University of Technology, Sydney

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Finite Element Analysis of Spur Gear

A thesis submitted for the degree of

Master of Engineering by Research

by

Gagandeep Singh

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Certificate of original authorship

I certify that the work in this thesis has not previously been submitted for a degree, nor has it been submitted as part of requirements for a degree, except as fully acknowledged within the text.

I also certify that the thesis has been written by myself. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

Gagandeep Singh
February 2017

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Nomenclature

Symbol	Description	Unit
Principal symbols and abbreviations		
a	Centre distance	mm
α	Pressure angle	^{0}
B	Total face width, double helical gear	mm
b	Face width	mm
β	Helix angle	^{0}
C	Constant, coefficient, relief of tooth flank	μm
c	Constant	-
γ	Auxiliary angle	^{0}
D	Diameter	mm
d	Diameter	mm
δ	Deflection	μm
E	Modulus of elasticity	$\mathrm{N/mm^2}$
e	Auxiliary quantity	
ε	Contact ratio	
ξ	Roll angle	
F	Force or load	N
f	Deviation, tooth deformation	μm
G	Shear modulus	$\mathrm{N/mm^2}$
g	Path of contact	mm
ϑ	Temperature	$^\circ c$
h	Tooth depth	mm
η	Effective dynamic viscosity of the oil	$\mathrm{mPa \cdot s}$
i	Transmission ratio	
K	Constant, factors concerning tooth load	
L	Length	mm
l	Bearing span	mm
Γ	Parameter of the line of action	

M	Moment of a force	Nm
m	Module, and mass	mm, Kg
μ	Coefficient of friction	
n	Rotational speed , number of load cycles	min^{-1}
v	Poisson's ratio	
P	Transmitted power	Kw
p	Pitch	mm
r	Radius	mm
ρ	Radius of curvature Density	mm, $\mathrm{Kg/mm^3}$
S	Safety factor	
s	Tooth thickness	mm
σ	Normal stress	$\mathrm{N/mm^2}$
T	Torque	$\mathrm{N/mm^2}$
τ	Shear stress Angular pitch	$\mathrm{N/mm^2}$ mm
u	Gear ratio (z_2/z_1)	
v	Tangential velocity	$\mathrm{m/s}$
w	Specific load (per unit face width, F_t/b)	$\mathrm{N/mm}$
ψ	Auxiliary angle	°
x	Profile shift coefficient	
χ	Running-in factor	
Y	Factor associated to tooth root stress	
y	Running-in allowance	µm
Z	Factor associated to contact stress	
z	Number of teeth	
ω	Angular velocity	$\mathrm{rad/s}$
c_γ	Mean value of mesh stiffness per unit face width	$\mathrm{N/(mm.\,\mu m)}$
$C_{Zv,ZR,ZL}$	Factor for determining lubricant film factor (ISO 6336-2)	
c'	Maximum tooth stiffness per unit face width of a tooth pair	$\mathrm{N/(mm.\,\mu m)}$
F_t	Transverse tangential load	N
d_a	Tip diameter	mm

d_b	Base diameter	mm
F_m	Mean transverse tangential load at the reference circle = $(F_t K_A K_v)$	N
K_A	Application factor	
K_V	Dynamic factor	
$K_{F\alpha}$	Transverse load factor (root stress)	
$K_{F\beta}$	Face load factor (contact stress)	
$K_{H\alpha}$	Transverse load factor (contact stress)	
$K_{H\beta}$	Face load factor (contact stress)	
K_γ	Mesh load factor	
N_L	Number of load cycles	
R_Z	Mean peak-to-valley roughness (ISO 4287 and ISO 4288)	Mm
ρ_C	Radius of relative curvature at the pitch surface	mm
S_H	Safety factor for pitting	
S_{Fn}	Tooth root chord at the critical section	Mm
S_F	Safety factor for tooth breakage	
s_R	Rim thickness	Mm
σ_B	Tensile strength	$\mathrm{N/mm^2}$
σ_F	Tooth root stress	$\mathrm{N/mm^2}$
$\sigma_{F\ lim}$	Nominal stress number (bending)	$\mathrm{N/mm^2}$
σ_{FE}	Allowable stress number = $\sigma_{F\ lim} Y_{ST}$	$\mathrm{N/mm^2}$
σ_{FG}	Tooth root stress limit	$\mathrm{N/mm^2}$
σ_{FP}	Permissible tooth root stress	$\mathrm{N/mm^2}$
σ_{F0}	Nominal tooth root stress	$\mathrm{N/mm^2}$
σ_H	Contact stress	$\mathrm{N/mm^2}$
$\sigma_{H\ lim}$	Allowable stress (contact)	$\mathrm{N/mm^2}$
σ_{HG}	Pitting stress limit	$\mathrm{N/mm^2}$
σ_{HP}	Permissible contact stress	$\mathrm{N/mm^2}$
σ_{H0}	Nominal contact stress	$\mathrm{N/mm^2}$
σ_S	Yield stress	$\mathrm{N/mm^2}$
x	Profile shift coefficient of pinion or wheel	

Y_{DT}	Deep tooth factor	
Y_F	Tooth form factor	
Y_R	Tooth root surface factor	
Y_S	Stress correction factor	
Y_{ST}	Stress correction factor, relevant to the dimension of the reference test gears	
Z_V	Velocity factor	
Z_B, Z_D	Single pair tooth contact factor for pinion or wheel	
Z_R	Roughness factor affecting surface durability	
Z_X	Size factor (pitting)	
Z_W	Work hardening factor	

Publications

Gagandeep Singh, "Increasing life of spur gear with the help of finite element analysis," *International Journal of Recent advances in Mechanical Engineering (IJMECH),* vol.3, no.3, August 2014.

Singh, J., Gagandeep Singh, "New gear locking design in synchromesh gearbox which reduces gear shift effort," *SAE Technical Paper 2014-01-2328*, doi:10.4271/2014-01-2328, 2014.

Abstract

This thesis evaluates the service life of the spur gear in industry, showing that innovative techniques are required to resolve the problem of gear failure that occurs due to flank surface pitting and tooth breakage. Such techniques involve theoretical calculation, finite element analysis, hardness testing and selecting the appropriate material for the spur gear. Calculations were performed to measure contact stress, bending stress, and safety factor of the spur gear. This was followed by a finite element analysis (FEA) and software simulation. Then, the hardness test to compare the hardness of the materials was conducted. The material for the spur gear is chosen based on its mechanical properties. In this dissertation, the mechanical properties of currently used material C45 is compared to a new material, 19MnCr5.

The aim of the research was to increase the service life of the spur gear pair using suitable and reliable material. To expand the purpose of the study, attention has also been paid to the ISO 6336 standard-based calculation for the load-carrying capacity of the spur gear; FEA simulation using ANSYS software, and Rockwell hardness test were both conducted. From material analysis, the study found that the 19MnCr5 material has more fatigue strength, tensile strength, and better yield point as compared to C45 material. Also, through mathematical and FEA comparison, the study establishes that the gear designed with 19MnCr5 material fulfils the prescribed safety limits and would operate for its recommended service life. Furthermore, it is clear from a series of Rockwell hardness tests conducted, that after achieving higher hardness values by using 19MnCr5 rather than the C45 grade material, the gear would work without breakage.

For future study, it is suggested that there is a need to assess the effect of stress distribution variance over the flank and root of the spur gears, as this aspect has not been covered in the current context. Also, the stresses over the sub-surface of the gear teeth should be investigated. Besides this, research to find compatible lubricants for 19MnCr5 material is also required. Finally, observed differences in the hardness value at the rim and the tooth of the gear call for deeper analysis of the hardness testing process.

CHAPTER 1

INTRODUCTION

This chapter commences with the background of the spur gear. It is followed by the problem statement that outlines the existing issues in regards to gear design, and suggests possible solutions. Then, the objective of the research is determined, which is the justification for this thesis. Furthermore, the project limitation minimises the scope of the study. In the end, a brief layout of all the chapters is discussed to outline the structure of the research.

1.1 Background and motivation

Gears are used in most types of mechanical machinery. Like nuts and bolts, gears are common machine elements that will be needed from time to time by almost all machine designers. They are mostly used to transmit torque and angular velocity. It would be appropriate to say that, because of compactness and high degree of reliability, gears will predominate in future industrial machines as the most effective means of power transmission. Furthermore, refinement in the application of gear technology is necessary due to the sudden shift from heavy industry, such as shipbuilding, to automobile manufacturing and office automation tools.

Designing gears is always a highly complicated and intellectual field. As there is always a demand for enhanced service life of gears in industry, more efficient, reliable, and light-weight gears need to be designed and manufactured. For decades, several measures have been adopted to enhance the service life of gears, such as heat treatment, adjusting micro geometry. But, even after spending millions of dollars on gear research and manufacture, while designing a gear there is still a possibility of failure.

Many physical factors accumulate to cause a gear failure, including the material of the gear. Selecting different materials for gears plays an important role in this study. The material preferred for manufacturing a gear depends on the environment in which the gear has to work. For example, various high-performance gears are carburised (case hardened) to enhance their service life. Some special purpose gears, such as those used in chemical and food processing machines are made of stainless steel or nickel-based alloys, because of their corrosion resistance properties. Material selected for making a gear must satisfy two conditions; (1) Manufacturability and processing requirement; (2) Achieving required service life. Manufacturability requirement includes its forgeability and its response to heat treatment. Whereas, to achieve required service life, gears should transmit power to a satisfactory level when working in loading conditions, as well as fulfilling mechanical property requirements such as fatigue, strength, and response to heat treatment (Handbook of Gear Design, Second Edition, 1994).

The factors that should be considered while selecting the material are the availability and suitability of the material and, most importantly, the cost of the material. Ignoring physical and chemical properties, this research focusses on the mechanical properties. Mechanical properties of gear materials are strength, stiffness, elasticity, fatigue, and hardness.

In industry, gear designers have been working hard for years to achieve precise gearing without error, and to produce maximum service life. To reach the most refined level of gear design, designers refer to the standards such as DIN, AMGA, IS, ISO. These standards are strongly influenced by several safety factors. In this thesis, detailed calculations with the help of International Organization for Standardization (ISO) standards are performed to support the theoretical purpose of this study.

The increasing demand of more precise gearing systems, with noiseless functioning of gears, has enhanced the need for detailed analysis of gear characteristics. To meet this requirement and to reduce the cost of actual prototypes and field testing of gears, analysis software was introduced. Analysis software such as ANSYS is capable of performing finite element analysis (FEA) over not only gears' teeth, but each part of the gear body, such as the rim. Also, this software provides information of bending stresses, contact stresses, along with transmission error. To minimise the modeling time, preprocessor software that helps to create the geometry required for FEA, such as SolidWorks 2016, could be used. SolidWorks generates the three-dimensional spur gears easily. After designing and saving the geometry in SolidWorks, it is easy to import the same file into ANSYS. Advances in software development have opened a new era of gear analysis simulation. Computer simulation results have helped to achieve more accurate gear tooth profiles before manufacturing a practical prototype of a gear. In this dissertation, finite element models and solution methods are used to get accurate spur gear contact stresses and bending stresses. Then the contact stresses and bending stresses are calculated with the help of ANSYS software. In the end, the results are compared with ISO standard calculation results. The aim of the study is to increase the service life of the spur gear by proposing a change in material selection from C45(Nitride Hardened) to 19MnCr5(Case Hardened).

As computers have become more powerful, gear designers prefer to use computer software such as MASTA and Kiss-SOFT to perform the numerical calculation and design simultion. These software packages help to develop theoretical models to predict the effect of transmission stresses, and to conduct analysis of advanced transmission systems. But, these packages are expensive and out of the reach of a student, university and start-up organisation. So theoritcal calculation methods are used in this research.

1.2 Problem statement

In the past, a large amount of research has been conducted to evaluate mechanical properties such as fatigue strength of the gear. Some researchers have experimented with automobile gearboxes, and some have examined wind turbine transmission systems to investigate fatigue strength; however, the focus of their research was on materials already in use. In this research, the focus is on the initial steps of manufacturing. This is the step in which a gear designer selects material for the gears by comparing several material options.

Each material has its specific load-carrying capacity. This capacity defines the type of function that a material could perform. The load-carrying capacity depends on a number of mechanical properties such as fatigue strength, tensile strength, Young's modulus, yield point and average roughness. During gear manufacturing, fatigue strength over the root and the flank of the gear is compared, to select the material. Although C45 (nitride hardened) material is very common and highly desired in the gear industry, its failure at certain load conditions is a problematic area. Material analysis comparing C45 material with more suitable and efficient materials, with the help of simulation software, could assist in the identification and resolution of such problems.

1.3 Project objective

In recent research on gear design and analysis, several methods were used to increase the service life of the spur gear, such as adjusting micro geometry, profile modification, analysing friction between gear meshing, flank modification, analysing crack initiation region, changing heat treatment and changing material. Also, numerous papers were published on how noise and vibration affect the life of spur gears. However, most of the literature only considers the possiblity of changing of gear material by discussing the effects due to change of gear material in general, rather than providing analysis of some specific gear material by perfoming FEA and other experiments.

In this study, an innovative and exciting approach to designing a gear with different gear material is suggested. The project is divided into three parts, in terms of design, analysis and experimental testing. In the past, gear analysis was conducted with the help of analytical methods, which involves theoretical calculations related to tooth stresses. Also, various assumptions, influence factor and simplifications were used. In this thesis, the same approach has been taken to exhibit the comparison between the spur pinion and gear of two different materials, by performing analysis of contact and bending stresses.

The objective of this thesis is to investigate the service life of spur pinion and gear using suitable and reliable material. Designing a strong and noiseless spur gear requires the analysis approach. This approach should easily be implemented and provide information on flank (contact region) and tooth root stresses. Finite element method (FEM) is used to provide analysis. This dissertation shows the FEA-simulation comparison between spur pinion and gears manufactured with two different materials, that is, C45 material and 19MnCr5 material. The work is summarised as follows:

- Calculation to evaluate the load-carrying capacity of spur gears by using formulas of flank safety against pitting, and root safety against tooth root breakage.
- Comparison of materials to assign correct alloy combination to the gear and pinion.
- Comparison to investigate the root bending stress and contact stress distribution over the gear and pinion at given operating speeds.

- Comparison of mathematical and FEA results with ISO 6336 (International standard).
- Performing practical tests, such as the Rockwell hardness test, to assure that the 19MnCr5 gear material is more efficient then the C45 material.

1.4 Project limitation

To restrict the scope of this thesis, the following limitation has been introduced:

- Due to constrained financial aid, limited testing is performed. For further research and implementation of new gear material leading to future industrial usage, gears need to go through several other experiments and must achieve satisfactory results.
- The value of some safety factors are taken as 1.
- Only spur pinion and gear mounted on the shaft of the gear box is inspected in this thesis.

Aspects of research beyond the scope of this research work are:

- Investigation of lubricants used in transmission systems.
- Practical impact of temperature on gears and bearings.

1.5 Thesis layout

Detailed simulation and mathematical analysis in this thesis have required diligent application of theories, leading to virtual and practical testing to adequately represent gearing systems and bring the issues of gear failure into the limelight. The detailed calculations in following chapters present gear failure data and gear upheld data for more service life, followed by simulation results. To focus reader attention, this thesis is divided into 5 chapters.

Chapter 2: This chapter includes a detailed literature review. Initially the background information is to introduce general topics of gears, followed by detailed literature research on gear technology, factors effecting service life and material characteristics, while strategies for improving the gear design and manufacturing are discussed.

Chapter 3: This chapter is devoted to the development of formulated models of gearing systems. Forces on spur gear pairs are consolidated to compare results with old and new gear materials. All the influencing factors and safety factors on gear calculation formulas are discussed and checked, to determine whether new gear material satisfies those conditions.

Chapter 4: This chapter starts off with the introduction of gear design methods and finite element method (FEM), followed by gear simulation, resulting in illustrations of faults within the design. This leads to the description of how material replacement can rectify the gear failure problem (virtual probe).

Chapter 5: The first pragmatic test is carried out in this chapter. Rockwell hardness test is performed in a university laboratory with different gear material. The first test is with C45 spur and helical gear, followed by 19MnCr5 spur and helical gear, and then the results are compared.

Chapter 6: The concluding chapter summarises each of the previous chapters and presents important results of the thesis, as well as suggesting the vital areas for extending this research.

CHAPTER 2

LITERATURE REVIEW

In this chapter, a survey has been conducted that peeps into the history of gears as well as seeks arguments in reigning paradigms for development of gear design with new-age materials. The use of new materials for making gears must pass the standard endurance tests. The main reason behind undertaking such research is that there is a continuous demand to improve performance, reduce overall weight of gearbox assemblies, and increase the temperature endurance of the gears. New designs are expected have more life and conform to highest possible safety standards.

2.1 Introduction

The use of gears is as old as the history of machine-making [1]. Initially, they were used as a device for transmitting rotatory motion for machines. These machines were made up of wooden or metal gears. Right from the ancient days of Greece, to the Roman Empire [2], to the current usage, the gear designs have undergone tremendous changes. The major chronological pointers that are worth mentioning from the history are as follows:

1) Most of the references [3] related to gear history mention the making of "Always South pointing" chariots by China as the first important milestone in the history of gear making. The mainstream medium was wood. The earliest design was based on the concept of pins engaged with each other.
2) Historical evidence has been found regarding the use of stone for building gears (Sweden) [2]. Romans and the Dutch have also made extensive use of gears made of stone and wood in their water and power mills.
3) Mechanical clocks [4] with gears started appearing in 1285 in Europe, and by 1656, gears became a part of the pendulum clocks.
4) In the Netherlands, windmill gears were made with the help of hard wooden material.

2.2 Review of technical advancements in making gears

The current industry exhibits hi-tech usage of robots [5], 3-D printing machines and software to build gear assemblies. But the penetration of such technologies is not extensive. Moreover, the type of materials that can be used in 3-D printing [6] is limited. The strength of the materials used in 3-D printed technologies does not find much application in designs that need to comply with long life with high/tight safety standards. The CNC [7] machine(s) are the industry norm. The gears made by the CNC process have better precision and strength than most 3-D printed machine work. The CNC machines, which may have spindle or rotary tools to cut away the wood or metal, may be used to build tools and parts of the gear assemblies. This can be done by either manually operated CNC machines, or fully automated tools. In the case of typical 3-D printed machines, molten material is pushed to form the desired object or component.

Then, there are laser cutting machines that can work in different axes in high precision cutting of the material that can be shaped into gears. An alternative to this technology is water jet technology, which can also be used to cut to form gears. But the precision will be low when compared to laser cutting machine work. The gears made from water jet technology cannot be used for precision machine assemblies. The cutting precision of the laser cutters can be compared to the level of CNC machines, especially in cases where the surface is flat and smooth. Where the gear material is stone, water jet technology [8] will be economical for making gears in 3 axes. However, to provide equal quality, a CNC router will give tighter tolerances for manufacturing the gears.

The early development of electronics and computers was dependent on the number of transistors that could be incorporated in the component. However, the robotics sector isn’t waiting around for transistors; it is waiting for high precision gears and assemblies. These gears must be designed to be highly flexible, with high grade endurance to wear and tear due to temperature, friction and other aspects. This is not possible unless material science contributes its research and resources. Today, plastic gears reinforced with carbon nano-materials [9] are taking their place in the industry to provide alternatives to heavy metal gears. Industrial research has found that these gears are easier and cheaper to produce than their metal counterparts. And because they’re not made from metal, they’re considerably lighter, which in turn makes the vehicle lighter and improves its fuel efficiency. This improvement in the gears has been achieved by Japan’s Gifu University [69] adding a thin layer of carbon fibre running through each tooth that adequately reinforces and strengthens the parts.

