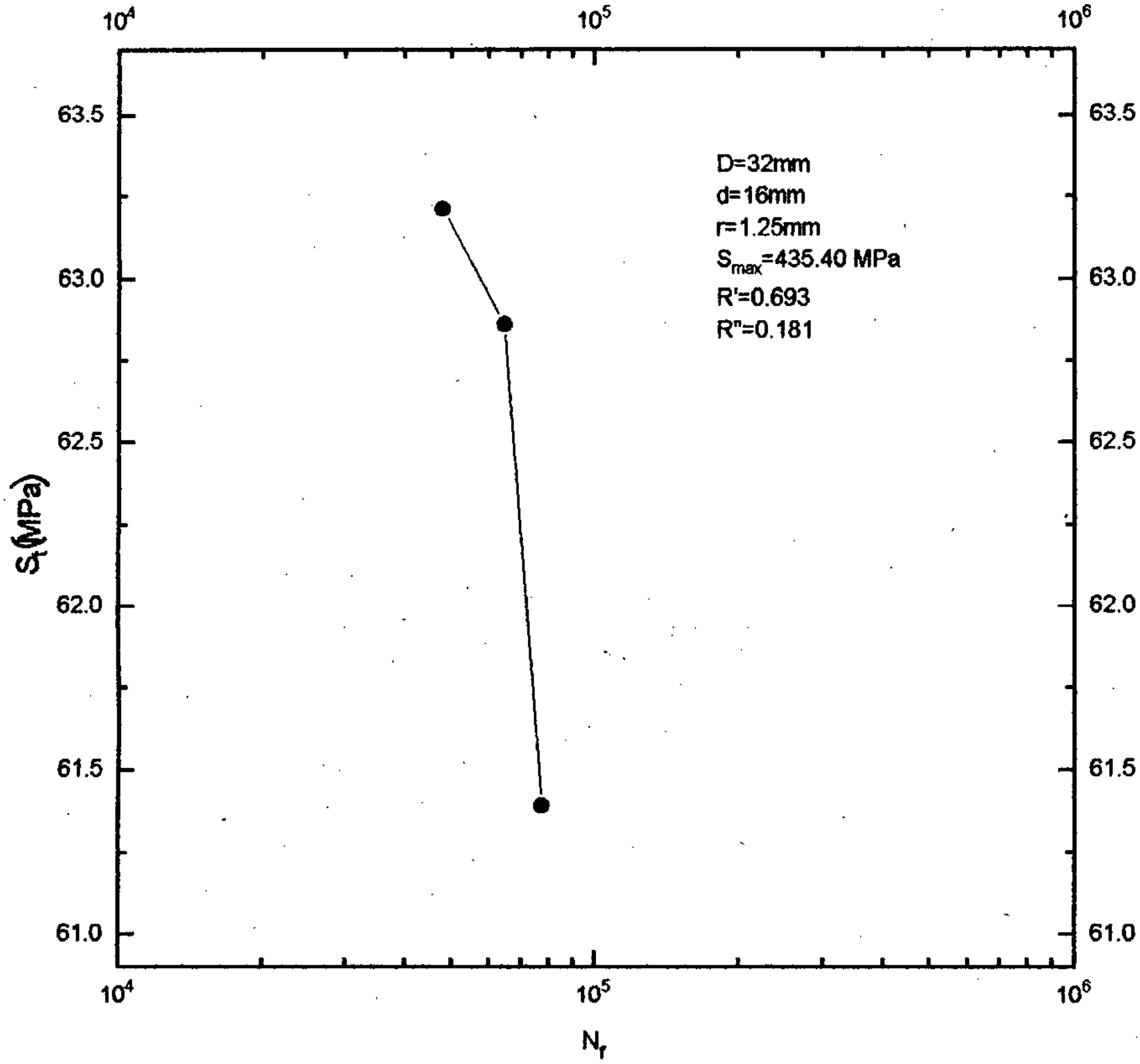


Figure 15 : Variation of fatigue life of 90° V-notched mild steel shaft at different tensile stress for the maximum stress of 435.4 MPa.

