CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF STUDY

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial machinery. It is possible that gears will become the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will require a refined application of gear technology [1].

Despite being a good power transmission mechanism, gears are also costly to manufacture and need lubrication system to prevent them from damage. There are many types of gears such as spur, helical, bevel, hypoid and crown. The simplest gear is the spur gear. It consists of a cylinder disk with teeth projecting radially, as shown in Figure 1.1.

Figure 1.1: A spur gear with 18 teeth

Any failure in gear transmission system will lead to the unexpected system failure. This will happen when any of the gear in a gear system exceeded the maximum limit of endurance; causing a failure in gear system. Gears can fail in many ways. Wear and pitting are some of the many defects found in a gear system that can lead to gear failure. These defects are due to surface fatigue, a result of repeated surface or subsurface stresses beyond the endurance limit of the material. Pitting is actually the fatigue failure of the tooth surface. Hardness is the primary property of the gear tooth that provides resistance to pitting. In other words, pitting is a surface fatigue failure due to many repetitions of high contact stress, which occurs on gear tooth surfaces when a pair of teeth is transmitting power. Problem arising from excessive wear and gear tooth surface pitting in gear transmission systems have been a concern for various gear users. Although the failure rate can be reduced by a regular maintenance, the cost and downtime required make such programs inefficient and not economical [3].

In most of the gear studies, commonly used method for stress analysis is Finite Element Method (FEM). This is because the method is accurate and the results obtained are acceptable. Nowadays the most common ways to conduct FEM is using computer simulation like ANSYS.

1.2 PROBLEM STATEMENT

Gear tooth failure due to wear and pitting is caused by surface contact stresses. These failures are the major area of concern and needs to be studied to avoid gear tooth failure. Therefore it is important to study and analyze the effect of wear and pitting on the contact stresses.

1.3 OBJECTIVE

The objective of this project is to investigate the effects of wear and pitting on the contact stresses using finite element method.

1.4 SCOPE OF STUDY

In this project, the author will be designing the 3D involutes spur gear pair by using SolidWorks software. In order to analyze the contact stresses of defected gears, the 3D gear will be simulated using finite element analysis software ANSYS. The results obtained by finite element analysis will be compared to the results from Hertz theory. The Hertz equation is the analytical formula used to determine the contact stress. Structural steel is used as the material of the gear.

CHAPTER 2

LITERATURE REVIEW

2.1 Overview of Gear Defect

Hardness is the primary property of the gear tooth that provides resistance to pitting. Pitting is surface damage from cyclic contact stress transmitted through a lubrication film that is in or near the elastohydrodynamic regime [2]. Pitting is one of the most common causes of gear failure. The literature available on the contact stress problems is extensive, but that available on the gear tooth contact stress problem is relatively small.

Pitting formation is a form of surface fatigue which may occur soon after operation begins and may be categorized to three types:

- Initial pitting/ Micro pitting
- Destructive pitting
- Normal Pitting

Initial pitting or sometimes called micro pitting as in Figure 2.1 is a contact fatigue phenomenon, caused primarily by localized severe stress concentrations occurring very near to the contacting surfaces [4]. These stress concentrations are the result of surface roughness or uneven surface on the gear tooth. This type of pitting can develop within a short period of time and it will reach a maximum state and will eventually reduce to a lesser severity state. Initial pitting usually occurs in a narrow band at the pitch line or just slightly below the pitch line [5]. Basically initial pitting will be considered as normal and less severe compared to destructive pitting where no correction can be made.

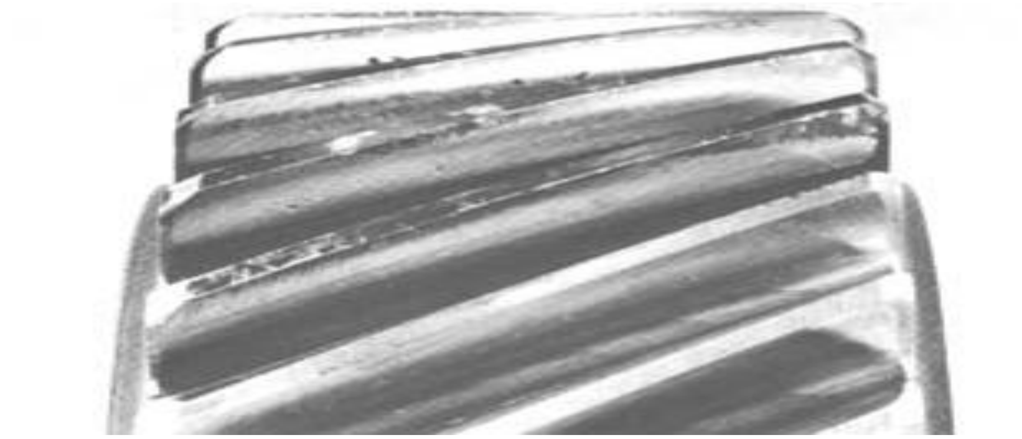


Figure 2.1: Initial Pitting [4]

Destructive or progressive pitting as in Figure 2.2 on the other hand usually starts below the pitch line and will progressively increase in both the size and number of pits until the surface is destroyed. Destructive pitting can be as severe as corrective pitting at the beginning but it will drastically increase as the time goes by. This kind of pitting usually resulted from the surface overload conditions but it is not the consequences of initial pitting.

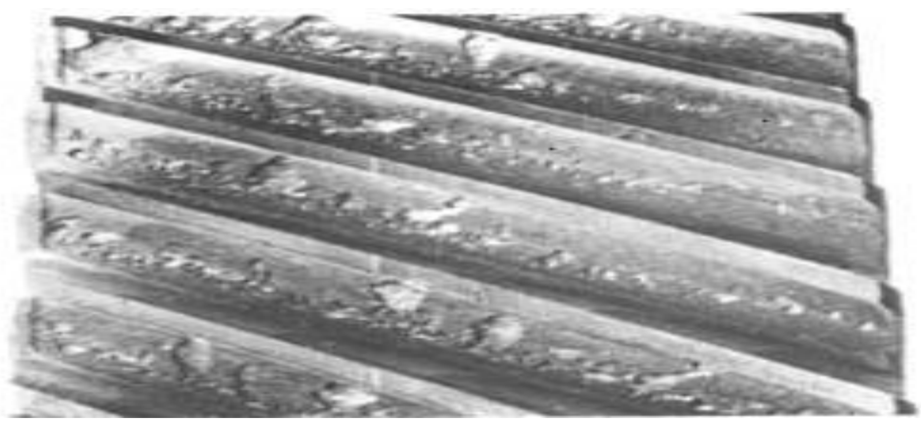


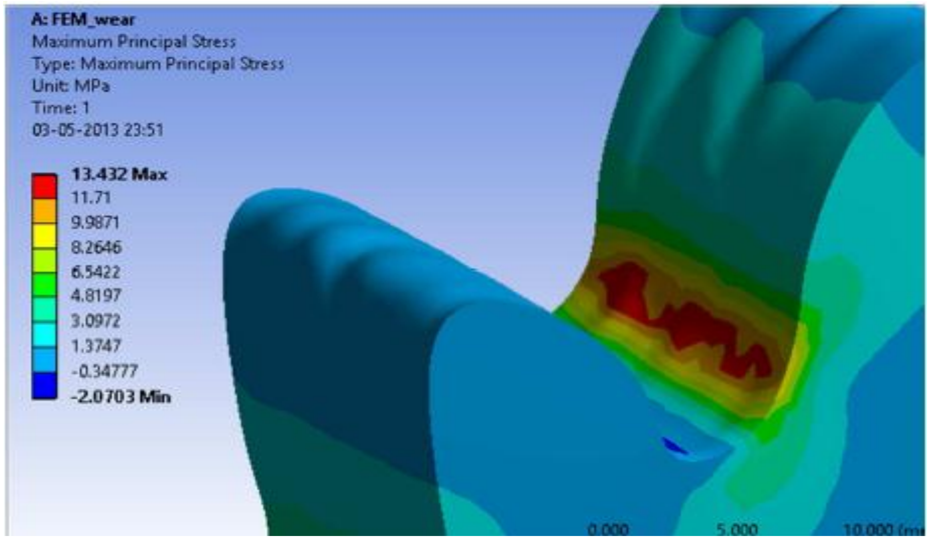
Figure 2.2: Destructive Pitting [4]