2.3 Types of gears, important differences, and similarities

Gears perform many functions, hence they are designed with the functions that increase or decrease the angular velocity, while simultaneously increasing or decreasing the torque so that the energy is conserved. Typical concerns of a person purchasing gears are its features in terms of its number of teeth. This measure gives the circumference, which in turn determines the radius. For example, if the number of teeth is increased by a factor of three, the circumference also increases three-fold. However, the type of gear and its application depends mainly on three aspects. The first one is the geometric properties, second is the material characteristics and the third is design suitability. Based on these three aspects the function and type of gear may be understood.

Geometric properties

The geometric properties refer to the gear shape in terms of its tooth, tooth spacing, tooth thinness/thickness, distance properties like distance from the centre, bore size, and other properties like flatness, bolting and doweling. The gear adds mechanical advantage to the machine based on these properties. These properties are incorporated by mapping the design goals to the desired functions of the gear assembly. Another way of classifying gear type is based on their manufacturing process. Broadly speaking, gears may be made by a machining (material removal) process like hobbing [10], milling [10], shaping and broaching [10], Classification can also be done based on the process of stamping and extrusion. Then, they may be classified on the basis of additive manufacturing methods such as powder metallurgy and die casting. It is critical to maintain the safety and service standards, especially when geometry is one of the key criteria used to define its functionality. The following sections review the geometric factors that influence the serviceability of the gears.

2.4 Review of geometric factors influencing the serviceability

The speed at which gears move, and their surface characteristics, influence the serviceability of the gears. However, in this section, only geometric features of the gears are discussed. The dimension of the thickness of the tooth root influences the bending stress in all types of gears. The life of a typical gearbox assembly of an agricultural tractor [11] is hard to calibrate because it is difficult to measure the precise load cycle as it is used for a variety of operations and in different soil types. Therefore, by conducting a series of simulations, the authors [11] were able to estimate the safety of bending strength and contact strength durability of the gears. The dimensions of the tooth thickness were varied to build a target serviceable gear box assembly. The contact ratio [12] is also important as it involves measurement related to the tooth engagement positions. The authors [13] basically experimented with different tooth engagements by varying various factors such line of action, plane of action, limit diameter. The FEA analysis gave insight into the right combination and position of gear parts to provide acceptable tooth wear per mesh for both standard spur gears and non–standard gears. By conducting this research exercise, the authors were able to identify tooth wear, surface wear and unbalanced tooth issues that influence the serviceability of the gears in long run.

During the gear design process, a specific centre diameter with respect to velocity ratio is desired. There is a need to check if there is any kind of interference in the system that may lead to reduced gear functionality in the long run. In research work, the authors worked on a number of geometric characteristics [14] to check this factor. These parameters include: face width, diameter pitch, number of teeth on the pinion, contact ratio, dynamic factors and dedendum ratio; the values of these factors were subjected to the optimisation algorithm (Genetic Algorithm [15]) so that an optimal centre can be achieved for improved performance of the helical gears.

Face width [14] is one factor that also influences the life and working of the gears. The spur gears mesh tangential and radial loads that act on the gear tooth. This force may lead to change in tooth dimensions including face width, thickness etc. The authors [16] have done an FEA analysis of the gear design to ascertain the bending stress limits that would influence the safety metric of the gears. Then, the profiles of the gear teeth that are involute to the circle also influence the working of the gears in long run. The contact between a pair of gear teeth occurs at a single point where the two involutes meet, and this also influences the stability of the gears while in service. The research work's [14] main focus is also the geometry of the gears working as pairs. This paper uses a computational model of load distribution. This model considers the rigidity of the gear teeth (Internal Tooth) material and path of the contact to build the model.

When the teeth of the gear are projected radially and are parallel to the axis of the gear, the parallel axis comes into play. In the paper [17] a computation model of modeling parallel axis gears is done with respect to the calculations of the bending stress. The focus is to establish a generic model that would be able to determine the bending strength geometry factor [17] called “J-Factor”. This model works for V-shaped, straight tooth and helical tooth-based gear designs. The computational geometric factor was also tested and simulated using the FEM method.

The imbalance of load cycles on the tooth creates many issues. To overcome these challenges, the geometry of the gears is changed so that the asymmetricity helps in maintaining and extending the life of the gears. This is exactly what is being done in this [18] work. The objective is to overcome the issue of imbalance of load on the tooth. This can be done by changing the geometry to an asymmetrical tooth which improves the performance and is able to overcome the functional difference. Such an arrangement helps in simultaneously increasing the contact ratio as well as the operating pressure angle beyond the conventional gear’s limits. The paper’s [18] motivation is on tooth geometry optimisation, which impacts the service of the gear assembly. The asymmetric tooth gear [19] design overcomes the conditions in which tooth load on one flank is greater than the other, and applied for a longer period of time. The role of micro-geometry is fundamental to the gear design and impacts the durability of the gear assembly, as do the service and safety limits. The emphasis in micro-geometry of the gears is on the optimisation of the macro parts of the gears, or just defining the durability of the gears in terms of maximum load and duration. The emphasis of [20] research work is on the bevel type of gears. It is an analysis of its micro-geometry [20] when load is applied. This is done by conducting analysis of “Tooth Contact Analysis” and dynamic bevel excitation behaviour. The outcome leads to an optimisation of distribution of load across the tooth face while simultaneously keeping the transmission error (or TE) low across the operating range.

It is apparent from the above discussion that the various parts of the gears are affected during their service life due to their geometric characteristics. Hence, the manufacturing process becomes supreme in deciding the quality of the gears. Gears made using the hobbing and stamping process [21] usually have major problems in tooth wear, and consequently have a negative impact on service life of the gears. High precision gears cannot be manufactured using die and casting methods, and the powdered metallurgy methods needs fine-gained metal powders, which are not easily available. Moreover, due to its geometry, this method of manufacturing is not suitable for spur gears. Wear of extrusion is the main root cause of failure in such cases of gears manufacturing. In all such cases the geometry and material are critical for targeting appropriate levels of serviceability of gear assemblies. The next section focusses on the possible materials that are being used to make gears and other parts of machinery, such as shafts, on which the serviceability and safety depend.

2.5 Material characteristics

The material characteristics refer to the properties of the material with which the gear is made. The material properties define the quality of the gearbox in terms of its endurance to temperature, water exposure, radiation exposure, air pressure on wheel cups etc. Basically, the input factors like horsepower or torque of a particular gear, the materials for the pinion and gear, the operating centre distance, number of teeth on the pinion and gear, the pressure angle, face width, pinion RPM, operating temperature, module or diametric pitch, are all important for selecting the right kind of gear material while designing the serviceability of the gears. The material characteristics in terms of allowable stress, Poisson's ratio and moduli of elasticity are considered in the selection process of the material. The gear casing and the shaft material must have desirable tensile strength, and the material should not develop cracks or voids in it, while in use. Hence, based on the application of the gears, appropriate materials are selected. The task of selecting material is quite complex due to the large number of options in materials.

The next section examines the recent usages of materials for manufacturing gear assemblies. For the sake of brevity and simplicity, the section is divided into multiple parts as follows, and the information below applies to all types of gears [3], including spur and its sub-types (metric, hubless, plastic, steel, injection molded), worm (stainless steel, steel, brass, plastic), helical (axial, hobbed), spiral (alloy steel, carbon, plastic, nylon), and bevel types of gears.

2.5.1 Steel-made gear assemblies

In the paper [22] the work has been done on worm and spur gears. These gears are part of an assembly of bending machines, which consist of rollers to flatten the metal sheets. The authors had simulated the machine design in the CATIA [23] software. Hence, this example shows that mild steel also finds many applications in making working machines.

There are many instances where Ca-treated carburised steel grades [24] [25] have been used to make gears. A major thrust of the research work [26] has been to increase the life of tools, so that the wear rate is minimised and there is reduction in wear patterns like flank and crater, micro chipping, edge fracture and nose wear. The gears made from these materials are used in polycrystalline cubic boron nitride (PCBN)-based [27] tools and gearboxes that are used for cutting, loading and transmission. PCBN composites are produced by sintering micron CBN (cubic boron nitride) powders with various ceramics, so as to produce extremely hard and thermally stable tooling materials. Most PCBN materials are integrally bonded to a cemented carbide substrate. CBN is the second hardest material known after synthetic diamond, but has high thermal and chemical resistance properties. PCBN composites provide extreme resistance to deformation and wear at high temperatures – typically an order of magnitude better than the nearest ceramic materials. In the case where the material is Ca-treated steel, the authors have claimed that the process doubles the tool life as compared to the standard steel grade material. This was because of increased surface hardness. This led to reduced economic cost of making gearboxes with increased machinability. The authors [26] have claimed to improve the life by using carburised steel grade 158Q also, and achieved increased endurance limits of the gears/tool by new material and investigating the role of clean steel. It should also be noted that conventionally-used carburised steel grades for gears include SAE 8620 [13] [28], 4320, and 9310. The research work [24] is basically a review of the gears used in helicopters. It covers a wide range of topics in the discussion of noise in the transmission, and stiffness of the material used in making such gears, etc. The gear types of spur, helical, magnetic, composite gears, face gears, shift type gear are formally discussed.

The cost of machining a typical gear is sometimes more than 50% of the total cost. This happens normally when a significant grinding after carburising is required. In such cases the manufacturer considers different options in terms of material or manufacturing process. This is one way to experiment and use new combinations of alloys, compositions and elements. High quality nitrating may be used instead of carburisation, as the nitrate gears with titanium as the main material [29] (Ti-6Al-4V or Ti-6Al-2Sn-4V-2Zr) will not require a case as deep as carburised gears. Then it can be used in case there is need to substitute steel gears for lowering weight, with a trade-off of strength. The aerospace industry may require reduction of weight, hence the authors [29] have worked on this aspect. According to research [30], the steel grade 158Q with carburising increased the fatigue life by 20% as compared to conventional steel alloys. Clean steel shows similar behaviour as per the machine trials done in this research work. Gears in consideration for this work are used in transmission operation in a machine.

2.5.2 Alloy-made gear assemblies

The material ASTM B505 (tin bronze alloy) is used in the applications of bearings as well as in gears. The work of [31] shows the design and conduct of an analysis with dynamic parameters of the servo press for improving overall safety limits of the helical gears. The FEA analysis shows that the press machine could be operated within safety limits due to good design and careful selection of material for each part. Then, in [32] literature bronze-steel alloys have also been mentioned. This paper also mentions CuSn12 as key material and 16MnCr (worm material) has been mentioned. In the paper [33] the claim is that experimenting with multiple conditions and combinations of factors related to safety can bring a better design of gears. The authors had tried variation of distance between worm shaft bearing, decreasing the worm tooth width, changing the pressure angle, and by checking with new types of lubrications, but kept the same material till the point of improvement. The paper also showed that in case of the worm gears [33], the selection of geometric features such as central distance, transmission ratio, diameter quotient, number of teeth, should be done judiciously and selection of lubrication (polyglykol in this case) is critical for improving the load capacity and safety related to pitting, wear, tooth breakage and worm shaft defection.

The authors [34], in order to achieve the objectives of selecting the right gear material, have compared three types of gear material properties using FEA analysis. They have conducted research work on housing of the gears, and the main focus has been to find the right method of evaluation and then check whether the aluminium and cast iron, as main materials, can work for defined safety in terms of displacement. The paper gives details on the outcomes of the stress analysis using the FEA method. The output clearly shows each type of material has its own advantage in its applications.

2.5.3 Plastic and polycarbonate gear assemblies

Plastic gears [35] [36] are normally made from nylon and acetal material. The nylon material engages chemically with the moisture. The acetal co-polymers-based gears material, however, provides long-term dimensional stability as well as high fatigue and chemical resistance over a wide range of temperature variation. The thermoplastic polyester gears also provide dimensionally stable life of the gears as compared to the nylon gears. Where no lubricant is used, nylon and polyester provide good lubricity when mated with polyacetal. Liquid crystal polymers [37] give high dimensional stability and chemical resistance, plus low mold shrinkage and high accuracy. To date they have been used only for small gears under light loads, such as watch gears. Linear polyphenylene sulphides [37] have high temperature and chemical resistance and good fatigue life. They work well in highly loaded parts molded with fine details. The long fibre reinforced plastics provide good dimensional repeatability and shrinkage consistency in large parts. Moreover, their high stiffness, plus creep and impact resistance, make them suitable for gear housings. The non-crystalline plastics have found limited success for gear applications. The ABS [38] is suitable only for lightly loaded gears. Polycarbonate usually requires glass reinforcement or a solid lubricant to obtain satisfactory lubricity, chemical resistance, and fatigue properties.

Gears are made in a variety of materials and each country has some different nomenclature. Hence, there is need to understand the equivalent material nomenclature in correct context. Steel is the most common material; in contemporary literature, multiple variations of steel alloy compositions have been used to achieve objectives like weight reduction, increasing life of the gears and increasing the safety limits. Then material-specific industry is also there, for example, titanium base material is used more in aerospace etc. It was also found that the plastic gear material is gaining more ground and use of carbon nanotubes/composites [39] is changing the course of many industries in making the machines etc. The next section gives the review of the materials strength factors that influence the selection of material for specific gear applications.

2.6 Review of material strengths factors which influence the gear design

Once the understanding of the properties of solids that influence the serviceability of the gears is done, it is necessary to do the study of the material's strength for matching the application of gears for a particular requirement. The section below gives the review of concepts employed to measure material strength and its role in designing the gears, especially in the context of spur gears.

Compression: It takes place if the resultant of the outer forces on one side of the section becomes a unique perpendicular to the section, and passing by its centre of elasticity. If the resultant force (the balance of the outer and the inner forces) is imbalanced, and it goes beyond the allowable stress for traction, it would lead to more deformation of gear parts, teeth and edges. This deformation may lead to longitudinal expansion and misalignment problems. This impacts the overall life of the gear assemblies. It means more wear and tear with increase in fatigue in the material. Many researchers [40] have worked on this problem by selecting materials with higher hardness values and strength; slow-speed tests have been helpful in identifying the wear rates and for taking remedial measures like adding/changing lubricants.

Shearing [41]**:** The deformation caused by the resultant of forces situated on one side of the section is a force, and is situated in the plane of the section of a gear object section. If the resultant force leads to deformation, and it goes beyond allowable shear, it could lead to problems of misalignment, and/or reduced load capacity. By observing the stress–strain relationship, the deformation of the gear material may be predicted. The service life of the gear assembly is reduced if this continues for long cycles of load. From various current literature, it was found that this challenge is overcome by choosing materials having appropriate Young's modulus, modulus of elasticity, so that the tensile strength and yield strength are maximum as per requirement. Gear fatigue may also occur due to stiffness in material. Hence, analysis based on Hooke's Law, Bauschinger Effect, plasticity, and the non-kinematic hardening rule may be helpful in identifying the issues.

Tooth Stress and **Tooth Force:** When the normal force can be resolved into two components: tangent force, which transmit the force, and radial component which does no work but tends to push the gears apart. The resultant of these forces may lead to misalignment of pinion, and noise and vibration [42]. The machinery remains under stress and exhibits reduced smoothness in operation. Typically, the Dolan and Broghammer model is used to understand the stress on the tooth gears. The highest stress occurs at regions where lines are bunched closest together (at a contact point where the normal force acts), at the fillet region near the base of the tooth. Higher tooth stress than the tolerance limit will lead to wear and tear of tooth, leading to material remove from the tooth. Hence the service life decreases.

Bending: Is a physical phenomenon modeled using International Organization for Standardization (ISO-6336) and the Lewis equation [11]. But this model has undergone many enhancements, hence, the bending analysis should be understood in the current context of this model. By definition, the bending analysis with the help of the Lewis equation model indicates that bending stress varies directly with respect to load, and is inversely proportional to the tooth width, the tooth size and tooth shape factors. This equation was modified to overcome drawbacks. The modified equation can be used in case the fatigue failure of the teeth is not that important. It considers dynamic load, pitch line velocity, shared load at the fillet, which means it is a better way to measure bending stress. The authors [43] have worked to improve the gear design based on this research. They had considered pitch line velocity, manufacturing accuracy, contact ratio [44], stress concentration, degree of shock, rigidity and moment of inertia of helical gears as indicators of improvements. As mentioned earlier in this paragraph, the tooth bending stress equation has been modified. As this modification incorporates additional geometric factors such as application, size, form, dynamic, idler and rim thickness, it can be considered a better method to estimate and predict the bending stress. Generally, the bending stress allowance values are 10 million cycles of tooth loading at 99% reliability, and may be adjusted as per the conditions. The material allowable bending stress values come in handy to complete the full-length analysis. Most of these allowable stress values are functions of the Brinell hardness metric [45], which means this analysis works fine with thin material as well. This clearly shows that selection of material is paramount in the making of gears.

Surface Stress: A gear tooth may not break due to bending, but may become faulty due to surface stress. It might develop pits on the tooth face due to high contact stress fatiguing its surface. The contact pressure is intense at pitch circle, where the contact is pure rolling with zero sliding velocity. In such cases, lubricants may help in easing out the surface stress. This condition is modeled as a pair of cylinders in line of contact, and a Hertzian stress [19]. Larger size gears have greater contact radii of curvature and will be lower stress. So increasing the size may be considered. Hardening the tooth face increases the contact life and overall serviceability. Safety is improved, as it will not lead to chances of slippage, misalignment etc. The number of lubricants, and combination or mixture of lubricants, can help in reducing the surface stress, and again the strength of the material is important for avoiding such problems due to surface stress. A hard material with appropriate rigidity needs to be selected. It should also be noted that the composition of stress is important for analysis, especially when the design of gears is optimised for high safety standards. The gear parts may also undergo problems of buckling under stress, hence its analysis is also required in many cases as these problems lead to higher order of fatigue and reduced life of the gears.

2.7 Review of hardness tests

The selection of the most suitable material for manufacturing spur gears cannot be done unless its material passes through a hardness test. This checks the property of the material in terms of resistance to indentation. It is measured by computing the permanent depth of the material sample. The outcome (a depression) of the indentation on the material helps in measuring the hardness. As a rule of thumb, the smaller the indentation, the higher is the hardness.

It is a crucial test, especially when safety is paramount. The shape, scale, sample size are the main factors to be considered while conducting a hardness test. The next paragraph gives an overview of the methods of conducting the hardness tests. This paragraph ignores the “scratch” testing methods for measuring hardness as they are not relevant in the present context.

What is the best way to establish the fact that gearbox components will not just survive, but last for the target service period in their intended application? To answer this question, there is need for a review of the well-proven methods in contemporary literature. The manufacturing company may adopt either a mechanical or optical method to determine all the factors associated with the material strength. These methods establish the strength of the material on the basis of mean pressure per unit material. The value of strength (hardness) is of tremendous significance in determining the quality of the gears. The Durometer hardness method [46] is a method of using a predetermined test force to assess conical or spherical shaped components for a predefined timeline. It is really useful in checking the hardness of plastic and rubber sample material. If the gear material is made up of polymers, elastomers and rubbers, this test is useful. Other aspects, like gear deflection behaviour, may be determined by using models such as Young's material model and hyperelastic Marlows model etc. The Knoop Testing method [46] is referred to as the micro hardness test method and is commonly used for testing the hardness of small parts, thin sections, or case depth work. It is mentioned in ASTM E-384. The pyramid type of diamond is used for conducting hardness tests and the indenter differs from the Vickers method as its indenter is more elongated or rectangular in shape. The test indentation is very small in a Knoop test, hence may not be a useful test for spur gear design in the current context of our research work. The Vickers Test [46] is based on an optical measurement system. The micro hardness test procedure, ASTM E-384, specifies a range of light loads using a diamond indenter to make an indentation which is measured and converted to a hardness value. It is most useful for material samples that have thin sections or small parts. Typically, the loads are very light, ranging from a few grams to a few kilograms. It can be used for materials like ceramics or composites, as it is a micro hardness method. The test procedure is subject to the handling of the operator that may influence the test result. It should also be noted that gears manufactured by ceramic and metal injection means find wide application in industry. Zirconium oxide (ZrO_2), is most commonly used in the injection process, or, for example, 316L composition of iron, copper, nickel, and molybdenum (FeCr19Ni9Mo2), may be used for its combined strength and corrosion resistance. Aluminum oxide (AlO_2) is another common material that may be used in manufacturing the gears. In such cases, the Vickers test is appropriate, especially when gears/parts that are made purposely for non-magnetic applications and their hardness needs to be checked. Now, the world is moving towards gears made up of bulk metallic glass [47] (BMG) as they have the combined mechanical properties of ceramics and crystalline metals. The Brinell Test [48] is typically useful for materials that are coarse in nature, for example casting and forging. The Brinell test often uses a very high test load (3000 kgf) along with the 10mm wide indenter so that the resulting indentation averages out most surface and sub-surface inconsistencies. In the context of our research work, the gears we intend to design do not have coarse or rough surfaces or uneven thinness, However, in some conditions it can be used in conjunction with the Rockwell test. Last, but not least, the most popular test for hardness is the Rockwell test. The Rockwell measures the permanent depth of indentation produced by a force/load on an indenter. It covers preliminary test loads (preloads) ranging from 3 kgf (used in the "Superficial" Rockwell scale) to 10 kgf (used in the "Regular" Rockwell scale) to 200 kgs (used as a macro scale and not part of ASTM E-18; see ASTM E-1842). Total test forces range from 15 kgf to 150 kgf (superficial and regular) from 500 kgf to 3000 kgf (macrohardness). It is useful in all conditions and materials except where vibrations in the metal sample are high or where indentations will be too large for application. It is defined in ASTM-E18, and for most situations is an easy and accurate way of getting measures of hardness.

