Republic of Iraq
Ministry of Higher Education and Scientific Research
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Numerical Simulation of Fatigue Life Estimation of Mild Steel Shaft under uniaxial loading

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A research paper submitted as partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering

The University of Misan

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بسم الله الرحمن الرحيم

{يَرْفَعِ اللَّهُ الَّذِينَ آمَنُوا مِنكُمْ وَالَّذِينَ أُوتُوا الْعِلْمَ دَرَجَاتٍ}

صدق الله العلي العظيم

[سورة المجادلة: ١١]

DEDICATION

I would like to express my deepest gratitude to those who have stood by me and supported me throughout the years:

To my supervisor. Ass.Lec Eman Mohamed, for his invaluable guidance, mentorship, and encouragement during this journey.

To my parents, for their unwavering love, support, guidance, and belief in me, even during the toughest times.

To my family, for their constant encouragement and understanding.

I extend my sincere thanks and deep gratitude to my respected professors, who generously shared their knowledge and guidance, and were always a source of inspiration and motivation. I also warmly thank my dear friends for their genuine support and constant encouragement, which played a significant role in overcoming challenges and achieving this accomplishment..

الاهــداء

إلَى حُجَّةَ اللهِ فِي أَرْضِهِ ،و عَيْنَهُ فِي خَلْقِهِ ، نُورَ اللهِ الَّذِي يَهْتَدِي بِهِ المُهتَدونَ ويُفَرَّجُ بِهِ عَنِ المُؤْمِنِينَ ، إلى المُهَذَّبُ الخائِفُ ، و الوَليُّ النّاصِحُ ، و سَفِينَةَ النَّجاةِ و عَيْنَ الحَياةِ ، الى مَوْلايَ يا صاحِبَ الزَّمانِ ، اهدي هذه الجهد المتواضع ، بغية رضاه.

إلى من غرَسوا في نفسي بذور الطموح، وسقوها بالصبر والدعاء والحنان، إلى من كانت دعواتهم سرّ نجاحي، إلى من سهِروا من أجلي، وفرحوا بفرحي، إلى والديّ الغاليَين، أهدي هذا العمل المتواضع عربون حب وامتنان، وعرفانًا بالجميل الذي لا يوفيه الكلام

ABSTRAC

Several mechanical engineering studied the varying on the load and its reason when the load were higher than the upper limit of the crack will be to start at the surface above these situation failure is initiate to occur and called fatigue failure. Fatigue failure offers no warning in progress and occurs suddenly which lead to accident and defeat of live.

This research Introduce a complete design and control data and other useful information of estimate the failure analysis and fatigue life for mild steel shaft under perpendicular cyclic bending moment.

The theoretical part also including the comparison between numerical results which be obtained by ANSYS 18.2 program and calibrated with standard device type (HI-TECH-HSM19Mk3) brass shaft which tests had been done on this machine takes high rang of stresses and life. The results that obtained should match the standard S-N curve.

Finally estimation the Basque's Equation Constants Results for brass and mild steel rotating shaft.

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INTRODICTION

Chapter One: Introduction

1.1 General

The failure of engineering materials is almost always an undesirable event for several reasons; these include human lives that are put in Jeopardy, economic losses, and the interference with the availability of Products and services. Even though the causes of failure and the Behavior of materials may be known, prevention of failures is difficult to Guarantee. The usual causes are improper materials selection and Processing and inadequate design of the component or its misuse. It is the responsibility of the engineer to anticipate and plan for possible Failure and, in the event that failure does occur, to assess its cause and then take appropriate preventive measures against future incidents. Fluctuation of stress interior materials structure and avoid it is the main aim of engineering mechanical industry because its lead to fatigue failure So, the fatigue is defined as phenomenon which caused by the repeated effort of loads to a component which finally consequences in the cracking and failure of that component which is meaning the fatigue of martial is appeared when subjected to dynamic stresses and it fails at stress below the yield point stresses [1]. To avoid the mechanical fatigue of the components its essential to description for the reaction of the materials to a number of loads possible to appeared for the period of the standard life time of the component [2].This is approved by limited mechanical test to determined S-N curves which represented the life time of materials as measured by the number of cycle (Nf) of fatigue life with the stress cycles required to cause failure [3].

1.2 Types of Loading

There are three main types of the dynamic fatigue loading on a specimen can be subjected: First is axial fatigue loading when the specimen subjected under tension or compression repetitively until failure. The second fatigue loading is a torsion loading where the torque is positioned at the specimen alternating until point of failure. Finally is fatigue due to bending which produces stresses are tension on the plane was decrease with growing distance into the element, and at certain point become compressive [4]. At the other hand the specimen may be subjected under combined load such as multi perpendicular bending moment which caused fatigue on the mechanical parts could be seen very clear in the bevel gear shaft.

1.3 Cycle Stress

In general, there are three types of cycle stresses caused the fatigue classified into: (a) fully reversed (b) repeated (c) random stress. The Figure (1.1) shown these types of cycle stresses **[5]**.

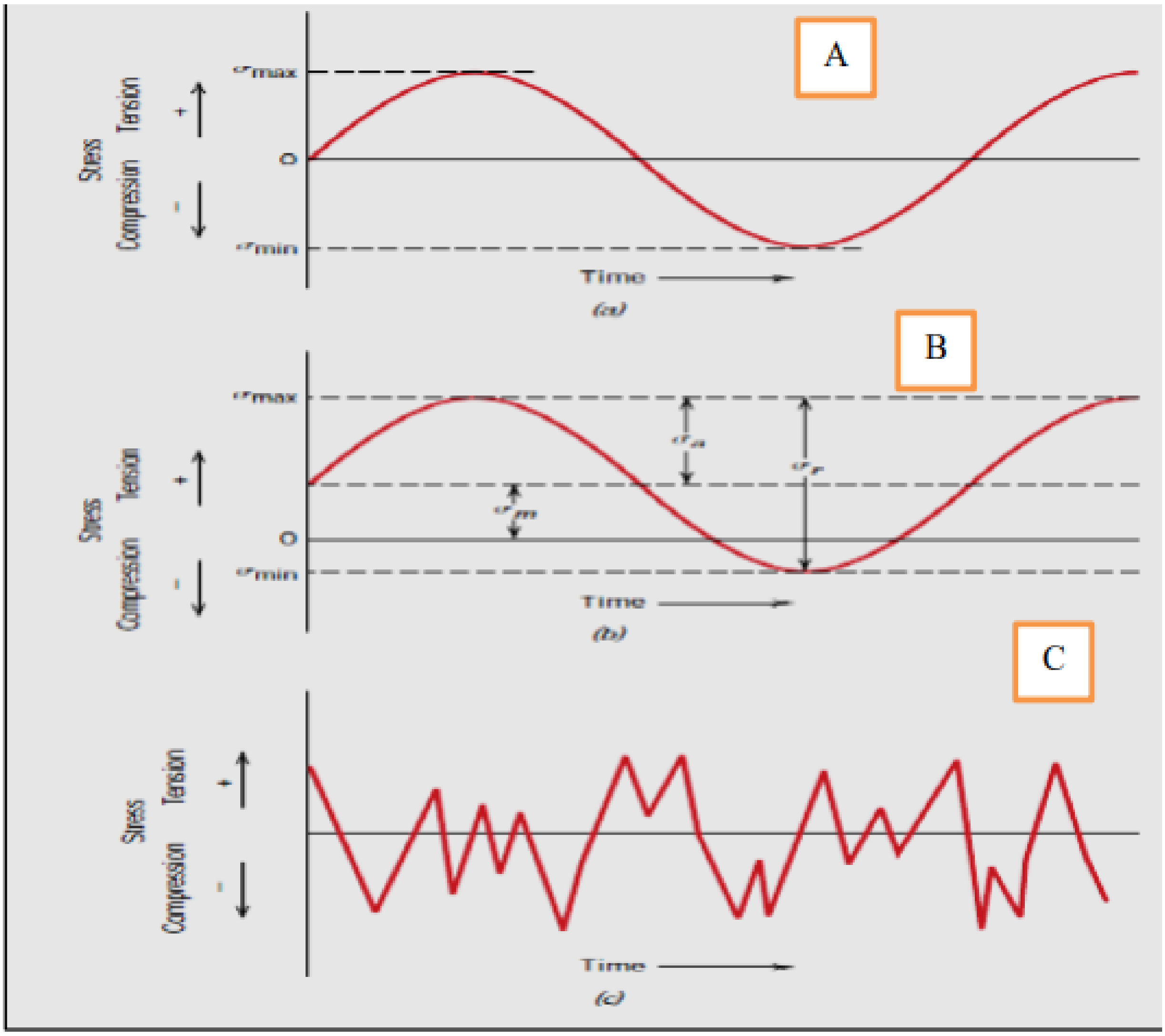

Figure 1 (1.1): Variation of stress with time that accounts for

1.4 THE S–N CURVE

As with other mechanical characteristics, the fatigue properties of Materials can be determined from laboratory simulation tests. A test apparatus should be designed to duplicate as nearly as possible the service stress conditions (stress level, time frequency, stress pattern, etc.). A schematic diagram of a rotating-bending test apparatus, commonly used for fatigue testing, is shown in Figure (1.3) the compression and tensile stresses are imposed on the specimen as it is simultaneously bent and rotated. Tests are also frequently conducted using an alternating uniaxial tension-compression stress cycle [3]. A series of tests are commenced by subjecting a specimen to the stress cycling at relatively large maximum stress amplitude (max), usually on the order of two thirds of the static tensile strength; the number of cycles to failure is counted. This procedure is repeated on other specimens at progressively decreasing maximum stress amplitudes. Data are plotted as stress S versus the logarithm of the number N of cycles to failure for each of the specimens. The values of S are normally taken as stress amplitudes on occasion, max or min values may be used. Two distinct types of S–N behavior is observed, which are represented schematically [5].

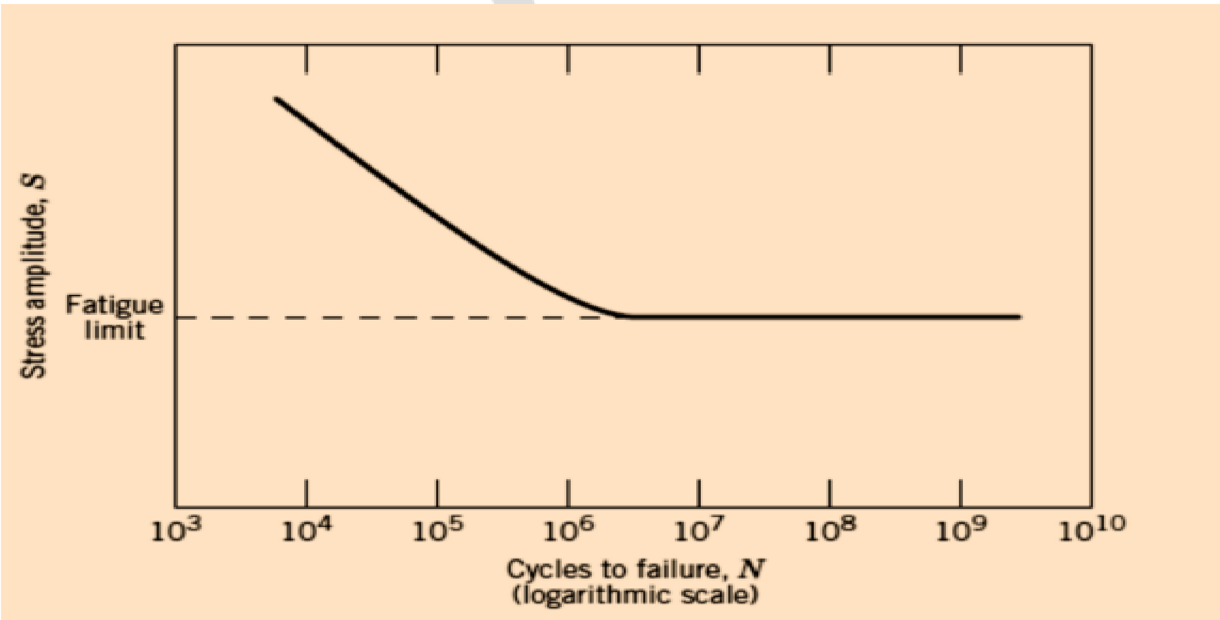

Figure 2 (1.2): Stress amplitude (S) versus logarithm of the number of cycles to fatigue failure (N). a material that displays a fatigue limit

As these plots indicate Two distinct types of S–N behavior are observed, which are represented. the higher the Magnitude of the stress, the smaller the number of cycles the material is Capable of sustaining before failure.

1.5 The Fatigue Life of the Shaft Estimation

Shaft is the important essential component used in practically mechanical system and machines. Past all power transmission parts shaft are the central component which should be designed with care to occur efficient operational of the machine.. The shaft is designed for different loading conditions used for many purpose of applications. The shaft design under multi loading is mainly critical problem in applied applications [8]. The multi loading subjected on the shaft at a actual point fluctuated among rotation of shaft which lead into fatigue. Even an enhancement component when subjected over and over again to loads of enough magnitude will in the end reproduce a fatigue crack in various highly stressed areas, usually located at the surface until final fracture [8]. In order to estimation the fatigue life of the shaft the finite element method analysis assistance locate the correct areas which presented the maximum loading stress, against the number of cycle curve (S-N Curve) is used to define the fatigue material properties.

1.6 Reserch Structural :

This thesis comprises from six chapters, which may be described briefly, and as indicated hereunder:

Chapter one provides a general introduction to illustrate some concepts that exhibit several beneficial issues of failure analysis and fatigue life of rotating shaft under an axial loading.

Chapter two reviews several historical studies that have relevance with the main topic's failure of the rotating shaft.

Chapter three includes theoretical consideration of fatigue analysis.

Chapter four involves the numerical analysis by using (ANSYS version 18.2) for model of the structure design and control of mild steel rotating under an axial loading. And discussion of the results.

Chapter five described the theoretical results and the recommendations of the aim of future work

Chapter Tow

LITERATURE REVIEW

CHAPTER TWO: LITERATURE REVIEW

General

In order to carry out this study, some technical research papers have been referred to indicate what has been done in investigating the phenomenon of fatigue failure in mechanical components. All basic concepts of this phenomenon were studied in the previous chapter, so the current chapter is to discuss some literature related to this study. These literature will be presented according to the time of the research by focusing on the studies that deal with designing and fabricating new rigs used to measure the fatigue in mechanical parts thereafter several remarks have been concluded accordingly.

2.1 Literatures Review for the Fatigue Analysis

Hereunder several important topics that are considered via researches to be described briefly as much as possible.