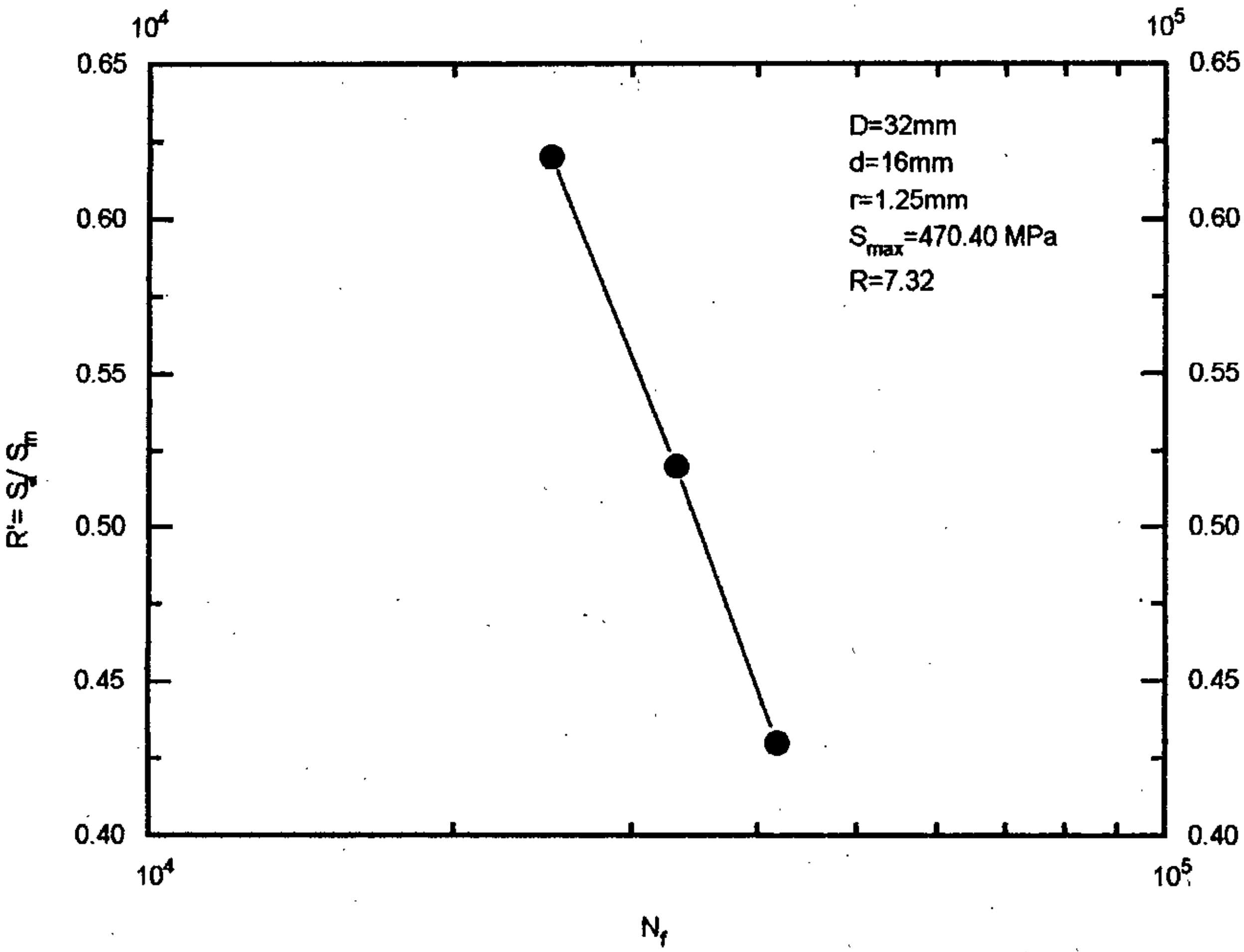


Figure 16 : Variation of fatigue life of 90° V-notched mild steel shaft at different alternating stress ratio for the maximum stress of 470.40 MPa.

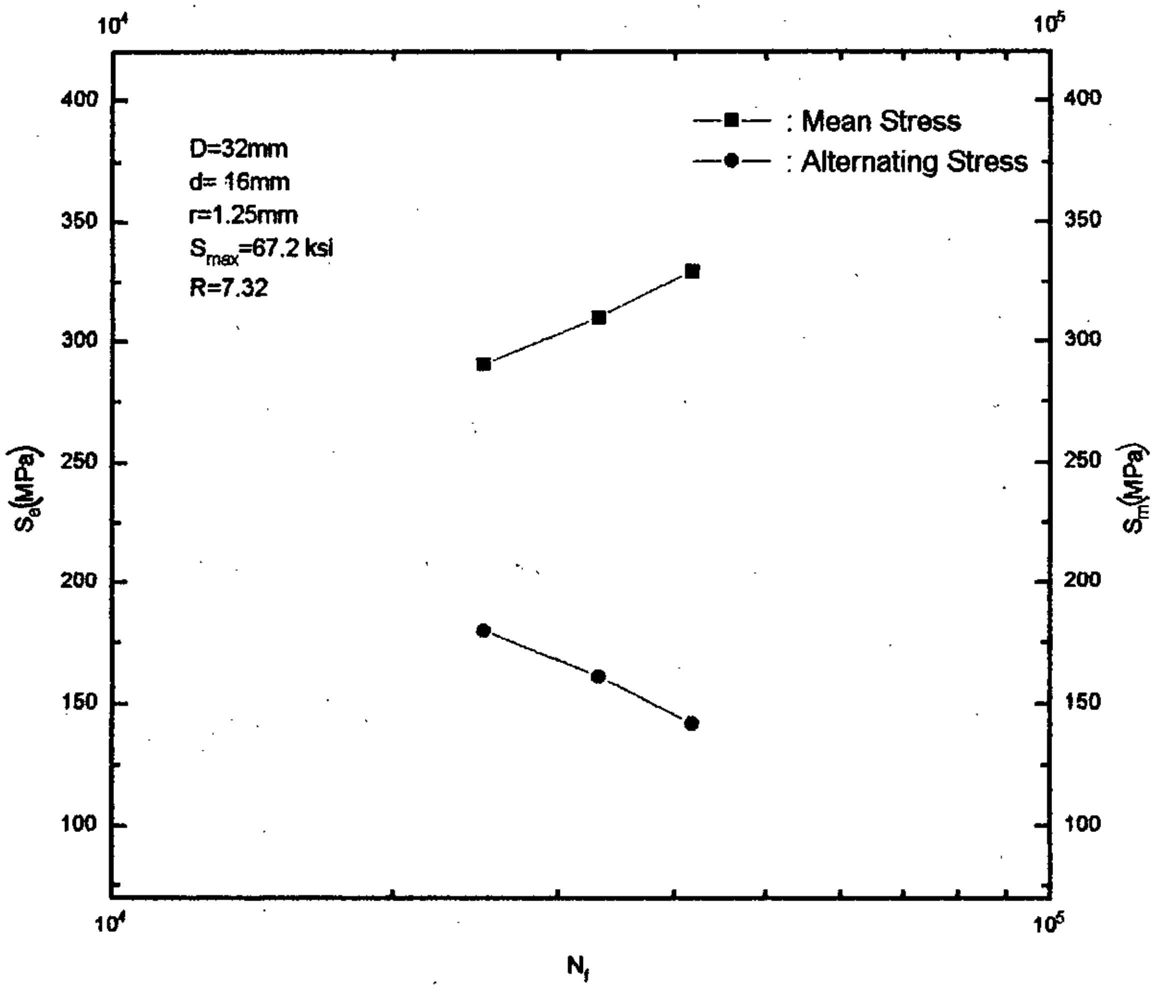


Figure 17 : Variation of fatigue life of 90° V-notched mild steel shaft at different alternating stress and mean stress for the maximum stress of 470.4 MPa.