2.8 Review of analytical methods

In order to determine the material for shafts and gears etc, there is always a need for performing in-depth analysis. The most popular method is FEA. It helps in conducting mechanical stress, mechanical vibration, motion, and fatigue analysis for selecting the right kind of material. Most of the FEA methods may be employed for selecting the right material for manufacturing. The process will involve analysing geometry; inclusion of dissimilar material properties and capturing the local effects may require some level of fine tuning in terms of discretisation method (h-version, hp-version, x-FEM, iso-geomeric analysis). However, the FEA algorithm is the most widely used in mainstream simulation software such as ANSYS [16]. These commercial simulators can be used for conducting tests for vibration, impact, durability, strength and optimisation of gearbox design.

2.9 Improving serviceability of the gears: Other factors

The service factors are used in both the analysis and design functions. For proper understanding of the properties of solids (materials) that impact serviceability of gears it is essential to consider other factors like friction, use of lubricants etc. Hence, the section below gives information and review of the properties that influence the serviceability of gears other than the material and geometric perspectives.

Friction Analysis [49]

The use of lubrication helps to manage the frictional and stiffness forces in teeth. Higher stiffness and rigidity may lead to higher degree of frictional forces coming into play, leading to removal of surface material of the gear's tooth. Other than this, it may lead to an increase in the temperature of the machine, more noise, and finally lead to operational incapacity. For a proper design to pass through the high quality standards, it is necessary to conduct this analysis. The frictional force works not just on internal parts of the gear assembly, but also on the surface and exterior parts of the gears.

Analysis of Centre of Gravity [50]

This is critical, especially for machines having gears that are moving and are operating at certain height, like helicopters. The erratic change in load and height may lead to changes in CoG, or Center of Gravity, potential and kinetic energies, and many other dynamic properties of the machine. High variability of the load, height, position, in the case where the gears are part of a moving machine, leads to inter-play of many forces that may generate vibrations and noise, and fatigue in the material. This ultimately leads to reduced serviceability and operational safety risks, because in moving parts, not only simple forces are acting but rectilinear movement and rotation movement also impacts the overall performance of the machine gears.

Analysis of Vibrations [13]

The Vibration and Acoustic Emission Analysis [20] is critical for machines that move up and down in height, and have gears prone to lot of vibration due to multiple factors like resonance, rotating movement etc. Frequent shocks and X, Y and Z-direction type vibrations in the gear assembly will lead to lowering of service and safety limits. Then the role of shock absorbers and dampers is important for gearbox protection, and increasing the service life, especially when the terrain over which the machine is moving has lot ups and downs in position.

Analysis of Torque

Sudden changes in torque [51] of the machine having gearboxes may lead to misalignment between the motor and load, mass and gear teeth. This may lead to frequent loud noise in the machine. The imbalance created may lead to multiple issues in terms of reduced service life and constant unwanted noise and vibrations. This analysis helps in identifying the correct positions in the conditions; when the input and output gear are revolving in clockwise directions, both the reaction force and weight of idler would act downward. Then, the preferred location of idler is on the top side, if the idler is to be mounted on a moveable arm and not on a rigidly-mounted shaft. A similar condition may have been analysed when the movement of output gear is outward, but in an anti-clockwise direction. Incorrect positions of the idler may lead to reduced alignment and life of the gears.

Design Suitability

Design suitability refers to fulfilling the customer needs or functionality for a particular application. Gears are classified as inline or right-angle drives. Inline gears (spur or helical) may have input and output shafts concentric or offset. Epicyclic [52] designs are preferred where concentrated power is required, and they find application in the aero-modeling industry. Another suitability factor is the number of gears the assembly worm gear can handle; this typically works best in the ratio range of (7:1) and (100:1). From other literature sources it is clear that spur and helical gears have minimum requirement of four teeth, and work in the ratio range of (1:1) and (9:1). And there are many other practical considerations of application of the gears that guide the suitability of the gears. Quality control inspectors are required at every stage of manufacturing, and for checking the conformance of the design. In addition to production, there is always a need for a geometric inspection team. Then, there is always a need to check the material properties or metallurgical properties. Most proactive organisations have adopted methods of using Quality Circle and TQM [53] [54] (Total Quality Management), to take care of this aspect. This technique not only helps to map to the government conformance and compliance specifications, but also supports in identifying the reasons of failures of gears, cost analysis and many other critical aspects. The quality class of a gear is a code which specifies manufacturing tolerances and relates to the accuracy of the gear and mounting accuracy requirements. In addition, the dynamic load varies with the quality code, with high dynamic load being associated with low quality codes. The next section discusses the strategies for improving the gear designs.

2.10 Review of strategies for improving gear design and manufacturing

The properties of solids, material strength and cost, all play crucial roles in the final design of gears. But, as the technological advancement occurs in other industries such as aerospace, the need to improve the existing designs become necessary. Therefore, there is always a need to adopt a systematic approach to continuous improvements of the gear designs. There are many ways in which an improvement may be bought to gear design, depending on the application. But the most popular method is to use the concept of Design of Experiments [55], [56] (DoE). This section discusses the various approaches adopted by researchers for doing experimentation for achieving a better design than previous ones.

The concept of Design of Experiments (DoE) was initially introduced by Ronald A. Fisher in his work (1920-1930) for agricultural research stations. He meticulously showed how design(s) and plan(s) of various steps to improve or observe some process can help in systematically solving lots of problems – in many cases, the problems and challenges that cannot be overcome without getting to experimental mode. In gear manufacturing processes, it is often of primary interest to explore the relationship between input factors and outcomes of the performance characteristics. One of the popular methods employed is OVAT (one variable at a time) [56]. It is an improved method based on the single input factor, and has limited scope of information used for experimenting and improving the gear design. Hence, it may lead to unreliable and false optimal solutions for gear designs.

The role of statistical thinking cannot be undervalued in planning and construction of the gears. It primarily helps in explaining how variability influences a certain characteristic of the gear output. In certain cases, the role of noise is a prime concern, for example in gear design; hence, this variable is more influential in impacting the serviceability of the gears under the design process. Identification of strong and weak factors needs to done, and then be carefully selected for improvement related goals. In real-life conditions, some of the factors may become unexplained variables and give rise to unwarranted results and outcomes. This can be overcome by following meticulous planning, and not by mere randomisation of experiments.

The replicability of the experiments is crucial. Some published research work is not replicable, and its proper extension is not possible. All this is done with the help of guidelines provided by methods such as Quality Function Deployment [55] (QFD). This method allocates weights as per the priority of the customer and other engineering requirements. Initially, a correlation matrix is made for finding how much each specification influences the customer's requirements. For example, the aim of the customer may be to maximise the life of the gear's assembly, maintaining high standard of safety and minimising the cost of production. Based on this input, the engineering specifications are finalised and executed. The FAST (Functional Analysis of System Technique) diagram is another way of finding exact specifications. This method maps the functions to the goals (there may be a group of goals) of producing new gear designs. Hence, the primary task to be undertaken after understanding and documenting the customer needs is mapping these customer needs to the engineering specifications. These specifications must cover the tolerances, geometric specifications, material specifications. It also contains improvement aims, so as to avoid failures in future due to factors such as noise, misalignment, impact of natural frequencies, rattle noise, machining error, load-sharing ratio impact etc.

2.11 Conclusions

After conducting this systematic literature survey and by evaluating the inputs from the industry, it was found that there has been limited explorative research in trying new materials (such as C45 material and 19MnCr5) in the context of designing spur gears. This may be due to the fact that less experimentation is done for understanding the performance in terms of contact, bending stresses and other endurance testing of these gears. Therefore, there is a need to investigate the service life of such combinations and upcoming designs. It was also found that the finite element analysis method is most suitable for conducting service and safety analysis. Based on FEA simulation outcomes and other important factors like cost of these new materials, loading capacity and safety factors, it could be concluded that this work will be useful for industry application. Endurance stress testing and hardness testing will ensure the work is validated. The design of experiments (DoE) methodology may be used for planning the different levels of factors such as load carrying capacity of gear tooth for experimenting, testing and improving the design of spur gears. It appears that the Rockwell Test is the most appropriate method to conduct hardness testing in the context of this research work.

CHAPTER 3

LOAD CAPACITY CALCULATION OF SPUR GEAR

This chapter would start up with some basic concepts of gear and its classification, followed by discussing influencing factors, safety factors and the load carrying capacity formulas over the tooth flank (contact stress) and tooth root (bending stress). At the end of this chapter, theoretical calculation using ISO 6336 standards and result comparison in the form of tables is presented.

3.1 Gear classification

Gear can be classified in three types. (1) according to the position of the shaft axis, which are: (a) parallel axis (b) intersecting axis (c) non-intersecting and non-parallel axis, (2) according to the peripheral velocity of the gears, which are: (a) low velocity (b) medium velocity (c) high velocity, and (3) according to the type of gearing, which are: (a) external gearing (b) internal gearing (c) rack and pinion.

From the above three classifications, different type of gears are determined such as; spur gear, helical gear, double helical (herringbone gear), bevel gear, hypoid gear (hyperboloids) and worm and worm-wheel. In this study, designing, analysing, and experimenting of spur gearing systems is discussed.

Gears are not defined only by their function. Instead, they are determined by their geometry, where geometry means the shape and relative arrangements of a body. With the combination of geometry, gear becomes relevant for mechanical use. Some important technical terms used in gear geometry are: pitch circle, pitch circle diameter (PCD), pitch point, pitch surface, pressure angle or angle of obliquity, addendum, dedendum, addendum circle, dedendum circle (root circle), circular pitch (PC), module (m), clearance, clearance circle, working depth, tooth thickness, tooth spaces, backlash, top land, flank, face width, profile, fillet radius, path of contact, and arc of contact (arc of approach, arc of recess).

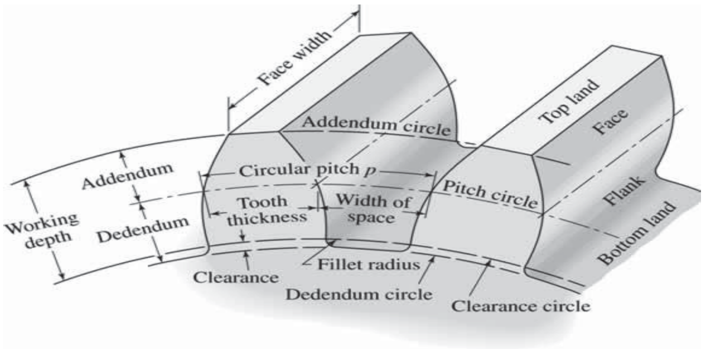


Figure 3.1 Gear micro geometry

(Fundamentals of Machine Elements, 3rd ed.Schmid, Hamrock and Jacobson)

Furthermore, some other definitions which are essential to discuss in this research are: basic rack, pressure angle, engagement of spur gear and contact ratio.

The basic rack represents the normal section of tooth in any gear-tooth system, and regulates the form or shape of the tooth as well as the various section of tooth form dimensions; namely, the module, the whole depth of the tooth, circular pitch, and the fillet radius. The rack is the foundation of a standard system of interchangeable gears.

To understand the concept of pressure angle assume that, if a tangent is drawn to the involute profile of a tooth at any point on the curve, and if a radial line is drawn through this point of tangency, connecting this point with the centre of the gear, then the acute angle included between this tangent and the radial is defined as the pressure angle at the point. It is also known as the working pressure angle. It can be different from the standard pressure angle depending on the correction factors involved and the mounting dimensions.

Every gear designer or gear related individual always mentions engagement of gears. The meaning of such statement is that, in a pair of meshing spur gears, the line of contact along the width of the gears is parallel to the gear axis and shifts its position along the tooth profile curve from top to bottom region of tooth height, or vice versa, as the engagement proceeds during action. The exact value of the contact ratio could be taken as the measure of the number of the pairs of teeth in mesh during the course of action.

During the gear meshing process, gear teeth engage along the path of action and from the starting point of contact to the end of the contact; the load is transmitted by a single tooth of the driving gear for part of the time and by two teeth the rest of the time. These two teeth are the new pair of teeth coming into action before first pair of teeth goes out of action. While contacting, they form an angle of contact which should be greater than the opposite angle at the centre, called the pitch angle. In other words, contact ratio is measured by the average number of teeth in contact during the period in which a tooth comes into and goes out of contact with the mating gear.

$$Contact\ Ratio\ =\ Length\ of\ contact/Base\ pitch$$

3.2 Introduction of safety factors in spur gear

Tooth breakage or exceeding the limit of the tooth surface durability of meshing flanks would end the service life of a gearbox. To overcome this problem, the value of the safety factor is chosen. The safety factor S_F, against tooth breakage (bending stress) should be larger than the safety factor S_H, against pitting (contact stress).

Evidently, if the performance of the gear could be achieved accurately through testing under actual load conditions, a lower safety factor and more economical manufacturing process may be allowed. Selecting the safety factor would be done after careful consideration of some influences: reliability of material data, reliability of load value used for calculation, variation in gear geometry due to manufacturing tolerances and variation in alignment. Before discussing formulas and their values, it is important to understand all the factors and their value evaluating process.

It is vital to distinguish between the safety factor relative to pitting, S_H and that relative to tooth breakage, S_F. Gear load capacity is demonstrated by the calculated values of S_H and S_F, which should be greater than or equal to the values of $S_{H\,min}$ and $S_{F\,min}$ respectively.

$$Safety\ Factor = \frac{Modified\ allowable\ stress}{Calculated\ stress}$$

Allowable stress is evaluated from a contact pressure that may be sustained for a specified number of cycles, without the occurrence of progressive pitting. The allowable stress can be calculated by the following equation: (ISO 6336-5, Equation 2, page 6):

$$\sigma_{H\ lim}, \sigma_{F\ lim} = A.x + B$$

Where,

x The surface hardness HV or HRC

A, B Constant from ISO 6336-5 page 6, table 1.

Calculated stress or contact stress is directly proportional to transmitted load. The calculation could be done by using equation 2, page 43 of this thesis.

Influence factor

Practically, the theoretical value of forces applied over the reference circle would differ from the value of the effective forces applied because of the internal and the external causes. To determine these causes and variation in the forces, influence factor is considered. In other words, the factors which influence the safety or life of the gear is known as the influence factor.

Although influence factors act independently, they nevertheless influence each other to such a degree that no numerical value could be assigned. There are several factors which influence the safety of the gear and pinion such as; Application factor ;, Internal dynamic factor K_v; Face load factors $K_{H\beta}$ and ;, Transverse load factors $K_{H\alpha}$ and $K_{F\alpha}$; Tooth stiffness parameter c' and c_γ; Zone factors, Z_H; Elasticity factor, Z_E; Contact ratio factor, Z_ϵ; Life factor, Z_{NT}; Lubricant factor, Z_L; Velocity factor, Z_V; Roughness factor, Z_R; Work hardening factor, Z_W; Size factor, Z_X; Single pair tooth contact factors, Z_B and Z_D; Form factor, Y_F; Stress correction factor, Y_S; Rim thickness factor, Y_B; Deep tooth factor, Y_{DT}; Life factor (at tooth root), Y_{NT}; Size factor, Y_X. In this thesis, all the factors are calculated with the help of ISO 6336.

Figure 3.2 shows the spur profile of the gear. The marked area over the flank and root of the gear is the investigating region. The formulas used to inspect the contact stress against the pitting, and root stress against the tooth breakage, were referred from ISO6336.

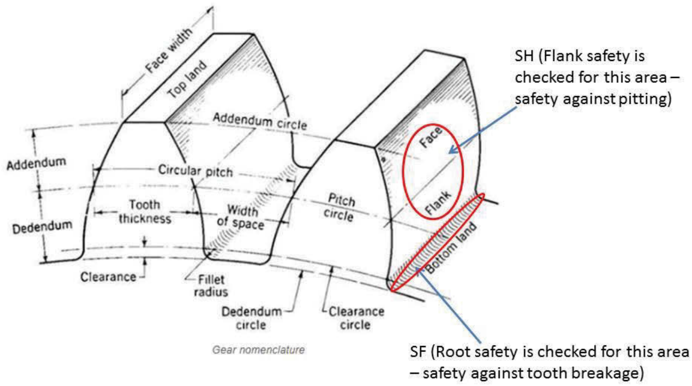


Figure 3.2: S_H over the flank and S_F over the root of the gear tooth

3.3 Calculation of surface durability or contact stress (Flank pitting)

This part of the chapter demonstrates the surface load carrying capacity calculation for a spur pinion and gear. This includes formulas for all the effects on surface durability for which numerical examination could be made. The formulas are applicable mainly to cylindrical gears which sustain sufficient lubrication (oil or grease) at all times over the teeth; for cylindrical gears obtaining tooth profile referred to basic rack discussed above in this chapter; and may also be used for teeth where the contact ratio is less than 2. The calculations of surface load capacity are dependent on the contact stress σ_H which generates at the pitch point or inner point of single pair tooth contact. Contact stress and the permissible contact stress are calculated separately for both mating gears (pinion–small gear and wheel–big gear). For gear to be safe in pitting, σ_H should be less than σ_{HP}.

3.3.1 Contact stress for the pinion (ISO-6336-2-page 3)

To measure contact stress in spur gear, the following formulas are used:

$$\sigma_H = Z_B \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \leq \sigma_{HP} \quad (1)$$

$$\sigma_{HO} = Z_H Z_E Z_\epsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{u+1}{u}} \quad (2)$$

Where,

σ_{HO} the nominal contact stress

Z_B the pinion single pair tooth contact factor (explained further in this chapter)

K_A the application factor

K_V the dynamic factor

$K_{H\beta}$ the face load factor for contact stress

$K_{H\alpha}$ the transverse load factor for contact stress

σ_{HP} the permissible contact stress

Z_H the zone factor

Z_E the elasticity factor

Z_ϵ the contact ratio factor

Z_β the helix angle factor for helical gear set.

F_t the nominal tangential load

b the face width

d_1 the reference diameter of pinion

u the gear ratio = Z_2/Z_1

3.3.2 Contact stress for the wheel

To measure the contact stress of wheel, the same formulas above are used, with one change.

$$\sigma_H = Z_D \sigma_{HO} \sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \leq \sigma_{HP} \tag{3}$$

Where

Z_D the single pair tooth contact factor of the wheel.