Łagoda, [11] (2001), studied The concerns verification of the energy model for the fatigue life determination under the uniaxial random and variable amplitude stress state. Application of the energy parameter for the fatigue life calculation using the algorithmic method under the random and variable amplitude uniaxial tension–compression proves that most results of the fatigue life calculation are included in the scatter band with the coefficient determined during the uniaxial cyclic tests. The strain energy density parameter seems to be efficient for the fatigue life determination in the ranges of high and low numbers of cycles. Conclusion of this study was consider the influence of the stress mean values on the fatigue life and other theory of fatigue failure simulation and experimental verification of usability of the new energy parameter for description of test results under uniaxial random and variable-amplitude tension–compression has been successful and may expect good results of verification of the parameter related to other materials.

Shin and Chen [12] (2004), studied the evaluation of fatigue crack propagation behavior using surface crack growth in a rotating bending rod has been attempted. Nine different rod geometries have been tested. The resulting fatigue crack propagation data is more sensitive to rod diameter than to rod length. Difference in crack growth behavior can largely be understood when crack closure is taken into account. All in all, the crack propagation data obtained from these small sized rods agree well with those obtained from standard testing employing compact tension specimens. A more precise picture about the crack growth behavior can be obtained if crack closure is considered. If crack closure is not monitored, rods with longer length and smaller diameter are more likely to give the conservative upper bound fatigue crack propagation behavior. Conclusion of this study was consider Crack growth behavior in the rod specimens underrotating bending is not sensitive to rod length except for the shortest (43 mm long) specimens with the smallest diameter (6 mm), which exhibited a slower rate than the longer rods with the same diameter.

Chen et al [13] (2006), studied the Low-cycle fatigue life tests are conducted on type 304 stainless steel with the loading sequence of (axial/torsional), (torsional/axial). The results show that cross hardening occurs in axial/torsional loading but not in torsional/axial loading. Much additional hardening is observed under out-of-phase loading. Prior loading in sequential inphase/out-of-phase loading produces a small degree of additional hardening in stress response in the latter loading compared with pure in-phase or out-of-phase loading. The (n1/N1, n2/N2) data points fall below the linear damage line in all cases. The double-linear damage rule, the damage curve approach and the plastic work model are found to give non conservative predictions. A modified damage model is proposed, which accommodates non proportional loading effects in sequential loading. Fatigue lives predicted based on the new model are within a factor-2 scatter band for all test conditions.

Beden et al [14] (2009), presented Structural components are frequently subjected to complex time histories of stress for which the prevalent mode of failure is fatigue. For reliable and cost-effective fatigue life calculations of metallic materials the systematic characterization of their fatigue behavior and the detailed understanding of basic fatigue mechanisms are of prime importance. Based on fatigue failure theories (stress-life theories such as; Goodman and Gerber, and strain-life theories ,this paper presents a technique to predict the fatigue life of a shell structure of different materials with application of measured variable amplitude loading. The finite element analysis technique was being used for the modeling and simulation. Numerical life prediction results of the shell materials, low and medium carbon steel (ASTM A533, AISI 1020, AISI 4340) are presented and discussed. There are many factors affecting the life predicted, a surface condition effect was shown here. Reasonable difference appears through the comparison of the above materials the Gerber mean stress correction model gives the conservative results when the time histories predominantly are zero mean. In the majority of the lifetime analysis, the deformation pattern remains stable and the material stiffness degrades due to the damage accumulated after each cycle. The fatigue resistance of steel depends more on material strength, the higher ultimate strength and hardness, the longer life.

Alaneme [15] (2011), studied the design principle was based on the adaptation of the technical theory of bending of elastic beams. Design drawings were produced and components/materials selections were based on functionality, durability, cost and local availability. The major parts of the machine: the machine main frame, the rotating shaft, the bearing and the bearing housing, the specimen clamping system, pulleys, speed counter, electric motor, and dead eights; were fabricated and then assembled following the design specifications. The machine performance was evaluated using test specimens which were machined in conformity with standard procedures. It was observed that the machine has the potentials of generating reliable bending stress –

number of cycles data. Also the machine has the advantages of ease of operation and maintenance, and is safe for use

Ince, and Glinka, [16](2014), involved the two different forms of an original multiaxial fatigue damage parameter related to the maximum fatigue damage plane are proposed for performing fatigue life prediction under various loading conditions loadings. The proposed fatigue damage parameters have been applied to uniaxial and multiaxial loading conditions for geometrically different bodies. Both the damage parameters are correlated to sets of experimental data published in the literature to verify the prediction accuracy of the damage parameters. the damage parameters show reasonably acceptable correlations with experimental fatigue data of steel notched shafts subjected to proportional and non-proportional loadings. Conclusion of this study can be attributed to the fact that less accurate notch stress and strain analysis of the notched shaft obtained from the approximate analytical approach.

Shreyas et al [17] (2015),studied testing different materials for their fatigue behaviour has been a major requirement before consider the material for use in any environment with continuous fluctuating load. There has been major evolution in the design of fatigue testing machines for over a century now. But the concept that was lacking in the testing industry was that of a fatigue testing machine that can simultaneously test the fatigue strength of two specimen. In this paper, the attempt to bring this concept to life has been explained. The design of this fatigue testing machine is different from other fatigue testing machines because it can hold two specimens at any given time. Conclusion of this research were : Specimens made of two different materials can be tested simultaneously and a basic comparative study can be made there itself and Takes comparatively less time for fatigue testing of two specimens.

Al-Shama [18](2016), investigated the numerical and experimental behavior of metal under multiaxial fatigue by arrangement of cycling bending with torsion loading , the specimens of metal using (L690M steel, C26800 brass, copper, 2117- T4 Aluminum) were recognized individuality and multiaxial fatigue life which is significant in sensible use as that metals at obtained the stress ratio constant for all metals, the studied shown that copper is the largely resistant to metal fatigue failure as soon as subjected to multiaxial loading and compared the result which were obtained from theoretical study and experimental work, the number of cycle to failure in instance of reversed bending fatigue test that the maximum percentage error is (22.4%) in (C26800) brass metal, and for the number of cycle to failure in case multiaxial bending with torsion fatigue is found that the maximum percentage error is (21.6%) in (C26800).

Chirag and Patel [19] (2016), designed the Fatigue test device are first classified in accordance with a number of features which include purpose, type of loading, and method of load application and transmit as well as control system. It is built on the understanding obtained from a vibration analysis of the force transfer function from the load cell of the machine to a fabricated calibration bar. Performing the dynamic characterization in practice consists of making a frequency sweep with the strain gage equipped calibration bar. All the proceeding steps of analysis required to predict an upper bound of the linear measurement error is conveniently handled by software. The machine made is concept machine and not taking into account any of the features like efficiency & output of the machine, servicing cost of the machine etc. The machine made by us is based on the machines used in the different industries, obtain the s-n curves on the basis of the testing of different types of material which are being tested. The result that get after the testing should match the standard s-n curve of that material. The conclusion is given that in the starting in initial condition at normal motor speed there is no breakdown in the component, But after some time if apply some load to the down side of the component then crack is initiated and at certain point and time there is sudden breakage in component.

Talemi et al [20](2016), investigated the effect of pre-bending procedure of HSS subject to low cycle fatigue loading conditions,. For this objective designed a original test set-up to obtained the results of pre-bending stress when the fatigue load had been applied. after bending and fatigue testing numerical procedure advance was applied to specimen the behavior of experienced material. The urbanized finite element method supplied additional information about the multiaxial stress. The main objective of this research study was to investigate the effects of pre-bending process on low cycle fatigue behaviour of different steel grades

Kopas et al [21](2016), the main objective of this paper the authors present and discuss results of fatigue life tests performed for reinforcing steel bars which is one of the most widely used construction material in the world. Endurances can be influenced by type of steel, geometry and size of the bars, nature of the loading cycle, welding and presence of corrosion. The experimental part deals with fatigue testing of specimens for identification of the strain-life behaviour of reinforcing bars and determining the number of cycles to failure. Conclusion of this study was The fatigue resistance of reinforcing steel increases with decreasing stress amplitude continuously in the cycles of number gion. It is possible to perform fatigue tests of reinforcing bars if the appropriate specimen preparation methodology is applied.

Engel, and Al-Maeeni, [22](2017),studied the failure modes of the shaft of the rotary draw bending machine are inspected. Furthermore, stress and deflection analysis of the shaft subjected to combined torsion and bending loads are carried out by an analytical method and compared with a finite element analysis method. The theoretical fatigue strength, correction factors, and fatigue life sustained by the shaft before damaged are estimated by creating a stress cycle (S-N) diagram. In conclusion, it is seen that the shaft can work in the second life, but it needs some surface treatments to increase the reliability and fatigue life. In conclusion, it is seen that the shaft can work in the second life, but it needs some surface treatments to increase the reliability and fatigue life

Aldeeb, and Abduelmula, [23](2018), discussed the test of the fatigue life of S275 mild steel at room temperature. Mechanism can be unsuccessful under cyclic loading for the period of time which is defined as the fatigue phenomenon. To prevent fatigue produce failures the behavior of material must be considered to establish the endurance. Limit of the metal in support of safe design and unlimited life as a result leading to reduced the efficient cost and loss in person life. The fatigue behavior of S275 mild steel was examined and investigated. The procedure of fatigue testing was related with constant load amplitude .The experimental results had been compared with the results of a Finite Element Analysis (FEA) by simulation software. The test results of the endurance fatigue limit of S275 mild steel (195.47 MPa).

Odaa, and Kahtan, [24](2018),studied the numerically simulation analysis of the fatigue behavior of stepped shaft made of Carbon steel SAE 1045_390_QT under a biaxial loading (torsion-bending moment) and uniaxial loading (only bending moment case), (only torsion case). This work included the complete analysis of stepped shaft by using ANSYSWORKBENCH 14.5 and the fatigue analysis using n Code Design Life 14.5,the numerical results showed how important is the effect of biaxial loading on fatigue life. Conclusion of this study was Since the fatigue life under biaxial loading conditions much lower than that of uniaxial loading, the fatigue analysis of mechanical parts which loaded by multi axial loadings must consider biaxial fatigue analysis and The effect of torque on the minimum fatigue life location is more than that of the bending, since the torque only condition gives difference location of the fatigue life

Ali, et al [25](2019), discussed this research work had been done in order to design and fabricate “Rotating Bending Fatigue Testing Machine” which is simple to understand, perform, portable, safe for use and more economical than others. A rotating bending fatigue testing machine was developed by keeping in mind the basic concepts of technical theory of bending of elastic beam. Different specimens of 6mm to 8mm diameter of different materials were tested under various loads to analyse the performance of this machine. The experimental readings and theoretical calculations indicate that the results were promising to agreement of the objective had been achieved. Conclusion of this research The advantage of the present design is the easiness of its modeling and ease of understanding. By using simple mechanism, the cost of this machine has been decreased to Rs. 31,500 (228.28$) without affecting the experimental results

Al-Alkawi and Faal, [26](2020), the main objective in this study design and construct a fatigue testing machine that is capable of examining the fatigue strength and life of various specimens from metals, such as aluminum alloys, brass composite materials and mild steel. The results that we obtained should match the standard S-N curve of that material. Five S-N curve tests were performed. These tests were

(RT), 60 days oil corrosion, SP (shot peening) prior to corrosion, elevated temperature (ET) 200 oC and finally SP prior to (ET) before using the fatigue test machine. A material selected which is AA6061-T6 to be the testing material. Mechanical properties tests were carried out under five conditions as mentioned before. The results revealed that. The Yield stress (YS), ultimate tensile strength (UTS), modulus of elasticity (E), ductility and hardness (HB) reduced by 4.65%, 6.41%, 2.85%, 659%, and 5.88%.

2.2 Summary

General points can be summarized with respect to above mentioned literatures:

1- To understand the future fatigue behavior of the used machine components, it is important to investigate the possible causes of machine parts failure through design, surface, and material inspections.

2- Rotating shaft is subjected to different conditions lead to fatigue in its components. Many researchers have worked on designing rotating shaft subjected to different loads in various application . The stress on the shaft at a particular point varies with rotation of shaft which it leads to cause fatigue.

3- The design of a rotting shaft that will be subjected to cyclic loading can be approached by adjusting the configuration of the part so that the calculated stresses fall safely below the required line on an S-N plot.

4- Verification of fatigue life of mild steel was conducted by using the FEA results. The FE analysis can predict the fatigue life with stress details in better and fast way, leading to save time and cost in industrial construction.

5- There are lacks of information regarding the effect of specimen subjected to two perpendicular bending moments. steel was conducted by using the FEA results. The FE analysis can predict the fatigue life with stress details in better and fast way, leading to save time and cost in industrial construction. 5- There are lacks of information regarding the effect of specimen subjected to two perpendicular bending momen

Chapter Three

THEORETICAL ANALYSIS

Chapter Three Theoretical Analysis

3.1 Introduction

This chapter contains two main parts: the first one is about theoretical considerations of fatigue analysis (Analytical or Empirical method), and presents a mathematical model of the mild steel shaft under multi perpendicular cyclic bending test. In fact, the analytical section is based on the fundamental of fatigue and some equation has been derived to calculate fatigue life and endurance limit theoretically. The second part is a numerical approach conducted using the finite element method in ANSYS Software.

3.2 Theoretical Considerations

Generally, fatigue is a problem occurring in 90% of moving mechanical components. Metal fatigue includes the initiation, propagation of crack and fracture. Residual stresses remaining from manufacture further play a significant part in increasing metal fatigue. Metal fatigue takes on greater significance with improving the level of yield strength due to greater strength alloys metal, and give excellent fatigue crack growth resistance, compared to materials of lower strength, yet are predicted to provide higher working stresses. The outcome of this fact is a tendency to overwork high strength alloys in fatigue conditions, therefore expressing fatigue breakdown [27]. Fatigue is by far the most usual cause of failure for mechanical parts, it is the most worry one, as it “happens suddenly without any obvious plastic deformation without warning”. It begins when a load is very adequate to create plastic deformation on a part, even when it occurs just at a small volume of material, limited plastic

deformation can happen at the highest stress location. When such load is repeated cyclically, the damage grows and ultimately a crack created [28]. If the cyclic load remains, the crack length increases until the stress on the part becomes sufficient to induce its unexpected break.

3.2.1 Fatigue Methodologies

There are three methods for studying fatigue behavior through predicting fatigue life as a number of cycles to fracture in specific loading level, where the fatigue life ($1 \leq N \geq 10^4$) is classified into the low cycle fatigue region, while ($N > 10^4$) in the high cycle fatigue region these methods are as follows:

1- Stress – life method.