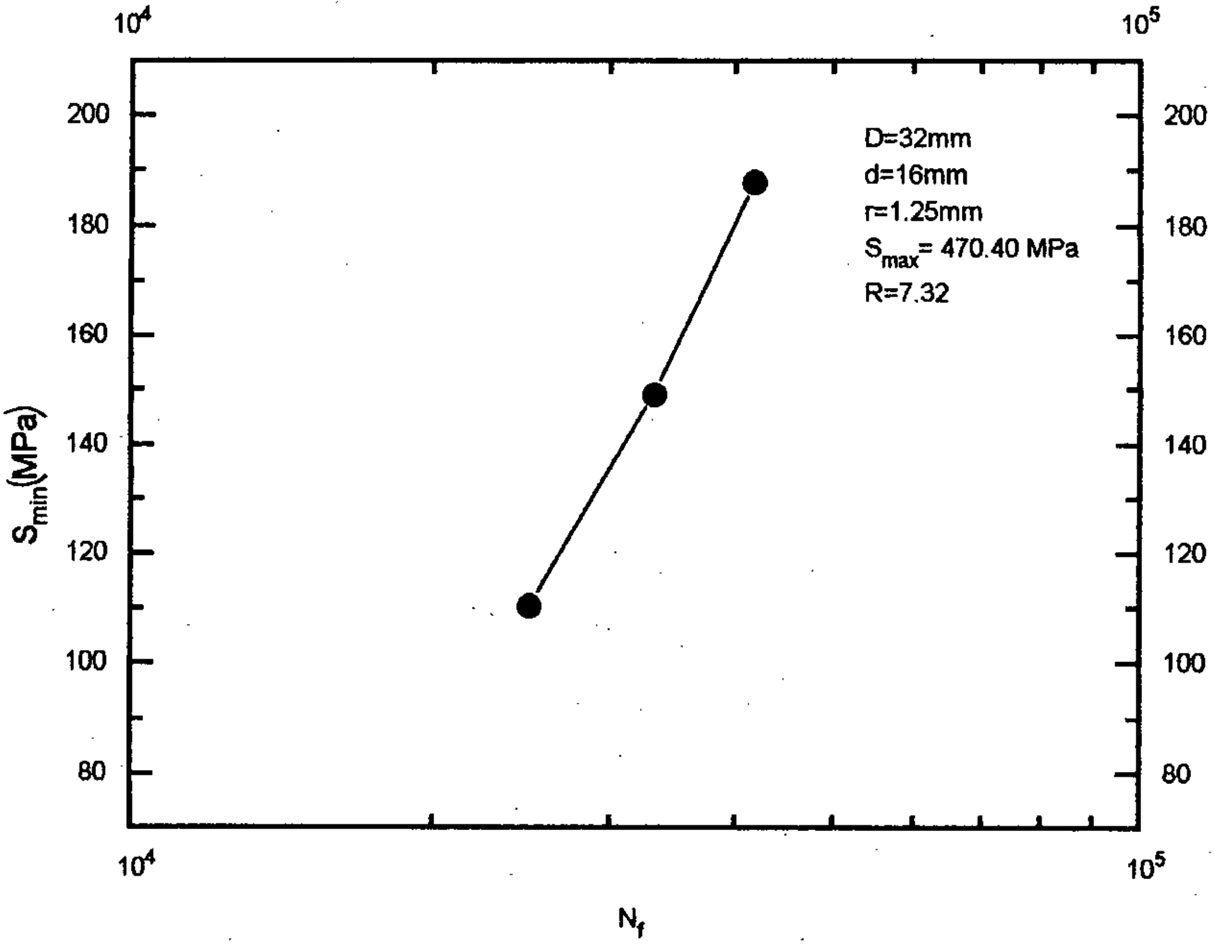


Figure 18 : **Variation of fatigue life of 90° V-notched mild steel shaft at different minimum stress for the maximum stress of 470.40 MPa.**

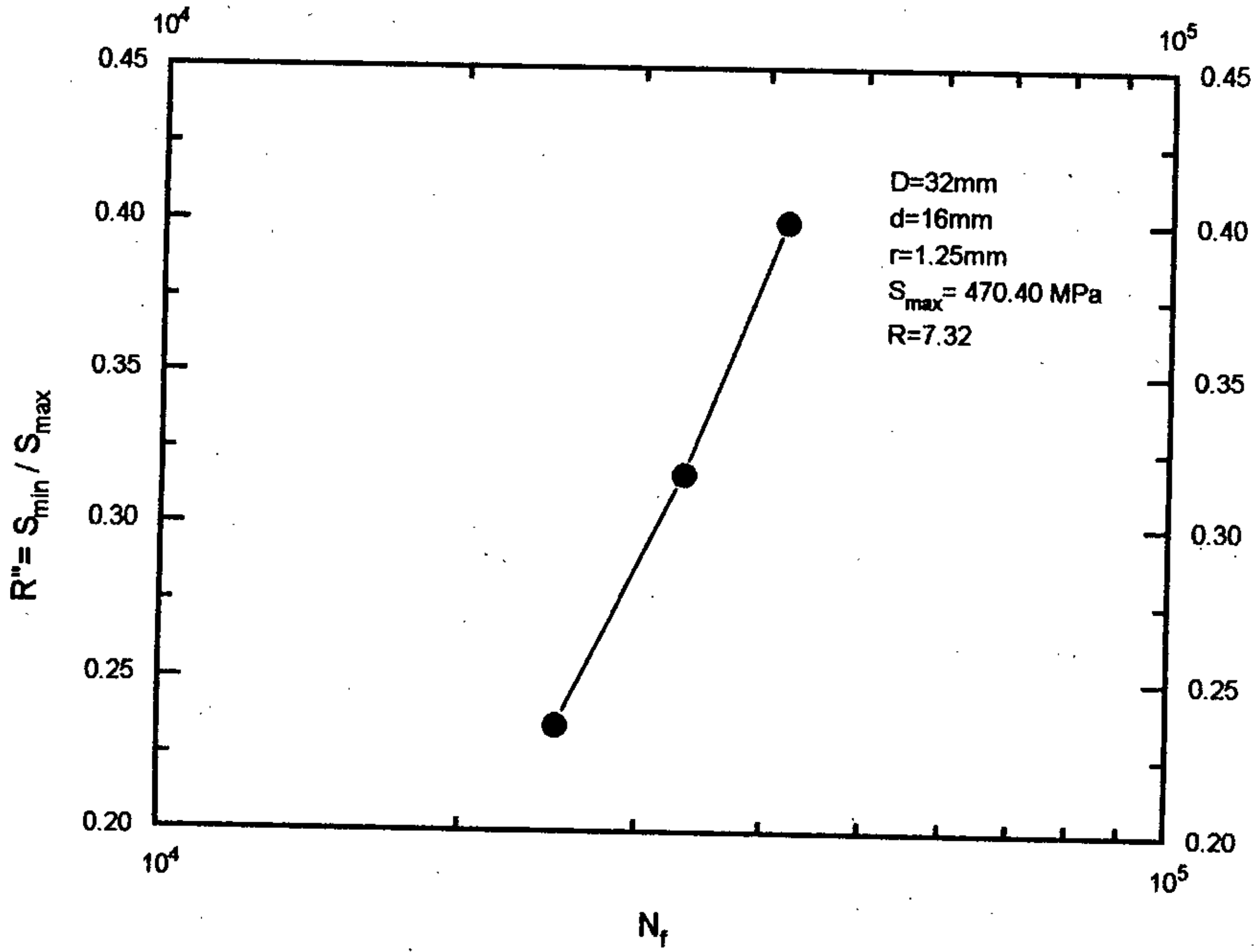


Figure 19 : Variation of fatigue life of 90° V-notched mild steel shaft at different stress ratio for the maximum stress of 470.4 MPa.

