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Influence of Gear Materials on Surface Fatigue Failure of Spur Gear

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(Mechanical Design Stream)

By
Sewnet Alemu
Advisor
Dr. Daniel Tilahun

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DEPARTMENT OF MECHANICAL ENGINEERING

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Approved by Board of Examining:

Daniel Tilahun (Dr.)	________________________	_____________________
Chairman	Signature	Date
Daniel Tilahun (Dr.)	________________________	_____________________
Advisor	Signature	Date
Zewudu Abdi (Dr.-Ing)	________________________	_____________________
Internal Evaluator	Signature	Date
Tamirat Tesfaye (Dr.-Ing)	________________________	_____________________
External Evaluator	Signature	Date

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ABSTRACT

The objective of this study is to investigate the effect of material on surface durability of spur gear. Current Analytical methods of calculating gear contact stresses use the contact stresses of spur gear. It’s necessary to develop and to determine appropriate models of contact elements, and to calculate contact stresses using ANSYS and compare the results with Hertzian theory. In addition parametric study to investigate the effect of material types in contact stress in gears was performed and Results from ANSYS Workbench, for a 3D model of involute spur gear, shows that contact stress results correlated with theoretical formulation, with tolerable accuracy. It is also shown that the development of finite element analysis model of the Spur gear assembly to simulate the contact stress between two gears reasonable ,and obtained result is compared with the Hertzian theoretical equation. Results of finite element approach for contact stress estimation in ANSYS and Hertz theory, for the material Grey cast iron GG-30,, Grey cast iron-20,and Chromium-molybdenum alloy steel (SCM 420) is compared. The result of the analysis summarized as follow. The maximum contact stress value for these material was, 1088Mpa, 977MPa and 1492Mpa respectively, which obtained by Hertz theory, the well known theoretical calculations method for contact problem. The stress analysis has been done by ANSYS software, where the results have been presented by contours and numerical values. The maximum contact pressure result from ANSYS for 3D model is 1032Mpa, 825MPa, 1414Mpa respectively. Based on the result from the contact stress analysis the hardness of the gear tooth can be improved to resist pitting failure.

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LIST OF SYMBOLS

$\boldsymbol{B}$	Half of contact width
$\boldsymbol{Dg}$	Gear pitch diameters
$\boldsymbol{Dp}$	Pinion pitch diameters,
$\boldsymbol{Ee}$	Equivalent modulus of material,
$\boldsymbol{E_1}$	Youngus Modulus for material 1
$\boldsymbol{E_2}$	Youngus Modulus for material 2
$\boldsymbol{F}$	Force acting over contact length L
$\boldsymbol{L}$	Contact length
$\boldsymbol{M}$	Module
$\boldsymbol{P_{max}}$	Maximum contact pressure
$\boldsymbol{r_{bg}}$	Base radius of gear
$\boldsymbol{r_{bp}}$	Base radius of pinion
$\boldsymbol{R_g}$	Radius of gear
$\boldsymbol{R_P}$	Radius of pinion
$\boldsymbol{R_e}$	Equivalent radius of curvature
$\boldsymbol{R_1}$	Radius of cylinder 1

R_2	Radius of cylinder 2
T_P	Torque on pinion
F_N	The normal force applied on the pinion
F_T	Tangential component of the force
φ	Pressure angle.
v	Poison ratio for material
σ_c	Contact stress

CHAPTER ONE

1. INTRODUCTION

1.1 BACKGROUND

Mechanical components such as gears, shafts, cams, and springs, are subjected to static and to rapidly fluctuating cyclic or periodic loads causing cyclic stresses. Under these conditions, the parts fail due to contact fatigue, at a stress lower than what failure would have occurred under static loading. Contact fatigue damage is very complicated, and during simulation the type of material, the geometry involved, the mode of lubrication, and the size should be considered.

The most common mode of gear failure encountered in practice is that of surface contact fatigue. This mode of failure leads to crack initiation at or near the contact surface, and may subsequently lead to damage varying in extent from microscopic pitting to severe spalling. The metal removed from the surface in such cases enters the machine system, and can, in turn, cause abrasive wear and failure of other components. Furthermore, the pits formed on the damaged surface lead to the formation of stress concentrations, and serve as initiation sites for other modes of gear failure. [20]

Whenever two curved (usually convex) surfaces are in contact under load, the contact occurs along a line or point, or, depending on the elastic constants of the materials concerned, along a very small circular or elliptical area. As a result of such small contact areas, the shear (Hertzian) stresses which develop at and near the surface are consequently very high. [21]

When the contacting stresses are repetitive, as is the case on the active flanks of gear teeth, the cyclic compressive stresses induced cause differing elastic and plastic behavior in the near-surface material. Depending on the microstructure and grain orientation of the material in this region, internal stress concentrations are formed which can ultimately lead to crack initiation. In practice, crack initiation usually occurs at inclusions in the stressed near-surface material, the most deleterious being those inclusions which are hard, brittle and angular in shape. [22]

Within geared transmissions, gear failures occur in many ways and often without advanced notice. While engineers have developed, over the years, a greater understanding of these failures, there is still a need for a thorough understanding of how gears fail and how they can be made to last longer. There are many types of gear failures, each influenced by a variety of gear, surface, lubricant and contact parameters. This thesis focuses surface failure, namely gear pitting and the influence of gear material pitting. [15] Pitting develops over time from recurring contact stresses between the teeth of two gears during rotation. Pitting can be described as visual surface fractures on the gear teeth usually preceded with hairline cracks that develop on or below the tooth surface. Once gear pitting has initiated, gear noise may become more prevalent and the gear surface continues to degrade until complete gear failure has occurred. [15]

Gears have been used by human for many years for various applications still a large number of unsolved problems exist. One of these problems is the assessment of the stress and strain status of the gear under rolling-sliding contact conditions for the investigation of the mechanically originated tooth surface damage of gears, prediction of wear and fatigue as well as change of material properties and different working conditions. [23]

In recent years, gear manufacturers have increased their technological knowledge of the production of quality gears. This knowledge has led to many improvements, including lower noise, lighter weight and lower cost, as well as increased load-carrying capacity to handle higher speeds and torque with a minimum amount of generated heat.

Gears are a critical component in the rotating machinery industry. Various research methods, such as theoretical, numerical, and experimental, have been done throughout the years regarding gears. One of the reasons why theoretical and numerical methods are preferred is because experimental testing can be particularly expensive.

After the investigation of shot peening to increase the tooth bending strength in gears, surface durability, in the form of macro and micropitting, is now considered the dominant restriction on gear life and performance as it is shown in figure 1. [24], In addition, a broken tooth tends to be catastrophic to a gear unit, so the designer usually makes the teeth large enough so that they are definitely less appropriate to fail in breakage mode than in a pitting mode. This makes the design life of a gear unit primarily dependent on its surface fatigue capacity (pitting resistance) rather than on its cantilever beam capacity (capacity to resist tooth breakage).

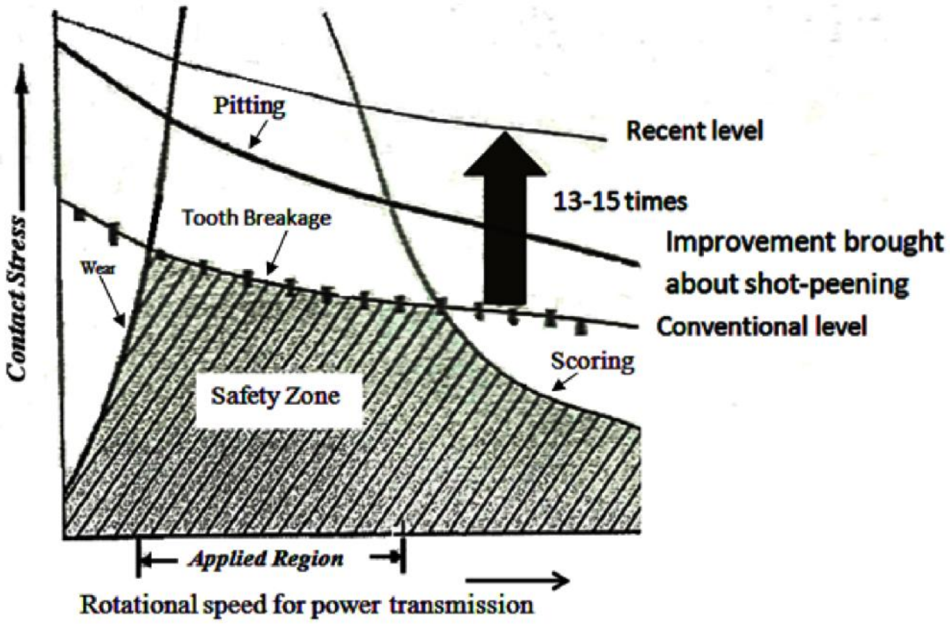

Figure 1.1: Failure of gears with respect to contact stress and rotational speed. [24]

As shown in the figure the vertical axis represents the contact stress and the horizontal axis represents the rotational speed of the gear. The applied region in the rotational speed range refers the rotational speed of the gears commonly used in power transmission gears.