2- Strain – life method.

3- Fracture mechanic method.

Where stress life method depends on stress level only. It is the least accurate especially for low cycle application, most traditional and simple. Also, it can be applied for a wide range of applications of high fatigue cycle and infinite cycle with being in safe region. In strain life method , strain and stress are considered for predicating fatigue life , hence, it needs more detailed analysis of plastic deformation and is used for low cycle formed and employed to predict crack with respect to stress intensity, and thus, it is suitable for practical application [29]

3.2.2 Mathematical Modeling

A number of applied function and moreover many fatigues theory test on materials includes cycling between maximum and minimum stress values which are constant. This is defined as constant amplitude stressing. The stress range ($\Delta_{\sigma}= \sigma_{max}- \sigma_{min}$) is the variation between the maximum and the minimum levels. Averaging the maximum and minimum values gives the mean stress, σ_m. The mean stress can be zero as in figure (3.1a) if it is not such as in figure (3.1b). A nonzero mean stress, which is the variation about the mean. Mathematical expressions for these basic definitions are [30].

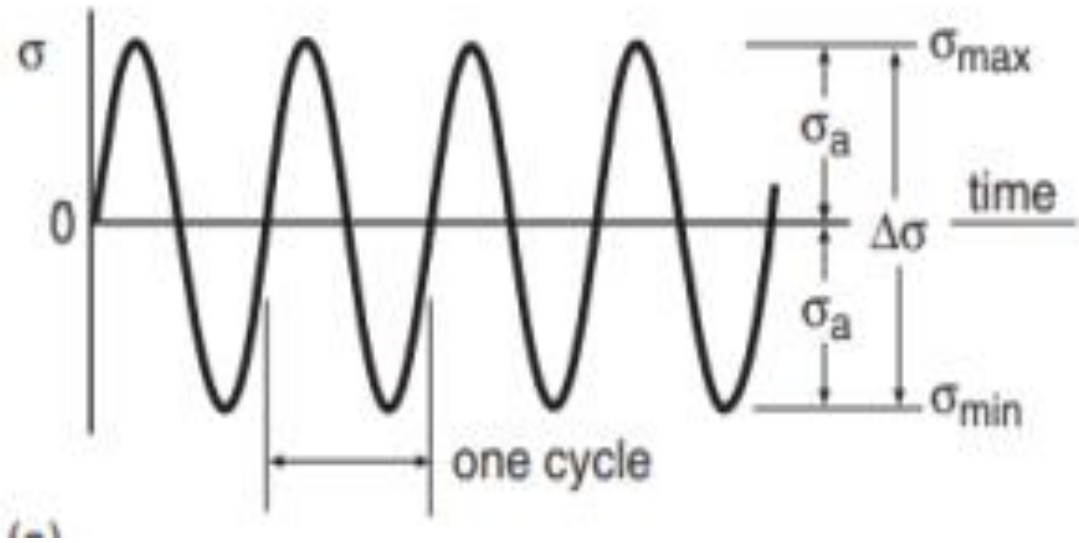

Figure 3 (3.1): Constant amplitude cycling and the associated nomenclature Completely reversed stressing

Figure. $\sigma m = 0$ [30]. (a)

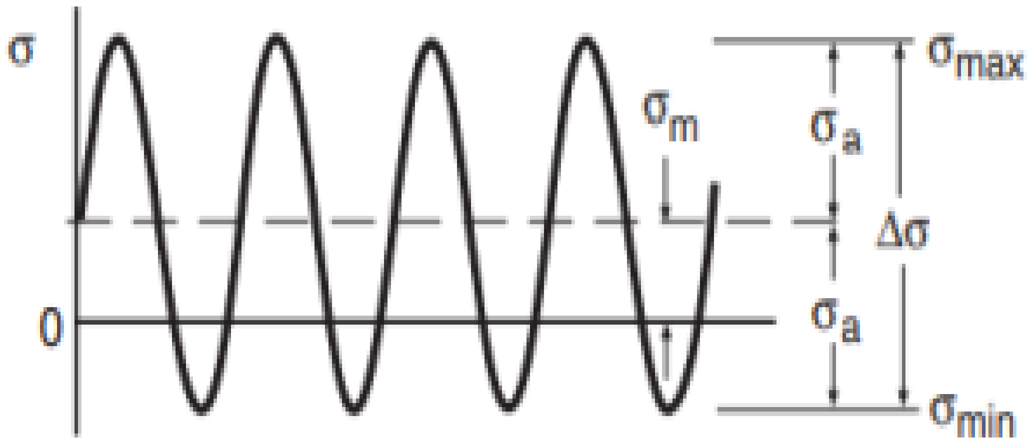

Figure 4 (3.1b): A nonzero mean stress

Figure $\sigma\ m$. [30]

(b)

Maximum stress $= \sigma_{max}$
Minimum stress $= \sigma_{min}$

$$\sigma_{\mathrm{m}} = \frac{\sigma_{max} + \sigma_{min}}{2}$$

Where:-

$\sigma_{\mathrm{max}} = \underline{1 + \mathrm{R}}$

Stress rang

$\Delta_{\sigma} = \sigma_{\mathrm{max}\_\text{-}}\ \sigma_{\mathrm{min}}$

Stress amplitude $= \sigma_a$

$$\sigma_{\mathrm{a}} = \frac{\sigma_{\mathrm{max}} - \sigma_{\mathrm{min}}}{2}$$

$$\sigma_a = \frac{\Delta\sigma}{2}$$

The stress ratio $\mathrm{R} = -1$ and $\mathrm{R} = \cdot$ are two common reference test conditions used for obtaining fatigue properties.

If $\mathrm{R} = -1, \sigma_{\text{mean}}$. the fatigue stress curve is called the fully reversed case since $(\sigma_{\min} = -\sigma_{\max})$

If $\mathrm{R} = \cdot$, where $\sigma_{\min} = \cdot$, the fatigue stress curve is called zero-to- tension stressing as shown in figure ($\Gamma.1\mathrm{c}$) .The R ratio is regularly less than '. May be larger than ' when $\sigma_{\max}$ is negative. A simply rotate cyclic examination form a relationship to $\mathrm{R} = -1$ a periodic cyclic examination in tension to $\mathrm{R} = \cdot$ in compression to $(R = \infty)$.

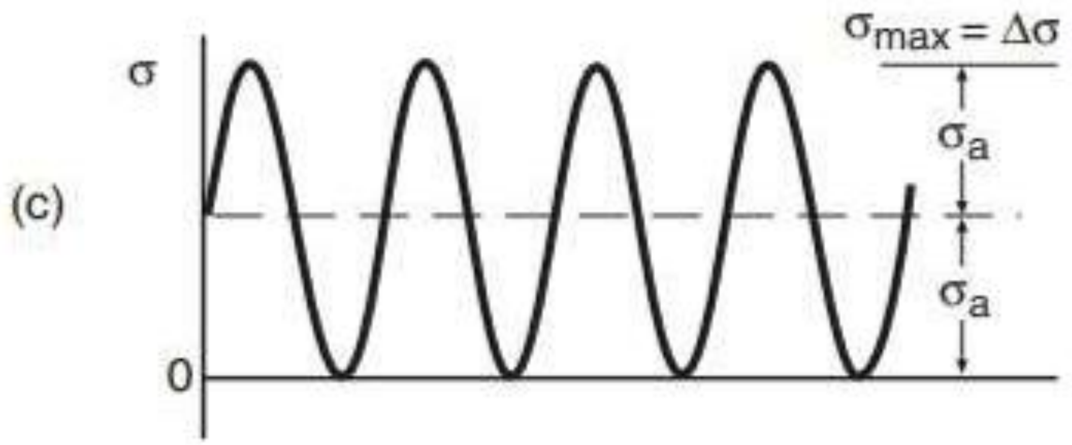

Figure 5 (3.1c): zero – to – tension stressing [30]

Stress ratio is equal as:

$$\mathrm{R} = \frac{\sigma_{\min}}{\sigma_{\max}}$$

The Amplitude ratio (A) is equal to as:

$$\mathrm{A} = \frac{\cdot\, \sigma_a}{\sigma m}$$

3.2.3 High Cycle Fatigue (HCF)

It is defined as a type of fatigue, which happens in advance failure. The material is subjected to a relatively critical quantity of cycle. In general, the fatigue procedure of high cycle fatigue occurs when the number of cycles required for failure exceeds 10^4 [31].

3.2.4 Low Cycle Fatigue (LCF)

This type of fatigue often happens after comparatively few cycles of less than 1. . cycles and it is known as low cycle fatigue. The mechanism of crack development in materials subjected to (Low-Cycle fatigue) is nearly identical to the crack expansion under constant stress of the material. In addition, the deformation is usually plastic. As the plastic strains in low- cycle fatigue are usually greater than in high-cycle fatigue, the surface deformations in material are not significant like the properties of bulk materials. [31].

3.2.5 Bending Moment of Cantilever

The equation refers to bending moment and the radius of cantilever shaft can be written as the formula based on simple theory of shaft. [32]

$$\frac{M}{I} = \frac{E}{r} = \frac{\sigma}{y}$$

Where:

M is the bending moment.

is the second moment of area. E is the modulus of elasticity. R is the radius curvature of cantilever. The radius curvature can be defined by following equation:

$$\frac{1}{r} = \frac{d^2y}{dx^2}$$

Hence $\frac{d^2x}{dy^2} = \frac{M}{EI}$

or $M = EI\frac{d^2y}{dx^2}$

3.2.6 Bending Stress in shaft

The maximum value of bending stress in a cross section of rotating bending shaft device is shown in figure. (3.2) [33]

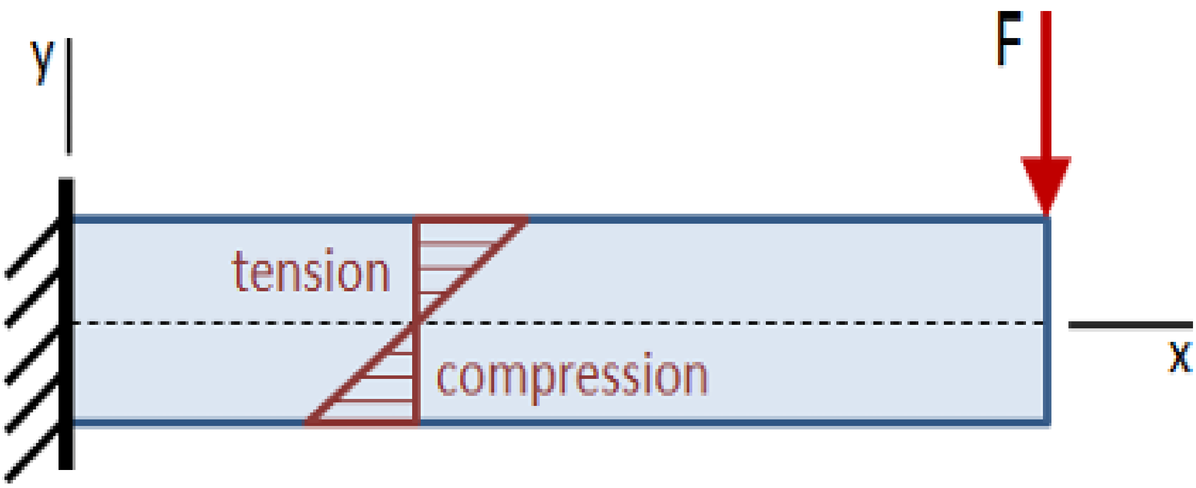

Figure 6 (3.2): Bending stress across the shaft diameter. [34]

The bending moment (M) which can be calculated from: [34]

where

M = $F * L$

M is the moment bending (N.mm).

F is the applied force (N)

L is the standard moment arm of fatigue device

$\sigma_I b = My$

Because there is multi perpendicular bending moment subjected on the shaft. Therefore, the bending stress has been written in two direction as follows:

$$\sigma_b = \frac{M_{zz}\gamma}{l_{zz}}$$ **, bending stress around z**

$$\sigma_b = \frac{M_{yy}Z}{l_{yy}}$$ **, bending stress around z**

Where, for circular cross-section shaft:

$$\gamma = z = r = \frac{d}{2}$$

$$l_{zz} = l_{yy} = \frac{1}{64} = \frac{\pi d^4}{2}$$

Now the equation of stress bending can be derived as

$$\sigma_b = \frac{F*L*Y}{\frac{\pi d^4}{64}} = \frac{F*L*\frac{d}{2}}{\frac{\pi\, d^4}{64}}$$

$$\sigma_b = \frac{F * L * 32}{\pi d3}$$

the stress of bending will be written as form fowling

σbzz=20 $*$ Fzz

σbyy=20 $*$ Fyy

3.2.7 The S-N Curve

The S-N curve measures the relation between the fatigue life against stress amplitude as shown in figure (3.3). Fatigue testing at little amplitude was usually concluded around 10^6 to 10^8 cycles. As the result of testing in this state is time uncontrollable, and often the S-N curves approach a constant value and the number of cycles at the point is usually defined the endurance limit, and the related stress amplitude is fatigue limit.Theoretical studied have found that fatigue failure may be occurred in the level at 10^6 to 10^8 cycles even for horizontal specimens subjected to constant amplitude stress. It is observed that the relationship between completely reversed stress and life N **[**35**]**

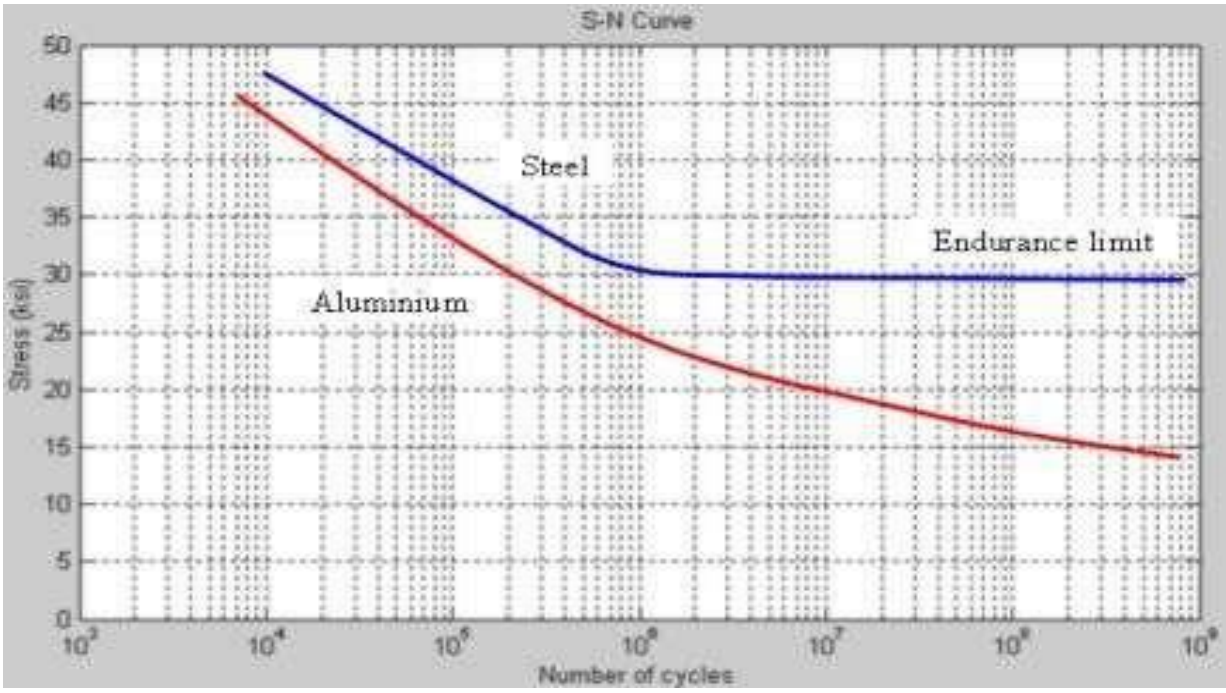

Figure 7 (3. 3): The relationship between stress and number of cycles to failure. [36]

3.2.8 Power Law Regression (Basquin's equation)

Fatigue life can be estimated by using the stress number of cycles to failure relation, called S-N curve, all the fatigue S-N curves of the metal at room temperature and elevated temperatures can be analyzed based on Basquin's equation form as [37]

$\sigma f = (Nf\ )\ \alpha$

Where :

σ_f is the reversed stress amplitude.