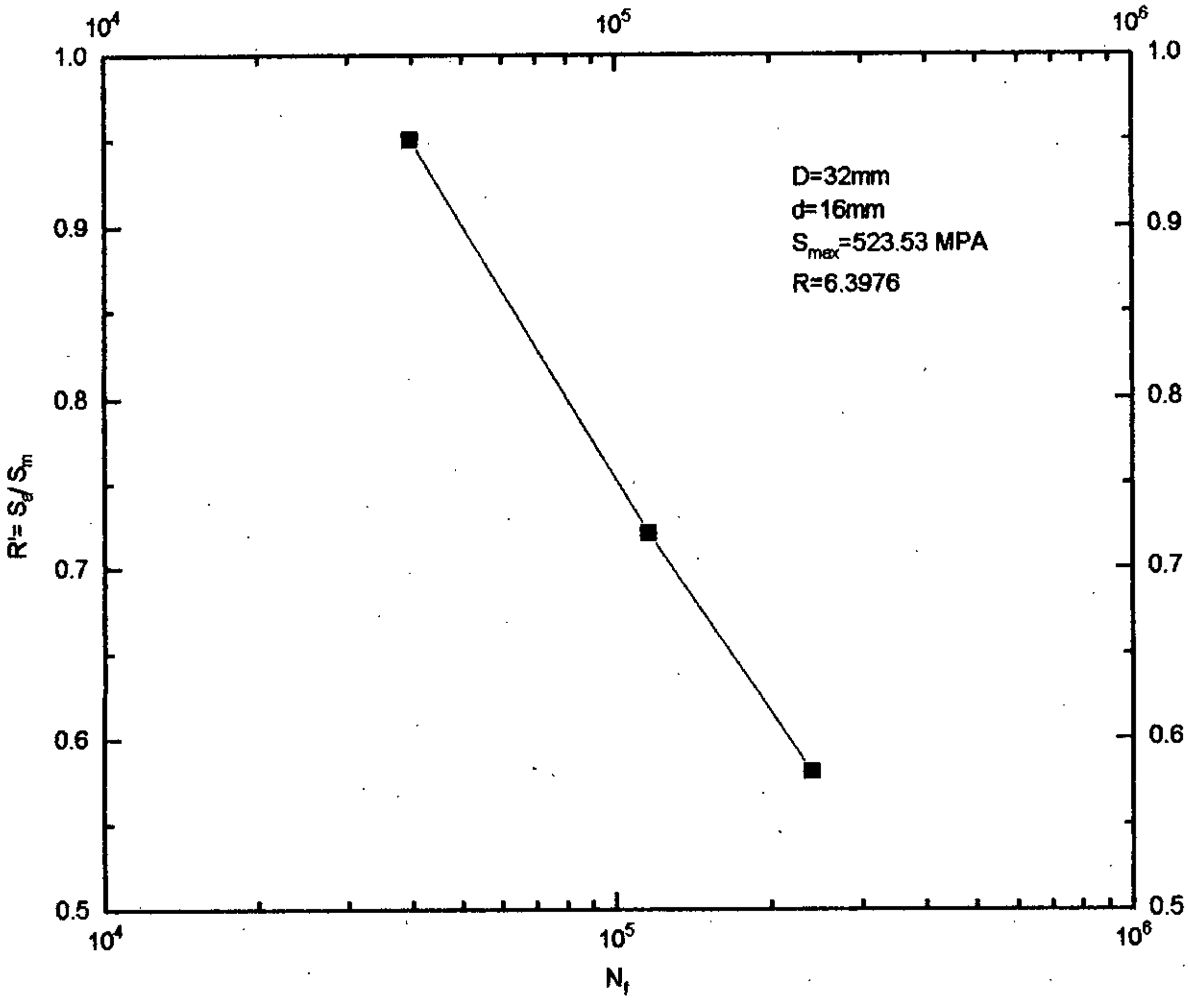


Figure 20 : Variation of fatigue life of plain mild steel specimen (shaft without notch) at different alternating stress ratio.

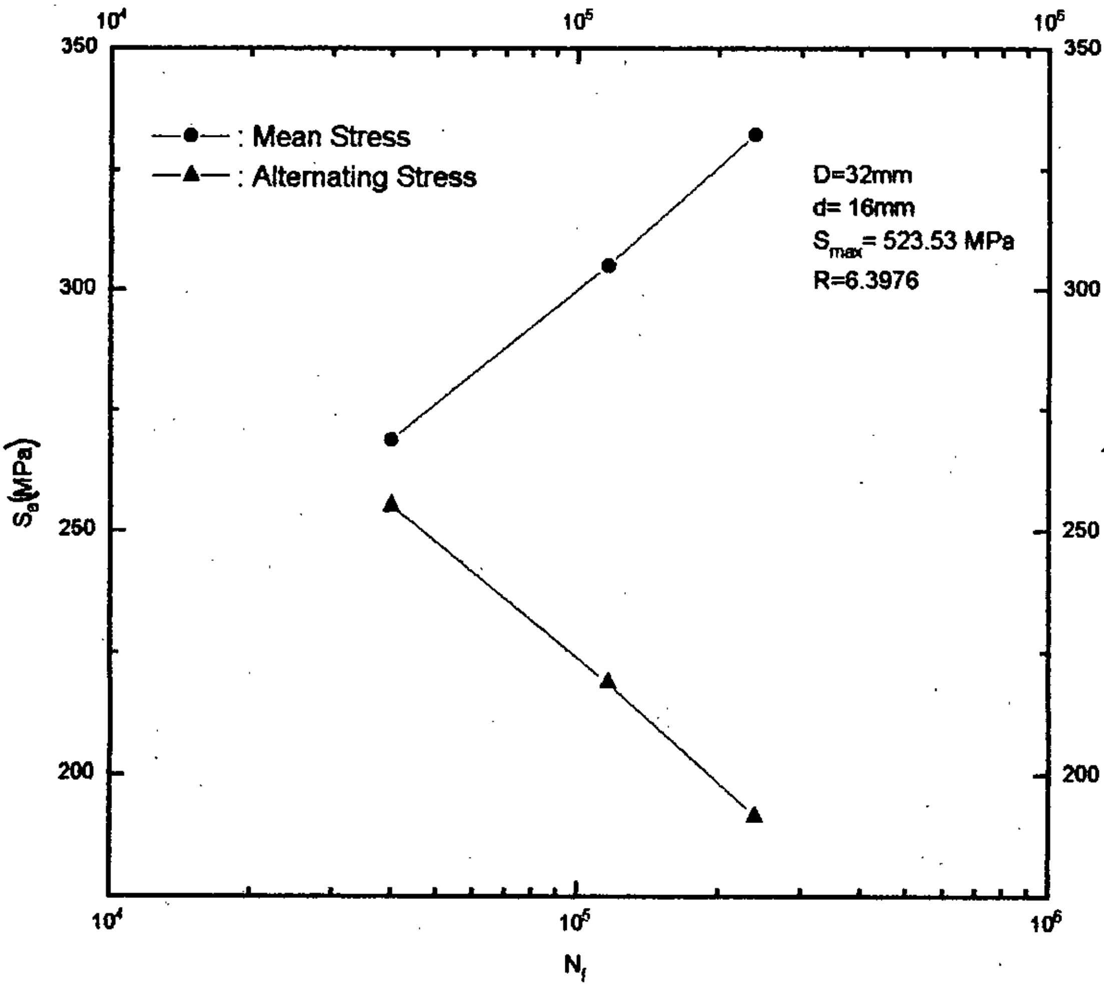


Figure 21 : Variation of fatigue life of plain mild steel specimen (shaft without notch) at different alternating stress and mean stress.