Improving the fatigue strength of gears is therefore of great importance in attaining increased load carrying capacities and in improving component reliability. Since most bending fatigue failures are initiated at or near to the surface, one of the most promising methods of improving fatigue strength is through the use of different treatments to modify the surface properties of the material. This study focuses primarily on the influence of material type on pitting fatigue performance of transmission gearing.

Gear pitting studies are very time-consuming and costly. For competitive reasons, results of highly specific studies of certain materials or surface treatments often do not reach the public domain. Therefore, though many pitting studies can be expected to take place globally, the amount of published (public-domain) data remains to be relatively limited.

1.2 OBJECTIVES OF THE THESIS

- ***General Objective:*** the general objective of this thesis is to study surface fatigue resistance of different materials using FEM & ANSYS and comparing results with theoretical formulation.
- ***Specific Objective***

✓ Geometrical Modeling of spur gear using ANSYS Workbench
✓ Analytical Contact Stress Analysis of Spur Gear using Hertzian theory
✓ Study the effect of material on surface fatigue

1.3 METHODOLOGY USED

✓ Modeling of spur gear geometry using ANSYS Workbench
✓ FEM and ANSYS analysis for determining The VonMises stress

1.4. ORGANIZATION OF THE THESIS

This thesis is organized in to six chapters. In the first chapter, background and justification of this thesis work and the objectives to be achieved and the methodology applied are discussed. In chapter two, a review of literature relevant to this thesis work, which has been investigated by different researchers, is given. Chapter three is about analytical method in contact stress analysis for spur gears. In chapter four, finite element method (FEM) is used, to develop involutes spur gear using ANSYS Workbench and 3D finite element analysis is done in ANSYS workbench. It also illustrates variation in stress results for different material types. In chapter five results of the analysis are summarized and discussions are made based on the outputs of the FEM. In addition, comparison of analytical method and experimental results from references. Finally, chapter six gives conclusion achieved from this thesis work and propose future work in this field of study.

CHAPTER TWO

2. LITERATURE REVIEW

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. [19]

Each tooth of a gear in a power flow path of the transmission is subject to cyclic loading as it continually enters and leaves its gear meshes. This intermittent loading on the tooth causes cyclic tooth bending stresses on the surface. Under such loading conditions, fatigue induced tooth breakage failures can take place depending on the state of stresses along the root region as well as the shape of the root fillet.

The same type of loading also causes cyclic contact stresses on the active tooth surfaces. Any arbitrary point on the tooth surface above the start of active profile is typically subjected to a combined rolling and sliding motion unless the point is exactly on the pitch-line where there is no sliding (pure rolling only). The instantaneous normal stresses combined with the corresponding surface shear stresses induced by the instantaneous traction result in multi-axial stress states on and below the tooth surface, which cause pitting failures, as described through modeling work by [1, 2] and [3]. Gear pitting remains to be one of the most common failure modes experienced by the gears of transmission systems. [17]

Pitting occurs when the cyclic stresses in a gear tooth causes small cracks at or below the surface of the tooth face. These cracks will propagate into pits as the gear material fractures and is removed from the tooth. The size of the pit then grows as the removed material presents further stress concentrations on the tooth face. While the parameters dictating tooth bending fatigue are limited to geometry, material and loading levels, a large number of parameters interact to define contact fatigue conditions of gears. Contact fatigue of gears is greatly influenced by tooth geometry as well as surface finish and lubrication parameters. These variables combined with gear material, operating speed, and temperature all define the conditions for cracks to nucleate and grow into sizable pits. [17]

Many gear pitting studies have been performed over the last several decades. The majority of these studies have focused on changing a single parameter or a sub-set of parameters to determine their influence on pitting fatigue life. The body of published work indicates that surface fatigue life is dependent on gear material, gear geometries, heat treatment, case hardening, surface finish, lubrication, and temperature, operating torque, and operating speed. Therefore the following literature reviews are organized according to ,the effect of surface condition on surface fatigue, the effect of material on surface fatigue and effect of operating condition on surface fatigue respectively

In one of the earlier studies, [4] performed experiments on the influence of the surface finishes and gear materials on gear pitting performance. The study used helical gears which were heat treated using quenching and annealing, flame hardening, or induction hardening processes, and spur gears which were heat treated through carburization. Helical gears were made of either CrMo steel or standard carbon steel while the spur gears were made of either NiCrMo or NiCr steel. Fatigue testing was performed on a power circulation type test machine. Other variables such as lubricant, temperature, and speed were held constant. Kaneko [4] did an extensive study on gear pitting with various carbon and alloy steels. In the study, quenched and annealed gears made of CrMO steel alloy and a carbon steel were subjected to various loads in a power circulation loop configuration similar to the FZG machines.. Flame hardened and induction hardened test specimens, which were ground afterward to remove heat treatment deformations, were also made of the same two materials. The hardened specimens were able to endure higher loads and the CrMo steel alloy had greater pitting fatigue life than samples made of carbon steel. Also in the study, NiCrMo and NiCr carburized steel gears were subjected to pitting tests and endured the highest loads. The NiCrMo carburized steel specimens had greater contact fatigue lives than those made of NiCr carburized steel.

In another experimental study, [5] investigated the micro-pitting behavior of case hardened and heavily loaded gears. They tested several different gears, which were either ground or polished, with five different lubricants. Tests were performed on a power circulating-type gear tester under varying conditions. In the ground gears tested, a common theme was the appearance of micro-pitted grey-stained zones leading to larger pits. Among other factors, it was suggested that a large factor in the occurrence of grey-staining was based on the surface roughness of the gear tooth surfaces.

Two different grinding technologies for AISI 9310 steel gears were tested in another fatigue study by [7]. The surface fatigue life was studied between carburized and hardened AISI 9310 gears finish ground with either a CBN (cubic boron nitride) or a vitreous grinding wheel. The newer CBN technology was thought to improve final gear geometries and provide fewer burn marks due to the high thermal conductivity of the CBN material. Aside from being a much faster grinding process, the CBN wheel finished the gear teeth at an average roughness (Ra) value of 0.30 µm versus 0.34 µm for the vitreous wheel. The test was setup on the same machine and conditions as the previous study [6]. The gears were run-in for one hour (at a lower load) before being loaded to provide a maximum Hertz stress of 248 ksi (1709 MPa).

The CBN ground AISI 9310 gears showed better pitting fatigue life over the standard vitreous ground AISI 9310 gears offering an approximate 50% improvement at the 10- percent fatigue life level.

Using the same experimental set-up, Townsend [8] later investigated the effect of high-intensity shot-peening on AISI 9310 ground gears as compared to a medium intensity shot-peening study performed in an earlier study [9]. The testing conditions and procedure were identical to those described in a previous study [7]. Among other factors analyzed in the study (i.e. variance in sub-surface residual stresses, etc.), the pitting surface fatigue life of the high-intensity shot-peened AISI 9310 gears was found to be approximately two times greater than the medium-intensity shot-peened AISI 9310 gears. This is in comparison to the medium-intensity shot-peened AISI 9310 gears, which exhibited a pitting life 1.6 times greater than the standard ground AISI 9310 gears in the previous study [9].

In the study by [10], he performed gear tests to quantify the fatigue life improvement of gears with improved surface finishes. In the study, a batch of AISI 9310 ground gears were super-finished to provide a near mirror finish. For fatigue life comparison, an older study of AISI 9310 ground gears was used as a baseline. The super-finishing process was shown to have removed 2 μm to 3 μm of material from each tooth face while improving the average surface roughness Ra values by a factor of 5 (0.38 μm baseline to 0.071 μm super-finished). The tests were performed on the same gear test machines used by [6-9]. The tests were run at 10,000 rpm under jet lubrication of which the inlet and outlet temperature were held constant near 47 ^{o}C. The statistical analysis of the data showed that the super-finished 9310 gears had a 10-percent failure life that was greater than the 10-percent failure life of the ground 9310 gears at a 91% confidence level. The study coupled with previous studies cited within the research suggested that super-finishing does significantly improve the surface fatigue lives of aerospace quality AISI 9310 gears.