N_f is the number of cycles to failure.

A is material constant.

This known as Basquin's equation constantans A and α can be determined from [37]:

$$\sigma = \frac{h\ \sum_{i=1}^{h} \log \sigma_f \log N_f - \left(\sum_{i=1}^{h} \log \sigma_f\right)\left(\sum_{i=1}^{h} \log N_f\right)}{h \sum_{i=1}^{h} \left(\log N_f\right)^2 - \left(\sum_{i=1}^{h} \log N_f\right)^2}$$

$$\log A = \frac{\sum_{i=1}^{h} \log \sigma_f - \sigma \sum_{i=1}^{h} \log \sigma_f}{h}$$

Where:

h is the number of stress levels.

α is the slope of S- N curve

N_f is the number of cycles to failure.

3.2.8 Fatigue Failure Criteria

There are different theories can be used to study the effect of mean stress. The theories that are commonly used to predict fatigue life under fluctuating are:

1. The Goodman theory.
2. The Soderberg theory.
3. The Gerber theory.

All equations for different theories can expressed [32]

$$\frac{\sigma a}{\sigma e}+\frac{\sigma m}{\sigma u}=\frac{1}{S.F}$$ **Goodman's**

$$\frac{\sigma a}{\sigma e}+\frac{\sigma m}{\sigma y}=\frac{1}{S.F}$$ **Soderberg's**

$$\frac{\sigma a}{\sigma e}+\left(\frac{\sigma m}{\sigma u}\right)^2=\frac{1}{S.F}$$ **Gerber's theory**

Where:

σm :Mean stress

σe: The endurance fatigue limit σy

:Yield stress

σu: The ultimate tensile strength

In the current work, a specimen of mild steel has been used. It is ductile metal where the Soderberg method is preferred to be used for this type of metal. Soderberg theory is most common compared with other theories, because it is the only one completely below the yield line [40] shows in figure (3.4)

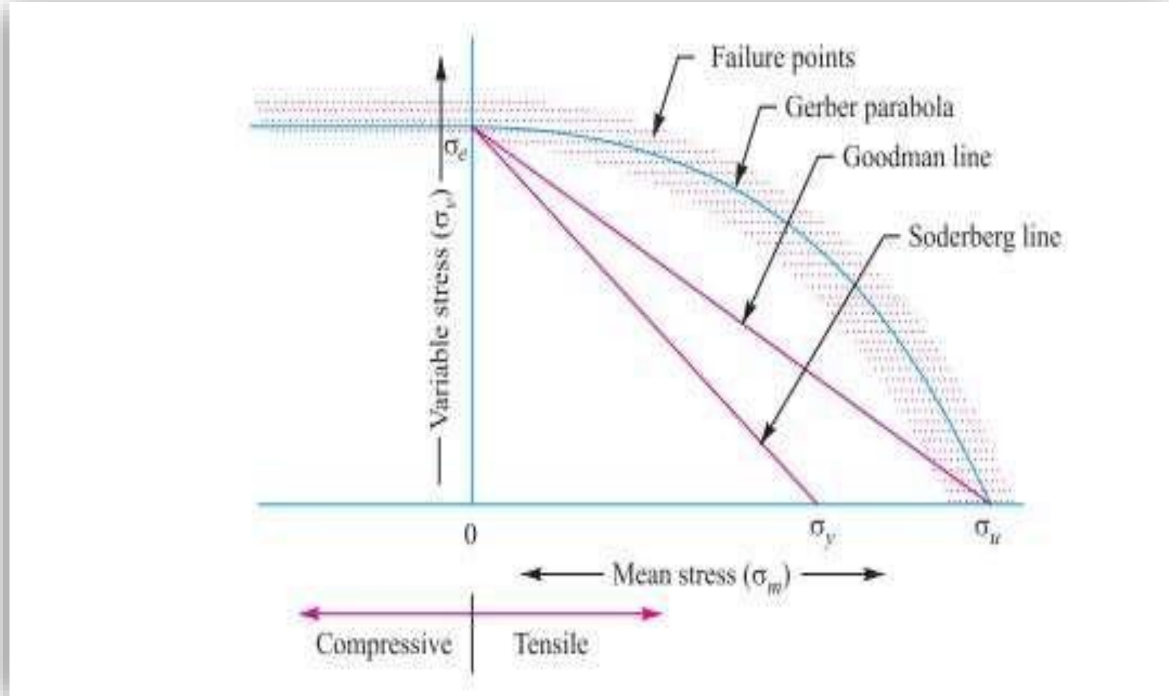

Figure 8 (3.4): Fatigue diagram showing various criteria of failure [41]

3.2.9 Palmgren-Miner rule (PMR)

This rule states the prediction of fatigue failure which is performed when the summation of the entire fatigue damage event by different stress level becomes unity as:

$$\sum \frac{N}{N_f} = \text{Soderberg's theory}$$

where:

n is number of cycles at given stress.

NF is number of cycles to failure obtained from S-N curve.

If equation (1-6) becomes equal to 1 the failure will occur and can be written as:

$$\sum \frac{n}{n_f} = 1$$

The main disadvantages in the above equation can be summarized thus:

1- The load sequins effect can't take into account.
2- The effect of the service environment like, corrosion fatigue creep interaction effect, erosion effect, etc is not taken into account
3- The surface treatment effect like shot preening is not considered

Chapter Four

NUMERICAL ANALYSIS

Chapter four: Numerical Analysis

Numerical Analysis

The most accurate numerical method used to solve engineering problems is the Finite Element Analysis (FEA).Currently, there are many commercial software Used this method for solving such problems. The most powerful software is ANSYS. Therefore, ANSYS workbench is adopted in this study.

4.1 Numerical Analysis of Brass Metal Specimen

Fatigue analysis in FE offers a way to predict the life of a material, or structure, when it is subjected to cyclic loading over a period. Use of FE analysis assists to locate the exact areas that accumulate the maximum amount of stress, or undergo highest deformation under cyclic loading. ANSYS Workbench is FE software, which defines a special fatigue tool for fatigue analysis of materials. The stress against number of cycle's graph (S-N curve) is used to define the fatigue material properties for stress life. The following steps obtain the life cycle of fatigue for the brass metal:

4.1.1 Geometry of the shaft

In order to perform the finite element analysis, it needs to make the specimen geometry with its elements this geometry has been created by design new modeler by soil works programming as shown in figure (4.1). It's fed the geometry of specimens and dimension from the standard brass specimen are supplied with rotating rig test (HI-TECH) as shown in figures (4. 2).

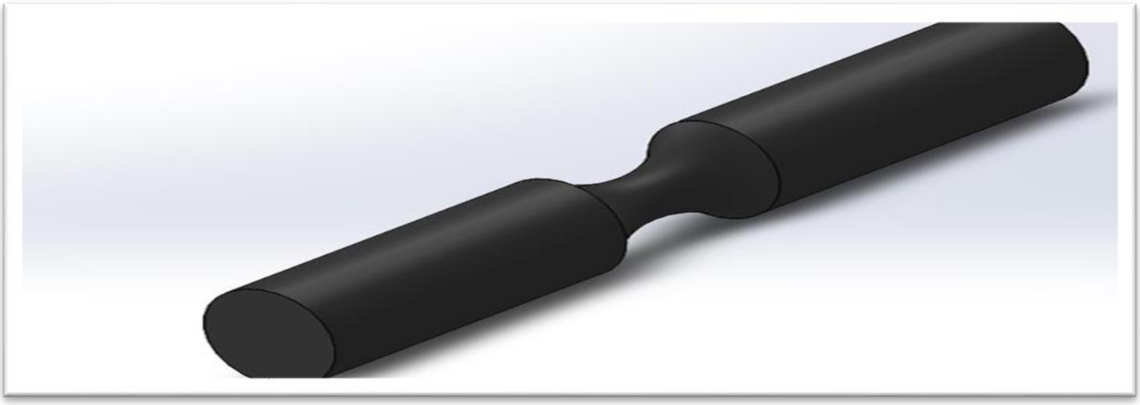

Figure 9 (4.1): The geometry of the shaft

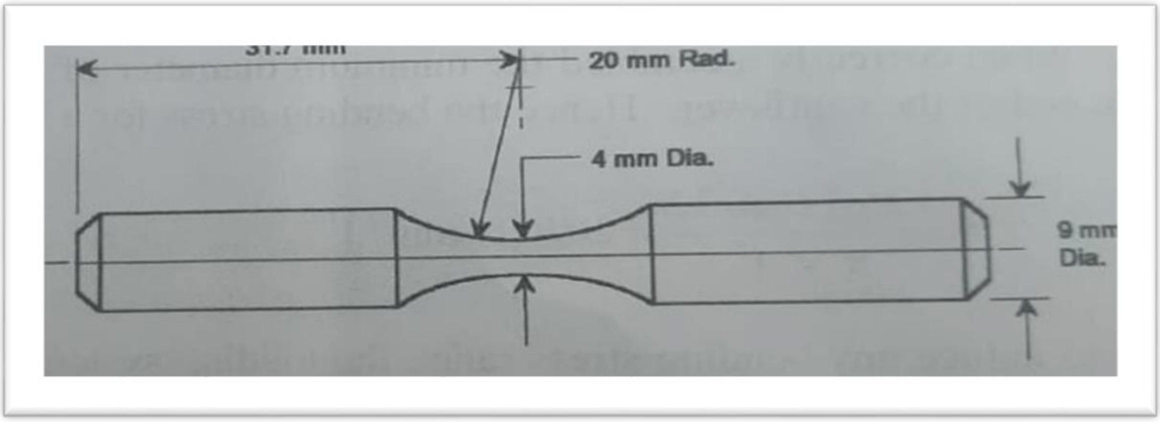

Figure 10 (4.2): Dimensions of fatigue brass specimens

4.1.2 Static structural

This is prepared by dragging "Static Structural" from "Toolbox" to project Schematic and then relating the created box to geometry box

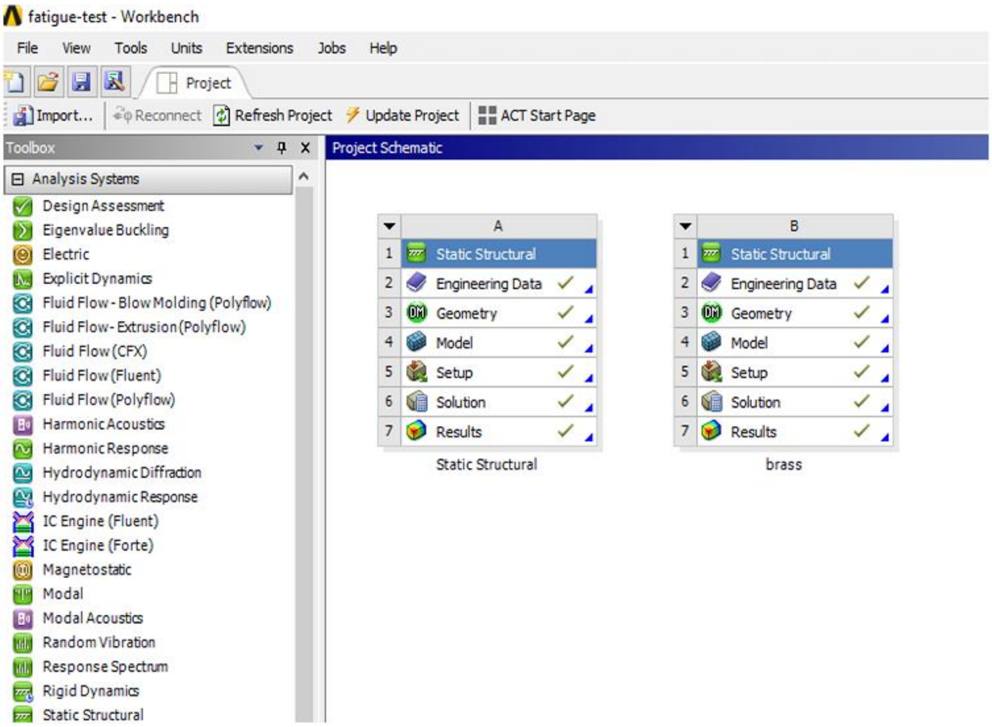

Figure 11 (4.3): The static structure modeling.

4.1.3 Material properties

The martial used in the most experimental work were from brass and mild steel and selected from available standard types, with different chemical and mechanical Compositions as shown in in APPEDIX A and APPEDIX B [38]. The selected mild steel and brass as follows: Table (4.4) shows listed the mechanical Properties of mild steel (St37-2(DIN1.0037)) and table (4.5) shows listed mechanical properties of standard brass metal.