The value of Z_D varies, but all other factors remain the same, as mentioned above in stress calculation for the pinion. In addition, to calculate permissible contact stress, σ_{HP}, the following formula is used:

$$\sigma_{HP} = \frac{\sigma_{H\,lim} Z_{NT}}{S_{H\,min}} Z_L Z_V Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H\,min}} \tag{4}$$

Where,

$\sigma_{H\,lim}$ the allowable stress number

Z_{NT} the life factor for contact stress

σ_{HG} the pitting stress limit (= $\sigma_{HP} S_{H\,min}$)

$S_{H\,min}$ the minimum safety factor needed for surface durability

Z_L, Z_R and Z_V the effect of the oil film over the tooth contact stress

Z_L the lubricant factor

Z_R the roughness factor

Z_V the velocity factor

Z_w the work hardening factor

Z_x the size factor for contact stress

Also, while computing the safety factor of the spur gear for surface durability specifically against pitting, S_H, then the following formula is used. Also, the diagram below shows which area shall be investigated.

$$S_H = \frac{\sigma_{HG}}{\sigma_H} > S_{H\,min} \tag{5}$$

All the values, such as torque, speed and power are considered from a tractor engine. Also, gear and pinion from a tractor gearbox is used for analysis because of noticeable amount of stress variation and the gearbox faces harsh conditions on each gear and pinion in the field. If a spur-gear pair, designed with 19MnCr5 material exceeds the minimum safety factor, then the same gear material combination could be recommended for industrial use.

Using all the formulas mentioned above, Table 3.1 is fabricated to show the results of the load carrying capacity or contact stress over the flank of C45 (nitride hardened) spur gear and pinion.

Table 3.1: Safety against the pitting of the flank of both, pinion, and gear (C45)

Teeth Geometry and Material	Pinion Driver	Wheel Driven	Units
No. of teeth (z)	26	28	
Gear ratio (u)	0.928		
Module	3		mm
Pressure angle (αn)	20		°
Helix angle (β)	0		
Pitch circle diameter (mm)	78	84	mm
Face width (d)	17.8	16	mm
Center distance (mm)	83.53		mm
Gear material	C-45	C-45	
Heat treatment	Nitrided	Nitrided	

Surface hardness	500/50	500/50	HV/HRC
Accuracy grade	8	8	

Note - All units are in mm unless otherwise specified

Load Parameter	Pinion Driver	Wheel Driven	Units
Power (Kw)	71		
Speed (RPM)	2919	2711	
Torque (Nm)	232.24	250.1	
Application factor (K_A)	1		
Required service life	20,000		Hours

Results	Pinion Driver	Wheel Driven	Units
Nominal tangebtial force at PCD	5953.97	5953.86	N
Fatigue strenght for Hertzian pressure $\sigma_{H\,lim}$	1000	1000	$\mathrm{N/mm^2}$
Nominal contact stress σ_{HO}	1150.13	1188.72	$\mathrm{N/mm^2}$
Contact stress σ_H	1290	1313.18	$\mathrm{N/mm^2}$
Permissible contact stress σ_{HP}	736.54	733.92	$\mathrm{N/mm^2}$
Limit strength pitting σ_{HG}	736.54	733.92	$\mathrm{N/mm^2}$
Safety for stress at single tooth S_H	0.58	0.56	
Min. required safety $S_{H\,min}$	1.00	1.00	
Status	FAIL	FAIL	

Influence Factor	Pinion Driver	Wheel Driven
Dynamic factor K_V	1.05	1.05
Face load factor $K_{H\beta}$	1.15	1.15
Transverse load factor $K_{H\alpha}$	1	1
Profile shift	0.43	0.43
Tooth form factor	1.45	1.48
Stress correction factor	2.02	2.01
Deep tooth factor	1	1
Zone factor Z_H	2.26	2.24
Elasticity coefficient factor Z_E	189.81	189.81
Contact ratio factor Z_ϵ	0.919	0.922
Helix angle factor Z_β	1	1
Lubricant coefficient factor	0.896	0.896
Speed coefficient	1.008	1.008
Roughness coefficient Z_R	0.8175	0.813
Work hardening factor Z_W	1	1
Size factor Z_X	1	1

Table 3.1 shows the gear geometry, load parameter, influence factors, followed by results. The geometry of the gear is given, such as number of teeth, module etc. The load parameters, such as torque, power, speed etc, are obtained from the tractor engine. Also, the influence factors, such as dynamic factor, face load factor are gathered by using formulas discussed earlier. Major concern is on the fatigue strength for Hertzian pressure $\sigma_{H\ lim}$, which is calculated from ISO 6336-5 using the formula:

$$\sigma_{H\ lim} = A.x + B$$

Where,

x the surface hardness HV or HRC

A, B constant from ISO 6336-5 page 6, table 1.

The value of $\sigma_{H\,lim}$ varies with change in material and this value is essential in this calculation. With the help of formulas, results are consolidated. The minimum safety factor, $S_{H\,min}$ for C45 spur-gear is set to be 1, but calculation result shows that the safety factor at the flank is 0.58. Such safety factor is leading towards failure of the gear on the flank against pitting. So that means the gear tooth flank would show the pitting failure on gears manufactured with C45 material, and would not work for a longer period of time.

To rectify the safety factor issue, material changing technique is proposed in this thesis. The 19MnCr5 material is examined under the same load parameters and with identical gear geometry. Table 3.2 is constructed to display 19MnCr5 gear material results.

Table 3.2. Safety against the pitting of the flank of both, pinion, and gear **(19MnCr5)**

Teeth Geometry and Material	Pinion Driver	Wheel Driven	Units
No. of teeth (z)	26	28	
Gear ratio (u)	0.928		
Module	3		mm
Pressure angle (αn)	20		°
Helix angle (β)	0		
Pitch circle diameter	78	84	mm
Face width (d)	17.8	16	mm
Center distance	83.53		mm
Gear material	19MnCr5	19MnCr5	
Heat treatment	Case-hardened	Case-hardened	
Surface hardness	697/60	697/60	HV/HRC
Accuracy grade	8	8	

Note - All units are in $\boldsymbol{mm}$ unless otherwise specified

Load Parameter	Pinion Driver	Wheel Driven	Units
Power	71		KW
Speed	2919	2711	RPM
Torque	232.24	250.1	Nm
Application factor (K_A)	1		
Required service life	20,000		Hours

Results	Pinion Driver	Wheel Driven	Units
Nominal force at PCD (F_t)	5953.97	5953.86	N
Fatigue strenght for Hertzian pressure ($\sigma_{H\,lim}$)	1500	1500	$\mathrm{N/mm^2}$
Nominal contact stress (σ_{HO})	1150.13	1188.72	$\mathrm{N/mm^2}$
Contact stress (σ_H)	1290	1313.18	$\mathrm{N/mm^2}$
Permissible contact stress (σ_{HP})	1348.72	1346.62	$\mathrm{N/mm^2}$
Limit strength pitting (σ_{HG})	1348.72	1346.62	$\mathrm{N/mm^2}$
Safety for stress at single tooth (S_H)	1.07	1.03	
Min. required safety ($S_{H\,min}$)	1.00	1.00	
Status	PASS	PASS	

Influence Factor	Pinion Driver	Wheel Driven
Dynamic factor K_V	1.05	1.05
Face load factor $K_{H\beta}$	1.15	1.15
Transverse load factor $K_{H\alpha}$	1	1
Profile shift	0.43	0.43
Tooth form factor	1.45	1.48
Stress correction factor	2.02	2.01
Deep tooth factor	1	1
Zone factor Z_H	2.26	2.24
Elasticity coefficient factor Z_E	189.81	189.81
Contact ratio factor Z_ϵ	0.919	0.922
Helix angle factor Z_β	1	1
Lubricant coefficient factor	0.896	0.896
Speed coefficient	1.008	1.008
Roughness coefficient Z_R	0.8175	0.813
Work hardening factor Z_W	1	1
Size factor Z_X	1	1

In Table 3.2, the minimum safety factor required, $S_{H\,min}$, is 1, and according to calculation after changing gear material, safety factor on pinion and gear is recorded as 1.07 and 1.03 respectively. That means that this gear material will not show pitting failure on the gear flank before estimated service life.

Furthermore, to compare C45 and 19MnCr5 gear material against the tooth breakage, tooth bending strength is calculated. To show the result, computation for tooth stresses has been done later in this thesis.

3.4 Calculation of tooth bending strength

This part of the thesis would determine the load carrying capacity over the tooth root on the basis of permissible bending stress which is known as tooth bending strength. The fracture over the gear tooth in most of the cases is initiated at the root diameter of the teeth. Breakage of tooth would end the service life of the entire transmission system. Sometimes, the path of transmission between output and input gear is damaged, and because of this, the value of safety factor, S_F, against tooth breakage should be greater than the safety factor against pitting, S_H. The formulas used in this section were referred from ISO 6336-3, page 2.

The tooth root stress, σ_F, and the permissible bending stress (tooth root), σ_{FP}, should satisfy one condition to pass the gear successfully, that is, σ_F should be less than σ_{FP}.

3.4.1 Safety factor for bending strength (safety against tooth breakage), $\boldsymbol{S_F}$

$$S_{F1} = \frac{\sigma_{FG1}}{\sigma_{F1}} \geq S_{F\,min} \text{ (Pinion)} \tag{6}$$

$$S_{F2} = \frac{\sigma_{FG2}}{\sigma_{F2}} \geq S_{F\,min} \text{ (Wheel)} \tag{7}$$

Where,

σ_{FG} tooth root stress limit

Basically, the maximum tensile stress over the surface in the root is defined as the tooth root stress, σ_F. To calculate the root stress, the following formula will be used:

$$\sigma_F = \sigma_{FO} K_A K_V K_{F\beta} K_{F\alpha} \tag{8}$$

$$\sigma_{FO} = \frac{F_t}{b\, m_n} Y_F Y_S Y_\beta Y_B Y_{DT} \tag{9}$$

Where,

σ_{FO} the nominal root stress. This stress is free from pre-stress such as shrink fitting

σ_{FP} the permissible bending stress

K_A the application factor

K_V the dynamic factor

$K_{F\beta}$ the face load factor for tooth root stress. This would consider the uneven load distributed over the face-width

$K_{F\alpha}$ the transverse load factor for tooth root stress

F_t the nominal tangential load

b the face width

m_n the normal module

Y_F the form factor

Y_β the helix angle factor (in this case, the value is 1)

Y_B the rim thickness factor

Y_{DT} the deep tooth factor.

3.4.2 Permissible bending stress, $\boldsymbol{\sigma_{FP}}$

The allowable value of the tooth root stresses, which could be investigated by some equations, are:

$$\sigma_{FP} = \frac{\sigma_{F\lim} Y_{ST} Y_{NT}}{S_{F\,min}} Y_{\delta\,rel\,T} Y_{R\,rel\,T} Y_X = \frac{\sigma_{FE} Y_{NT}}{S_{F\,min}} Y_{\delta\,rel\,T} Y_{R\,rel\,T} Y_X = \frac{\sigma_{FG}}{S_{F\,min}} \tag{10}$$

Where,

$\sigma_{F\lim}$ the nominal stress value, which is bending stress limit value influence to the material, heat treatment and surface roughness

σ_{FE} the allowable stress = $\sigma_{F\lim} Y_{ST}$

Y_{ST} the stress correction factor

Y_{NT} the life factor for tooth root stress

σ_{FG} the tooth root stress limit = $\sigma_{FP} S_{F\,min}$

$S_{F\,min}$ the minimum required safety factor for tooth root stress.

$Y_{\delta\,rel\,T}$ the relative notch sensitivity factor

$Y_{R\,rel\,T}$ the relative surface factor

Y_X the size factor

Using the above formulas, Table 3.3 is formed to illustrate the stresses generated on the root of the gear tooth. Also, comparison of root stresses would be discussed between C45 gear and 19MnCr5 gear.

Table 3.3: Safety against bending of the teeth root of both pinion and gear (C45).

Teeth Geometry and Material	Pinion Driver	Wheel Driven	Units
No. of teeth (z)	26	28	
Gear ratio (u)	1.077		
Module	3		mm
Pressure angle (αn)	20		°
Helix angle (β)	0		mm
Pitch circle diameter (mm)	78	84	mm
Face width (d)	17.8	16	mm
Center distance	83.53		mm
Gear material	C45	C45	
Heat treatment	Nitrided	Nitrided	
Surface hardness	500/50	500/50	HV/HRC
Accuracy grade	8	8	

Note - All units are in mm unless otherwise specified

Load Parameter	Pinion Driver	Wheel Driven	Units
Power (KW)	71		KW
Speed (RPM)	2919	2711	RPM
Torque (Nm)	232.24	250.1	Nm
Application Factor, K_A	1		
Required Service Life	20,000		Hours

Results	Pinion Driver	Wheel Driven	Units
Nominal force at PCD (F_t)	5953.97	5953.86	N
Tooth root stress (σ_F)	381.54	425.13	$\mathrm{N/mm^2}$

Material fatigue strength (at root) ($\sigma_{F\,lim}$)	370	370	$\mathrm{N/mm^2}$
Limit strength tooth root (σ_{FG})	507.72	726.04	$\mathrm{N/mm^2}$
Permissible tooth root stress ($\sigma_{FP} = \sigma_{FG}/S_{F\,min}$)	362.66	518.60	$\mathrm{N/mm^2}$
Tooth root safety (S_F)	1.33	1.71	
Min. required safety ($S_{F\,min}$)	1.4	1.4	
Status	FAIL	PASS	

Note - All units are in mm unless otherwise specified

Influence Factor	Pinion Driver	Wheel Driven
Dynamic factor K_V	1.05	1.05
Face load factor $K_{H\beta}$	1.15	1.15
Transverse load factor $K_{H\alpha}$	1	1
Profile shift	0.43	0.43
Tooth form factor	1.45	1.48
Stress correction factor	2.02	2.01
Deep tooth factor	1	1
Zone factor Z_H	2.26	2.24
Elasticity coefficient factor Z_E	189.81	189.81
Contact ratio factor Z_ϵ	0.919	0.922
Helix angle factor Z_β	1	1
Lubricant coefficient factor	0.896	0.896
Speed coefficient	1.008	1.008
Roughness coefficient Z_R	0.8175	0.813
Work hardening factor Z_W	1	1
Size factor Z_X	1	1

The above Table 3.3, illustrates the safety against the bending of the teeth of both the pinion and the gear with C45 material. The minimum required safety at root of the tooth is 1.4. According to the result, pinion safety factor value is 1.33, which is unacceptable for the gear. But, at the same time, a gear with 28 teeth in same gearing system shows the acceptable safety factor value, that is, 1.71. Failure in one gear could damage the adjacent gears so, use of one safe gear against a failing gear is not recommended. Like contact stress in Table 3.2, used to solve the root failure problem, material change is suggested. Table 3.4 is structured to illustrate the safety against root bending of the gear with 19MnCr5 material.

Table 3.4: Safety against bending of the teeth toot of both, pinion, and gear (19MnCr5).

Teeth Geometry and Material	Pinion Driver	Wheel Driven	Units
No. of teeth (z)	26	28	
Gear ratio(u)	1.077		
Module	3		mm
Pressure angle (αn)	20		°
Helix angle (β)	0		
Pitch circle diameter	78	84	mm
Face width (d)	17.8	16	mm
Center distance	83.53		mm
Gear material	19MnCr5	19MnCr5	
Heat treatment	Case hardened	Case hardened	
Surface hardness	697/60	697/60	HV/HRC
Accuracy grade	8	8	

Load Parameter	Pinion Driver	Wheel Driven	Units
Power (KW)	71		
Speed (RPM)	2919	2711	
Torque (Nm)	232.24	250.1	Nm

Application Factor K_A	1	
Required Service Life	20,000	Hours

Results	Pinion Driver	Wheel Driven	Units
Nominal force at PCD (F_t)	5953.97	5953.86	N
Tooth root stress - σ_F	381.54	425.13	$\mathrm{N/mm^2}$
Material fatigue strength (at root) $\sigma_{F\,lim}$	430	430	$\mathrm{N/mm^2}$
Limit strength tooth root - σ_{FG}	575.21	821.90	$\mathrm{N/mm^2}$
Permissible tooth root stress $\sigma_{FP} = \sigma_{FG}/S_{F\,min}$	410.86	587.07	$\mathrm{N/mm^2}$
Tooth root safety S_F	1.53	1.93	
Min. required safety $S_{F\,min}$	1.4	1.4	
Status	PASS	PASS	

Note - All units are in mm unless otherwise specified

Influence Factor	Pinion Driver	Wheel Driven
Dynamic factor K_V	1.05	1.05
Face load factor $K_{H\beta}$	1.15	1.15
Transverse load factor $K_{H\alpha}$	1	1
Profile shift	0.43	0.43
Tooth form factor	1.45	1.48
Stress correction factor	2.02	2.01
Deep tooth factor	1	1
Zone factor Z_H	2.26	2.24
Elasticity coefficient factor Z_E	189.81	189.81
Contact ratio factor Z_ϵ	0.919	0.922
Helix angle factor Z_β	1	1
Lubricant coefficient factor	0.896	0.896
Speed coefficient	1.008	1.008
Roughness coefficient Z_R	0.8175	0.813
Work hardening factor Z_W	1	1
Size factor Z_X	1	1

Table 3.4 shows that the influence factor has major impacts on the safety of the gear and pinion. Safety factor at tooth root is 1.53 and 1.93 of pinion and gear, respectively. The minimum safety value for tooth root is 1.4. After consolidating theoretical data, it is clear that the pinion and gear will be safe and give maximum service life with 19MnCr5 material.

3.5 Conclusion and findings

Theoretical comparison of C45 and 19MnCr5 material has generated very interesting conclusions and contributed to the finding of the research. First, the difference in mechanical properties such as fatigue strength leads to new area of research. When compared with 19MnCr5 material, the fatigue strength of C45 material at flank and root of the gear and pinion is found to be lower, by 15% and 40%, respectively. This has major implications for the safety factor of the gear and the pinion.

Second, due to switching of the materials, there would be not much difference in the cost of the gear manufacturing. All the physical and machining environments remain the same while comparing 19MnCr5 and C45 gear manufacturing processes. The only change is in the alloy combination of magnesium and chromium to form 19MnCr5 material.

Finally, after careful consideration, one important fact has been verified. Every point over the tooth of the pinion or gear has different stress distribution. This variation of stress over the tooth made analysing the cause of failure more challenging. In this thesis, only flank area and root area of pinion and gear is investigated. Furthermore, it would be interesting to compare software simulation results with mathematical results.

CHAPTER 4

FINITE ELEMENT ANALYSIS OF SPUR GEAR

This chapter discuss the comparison between two gear materials with the help of FEA. Initially, a brief introduction of finite element analysis is presented. This is followed by an analysis of spur gear with combination of two software packages. In addition, FEA over the flank and root of the gear and pinion with C45 and 19MnCr5 material is performed. In the end, the conclusion and findings of the chapter are discussed.

4.1 Brief introduction to finite element analysis

Finite element analysis was originally introduced by Turner et al, in 1956. This method proved to be most powerful computational technique to calculate approximate solutions to several practical engineering issues containing critical domains subjected to general boundary conditions.

The FEA method requires the following steps:

Discretisation of the unknown boundary into a finite number of subdomains.

- Selection of nodes.
- Constructing an element matrix for each subdomain.
- Assembly of each element matrix to obtain the global matrix for the entire domain.
- Implementing the boundary conditions.

4.2 Finite element analysis of spur gear.

Before preforming an FEA test on the ANSYS 16.2, the engineer should be acquiring answers to the following questions:

- What is the aim and objective of the analysis?
- Should the whole system be modeled, or just a small portion would be enough?
- How much detail should be included?
- How detailed the refinement of finite element mesh would be?

Although, ANSYS 16.2 is designed to perform a wide variety of simulation in almost every engineering discipline, this thesis discusses only the structural analysis. The structural analysis includes deformation, stress and strain fields, also reaction forces in a solid body. In addition, this analysis deals with a number of structural problems, such as static analysis, modal analysis, harmonic analysis, transient dynamic and eigenvalue buckling.