Normal Pitting or the dedendum pitting is a small or intermediate size pit which is covering the entire dedendum portion of the tooth flanks. Continued operation results in pit rims being worn away with virtually no further pitting occurring [4]. This type of pitting occurs when the load is approximately close to maximum allowable surface loading values.

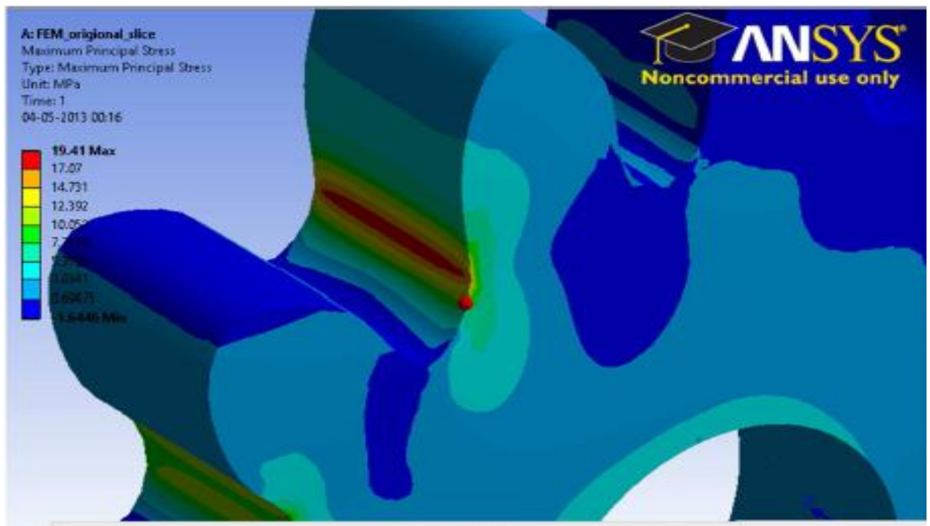
This contact stress defect also leads to loss of serviceability of the machine part and this will result to the loss of the performance of the system and eventually caused more losses to the industry related to this gear system. Thus a comprehensive study about this fatigue phenomenon and its effect on the gear transmission is crucially required.

Choy *et al.*[3] presented a comprehensive procedure to stimulate and analyze the vibrations in a gear transmission system with surface pitting, wear and partial tooth fracture of the gear teeth. They developed an analytical model where the effect of surface pitting and wear of the gear were simulated by phase and magnitude changes in the gear mesh stiffness. The authors concluded that the numerical analysis method gave good accuracy results and gear tooth damage due to wear and pitting can be simulated by amplitude and phase changes in gear mesh stiffness model. On the other hand, simulation of various degree of pitting and wear damage could provide a comprehensive database for gear fault detection and damage estimation research.

A study about the comparison between a normal gear and worn out gear have been conducted by Jain *et al.*[6]. Reverse engineering have been used to get a 3D CAD model design of a normal and worn out gear. The results obtained show that the shear stress and equivalent stress are low for original gear tooth and higher for worn-out tooth as shown in Figure 2.3 (a) and (b), respectively.



(a)



(b)

Figure 2.3: Effective stress distribution on a loaded gear tooth (a) along Z-axis for original gear (b) along Z-axis for worn-out gear.

The study shows that the maximum stress occurs near the roots of the gear and the maximum principal stress of worn-out gear tooth is more than the original tooth.

2.2 Overview of Finite Element Method and Hertz Theory

Gears analyses in the past were performed using analytical methods, which required a number of assumptions and simplifications. In general, gear analyses are multidisciplinary, including calculations related to the tooth stresses and failures such as wear and pitting. In this project, contact stress analyses will be performed.

Current methods of calculating gear contact stresses use Hertz's equations, which were originally derived for contact between two cylinders. The Hertz theory is based on the following assumptions [7]:

- Surfaces are continuous and non-conforming
- Contact area are elliptical
- Each body is an elastic half-space loaded over small elliptical region of its plane surface
- Surfaces are assumed to be frictionless

The example of comparison between two mating involutes gears and two cylinder concepts is shown in Figure 2.4 below.

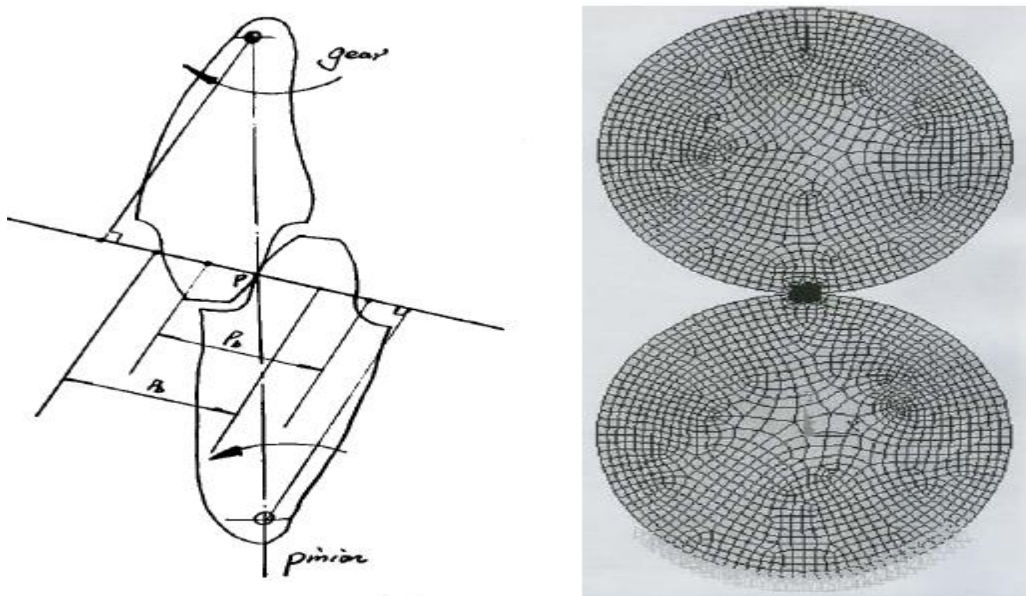


Figure 2.4: Two steel cylinders are pressed against each other **[1]**

To investigate the contact problems with FEM, the stiffness relationship between the two contacts areas is usually established through a spring placed between the two contacting areas. This can be done by inserting a contact element placed in between the two areas where contact takes place. Eventually the result of stress from simulation process of the two contact area was compared with the theoretical values. Both results of analytical and simulation process should tally with each other. This is to indicate that the FEM model is accurate and applicable to contact stress analysis.