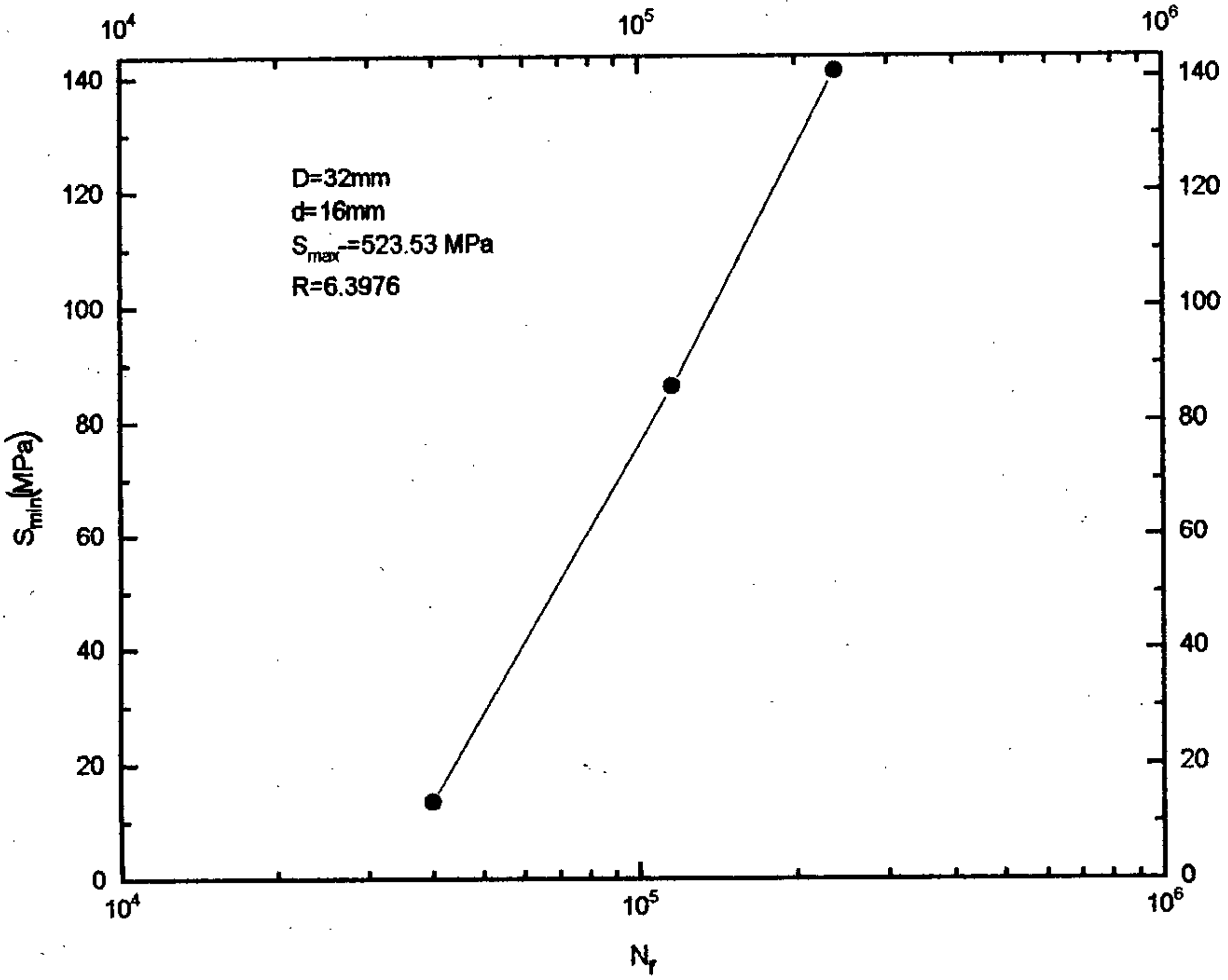


Figure 22 : Variation of fatigue life of plain mild steel specimen (Shaft without notch) at different minimum stress.