Table 1 (4.4) Mechanical properties of mild steel St37-2(DIN1.0037)

$\sigma_u(MPa)$	$\sigma_y(MPa)$	$E(GPa)$	$G(GPa)$	Poisons ratio (v)
640	425	207.88	74.463	0.3

Table 2 (4.5) Mechanical properties of standard brass metal

$\sigma_u(MPa)$	$\sigma_y(MPa)$	$E(GPa)$	Poisons ratio (v)
350	180	97	0.31

These properties are applied into ANSYS software as shown in figure (4.6). The material properties can be fed to the model using the following commands sequence: Engineering Data, General Material, Add Material, and Input Properties Save Data

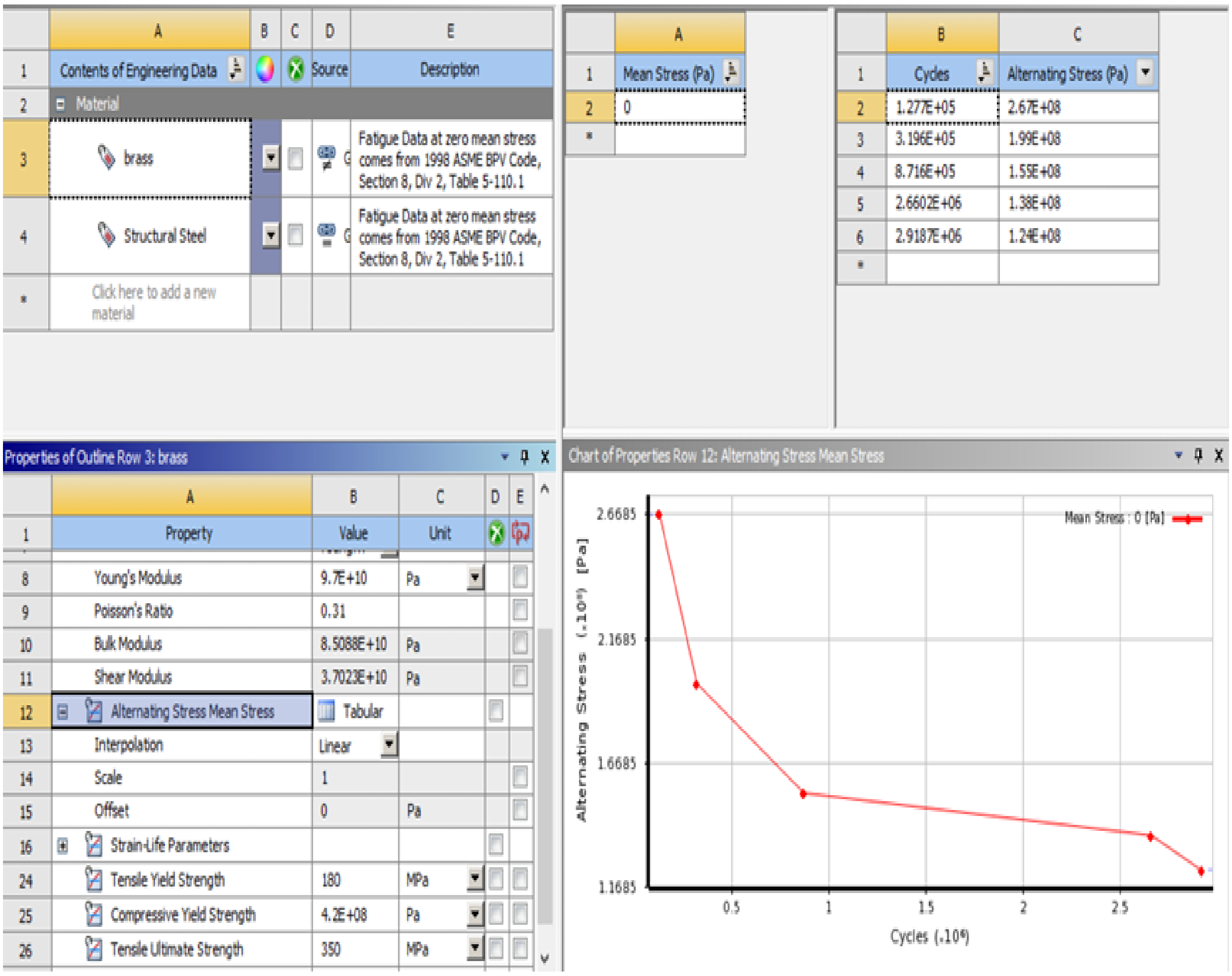

Figure 12 (4.6): Material properties modeling

4.1.4 Meshing process

The specimen can be meshed using the following running: Mesh, Sizing, Element Size, Input Number and element shape hexahedron. The meshed model can be seen in figure as shown below (4.7).

validation of Meshing ANSYS Workbench brings meshing tool platform and has a variety of different meshing method for three-dimensional geometry. The validation process its important point, where More than one attempt was made with the reduce the element area where started from 5mm to 0.5 mm. We noticed that there is no change in the value of the max stress. Hence when the local unit element size 0.5 mm, the unit number is 386908, node number is (93783) while the value Von Misses stress was (257 MPa) as shown in figure (4.7b).

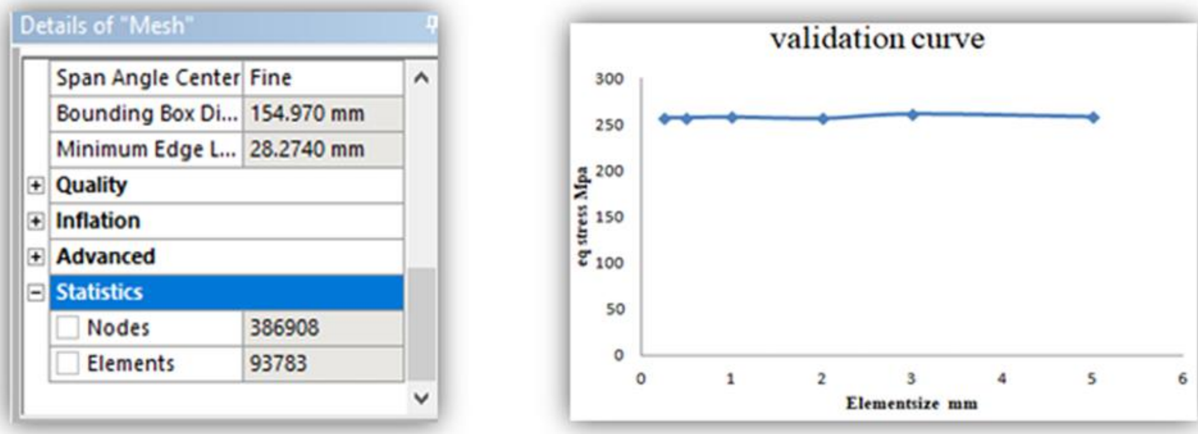

Figure 13 (4.7b): Validation curve

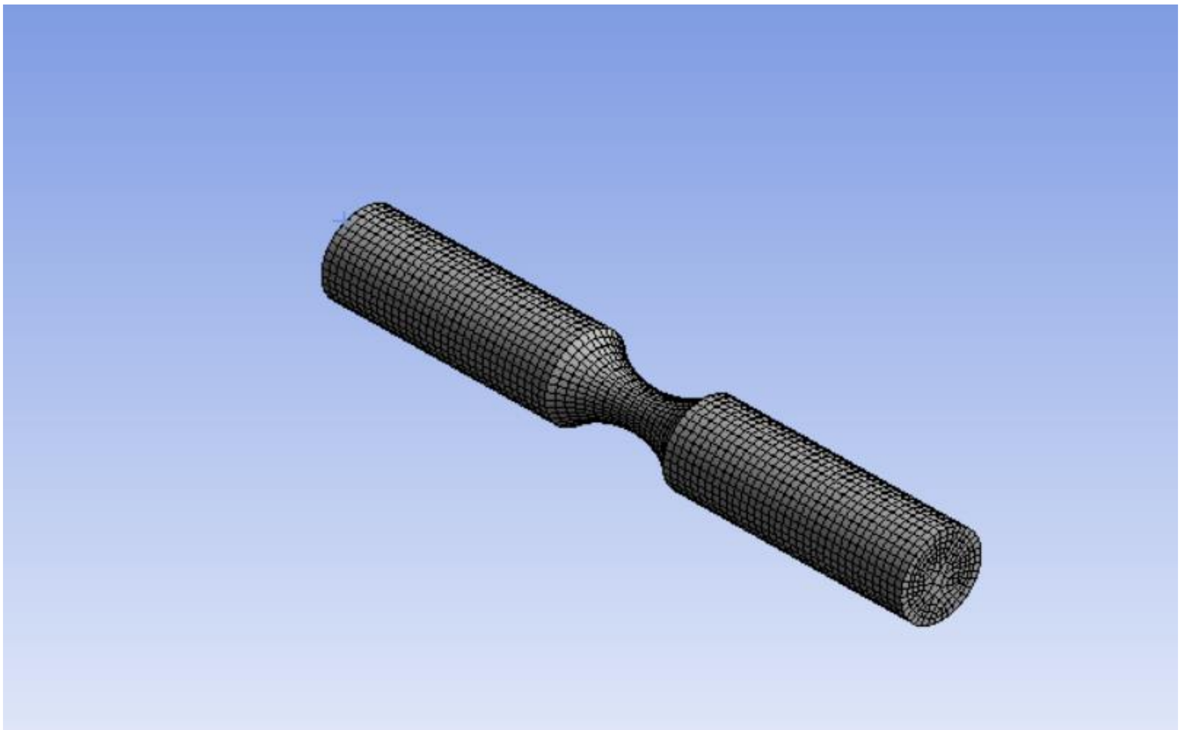

Figure 14 (4.7): Meshing process modeling

4.1.5 Boundary Conditions

The boundary condition of the specimen using the following running: static structural, insert: fixed support at one end of the shaft as shown in the figure (4.8), insert force was applied bending load vertically at Y direction and rotation velocity (6000 r.p.m) as shown below in the figure (4.9). The loads were calculated using equation 3.11 in chapter 3 based on the stresses provided with the standard brass specimen in Appendix C. The value of the bending stress (σb measured in (N/mm^2) for a known value of load (P), measured in Newton (N) is calculated from the relative below:

$$\boldsymbol{\sigma}_{b=}\frac{M_{b\times Y}}{I}$$

$$\boldsymbol{\sigma}_{b=}\frac{F\times L\times Y}{\frac{\pi\times d^4}{64}}=\frac{F\times L\times\frac{d}{2}}{\frac{\pi\times d^4}{64}}$$

Where:-

σb : The applied stress (N/mm^2)

Mb: The bending moment (N.mm)

Y: The distance from the neutral position (mm)

I: The moment of inertia (mm^4)

d: Minimum diameter of fatigue specimen (mm)

For Rotating bending fatigue machine

L=125.7 mm, d=4 mm

therefore σb = 20F (N/mm2)