As technology have become more advanced, people are likely to use numerical approaches to develop theoretical models to predict the effect of whatever being studied. This has improved gear analyses and computer simulations. Numerical methods can provide more accurate solutions since they require much less complicated assumptions. However the model and method must be carefully chosen to get an accurate result and a reasonable computational time. In this case, the finite element method is very often used to analyze the stress state of an elastic body with complicated geometry, such as a gear.

CHAPTER 3

METHODOLOGY

3.1 PROJECT METHODOLOGY

At the end of this project, a comparison between each failure mode will be established. The comparison is about the effects of failure modes to the maximum contact stress of the gear. There are two types of gear fault that will be considered, namely wear and pitting. It is believed that they will give different results. The maximum contact stress result will be used to indicate the effect of gear fault on gear performance.

In order to achieve good results, a detailed study of the gear and finite element analysis is crucial. First and foremost, before the literature review, it is very important to understand and focus on the objective and scope of the project. For this project, a good understanding of the Hertz theory and finite element analysis is very important.

In this project, the finite element models and solution methods needed for the accurate calculation of three dimensional spur gear contact stresses will be determined. Then, the contact stresses calculated using ANSYS will be compared to the results obtained from existing methods which are Hertz theory and from literature review. The purpose of this project is to develop a model to study and predict contact stresses of gears in mesh using the ANSYS software package based on numerical method. The aim is to analyze the effect of the gear defect on the contact stress of the gear pair. Thus two important tools or software that will be used are SolidWorks and ANSYS. SolidWorks is used to create the 3D model before simulation and ANSYS software will be used to simulate the 3D model created in SolidWorks.

The project will start by designing of a healthy spur gear for validation purpose. For the stress analysis, the procedure was taken from previous study by Kumar and Tiwari [8]. Only the analysis of the healthy gear will be referred for validation purpose. In their studies, Kumar and Tiwari compared the results of contact stress by using Finite Element Analysis to the calculated result by AGMA and Hertz Theory equation. From the result, it was approved that the contact pressure value obtained by using FEM was comparable to the result of AGMA and Hertz Theory equation.

Hertz Equation for contact stress is given by [9]:

$$\sigma_o = \sqrt{\frac{W\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{F\pi[\left(\frac{1-v_1^2}{E_1}\right)+\left(\frac{1-v_2^2}{E_2}\right)}}$$

Where

W= load

E_1= modulus of elasticity of pinion

E_2= modulus of elasticity of gear

v_1= Poisson’s ratio of pinion

v_2= Poisson’s ratio of gear

F= gear face width

R_1= $r_{p1}\sin\theta$

R_2= $r_{p2}\sin\theta$

r_{p1}= pitch radii of pinion

r_{p2}= pitch radii of gear

The results of the analysis by Kumar and Tiwari are shown in Table 3.1. Table 3.1 shows the results for contact stress using Hertz equation, AGMA contact stress equation and FEA.

Table 3.1: Contact stress for different method of analysis

Method	Contact stress (MPa)
Hertz Equation	562.67
AGMA contact stress Equation	572.00
FEA	567.75

The main purpose of using previous study as a validation is to ensure that the steps taken or methodology used to analyze the gear is correct. From the healthy gear analysis by previous study, further analysis about the effect of pitting formation on the contact stress of the gear will be done. This will be done by designing wear and pitting formation on the gear.

3.1.1 Project Flow Chart

Overall steps for conducting the project can be seen in Figure 3.1.

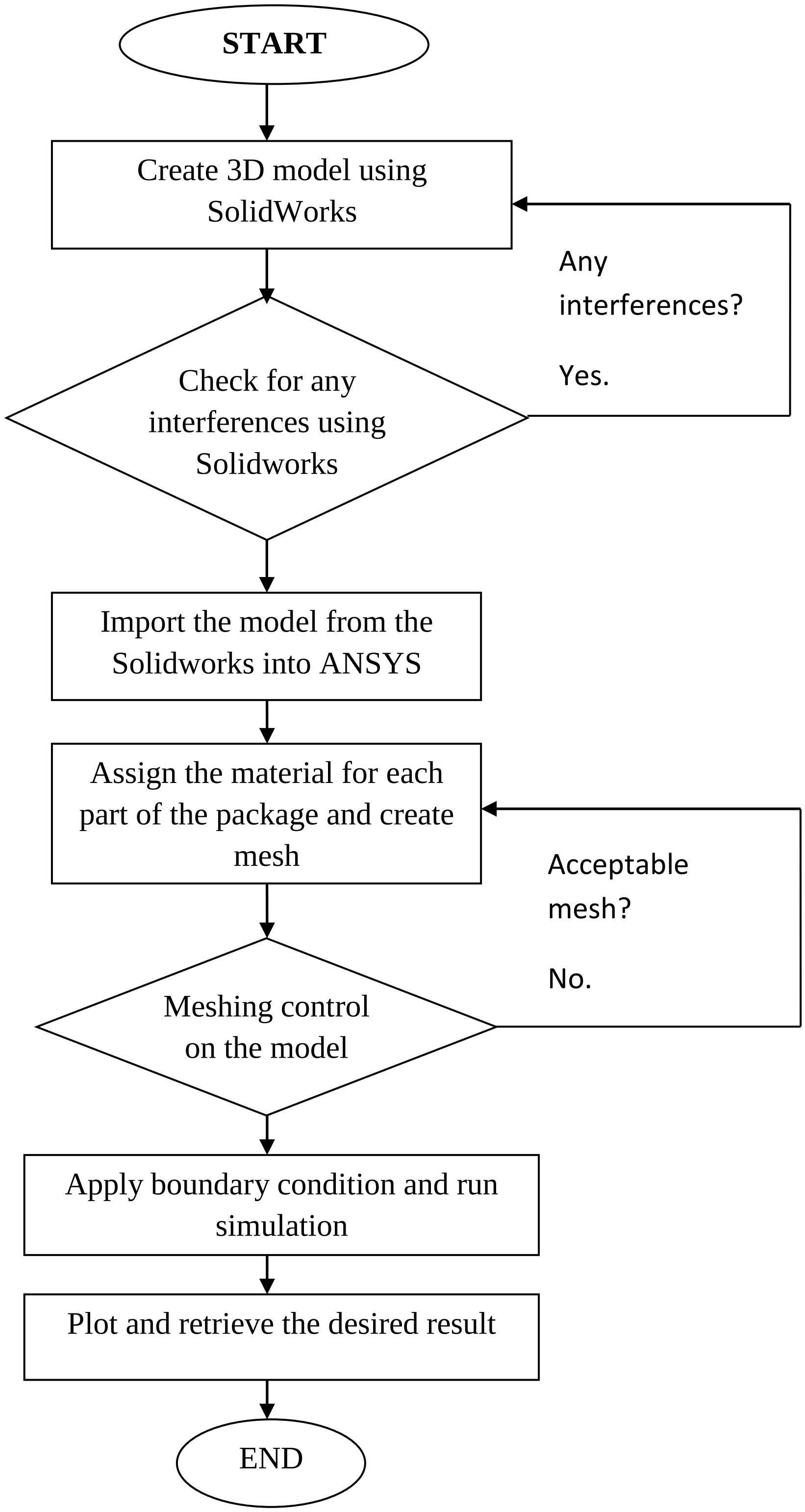


Figure 3.1: Project flow chart

3.1.2 Key Milestone

Project key milestone was established in order to ensure the project is on track. The project key milestones are shown in Table 3.2 and Table 3.3 for FYP 1 and FYP 2, respectively.