Research preceding this study at the Ohio State University was carried out by [14] and [15]. [14] Investigated the influence of varying engineered surface finishes on gear pitting lives of gears. Using specially designed spur gears provided by the sponsor with standard FZG test machines, stress-life curves for various surface treatments were successfully generated. All of his specimens were made out of AISI 8620 gear steel. The temperature and operating speed were held constant at 90^{o}C and 1440 rpm, respectively. [14] Performed tests with gear specimens having four different surface finishing processes: baseline shaved surfaces, chemically polished surfaces, shot-peened and plastic honed surfaces, and chemically polished and CrN coated surfaces. Multiple tests were run at specified load levels to obtain a statistically relevant analysis of pitting life expectancy. The study found that chemically polishing the gears yielded an approximate three-fold increase in life expectancy over the baseline shaved gears. Further finishing the chemically polished gears with a CrN coating provided a slight increase in pitting life. Shot-peening and plastic honing gears did not provide any noticeable increase in fatigue life over the baseline gears.

Research by [14] indicated that further testing must be performed on the shot-peened and plastic honed gears and the chemically polished and CrN coated gears to obtain statistically relevant fatigue lives and confidence intervals. Klein [15] continued the research from Bluestein using the same test equipment and procedures. The study included the same gear finishes as Bluestein [14] with the exception of the chemically polished and CrN coated gears which were not tested further. In addition to these gears, the study also included ground gears from the baseline 8620 steel, and ground gears made of different gear steel (4620M). Findings from this study further confirmed Bluestein’s initial findings with respect to greater fatigue life obtained with chemically polished (super-finished) gears. Additionally, testing of shot-peened and plastic honed gears showed greater life expectancy over the baseline shaved gears. Initial testing of the AISI 8620 and 4620M ground gears, though not statistically relevant, also suggested improved pitting fatigue life over the baseline gears.

A contact fatigue study was performed by [13] to investigate the performance of five common gear materials, namely 20MC5 (Chromium-Manganese alloy), XC18 (carbon steel, not alloyed), 16NC6 (Chromium-Nickel alloy), 17CrNiMo6 (Chromium-Nickel-Molybdenum alloy), and 18NCD6 (Chromium-Nickel-Molybdenum alloy). The case carburized gears were finished to an ISO 6 quality standard. The surface roughness Ra values of the gears were within the range of 0.5 to 0.7 μm. Tests were performed on four-square test machines with EP additive oil held at a constant temperature of 70 ^{o}C. Test results were analyzed at two different load levels specified as low load (1,700 MPa) and high load (2,300 MPa). 18NCD6 material was shown to exhibit the best pitting resistance at low load while 20MC5 performed best at high load.

The carbon steel, XC18 performed the worst at all load levels. From the study, 18NCD6 was recommended for long cycle, low load applications while 20MC5 was recommended for high load, short duty applications.

A large number of pitting studies have been performed by Townsend using low contact ratio, unity-ratio spur gear specimens operated in a four-square arrangement [9]. As the main concern of Townsend was aerospace applications, the majority of his pitting studies used aerospace grade AISI 9310 gears. In an earlier study by [6], the potential of using hot forged powder metal (P/M) manufacturing for gears was quantified.

This study considered four different groups of test gears to compare to previously tested machined and ground AISI 9310 gears: hot forged P/M AISI 4620 gears which were carburized and hardened, hot forged P/M AISI 4620 gears which were carburized, hardened, and ground, hot forged P/M AISI 4640 gears which were carburized and hardened, and standard machined AISI 4340 gears which were hardened and ground.

Tests were performed at 10,000 rpm and synthetic paraffinic oil (with a 5% extreme pressure additive) at 77 ^{o}C was used in jet lubrication. Tests were performed at a maximum Hertz stress of 248 ksi (1709 MPa). Due to tooth fracture in the AISI 4640 hot forged gears, the testing torque was lowered until the maximum Hertz stress was 222 ksi (1530 MPa). The standard machined AISI 4340 gears were through hardened and ground, but not able to withstand the higher torque load and were therefore carburized and hardened. This affected the surface geometries of the gears as they did not receive an additional grinding process and this is believed to be the cause of the low life expectancy, and statistical scatter, discovered in the study. Overall, the results of the study showed that the hot forged powder metal gears did not offer the same surface fatigue life as the previously studied AISI 9310 gears. The hot forged P/M AISI 4620 gears that were carburized hardened, and ground showed the best pitting life of the test samples at nearly 70% of the 10-percent pitting life of the AISI 9310 gears.

A later study by [11] provided a dynamic analysis of the test machines used by [6-9] and [10] to point to certain vibration and force transmissibility issues with these NASA machines. The analysis suggested that the forces induced by the errors (eccentricities) of the reaction gears influence the tooth loads of the test gears significantly since the test and reaction gear boxes of the NASA rigs were connected via short, rigid shafts. Later examination of failed test specimens from these earlier studies by [12] pointed to excessive and uneven surface wear patterns associated with the eccentricities of the reaction gears.

CHAPTER THREE

3. ANALYTICAL METHODS AND CONDITION

3.1. INTRODUCTION

The most common mode of gear failure encountered in practice is that of surface contact fatigue. [27] This mode of failure leads to crack initiation at or near the contact surface, and may subsequently lead to damage varying in extent from microscopic pitting to severe spalling. The metal removed from the surface in such cases enters the machine system, and can, in turn, cause abrasive wear and failure of other components. Furthermore, the pits formed on the damaged surface lead to the formation of stress concentrations, and serve as initiation sites for other modes of gear failure.

There are many types of gear failures when someone analyzes actual gear in services however; these failures are categorized in two general groups. One is the failure of the root of the teeth when the bending strength is inadequate and the other is created on the surface of the gear. There are two theoretical formulas, which deal with the above two fatigue failure problems. One is the Lewis formula that is used to calculate the bending stress and Hertezian equation that is used to compute the contact stress. Various researchers are used variety of method to determine these stresses. The finite element method is very often used to analyze the state of stress in elastic bodies, which have complicated geometry like gears. [28]

In many engineering applications, such as rolling bearings, gears, cams, etc., machine components whose functioning depends upon rolling and sliding motion in contact along surfaces while under load. In this case, the contacting surfaces are non-conformal, hence the resulting contact areas are very small and the pressures are very high. From the point of view of machine design it is essential to know the values of stresses acting in such contacts. These stresses can be determined, from the analytical formulae, based on the theory of elasticity, developed by Hertz in1881 [25, 26].

Mechanism of contact fatigue

Whenever two curved (usually convex) surfaces are in contact under load, the contact occurs along a line or point, or, depending on the elastic constants of the materials concerned, along a very small circular or elliptical area. As a result of such small contact areas, the shear (Hertzian) stresses which develop at and near the surface are consequently very high. The maximum shear stress occurs at some distance below the surface [16] as illustrated in Fig.3. 1.

When the contacting stresses are repetitive, as is the case on the active flanks of gear teeth, the cyclic compressive stresses induced cause differing elastic and plastic behavior in the near surface material. Depending on the microstructure and grain orientation of the material in this region, internal stress concentrations are formed which can ultimately lead to crack initiation. In practice, crack initiation usually occurs at inclusions in the stressed near-surface material, the most deleterious being those inclusions which are hard, brittle and angular in shape.

When the contacting stresses are repetitive, as is the case on the active flanks of gear teeth, the cyclic compressive stresses induced cause differing elastic and plastic behavior in the near surface material. Depending on the microstructure and grain orientation of the material in this region, internal stress concentrations are formed which can ultimately lead to crack initiation. In practice, crack initiation usually occurs at inclusions in the stressed near-surface material, the most deleterious being those inclusions which are hard, brittle and angular in shape.

Damage due to contact fatigue in gear teeth usually occurs in one of three areas, viz along the pitch-line, in the addendum (i.e. above the pitch-line), and in the dedendum (i.e. below the pitch line) [17]. Along the pitch-line, only pure rolling stresses exist (Fig. 3.1), while away from the pitch-line, both rolling and sliding stresses are experienced. Stresses which develop at and near the surface are consequently very high. The maximum shear stress occurs at some distance below the surface [16] as illustrated in Fig.3. 1.