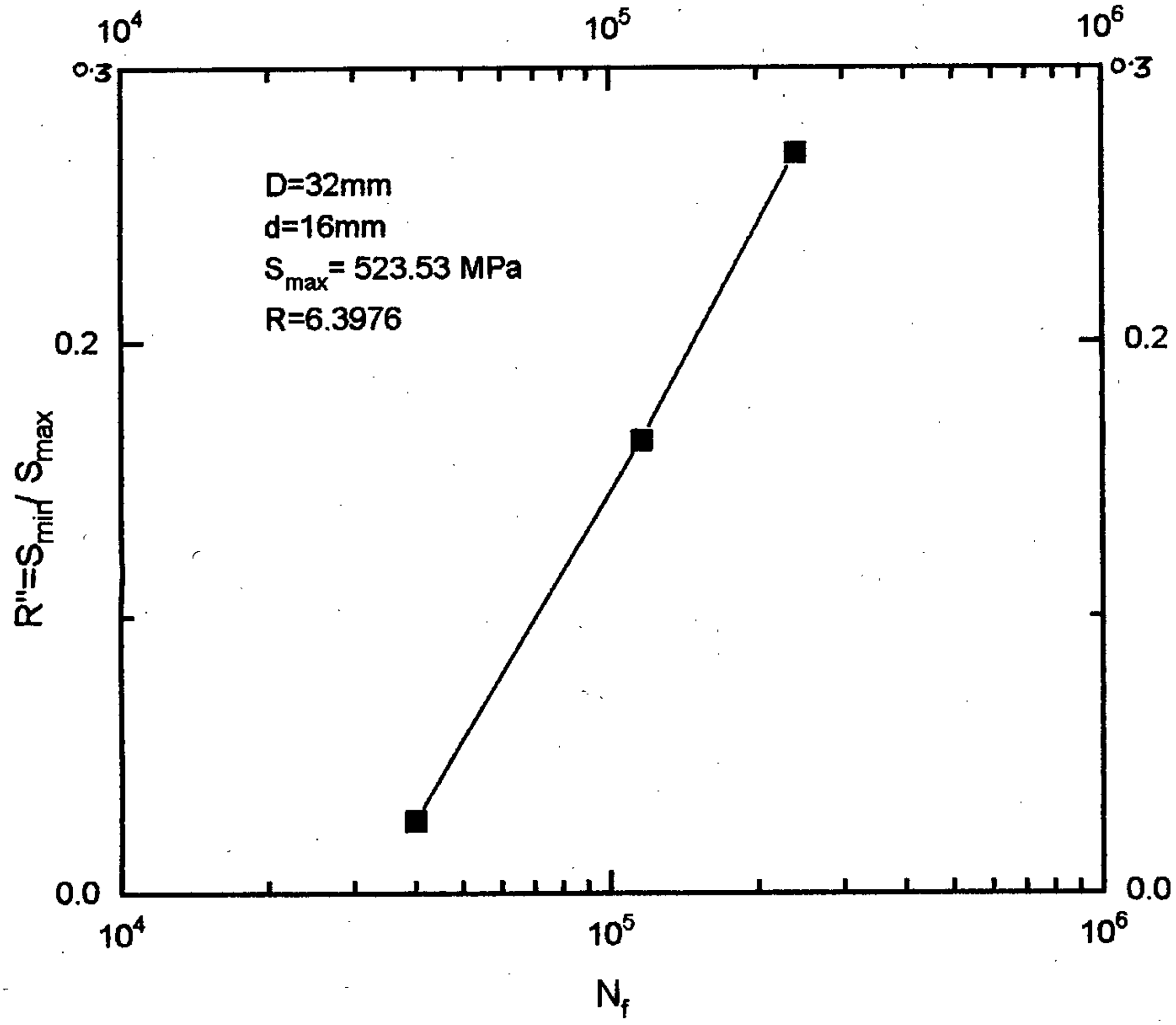


Figure 23 : Variation of fatigue life of plain mild steel specimen (shaft without notch) at different stress ratio.

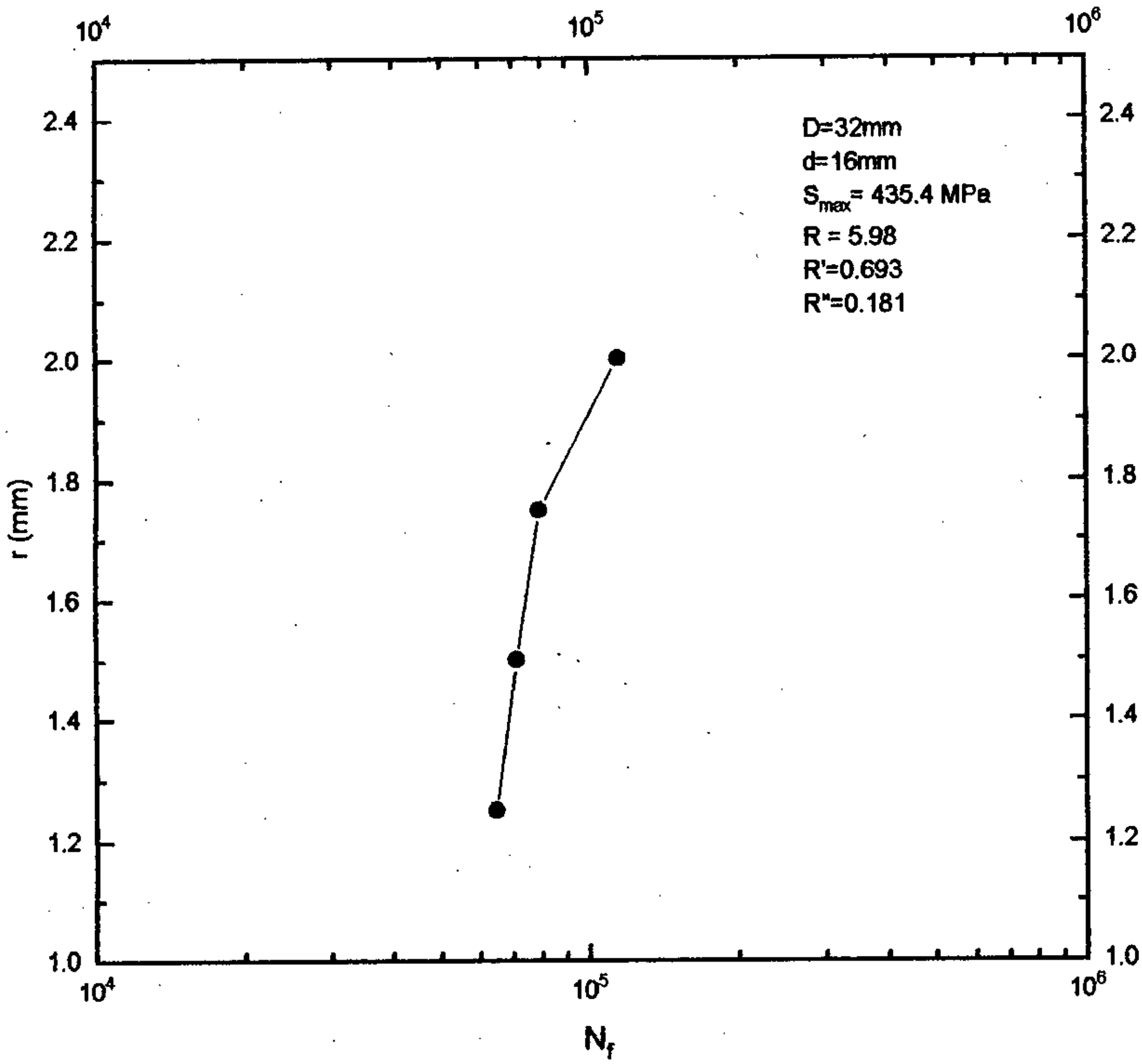


Figure 24: Effect of notch-root radius on fatigue life of 90° V-notched mild steel shaft for the maximum stress of 435.4 MPa .

APPENDIX - A

Table A-1 : Corelation between load and strain for the strain gage attached on the connecting bar.

Load (kg)	0	100	200	400	800	1600	3200	6400
Strain (in/in)	0	0.04×10^{-4}	0.08×10^{-4}	0.16×10^{-4}	0.32×10^{-4}	0.64×10^{-4}	1.28×10^{-4}	2.56×10^{-4}

Table A-2 : Corelation between voltage change and strain for the same strain gage used on the connecting bar

Change of Voltage (volt)	0	5×10^{-4}	1×10^{-3}	2×10^{-3}	4×10^{-3}
Strain (in/in)	0	5×10^{-4}	1×10^{-3}	2×10^{-3}	4×10^{-3}

Table A-3 : Corelation between load and voltage change for the strain gage used on the connecting bar

Load (kg)	0	100	500	2500	5000
Voltage Change (Volt)	0	0.04×10^{-4}	0.2×10^{-4}	1×10^{-4}	2×10^{-4}

Table A-4(a) : Chemical composition of mild steel used for the experiments.

