ADDIS ABABA INSTITUTE OF TECHNOLOGY SCHOOL OF

MECHANICAL AND INDUSTRIAL ENGINEERING

EFFECT OF STRESS DISTRIBUTION ON SURFACE FAILURE OF SPUR GEAR

BY Shikur Ali

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Effect of Stress Distribution on Surface failure of Spur Gear

Master's Thesis in Mechanical Engineering (Mechanical Design Stream)

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ABSTRACT

One of the main gear damage mechanisms is the formation of pitting and spalling on the tooth flank. Several factors have significant influence on the damage formation, such as: contact stress level; tooth profile type; relative contact speed; surface finish and lubrication conditions.

For this particular study the contact stress level (stress distribution) is the main factor which is considered for the surface failure.

Surface failure is the critical failure mechanism that happens between two contacting solid bodies in sliding-rolling contacts. The stress distribution has a great effect on the surface failure of the spur gear teeth contact. In this thesis work the gear teeth contact simulation have done using two rollers contact having low and high speeds where the high speed roller is being tapered at the end in both sides.[12] The two rollers geometry have modelled using solid work premium and then imported to ANSYS Workbench 15 for analysis. The normal external load 5.7 KN ~ 29.6 KN is applied and the corresponding stress distribution have been evaluated. So that the maximum von misses stress distribution happened at the tapered end of the contact and the crack has been generated. As a result at tapered end of the roller a surface failure happened due to the maximum equivalent stress. The application area of the gear is for automobile power transmission systems.

Keywords- Contact stress, spur gear, roller, rolling-sliding contact, stress distribution, Surface failure.

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LIST OF SYMBOLS	NAMES
FEM	Finite Element Method
3D	Three Dimensional
2D	Two Dimensional
E	Modulus of Elasticity
υ	Poison‟s ratio
ρ	Density
ID	Internal Diameter
OD	Outer Diameter
rpm	revolution per minute
SEM	Scanning Electron Microscope
Ch-Mo	Chromium Molybdenum
SCM	Steel Chromium Molybdenum
σ	Stress
HSR	High Speed Roller
LSR	Low Speed Roller
CAD	Computer Aided Design
IGES	Initial Graphic Exchange System
WB	Workbench
GUI	Graphical User Interface

CHAPTER ONE

1. INTRODUCTION

1.1 BACKGROUND

The subject of this thesis is the effect of stress distribution on surface failure of a spur gear. Gear is a commonly used mechanical component in power transmissions and is frequently responsible for failure. Surfaces that are repeatedly loaded by rolling and sliding motion may be exposed to contact fatigue. Typical mechanical applications where such contact arises are: gears; cams; bearings; rail/wheel contacts.[1] For these mechanical components application we can see the rolling/sliding effect using rollers. These components are in general exposed to high contact stresses.[3] Despite the different purposes, they will suffer to contact fatigue damage if other failure modes are avoided. Even if the contact stresses may be of the same magnitude for different applications the resulting fatigue damage may vary. This is due to the fact that the contact conditions differ between the applications.

The main focus has been on surface fatigue damage or wears in the rollers contact. However, similar surface damage has been found in the other applications as well. The hypothesis was that local surface contacts can be of importance for surface initiated contact damage. The hypothesis can be visualized as local contact between asperities. When two surfaces are brought into contact the real contact area will be smaller than the apparent, this due to the fact that the contact surface is not perfectly smooth. Thus, the asperities will support the contact. The local asperity contacts will introduce stress concentrations, which will change the nominal stress field. For non-conformal mechanical contacts that are susceptible for rolling contact fatigue the nominal stress fields are of hertzian type. [1]

Technological progress usually leads to increasing demands on all fields of design and manufacture, and consequently on the design and application of contacts in relative motion supporting load. The solution of the diversified contact problems involved occasionally requires a detailed knowledge of friction and wear mechanisms and theories on the part of the designer. This knowledge is a part of an ever growing area called tribology.

The interdisciplinary nature of tribology, with knowledge drawn from different disciplines such as mechanical engineering, materials science, chemistry, and physics, leads to a general tendency for the chemist to describe in detail, for instance, lubricant additives, and for the mechanical engineer to discuss, for example, sliding journal bearings, with no overall guide to the subject. Also it is difficult to find in books dealing with tribology problems in general, a focused and advanced treatment of certain practically important topics in a comprehensive and thorough way. This is certainly true with rolling contacts.

The ancient Egyptians and Greeks are believed to have made effective use of the principle of the rolling contact, and a reference exists to a rolling bearing devised by the Greek Diades in 330B.C .for a battering ram which incorporated the essential principles of a rolling bearing as made at the present time. Fragments of what appears to resemble a ball thrust bearing were found in Lake Nemi, Italy, in1928[1]. It was speculated that it was used to support a rotatable statue and may have been made about 12A.D. Leonardo daVinci [2] studied among other things, the differences between sliding and rolling, but this aspect of his work was not generally known until a publication that appeared in the late nineteenth century.

On the British scene, an iron ball thrust bearing with many design characteristics of a modern bearing made its appearance about 1780 for use in a post mill in the Norwich area. A book published by Varlo in 1772 [3] describes a ball bearing he designed and fitted to his post chaise. British Patent 1580 was assigned to John Garnett of Gloucester in1787for interesting arrangements of various types of rolling element to form a bearing. In 1794, British Patent 2006 was granted to Philip Vaughan of Carmarthen for a radial ball bearing, the first of its kind on record. Important engineering developments of the rolling bearing continued in the early and middle part of the eighteenth century, but the main impetus that led to the foundation of the rolling bearing manufacturing industry came from the invention of the bicycle in Scotland in about 1840.

In1881,Heinrich Hertz[4] published in Germany his study on deformation of curved elastic bodies in contact , providing the growing industry with a mathematical theory that is used to the present time.

Modern gears are an extension of primitive friction wheels which transmitted motion from one revolving shaft to another. Under low load conditions where constant velocity was not a requisite, friction wheels performed adequately- but as greater demands were placed on mechanical power transmission systems, first cogwheels and then gears were developed.

Before the year 1700, there were numerous tooth designs in use. However, shortly after the turn of the 18^{th} century, the involute tooth was developed and it remains the most common form in use today. Most other tooth forms had to be custom cut for each application, but the involute curve made it possible to use one tool to generate gears having the same pitch (or size) having anything from ten to hundreds of teeth. Other benefits of the involute are that the centre distance between two involute may vary somewhat, without loss of operating smoothness, and that similar gears may be interchanged without loss of efficiency [11].

The common normal to the tooth profile at the point of contact must always pass through a fixed point called the pitch point in order to maintain a constant angular velocity ratio of the two gears. The involute curve satisfies this fundamental law of gearing and is commonly used for gear teeth profiles [3].

The load carrying capacity of any gear drive may be limited by several factors like:- excessive heat of operation, breaking of gear teeth, insufficient lubrication, misalignment etc.

1.2 LITERATURE REVIEW

1.2.1. Introduction

This chapter is a review of some of the relevant literature gathered on gears. Surface failure of gear addressed in general from different point of views.

1.2.2 Definitions of gear terms

The terminology of gear teeth is illustrated in fig.1

Addendum is the height of the tooth above the pitch circle.

Base circle is the circle from which the involute tooth curve is generated.

Circular pitch Pc is the length of the arc of the pitch circle between the centres or corresponding points of adjacent teeth

Clearance is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.

Dedendum is the depth of tooth space below the pitch circle or the radial dimension between the pitch circle and the bottom of the tooth space.

Involute curve is the shape of a gear tooth traced out by knot on a taut in extendable cord unwound from a circle

Module m is the ratio of the pitch diameter to the number of teeth.

Pitch diameter d_p is the diameter of the pitch circle

Pitch point of tooth profile is at its intersection with the pitch circle.

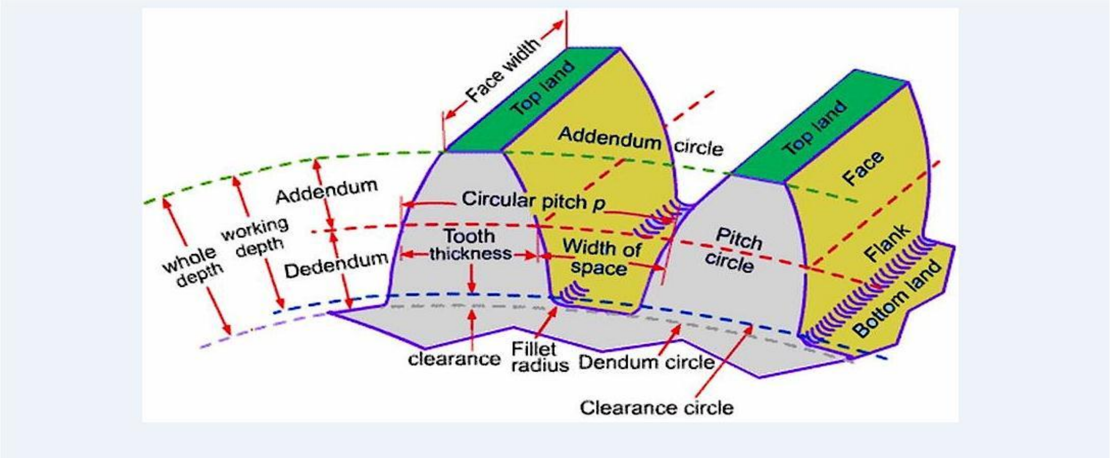


Figure 1 Nomenclature of gear teeth

1.2.3 Gear Materials

Throughout history many materials have been successfully utilized to manufacture gears. At one time millwrights made wooden cogs from applewood or hombeam. Cast iron was an early favourite when tooth loadings were moderate. Modern technology in cast iron design has brought this material to the forefront again, because it has advantageous surface properties in that its crystalline, slightly porous structure retains lubricant well.

However, steel is the most widely used material today, and the main point in its favour is that it can be surface hardened. Alloy or carbon steels are used for spur, helical and bevel gears. The pinion is usually manufactured from harder steel than the wheel, as there is greater tooth loading on the pinion, and the pinion, as the smaller gear, has teeth in mesh more often than the teeth on the wheel. Mild steel is not very satisfactory as a gear material, because its load carrying capacity in the line contact area is low. If the lubricant film fails, or there is dirt in the oil, welding of the mating surfaces can occur. The lowest carbon content in steel for industrial gears is 0.4% when used in an unhardened state [11].

In order to harden low- carbon steel it is necessary to increase the carbon content of the surface of the steel so that a thin outer case can be hardened by heating the steel to the hardening temperature and then quenching it. The process, therefore, involves two separate operations. The first is the carburizing operation for impregnating the outer surface with sufficient carbon, and the second operation is that of heat treating the carburized parts so as to obtain a hard outer case and, at the same time, give the core the required physical properties. The term case- hardening is ordinarily used to indicate the complete process of carburizing and hardening [11].

For highly loaded applications, case-hardened steels are employed. Hard and wear-resistant steels of the homogenous variety tend to the brittle and lack the strength and ductility required for many engineering applications. The required combination of hardness and strength is obtained when only the outer layers of a component are harder than the core-very necessary in the case of gears. Case-hardening provides this important characteristic-the molecular structure of the steel is changed to give a hard, martensitic formation on the outer layers. Various methods of hardening are used, but they all have to be followed by a machining process because of the distortion caused by the heat. The hard case must be thick enough to allow for this machining.

1.2.4 Surface failure of gear teeth

Surface deformation of gear teeth is a sign that the surface stresses associated with the tooth loading have been too high for material, and the reason lies in any of: defective material of the gear, defect in the manufacturing of the gear, defective lubricant foreign matter in the lubricant, defect in the mounting of the gears dimensions and inadequate for the applied loads.

Considering the character or the smoothness of the surface, there are six distinct types of wear that have been noted on gear-tooth profiles in service. Some of the identified types of wear appear to be caused by failure of the lubricant. Other types of wear appear to be substantially independent of the lubricant, although some of them may be accentuated by inadequate lubricant [5].

a) Abrasive wear

Abrasive wear is due to the presence of foreign material within the gear case

b) Scoring wear

Scoring wear is happened when high and localized flash temperatures cause tiny spots on the contacting teeth weld together. As the teeth separate, the welds are torn apart, leaving roughened surfaces

c) Pitting

Pitting is happened due to the surface fatigue failure of a material as a result of repeated surface stresses that exceed the endurance limit of the material.

d) Spalling or flaking

In flaking (called spalling in the USA), intense local loads compress the metal below the surface. In time small cracks appear which eventually join up to loosen flakes from the surface.

e) Scuffing

Scuffing occurs under sever compressive stress and high sliding velocity. When numerous fine scratch lines are found in the direction of sliding, the condition is known as ‘scuffing’.

f) Seizing

Gears become stuck from undue heat and friction.

1.2.5 Surface contact fatigue failures in gears

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. e.g tooth bending fatigue

1.2.6 Mechanism of contact fatigue

When two bodies not mechanically joined touch each other without becoming rigidly connected, it is said that they are in contact. They can come in to contact either at a point or along a line or over a surface or a combination. Contact stresses are caused by the pressure of one solid body on another over limited areas of contact. In some cases, the contact stresses are experienced when two surfaces are pressed together by external loads. Contact stresses may be considered the major cause of failure of one body or both contacting bodies. The contact region transmits the forces from one body to the other by means of compressive and tangential or shear stresses if friction is presented. Typical failures are seen as cracks, pits or flaking in the surface of the material. [13]

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 (hertizian) stresses which develop at and near the surface are consequently very high. The maximum shear stress occurs at some distance below the surface, as illustrated in fig 8.

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 behaviour 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). Along the pitch line, only pure stresses exist (fig 5), while away from the pitch line, both rolling and sliding stresses are experienced. In the subsequent sections of this paper, surface damage under both pure rolling and rolling-sliding conditions, as well as a more advanced and severe mode of surface damage called spalling.

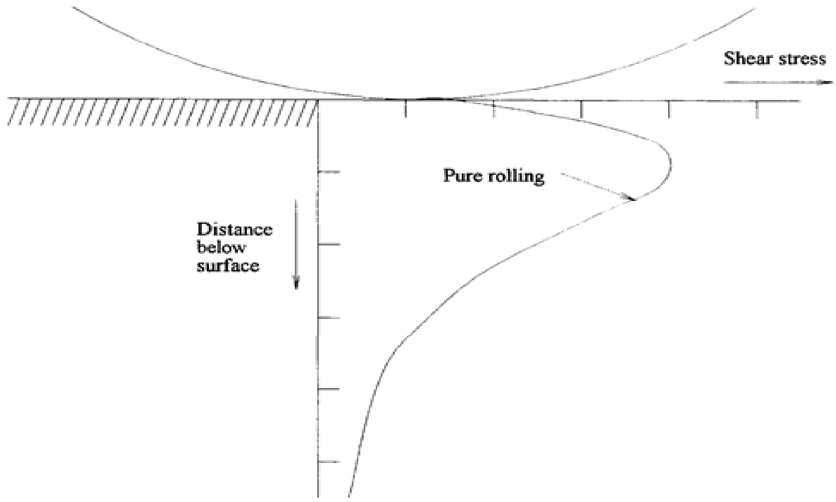


Figure 2 stress distribution at and near two contacting surfaces under pure rolling[13]

The stress distribution resulting from pure rolling conditions prevalent at the pitch line is shown in fig.2. The maximum shear stress occurs at some distance, usually 0.18-0.3 mm [3], below the surface, and just a head of the contact point. Cracks initiate at the point of maximum stress, and propagate essentially parallel to the surface. Continued rolling may cause the cracks to deviate up towards the contact surface, resulting in the removal of metal from the surface. The pits formed in this manner initially have side's perpendicular to the contact surface, as illustrated schematically in fig.3 Continued operation of the pitted surface, however, may cause a breakdown in the pit shape.

Pitting under pure rolling can occur even under proper lubrication conditions, since oil, as an incompressible fluid, will merely transmit the contact loads. The pits formed are generally very small and seldom give more than a „frosted‟ appearance to the damaged surface. In some cases, the pits may not progress beyond their point of origin, and may even be self-healing. There are two unique characteristics of rolling contact fatigue pits which can be used to distinguish this type of damage from other forms of pitting.

Firstly, the formation of rolling contact fatigue pits occurs with no surface plastic deformation. This is contrary to pitting under sliding-rolling conditions. Secondly, in components with a case-hardened layer consisting of martensite with little or no retained austenite, rolling contact fatigue leads to the formation of a microstructural feature referred to as “butterfly wings”. These are formed when plastic deformation is constrained by the surrounding material, and is more common when shear stresses are extremely high. [3].

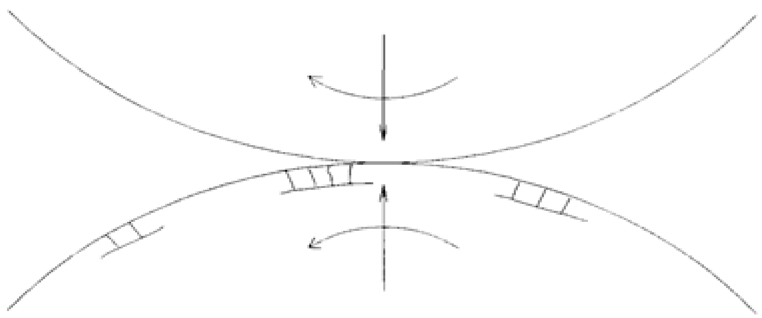


Figure 3 Pits formed initially perpendicular to surface contact [13]

1.2.7 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 direction (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. The modified stress in the surface and near-surface material resulting from combined rolling and sliding is shown in fig.8. The position of maximum shear stress is moved closer to the contacting interface, and crack initiation therefore occurs at the surface.

Gear teeth have complex combinations of sliding and rolling, which vary along the profile of each tooth, as illustrated in fig.11. 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.

Sliding motion is developed during gear contact as the driving gear tooth contacts the driven gear tooth. The first point of contact occurs below the pitch line of the driving gear tooth, otherwise known as the dedendum. Conversely the initial point of contact on the driven gear tooth occurs above the pitch line in the tooth addendum. The pitch line is defined as the singular point on the gear tooth profile where pure rolling occurs. The most sever sliding (highest slide to roll ratio) occurs at the beginning and the end of the contact. The extent of sliding decreases as the tooth profiles roll/slide toward their respective pitch lines. The driving gear tooth experiences negative slide, that is, the sliding direction is opposite to the rolling direction. The driven tooth experiences positive slide because the rolling and sliding motions are in the same direction. After the gear tooth roll past their pitch lines during contact the positive and negative sliding behaviour is reversed. The driving gear is contacted on its dedendum and experiences negative slide. This concept is shown schematically in fig.7 Negative sliding always occurs in the tooth dedendum.