Table 3.2: FYP 1 key milestone

Event	Period (Week)	Responsibility
Project title selection	1-2	Choose a project title to work on from a list of title provided.
Project title approval	2-3	Submission of project title. The title approved is Stress Analysis of a Defected Gear Pair Using Finite Element Analysis (FEA)
Submission of Extended Proposal	3-6	Prepare an extended proposal and submit to the coordinator and supervisor.
Proposal defence	6-8	Conduct presentation; verbally report the progress of the project to the supervisor and examiner.
Submission of interim report first draft	12-13	Show supervisor the first draft of interim report.
Submission of interim report	14	Complete the interim report and submit to the supervisor.

Table 3.3: FYP 2 key milestone

Event	Period (Week)	Responsibility
Progress report submission	8	Complete progress report and submit to the coordinator. Coordinator will submit to the supervisor.
Pre SEDEX poster and presentation	11	Poster presentation to the external examiner.
Submission of dissertation draft	12	Show supervisor the first draft of dissertation.
Submission of dissertation (soft bound)	13	Submit the dissertation in soft copy to the supervisor.
Submission of technical paper	13	Submit the technical report in soft copy to the supervisor.
Oral Presentation	14	Conduct presentation; verbally report the finding of the project to the supervisor and examiner.
Submission of dissertation (hard bound)	15	Submit the final dissertation to the supervisor.

3.1.3 Gantt Chart

A proper project plan has been developed to ensure it can be completed within 28 weeks time. The project Gantt charts are shown in Table 3.4 and Table 3.5 for FYP 1 and FYP 2, respectively.

Table 3.4 FYP 1 Gantt chart

Activity/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
FYP 1 first briefing														
Project selection														
First meeting with supervisor														
FYP 1 second briefing														
Literature review														
Working on extended proposal														
First draft of extended proposal														
Submission of extended proposal														
Literature review continue														
ANSYS training														
Proposal defense														
ANSYS training														
Interim report														

Table 3.5 FYP 2 Gantt chart

Activity/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Design healthy gears in SolidWorks														
Import design into ANSYS														
Set up boundary condition														
Simulation of FEA Model														
Validation with Hertz theory														
Design pitting on the gear														
Design wear on the gear														
Simulate the models														
Compare all the data gathered														
Brief analysis on the data collected														
Final documentation														

3.1.4 Modeling and simulation of gear

Modeling of healthy gear was done using the SolidWorks software. In this project, 1:1 gear ratio is used. After each pinion and gear was modeled, an assembly was created in SolidWorks as shown in Figure 3.2. For this project, the gear design will follow the dimension and specification as shown in Table 3.6:

Table 3.6: Dimension and specification for the involute spur gear pair

Parameter	Pinion	Gear
Number of Teeth	12	12
Module	2.5	2.5
Material	Grade 1 steel	Grade 1 steel
Diameter of Pitch Circle (mm)	91.5	91.5
Diameter of base circle (mm)	79.0	79.0
Diameter of Addendum circle (mm)	86	86
Diameter of Dedendum circle (mm)	101.5	101.5
Face Width (mm)	20	20
Bore diameter (mm)	27.46	27.46
Young's Modulus (GPa)	210	210
Poisson's ratio	0.3	0.3

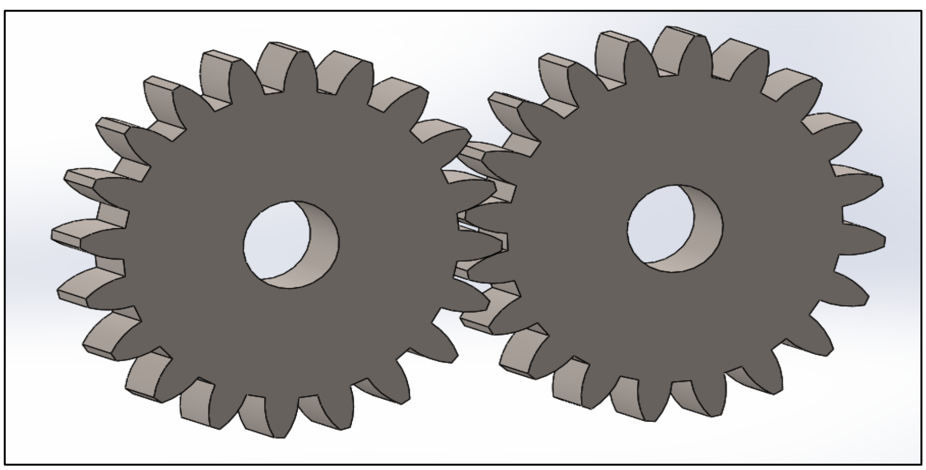


Figure 3.2: Assembly of spur gear pair in SolidWorks

The gear mesh configuration for FEA is shown in Figure 3.3. The model consisted of two gears, the pinion and the driven gear.

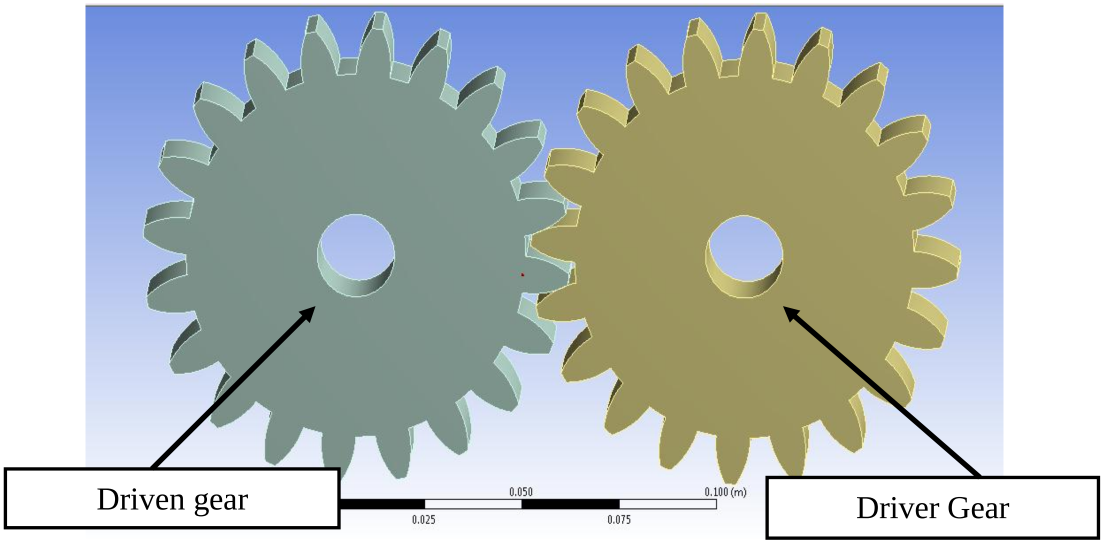


Figure 3.3: Gear mesh configuration

The material used was steel. The driver will be applied a moment of 150 Nm and the driven gear will rotate accordingly. A body-to-ground support was applied at the centre of both gears so that it only allowed rotational motion in Z axis. X, Y, Z displacement will be fixed for both gears since there is no translational movement involved (one degree of freedom).