% of Carbon	% of Manganese	% of Phosphorus	% of Sulfur	% of Silicon
0.3	0.45	0.041	0.062	0.284

Table A-4(b) : Mechanical properties of mild steel used for the experiments.

Hardness	Ultimate Tensile stress (MPa)	0.1% Proof stress (MPa)	% Elongation	% Reduction of Area
HR B-84	616	546	16	46

Table A-5 : Values of different parameters of V-notched and circular notched mild steel specimen

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	5.72	11.19	90	3.63	22370
2.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	5.72	11.19	60	3.63	27010
3.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	5.72	11.19	30	3.63	33007
*	346.64	141.54	0.41	488.18	205.10	0.42	448.14	5.72	11.19	Circular notch	1.5	92005

Table A-6 : Values of different parameters of V-notched mild steel shaft for the value of maximum stress of 454.79 MPa for D = 38 mm, d = 19 mm and r = 0.25 mm.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
4.	311.29	143.5	0.46	454.79	167.79	0.369	413.28	41.51	9.96	90	3.63	23668
5.	311.29	143.5	0.46	454.79	167.79	0.369	413.28	41.51	9.96	60	3.63	28111
6.	311.29	143.5	0.46	454.79	167.79	0.369	413.28	41.51	9.96	30	3.63	34996

* Values of different parameters of circular notched mild steel specimen.

Table A-7 : **Values of different parameters of V-notched mild steel shaft for the value of maximum stress of 391.58 MPa for D = 38 mm, d = 19 mm and r = 0.25 mm.**

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
7.	267.26	124.32	0.47	391.58	142.94	0.365	245.31	46.27	7.46	90	3.63	35766
8.	267.26	124.32	0.47	391.58	142.94	0.365	245.31	46.27	7.46	60	3.63	42919
9.	267.26	124.32	0.47	391.58	142.94	0.365	245.31	46.27	7.46	30	3.63	58202

Table A-8 : **Values of different parameters of V-notched mild steel shaft for the value of maximum stress of 470.40 MPa for D = 32 mm, d = 16 mm and r = 1.25 mm.**

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
10.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	90	2.46	24996
11.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	60	2.46	29921
12.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	30	2.46	34205

Table A-9 : Values of different parameters of V-notched mild steel shaft for the value of maximum stress of 435.40 MPa for D = 32 mm, d = 16 mm and r = 1.25 mm.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
13.	257.18	178.22	0.69	435.40	78.96	0.181	372.54	62.86	5.93	90	2.46	64835
14.	257.18	178.22	0.69	435.40	78.96	0.181	372.54	62.86	5.93	60	2.46	80651
15.	257.18	178.22	0.69	435.40	78.96	0.181	372.54	62.86	5.93	30	2.46	99813

Table A-10 : Values of different parameters of V-notched mild steel shaft for the value of maximum stress of 503.51 MPa for D = 32 mm, d = 16 mm and r = 1.25 mm.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
16.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	90	2.46	11329
17.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	60	2.46	13959
18.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	30	2.46	19502

Table A-11 : Values of different parameters of V-notched mild steel shaft for D = 38 mm, d = 19 mm and r = 0.25 mm.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	267.26	124.32	0.47	391.58	142.94	0.365	345.31	46.27	7.46	30	3.63	58202
2.	311.29	143.50	0.46	454.79	167.79	0.369	413.28	41.51	9.96	30	3.63	34996
3.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	40.04	11.19	30	3.63	33007
4.	267.26	124.32	0.47	391.58	142.94	0.365	345.31	46.27	7.46	60	3.63	42919
5.	311.29	143.50	0.46	454.79	167.79	0.369	413.28	41.51	9.96	60	3.63	28111
6.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	40.04	11.19	60	3.63	27010
7.	267.26	124.32	0.47	391.58	142.94	0.365	345.31	46.27	7.46	90	3.63	35766
8.	311.29	143.50	0.46	454.79	167.79	0.369	413.28	41.51	9.96	90	3.63	23668
9.	346.64	141.54	0.41	488.18	205.10	0.42	448.14	40.04	11.19	90	3.63	22370

Table A-12 : Values of different parameters of V-notched mild steel shaft for D = 32 mm, d = 16 mm and r = 1.25 mm.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	30	2.46	19502
2.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	30	2.46	34205
3.	257.18	178.22	0.693	435.40	78.96	0.181	372.19	63.21	5.888	30	2.46	99813
4.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	60	2.46	13959
5.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	60	2.46	29921
6.	257.18	178.22	0.693	435.40	78.96	0.181	372.19	63.21	5.888	60	2.46	80651
7.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	90	2.46	11329
8.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	90	2.46	24996
9.	257.18	178.22	0.693	435.40	78.96	0.181	372.19	63.21	5.888	90	2.46	48228

Table A-13 : **Values of different parameters of 90^0 V-notched mild steel shaft for D = 32 mm, d = 16 mm and r = 1.25 mm for the maximum stress of 503.51 MPa and alternating stress ratio of 0.4611.**

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	344.61	158.90	0.4611	503.51	185.71	0.369	440.93	62.58	7.046	90	2.46	6255
2.	344.61	158.90	0.4611	503.51	185.71	0.369	442.26	61.25	7.221	90	2.46	7851
3.	344.61	158.90	0.4611	503.51	185.71	0.369	443.10	60.41	7.335	90	2.46	11329

Table A-14 : **Values of different parameters of 90^0 V-notched mild steel shaft for D = 32 mm, d = 16 mm and r=1.25 mm for the maximum stress of 503.51 MPa and alternating stress ratio of 0.4215.**

Obs .no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	354.20	149.31	0.4215	503.51	204.89	0.407	441.56	61.95	7.128	90	2.46	8545
2.	354.20	149.31	0.4215	503.51	204.89	0.407	442.26	61.25	7.221	90	2.46	11247
3.	354.20	149.31	0.4215	503.51	204.89	0.407	443.10	60.41	7.335	90	2.46	16556

Table A-15 : Values of different parameters of 90^0 V-notched mild steel shaft for D = 32 mm, d = 16 mm and r=1.25 mm for the maximum stress of 435.40 MPa and alternating stress ratio of 0.693.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	257.18	178.22	0.693	435.40	78.96	0.181	372.19	63.21	5.888	90	2.46	48228
2.	257.18	178.22	0.693	435.40	78.96	0.181	372.54	62.86	5.93	90	2.46	64835
3.	257.18	178.22	0.693	435.40	78.96	0.181	374.01	61.39	6.092	90	2.46	77759