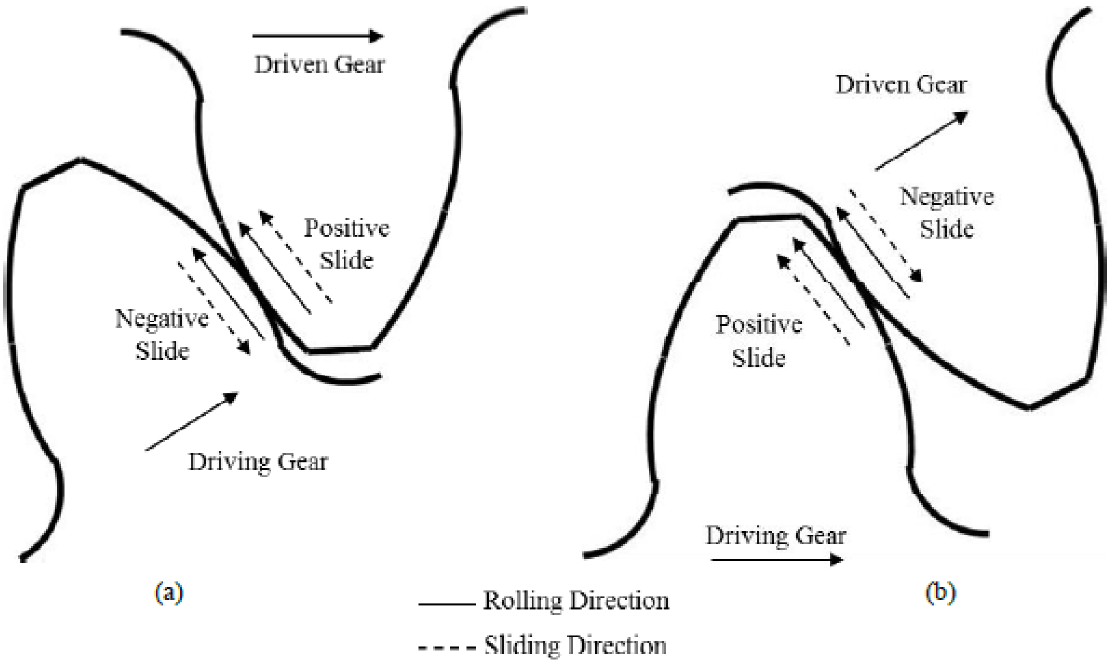


Figure 4 Schematic drawing of gear tooth contact, (a) start of contact, (b) end of contact [17]

The primary mechanism related to the negative sliding experienced by a gear tooth below its pitch line in the dedendum. Higher stresses are encountered because the surface material is being rolled in one direction and pushed in another. The combination of pure rolling and traction forces alters the subsurface stress profile by increasing the maximum shear stress and moving the location where it occurs closer to the contact surface [16]. For this reason, contact fatigue damage is usually located in the dedendum of the gear tooth. Another reason why sliding decreases contact fatigue resistance was proposed by Kim et al.[17]. Kim proposed that with sliding the number of stress cycles experienced locally by individual surface asperities increases for each contact cycle. This occurs because the asperities slide past each other and interact with more surface defects than they would if pure rolling was occurring and each asperity only interacted with one point on the other contacting body.[18]

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.[5]. 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.

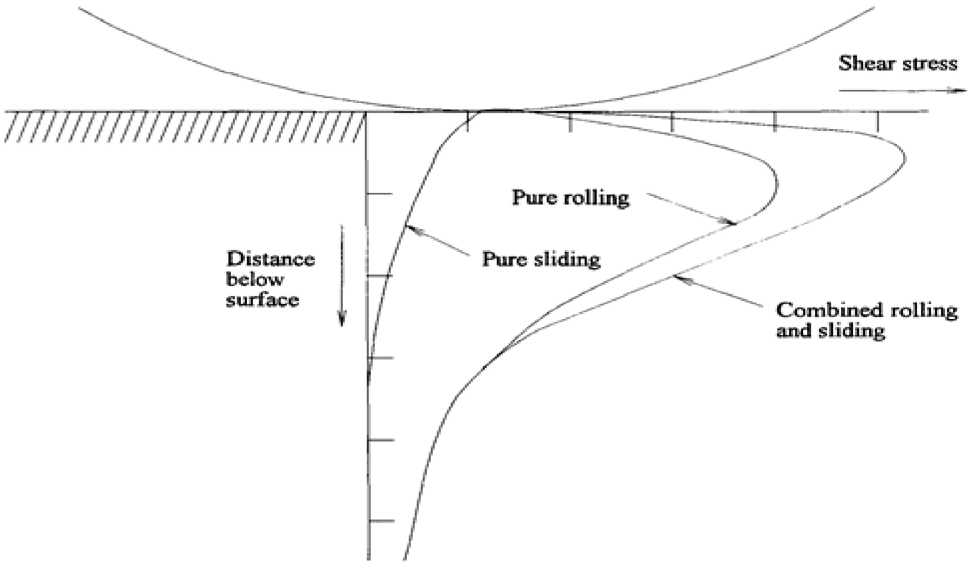


Figure 5 stress distribution at and near two contacting surfaces under sliding –Rolling conditions[5]

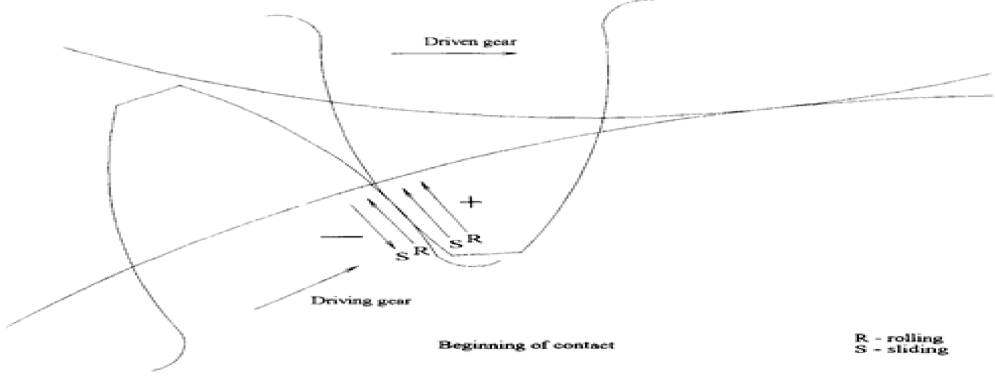


Figure 6 Combination of sliding and rolling contact of gear teeth at the beginning of contact [17]

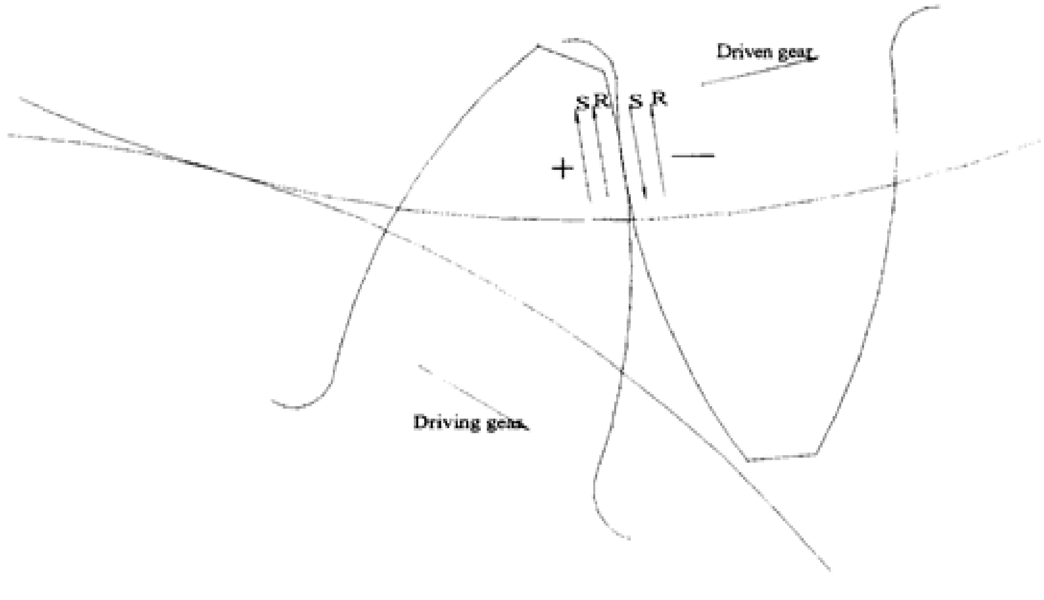


Figure 7 Combination of sliding and rolling in gear teeth at the end of contact[5]

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. 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.

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.

There are basically two gear teeth failure modes: bending fatigue at the teeth root and contact (or surface) fatigue at the teeth flank. The contact fatigue is caused by the stresses developed at the region of contact between the teeth flanks, which, after several cycles, will lead to crack initiation. The contact conditions are responsible for the nucleation of these cracks on the surface or subsurface of the teeth flanks. The cracks propagation may result in failure by pitting and/or spalling [Cheng, 1983].

The contact stresses on non- conformal surfaces i.e nominally (under zero load), such as in gears, can be estimated by analytical equations based on the elasticity theory developed by Hertz in 1881 [Stachowiak and Batchelor, 2005]. In this case the contact between two teeth is usually simulated using the contact between two equivalent cylinders/or rollers with the radii identical to the flanks radii curvatures in the contact point. It is very important to find which parameters of the tribological system affect the hertizian stress levels along the contact action line. Among the main influence factors it can be mentioned: the geometric profile of the tooth flank (modulus, number of teeth, pressure angle), the gear materials, the lubricant properties, the load transmitted and the kinematics of the gear teeth movement.

The surface finishing has a strong influence on the gear life, and the roughness behaves as a stress concentration factor for crack initiation; therefore, this is a relevant issue when analysing the gear flanks wear. Several studies have been conducted to verify the roughness effect on the resistance to contact fatigue: according to Zafosik et al. [2007], at the rolling and sliding contact, the resistance to fatigue depends on different factors, such as stress and elastoplastic strain, material properties, and physicochemical properties of lubricant, surface roughness, residual stress and contact kinematics. The surface cracks could be initiated near the deformed surface zone, in the region of maximum cyclic shear stress caused by rolling-sliding contact or alternatively, near defects such as notches or scratches.

As shown in fig., three steps are identified in the gear movement kinematics. At the beginning, the contact occurs through a combination of rolling and sliding (friction) between the teeth. In the pitch diameter region there is pure rolling, and after this point, rolling and sliding contact will occur again.

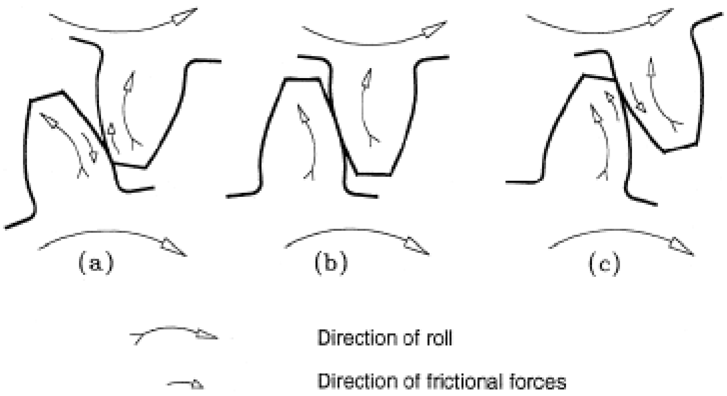


Figure 8 Mechanics of gear tooth contact: (a) at point of first contact; (b) at pitch point and (c) at last point of contact (Adapted from Walton and Goodwin, 1988).

The friction provokes changes in the stress field generated by the contact between bodies, thus exerting great influence on the contact fatigue failure. It is known from studies conducted by Honh (2004) that friction is also greatly influenced by the additive types present in the lubricants. Another important factor mentioned by Honh is related to the gears coating. Therefore it can be stated that the proposed models may show some changes when working with gears coated with fortified lubricants.

Lubrication is aimed at introducing a low shear strength film, which ends up weakening the resistance of these joints, and thus reducing friction. In some cases, the lubricant may not fully prevent contact between the asperities, although it may reduce the severity of contact conditions. In other situations, the lubricant separates the surfaces completely, and joints with asperities are not formed. Thus to a greater or lesser extent, the use of lubricant will always reduce the wear rate, and this will be a direct function of this type of lubrication.

There are basically three different lubrication regimes: hydrodynamic (HD), elasto-hydrodynamic (EHD) and boundary lubrication.

The contact between the gear teeth surfaces is “non-conformal”, i.e., it nominally (under zero load) involves a line or point of contact, generating small-area concentrated contacts. Under these conditions, elasto-hydrodynamic (EHD) is the predominant lubrication regime. Whenever the oil film breaks, the lubrication regime turns into boundary lubrication, where almost the entire load is supported by the asperities [Grupin, 1949].

The contact stress of the rollers model should represent the contact stress between two gear teeth.

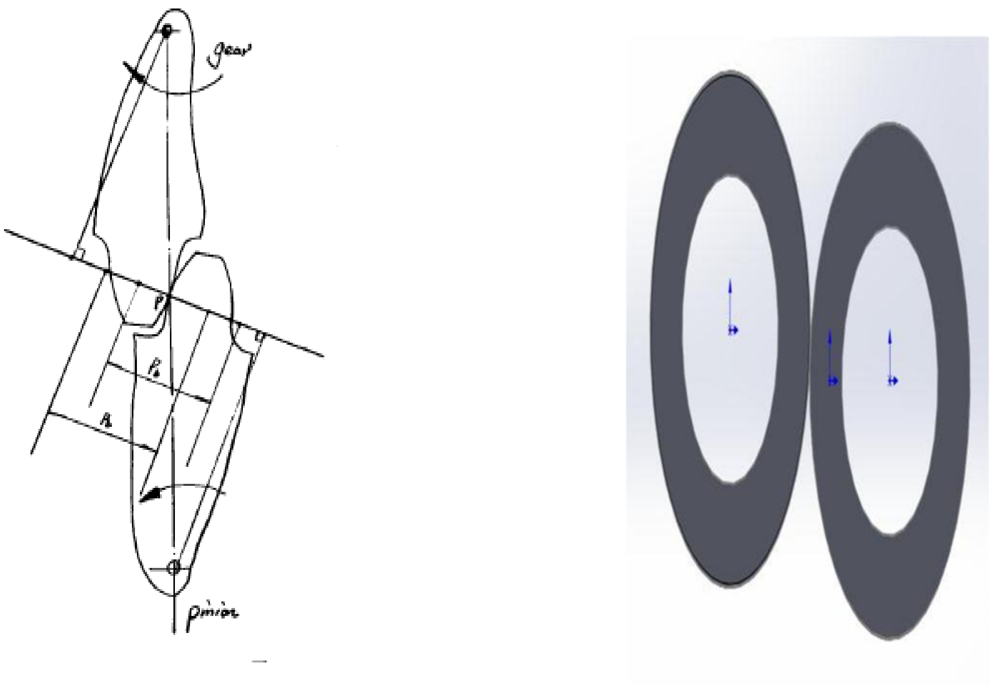


Figure 9 gear teeth contact and rollers contact

Wu and Cheng [18] analyse the sliding in spur gears, including the consideration of gear dynamics and rough-elastohydrodynamic lubrication (EHL). Equations for equivalent wear rate and tooth wear profile along the line of action are derived by utilizing a sliding wear model developed before by Wu and Cheng. The sliding wear rate is defined as the ratio of wear volume to the distance slid. The equivalent wear rate is the dynamic change in wear in a constant interval (or duration) at any contacting position along the line of action, and consequently, at the corresponding contacting point along the tooth profile. Therefore, the distribution of wear rate and wear profiles indicate that although the sliding wear rate has higher values around the pitch point because of the small sliding velocity, it is at the commencement of an engagement that most wear occurs because of the higher sliding and higher loading [7].

The effect of gear speed on wear is twofold: The increase in speed, on one hand increases the sliding velocity which tends to raise the contacting temperature, and therefore, promotes more wear. On the other hand, the increase in gear speed increases also the rolling velocity which increases the film thickness ; and this increase in film will reduce not only the percentage of asperities in contact but the contacting temperature as well, and thus decrease wear [18],[6].

In many gears, the maximum tooth loading is considerably higher than that corresponding to the transmitted power because of the dynamic effects of the unavoidable imperfections in the gears [4]. In general, the faster the gears are running, the more shock due to tooth errors and the more dynamic effects due to imbalance and torque variations in the driving and driven apparatus [12].

A considerable amount of research has been carried out to determine the amount of dynamic gear-tooth loads. A research committee of the American Society of Mechanical Engineers, headed by Earle Buckingham, published the first authoritative work on dynamic load in 1931. A simplified equation that makes possible a quick but approximate calculation of dynamic load has been developed from this work. Buckingham has given equations that permit dynamic load to be calculated using actual shaft stiffness and inertias. This work covers dynamic-load calculations on several kinds of gear teeth [12].

Most mechanical devices involve sliding of a moving member over a stationary surface and voitik et al.‟s [6] collective lubrication experience is based primarily on this concept. However, there are at least two mechanical functions in machinery that differ slightly from the definition above. These two are gears and cams. A successful engine or transmission can suffer failure if the special lubrication requirements of these two mechanisms are overlooked [6].

Compared with pure rolling condition, sliding as well as rolling provides addition EHD effects. The presence of sliding may increase the stress concentration at the trailing edge. The leading edge, according to Chiu [23], may benefit from a reduction in local pressure under sliding conditions. The influence of sliding on sub-surface stresses and the subsequent contact failures are mainly introduced through contact pressure alterations rather than surface friction.

The important role of liquid in creating the crack tip stress histories during cyclic contact loading has been discussed by Way [24]. It is widely believed that liquid can influence fatigue crack growth by reducing friction between crack faces, by pressure on the crack faces as fluid flows into the crack, and by the "fluid entrapment effect" exerting hydrostatic pressure at the crack tip [2].

Ali Raad Hassan [4] did a research study in which Contact stress analysis between two spur gear teeth was considered in different contact positions, representing a pair of mating gears during rotation. A programme has been developed to plot a pair of teeth in contact. Each case was represented a sequence position of contact between these two teeth. The programme gives graphic results for the profiles of these teeth in each position and location of contact during rotation. Finite element models were made for these cases and stress analysis was done. The results were presented and finite element analysis results were compared with theoretical calculations, wherever available.

Bharat Gupta, et al. [7] presented a paper to suggest that, through study of contact stress developed between the different matting gears are mostly important for the gear design. They used Hertz‟s equations which are current analytical methods of calculating gear contact stresses, originally derived for

Contact between two cylinders. So for contact stress they developed and determined appropriate models of contact elements, and calculated contact stresses using ANSYS and compared the results with Hertzian theory. Conclusions suggest that with increasing in module, contact stresses decreases for a pair of spur gears.