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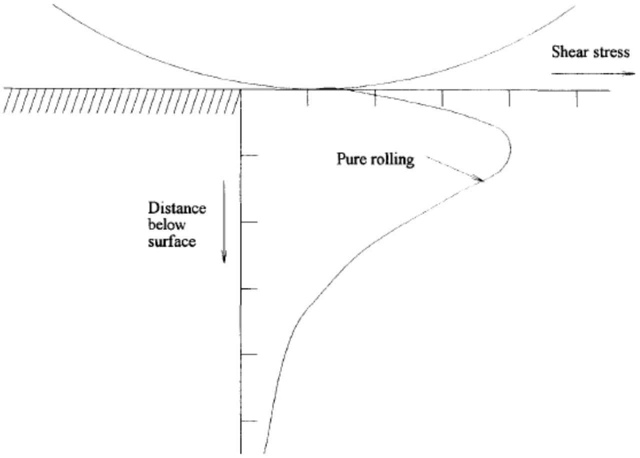


Fig.3.1 Stress distribution at and near to contacting surfaces under pure rolling

Sliding- Rolling Contact Fatigue

Pure rolling conditions prevail when the surface velocities of two contacting curved bodies are the same. However, if these velocities are different, an element of sliding is introduced which significantly alters the stress distribution in the surface and near-surface material. Depending on the relative velocities of the contacting bodies, rolling and sliding may occur in the same direction, positive sliding or in opposite directions negative sliding. The effect of the latter is that the surface material is rolled in one direction, and pushed (sliding) in another, therefore resulting in higher stresses than those encountered in positive sliding [18]. The modified stress distribution in the surface and near-surface material resulting from combined rolling and sliding is shown in Fig. 3.2.The position of maximum shear stress is moved closer to the contacting interface, and crack initiation therefore occurs at the surface.

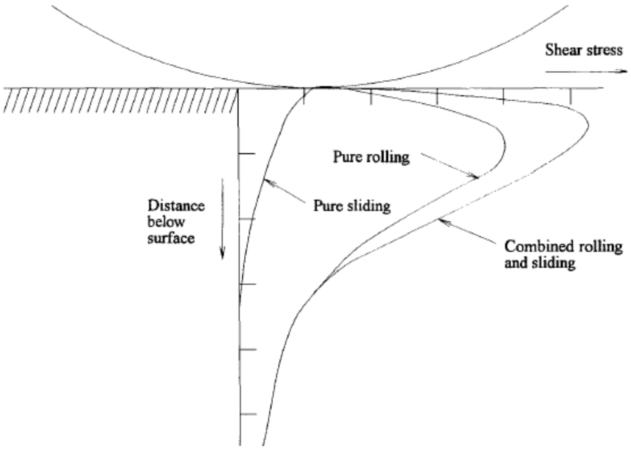


Fig 3.2 Stress distribution at and near to contacting surfaces under pure sliding-rolling condition

Gear teeth have complex combinations of sliding and rolling, which vary along the profile of each tooth, as illustrated in Fig. 3.3. In the addendum, the direction of rolling and sliding is the same, and positive sliding conditions therefore prevail. In the dedendum, however, the direction of rolling is opposite to that of sliding, and negative sliding conditions exist. Contact fatigue is therefore more likely to initiate in the dedendum, and pitting in this region is usually very severe, and often acts as a precursor to tooth bending fatigue [19].

In practice, it is common that contact fatigue damage will first occur in the dedendum of the smaller gear (which is usually the driving gear) of a gear set. This is explained by the fact that the smaller gear will undergo more revolutions, and therefore each tooth will experience a larger number of stress cycles. In order to prevent premature failure in such cases, it is common to make the smaller, driving gear harder than the other gears in the gear set.

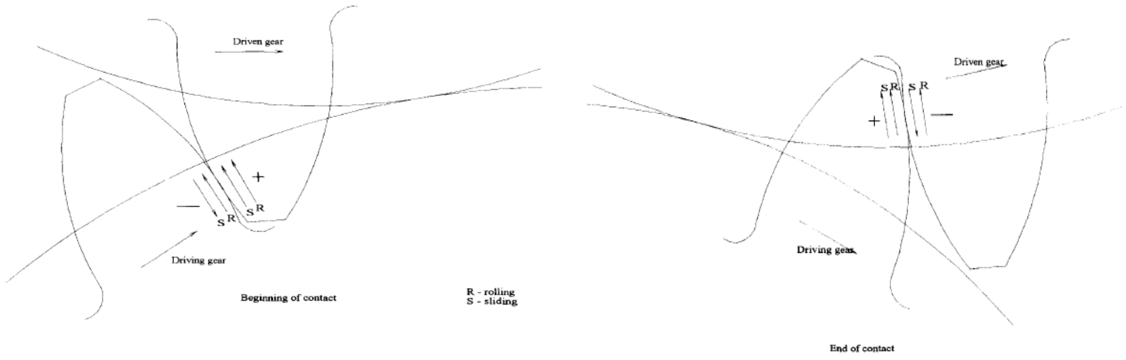


Fig 3.3 Combination of sliding and rolling in gear teeth

Another region in which contact fatigue damage is frequently encountered is at the lowest point of single tooth contact, i.e. the point at which contact is made with the tip of the matching tooth [17]. Since the contact area in this case is very small, high stresses are generated, even under normal loads. Moreover, the lowest point of single tooth contact is always in the dedendum of the matching tooth, and sliding speeds, both in approach and recess, are at a maximum. The negative sliding conditions in these regions, together with the high stresses and high sliding speeds, lead to rapid initiation of damage. The study by [20] shows in his testing report , the driving gear of a 1.5 ton lever hoist in which severe pitting in the dedendum has occurred at the point of single tooth contact . The extent of damage observed occurred after only 500 cycles under test conditions.

Unlike contact fatigue damage under pure rolling conditions, the sliding rolling action causes plastic deformation of the surface material, and this can usually be detected using metallographic analysis. The extent of plastic deformation, and hence of contact fatigue damage, can be reduced effectively by ensuring that correct lubrication conditions are maintained. Furthermore, it has been shown that sliding-rolling damage can be minimized by achieving a surface hardness on the matching bodies greater than HRC 60 [18] Retained austenite, at levels of 10-20%, in the case-hardened layer also minimizes the extent of contact fatigue since this micro-structural constituent deforms plastically to form larger contact areas, and hence lower contact stresses. However, austenite reduces the fatigue strength of the material, and can therefore introduce other deleterious effects.

3.2. METHODS

Most machine components are designed on the basis of stress in the main body of the member, that is, in portions of the body not affected by the localized stresses. In other words, damage to most mechanical components is associated with stresses and strains in portions of the component far removed from the points of application of the loads. However, in certain situations the contact stresses between the surfaces of two externally loaded bodies (e.g., meshing gear teeth) can be the significant stresses; that is the stresses on or somewhat below the surface of the contact are the major causes of damage to one or both of the bodies. Therefore, analysis Hertzian stresses provide important information about surface and subsurface stresses for static loading of concentrated contacts.

Modeling contact problem using parallel cylinders in contact

The radius of an involute gear tooth will change slightly across the width of contact with a mating tooth it is normal to ignore this and take the contact of spur gear teeth as equivalent to the contact of parallel cylinders with the same radius of curvature at the point of contact. The Hertzian equations can thus be applied to spur gears. Buckingham [30] shows that two contacting parallel cylinders can be used to fair accuracy to study contact stresses of spur gears. Hertz theory assumes that the gear teeth as an equivalent contacting cylinder as shown Fig (3.4).

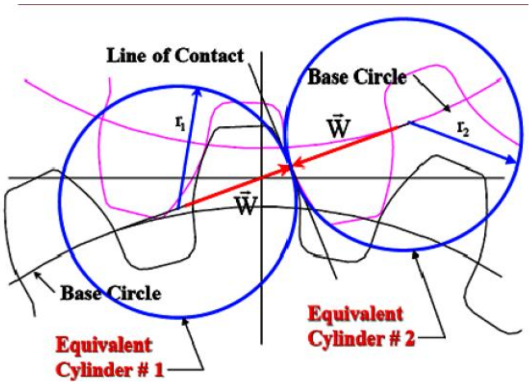


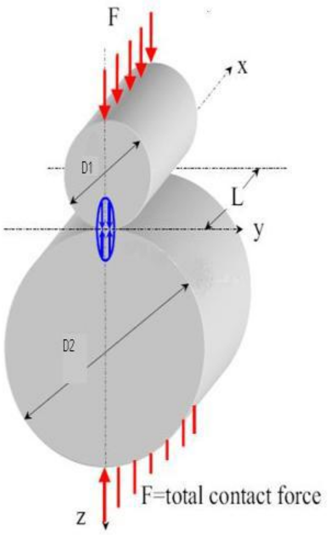
Figure 3.4: Equivalent Contacting Cylinders.

r_1 and r_2 is the radii of equivalent cylinders

$\overrightarrow{W}$ is the normal force

Hertz Theory of Contact

The contact between two surfaces gives rise to an area of contact, and a pressure distribution over this contact area. If the surfaces have a simple geometry, the contact theory of Hertz (1881) can be used for calculating the contact area and the pressure distribution. The shape of the contact area depends on the shape (curvature) of the contacting bodies. To illustrate the contact between two elastic bodies, the simpler contact of two parallel cylinders (Figure 3.2) is presented based on the Hertz theory.