Table A-16 : Values of different parameters of 90^0 V-notched mild steel shaft for D = 32 mm, d = 16 mm and r = 1.25 mm for the maximum stress of 470.40 MPa and flexural stress ratio of 7.32.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress Ratio (R)	Notch angle (α deg.)	Stress concentration factor (K_t)	Life cycles (N_f)
1.	290.43	179.97	0.62	470.40	110.46	0.235	413.84	56.56	7.32	90	2.46	24996
2.	309.75	160.65	0.52	470.40	149.10	0.317	413.84	56.56	7.32	90	2.46	33247
3.	329.14	141.26	0.43	470.40	187.88	0.40	413.84	56.56	7.32	90	2.46	41787

Table A-17 : Values of different parameters of plain mild steel shaft for d = 16 mm for the maximum stress of 523.53 MPa and alternating stress ratio of 6.3976.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural stress ratio (R)	Life cycles (N_f)
1.	268.52	255.01	0.95	523.53	13.51	0.026	452.76	70.77	6.3976	39918
2.	304.78	218.75	0.72	523.53	86.03	0.164	452.76	70.77	6.3976	116402
3.	332.15	191.38	0.58	523.53	140.77	0.269	452.76	70.77	6.3976	240035

Table A-18 : Values of different parameters of 90^0 V-notched mild steel shaft for D = 32 mm, d = 16 mm with different notch-root radius for the maximum stress of 435.40 MPa and flexural stress ratio of 5.98.

Obs. no.	Mean stress (s_m MPa)	Alternating stress (S_a MPa)	Alternating stress ratio (R′)	Maximum stress (S_{max} MPa)	Minimum stress (S_{min} MPa)	Stress ratio (R″)	Flexural stress (S_f MPa)	Tensile stress (S_t MPa)	Flexural tress Ratio (R)	Notch angle (α deg.)	Notch-root radius (r mm)	Stress concentration factor(K_t)	Life cycles (N_f)
1.	257.18	178.22	0.693	435.4	78.96	0.181	372.54	62.86	5.98	90	1.25	2.46	64835
2.	257.18	178.22	0.693	435.4	78.96	0.181	372.54	62.86	5.98	90	1.50	2.30	71332
3.	257.18	178.22	0.693	435.4	78.96	0.181	372.52	62.86	5.98	90	1.75	2.17	79182
4.	257.18	178.22	0.693	435.4	78.96	0.181	372.52	62.86	5.98	90	2.00	2.07	115027

APPENDIX - B
Sample Calculation

Experiments were done by using tension-tension cyclic bending stressing pattern. One end of the specimen was kept fixed and inclined load was applied at the other end to develop required stress condition. Load P, loading angle θ and moment arm 'l' can be changed as required.

For sample calculation data for Fig. 14 is taken. The data are S_m =257.18 MPa, S_a = 178.22 MPa, S_f = 372.54 MPa, S_t = 62.86 MPa and N_f = 64835 cycles.

Calculation of total mean stress (S_m) : An initial tension was applied to the specimen before starting the machine. The specimen had the dimensions of D=32 mm, d=16 mm(0.629") r = 1.25 mm, l = 44.2 mm (1.74 inch) and $\theta = 15^\circ$. Initial tension was measured by the strain measurement of connecting bar. The strain was (31×10^{-6} m/m). From load- strain relation the corresponding load was found to be 775 kg. The steady tensile stress component (S_{to}) of this load at the notch section can be found as follows-

Steady tensile force = $P_0 \cos\theta = 775 \times 2.2 \cos 15^\circ = 1646.9$ lb.

Area = $\pi/4 \times d^2 = \pi/4 \times (0.629)^2 = 0.3107357$ in^2

$\therefore$ Steady tensile stress component due to initial load of 775 kg, S_{to} = 1646.9/(0.3107357 x 1000) = 5.31 ksi. =37.17 MPa

Steady flexure stress component of initial load of 775 kg at test section can be found from Mc/I as follows-

Moment (M) = $P_0 \sin\theta \times$ 'l' = $775 \times 2.2 \sin 15^\circ \times 1.74$ inch. = 771.77 in-lb.
c = d/2 = 0.629 inch./2 = 0.3145 inch.
$I = \pi d^4/64 = \pi/64 \times (0.629)^4 = 7.6837368 \times 10^{-3}$ (inch.)^4

$\therefore$ Steady flexural stress component of initial load of 775 kg, S_{f0} = Mc/(I x 1000)
= 31.43 ksi =220.01 MPa

Therefore total mean stress, $S_m = S_{t0} + s_{f0}$ = 5.31 + 31.43 = 36.74 ksi =257.18 MPa.

Total Alternating stress (S_a) calculation : Fluctuating load, for the calculation of alternating stress was measured by FFT analyzer. Before running the machine (without dynamic loading) there was a voltage signal of strain gage of connecting bar. The peak value was 0.0039 mv. Arranging the centrifugal mass and dead weight of upper load arm and during running of the machine, the peak value of voltage signal was recorded by FFT analyzer as 0.0254 mv. Therefore net peak for the fluctuating load calculation was 0.0254-0.0039 = 0.0215 mv. From voltage strain correlation the maximum strain corresponding to the voltage of 0.0215 mv is (2.15×10^{-5} m/m) and the maximum value of fluctuating load is 537.5 kg.

Therefore tensile stress component of fluctuating load (P_a) of 537.5 kg, S_{ta} $= (537.5 \times 2.2 \times \cos 15^\circ)/\{\pi/4 \times (0.629)^2 \times 1000\} = 3.67$ ksi =25.69 MPa

Flexural stress component of fluctuating load (P_a) of 537.5 kg, S_{fa} $= \{537.5 \times 2.2 \times \sin 15^\circ \times 1.74 \times (0.629/2)\}/\{(\pi/64) \times (0.629)^4\} = 21.79$ ksi =152.53 MPa.

$\therefore$ Total alternating stress, $S_a = S_{ta} + S_{fa} = 3.67 + 21.79 = 25.46$ ksi =178.22 MPa.

Therefore total flexural stress, S_f = steady flexure stress component + alternating flexural stress component $= S_{f0} + S_{fa} = 31.43 + 21.79 = 53.22$ ksi =372.54 MPa.

Total tensile stress, S_t = steady tensile stress component + alternating tensile stress component = $S_{t0} + S_{ta} = 5.31 + 3.67 = 8.98$ ksi = 62.86 MPa.

Flexural stress ratio (R) = (total flexure stress)/(total tensile stress) = S_f / S_t =53.22/8.98 = 5.93.

Alternating stress ratio (R′) = (Total alternating stress)/(Total mean stress)
$= S_a/S_m = 25.46/36.74 = 0.693$
Maximum stress = $S_m + S_a = 25.46 + 36.74 = 62.2$ ksi = 435.40 MPa.
Minimum stress = $S_m - S_a = 36.74 - 25.46 = 11.28$ ksi = 78.96 MPa.
Stress ratio (R″) = (minimum stress)/(maximum stress) = 11.28/62.2 = 0.181.

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