Sushil Kumar Tiwari, Upendra Kumar Joshi [8] this paper presented analysis of Bending stress and Contact stress of Involute spur gear teeth in meshing. Bending stress and contact stress (Hertz stress) calculation is the basic of stress analysis. It is difficult to get correct answer on gear tooth stress by implying fundamental Hertz equation for contact stress. They primarily prefer Theoretical and Numerical methods because Experimental testing can be expensive. In this study we use a 3D model of gear and finite element analysis to conduct RBS (Root Bending Stress) and SCS (Surface Contact Stress) calculation for mating involute spur gears. A pair of involute spur gear without tooth modification and transmission error is define in a CAD system (CATIA V5 and AUTODESK INVENTOR etc.) and FEA is done by using finite element software ANSYS. Obtained FEA results is comparable with theoretical and AGMA standard. It is found that Lewis formula and Hertz equation is used for quick stress calculation for gear, whereas the AGMA standards and FEM is used for detailed gear stress calculation for a pair of involute spur gear.

Vivek Karaveer, et.al [9] this paper presents the stress analysis of mating teeth of spur gear to find maximum contact stress in the gear teeth. The results obtained from Finite Element Analysis (FEA) are compared with theoretical Hertzian equation values. For the analysis, steel and Grey cast iron are used as the materials of spur gear. The spur gears are sketched, modeled and assembled in ANSYS Design Modeler. As Finite Element Method (FEM) is the easy and accurate technique for stress analysis, FEA is done in finite element software ANSYS 14.5. Also deformation for steel and Grey cast iron is obtained as efficiency of the gear depends on its deformation. The results shown that the difference between maximum contact stresses obtained from Hertz equation and Finite Element Analysis is very less and it is acceptable. The deformation patterns of steel and Grey cast iron gears depicted that the difference in their deformation is negligible.

Ramalingam Gurumani, et.al [10] This paper provides a novel method to model lead crowned spur gears. The teeth of circular and involute crowned external spur gears are modelled for the same crowning magnitude.

Based on the theory of gearing, mathematical model of tooth generation and meshing are presented. Effect of major performance characteristics of uncrowned spur gear teeth are studied at the pitch point and compared with longitudinally modified spur gear teeth.

The results of three dimensional FEM analyses from ANSYS are presented. Contact ellipse patterns and other contact parameters are also studied to investigate the crowning effects.

The relative motion between a pair of involute spur gear teeth is one of combined rolling and sliding except at the pitch point, where the sliding velocity is zero and changes from one direction to the other. It follows that the tangential velocity of points on the flank (dedendum) of a tooth relative to the point of contact is always less than that of the mating point on the tip (addendum) of the adjacent tooth.

During research into the wear of toothed gears Barwell [14] observed that wear debris showed a strong tendency to be removed from the slower surface and to become attached to the faster one. This observation was picked up by Dawson [15], who worked in the research laboratory of a leading gear manufacturer. He conducted experiments in a small disc machine in which spring loaded discs were geared together with a ratio of peripheral speeds of 5:7. Small pieces of plasticine were stuck to the surface of one or other of the discs and passed through the nip between the discs. In the vast majority of tests it emerged sticking to the faster surface, irrespective of the disc to which the plasticine was originally attached.

According to Robert B. and et al.,[19] study pitting of gears teeth flanks is one of the most pronounced types of damage that occurs in gears. Pitting is phenomenon in which small particles are removed from the surface of the tooth because of the high contact forces that are present between mating teeth. It is actually the fatigue failure of the tooth surface. Complex stresses with in the contact zone cause surface and subsurface fatigue failures. The pits seen on the teeth grow in size and depth, ultimately resulting in tooth fracture and spalling. Pitting occurs after a large number of repeated loading on the contact surfaces of the teeth. It is found to occur most frequently at the pitch circle where relative sliding of the teeth is zero and the hydrodynamic lubricant film tends to break down.

P.J.L Fernandes, et al., [20] referred that surface contact fatigue is the most common cause of gear failure. It results in damage to contacting surfaces which can significantly reduce the load-carrying of components, and may ultimately lead to complete failure of a gear. The types of contact fatigue damage are discussed, and a number of actual examples are presented to illustrate this failure mode in practice in his study.

Robert Basan. Et al., [21] focuses on pitting of gears teeth flanks is one of the most pronounced type of damage that occurs in gears. Although it can manifest itself in a number of different forms, it is direct consequence of highly concentrated cyclic stresses and strains which develop in tooth flank material during the mesh in conditions of simultaneous rolling and sliding contact. Due to such rather complex state of stress and strain that changes during the mesh, identification of critical locations in regard to fatigue damage and crack initiation is difficult task and requires detailed numerical analysis. For the determination of mentioned stresses and strains and their evolution during the mesh, a numerical model gear pair in mesh was developed and complete analysis procedure defined.

Solomon Tekeste, [22] come up to the two main factors that affect the service life of gears: surface durability and tooth bending strength. Pitting and scoring is a phenomenon in which small particles are removed from the surface of the tooth due to the high contact stresses that are present between mating teeth. After the investigation of shot pinning to increase the tooth bending strength in gears, surface durability, in the form of macro and micro pitting, was considered as the dominant restriction on gear life and performance. The main objective of his thesis is to develop a method for the investigation of surface-fatigue in involute spur gears using finite element analysis. In achieving this two dimensional and three dimensional equivalent contacting cylinders to model involute spur gear in contact and three dimensional model of involute spur gear is used.

Ruben D. Chacon, et al., [23] in this paper a study of the stresses in the contact zone among a couple of spur gears is realized using the finite elements method. The analysis is done by using a plane model involving the contact between two teeth. The geometry is defined according to the standards of the American Gear Manufacturers Association (AGMA). The results obtained are compared with the value given by two others approaches. The first is the theory of Hertz when it is applied to two curved segments in contact. The second approach is the AGMA procedure for calculating contact stresses in spur gear. The results obtained are very similar either using FEM and Hertz‟s theory. The contact pressure obtained by FEM is lower than the one obtained by means of

Hertz"s theory, this fact reflects the influence of the geometry profile which allows the occurrence of contact zones with greater area, and therefore lower pressures.

Shreyash Patel, [24] developing an analytical approach and modelling procedure to evaluate stress distribution under velocity and moment would provide a useful tool to improve spur gear design with high efficiency and low cost. The purpose of this work is to analyze and validate the stress distribution in spur and helical gears using contemporary commercial FEM program ANSYS coupled with the Pro/E solid modeler. Gear profiles are created in Pro Engineer using the relation and equation modeling procedure so as to make it a general model, dependent on certain key features like number of teeth, diametral pitch, and pressure and helix angle. These models were then analyzed for 3D and 2D bending stresses. Calculated results obtained when compared to standard AGMA stresses show good agreement.

Daniel Tilahun Redda, et al. [12] Presented a paper of Surface Durability of Developed Cr-Mo-Si Steel Under Rolling-Sliding Contact. In this paper they develop case-carburized gears with high loading capacity by increasing the surface durability of a gear by adding Si in the normal steel alloy of Cr-Mo alloy steel. The experiment is done using two rollers having different speed of rotation and profiles.
Figure 2 shows the dimensions of the test rollers. The outer and inner diameters of the test rollers were 70 and 45 mm respectively. The axial length of all test rollers was 28 mm and the outer surface of the high speed roller was modified to provide an effective contact width of 8.5 mm.

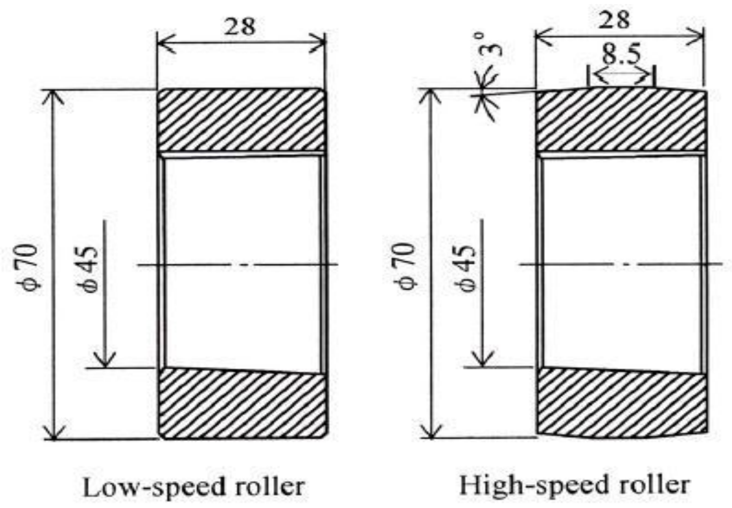


Figure 10 2D view of High speed and Low speed rollers

From the experimental result, due to the addition of Si, the surface failure of the roller is almost yet happened.

The following figure shows the experimental result of the rollers contact (high speed and low speed rollers contact) of Cr-Mo-Si alloy steel and Cr-Mo alloy steel under the same Stress, Force and Temperature. As we have seen from the figure, even if the number of cycle of Cr-Mo alloy steel is short, the pitting is happened. Whereas no pitting in Cr-Mo-Si alloy steel.

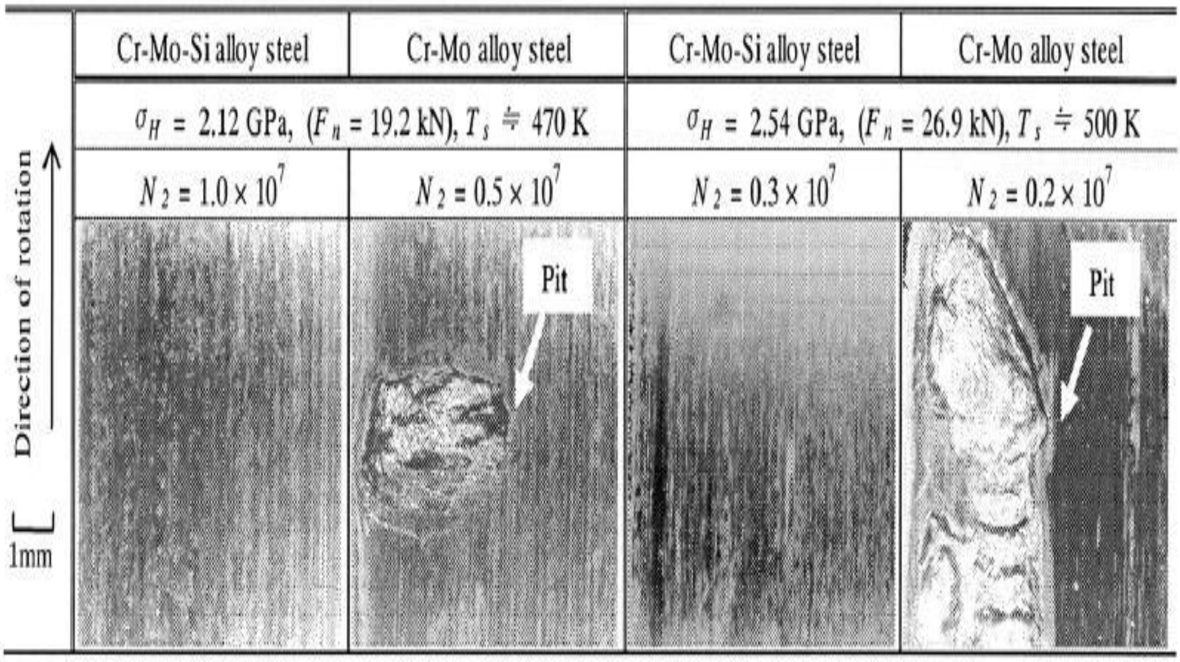


Figure 11 Stereoscopic microscope photographs of surface of roller after running

Pitting is a surface fatigue failure resulting from repetitions of high contact stress. The surface fatigue mechanism is not definitively understood. The contact affected zone, in the absence of surface shearing tractions, entertains compressive principal stresses.

Most of the researchers have done on the surface fatigue failure of the gear teeth contact using experimental approach and it is expensive. One research has done using two rollers contact using experimental approach to see the durability of the material. In this paper the surface fatigue failure of the teeth analysed using two rollers contact that happened at the edge of the roller due to maximum stress distribution through FEM.

1.3 STATEMENT OF THE PROBLEM

The current problem which is not investigated in the literature is that the 3D geometry of the rollers is not modelled and the contact analysis is not done using FEM. The expected outcome is the effect of stress distribution to be at the end of the contact region on the rollers contact.

1.3 OBJECTIVE

1.3.1 General objective:-

The general objective of this study is to investigate the effect of stress distribution on surface failure of involute spur gear teeth by simulating using two rollers contact.

1.3.2 Specific objective

The specific objectives of the research will be to:

- Modeling, Assembling and importing the rollers using solid work
- Study stress distribution during rollers contact
- simulating the contact stress of a gear teeth using rollers contact
- investigating the maximum von mises stress which causes failure and know the region at which failure will be happened
- validating the surface failure using ANSYS with the experimental result

1.4 Methodology

The methods employed to achieve the above objectives are:

- Overview of experimental result which is done on contact stress of a gear using rollers contact
- modelling of the 3D geometry using solid work
- simulating the effect of contact stress distribution using ANSYS Workbench

1.5 Organization of the Thesis

This thesis is comprised of a total of five chapters. Chapter one presents a general introduction, literature reviews, statement of the problem, the objectives to be achieved and the methodology of the thesis. Finally the layout of the thesis is described. Chapter two describes the modeling, importing and analysis of the rollers contact have been discussed. Chapter three begins with results of the von misses stress distribution and discussions have been takes place. Chapter four gives the conclusions and recommendations of this thesis and suggests future work.

CHAPTER TWO

2. MATERIALS, METHODS AND CONDITIONS

2.1 Material

In this step it's necessary to define the material properties used in this study. The necessary material properties to be defined are young's modulus, poison's ratio and density of the material. This can be done by selecting the Engineering Data from the analysis tab of the ANSYS Workbench and inserting the corresponding values for the material.

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 material that were used for the analysis was listed in the table has been added to the engineering data and used for the stress evaluation in ANSYS Workbench. [30]

In this thesis work the material used is SCM 420 alloy steel; SCM 420 is an alloy containing Chromium and Molybdenum. Since it is commonly used in vehicle technology, this improved alloy steel used for manufacture of gears which has improved strength and toughness while exhibiting enhanced machinability. This material suitably used for producing gears for automotive power transmissions, it can also be used for gears other than the transmission gears. 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. [27]

SCM 420 steel's is used to make all kinds of fasteners, high pressure pipe and more advanced carburized parts, such as gear, shaft etc., which work in the noncorrosive medium and temperature below 250 ^{o}c, working with nitrogen hydrogen mixture in the medium.

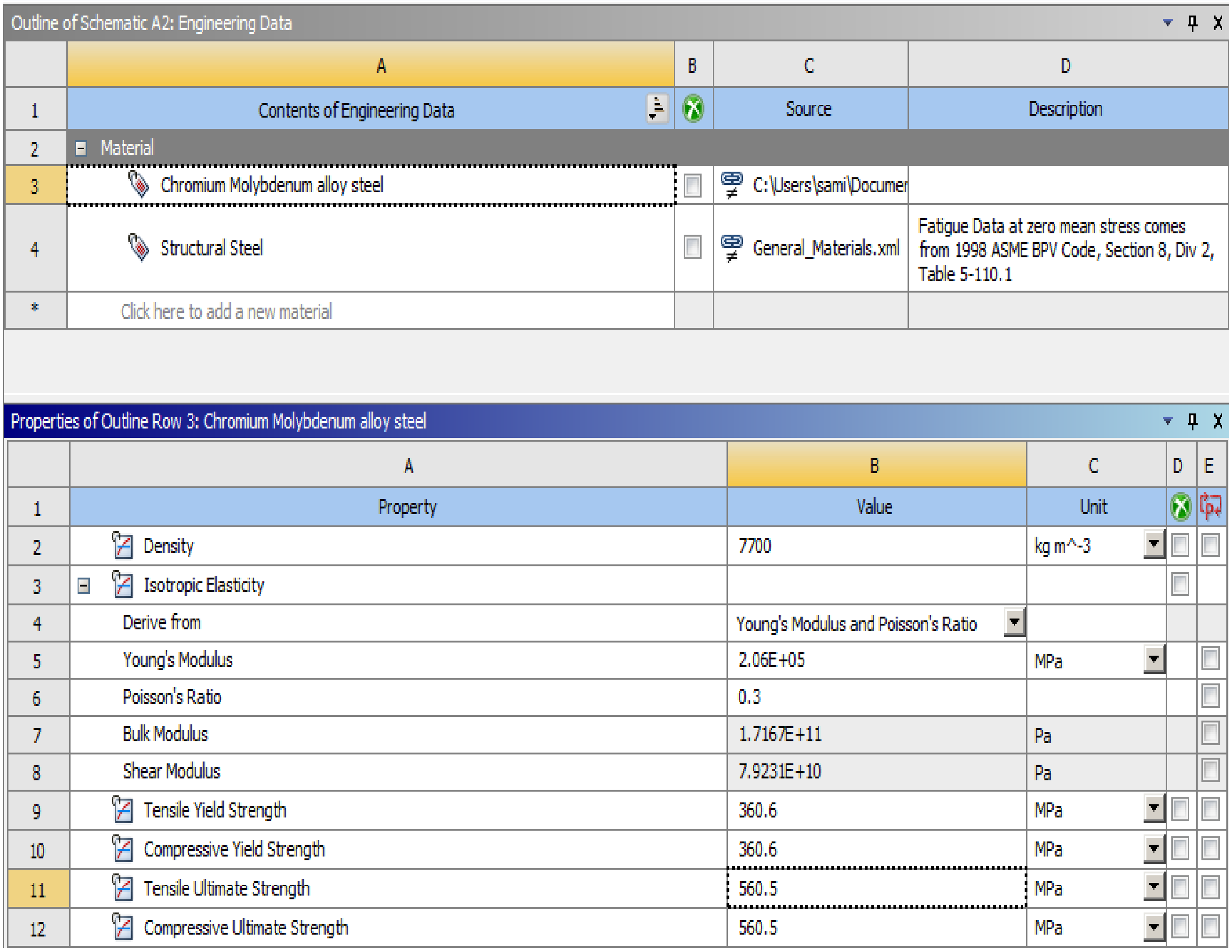
SCM 420 steel's quench-hardening ability is higher, no temper brittleness. The weldability of SCM 420 Steel is quite good. The tendency of cold crack formation is very small, but machinability and cold straining plasticity is good.