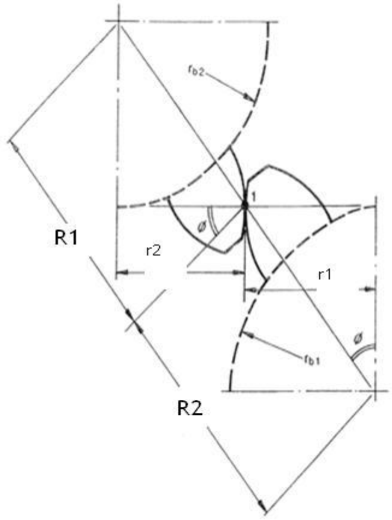


Fig 3.5 Cylinders in contact under compression Fig 3.6 Contacting gears at pitch point

In Fig.3.6 two gear teeth are shown in mating condition at the pitch point. Referring to Fig.3.7, the area of contact under load is a narrow rectangle of width 2B and length L. The stress distribution pattern is elliptical across the width as shown in figure 3.5.

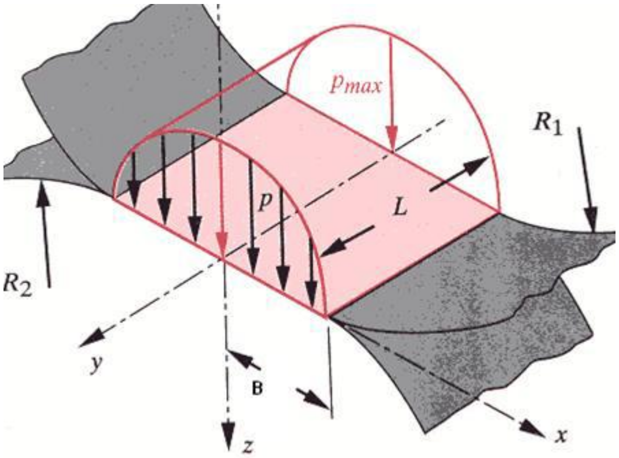


Fig 3.7 Ellipsoidal–prism pressure distribution

From Figure 3.7 the width of the contact zone is 2B. If total contact force is F and contact pressure is $pmax$, there is a formula [21], which shows the relationship between the force F and the pressure $pmax$

$Pmax$=$2F\pi BL$ --(3.1)

Contact width $B = \sqrt{\frac{4F}{\pi L} * \frac{{(1-v_1^2)}/{E_1} + {(1-v_2^2)}/{E_2}}{\frac{1}{R_1}+\frac{1}{R_2}}}$ --(3.2)

Half of the contact width can be written as$B = \sqrt{\frac{4*F*R_e}{\pi*L*E_e}}$ --------------------------------------(3.3)

Where Re is the equivalent radius of curvature, given by, $R_e = \frac{1}{R_1} + \frac{1}{R_2}$

Where $R1$ and$R2$ the instantaneous values of the radii of curvature on the pinion- and gear tooth profiles, respectively, at the point of contact.

At the pitch point R1= Rp & R2= Rg and given by

$$R_p = d_p sin\frac{\varphi}{2} \tag{3.4}$$

$$R_g = d_g sin\frac{\varphi}{2} \tag{3.5}$$

Where dp & dg are the pitch diameters, for Pinion and gear respectively and φ is the pressure angle. Therefore, at the pitch point Re becomes

$$R_e = \frac{1}{\frac{1}{R_p}+\frac{1}{R_g}} = \frac{1}{\frac{2}{\sin\varphi}*(\frac{1}{d_p}+\frac{1}{d_g})} \tag{3.6}$$

And Ee is the equivalent modulus of the material, given by

$$E_e = \frac{1}{{(1-v_1^2)}/{E_1}+{(1-v_1^2)}/{E_2}} \tag{3.7}$$

Substitute the value of B from equation (3.2) into (3.1), to find the maximum contact pressure.The maximum contact stress

$$P_{max} = \sqrt{\frac{FE_e}{\pi LR_e}} \tag{3.8}$$

For the driver pinion, and gear follower,

$$F = F_N = \frac{F_T}{cos\varphi} = \frac{T_p}{R_p cos\varphi} \tag{3.9}$$

$$P_{max} = \sqrt{\frac{T_p E_e}{\pi LR_e cos\varphi R_p}} \tag{3.10}$$

Where

Rp is the radius of the pinion and

Tp is the torque on the pinion

FN the normal force applied on the pinion

FT Tangential component of the force

Mechanical properties and its influence on fatigue life

The most common mechanical properties which are used in order to classify and identify materials are strength, ductility, hardness, impact resistance, and fracture toughness. These properties involve a reaction to applied loads. Structural materials are mostly un-isotropic and a material property varies with the orientation. One reason for the variation in properties of a material can be due to the directionality of the microstructure (texture) formed during the manufacturing processing of the material. Ductility is used as a quality control measure to know the impurities and proper processing of a material. It is defined as measure of the extent to which a material will deform before fracture.[32].

The conventional measures of ductility are the engineering strain at fracture (usually called the elongation) and the reduction of area at fracture. Hardness is defined as resistance of a material to localized deformation. The hardness is not considered as a basic property of a material, but rather a composite one with contributions from the yield strength, work hardening, true tensile strength, modulus, and others factors. The hardness measurements is very quick and considered as non destructive testing of materials when the marks or indentations produced by the test are in low stress areas and widely used for the quality control of the material

Toughness is defined as the ability of a metal to absorb energy during the deformation process before fracture. A good ductile material does not become a tough material. A combination of good strength and ductility leads to toughness. A material which shows high strength and high ductility will have higher toughness than a material having low strength and high ductility.

A metal can fail due to cyclic or dynamic loading conditions, but may possess satisfactory toughness results under static loads and therefore it can be said that the toughness of a material decreases with increase in loading and vice versa.

Tension, compression, bending, shear, and torsion are the five-basic and fundamental loading conditions that can be applied to a material.

The way in which the material is loaded will greatly affect its mechanical properties and largely determines how, or if, a component will fail; and whether it will show warning signs before failure actually occurs. Tensile strength is one important property of the material in analyzing the fatigue failure. Tensile strength is correlated to hardness. This correlation depends upon specific test data and cannot be extrapolated to include other materials not tested.

Materials

With improvements in steel making, there have been significant advances in the development of steels for fatigue-sensitive automotive applications, such as gears and shafts. Automotive design requirements that demand smaller and lighter components without sacrificing torque or force capacity have led to the need for components with significantly improved fatigue performance. To satisfy these needs, material/processing combinations that concentrate on surface microstructure have been developed. [34]

Improvements in fatigue performance in components are derived primarily by decreasing the surface cyclic tensile stress or by increasing the surface yield stress, thereby increasing the resistance to fatigue crack nucleation. To achieve these goals, common surface modification processes, which often simultaneously increase the surface yield stress and introduce a residual compressive stress to decrease the surface cyclic tensile stress, are based on heat treating (e.g. carburizing, carbo-nitriding, laser hardening, and induction hardening), non-uniform plastic deformation (e.g. peening and deep rolling), or selected surface alloy modification (e.g. ion implantation and chemical or physical vapor deposition). To realize the maximum improvement in fatigue performance these surface modification techniques must be carefully controlled and matched with the particular alloy of interest to ensure that undesirable features (e.g. micro-cracks, incorrect microstructure, etc) are not introduced during manufacture.[33]

The failure of carburized gears normally occurs through a process of fatigue and is usually attributed to a combination of the material properties along with gear design and mechanical misalignment. Improving the fatigue strength of gears is therefore of great importance in attaining increased load carrying capacities and in improving component reliability. Since most bending fatigue failures are initiated at or near to the surface [33], one of the most promising methods of improving fatigue strength is through the use of different treatments to modify the surface properties of the material.

Performance Requirements

- ✓ Transmission gears require:
- ✓ Hard wear resistant surfaces
- ✓ Resistance to tooth root bending fatigue
- ✓ Resistance to surface fatigue which leads to pitting
- ✓ A tough core
- ✓ Dimensional accuracy for smooth meshing and reduced (noise, vibration and harshness)
- ✓ Transmission of higher loads without increasing size and weight

Material Requirements

The majority of transmission gears are case hardened by carburizing to achieve a high surface hardness and a tough core. In addition to the standard grades, has been developed steel for transmission gears with all or some of the following enhancements:

Controlled hardenability steels ensure repeatability of mechanical properties and heat treatment distortion behavior. Controlled low silicon steels improve the bending fatigue life by reducing internal oxidation during carburizing.

Optimized sulphur content balances the conflicting benefits of low sulphur for improved bending fatigue properties and high sulphur for improved machinability. Clean steels provide good fatigue resistance from low overall inclusion content.