In this thesis, after discussing theoretical approaches to solve the mathematical problem in Chapter 3, spur pinion and gear are 3D-modeled using SolidWorks 2016. and finite element analysis is preformed over the mating teeth of the gears using ANSYS 16.2 software. The structural analysis is done over the flank and root of the spur pinion and gear tooth. The main purpose is to study the root stress and stress at the flank of the spur pinion and gear with new material. First, pinion and gear of C45 material is analysed. and then the simulation result is compared with second set of spur pinion and gear having identical geometry, but different gear material (19MnCr5). The comparison exhibits almost the same value as the mathematical calculation performed in Chapter 3. After comparison, the conclusion of the chapter is discussed.

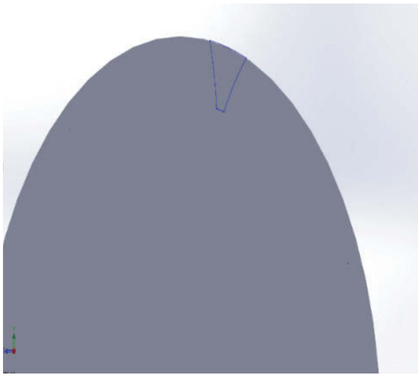


Figure 4.1 : Spur gear teeth formation

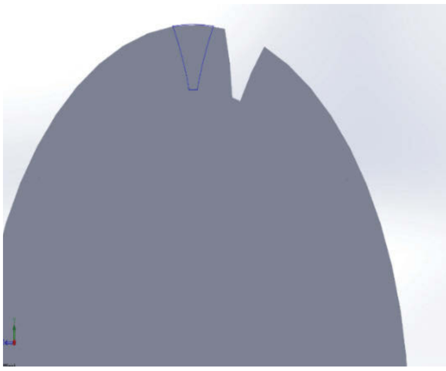


Figure 4.2: Spur gear teeth formation

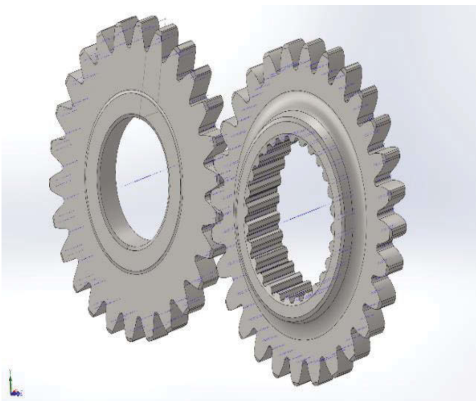


Figure 4.3: C-45 spur gear assembly

Figure 4.4: 19MnCr5 spur gear assembly

Figure 4.1 shows the involute profile curve on a cylindrical shaped base. This curve is passing through pitch circle diameter, base diameter, and tip diameter. The cylindrical base is used to draw pitch diameter, root form diameter, root diameter and operating pitch circle diameter. Following this, Figure 4.2 explains how 26 teeth in pinion or driving, and 28 teeth in gear or driven, are created. Furthermore, Figures 4.3 and 4.4 present the spur gear assembly with C45 and 19MnCr5 materials, respectively.

Applying material to a 3-D design model in SolidWorks 2016 is a simple task. Figure 4.5 shows the properties of C45 gear material. In this case, C45 material is selected which is in SolidWorks DIN materials-DIN steel (nitriding alloy)-C45G. All the material properties are discussed in the figures, such as elastic modulus, Poisson's ratio, shear modulus. Also, the units are in $\mathrm{N/mm^2}$, $\mathrm{Kg/m^3}$,$\mathrm{W/(m.K)}$ and $\mathrm{J/(Kg.k)}$. The appearance of the C45 material is polished steel. The same steps are taken to assign 19MnCr5 material to a spur gear.

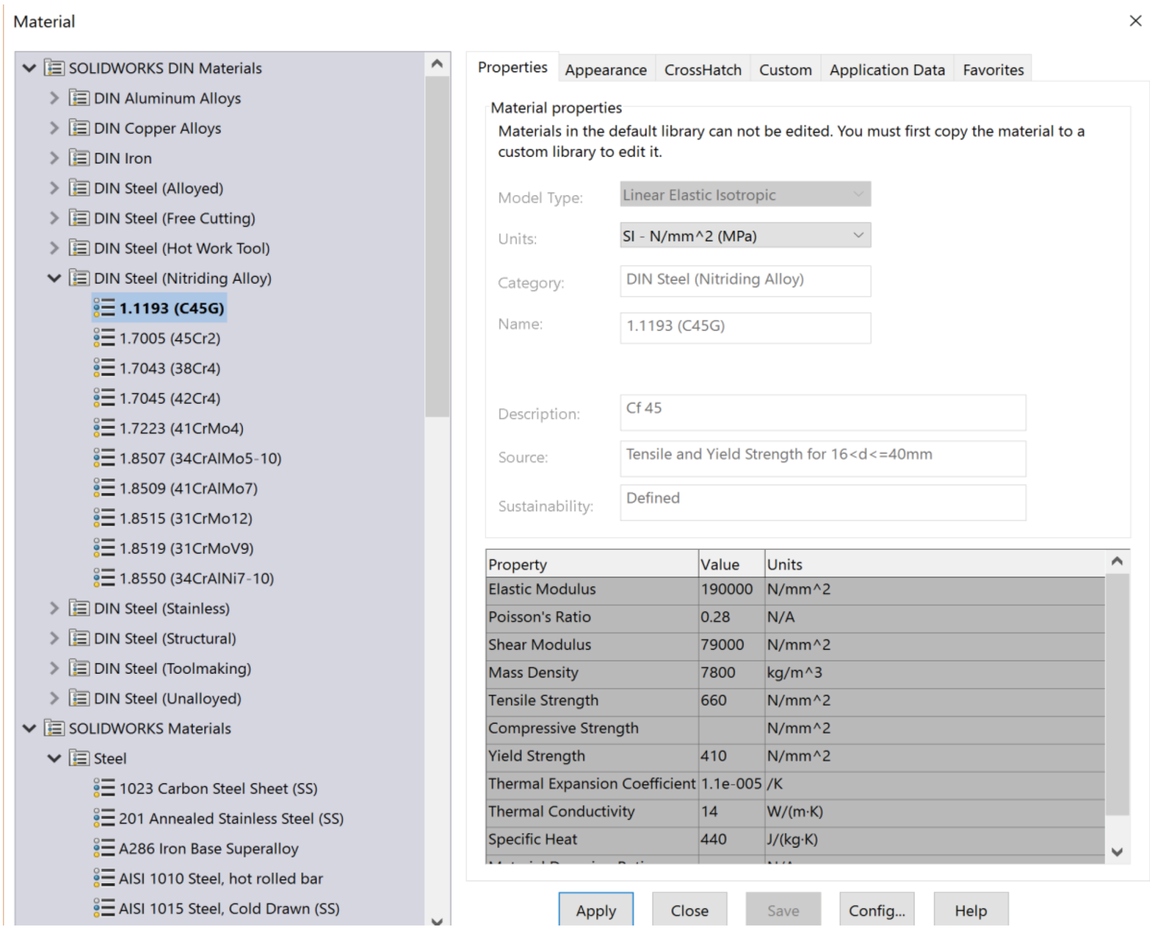


Figure 4.5: Applying material to the spur gear

Table 4.1 shows all the major parameters used to design a spur gear. The dimensions, such as number of gears, face-width, pitch circle diameter, are calculated with the help of basic rack, which is discussed in Chapter 3.

Table 4.1 Teeth Geometry of spur gear (C45)

Teeth geometry	Pinion Driving	Gear Driven	Units
Number of teeth	26	28	mm
Module	3	3	mm
Pressure angle	20	20	°
Face width	17.80	16	mm
Type of gear	Spur gear	Spur gear	
Helix angle	0	0	°
Surface Hardness	500	500	HV
Gear ratio	1.07	1.07	μ
Working pressure angle at normal section	23.953	24.324	°
Contact ratio	1.46	1.46	
Pitch circle diameter	78	84	mm
Base diameter	73.296	78.934	mm
Tip diameter	86.588	92.574	mm
Operating pitch diameter	80.203	86.624	mm

Root diameter	72.188	78.174	mm
Active root diameter	76.094	82.137	mm
Addendum	4.294	4.287	mm
Dedendum	2.906	2.913	
Tooth height	7.2	7.2	mm
Pitch on reference circle	9.425	9.425	mm
Base pitch	8.856	8.856	mm
Length of path of contact	8.856	8.856	mm
Root form diameter	74.656	80.579	mm

In Table 4.1a, both the materials are discussed to compare their strength, stress, Poisson ratio etc. The hardness of the 19MnCr5 gear is measured in HRC units, which is then converted into HV units. The practical test is performed to evaluate the hardness of the material, which is discussed in Chapter 5. In general, most of the values from Table 4.1a are the natural values of the material. The values such as tooth root stress, $\sigma_{F\ lim,}$ and Hertzian pressure, $\sigma_{H\,\mathrm{lim}}$, are gathered from the International Organization for Standardization (ISO) 6336-5. Also, life factor, Z_{NT} and Y_{NT} is taken from the standards. To understand the derivation for life factor, ISO 6336-3, page 21 is referred.

Moreover, CrMn-alloyed case-hardening steel is highly recommended for components in mechanical and automotive engineering, with relatively high core strength in elements such as camshaft and gear wheels. This encourages more research in that area.
After conducting the survey on 19MnCr5 material, the major advantage noticed is that the material with the right hardening process could provide improved load capacity and safety related to pitting, wear, tooth breakage and worm shaft defection, compared to C45 material. There is not much difference in the cost of both materials.

Table 4.1a: Material properties comparison

Material	C45	19MnCr5	Unit
Surface hardness	50 = 500	60 = 697	HV/HRC
Life factor Z_{NT} and Y_{NT}	1	1	
Fatigue strength: tooth root stress $\sigma_{F\ lim}$	370	430	$\mathrm{N/mm^2}$
Fatigue strength for Hertzian pressure $\sigma_{H\ lim}$	1000	1500	$\mathrm{N/mm^2}$
Tensile strength	660-700	1200	$\mathrm{N/mm^2}$
Yield strength	410	850	$\mathrm{N/mm^2}$
Poisson ratio	0.3	0.3	

As specified earlier in the thesis, the power, torque, and forces are considered from a tractor engine, and spur pinion and gear is chosen from the same tractor gearbox. All stress related values are mentioned in Chapter 3 in detail. By using the torque and force from Tables 3.1 and 3.2 into the 3-D designed model, which is imported in ANSYS 16.2 simulation software from SolidWorks 2016 software, the finite element analysis has begun.

4.2.1 Contact stresses of spur pinion and gear with C45 material (tooth flank)

This part of the chapter reflects the comparison between stress distribution over the flank (against pitting) of C45 spur pinion and gear, and 19MnCr5 spur pinion and gear. Attaching geometry to static structural tab in ANSYS 16.2, defining the contact region between two involute gears is vital. The contact between the two teeth is assumed to be frictionless. Importantly, the interface treatment should be changed to “adjust to touch”.

Scope	
Scoping Method	Geometry Selection
Contact	1 Face
Target	1 Face
Contact Bodies	Part 2
Target Bodies	Part 1
Definition	
Type	Frictionless
Scope Mode	Manual
Behavior	Program Controlled
Trim Contact	Program Controlled
Suppressed	No
Advanced	
Geometric Modification	
Interface Treatment	Adjust to Touch
Contact Geometry Correction	None
Target Geometry Correction	None

Figure 4.6: contact region

Assembly discretisation requires complex meshing. Meshing with the default setting does not guarantee accurate results. For current assembly, tetrahedral element is utilised. Fine meshing command is selected in this analysis. from the sizing command. At the bending surface of pinion and gear (at tooth root), refinement is done to achieve finer mesh and continuous stress value. Figure 4.7 shows the meshing of spur pinion and gear. Left side body is pinion and right hand body is gear.

Following with the supports and loads, the pinion and gear pairs are given frictionless support. Also, the pinion rotates in a clockwise direction. The pinion provides tangential force, F_t, to the gear. The value of force is considered from Chapter 3. Figure 4.8 illustrates the boundary condition and force direction, respectively.

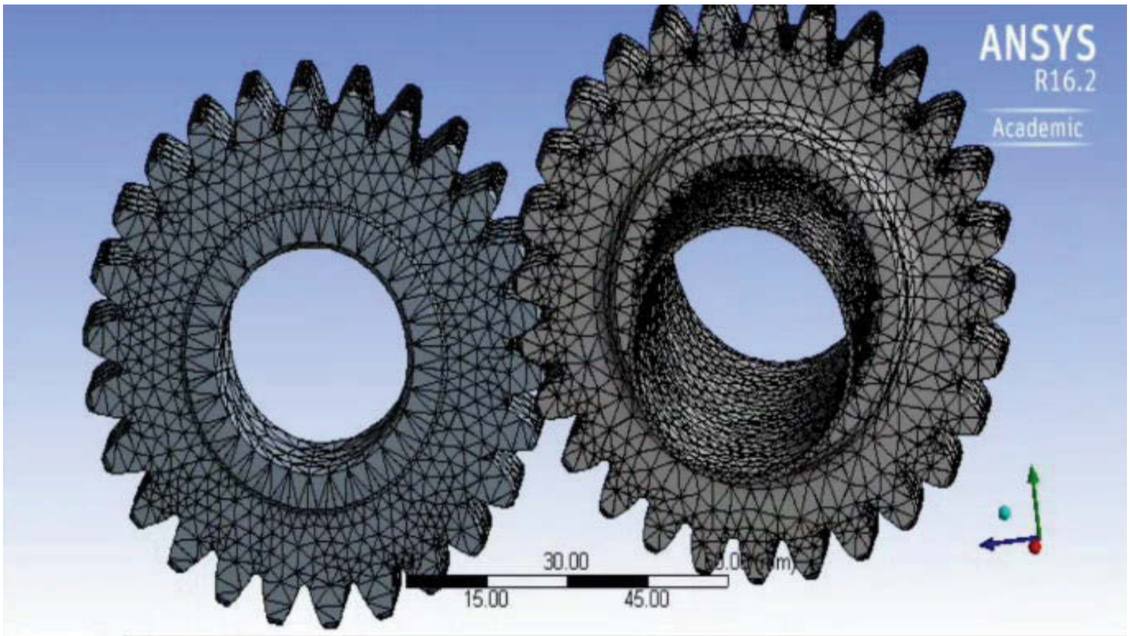


Figure 4.7: Meshing of spur-gear pair

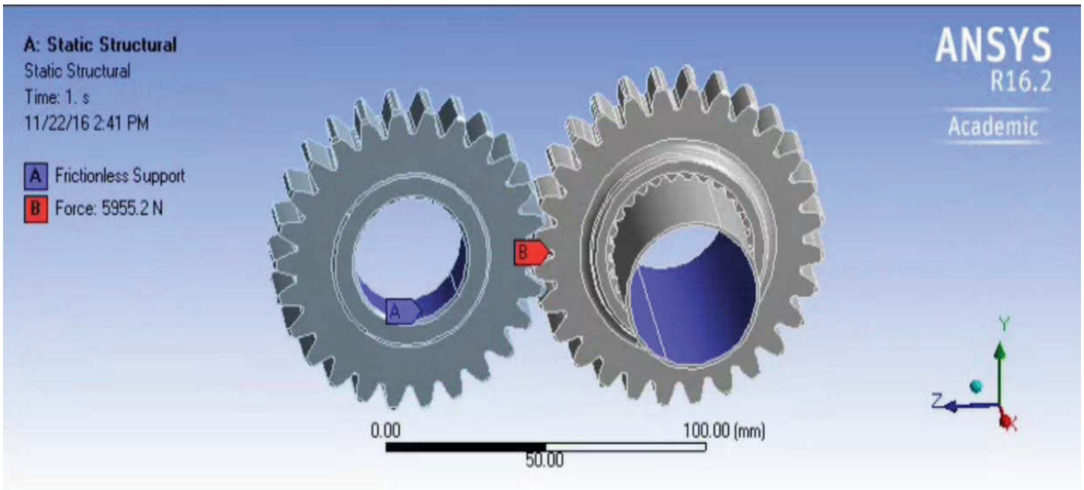


Figure 4.8: Boundary condition

Under the above boundary conditions, the equivalent stress (von Mises) and stress tool (safety factor) solution is conducted. The analysis result over the flank of the spur pinion and gear is determined in Figure 4.9. In other words, the contact stress equation used in Chapter 3 has been proven in this section.

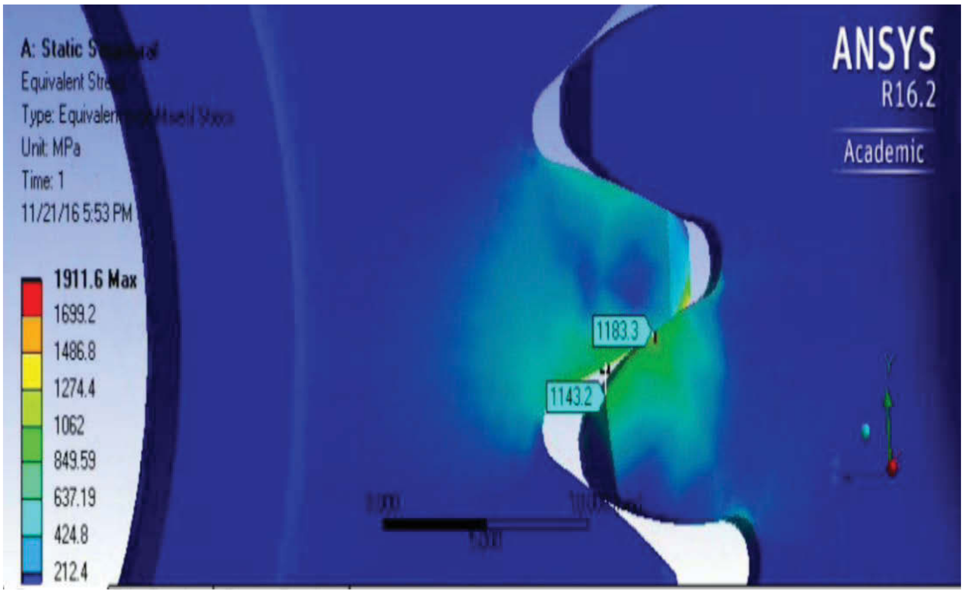


Figure 4.9: Three dimension von Mises contact stress

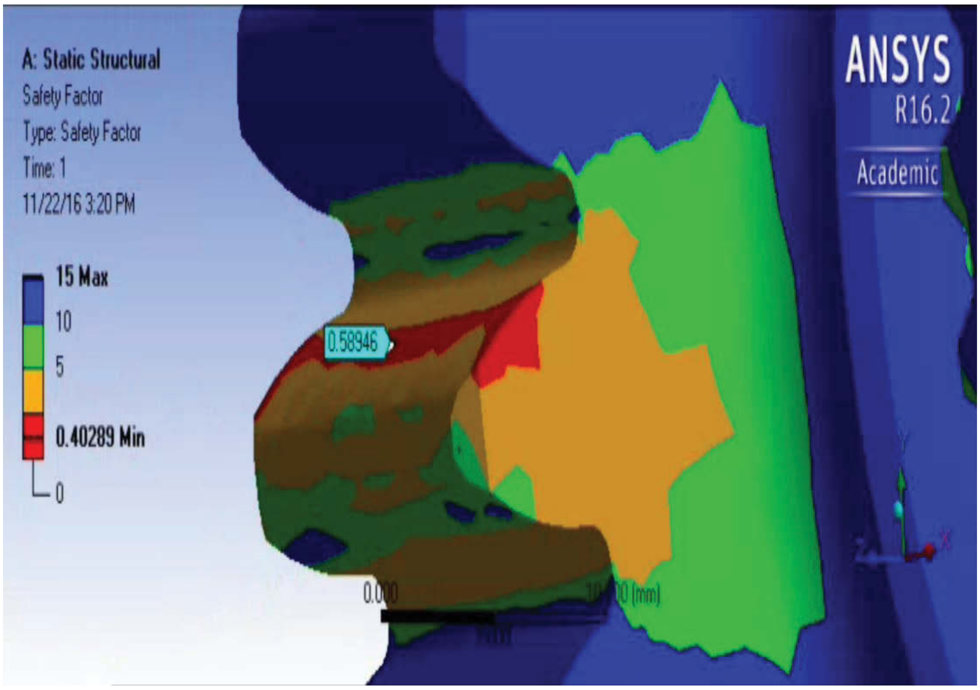


Figure 4.10: Gear safety (against pitting)