Table 1 Chemical composition of SCM 420 steel alloy

Element	Minimum	Maximum
Carbon, C	0.18	0.23
Manganese, Mn	0.60	0.90
Silicon, Si	0.15	0.35
Nickel, Ni	0.00	0.25
Molybdenum, Mo	0.15	0.25
Chromium, Cr	0.90	1.20
Vanadium, V	-	-
Copper, Cu	0.00	0.30
Sulphur, S	0.00	0.03
Phosphorous, P	0.00	0.03

Table 2 Mechanical and Physical properties of Chromium Molybdenum alloy steel (SCM420) under study

S.No	Mechanical and physical properties of SCM420 Alloy structural steel	Values	Unit
1	Modulus of Elasticity	206	Gpa
2	Tensile yield strength	360.6	Mpa
3	Compressive yield strength	360.6	Mpa
4	Ultimate tensile strength	560.5	Mpa
5	Ultimate compressive strength	560.5	Mpa
6	Elongation	28.2	%
7	Poisons raio	0.3	
7	Thermal conductivity	25	w/m.k
8	Specific heat	460	J/Kg.K
9	Melting temperature	1480	^{o}c
10	Density	7700	Kg/m^3



Outline of Schematic A2: Engineering Data

	A	B	C	D
1	Contents of Engineering Data		Source	Description
2	Material			
3	Chromium Molybdenum alloy steel		C:\Users\sami\Documer	
4	Structural Steel		General_Materials.xml	Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1
*	Click here to add a new material			

Properties of Outline Row 3: Chromium Molybdenum alloy steel

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	7700	kg m^-3		
3	Isotropic Elasticity				
4	Derive from	Young's Modulus and Poisson's Ratio			
5	Young's Modulus	2.06E+05	MPa		
6	Poisson's Ratio	0.3			
7	Bulk Modulus	1.7167E+11	Pa		
8	Shear Modulus	7.9231E+10	Pa		
9	Tensile Yield Strength	360.6	MPa		
10	Compressive Yield Strength	360.6	MPa		
11	Tensile Ultimate Strength	560.5	MPa		
12	Compressive Ultimate Strength	560.5	MPa		

Figure 12 Chromium Molybdenum alloy steel assigned as a material in engineering data

2.2 Dimensions

2.2.1 Dimensions of contact rollers

The rollers are prepared to show the contact simulation that happened during gears contact. The low speed roller (2630 rpm) has an axial length of 28 mm and has 45 mm and 70 mm of internal and external diameters respectively. Whereas the high speed roller (3190 rpm) has an axial length of 28 mm and has 45 mm and 70 mm of internal and external diameters respectively. In addition to this the high speed roller is tapered at the edge. So the true contact length of the high speed roller with the low speed roller will be 8.5 mm.

2.2.2 Stress Distribution in Rollers Contact

When the two rollers rotating, the stress due to the contact fatigue becomes developed and it searches a free space, so that the stress concentration becomes increase at the region of the curved area of the rollers and there is a thermal effect at this region. Any components with relative motion are subjected to surface contact due to loading and additional thermal effects on the contact bodies. Those applied load and relative motion results high stress and deformation on the contact surfaces which result failure.[21] The surface of the roller is going to failure. [18] The result will be seen using ANSYS Workbench 15.

For this particular study, consider a roller contact to simulate gear teeth contact and therefore the following parameters were considered.

Table 3 Geometrical parameters for the two rollers

S.N	Type of rollers	ID	OD	Axial length	True contact length	rpm
1	HSR	45 mm	70 mm	28 mm (9.75 mm tapered both sides of the edge)	8.5 mm	3190
2	LSR	45 mm	70 mm	28 mm (no tapering)	8.5 mm	2630

2.3 Conditions

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

During the rollers contact the following simplifying conditions are considered

- the material in contact are homogenous
- the contact area (8.5 mm) 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
- it considers only static loading
- the friction forces are considered due to difference in relative velocity between contacting rollers.

2.4 FINITE ELEMENT METHOD (FEM) ANALYSIS

2.4.1 Modelling rollers geometry using solid work

We can use two different methods to generate our model: solid modelling and direct generation. With solid modelling, we describe the geometric boundaries of our model, establish controls over the size and desired shape of our elements, and then instruct the ANSYS program to generate all the nodes and elements automatically. By contrast, with the direct generation method, we determine the location of every node and the size, shape and connectivity of every element prior to defining these entities in our ANSYS model.

Although some automatic data generation is possible, the direct generation method is essentially a hands-on, ”manual” methods that requires we to keep track of all our node numbers as we develop our finite element mesh. This detailed bookkeeping can become tedious for large models, contributing to the potential for modelling errors.

Solid modelling is usually more powerful and versatile than direct generation, and is commonly the preferred method for generating our model.

In spite of the many advantages of solid modelling, we might occasionally encounter circumstances where direct generation will be more useful. We can easily switch back and forth between direct generation and solid modelling, using the different techniques as appropriate to define different parts of our model. Solid modelling is generally more appropriate for large or complex models, especially 3D models of solid volumes

Solid modelling can be modelled in different Computer Aided Design (CAD) systems, like Solid work, Pro/Engineer, UG, Parasolid, Auto cad, CATIA, Design Modeller etc.

For this thesis particular study the 3D geometry of the rollers have been modelled using solid works.

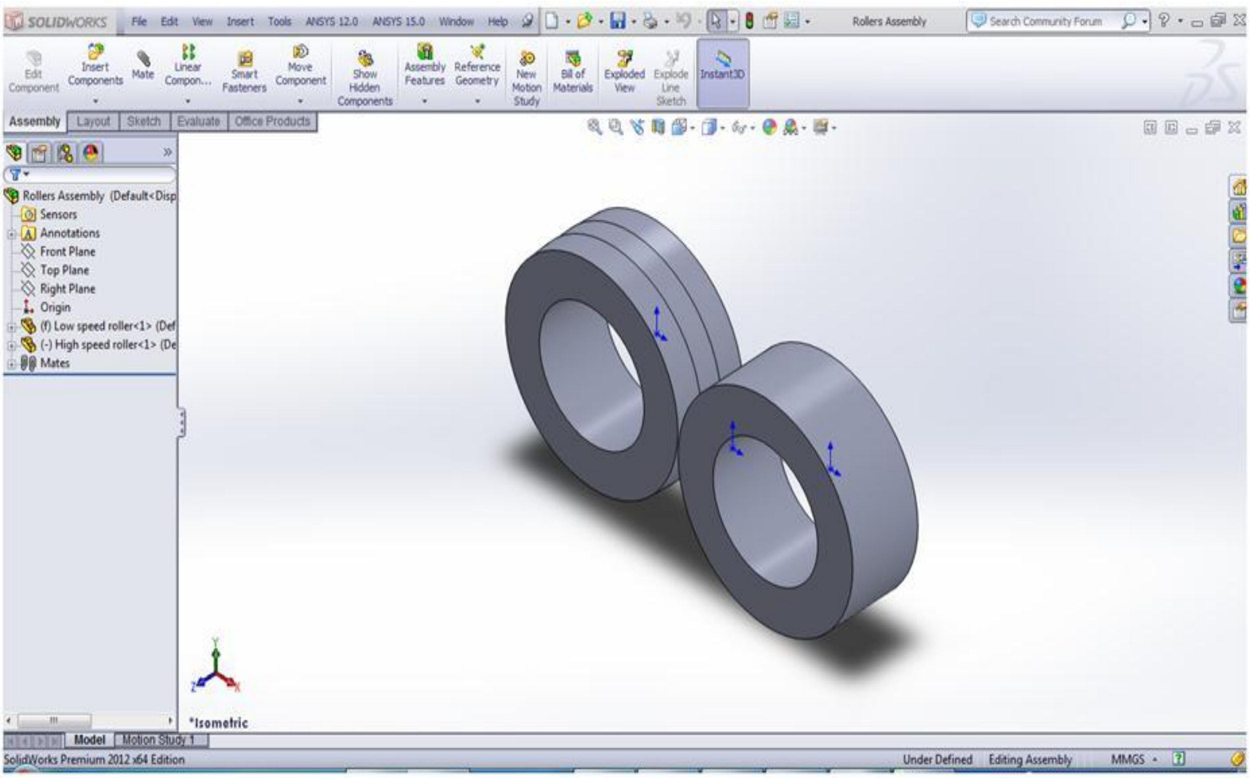


Figure 13 Assembly of two rollers contact using solid work

2.4.2 Importing solid models created in CAD systems to ANSYS Mechanical Workbench

Design Modeler is the workbench geometry application to design the 3D model and is basically solid model editing platform in ANSYS. Also it supports all the functions capabilities listed for commercial CAD systems.

As an alternative to creating our solid models within ANSYS, we can create them in our favourite CAD system and then import them into ANSYS Mechanical Workbench for analysis, by saving them in IGES (Initial Graphic Exchange System) file format or in a file format supported by an ANSYS connection product. Creating a model using a CAD package has the following advantages:

- we avoid a duplication of effort by using existing CAD models to generate solid models for analysis.
- we use more familiar tools to create models.

However, models imported from CAD systems may require extensive repair if they are not of suitable quality for meshing

When an assembly is imported contact surfaces are automatically detected and created. The proximity of surfaces is used to detect contact. Contact „surfaces" in 2D geometry are represented by edges.

Generally, when importing assemblies of solid parts, contact regions are automatically created between the solid bodies. Parametric CAD dimensions also can be imported into Mechanical Workbench.

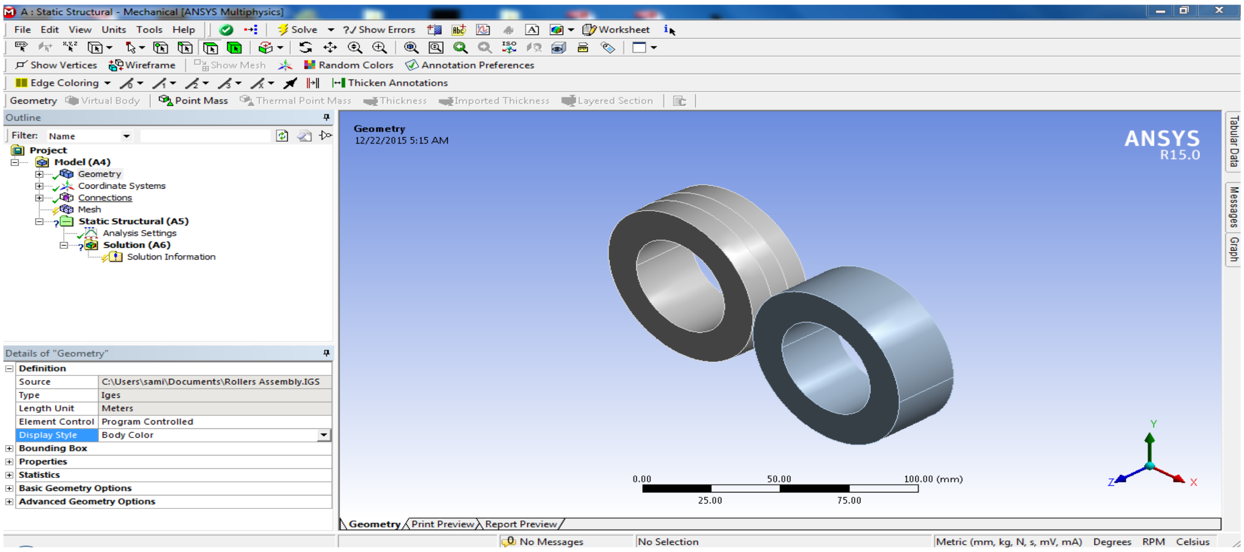


Figure 14 Geometry import to ANSYS Workbench from solid work

2.4.3 Static structural Contact stress analysis using ANSYS WB

ANSYS is an integrated program with all operations performed under one GUI. Creating model, running it, and post processing the results are all done without leaving the ANSYS environment.

ANSYS Workbench is a brand new GUI with an emphasis on CAD connectivity, ease of use, and easy management of assembly contact. This GUI is covered in a separate guideline.

2.4.3.1 ANSYS Workbench Overview

ANSYS Workbench is a project management tool which can be considered as the top level interface linking all our software tools.

Workbench handles the passing of data between ANSYS Geometry /Mesh/Solver/Post processing tools.

This greatly helps project management. We do not need worry about the individual files on disk (geometry, mesh etc.). Graphically, we can see at a glance how a project has been built. Because Workbench can manage the individual applications and pass data between them, it is to automatically perform design studies (parametric analyses) for design optimization.

Workbench geometry properties control the import of numerous CAD items in addition to geometry: parameters, coordinate system, material properties etc.

2.4.3.2 Static structural type of analysis

Structural analysis is probably the most common application of the finite element method. The term structural (or structure) implies not only civil engineering structures such as bridges and buildings, but also naval, aeronautical, and mechanical structures such as ship hulls, aircraft bodies, and machine housings, as well as mechanical components such as pistons, machine parts and tools.

Static structural analysis used to determine displacements, stresses etc. Under static loading conditions, both linear and nonlinear static analysis. Nonlinearities can include plasticity, stress stiffening, large deflection, large strain, hyper-elasticity, contact surfaces and creep.

A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time-varying loads. A static analysis can, however, include steady inertia loads (such as gravity and rotational velocity), and time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind and seismic loads commonly defined in many building codes).

Static analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not include significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structures response are assumed to vary slowly with respect to time. The types of loading that can be applied in a static analysis include:

- externally applied forces and pressures
- Steady-state inertial forces (such as gravity and rotational velocity)
- Imposed (nonzero displacement)
- Temperatures (for thermal strain)
- Fluences (for nuclear swelling)

In the static structural, there is a question mark because we don't define the load and the boundary condition. After defining the necessary loads and boundary conditions, the question mark will be changed to check marked.

2.4.3.3 Contact Interpretation between contacting bodies

Contact overview

Contact problems are highly nonlinear and require significant computer resources to solve. It is important to understand the physics of the problem and take the time to setup our model to run as efficiency as possible.

ANSYS supports both rigid-to-flexible and flexible-to-flexible surface to surface contact elements. These contact elements use a "target surface" to form a contact pair. The target surface is modelled with TARGE170 for 3D and the contact surface is modelled with CONTA174 surface element.

A. Contact region

In simulation, the concept of contact and target surfaces is used for each contact region:

- One side of a contact region is referred to as a contact surface; the other side is referred to as a target surface.
- The contact surfaces are restricted from penetrating through the target surface

. When one side is designated the contact and the other side the target, this is called asymmetric contact.

. If both sides are made to be contact and target this is called symmetric contact.

Asymmetric contact is defined as having all contact elements on one surface and all target elements on the other surface. This is sometimes called" one-pass contact". This is usually the most efficient way to model surface to surface contact. However, under some circumstances asymmetric contact does not perform satisfactory. In such cases, we can designate each surface to be both a target and a contact surface.. we can then generate two sets of contact pairs between the contacting surfaces. This is known as symmetric contact (or "two-pass contact"). Obviously symmetric contact is less efficient than asymmetric contact.[11]

By default, ANSYS uses symmetric behaviour, this means that the contact surfaces are constrained from penetrating the target surfaces and the target surfaces are constrained from penetrating the contact surfaces. If we use asymmetric behaviour, only the contact surfaces are constrained from penetrating the target surfaces.

For asymmetric behaviour, the nodes of the contact surface cannot penetrate the target surface. This is an important rule to remember.

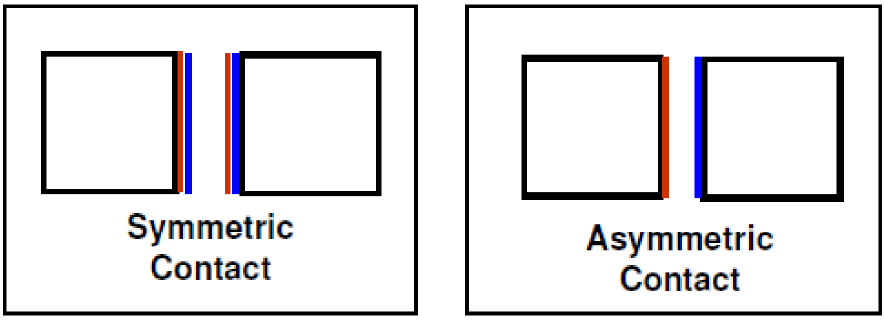


Figure 15 Symmetric and asymmetric type of contact

B. Contact and Target bodies

Contact elements are constrained against penetrating the target surface. However, target elements can penetrate through the contact surface**.** For rigid-to-flexible contact, the designation is obvious: the target surface is always the rigid surface and the contact surface is always the deformable surface.

General assumptions for contact and target designations

- If a convex surface comes into contact with a flat or concave surface, the flat or concave surface should be the target surface
- If one surface has a coarse mesh and the other and the other a fine mesh, the surface with a coarse mesh should be the target surface
- If one surface is stiffer than the other, the stiffer surface should be the target surface
- If one surface is higher order than the other, the low order should be the target surface
- If one surface is larger than the other, the larger surface should be the target surface.[17]

The contact and the target parts of the body will be shown in the details part as shown in the table.