Since the objective of this thesis is to investigate the influence material on the contact stress of spur gear, for analysis purpose three materials with different chemical compositions was considered.

3.3. CONDITIONS

- ✓ Hertz’s model of contact stresses are based on the following simplifying assumptions [29]
- ✓ The materials in contact are homogeneous and the yield stress is not exceeded,
- ✓ Contact stress is caused by the load which is normal to the contact tangent plane which effectively means that there are no tangential forces acting between the solids,
- ✓ The contact area is very small compared with the dimensions of the contacting solids,
- ✓ The contacting solids are at rest and in equilibrium,
- ✓ The effect of surface roughness is negligible.

For this particular study, spur gear with the following parameters was considered

Table 3.1 Geometrical parameters for spur gear

Sr.No	Input Parameters	Symbols	Values
1	Module	M	2.5mm
2	Nominal Input power	P	2KW
3	Pinion speed	N1	50rpm
4	Number of pinion teeth	Z1	20
5	Number of gear teeth	Z2	20
6	Pitch circle diameter of pinion	d1	50mm
7	Pitch circle diameter of gear	d2	50mm
8	Pressure angle	φ	20

Table 3.2 Properties of materials under study

Gear Material	Modulus of elasticity (Gpa)	Poisson’s ratio
Chromium-molybdenum alloy steel (SCM 420)	206	0.3
Grey cast iron GG-20	113	0.25
Grey cast iron GG-30	91	0.25

3.4. CONTACT STRESS ANALYSIS

1. Material: Chromium-molybdenum alloy steel (SCM 420)

Modulus of elasticity (Gpa) = 206Gpa

Poisson's ratio =0.3

Nominal torque on the pinion shaft

$$T_p = 9550 * \frac{P_1}{N_1} = 9550 * \frac{2}{50} = 382Nm$$

The Hertizian contact stress is given by

$$P_{max} = \sqrt{\frac{T_P E_e}{\pi L R_p cos\varphi R_e}}$$

The material co-efficient Ee

$$E_e = \frac{1}{(1-v_1^2)\big/_{E_1} + (1-v_1^2)\big/_{E_2}}$$

$$E_e = \frac{1}{(1-0.3^2)\big/_{(2.06*10^5)} + (1-0.3^2)\big/_{(2.06*10^5)}}$$

$$E_e = 113.187 \times 10^3$$

Re The Pitch co-efficient

$$R_e = \frac{1}{\frac{1}{R_p} + \frac{1}{R_g}} = \frac{1}{\frac{2}{\sin\varphi} * (\frac{1}{d_p} + \frac{1}{d_g})}$$

$$R_e = \frac{1}{\frac{2}{\sin 20} * \left(\frac{1}{50} + \frac{1}{50}\right)} = 4.275$$

$$P_{max} = \sqrt{\frac{382000 * 113.187 \times 10^3}{\pi * 25 * 20 * cos20 * 4.275}}$$

$$P_{max} = 2617.7Mpa$$

$$\sigma_{VonMiss} = 0.57 * 2617.7 = 1492Mpa$$

2. Material: Grey cast iron GG-30

Modulus of elasticity (Gpa) = 113Gpa

Poisson's ratio =0.25

Nominal torque on the pinion shaft

$$T_p = 9550 * \frac{P_1}{N_1} = 9550 * \frac{2}{50} = 382Nm$$

The Hertizian contact stress is given by

$$P_{max} = \sqrt{\frac{T_P E_e}{\pi L R_p cos\varphi R_e}}$$

The material co-efficient Ee

$$E_e = \frac{1}{(1-v_1^2)\Big/E_1 + (1-v_1^2)\Big/E_2}$$

$$E_e = \frac{1}{(1-0.25^2)\Big/(1.13*10^5) + (1-0.25^2)\Big/(1.13*10^5)}$$

$$E_e = 60.267 \times 10^3$$

Re The Pitch co-efficient

$$R_e = \frac{1}{\frac{1}{R_p}+\frac{1}{R_g}} = \frac{1}{\frac{2}{\sin\varphi}*(\frac{1}{d_p}+\frac{1}{d_g})}$$

$$R_e = \frac{1}{\frac{2}{\sin 20}*\left(\frac{1}{50}+\frac{1}{50}\right)} = 3.42$$

$$P_{max} = \sqrt{\frac{382000*60.267\times 10^3}{\pi*25*20*cos20*3.42}}$$

$$P_{max} = 1910$$

$$\sigma_{VonMiss} = 0.57\sigma_H = 1088.75Mpa$$

3. Material: Grey cast iron GG-20

Modulus of elasticity (Gpa) = 91Gpa

Poisson’s ratio =0.25

Nominal torque on the pinion shaft

$$T_p = 9550 * \frac{P_1}{N_1} = 9550 * \frac{2}{50} = 382Nm$$

The Hertizian contact stress is given by

$$P_{max} = \sqrt{\frac{T_P E_e}{\pi L R_p cos\varphi R_e}}$$

The material co-efficient Ee

$$E_e = \frac{1}{(1 - v_1^2)\Big/_{E_1} + (1 - v_1^2)\Big/_{E_2}}$$

$$E_e = \frac{1}{(1 - 0.25^2)\Big/_{(0.91 * 10^5)} + (1 - 0.25^2)\Big/_{(0.91 * 10^5)}}$$

$$E_e = 48.533 \times 10^3$$

Re The Pitch co-efficient

$$R_e = \frac{1}{\frac{1}{R_p} + \frac{1}{R_g}} = \frac{1}{\frac{2}{\sin\varphi} * (\frac{1}{d_p} + \frac{1}{d_g})}$$

$$R_e = \frac{1}{\frac{2}{\sin 20} * \left(\frac{1}{50} + \frac{1}{50}\right)} = 3.42$$

$$P_{max} = \sqrt{\frac{382000 * 48.533 \times 10^3}{\pi * 25 * 20 * cos20 * 3.42}}$$

$$P_{max} = 1714Mpa$$

$\sigma_{VonMiss} = 0.57 * 1714 = 977Mpa$

Table 3.3 Hertzian analytical results for different materials

Sr.No	Material	Maximum contact stress(Mpa)
1	Chromium-molybdenum alloy steel (SCM 420)	1492
2	Grey cast iron GG-30	1088
3	Grey cast iron GG-20	977

The above table summarized the Hertizian analytical result for three materials under study.As seen from the result the chromium molybdenum alloy have the maximum Hertizian stress, this means that this material can resist higher stress than the other materials which is considered in this study.

CHAPTER FOUR

4. FEM ANALYSIS METHODS AND CONDITIONS

4.1 INTRODUCTION

With the rapid development of computational mechanics, however, great progress has been made in numerical analysis of the problem. Using the finite element method, many contact problems, ranging from relatively simple ones to quite complicated ones, can be solved with high accuracy. The Finite Element Method can be considered the favorite method to treat contact problems, because of its proven success in treating a wide range of engineering problem in areas of solid mechanics, fluid flow, heat transfer, and for electromagnetic field and coupled field problems. [28]

ANSYS is the name commonly used for ANSYS mechanical, general-purpose finite element analysis (FEA) computer aided engineering software tools developed by ANSYS Inc. ANSYS mechanical is a self contained analysis tool incorporating pre-processing such as creation of geometry and meshing, solver and post processing modules in a unified graphical user interface. ANSYS is a general-purpose finite element-modeling package for numerically solving a wide variety of mechanical and other engineering problems. These problems include linear structural and contact analysis that is non-linear. [31]

Among the various FEM packages, in this work ANSYS is used to perform the analysis. The following steps are used in the solution procedure using ANSYS:

- ✓ The geometry of the gear to be analyzed is imported
- ✓ The element type and materials properties such as Young's modulus and Poisson's ratio are specified.
- ✓ Meshing the three-dimensional gear model.
- ✓ The boundary conditions and external loads are applied.

- ✓ The solution is generated based on the previous input parameters.
- ✓ Finally, the solution is displayed.

4.2 FEM ANALYSIS METHODS

Here the methods for the analysis of finite element will be discussed

4.2.1MODELING OF MATTING SPUR GEAR

With the basic understanding of how finite element programs work, a finite element model must be created with appropriate parameters such as dimensions, loads, constraints, element choice, mesh selection, etc. In a way, creating the finite element model is the most time consuming step of finite element analysis. Users should spend time to create the model as accurately as possible since geometry is one of the critical aspects in FEA. In ANSYS, there are two different methods to construct the model. The first method is to build the model in a computer-aided design (CAD) environment such as SOLIDWORKS, Pro/ENGINEER, or CATIA, and export the model with a file format such as IGES, ACIS, or Parasolid. The file is then imported into ANSYS for set up and analysis. However, the main disadvantage for this method is the CAD geometry data could be lost during the translation of the model, which means the dimensions of the model are no longer exact. The second method is to use ANSYS internal drawing capabilities to build the model. In this method, no geometry data is lost since the file does not need to be translated. However, the modeling functions in ANSYS are not as good as the other CAD programs; users often encounter difficulties for building complex models due to the interface limitations.