Defining the contact region is the main step in simulating the gear using FEA. Once the geometry is attached with static structural analysis tab, we must define the contact between the two involutes teeth. ANSYS has inbuilt option, which will read the attached geometry automatically for any predefined contacts or other boundary definitions as shown in Figure 3.4.

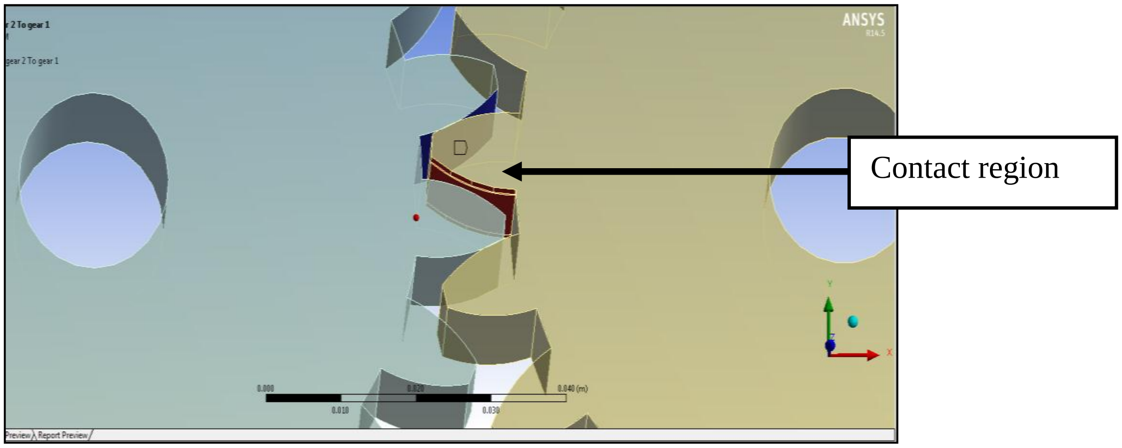


Figure 3.4: Contact region between two gears

Applied boundary conditions can be seen in Figure 3.5 and Figure 3.6. The model will be simulated in static structural analysis. From this analysis, maximum contact stress can be obtained.

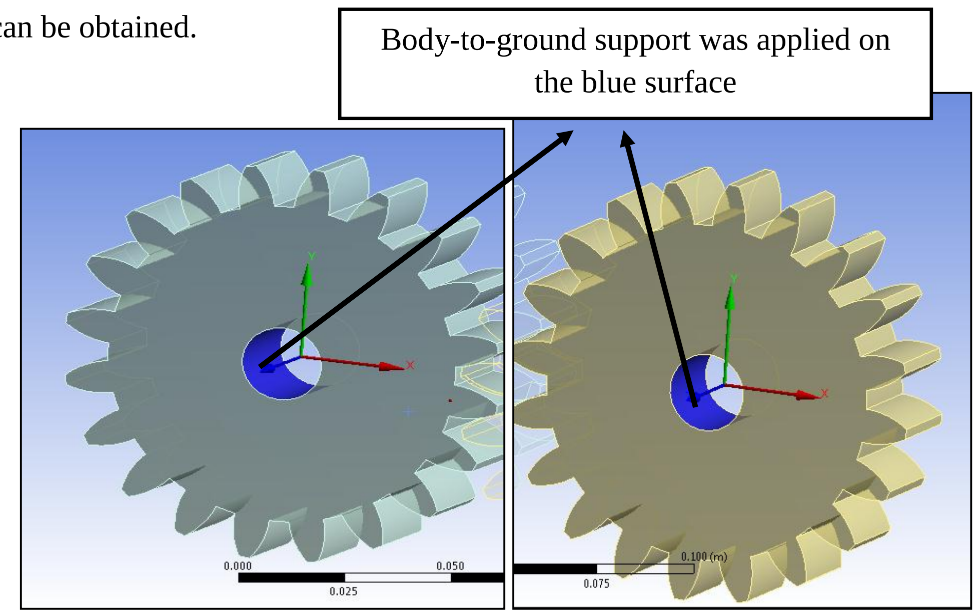


Figure 3.5: Applied boundary conditions to the gears; Body-to-ground supports

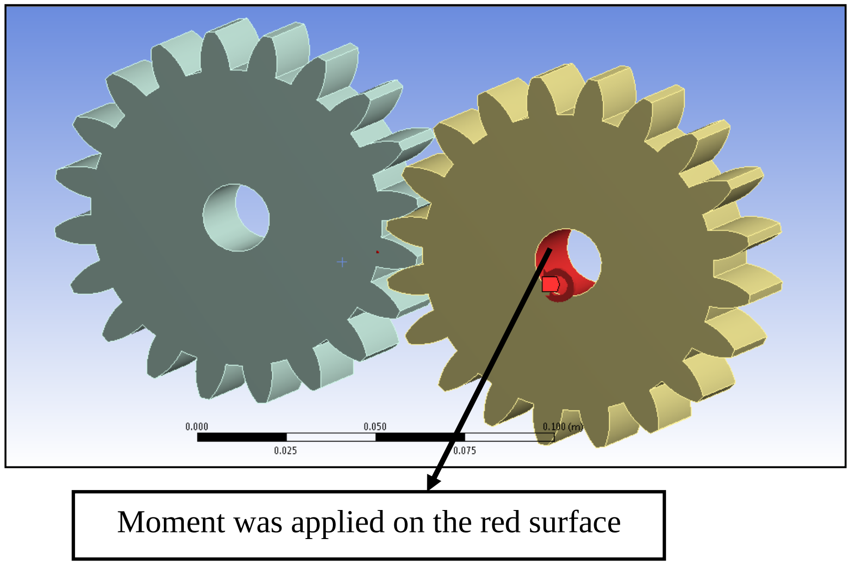


Figure 3.6: Applied boundary conditions to the gears; Moment to the driver

Supports and moment are applied to the both side of driver and driven gear. There are a total of 2 faces of support and 1 face of moment.

3.1.5 Designing failure modes

There will be three designs in the gear assembly, the healthy gear design, pitting design and wear design. The defects will be designed on a tooth of the driver gear. The affected teeth location is shown in Figure 3.7.