Table 4 Contact and Target bodies

Details of "Frictional - High speed roller To Low speed roller"

Scope	
Scoping Method	Geometry Selection
Contact	2 Faces
Target	2 Faces
Contact Bodies	High speed roller
Target Bodies	Low speed roller
Definition	
Type	Frictional
Friction Coefficient	0.15
Scope Mode	Manual
Behavior	Asymmetric
Trim Contact	Program Controlled
Suppressed	No
Advanced	
Formulation	Augmented Lagrange
Detection Method	Nodal-Normal To Target
Penetration Tolerance	Program Controlled
Elastic Slip Tolerance	Program Controlled
Normal Stiffness	Program Controlled
Update Stiffness	Program Controlled
Stabilization Damping Factor	0.
Pinball Region	Program Controlled
Time Step Controls	None
Geometric Modification	
Interface Treatment	Adjust to Touch
Contact Geometry Correction	None

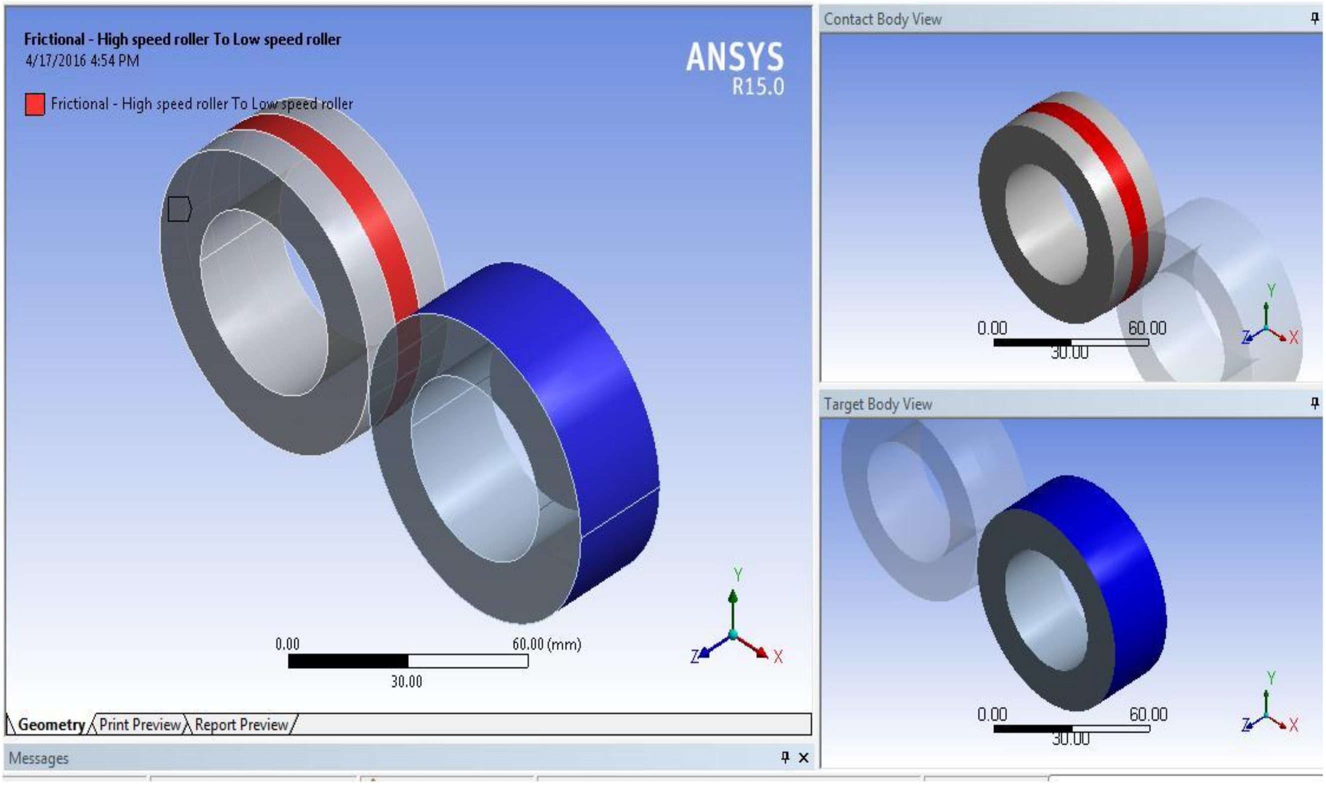


Figure 16 Contact and Target type of contact region

C. Contact bodies behaviour

The five common forms of contact behaviour available in WB:

Table 5 Contact of two contact bodies' behavior

Contact Type	iterations	Normal behaviour (separation)	Tangential behaviour (sliding)
Bonded	1	No gaps	No sliding
No Separation	1	No gaps	Sliding allowed
Frictionless	Multiple	Gaps allowed	Sliding allowed
Rough	Multiple	Gaps allowed	No sliding
Frictional	Multiple	Gaps allowed	Sliding allowed

- Bonded and No separation contact are linear and require only 1 iteration.
- Frictionless, Rough and Frictional contact are nonlinear and require multiple iterations. In contact between two contacting surfaces can transfer compressive and tangential friction forces but do not transfer tensile normal forces.[28]

In general, the tangential or sliding behaviour of two contacting bodies may be frictionless or involve friction. Frictionless behaviour allows the bodies to slide relative to one another without any resistance. When friction is included, shear forces can develop between the two bodies. Frictional contact may be used with small deflection or large deflection analyses.

D. Contact Formulation

Some of the primary aspects of contact formulations are compared below.

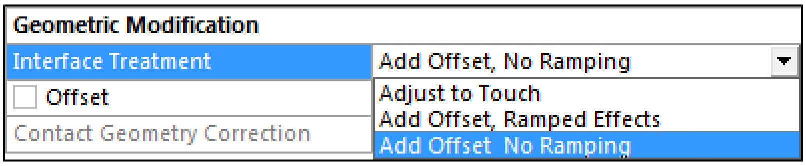
Table 6 Contact Formulation comparison

<table>
<tr><th>Pure penalty</th><th>Augmented Lagrange</th><th>Normal Lagrange</th><th>MPC</th></tr>
<tr><td>Good convergence behaviour(few iterations)</td><td>Additional iteration is needed if penetration is too large</td><td>Additional iteration is needed if chattering is present</td><td>Good convergence behaviour(few iterations)</td></tr>
<tr><td>Sensitive to selection of normal contact stiffness</td><td>Les sensitive to selection of normal contact stiffness</td><td colspan="2">No normal contact stiffness is required</td></tr>
<tr><td>Contact penetration is present and un controlled</td><td>Contact penetration is present but controlled to some degree</td><td>Usually- penetration is near zero</td><td>No penetration</td></tr>
<tr><td colspan="3">Useful for any types of behaviour</td><td>Only Bonded and No separation behaviour</td></tr>
<tr><td colspan="2">Iterative or Direct solvers can be used</td><td>Only Direct solvers can be used</td><td>Iterative or Direct solvers can be used</td></tr>
<tr><td colspan="2">Symmetric or asymmetric contact available</td><td colspan="2">Asymmetric contact only</td></tr>
<tr><td colspan="2">Contact detection at integration points</td><td colspan="2">Contact detection at nodes</td></tr>
</table>

Interface Treatment

The interface treatment property defines how the contact interface of a contact pair is treated. It becomes active when contact type is set to Frictionless, Rough or Frictional (nonlinear).

When active, the interface treatment option provides the following properties.



- Add offset, No ramping: this is the default setting. This option is the same as Add offset, Ramped effects but loading is not ramped.
- Adjust to Touch: any initial gaps are closed and any initial penetration is ignored creating an initial stress free state. Contact pairs are just touching as shown.

Contact pair before any interface treatment. Gaps exist.

Contact pair after Adjust to Touch Treatment. Gap is closed and pair is touching.

Contact pair before any interface treatment. Penetration exist

Contact pair after adjust to touch treatment. Gap is closed automatically. Pair is just touching

Figure 17 Interface treatment of contact pairs [19]

2.4.3.4 Meshing in Mechanical ANSYS Workbench

Meshing is the process in which the geometry is spatially discretized into elements and nodes. This mesh along with material properties is used to mathematically represent the stiffness and mass distribution of the structure.

Defining a proper mesh is critical to contact conditions. A well-defined mesh ensures accurate stress measurements at a contact region. Furthermore, a quality mesh is essential for nonlinear contact conditions in order to obtain an accurate solution. This is especially true for curved surfaces. Use local mesh controls, such as proximity controls and contact sizing controls to better ensure mesh quality.

The nodes and elements representing the geometry model make up the mesh because the model in CAD package is idealized discretize. A default mesh is automatically generated during a solution. It is generally recommended that additional controls be added to the default mesh before solving.

The contact conditions of the gear teeth/ or rollers are sensitive to the geometry of the contacting surfaces, which means that the element near the contact zone needs to be refined. It is not recommended to have a fine mesh everywhere in the model, in order to reduce the computational time requirements. There are two ways to build the fine mesh near the contact surfaces. One is a fine mesh of rectangular shapes were constructed only in the contact areas. The other one “smart size” in ANSYS, was chosen and the fine mesh near the contact area was automatically created.

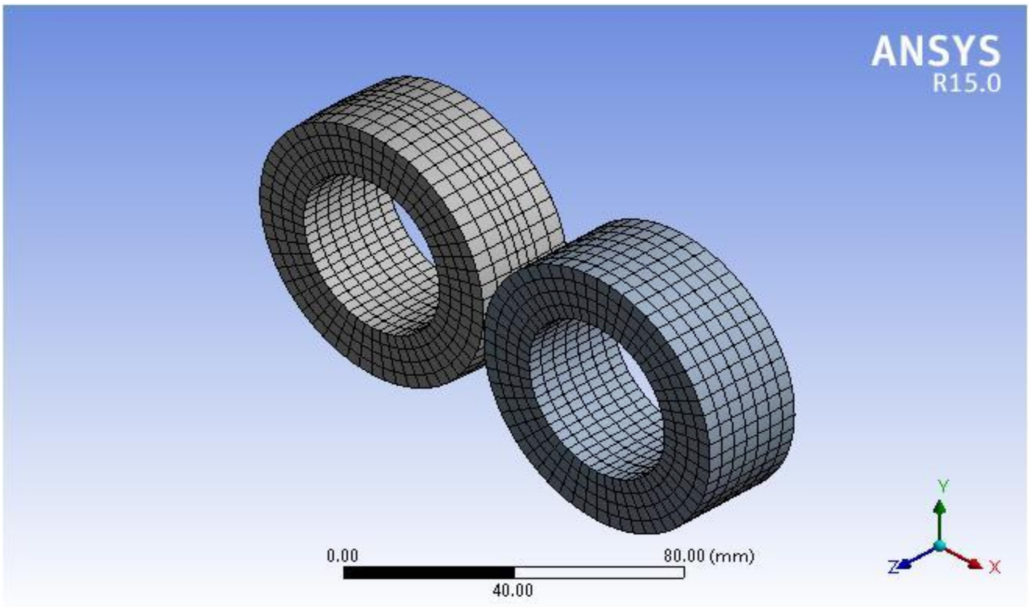


Figure 18 Meshing of two contact bodies

2.5.3.5 Applying boundary conditions

The force and rotational velocities of the rollers have given with their respected values.

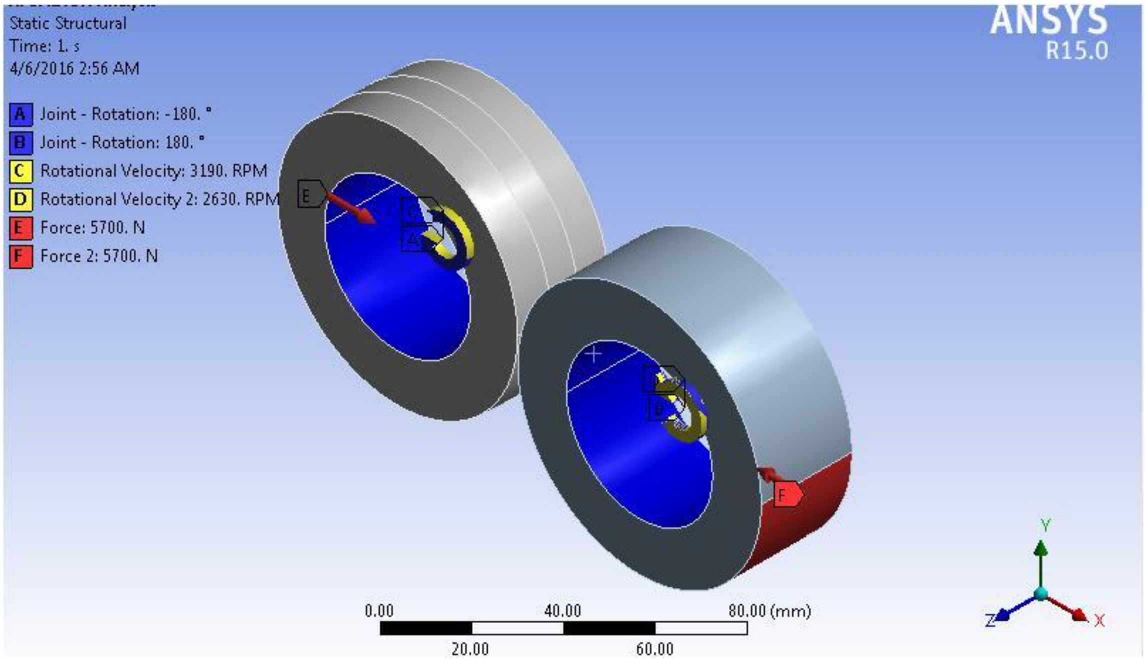


Figure 19 Normal Force and Rotational velocity of rollers

In defining different rotational velocity of the two contacting rollers is not the same way as defining of rotational velocity having the same rpm. This is the most difficult and challenging part during the analysis.

- Global coordinate system – the default coordinate system at 0,0,0
- Coordinate system- the defined coordinate system for high speed roller at -46.647, 7.4223, -0.0042803 mm
- Coordinate system 2- the defined coordinate system for low speed roller at 27.978, -0.07441, 0.01023 mm

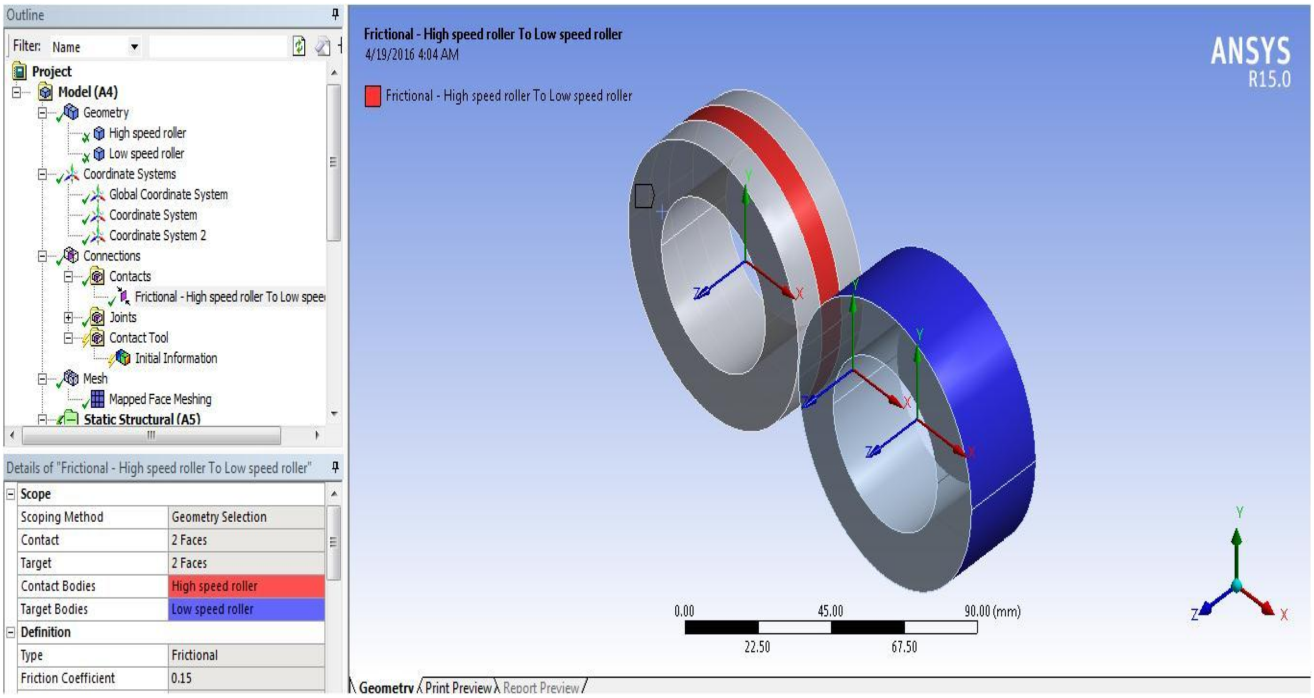


Figure 20 Defining of coordinates for different bodies

After setting of all the required boundary conditions for the rollers contact, we can start the solver to compute the required results.

CHAPTER THREE

3. RESULTS AND DISCUSSIONS

3.1 RESULTS

3.1.1 Stress Distribution Analysis

In the previous chapter, input parameters and steps that are used in the analysis of the contact between high speed roller and low speed roller discussed briefly. The analysis has done using Finite Element Method by using ANSYS Workbench release 15. Static structural analysis is carried out for determining the stress distribution and the maximum von misses stress on the rotating rollers to know the surface where surface failure /or pitting will be happened.

Using ANSYS Workbench 15 software, along with the boundary conditions, applied loads, respected rotational velocities is given then I have got the values of equivalent von misses stress.

Von misses stress is widely used by designers to check whether their design will withstand given load conditions and it arises from distortion energy failure theory. Then the maximum and minimum value is noted and the portion where it’s value is obtained, i.e some portion of the roller showing the maximum equivalent stress. In real condition that portion undergo a surface failure/ or pitting.

The result from the analysis in chapter three will be discussed now.