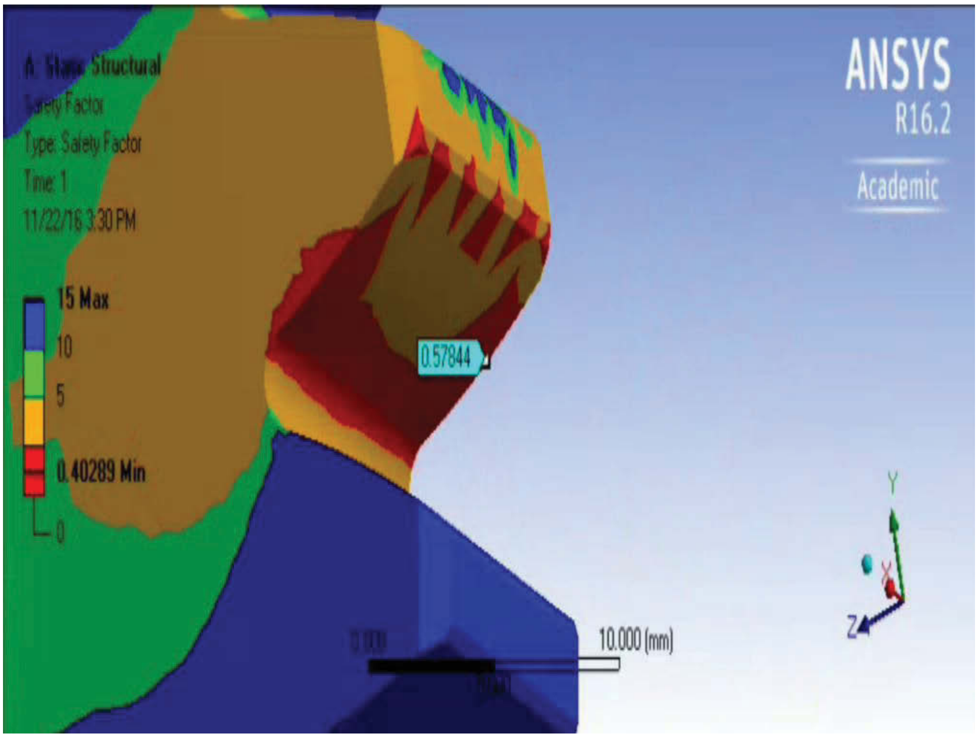


Figure 4.11: Pinion safety factor

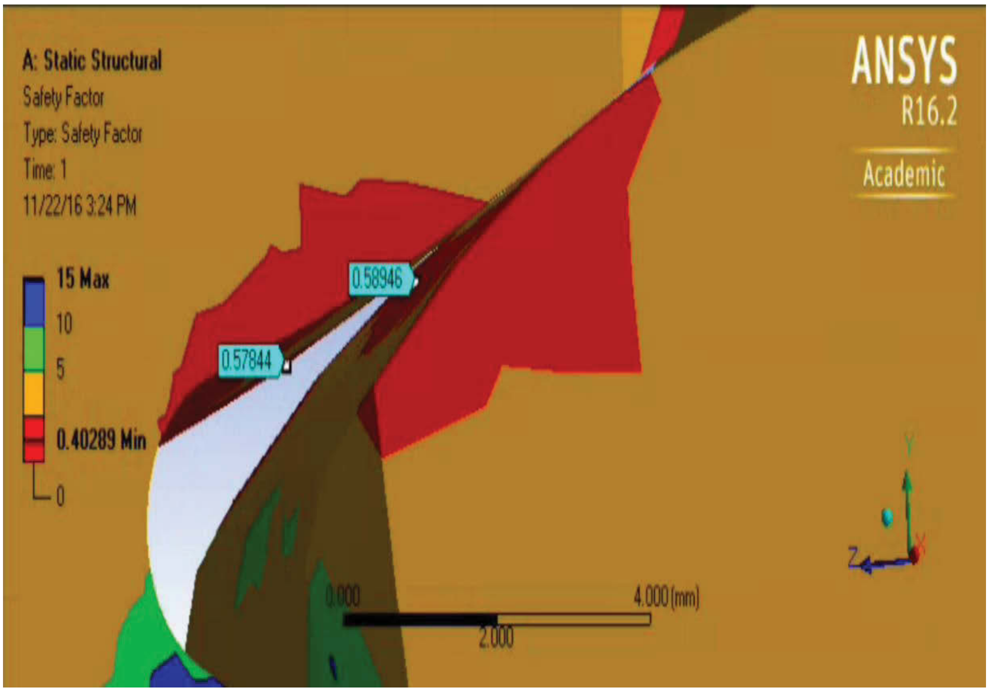


Figure 4.12: Safety factor at the flank (against pitting)

According to the contact stress formula, the nominal contact stress or flank pressure at pinion and gear is 1150.13 $\mathrm{N/mm^2}$ and 1188.72 $\mathrm{N/mm^2}$, respectively. The maximum contact stress is at the centre of the pinion tooth flank or pitch circle diameter.

$\sigma_{HO} = Z_H Z_E Z_\epsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{u+1}{u}}$ (Chapter 3, Equation 2)

$\sigma_{HP} = \frac{\sigma_{H\,lim}\, Z_{NT}}{S_{H\,min}} Z_L Z_V Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H\,min}}$ (Chapter 3, Equation 4)

The safety for stress at single tooth contact (flank), using the equation below, is 0.57 and 0.58 on pinion and gear, respectively. The minimum required safety factor is 1, but the result after analysis shows less than 1 — the gear and pinion are unsafe. ANSYS 16.2 shows the difference error of safety factor, with the variation of 1.7% to 3.4%. The formula used is:

$S_H = \frac{\sigma_{HG}}{\sigma_H} > S_{H\,min}$ (Chapter 3, equation 5)

Tables 4.2 and 4.2.1 have listed the pinion and the gear contact stresses and safety factors, respectively. Also, the percentage error between 3-D models in ANSYS and ISO-6336 values is defined.

Table 4.2: von Mises (contact) stresses for spur gear C45 model

	ISO contact stresses ($\mathrm{N/mm^2}$)	3D contact stress (ANSYS) ($\mathrm{N/mm^2}$)	Permissible Contact stress ($\mathrm{N/mm^2}$)	Difference error (%)
Pinion Driver	1150.13	1143.2	734.82	0.60
Gear Driven	1188.72	1183.3	734.82	0.47

Table 4.2.1: Comparison of safety factor

	Safety at tooth contact (ISO6336)	Required safety	Safety at tooth contact(ANSYS)	Difference error (%)
Pinion Driver	0.58	1	0.57	1.7
Gear Driven	0.56	1	0.58	3.4

4.2.2 Tooth root bending stress of spur gear (C45)

Spur pinion and gear assembly are imported into ANSYS 16.2 and the same boundary conditions are applied as the contact stress spur gear model. Figure 4.13 below, from ANSYS 16.2, shows the stress distribution plot along the root tooth of the mating gear. Theoretical example of the bending stress for gear model could be calculated using ISO 6336 formula (Equations 9, 8 and 7, Chapter 3):

$$\sigma_{FO} = \frac{F_t}{b\, m_n} Y_F Y_S Y_\beta Y_B Y_{DT} = \frac{5954.205 \times 1.48 \times 2.01 \times 1 \times 1 \times 1 \times 1}{16 \times 3} = 369.07\ \mathrm{N/mm^2}$$

$$\sigma_F = \sigma_{FO} K_A K_V K_{F\beta} K_{F\alpha} = 367.54 \times 1.05 \times 1.10 \times 1 \times 1 = 424.50\ \mathrm{N/mm^2}$$

$$\sigma_{FP} = \frac{\sigma_{F\lim} Y_{ST}\, Y_{NT}}{S_{F\,min}} Y_{\delta\,rel\,T} Y_{R\,rel\,T} Y_X = \frac{\sigma_{FE} Y_{NT}}{S_{F\,min}} Y_{\delta\,rel\,T}\, Y_{R\,rel\,T}\, Y_X = \frac{\sigma_{FG}}{S_{F\,min}} = 518.60\ \mathrm{N/mm^2}$$

$$S_{F1} = \frac{\sigma_{FG2}}{\sigma_{F1}} \geq S_{F\,min} = \frac{726.04}{424.50} = 1.71$$

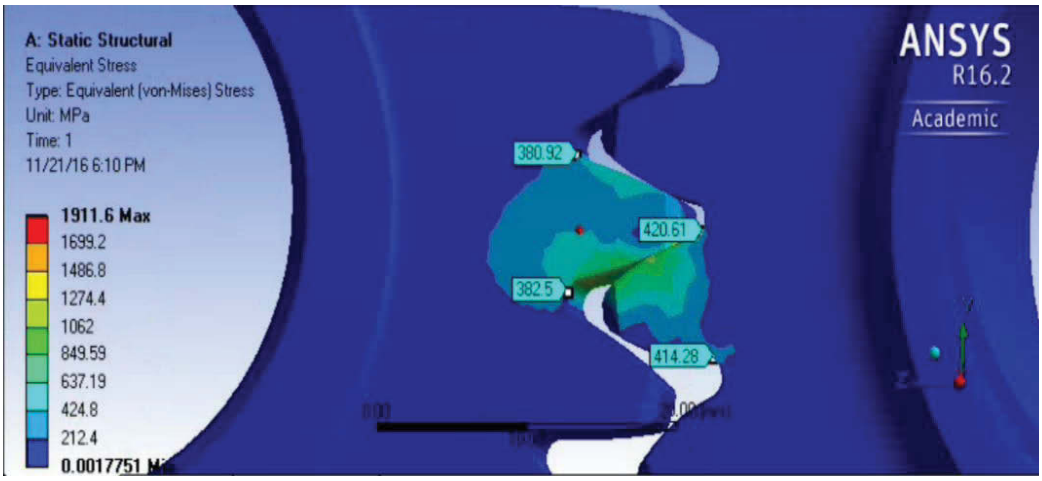


Figure 4.13: Tooth bending stress

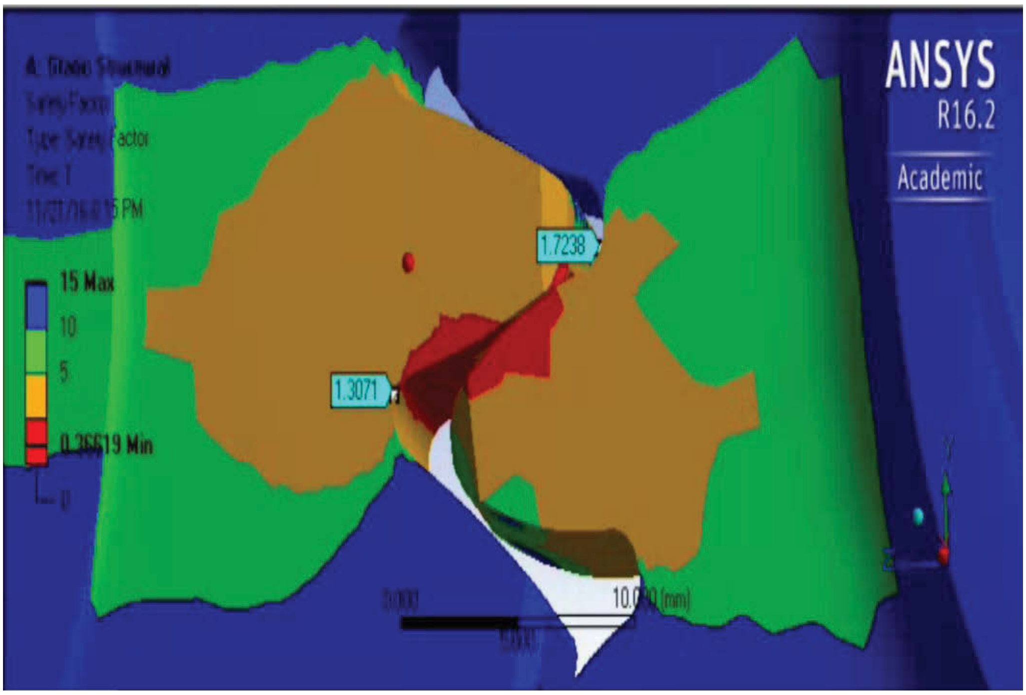


Figure 4.14: Safety factor at the root (against tooth breakage)

Tables 4.2.2 and 4.2.3 show the comparison of von Mises stress distribution for pinion and gear, safety factor over the root of the pinion and gear, and the difference error percentage between the ISO 6336 result and FEA result by ANSYS 16.2.

Table 4.2.2: von Mises (bending) stress for spur pinion and gear (C45)

	ISO bending stress N/mm^2	3D bending stress (ANSYS) (MPA) (N/mm^2)	Permissible tooth root stress (N/mm^2)	Difference error (%)
Pinion Driver	381.54	380.92	362.66	0.16
Gear Driven	424.50	420.61	518.60	0.92

Table 4.2.3: Comparison of safety factor between pinion and gear (C45)

	Safety at tooth root (ISO 6336)	Minimum safety	Safety at tooth root (ANSYS)	Difference error (%)
Pinion Driver	1.33	1.40	1.30	2.2
Gear Driven	1.71	1.40	1.72	0.5

The above compared results illustrates that the C45 gear shows acceptable safety at the root of the tooth, which is 1.71. But safety at the root of C45 pinion is 1.33, which is less than minimum required safety. Gear and pinion have to have at least minimum safety value to remain safe for usage. Also, safety against the pitting over the flank is 0.58 and 0.56 of pinion and gear, respectively, and the minimum required safety at the flank is 1. So, gear and pinion with C45 material fails at flank or pitch circle diameter.

Therefore, C45 gear and pinion are not advocated for use in industry where tangential force, $F_{t,}$ is more than 5954.205 N. In addition, material change is highly recommended to achieve more satisfactory results.

4.2.3 Contact stress of spur gear and pinion (19MnCr5).

Stress at the flank, and stress at the tooth root of gear and pinion studied in this section, are calculated in the same manner as stresses of pinion and gear with C45 material, but the material would be 19MnCr5. The von Mises stress distribution would be obtained at the contact region and at the root of the pinion gear tooth. Figures 4.15, 4.16, 4.17 and 4.18 below, present the stress distribution over the gear flank and gear root with 19MnCr5 pinion and gear material.

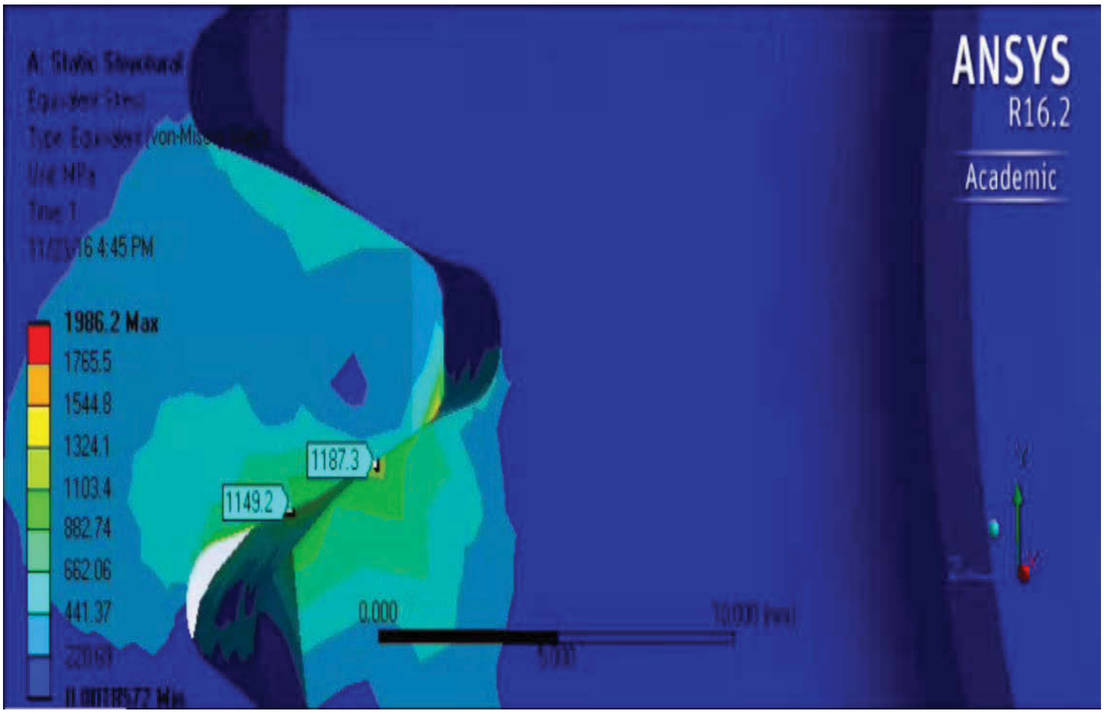


Figure 4.15: Stress at flank (19MnCr5)

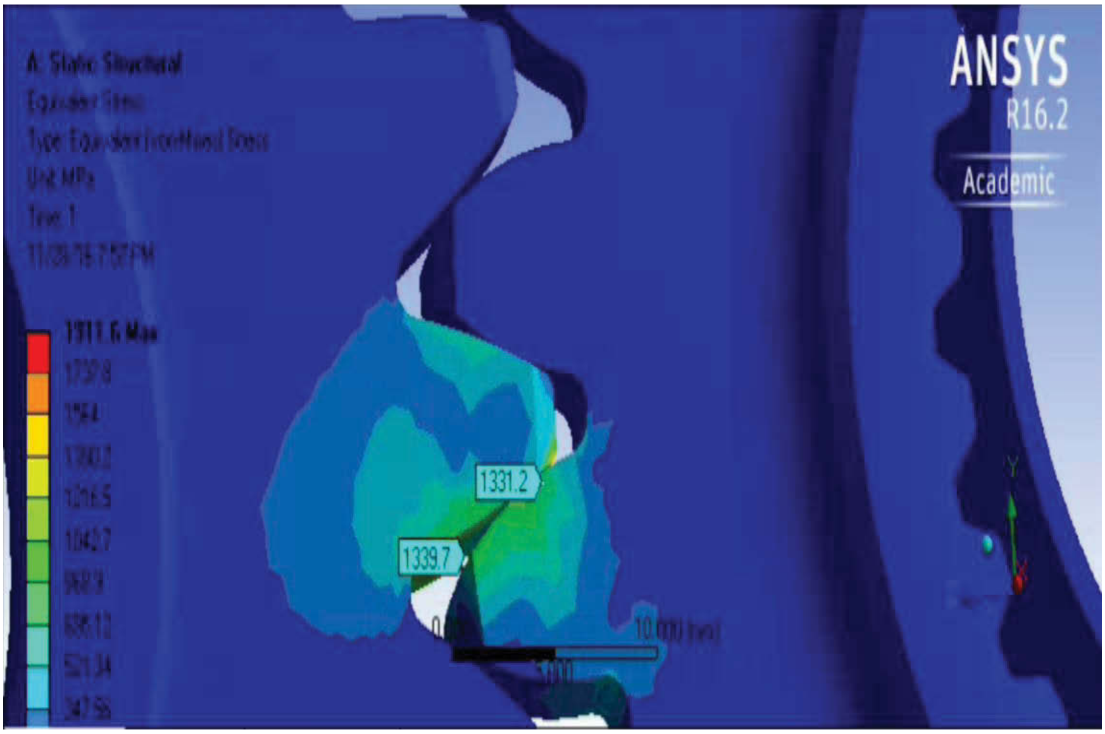


Figure 4.16: Permissible stress at flank (19MnCr5)