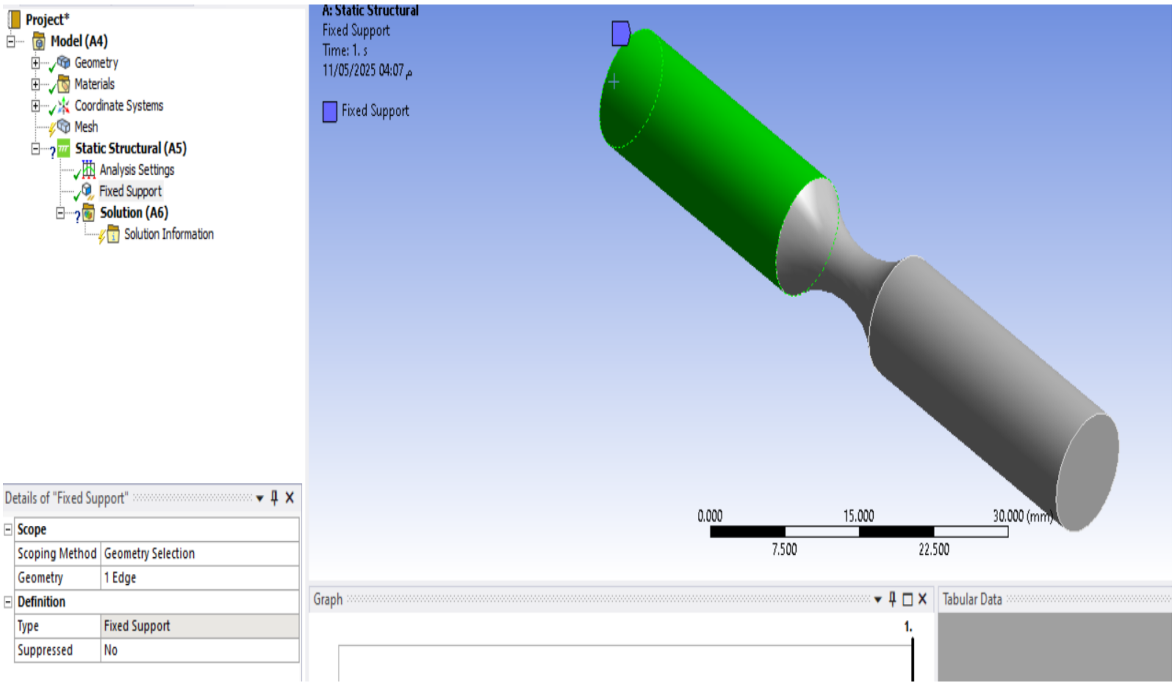

Figure 15 (4.8): fixed support

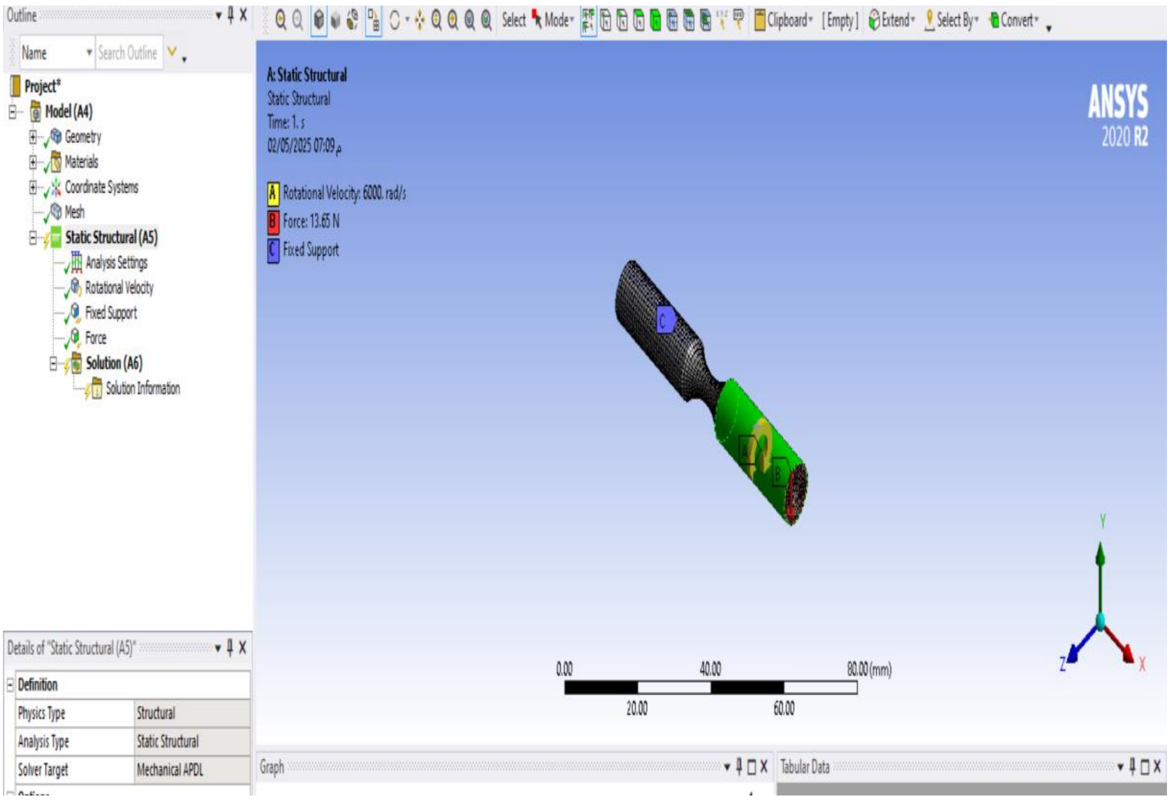

Figure 16 (4.9): boundary condition

4.1.6 Modal Analysis

In this section, solution information using the following running insert fatigue tool to analyzed the following as shown in figure (4.10) the Analysis the Equivalent max stress (von-Misses) was valued (258.15 Map) and fatigue life valued was (1.5268e5) as shown below in the figures (4.11) and (4.12) as respectably .this results was made for point (1) than repeated all steps with changed the applied load for (5) points. For drawing the S-N curve.

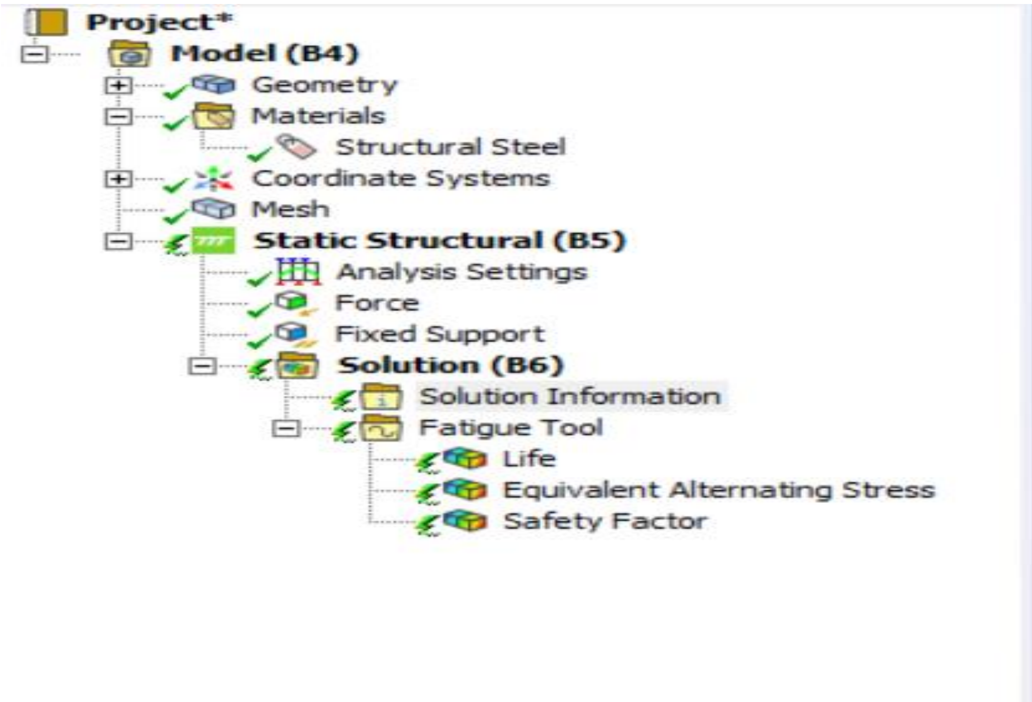

Figure 17 (4.10): fatigue tool

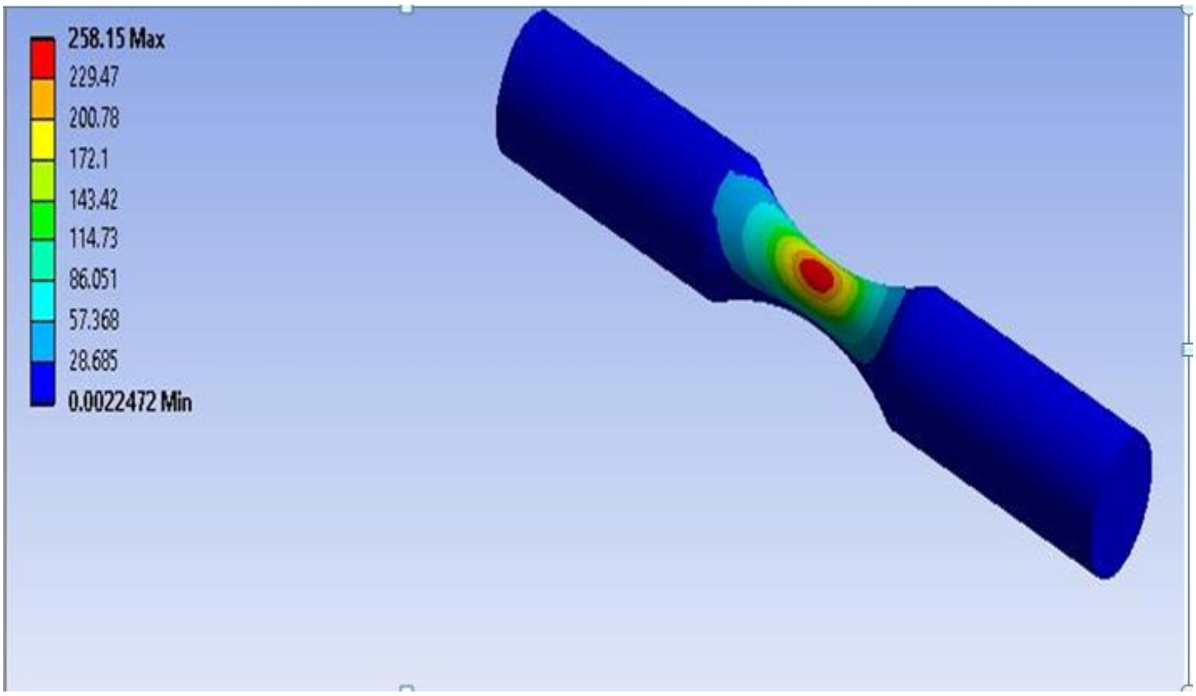

Figure 18(4.11): Max stress (von misses) of the bras shaft

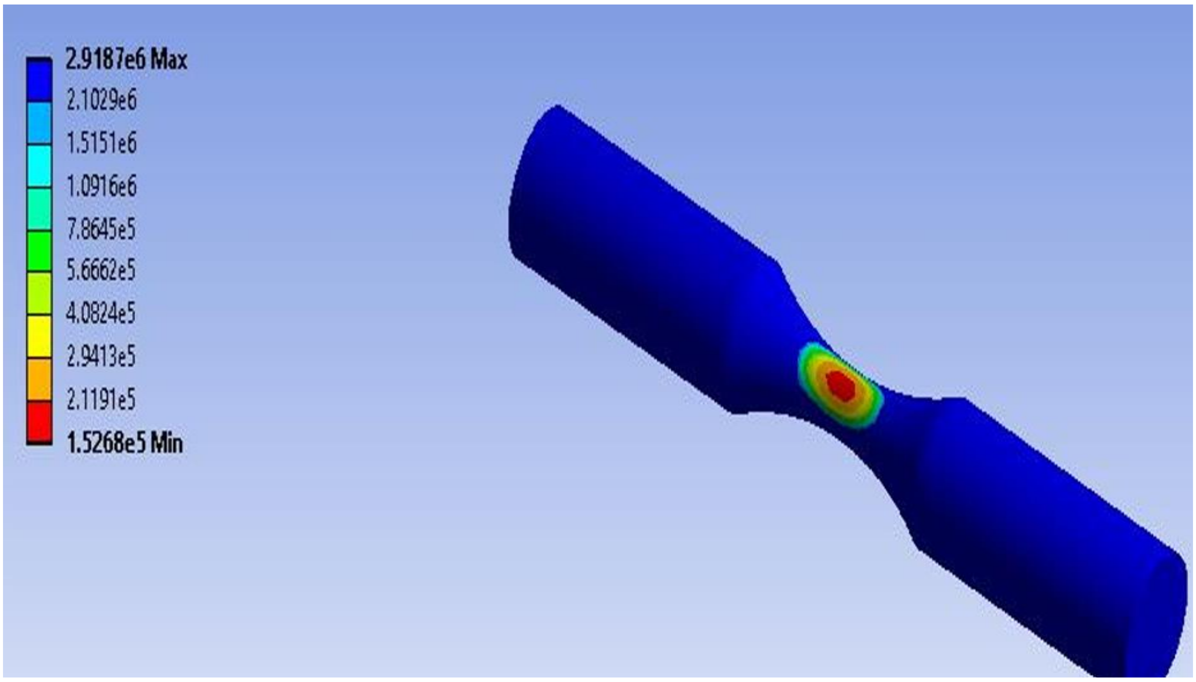

Figure 19 (4.12): fatigue life of bras shaft

4.2 Numerical Analysis of Mild steel

The same steps that were performed on the brass metal specimen in order to calculate the fatigue life will be applied to mild steel specimen with boundary condition of the specimen were considered fixed support at one end, and then a force was applied bending load vertically at Y direction) and rotation velocity (6000 rpm). As shown below in the figure (4.13) . The loads were calculated using equation 3.11 in chapter 3 based on multiplying the ultimate stress of mild steel (640 MPa) by five factors of (0.8, 0.6, 0.4, 0.3and 0.2) to obtain five loads. These calculations of cycle loading were conducted because the stress applied on the mild specimen must be less than its ultimate stress as shown in table (5.3).

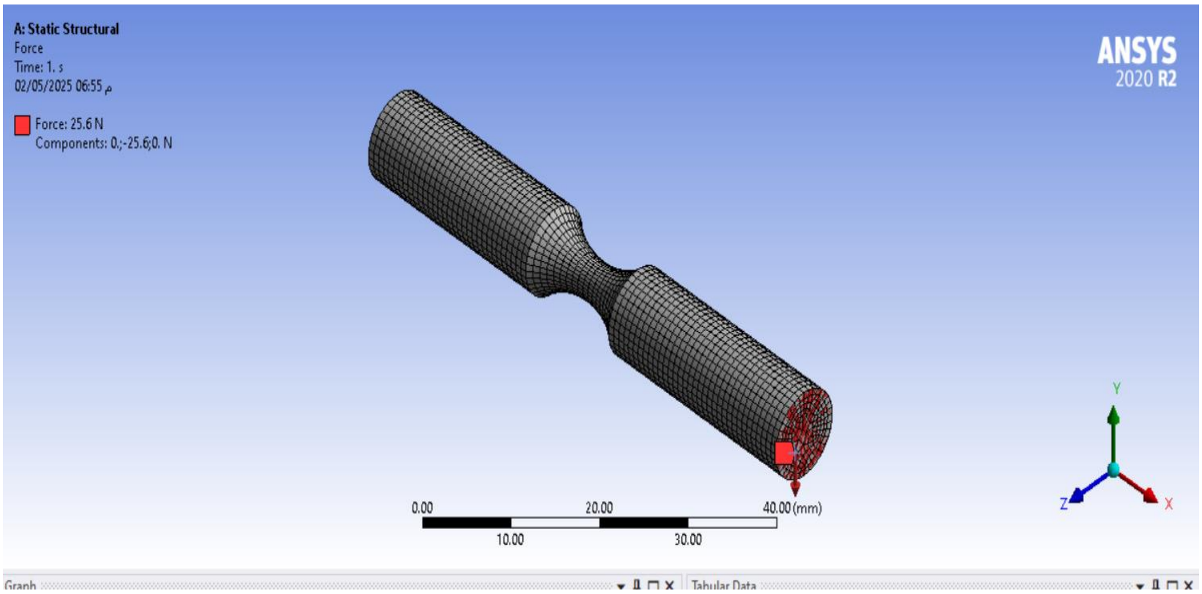

Figure (4.13): Boundary Conditions Modeling of mild steel

4.2.1 Model Analysis Results

In this section, analyzer the Equivalent max stress (von-Misses) was valued (493.39 N/mm^2) and fatigue life valued was (1493.6) as shown below in the figures (4.14) and (4.15) as respectably.