The Von-Mises stress at the constant normal force of 5.7KN

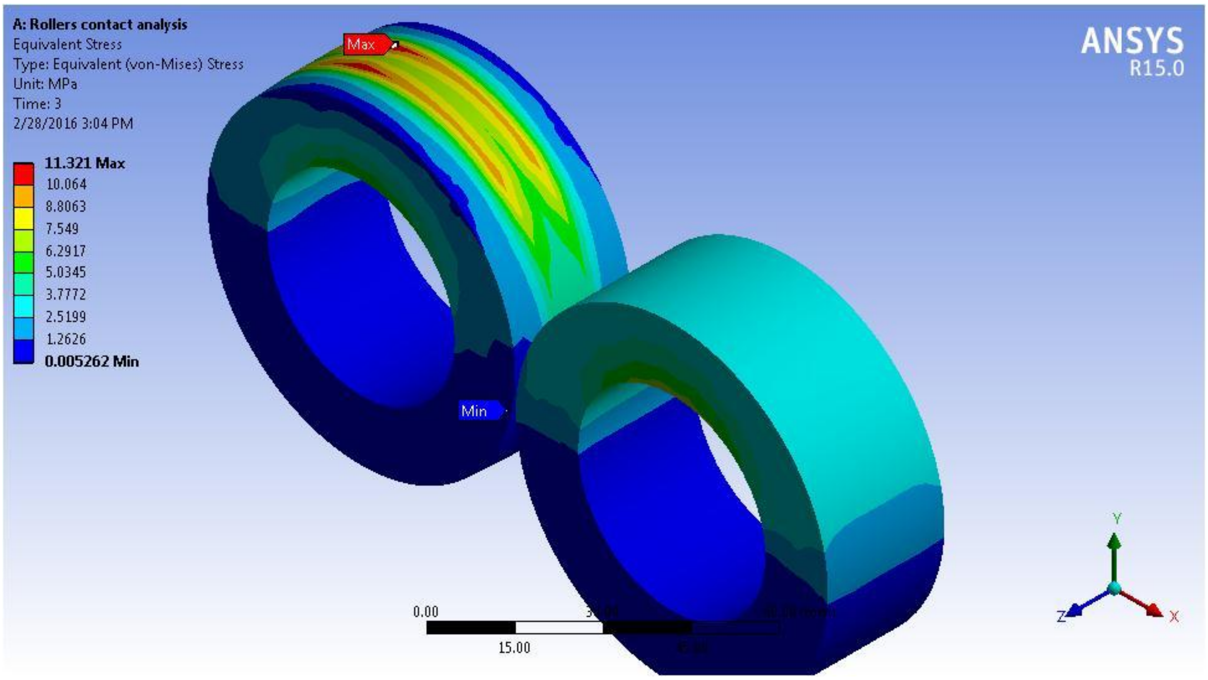


Figure 21 Von-Misses stress distribution at 5.7 KN

The Von-Misses stress at the constant normal force of 7.9 KN

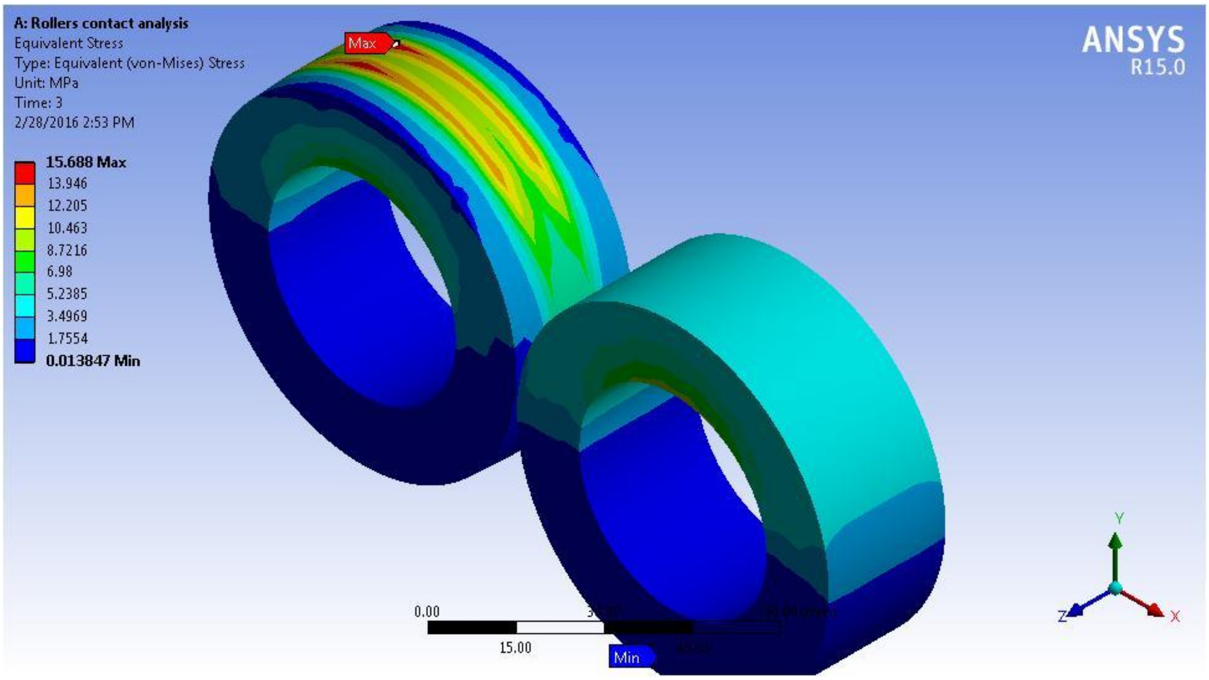


Figure 22 Von-Misses stress distribution at 7.9 KN

The Von-Misses stress at the constant normal force of 19.2 KN

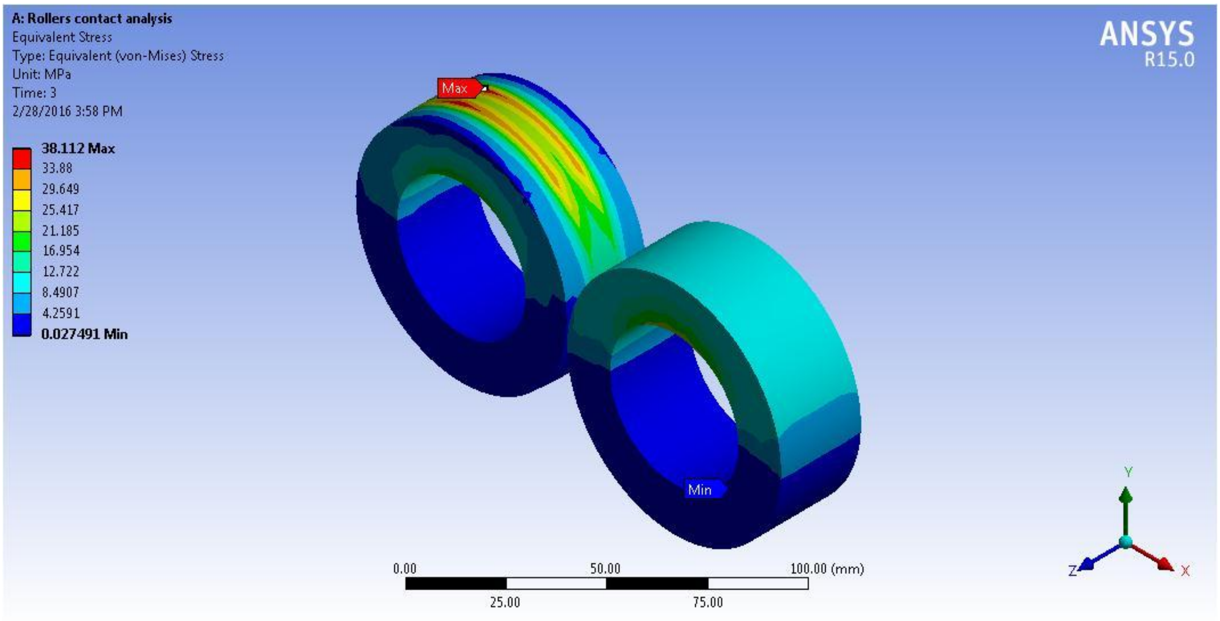


Figure 23 Von-Misses stress distribution at 19.2 KN

The Von-Misses stress at the constant normal force of 26.9 KN

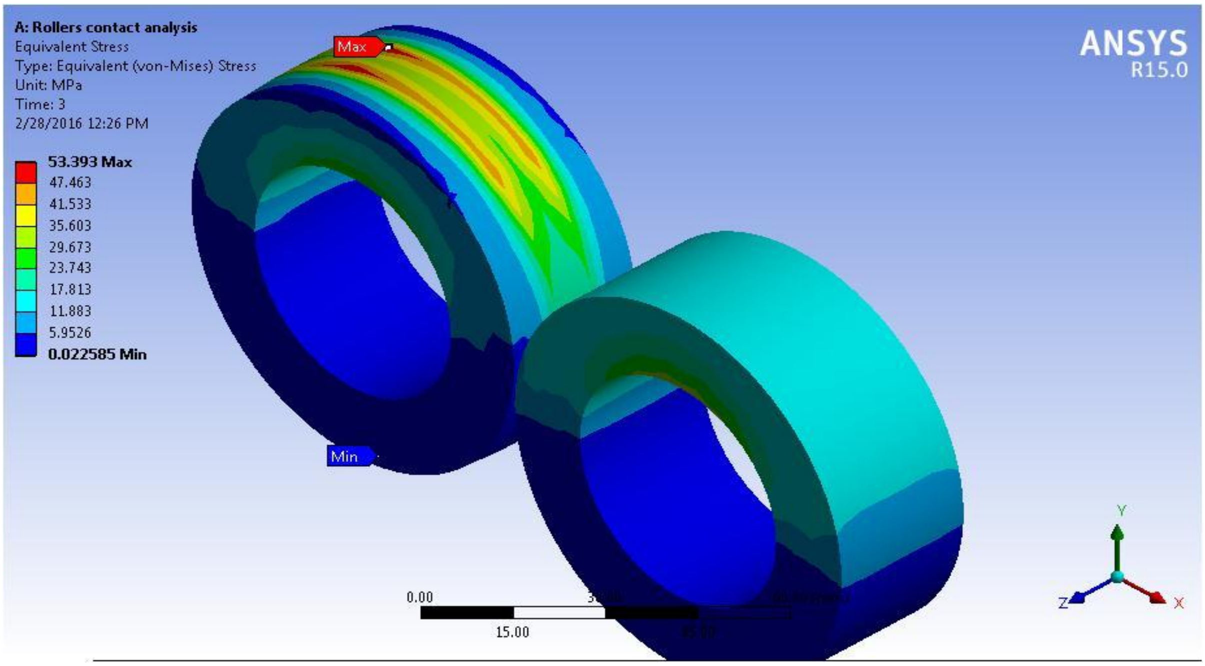


Figure 24 Von-Misses stress distribution at 26.9 KN

3.1.2 Defining a crack

Crack will be formed at the region of tensile zone where there is a maximum principal stress.

At the beginning, the coordinate system should be created at the surface of the roller in which the maximum principal tensile stress is located in order to generate a crack.

Create a local coordinate system:

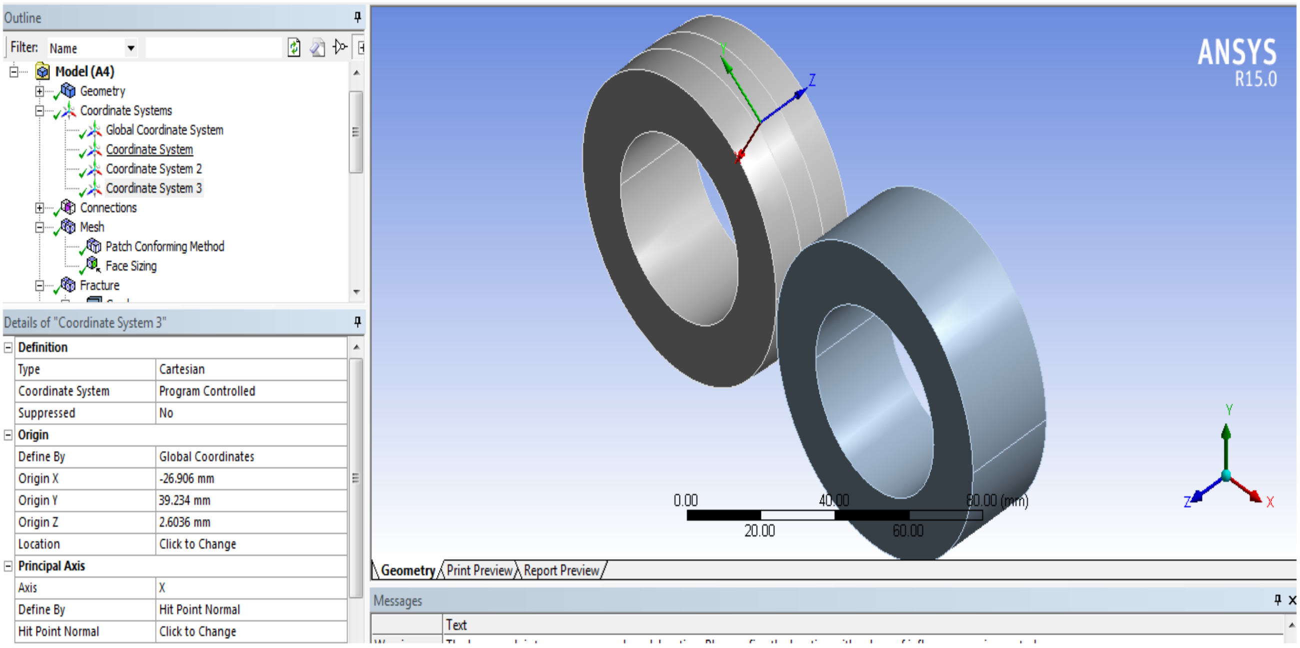


Figure 25 creating local coordinate system for crack

To define a crack the following steps are presented: select the Module object in the Tree outline, insert a Fracture object in to the tree by right clicking on the module object and selecting insert> Fracture, insert a crack object into the tree by right clicking on the fracture object and selecting Insert> crack. Then fill the crack details with related to the geometry.

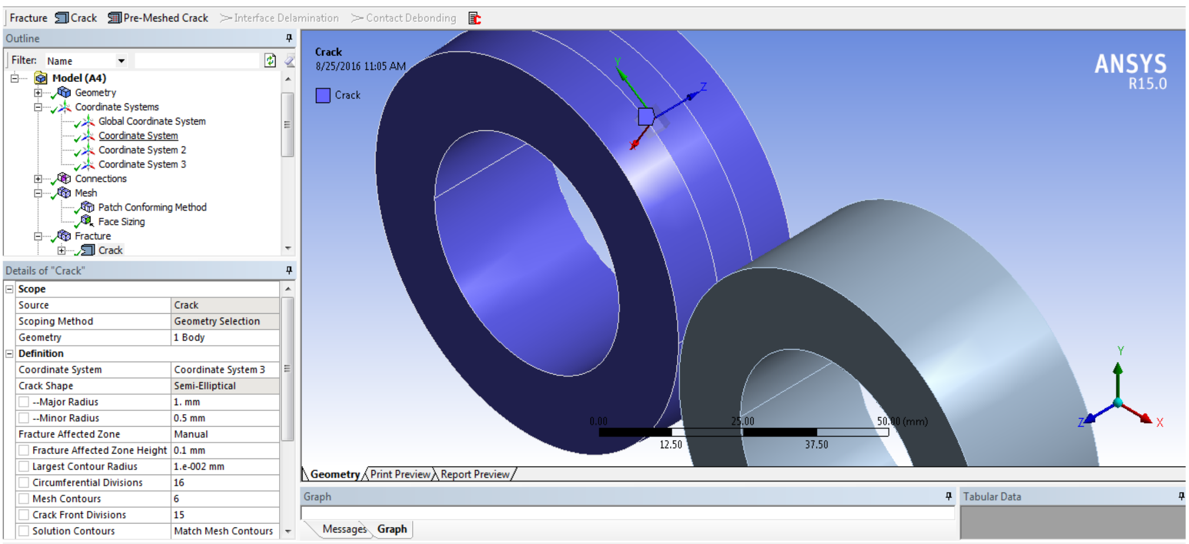


Figure 26 defining a crack

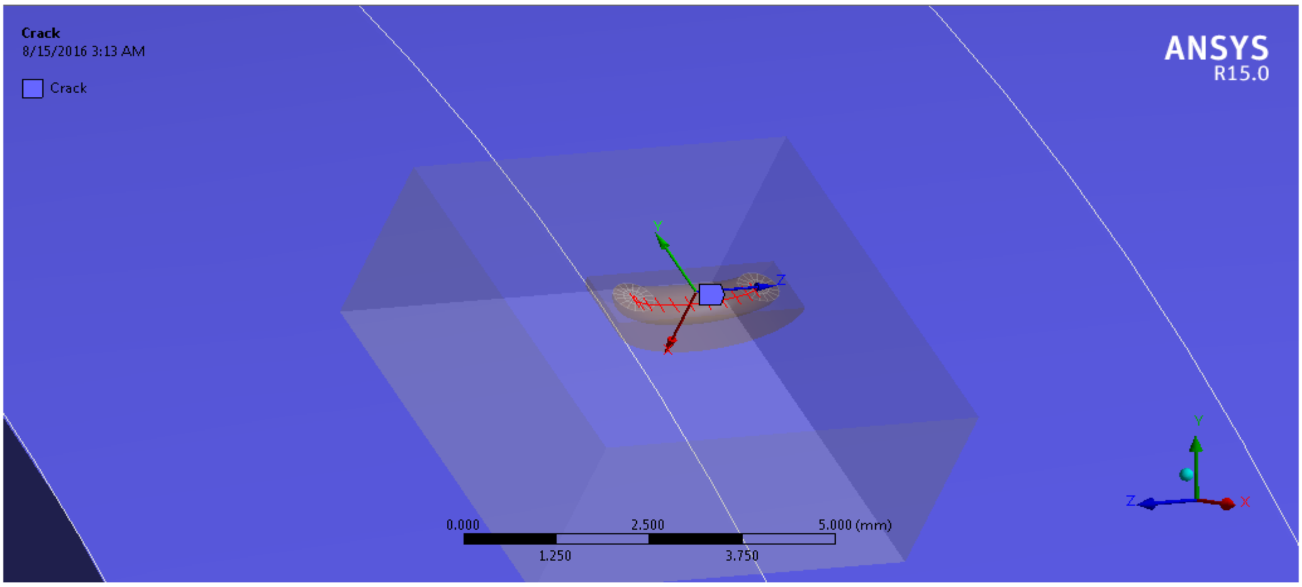
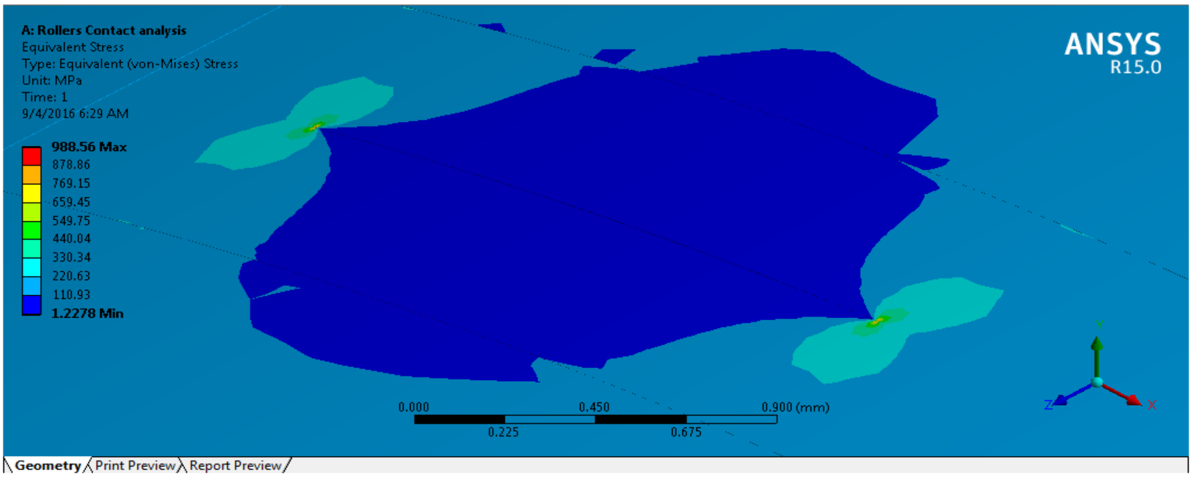
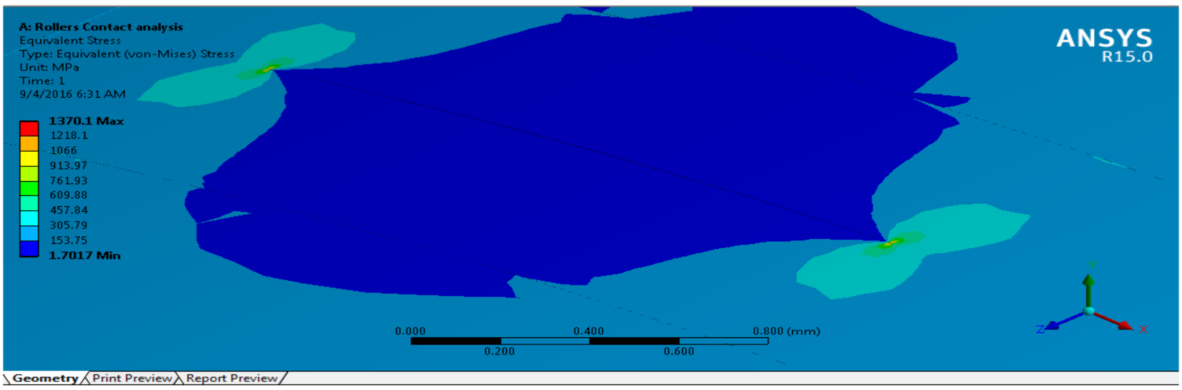