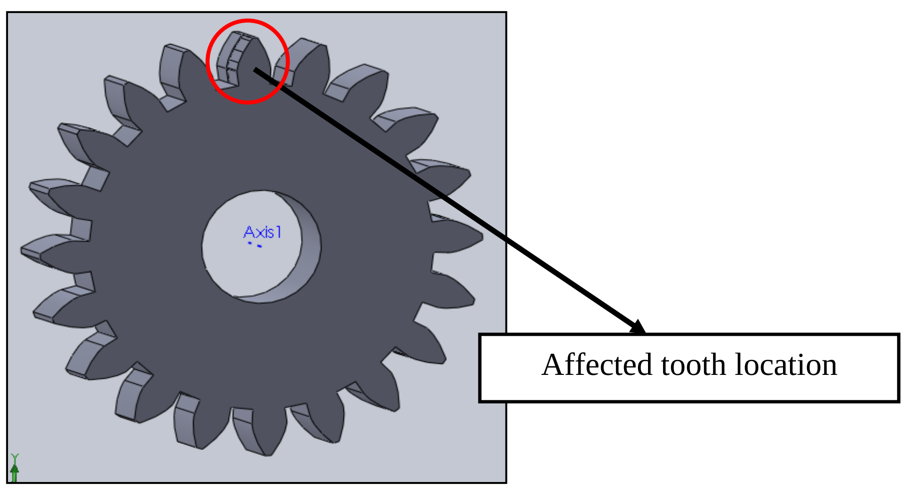


Figure 3.7: Affected tooth location

Pitting Design

Figure 3.8: Pitting formation on gear teeth [10]

Pitting is a surface fatigue failure due to many repetitions of high contact stress, which occurs on gear tooth surfaces when a pair of teeth is transmitting power. Pitting formation in gear is shown in Figure 3.8. In designing the pitting, the tooth surface area was decreased by percentage. Table 3.7 shows the area reduction values. The area reductions were made by removing parts from the surface of the teeth, as shown in Figure. 3.9.

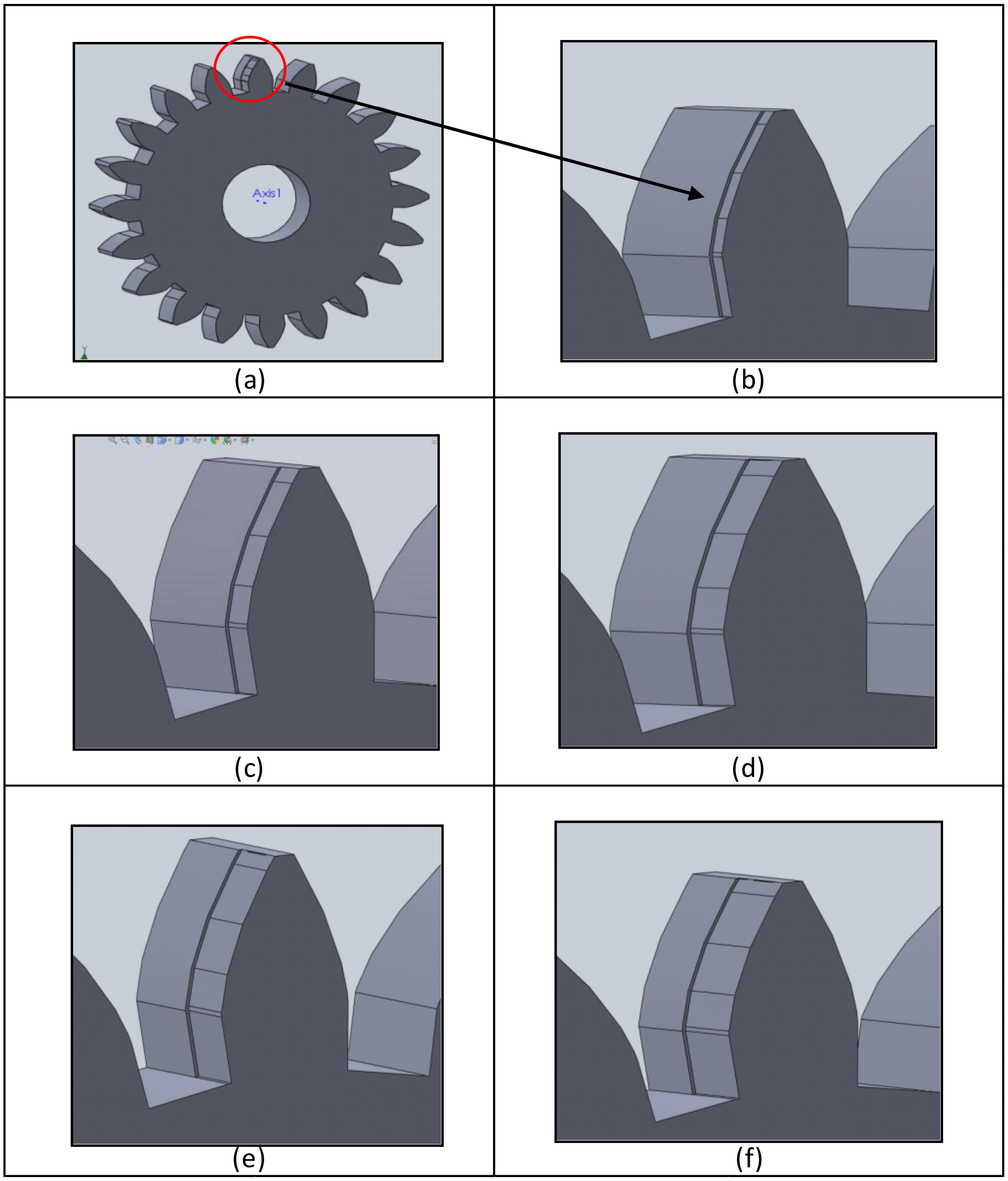


Figure 3.9: Surface area reduction by percentage - a) Full figure of the pitting, b) 10%, c) 20%, d) 30%, e) 40% and f) 50% reduction

Table 3.7: Surface area reduction

Area removal, %	Area reduction, mm^2 Width x thickness
10	2.5 x 0.1
20	5.0 x 0.1
30	7.5 x 0.1
40	10.0 x 0.1
50	12.5 x 0.1

Wear Design

Figure 3.10: Gear undergoing wear failure

Wear is related to the interactions between surfaces and more specifically the removal and deformation of material on a surface as a result of mechanical action of the opposite surface. Wear in gear is shown in Figure 3.10.

For the wear analysis, defected part was presented by the reduction of tooth width. Some reduction in the teeth width has been made in order to represent wear formation on the gear tooth. The order of the teeth volume reduction was in increasing order. For this analysis, the reduction of the width is made by percentage, namely 10%, 20%, 30%, 40% and 50% as shown in Figure 3.11. The value for area removal is shown in Table 3.8.