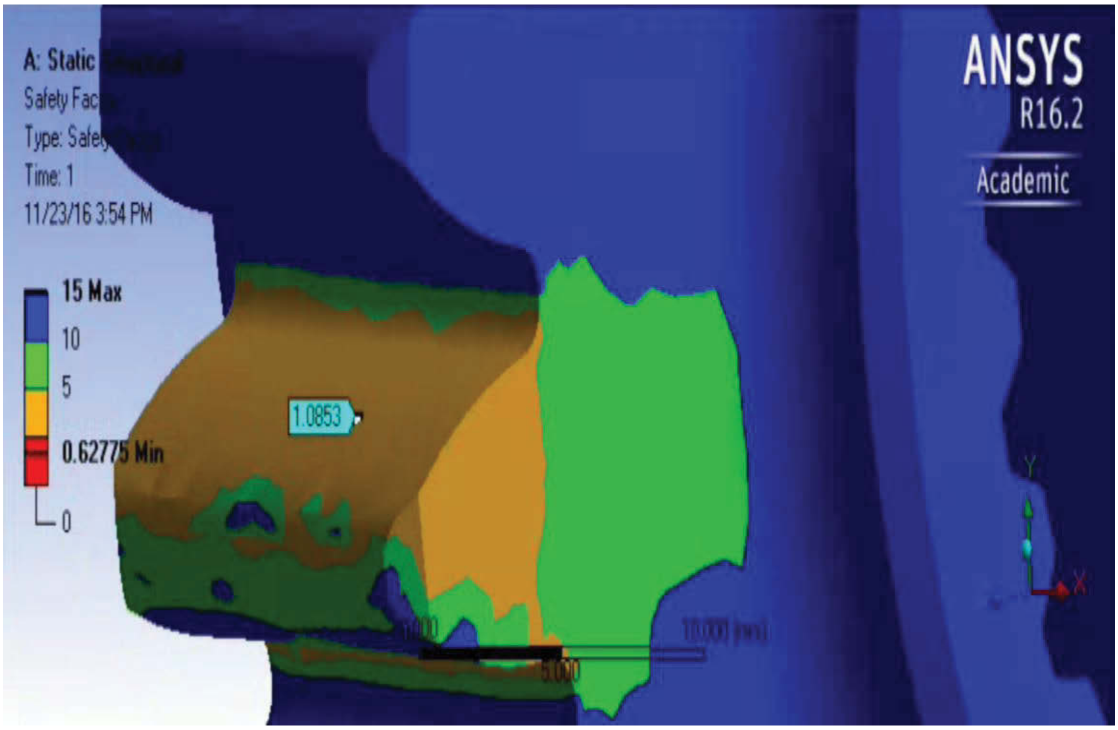


Figure 4.17: Safety at flank on gear (19MnCr5)

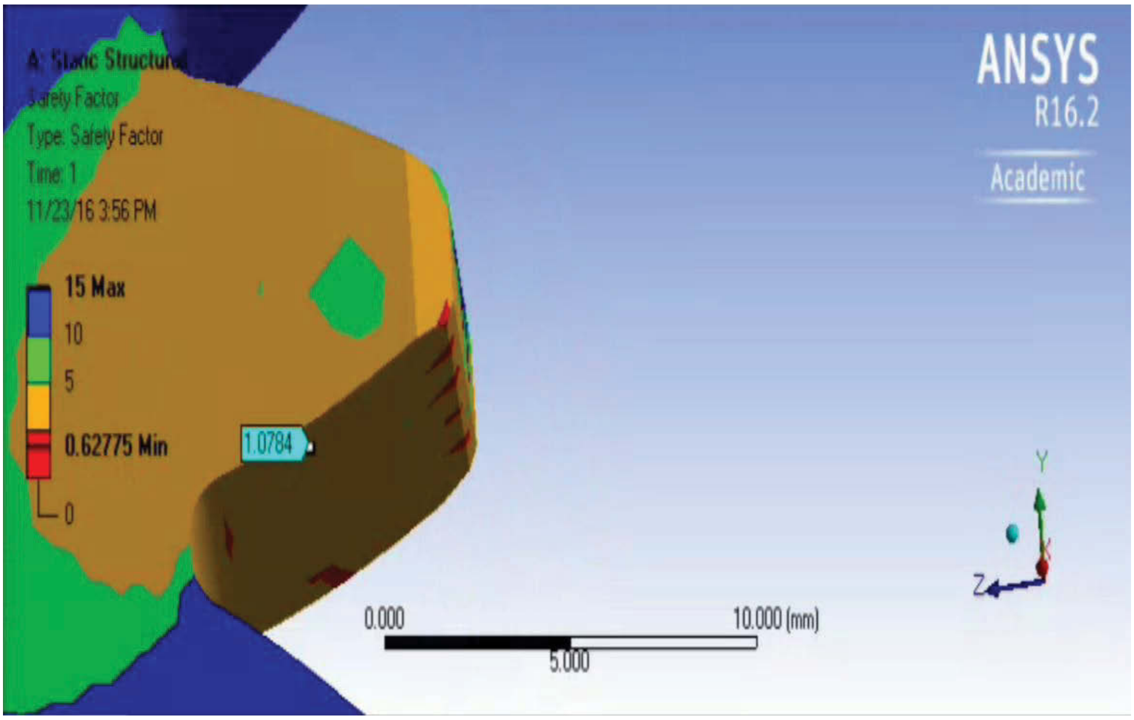


Figure 4.18: Safety at flank on pinion(19MnCr5)

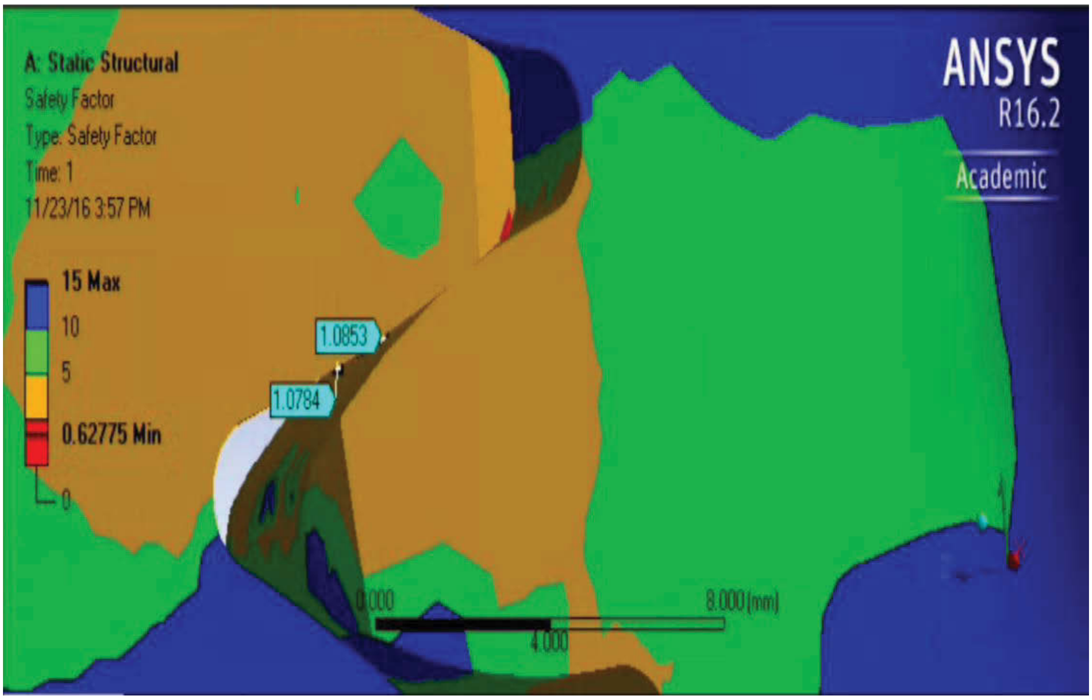


Figure 4.19: Safety factor against flank pitting (19MnCr5)

The result explicitly shows that the mathematical and ANSYS simulation results have agreed with each other, with minor difference error. The theoretical values (contact stress) are obtained by using Equation 2 and Equation 4 in Chapter 3.

The safety factor at the tooth contact with 19MnCr5 pinion and gear material is 1.07 and 1.08, respectively. Also, minimum required safety factor is 1 at the flank or pitch circle diameter. The formula used to compute safety is referred from Equation 5 in Chapter 3.

Table 4.2.4 compares the ISO 6336 standard values with ANSYS simulation results at the flank and root of pinion and gear, with 19MnCr5 material. Also, the difference error between ISO standard and ANSYS 3D models is compared in percentage.

Table 4.2.4: von Mises (contact) stress of spur pinion and gear (19MnCr5)

	ISO contact stress N/mm^2	3D contact stress (ANSYS) (MPA)	Permissible contact stress (N/mm^2)	Difference error (ISO and 3D) (%)
Pinion Driver	1150.13	1149.2	1346.62	0.08
Gear Driven	1188.72	1187.3	1346.62	0.11

Table 4.2.5: Safety factor at flank of pinion and gear(19MnCr5)

	Safety at tooth contact (ISO 6336)	Minimum safety required	Safety at tooth contact (ANSYS)	Difference error (%)
Pinion Driver	1.07	1	1.07	0
Gear Driven	1.03	1	1.08	3

Table 4.2.5 proves that the pinion and gear with 19MnCr5 material exceed the minimum required safety. Under such conditions, pinion and gear would work perfectly fine for the recommended service life. In this study, the required service life is 20,000 hours. Furthermore, if the safety factor at the root exceeds the minimum required value, then the pinion and gear are safe to use.

4.2.4 Stress at the tooth root (19MnCr5)

Using ANSYS 16.2, three-dimensional root stress is obtained. Stress results are then compares with ISO 6336 theoretical stress. Figure 4.20 illustrates the von Mises stress distribution in 3-D models. Also, the Tables 4.2.6 and 4.2.7 consolidate the results of safety factor and stress distribution along with percentage of error between 3-D ANSYS value and ISO 6336. Tooth root stress and safety factor for spur pinion and gear are calculated with the reference Equations 6, 7, 9 and 10 in Chapter 3.

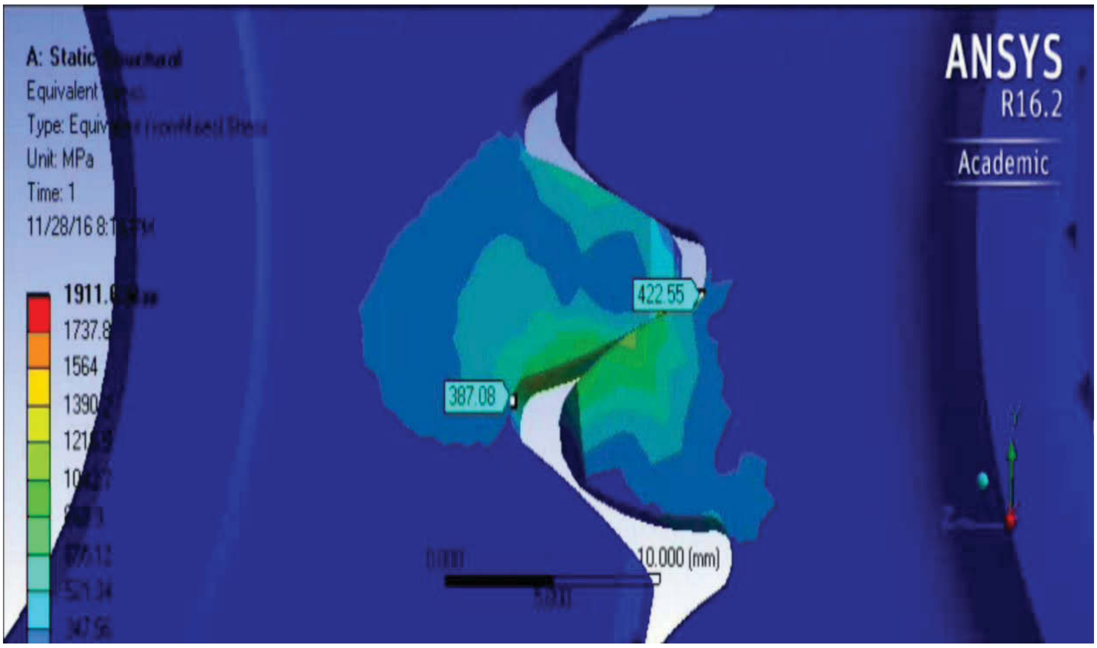


Figure 4.20: von Mises stress at the root (19MnCr5)