For this research, the second method was chosen because a large amount of geometry data was lost during translate from CAD files. The gear tooth involutes are no longer exact after being imported into ANSYS. Therefore, gears will be drawn entirely in ANSYS workbench.

The remaining parts should be relatively straight forward. The generated tooth was saved as a geometry file and another ANSYS workbench window opened and the geometer is imported to the workbench interface. The pinion and the gear will need to be assembled correctly in order to perform analysis. Usually, the actual position of the contact is unknown. However, the pitch circles of pinion and gear should be in contact once they are assembled. Use the Translate Instance function to adjust the distance so that the pitch circles are in contact. The following figures have shown the assembled gear set and its proper positions.

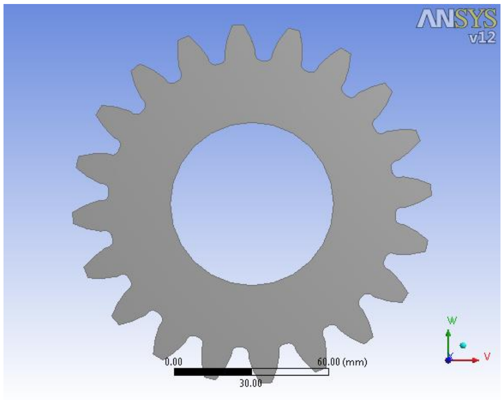


Fig: 4.1. Geometry of Spur gear

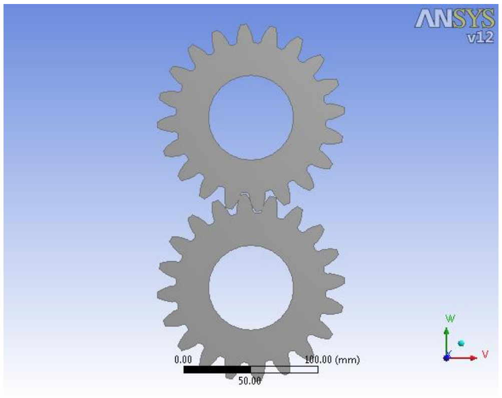


Fig: 4.2. Assembled gear set

After the assembly is imported in ANSYS Workbench 12, assembly is subjected to the boundary conditions. In this thesis it is assumed that the one gear is fixed and the other gears is given torque along its axis.

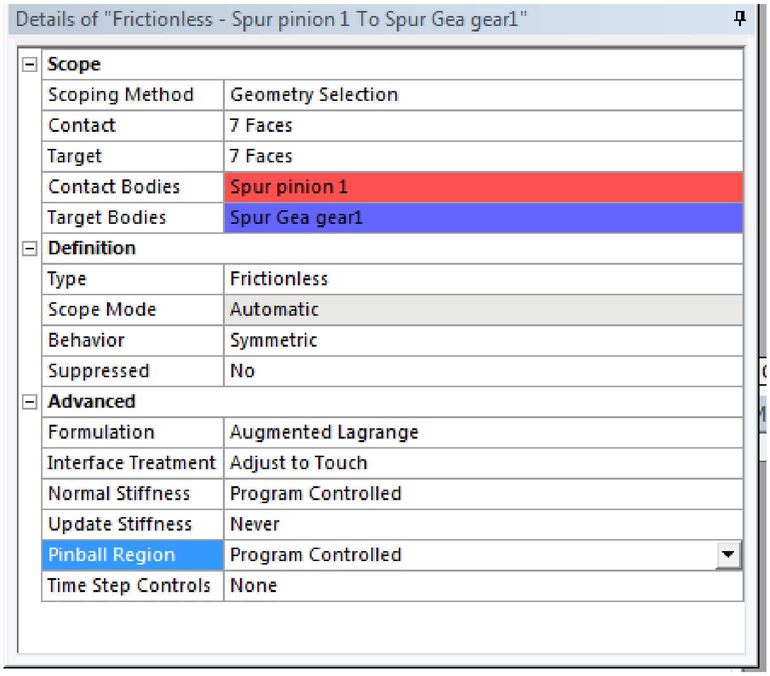
ANSYS has many type of analysis, so it is necessary to select the correct type of analysis from the menu bar. As the imported geometry is 3-Dimensional, select 3-D and Static Structural Analysis from menu and connect the geometry to the analysis tab.

4.2.2. DIFINING MATERIAL PROPERTIES

Then the next step is to enter the Young's Modulus and Poisson's ratio and density of the gear material that has been used for analytical calculation in the previous chapter. This can be done by selecting the Engineering Data from the analysis tab of the ANSYS Workbench and inserting the corresponding values.

4.2.3. DEFINING CONTACT REGION

Once the geometry is attached with Static Structural analysis tab, next thing is to define the contact between the two involutes teeth. ANSYS has an inbuilt option, which automatically reads the attached geometry for any predefined contacts or other boundary definitions. The contact between the two teeth is assumed to be frictionless; the figure below shows the contact being defined as frictionless. One of the most important things is to change the Interface Treatment to ‟ Adjust to touch‟. This option defines the kind of contact between the selected bodies. The figure below shows the image from ANSYS showing the contact defined for the two spur gear teeth in mesh.



Details of "Frictionless - Spur pinion 1 To Spur Gea gear1"

Scope	
Scoping Method	Geometry Selection
Contact	7 Faces
Target	7 Faces
Contact Bodies	Spur pinion 1
Target Bodies	Spur Gea gear1
Definition	
Type	Frictionless
Scope Mode	Automatic
Behavior	Symmetric
Suppressed	No
Advanced	
Formulation	Augmented Lagrange
Interface Treatment	Adjust to Touch
Normal Stiffness	Program Controlled
Update Stiffness	Never
Pinball Region	Program Controlled
Time Step Controls	None

4.2.4. MESH GENERATION

The mesh with the default settings is not adequate to get the accurate results. In this analysis both the gear were finely meshed with “Sizing‟ option in menu. The element size was chosen to be 0.001 and it was then refined at the contacting surfaces to get the finer mesh and continuous stress values. The image below shows the meshed assembly according to the default size.

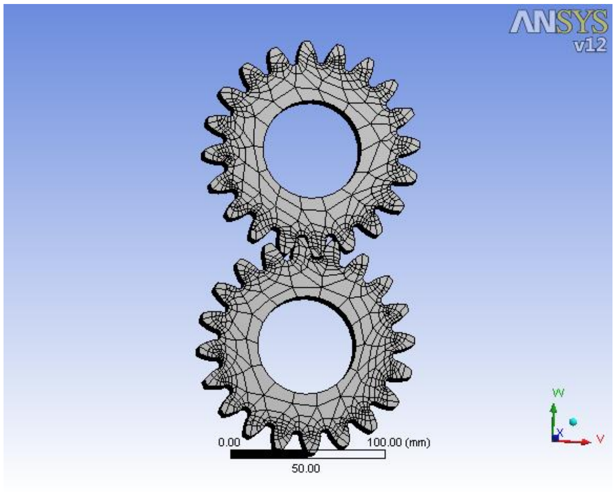
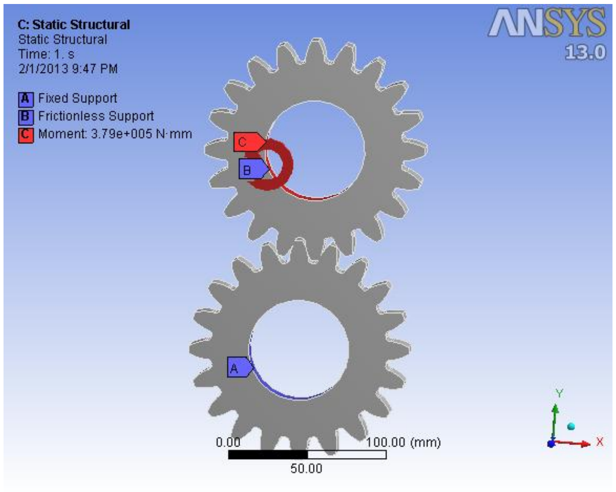


Fig: 4.4 Meshed assemblies

4.2.5. SUPPORTS AND LOADS

The lower gear is given a fixed support and the top gear is given frictionless support. The top gear is also given a toque or a moment in clockwise direction. The image below shows how the supports and loads were applied to the gear model.