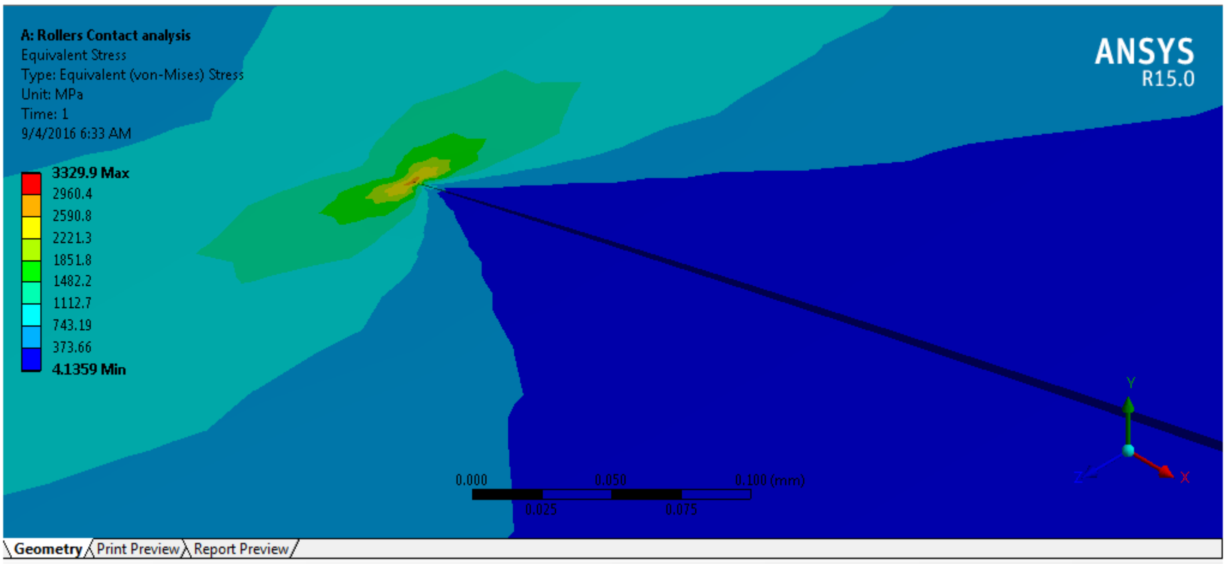
Figure 27 Magnified semi elliptical crack generation



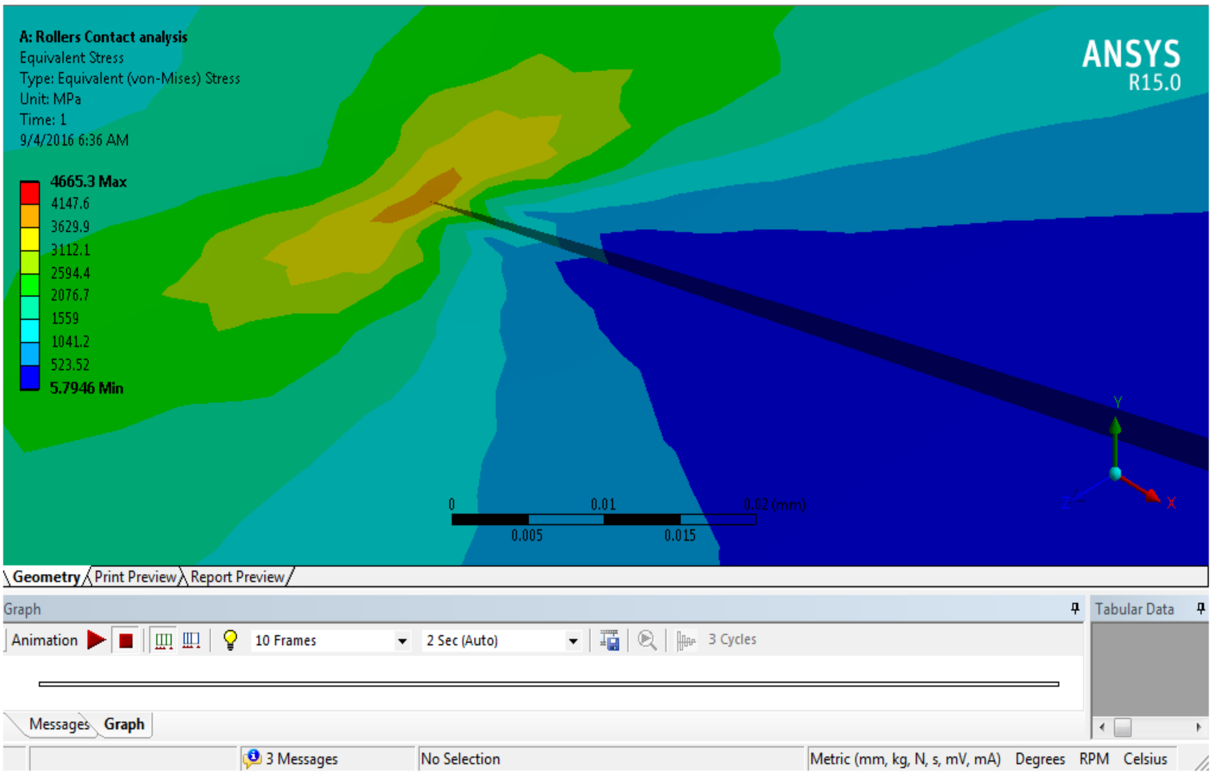
a)



b)



c)



d)

Figure 28 Magnified crack propagation (surface failure) at a) 5.7 KN b) 7.9 KN c) 19.2 KN d) 26.9 KN

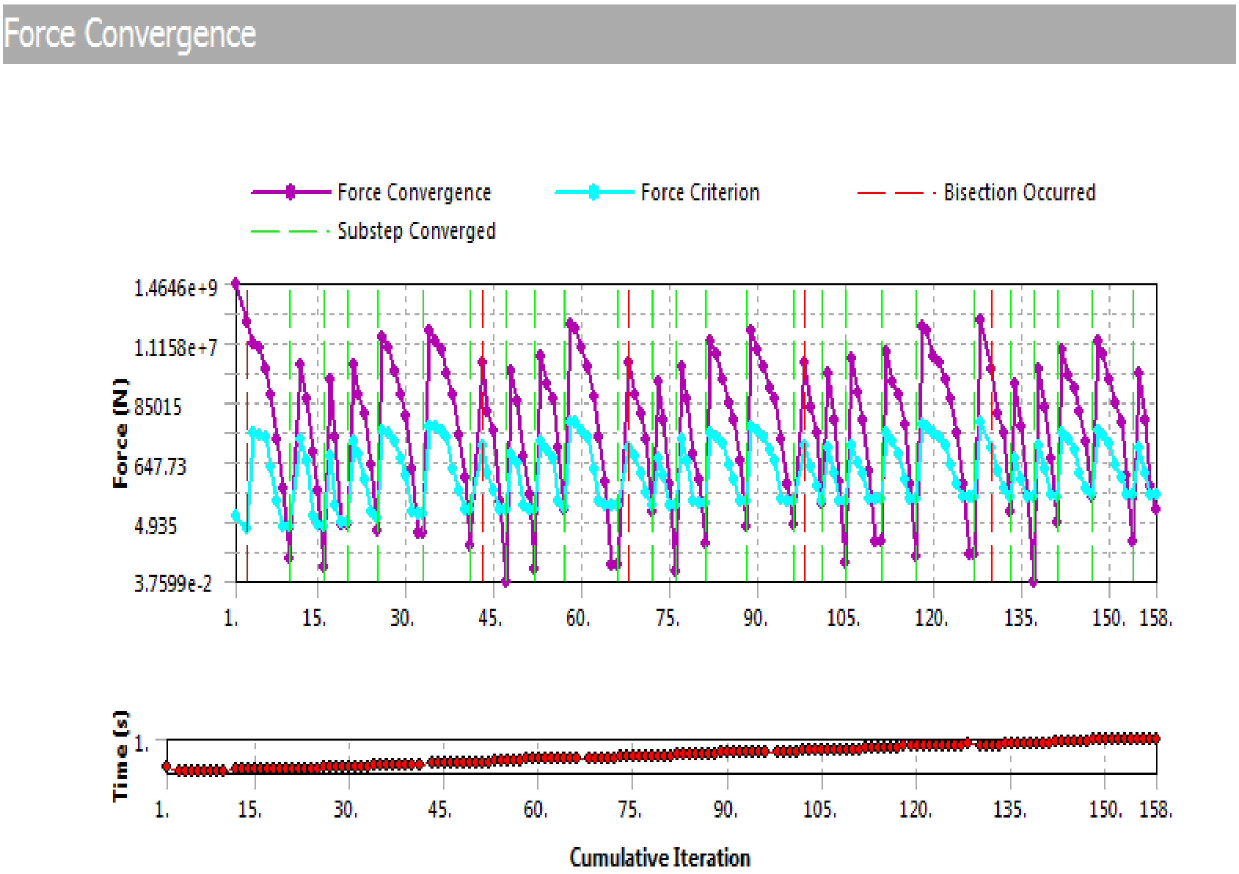


Figure 29 Force convergence

3.2 DISCUSSIONS

The analysis starts by defining the material properties in the engineering data, by applying the loads and boundary conditions in the static structural analysis. After all the results are found using ANSYS Workbench. The stress distribution that happened during rollers contact has shown from Fig. 21 to 24. Each figure shows that the Von Misses stress distribution that happened due to surface contact fatigue. As the applied external load is increased, the stress distribution also increased. It is noted that the maximum stress distribution (stress concentration) relatively close to the region of the tapered end of the roller. At the tapered region the principal stress is maximum which leads surface failure by defining a crack. After defining a crack and applying a normal load the equivalent stress becomes maximum around the crack front and the crack grows up. So that the material failure happened at the end of the contact region through crack propagation.

CHAPTER FOUR

4. CONCLUSION, RECOMMENDATION AND FUTURE WORK

4.1 CONCLUSION

This thesis is analysis of effect of stress distribution on surface failure of gears/ or rollers contact using ANSYS Workbench. The study focused from the experimental result of surface failure of rollers contact. In the experimental result the surface failure is happened at the tapered end of the roller and also in ANSYS WB it is happened at the tapered end of the roller contact. The general objective of this work is to know the reason where the surface failure being happened and to improve the material profile and validate with the experimental result. The load from 5.7 KN to 26.9 KN applied on the rollers and found that a corresponding maximum equivalent stress. Generally from this thesis work the difference of rollers relative speed increases the sliding velocity which tends to raise the contacting temperature, and therefore promotes more pitting.

On this study it is found that the maximum stress occurs at the end of the tapered roller of the rollers contact. The cracks that begin below the surface eventually grow to the point that the material above the crack breaks out to form a pit.

4.2 RECOMMENDATION

Knowing of the stress distribution in the gear teeth contact is significant. This investigation confirmed that it is possible to obtain the maximum von misses stresses to reduce the stress that leads to a surface failure by developing the material properties and reducing of the sliding ratios in the contact zone between the two contacting bodies and make the contact surface profile continuous and smooth. It means that reduction of the Stress distribution results in better load carrying capacity, micro pitting resistance and prolongs gear service life.

4.3 FUTURE WORK

In this thesis work the effect of stress distribution on surface failure of a spur gear has been investigated using the high speed roller and low speed roller contact, other influencing factors are not studied. So this work was restricted to the specified case. However, this paper can be extended to other situation listed below. Further study will be conducted on:

- Effect of temperature in surface fatigue failure
- Effect of surface finish on surface failure
- Contact mechanics in gears under lubricated condition and its effect in surface fatigue failure.
- Effect of temperature on fatigue life

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Appendix I

Generation of Rollers using Solid Work

Step1. Creat a new part in a part file

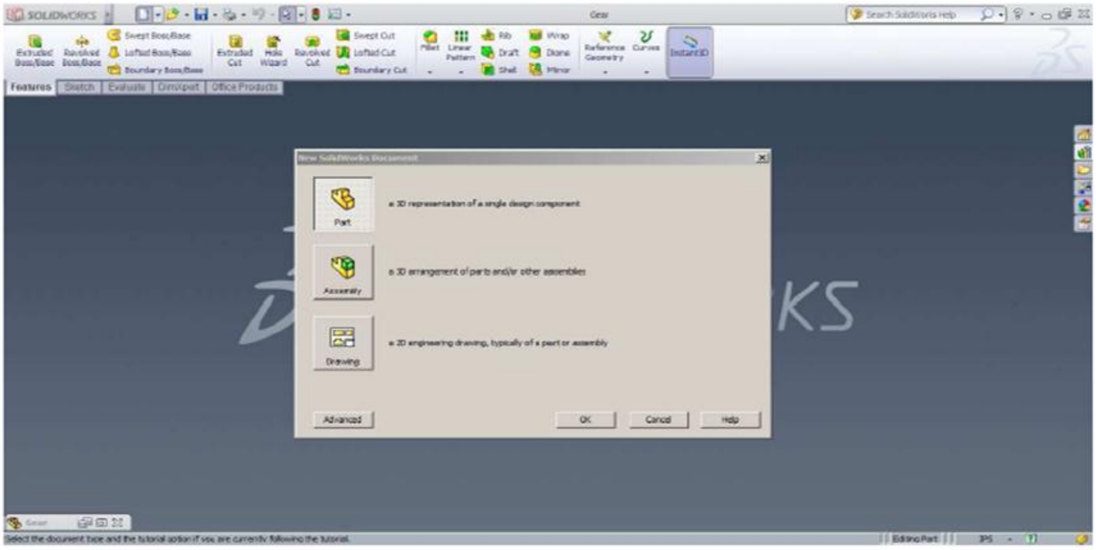

Figure 30 New part of solid work

Step 2. Select the front plane and start the sketch of the low speed roller

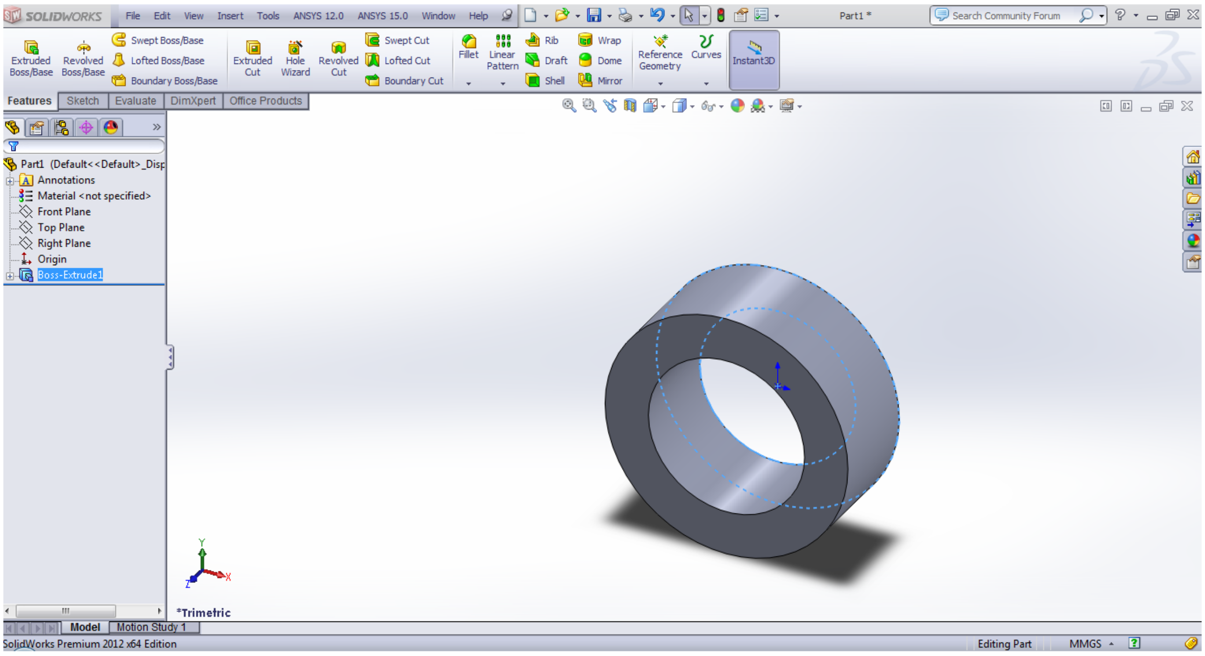

Figure 31 Low speed roller in solid work

Step 3. Again sketch the high speed roller, the high speed roller is sketched by tapering at the edge and hence the effective true contact length will be 8.5 mm from the total axial length of 28 mm.