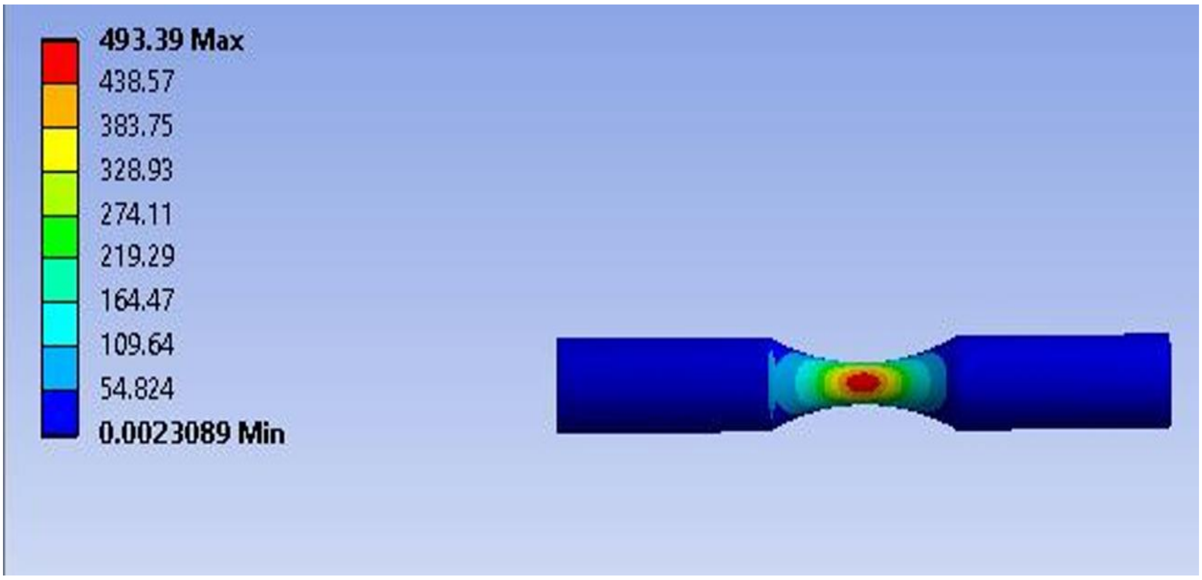

Figure 20 (4.14): Max stress (von misses) of the mild steel shaft misses)

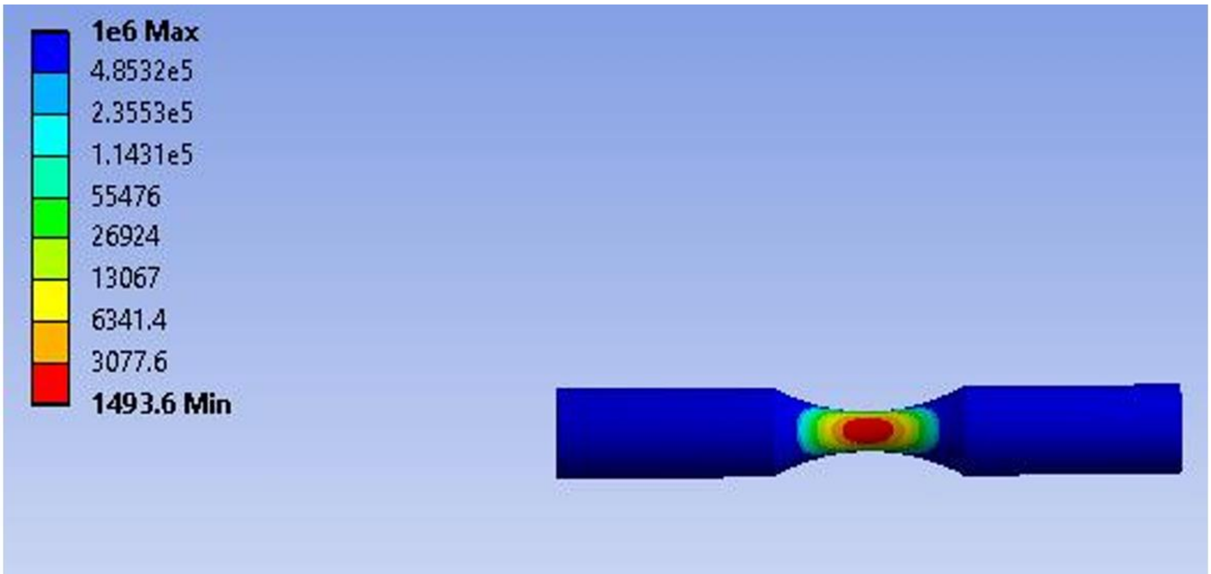

Figure 21 (4.15): fatigue life of mild steel shaft

Same way was conducted for all other points. The maximum stress (Von-Misses) and fatigue life of point (5) are (123.35 N/mm2) and (1.5025e5) respectively as shown figures (4.16) and (4.17)

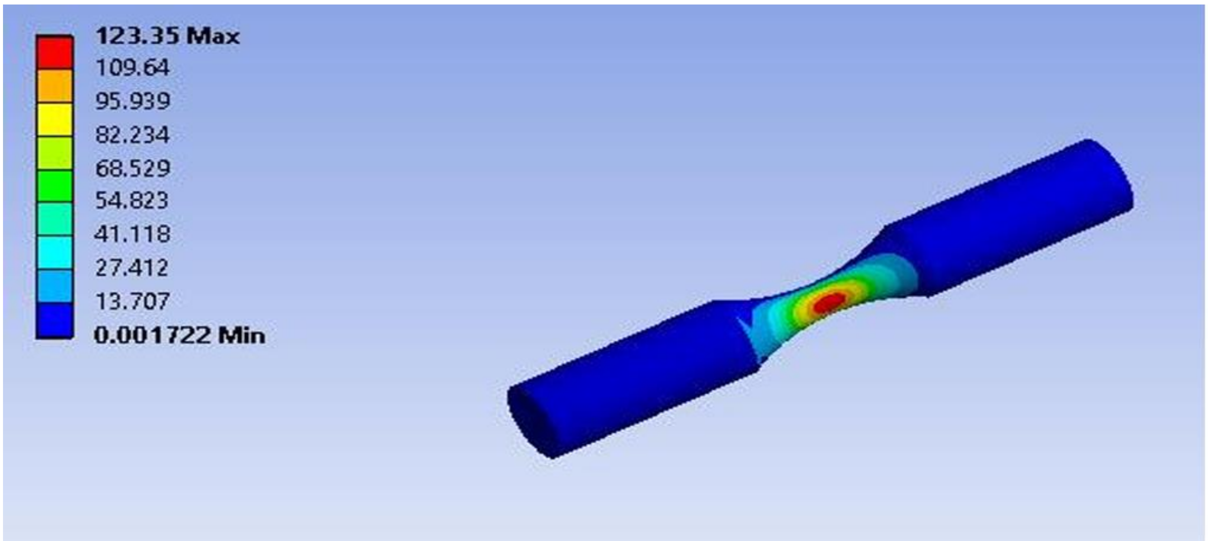

Figure 22 (4.16): Max. Stress (Von Misses) of the mild steel shaft (point 5)

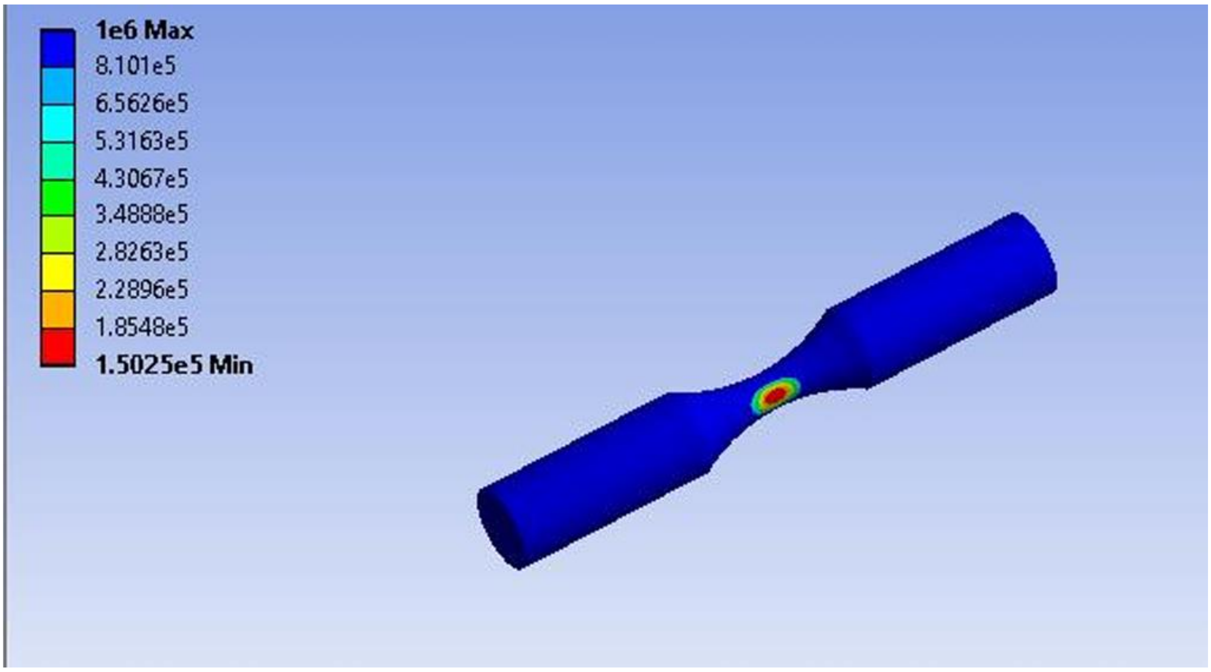

Figure 23 (4.17): Fatigue life of mild steel shaft (point 5)

Chapter Five

RESULTS AND DISCUSSIONS

Chapter Five

Results and Discussions

This chapter describes the numerical solution results of ANSYS Workbench 18.2 for two materials, brass and mild steel which subjected to bending moment, so the current chapter includes the following:

1-Numiercal result of the brass and the mild steel Material (S-N Curve).

2-Basque's Equation Constants Results.

5.1 Numiercal result of the Brass Material (S-N Curve)

The S-N graph is between the number of cycle and the stress. The number of Cycles are taken from ANSYS when applying a continuous cycle loading, until the

Specimen reaches the failure. The calculations of cycle loading shown in table (5.1) were made by verified the load applied on the shaft because ANSYS only uses load not stress (for four trail). Figure (5.1) represents the S-N curve for brass metal specimen.

Table 3 (5.1) the stress and No of cycle value (Numerical) for brass metal

Reversed bending test		
No	σ_bMpa	N_f
1	258	152680
2	207	362430
3	159	981200
4	127	3492245

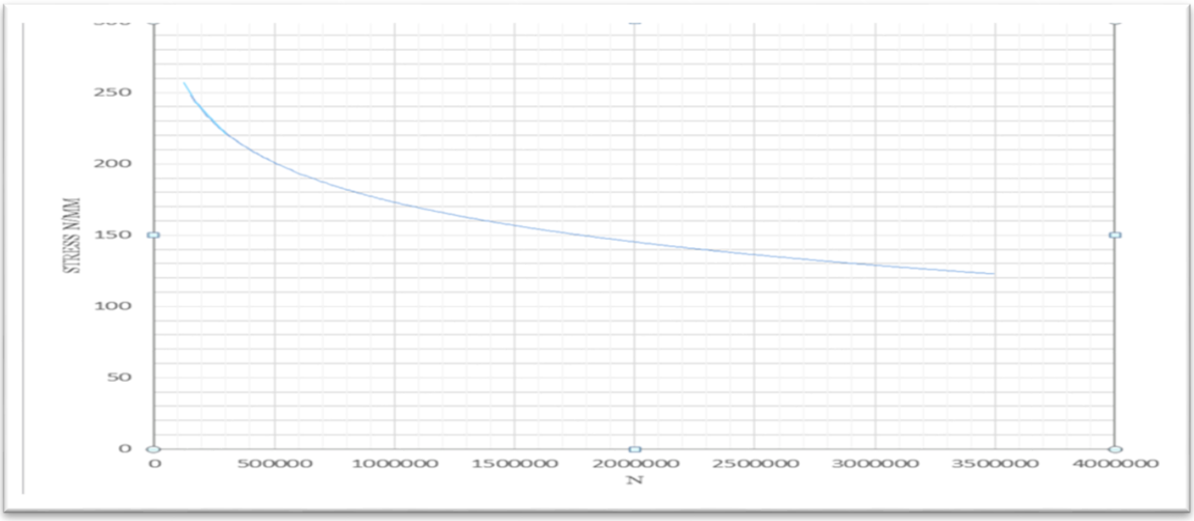

Figure 24 (5.1) S-N Curve for brass metal

This result which calibrated with the stander curve of Hi- TECH machine as shown in APPEDIX C. The table (5.2) and the figure (5.2) shown stander S-N curve for brass metal. The purpose of the calibration of brass metal shaft was applied alternating or fluctuating bending to a cantilevered strip of material to determined fatigue performance which was slandered values calibrated with numerical results of brass shaft.

Table 4 (5.2) the stress and No of cycle (S-N stander) value for brass metal

Reversed bending test			
No	σbMpa	N_f	Nf_{ave}
1,2,3	267	127700, 107800,65200	100233
4,5,6	199	319600,226200,303600	283133
7,8,9	155	871600,663700,558500	697933
10,11,12	138	2918700,1119600,2065800	2034700
13,14,15	124	2660200,1366700,2919000	2315300

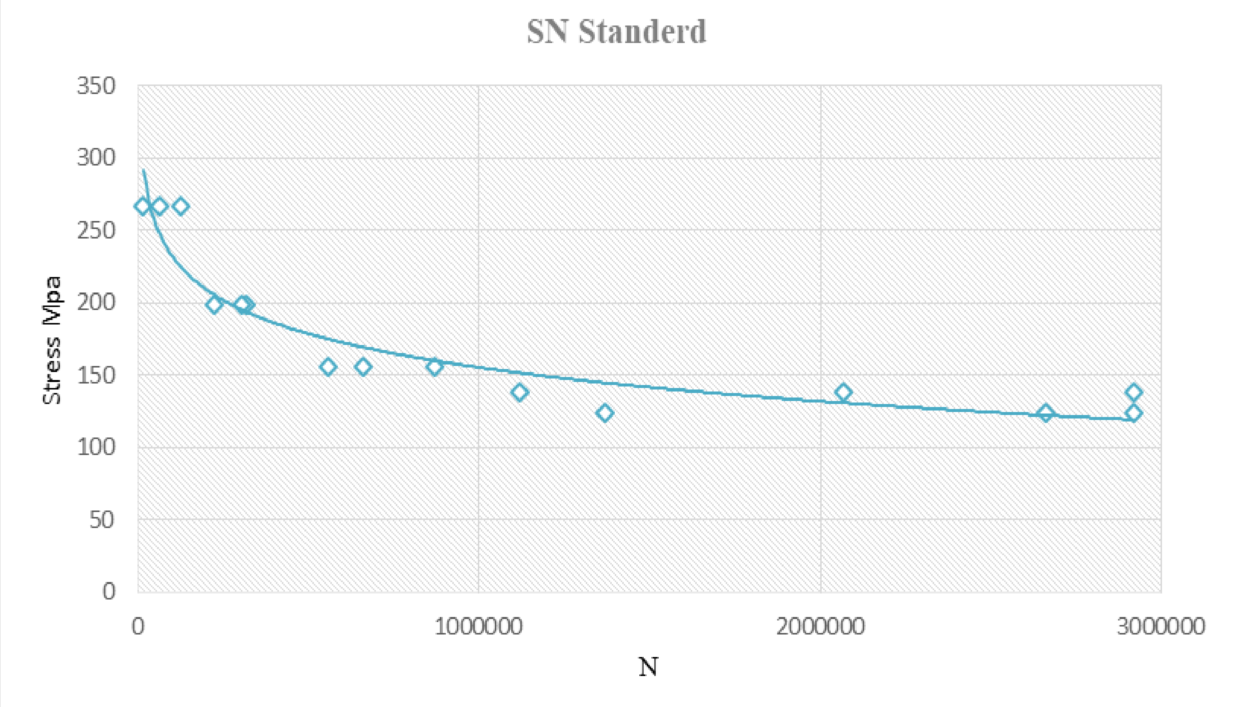

Figure 25 (5.2): S-N curve stander of brass metal

Table (5.3) shown the error percentage between numerical and stander number of cycle for brass metal appear to be of acceptable quality from the point of view of the of the number of cycle and load stress.