C: Static Structural
Static Structural
Time: 1. s
2/1/2013 9:47 PM

A Fixed Support
B Frictionless Support
C Moment: 3.79e+005 N·mm

Fig 4.5.Applied Boundary Condition

4.3. CONDITIONS

Here the conditions that has been considered for the FEM analysis is discussed

The following geometrical data are used to develop the geometry of the spur gear in ANSYS workbench

Table. 4.1. Geometrical data for geometry of the spur gear

Sr.No	Input parameters	Values
1	Module	2.5
2	Pitch circle diameter	50mm
3	Outside diameter	140mm
4	Shaft diameter	64mm
5	Number of teeth	20mm
6	Face width	20mm

Table .4.2. Material properties

Gear Material	Modulus of elasticity (Gpa)	Poisson’s ratio
Chromium-molybdenum alloy steel (SCM 420)	206	0.3
Grey cast iron GG-20	113	0.25
Grey cast iron GG-30	91	0.25

4.4. CONTACT STRESS ANALYSIS

Fatigue analysis requires an accurate description of the fatigue material properties. In ANSYS Workbench, sample fatigue curves are available only for structural steel and aluminum alloy materials. This is because, fatigue analysis is empirical and it becomes very difficult to take in to account all materials type. Therefore in doing fatigue analysis in ANSYS Workbench, users must provide their own fatigue data.

Engineering data is a resource for material properties used in an analysis system. Engineering data can be used as repository for company or department data, such as material data libraries. The engineering data workspace is designed to allow to create, save and retrieve material models, as well as to create libraries of data that can be saved and used in subsequent project and by other users. Therefore the materials that were used for analytical determination of stress that was listed in table 4.2 has been added to the engineering data and used for the stress evaluation in ANSYS workbench.

As already mentioned high contact stresses results in pitting failure of the gear tooth, it is necessary to keep contact stresses under limit. In this thesis VonMises Contact stresses are obtained at the contact region. The image below from ANSYS shows the VonMises the contact for three different materials.

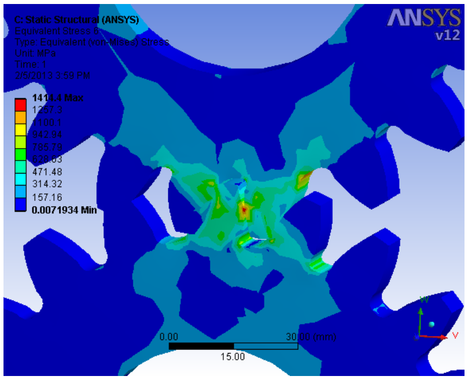


Fig 4.6.VonMises Contact stresses for Chromium-molybdenum alloy steel (SCM 420).

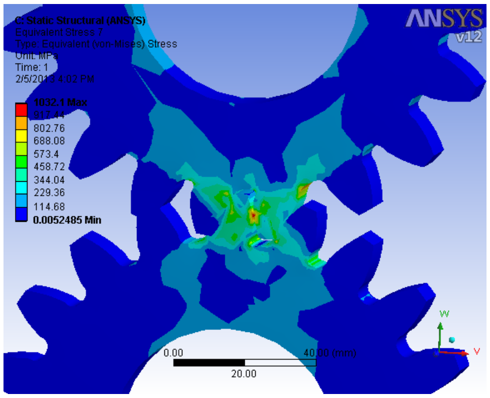


Fig 4.7. VonMises Contact stresses Grey cast iron GG-30.

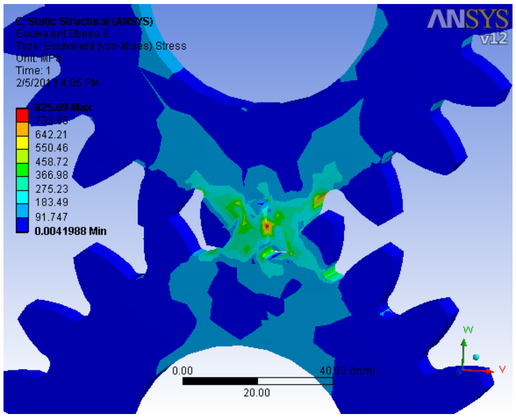


Fig.4.8. VonMises Contact stresses for Grey cast iron GG-20

The goal of stress analysis presented in this section is to investigate the change in stress with material types using ANSYS. Maximum contact pressure results for three different materials are presented using contours and numerical values. Further interpretation of the result and comparison with the Hertzian theory will be discussed on the next chapter.

CHAPTER FIVE

5. RESULTS AND DISCUSSION

5.1 RESULTS

The goal of stress analysis presented in this section is to investigate the change in stress with material types. Maximum contact pressure results for different material, using both ANSYS and Hertzian theory under a constant loading condition of torque 382 Nm is plotted in Figure 5.1 As it is seen in the figure the ANSYS model agrees well with the theoretical results. The error for all loading condition is computed and shown in table 5.1.

Table 5.1: Comparison of Maximum contact pressure for different material

Gear material	ANSYS Max Contact Pressure (Mpa)	Hertzian Max Contact Pressure (Mpa)	Error (%)
Chromium-molybdenum alloy steel (SCM 420)	1414	1492	-5.23
Grey cast iron GG-30	1032	1088	-5.147
Grey cast iron GG-20	825	977	-15.4

5.2 DISCUSSION

In this section, results of finite element approach for contact stress estimation in ANSYS and Hertz theory, for the material Chromium-molybdenum alloy steel (SCM 420) is compared. The result of the analysis summarized as follow. The maximum contact stress value for this material was 1492MPa which obtained by Hertz theory, the well known theoretical calculations method for contact problem. The stress analysis has been done by ANSYS software, where the results have been presented by contours and numerical values the maximum values are presented red color in the contour diagram and minimum by blue. The maximum contact pressure result from ANSYS for 3D model is 1414MPa. The error between the theoretical and ANSYS 3D model, is about -5.23%.

Results of finite element approach for contact stress estimation in ANSYS and Hertz theory, for the material Grey cast iron GG-30, is compared. The result of the analysis summarized as follow. The maximum contact stress value for this material was 977MPa which obtained by Hertz theory, the well known theoretical calculations method for contact problem. The stress analysis has been done by ANSYS software, where the results have been presented by contours and numerical values. The maximum contact pressure result from ANSYS for 3D model is 825MPa. The error between the theoretical and ANSYS 3D model, is about -15.4%.

Results of finite element approach for contact stress estimation in ANSYS and Hertz theory, for the material Grey cast iron GG-20 is compared. The result of the analysis summarized as follow. The maximum contact stress value for this material was 1088MPa which obtained by Hertz theory, the well known theoretical calculations method for contact problem. The stress analysis has been done by ANSYS software, where the results have been presented by contours and numerical values. The maximum contact pressure result from ANSYS for 3D model is 1032MPa. The error between the theoretical and ANSYS 3D model, is about -5.147%.

6. CONCLUSION AND FUTURE WORK

6.1 CONCLUSION

Analytical method of gear analysis uses a number of assumptions and simplifications and it is intended to determine the maximum stress values. In this paper, numerical approach has used for predicting surface contact stresses of involute spur gear. A parametric study is also made by varying the material to investigate their effect on the pitting resistance of spur gears.

The surface strength of the gear tooth is a crucial parameter to prevent failure. In this study, it is shown that the effective method to estimate the contact stresses using three dimensional model of the gear and to verify the accuracy of this method the results with different materials are compared with the analytical Hertizian standard formula

It is also shown that the development of finite element analysis model of the Spur gear assembly to simulate the contact stress between two gears reasonably and obtained result is compared with the Hertzian theoretical equation. Based on the result from the contact stress analysis the hardness of the gear tooth can be improved to resist pitting failure.

Finally 3D meshed involute spur gear was modeled to investigate, contact stress, the change in contact pressure with applied torque, the relation between elasticity of material with different stress component and maximum contact pressure for three different materials was studied and the results were compared. FEA results are correlated with theoretical formulation, with tolerable accuracy.

6.2 FUTURE WORK

In this thesis work surface contact stress is studied for different materials using analytical and FEM numerical method , it can be further studied by performing experiments for a better result in addition to that other influencing factor is not studied in this paper . So this work is restricted to the specified cases. However, this paper can be extended to other situation listed below. Further numerical method investigations should be conducted on:

- ✓ The effect of surface finish on the contact stress of gears
- ✓ The effect of loading on the contact stress of gears
- ✓ Effect of temperature in surface contact stress of gears.
- ✓ Fracture mechanics approach to study surface related to initiation and propagation of cracks.
- ✓ Contact mechanics in gears under lubricated condition and its effect in surface contact stress
- ✓ Effect of misalignment, sliding speed and friction on surface contact stress.

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