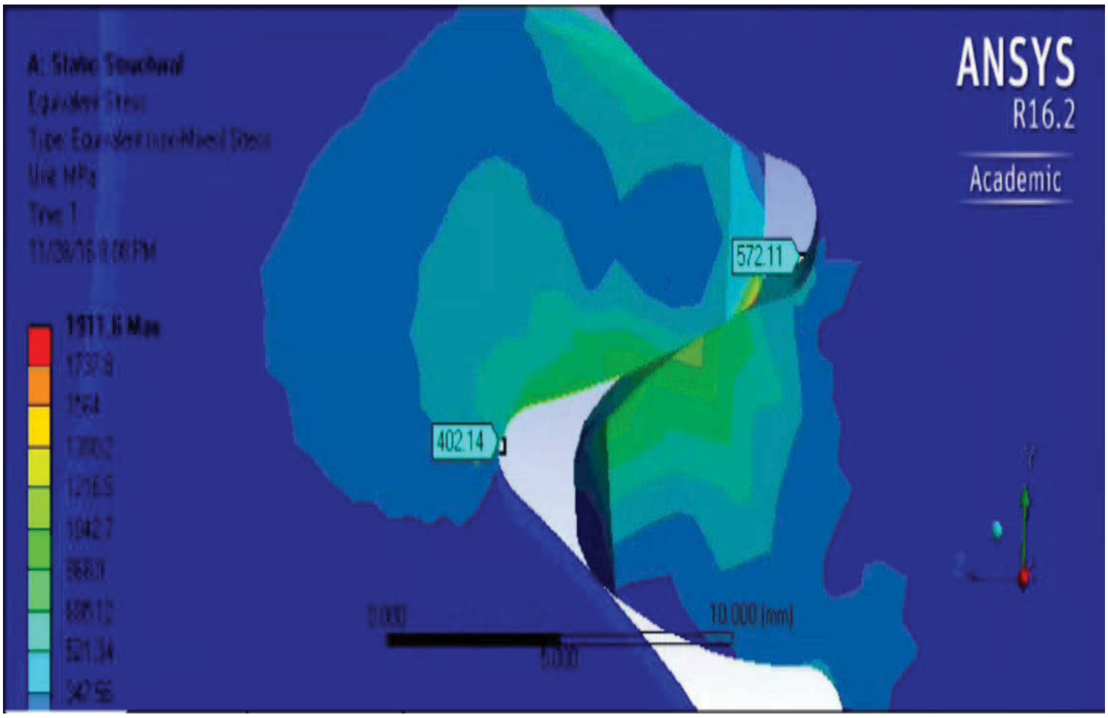


Figure 4.21: Permissible tooth root stress

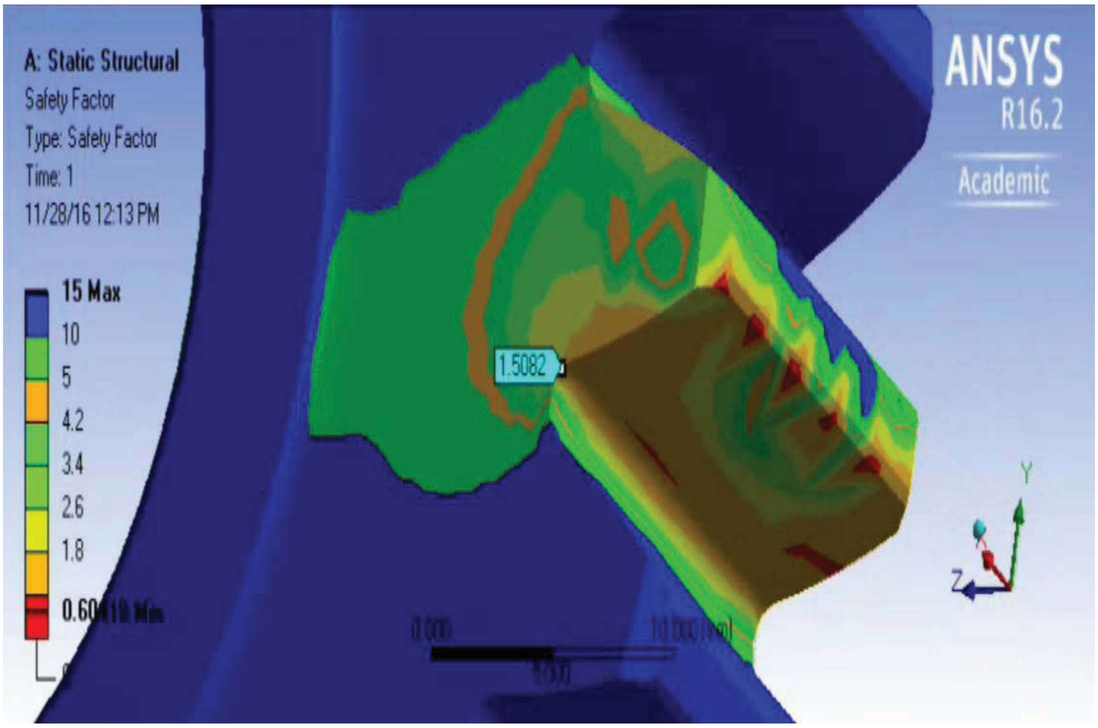


Figure 4.22: Safety factor at pinion root

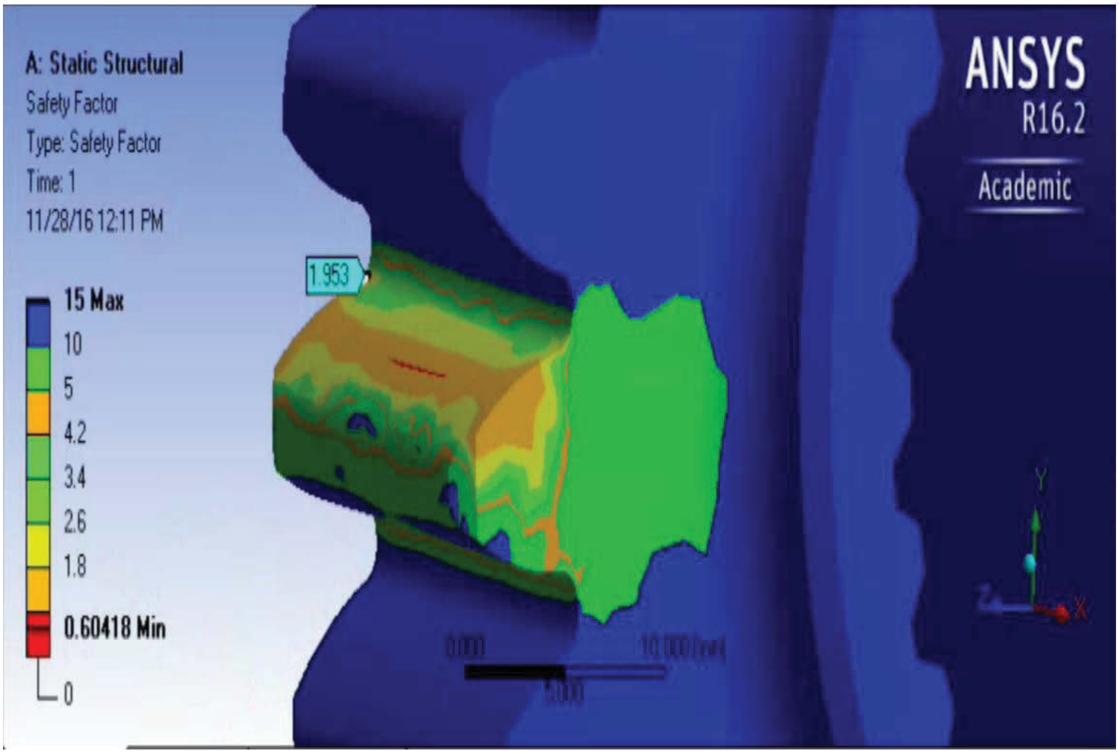


Figure 4.23: Safety factor at gear root

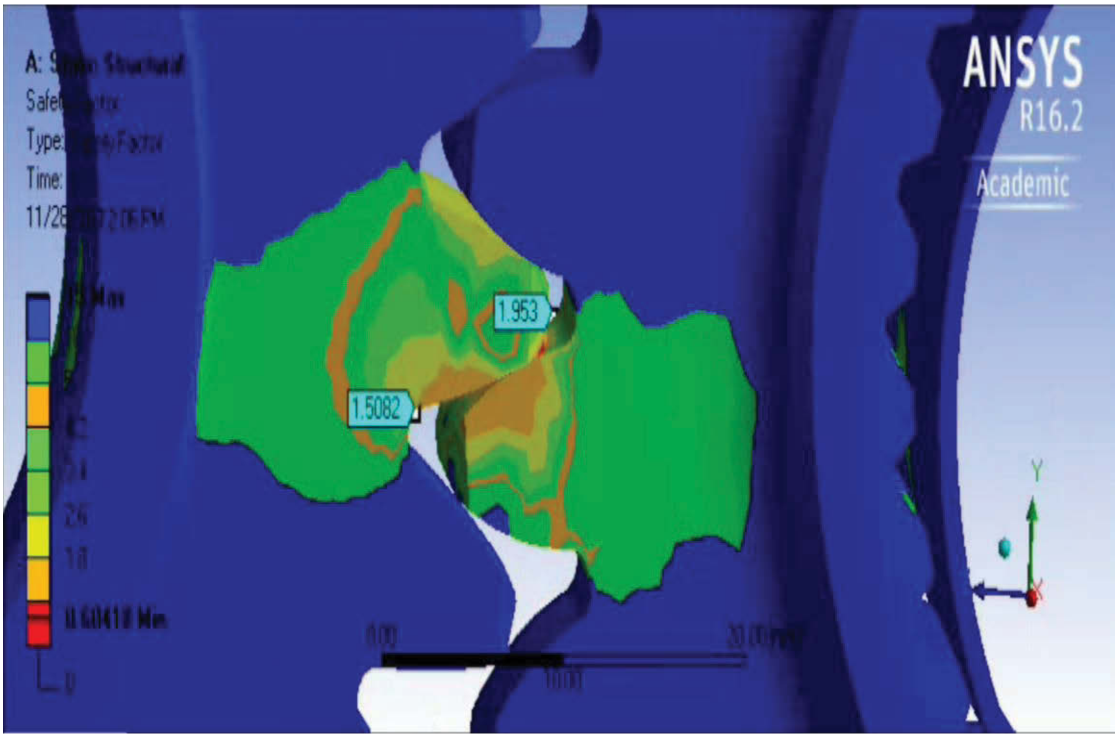


Figure 4.24: Stress at tooth root (against tooth breakage)

Table 4.2.6: Comparison of von Mises stress at the root

	ISO bending stress N/mm^2	3D bending stress (ANSYS) (MPA)	Permissible bending stress N/mm^2	Difference error (ISO and 3D) (%)
Pinion Driver	381.54	387.4	410.86	1.5
Gear Driven	424.50	420.86	587.07	0.86

Table 4.2.7: Safety factor comparison at tooth root (19MnCr5)

	Safety at tooth root (ISO-6336)	Minimum safety	Safety at tooth root (ANSYS)	Difference error (%)
Pinion Driver	1.51	1.40	1.50	0.66
Gear Driven	1.93	1.40	1.95	1

Table 4.2.7 illustrates that the safety at the pinion and gear root with 19MnCr5 material is more than the minimum safety value, that is, 1.40. Also, Table 4.2.5 shows that the safety factor at the flank of the pinion and gear exceed the minimum required value.

4.3 Conclusion and findings

In this chapter, the FEA stress analysis result is compared with theoretical results (not in running scenario) to consolidate the contribution and finding of the chapter. First, the FEA result proves that the software simulation could detect the high percentage of gear failure, which minimises the requirement of practical prototypes. In material comparison, the percentage of error difference is between 0.08% and 3.4%, which proves the accuracy of the software.

Second, in ANSYS software, theoretical values such as allowable stress, contact stress, permissible stress, etc. were used to create properties for the material. After analysis, it was clear that the numerically-derived values obtained by ISO 6336-3 and -2 standards were in good agreement with finite element root stresses, and von Mises contact stresses under maximum value.

Finally, in this study, the minimum service life of pinion and gear is recommended as 20,000 hours. The gear and pinion with 19MnCr5 material has shown the sustainable stresses and safety factor over the flank and root within minimum service life. However, it would be challenging to prove all the results practically, but that would be next step.

CHAPTER 5

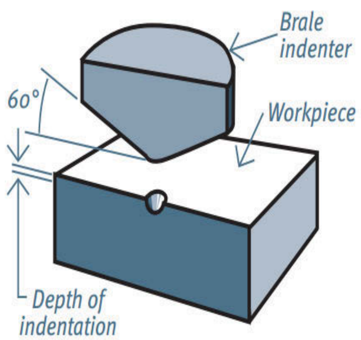
HARDNESS TEST

In this chapter, a practical test is performed soon after comparing simulation results. Initially, the brief introduction about the type and procedure of the hardness test is discussed. Then, the practical Rockwell hardness test is performed over the spur pinion and gear with two different materials. This is followed by the comparison between C45 and 19MnCr5 materials results. In the end, the conclusion and findings are discussed to validate the purpose of the test.

5.1 Introduction

The hardness is known as resistance of a material to plastic deformation, which is measured with the depth or area of indentation. The smaller the indentation, the higher will be the hardness. Hardness of the material has long been investigated by resistance to scratching or cutting. The capability of the material to resist cutting by another material could be graded by referring to the Mohs scale, that assesses relative hardness of materials. Hardness is a mechanical property, which can be dramatically changed by processing and heat treatment. In this research, nitriding and case-hardening processes are performed over the pinion and gear before the hardness test.

Several hardness tests exist, some of which are for special purposes only. In defining hardness value, it is necessary not only to give numerical value but also to indicate the scale or type of test used. Hardness testing abbreviation starts with H ("hardness") followed by additional letters and numbers indicating the specific type of test.



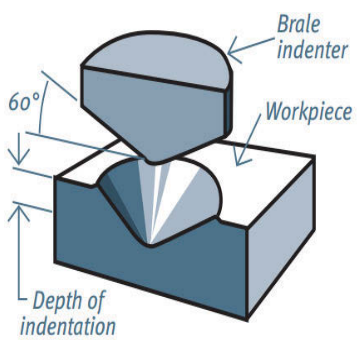


Figure 5 .1 Indentations made by minor load   Fig 5.2 Indentation by major load

Testing is conducted by pressing or indenting one material into another with a known amount of mechanical force. Since the ability of the material to resist deformation is related to the yield point and the material's capacity for work-hardening, the result is actually a measurement of relative hardness. The shape of the indenters is defined by the respective standard of hardness testing and can be very different, depending on whether the indenter is cone, pyramid or sphere. At the point of contact between test material and indenters, the stress easily exceeds the yield strength of the tested material, which is plastically deformed as the indenter moves into the material.

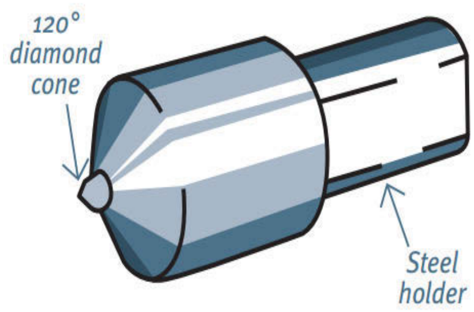


Figure: 5.3 Diamond indenter

(Source: Chandler, H., Metallurgy for the non-metallurgist, ASM handbook, 1998)

The three most common hardness tests are Brinell, Vickers and Rockwell hardness test. Rockwell hardness test is adopted in this research. The hardness values of this test are based on the difference of indenter depths from two load applications.

Major principles of the Rockwell hardness test:

- Position the surface area to be measured close to the indenter.
- Apply the minor load and establish zero reference position.
- Apply the major load for a specified time period (dwell time) beyond zero.
- Release the major load leaving the minor load applied.

5.2 Test on spur pinion and gear with C-45 and 19MnCr5 material:

Initially, a minor load is applied on the surface of the pinion tooth profile which serves as a starting position. Then a major load is applied for a certain period of time, which increases the penetration depth after a specified dwell time for the major load; the major load is removed from the pinion surface, but the minor load still maintained. The difference in depth is the Rockwell hardness number. The initial application of the minor load increases the accuracy of the testing, since it eliminates the effect of the surface layers, which may not be representative of the bulk material. The same procedure is followed for the gear surface. Figure 5.4 shows spur pinion in vertical position.

Rockwell hardness values are expressed as a combination of hardness number and a scale symbol representing the indenter and the minor and major test forces. The Rockwell hardness is expressed by the symbol HR and the scale designation. With the mixture of three different indenters and three different loads, there are nine scales available for testing (HRA, HRB, HRC, HRD, HRE, HRG, HRH and HRK).

Figure 5.4 Spur pinion in vertical position

The majority of application for testing steel are covered by the Rockwell C and B scales. For example, a diamond cone indenter is used for scale C and a ball indenter for scale B (HRB, scale value 20-100). Indenter size and test force are selected according to the type of material, sample thickness, test location and scale limitations. Advantages of Rockwell test are:

- Various materials can be tested with the same testing method for comparison.
- Hardness test is non-destructive, the specimen is neither fractured nor excessively deformed.
- HRC measurements are preferred methods for hard steels.
- The procedure is fast; the entire testing takes only 5-10 seconds.
- Hardness results are directly read on the measuring device.

In order to get a reliable reading, the thickness of the test-piece should be at least 10 times the depth of the indentation. Also, readings should be taken from a flat perpendicular surface, because convex surfaces give lower readings. A correction factor can be used if the hardness of a convex surface is to be measured. The test is performed in the laboratory of the University of Technology Sydney, Australia.

Table 5.1: Rockwell hardness test reading from lab

Test	Gear (C45) tooth profile HRC	Gear (19MnCr5) tooth profile HRC
Test 1	50	59.4
Test 2	52.5	60.7
Test 3	52	61.5

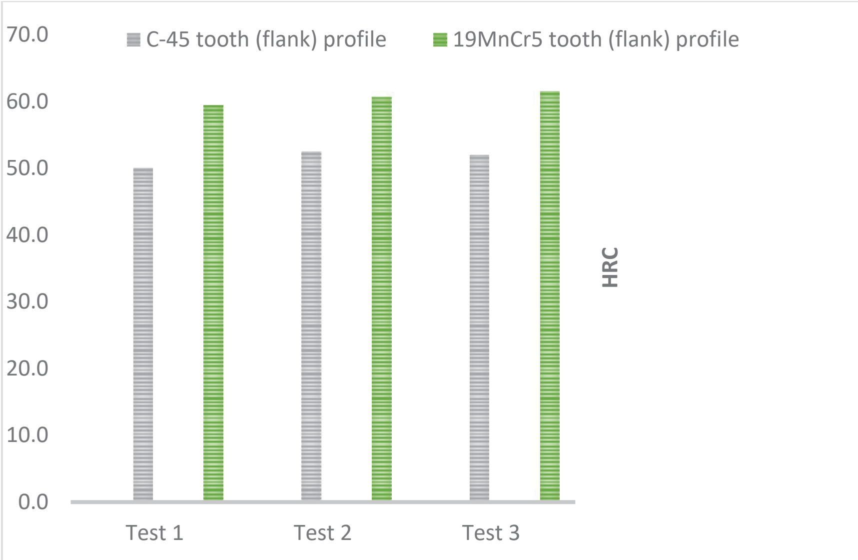


Chart 5.1: Comparison of hardness test results

Table 5.1 illustrates Rockwell hardness test results. Scale "C" for Rockwell hardness test is referred. Although several tests on the various parts of the spur pinion and gear were performed, the test over the profile or flank of the teeth is the major concern in this thesis. The flank is the area at the pitch circle diamter (PCD) of pinion and gear, where the contact between mating pinion and gear is meaured. Three tests were performed on spur pinion and gear, with two different materials, C45(nitride hardned) and 19MnCr5 (case-hardened). Test 1 on involute profile of pinion with C45 materal has given a 50 HRC value. In the following tests 2 and 3, the HRC value measured 52.5 and 52.0, respectively, on two other teeth of the pinion. On the other hand, all the three tests over the gear flank with 19MnCr5 material measured 59.4, 60.7 and 61.5 HRC. The hardness (not near to brittle value) at the flank of the gear or pinion gives higher load carrying capacity to the gear, which provides a longer service life.

5.3 Conclusion and findings

Although the hardness test does not directly contribute to enhancing the life of the spur gear, there are some major indirect contributions to the research.

First, the hardness test results show that 19MnCr5 material has much more hardness in comparison to C45 material. The hardness percentage difference between the two materials is 18.18%. Also, more hardness means much more fatigue strength at the root and at the flank of the gear, higher tensile strength, and more yield point. Therefore, better material for gear application. In summary, the wear resistance of the gear can be adjusted by controlling the surface hardness. The allowable stress and nominal stress are calculated from ISO 6336-5 using the formula:

$$\sigma_{H\ lim}, \sigma_{F\ lim} = A.x + B$$

Where,

x the surface hardness HV or HRC

A, B constant from ISO 6336-5 page 6, table 1.

Second, the hardness values achieved from the experiment were almost the same as the values measured from ISO 6336-5, ISO 6508-1 and ASTME 18 tables, which validate the research work.

Finally, hardness value at the rim area of the gear and pinion shows different hardness values compared to teeth, which is worth noting, because after case-hardening or nitriding processing, hardness of the teeth of the gears is comparatively higher. This means that the hardening process is deliberately performed over the tooth of the gears and not the entire body, which opens a new pathway for this research.

CHAPTER 6

THESIS CONCLUSION

6.1 Summary of thesis

The service life of spur pinion and gear is investigated within this thesis, which includes calculation of spur gears, finite element analysis using simulation software, comparing two materials and comparison of data acquired from analysis with ISO 6336 standard. Initially, a literature survey brief was conducted about the application of gears, advancement of gear technology, material characteristic and strategy for improving gear designing and manufacturing. Most of the dissertation has been devoted to gear calculation, gear modelling, simulation, material selecting and performing experiments.

In this thesis, load carrying capacity calculations of spur gears with C45 and 19MnCr5 material is performed in Chapter 3, followed by 3-D modeling and finite element method in Chapter 4. International Organization of Standardization (ISO 6336) standards are referred to perform mathematical calculations with every influence and safety factor. To prove the theoretical data authentic, finite element method (FEM) is used. Stress is measured at the flank and tooth root of the pinion and gear with the same geometry, and two different gear materials were shown in Chapter 4. Also, the safety factors of pinion and gear with both materials are compared. After mathematical results, hardness test is performed in Chapter 5, to show the material hardness property. Material properties play a significant role in this research.

6.2 Summary of findings and contributions

The objectives of this thesis identified in the introduction are restated here along with the primary finding and contributions that have emerged from this investigation:

1. *Material analysis using mathematical formulas and simulation software to select material for pinion and gear with precise alloy combination.*

During comparison between C45 and 19MnCr5 material, some mechanical properties such as fatigue strength have shown noticeable difference. This difference is about 15% to 40%, which is arguably within variance (C45 has less fatigue strength compared to

19MnCr5). Also, with the change of the material, the physical and environmental conditions for gear manufacturing are not affected, which could improve efficiency from a financial point of view.

Evidently, not every region of gear and pinion has an equal amount of load carrying capacity. This variation in stress distribution could lead to new areas of research.

2. *Comparing mathematical and FEA results.*

The FEA results have proved that the simulation software, such as ANSYS, could anticipate high percentage of gear failure which reduces the cost of practical prototypes. The accuracy of the result is between 0.08% and 3.4%. Also, the results obtained from mathematical and FEA analysis were in good agreement with international standards (ISO 6336) under all load conditions.

In addition, contact and bending stresses, and safety factor of the gear over the flank and root of the gear teeth with 19MnCr5 material, meets the requirement to operate at minimum service life. In this research, the recommended minimum service life of gear is 20,000 hours.

3. *Rockwell hardness test is performed to support the theory.*

To get reliable results under the hardness test, the object at hand should be at least 10 times the depth of the indentation, which is mentioned in Chapter 5. This condition is totally accomplished by 19MnCr5 material. Hardness at the flank of the gear is 59 to 62 HRC, which is higher in comparison to material C45. Also, hardness values measured by the experiment were compatible with ISO standards. Moreover, no direct correlation is observed between FEA and the hardness test. This test shows only the relation between hardness and gear safety, which assists in selecting the most reliable material for gear manufacturing.

Interestingly, there is variation in the hardness value between the rim and the teeth of the gear. This variation has opened different perspectives for future research. Also, applying hardness test results in the FEA would be a major area for study.

6.3 Further research

A line of approach for further research into improvement of gear life includes:

- The variation of stress over the flank and root of the gear tooth opens new areas for research. Further study would concentrate on the effect and causes of stress distribution variance.
- It would be appropriate to analyse sub-surface stresses over the flank and root of the gear teeth with 19MnCr5 material. Also, to compare the sub-surface stress of 19MnCr5 material with other materials.
- It would be very interesting to form and use a compatible lubricant for 19MnCr5 material. More exciting would be to test such lubricant under different practical conditions. Although there are several high-grade lubricants available, materials such as 19MnCr5 are alloys, and require many suitability tests.
- Hardness at the rim of the gear and pinion are different from the teeth of the gear. It would be appropriate to analyse gear hardness processes to generate the same hardness value at every part of the gear, and then test the gear and pinion efficiency with similar gear geometry.
- The abstract for this research work has already been submitted to an SAE 2017 International Powertrain, Fuels and Lubricants meeting. In future, other than working on variation of stress distrubution, sub-surface stresses, new lubricant, and hardness value over the rim of the gear, the author would also concentrate on the publishing of quality journal papers.

6.4 Conclusion

The objective of the thesis has been achieved, as can be seen starting with Chapter 2, which gives the motivation to study in such a specific area. After conducting this survey, it was clear that not much research has been done in trying material such as C45 and 19MnCr5 in designing spur gears. This literature review helps to conclude that the finite element method is preferable for conducting service and safety analysis. Also, the design of experiment (DoE) methodology has been used to fabricate the structure of the thesis. This is outlined in Chapter 3, which gives the authenticity to the assumptions made in regards to C45 and 19MnCr5 material with the help of mathematical calculations. This is the assumption that 19MnCr5 material is safer to use for gear manufacturing and provides better results compared to C45 material. After theoretical computation, Chapter 4 helps to predict the safety of the gear and pinion with the help of finite element analysis. The comparison between ISO 6336 calculations and FEA results in this chapter shows that the safety factor and stress distribution values at the flank and root of the gear teeth are almost the same, with maximum 3.4% of error difference. Finally, the hardness test is perfomed in Chapter 5. This test concludes that 19MnCr5 material has much more hardness in comparison to C45 material. This study confirmed that theoretical, FEA, and hardness test results are all in good agreement with ISO standards.

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