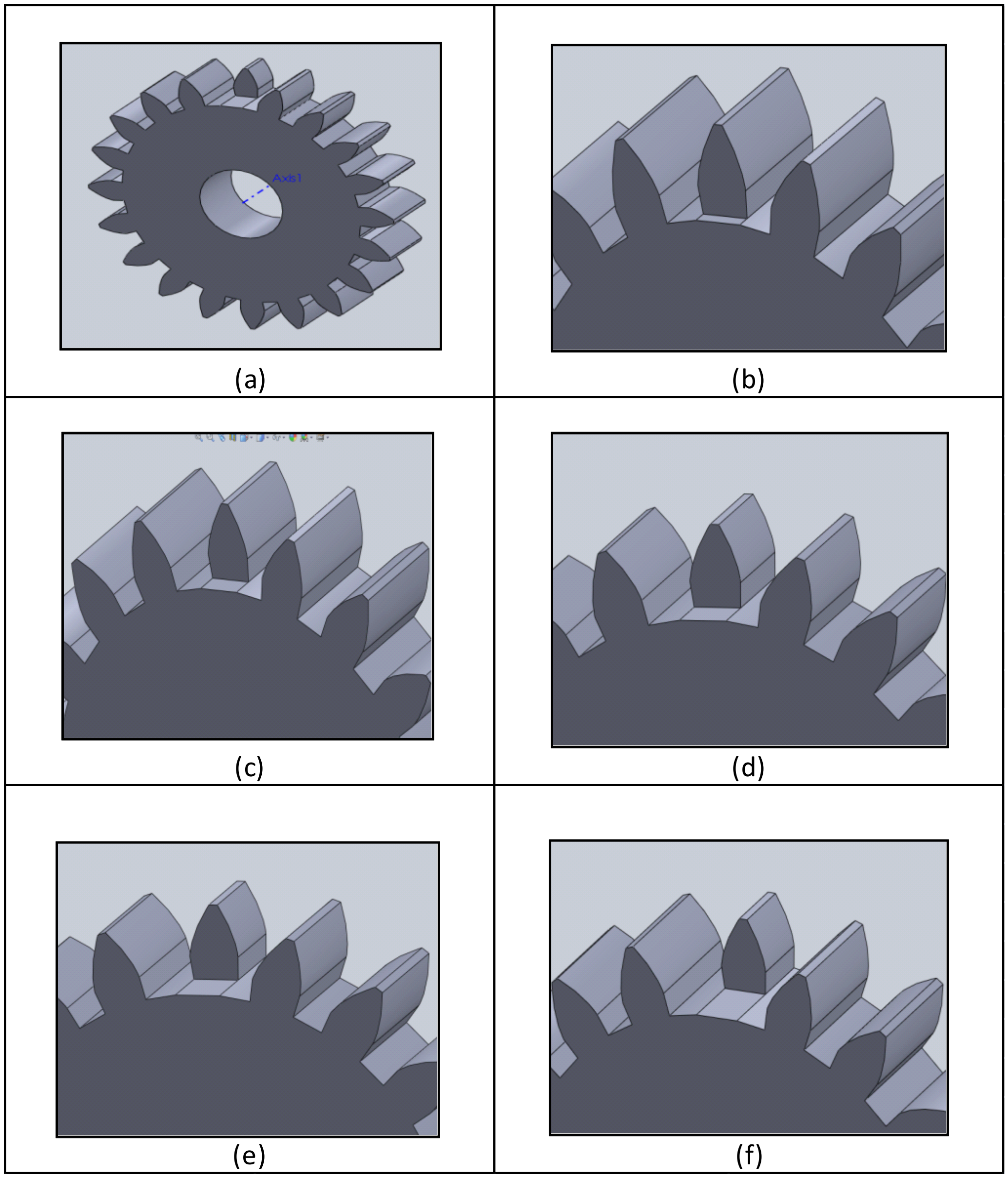


Figure 3.11: Tooth width reduction by percentage- a) Full figure of the wear gear, b) 10%, c) 20%, d) 30%, e) 40% and f) 50% reduction

Table 3.8: Width removal of wear

Width removal, %	Width reduction, mm
10	2.5
20	5.0
30	7.5
40	10.0
50	12.5

After the simulation, the maximum contact stress was recorded. The trend of the maximum contact stress is plotted in graph.

3.1.6 Data Collection and Analysis

For each type of failure, the maximum contact stress was obtained. The maximum stress was recorded for every case.

After all the results were collected, they were compared in one line graph. An analysis was carried out to determine the defect that gives higher contact stress. From the results and analysis, we can conclude which type of failure is more severe to the gear mechanism.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Healthy gear simulation

For FEA analysis, the maximum contact stress value was considered in order to determine the failure type that gives the greater detrimental effect to the gear. Figure 4.1 shows the maximum contact stress pattern for the simulation of healthy gear. The maximum equivalent stress obtained from FEA model was 4.813 MPa. The equivalent stress pattern results by the simulation are comparable to the pattern results by Kumar and Tiwari as shown in Figure 4.2.

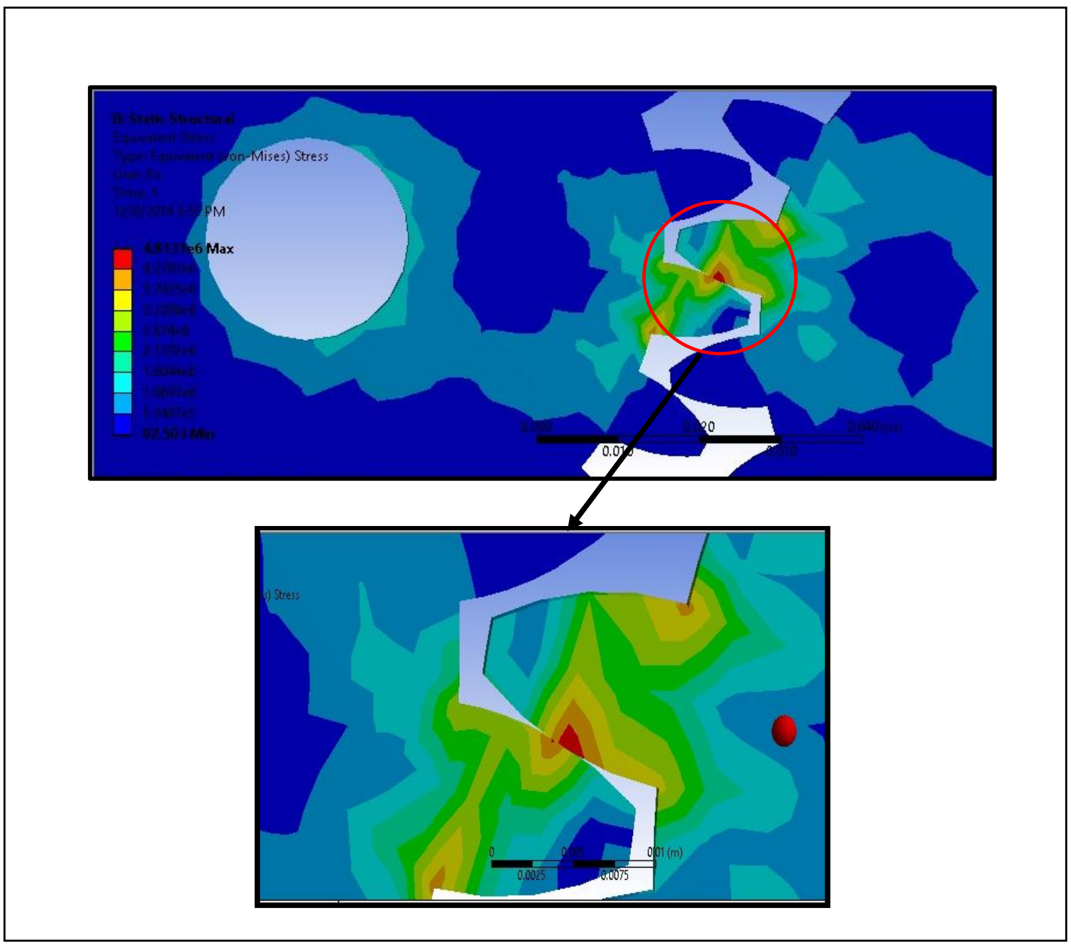


Figure 4.1. Equivalent stress pattern and maximum contact stresses from the simulation of healthy gear