Table 5(5.3): error percentage between numerical and stander number of cycle for brass metal.

No	σbMpa	Nf(numerically)	Nf(stander)	Error %
1	267	152680	127700	19%
2	199	362430	319600	13%
3	155	981200	871600	12%
4	138	3492245	2918700	19%
5	124	3398700	2919000	16%

5.2 Numirecal result of the mild steel material under bending moment (S-N curve)

In the theoretical analysis of the reversed bending fatigue using ANSYS program for calculate the fatigue life and equivalent stress was applied under bending stress The S-N graph is between the number of cycle and the stress. The number of Cycles are taken from ANSYS when applying a continuous cycle loading, until the

Specimen reaches the failure. These calculations of cycle loading were conducted because the stress applied on the mild specimen must be less than its ultimate stress as shown in table (5.4). As a result of this, the S-N curve was drawn as shown in figure (5.3).

Table 6 (5.4) the stress and No of cycle value for mild steel metal

No	σ Mpa	Load(N)	σ(from ANSYS)	$N_{f(Cycles)}$
1	512	25.6	493	1493
2	384	19.2	367	3407
3	256	12.8	244	12164
4	192	9.6	188	33735
5	128	6.4	123	150000

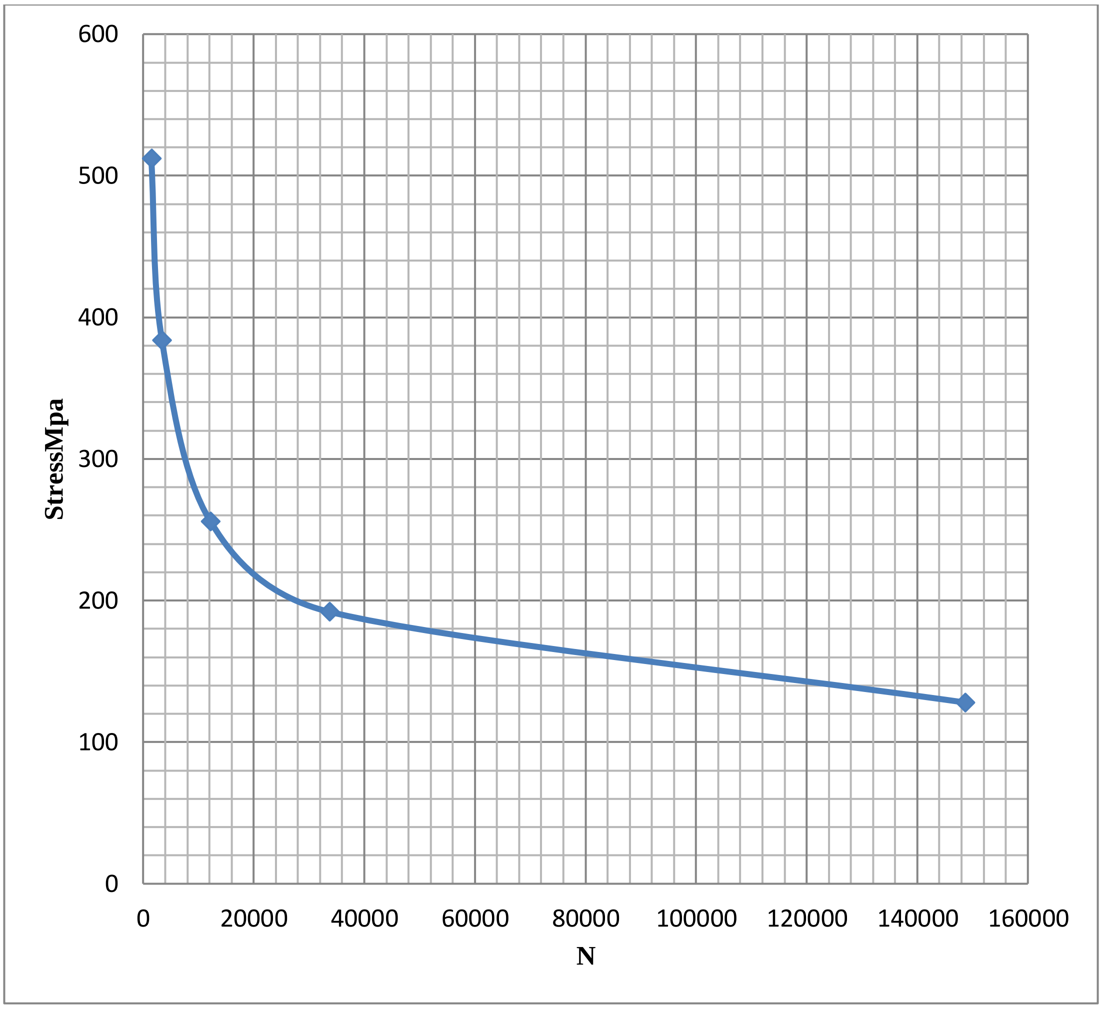

Figure 26 (5. 3) S-N Curve for mild steel metal

5.1 Basque's Equation Constants Results

5.3.1 Basque's Equation Constants Results for mild steel

Applying the Basque's equation constantans A and α can be determined from [37].

$$\sigma = \frac{h\ \sum_{i=1}^{h} \log \sigma_f \log N_f - \left(\sum_{i=1}^{h} \log \sigma_f\right)\left(\sum_{i=1}^{h} \log N_f\right)}{h \sum_{i=1}^{h} \left(\log N_f\right)^2 - \left(\sum_{i=1}^{h} \log N_f\right)^2}$$

$$\log A = \frac{\sum_{i=1}^{h} \log \sigma_f - \sigma \sum_{i=1}^{h} \log \sigma_f}{h}$$

Table 7 (5.5) shows the results when apply the Basque's equation

No	N_f	$logN_f$	σf	$log\sigma_f$	$log\sigma_f logN_f$
1	1493	3.17	493	2.69	8.52
2	3407	3.53	367	2.56	9.03
3	12164	4.08	244	2.38	9.71
4	33735	4.52	188	2.27	10.26
5	150000	5.17	123	2.08	10.75

h=5

$\sum(logN_f)^2$ =419.020

α = - 0. ٢٩٨٣

A=٤٢١٢

$\sigma_{f=4212\,Nf^{-0.30}}$

5.3.2 Basque's Equation Constants Results for brass

Applying the Basque's equation constantans A and α can be determined from [37].

$$\sigma = \frac{h\ \sum_{i=1}^{h} \log\sigma_f \log N_f - \left(\sum_{i=1}^{h} \log\sigma_f\right)\left(\sum_{i=1}^{h} \log N_f\right)}{h\sum_{i=1}^{h}\left(\log N_f\right)^2 - \left(\sum_{i=1}^{h} \log N_f\right)^2}$$

$$\log A = \frac{\sum_{i=1}^{h} \log\sigma_f - \sigma\sum_{i=1}^{h} \log\sigma_f}{h}$$

Applied this equation to the results of brass metal from table (5.1).

h=4

α= - 0. 228

A=3858

$\sigma_{f=3858\,Nf^{-0.228}}$

Chapter Six

Conclusion and Recommendation

Chapter Six

Conclusion; and Recommendation

6.1 Conclusions:

These chapter Considerable conclusions have been pointed out hereunder by virtue of outcomes

1- An estimation of fatigue life cyclic of mild steel and brass metal under bending moment moment.
2- Numerical results had been obtained by using Ansys program to estimation the fatigue life by knowledge the relation between number of cycle with stresses (S-N) curve.
3- The calculation Basque's Equation Constants Results.

6.2 Recommendations for Future the Work

In order increase the field that covers the research in the future, have been many topic shown below:

1- Study the effect the cumulative fatigue damage under multi fatigue bending
2- Study fatigue crack growth and estimation fatigue life under multi perpendicular bending moment.
3- This work plane can be repeated for another material such as stainless steel , or another series of aluminum allo

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APPENDIX A

	Al-Karama Engineering test Lab.	**Document No: F37-16**
		Rev. No: 00
	Chemical Composition Analysis Report	**Issue Date: 01/01/2017**

No. :
Date: / / 2019

02/09/2020	تاريخ الشهادة		رقم شهادة الفحص
مهندس حيدر عباس	اسم الجهة المستفيدة	Shaft	مادة الفحص
الاسوا ق المحلية	اسم الجهة المنفذة	1	عدد النماذج
SPECTROPORT	نوع جهاز الفحص	--------	الكمية
00 / 08 /2020	تاريخ الكتاب الوارد	Chemical Composition Analysis	نوع الفحص
00 / 08 /2020	تاريخ استلام النموذج		اسم المشروع

Chemical Composition: Samples No. (1, 2& 3)

Samples No. (1)	C%	Si%	Mn%	P%	S%	Cr%	Mo%	Ni%	Al%	Co%
	0.093	0107	0.43	0.0058	0.0291	0.105	0.0141	0.082	<0.0040	0.0064
	Cu%	Nb%	Ti%	V%	W%	Pb%	Sn%	Zr%	B%	Fe%
	0.260	<0.0070	<0.0020	<.0040	<0.040	<0.0110	0.0165	<0.0070	<0.0005	98.8

Conformable to Material of St37-2(DIN1.0037)	نتيجة الفحص

Note:-This Type for Steel
1-Low Carbon Steel

- تم جلب النموذج من قبل المهندس
- المركز مسؤول عن النموذج المفحوص وليس عن الكمية الكلية في الموقع

نسخة منه الى:
أدارة الشركة

Prepared by: Eng. Ali M. Jassim	Approved by: Eng. Mohamed A. Attia
Sign:	Sign:
Date: 01 / 09 / 2020	Date: 01 / 09 / 2020

APPENDIX B

Al-Karama Engineering test Lab.		Document No: F37-16
		Rev. No: 00
Chemical Composition Analysis Report		Issue Date: 01/01/2017

No. :

Date: / / 2019

13/09/2020	تاريخ الشهادة		رقم شهادة الفحص
مهندس حيدر عباس	اسم الجهة المستفيدة	Brass Shaft	مادة الفحص
الاسوا ق المحلية	اسم الجهة المنفذة	1	عدد النماذج
SPECTROPORT	نوع جهاز الفحص	———	الكمية
00/08/2020	تاريخ الكتاب الوارد	Chemical CompositionAnalysis	نوع الفحص
00 / 08 /2020	تاريخ استلام النموذج		اسم المشروع

Chemical Composition:Samples No. (1)

Samples No. (1)	Zn%	Pb%	Sn%	P%	Mn%	Fe%	Ni%	Si%	Mg%	Cr%
	39.03	2.74	0.22	0.035	<0.004	0.17	0.098	0.004	0.0006	0.003
	As%	Sb%	Cd%	Bi%	Ag%	Co%	Al%	S%	Be%	Cu%
	0.040	0.10	0.005	0.009	0.005	0.009	0.003	0.013	<0.0001	57.5

Conformable to Material of DIN(CS629N)-EN(S-CuZn-6)	نتيجة الفحص

- تم جلب النموذج من قبل المهندس
- المختبر مسؤول عن النموذج المفحوص وليس عن الكمية الكلية في الموقع

نسخة منه الى:
أدارة الشركة

Prepared by: Eng. Ali M. Jassim	Approved by: Eng. Mohamed A. Attia
Sign:	Sign:
Date: 13 / 09 / 2020	Date: 13 / 09 / 2020

APPENDIX C

Table 8 Bending Fatigue Tests on Brass

Length (l) (mm)	Stress (S) (N/mm^2)	Reversals (N) ($\times 10^3$)	Log *S*	Log *N*
35.0	267	127.7	2.43	5.11
		107.8		5.03
		65.2		4.81
40.0	199	319.6	2.30	5.50
		226.2		5.35
		303.6		5.48
45.0	155	871.6	2.19	5.94
		663.7		5.82
		558.5		5.75
47.5	138	2918.7	2.14	6.47
		1119.6		6.05
		2065.8		6.32
50.0	124	2660.2	2.09	6.42
		1366.7		6.14
		>2919.0		>6.47

جمهورية العراق
وزارة التعليم العالي والبحث العلمي
جامعة ميسان/كلية الهندسة
قسم هندسة الميكانيك

Numerical Simulation of Fatigue Life Estimation of Mild Steel Shaft under uniaxial loading

بحث تقدم به كل من الطالب
زينب عبد الأمير محمد
رؤى جعفر بنيان
مريم قيس دگس

كجزء من متطلبات الحصول على درجة بكالوريوس العلوم في الهندسة الميكانيكية
جامعة ميسان
المشرف على البحث: م.م إيمان محمد

الخلاصة

قامت العديد من البحوث في الهندسة الميكانيكية بدراسة الحمل المتغير وسببه الذي يحدث عندمل يصل اعلى من الحد الاعلى ، حيث يبدأ التصدع من السطح وينتشرويؤدي الى حدوث الفشل ويسمى فشل التعب. فشل التعب لا يعطي أي تحذير في التقدم ويحدث فجأة مما يؤدي إلى حوادث.

يقدم هذا البحث بيانات تصميم وتحكم كاملة، بالإضافة إلى معلومات مفيدة أخرى لتقدير تحليل الفشل وعمر التعب لعمود فولاذي خفيف تحت عزم انحناء دوري عمودي.

يتضمن الجزء النظري أيضًا مقارنة بين النتائج العددية التي تم الحصول عليها بواسطة برنامج ANSYS 18.2، والتي تمت معايرتها باستخدام جهاز قياسي من نوع (-HI-TECH HSM19Mk3) لعمود نحاسي، حيث تم إجراء الاختبارات على هذه الآلة التي تتحمل نطاقًا كبيرًا من الإجهادات وعمرًا أطول. يجب أن تتطابق النتائج التي تم الحصول عليها مع منحنى S-N القياسي.

ايضا تم تقدير ثوابت معادلة الميل لنتائج عمود الدوران النحاسي والفولاذي الخفيف.