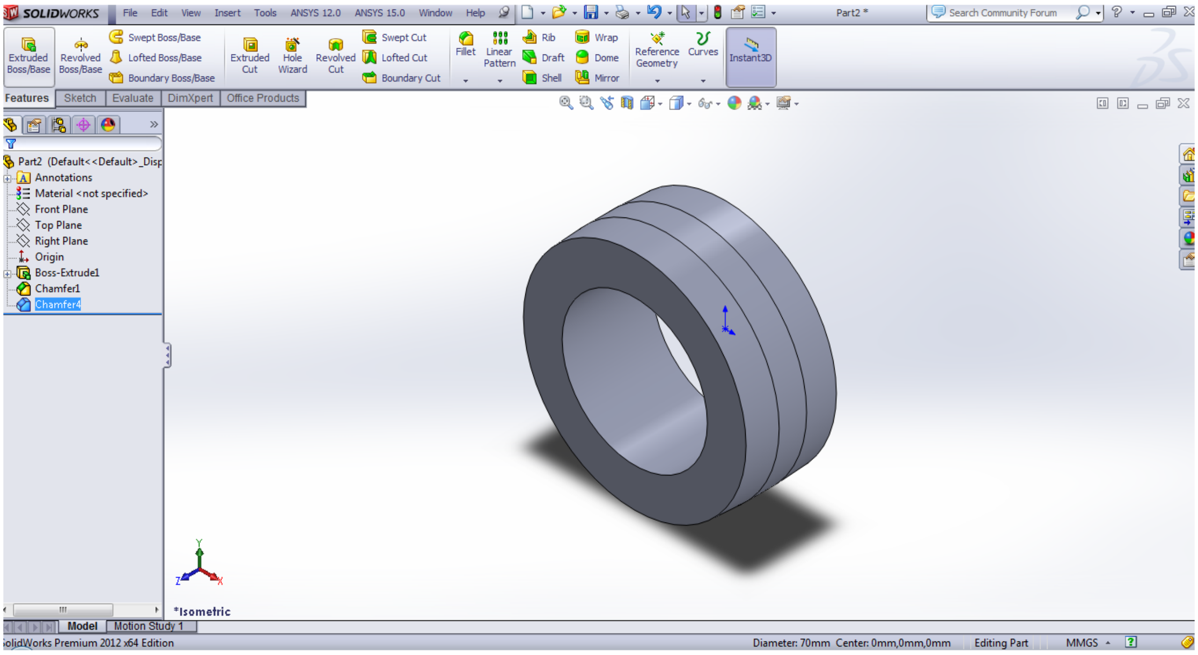

Figure 34 High speed roller in solid work

Step 4. Assembling of the two rollers

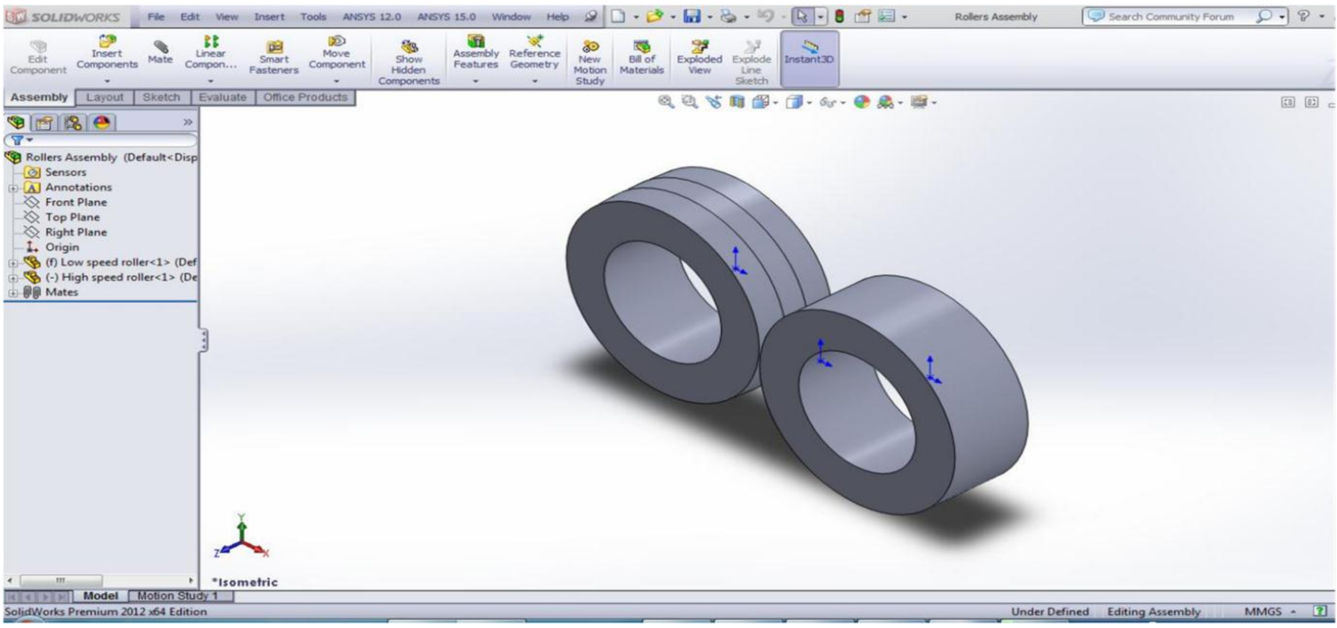

Figure 30 Assembly of two rollers in solid work

Appendix II

Steps during structural analysis

Step1. Double click "static structural" type analysis to add a new system

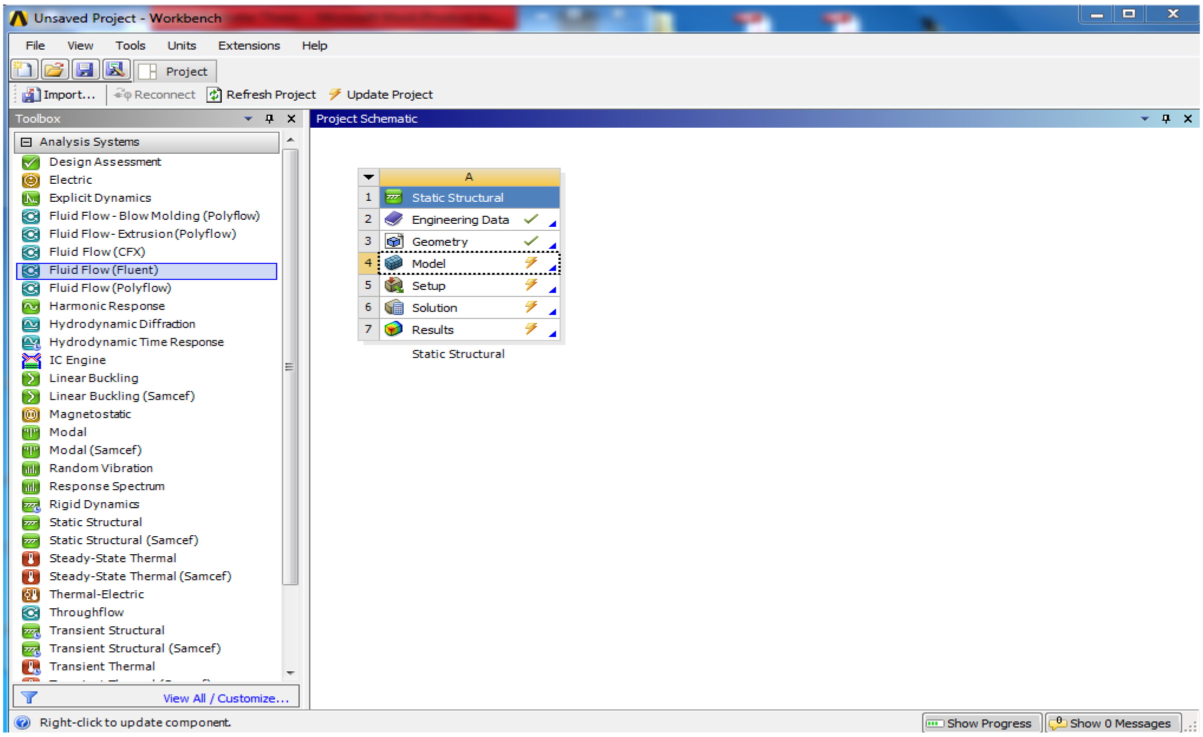

Figure 35 Begin static structural analysis

Step2. Double click"Engineering data" to adjust the material used for the analysis. The Engineering data application provides control for material properties. Once the engineering data open, the engineering data source (material library) displayed. In ANSYS Work bench the default material is structural steel but we can add in any material from the engineering data source and we can adjust the properties of the material. For this project I have taken the material to be a default structural steel.

Note that: only materials shown (or added) in the project view will be available in the analysis.

Outline of Schematic A2: Engineering Data

	A	B	C	D
1	Contents of Engineering Data		Source	Description
2	Material			
3	Chromium Molybdenum alloy steel		C:\Users\sami\Documer	
4	Structural Steel		General_Materials.xml	Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1
*	Click here to add a new material			

Properties of Outline Row 3: Chromium Molybdenum alloy steel

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	7700	kg m^-3		
3	Isotropic Elasticity				
4	Derive from	Young's Modulus and Poisson's Ratio			
5	Young's Modulus	2.06E+05	MPa		
6	Poisson's Ratio	0.3			
7	Bulk Modulus	1.7167E+11	Pa		
8	Shear Modulus	7.9231E+10	Pa		
9	Tensile Yield Strength	360.6	MPa		
10	Compressive Yield Strength	360.6	MPa		
11	Tensile Ultimate Strength	560.5	MPa		
12	Compressive Ultimate Strength	560.5	MPa		

Figure 36 Defining material properties in engineering data

Step3. Right click "Geometry" to import the CAD geometry by picking the browse button. But if we double click the "Geometry" the Design Modeler will be opened to model a new geometry. Remember that when the CAD geometry is saved it should be in „IGES" file format. If the green check next to the geometry cell is marked, we can assure that the geometry is loaded in the Mechanical workbench.

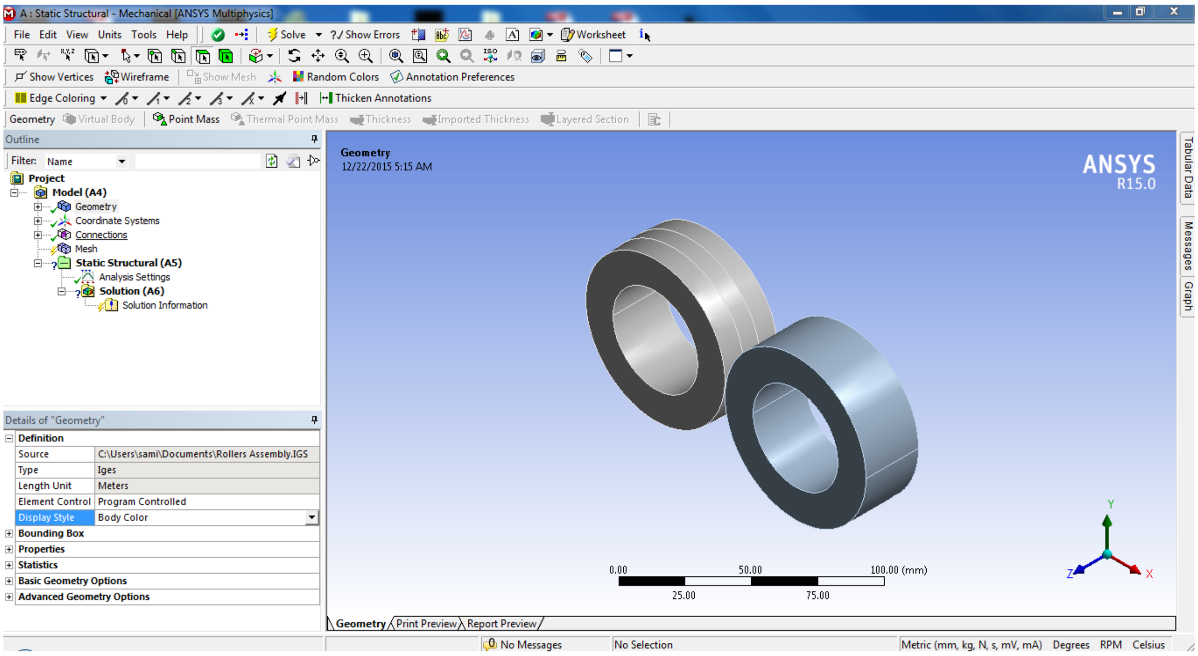

Figure 37 Import CAD geometry to ANSYS Workbench

We can identify the high speed and low speed roller parts from the assembly of the geometry.

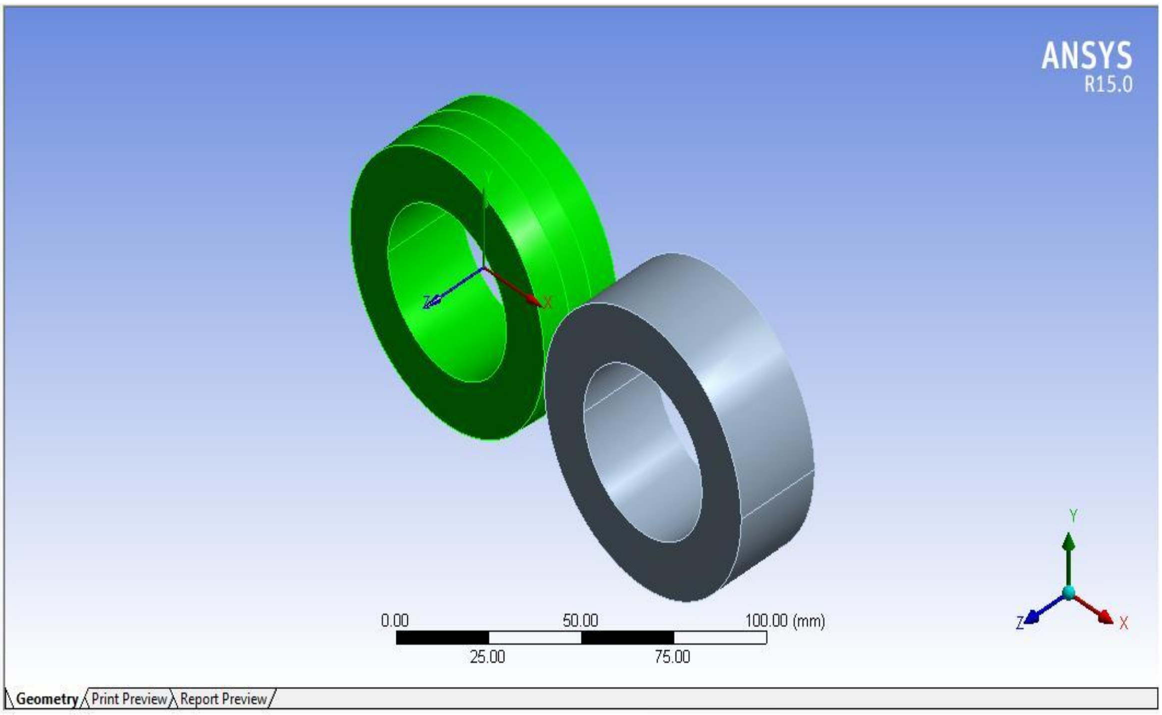

Figure 38 High speed roller highlighted in green color

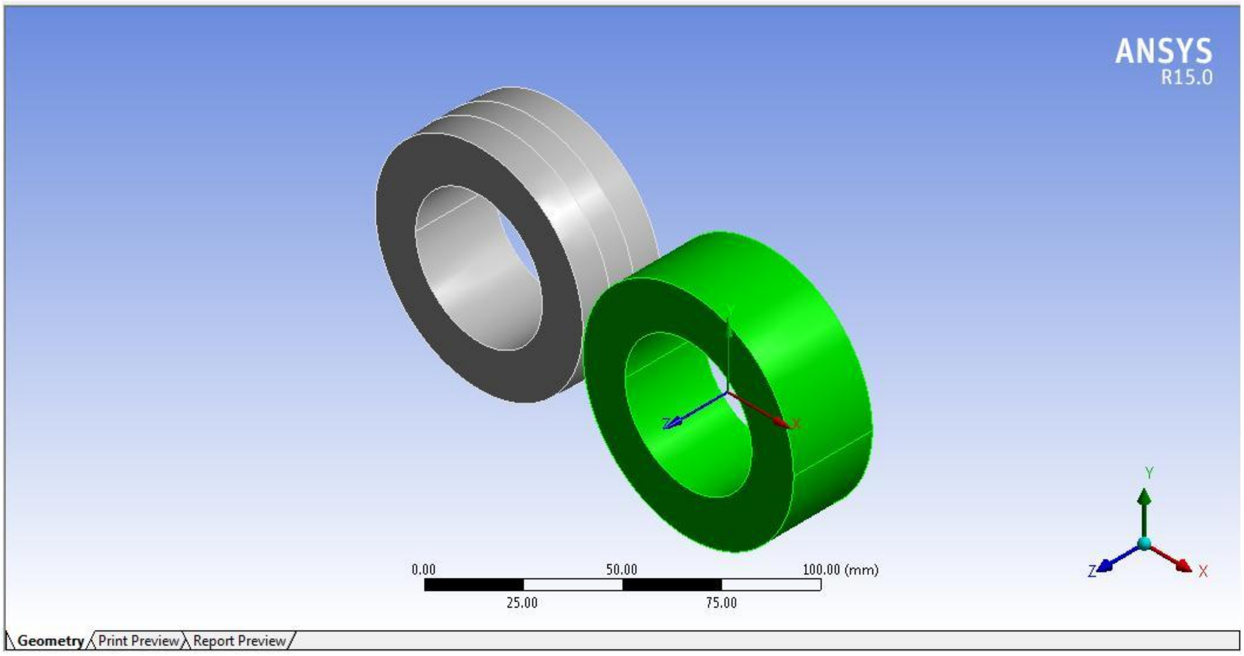

Figure 39 Low speed roller highlighted in green color

Step4. Double click"Model" to start the static structural analysis in Mechanical workbench. Here we can adjust the units to be Metric or Us customary units and we can edit anything that we have made in CAD modelling. Really this is the last phase of the whole procedures to solve the engineering problems in ANSYS. Almost all the analysis will do at this step.

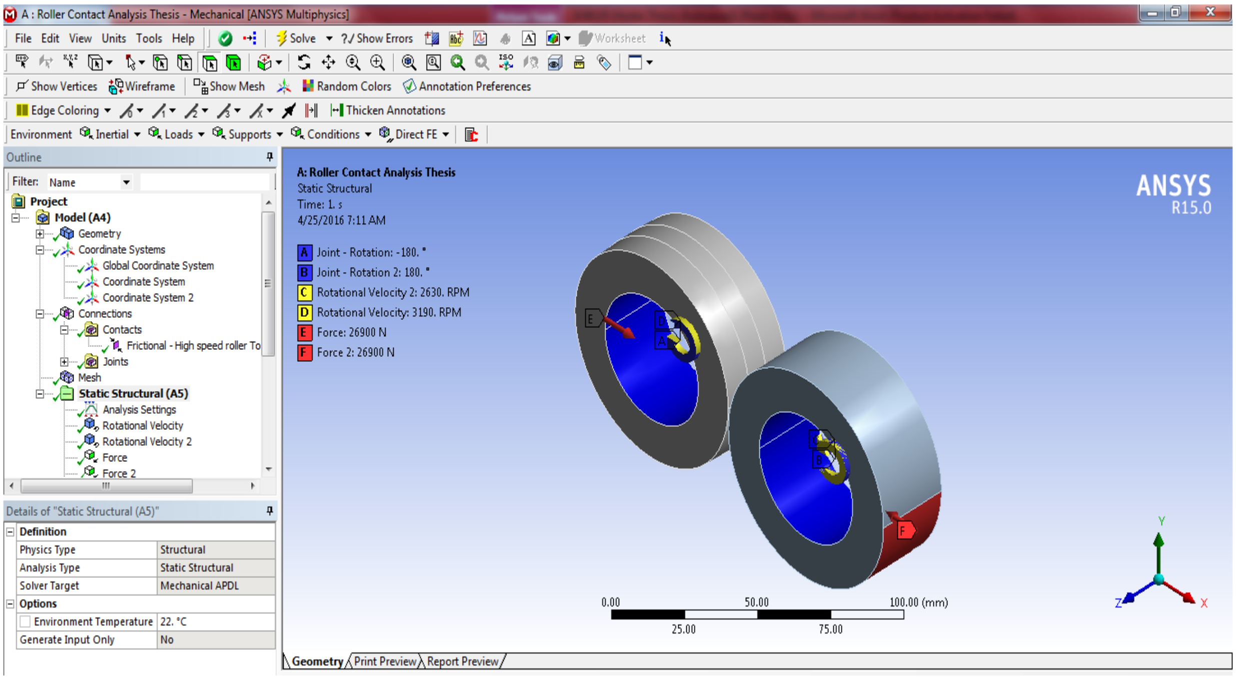

Figure 40 **start static structural analysis**