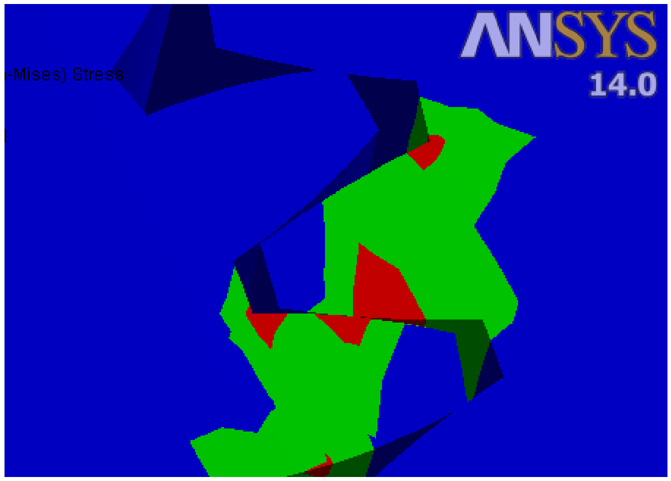


Figure 4.2: Equivalent stress pattern by Kumar and Tiwari [4]

The results pattern is comparable to the results of Kumar and Tiwari even though the value is not the same. This is because the geometry of finite element model was not exactly the same as the literature model and the shape of the teeth was different due to lack of gear specification information. Besides, the meshing of these two gears also did not follow the meshing of the literature due to some error in simulation results when using the Kumar and Tiwari simulation setup.

The next step is to design the faults on the driver gear, first will be pitting formation, followed by wear defect. The fault area was increased and the maximum contact stresses were recorded. The results were compared in order to see the difference in the maximum contact stress of a single tooth.

4.2 Maximum Contact Stresses

The maximum contact stresses for healthy gear, gear with pitting and gear with wear were obtained from the simulation. Figure 4.3 shows the maximum contact stresses for the healthy gear condition. The main focus is to identify the maximum contact stress for every cases and its trend. The results of the simulation of pitting and wear were also plotted as in Figure 4.3 and Figure 4.4. The percentage of difference between the healthy gear and defected gear is calculated as;

$$\% \text{ of difference} = \frac{\text{Max contact stress of defected gear} - \text{Max contact stress of healthy gear}}{\text{Max contact stress of healthy gear}} \times 100\ \%$$

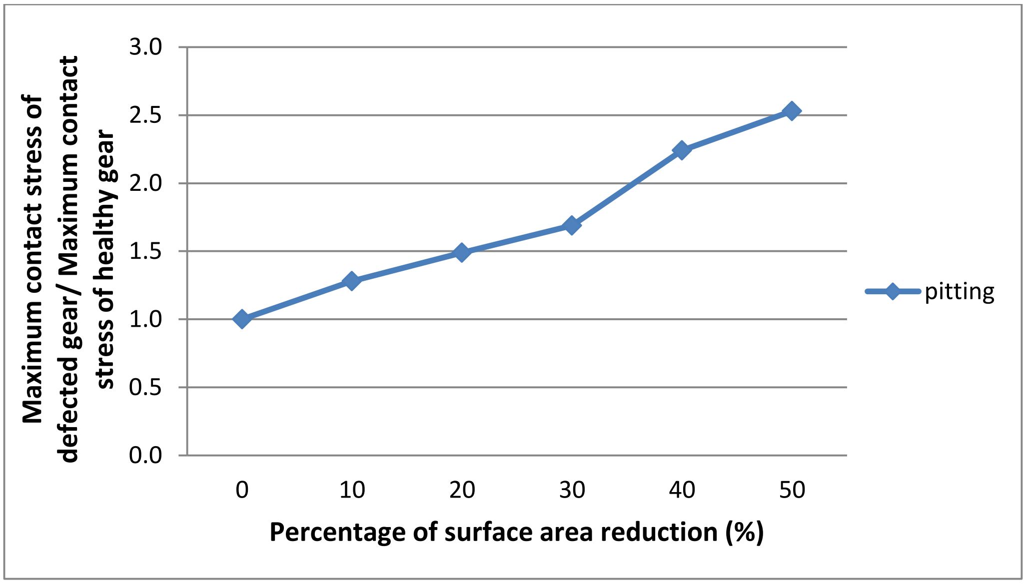


Figure 4.3: Maximum contact stress ratio for pitting defect

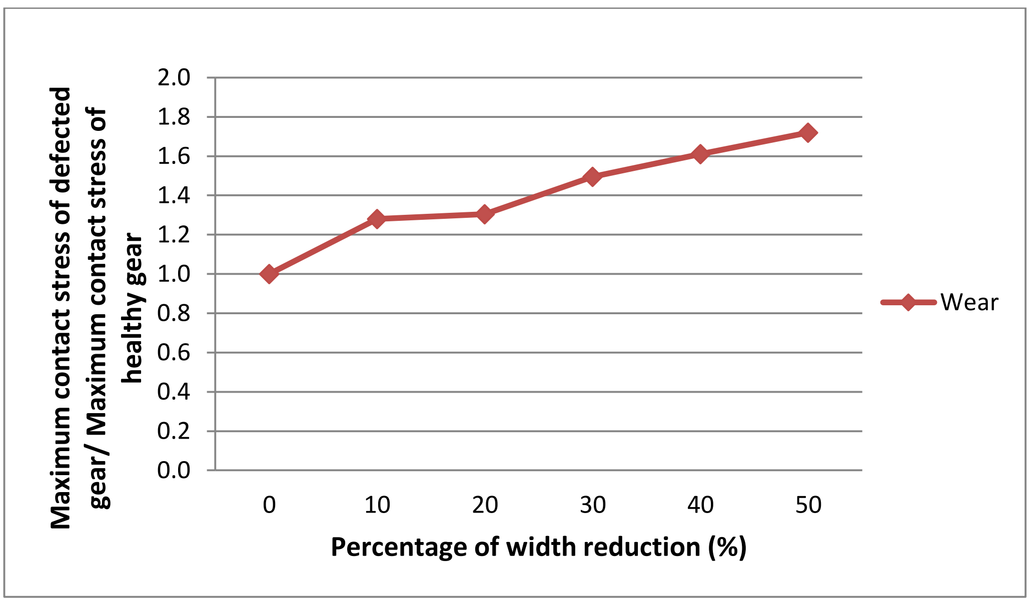


Figure 4.4: Maximum contact stress ratio for wear defect

Table 4.1: Results of maximum contact stress for pitting defect

Area reduction (%)	Maximum contact stress (MPa)	Percentage of increase (%)
0	4.813	0
10	6.182	28.01
20	7.204	49.61
30	8.147	69.35
40	10.772	123.82
50	12.166	152.77

Table 4.2: Results of maximum contact stress for wear defect

Width reduction (%)	Maximum contact stress (MPa)	Percentage of increase (%)
0	4.813	0
10	6.181	28.42
20	6.279	30.45
30	7.205	49.69
40	7.755	61.13
50	8.274	71.91

The summary of the maximum contact stress ratio for gear with pitting and wear are shown in Table 4.1 and Table 4.2, respectively. The tables show the changes in maximum stress when the fault area increased. These values will be able to tell which failure mode is riskier to the gear mechanism. Higher contact stress will be more risky to the gear system mechanism as it will lead to gear failures. Theoretically, the contact stress should be increasing when the pitting and wear area increased. This is because the contact stresses will cause pitting and eventually this pitting cause detrimental effect to the the contact surface, hence contact stresses will increase. This leads to more pitting and this cycle will continue. Therefore more contact stress will cause more pitting and more pitting leads to high contact stress.

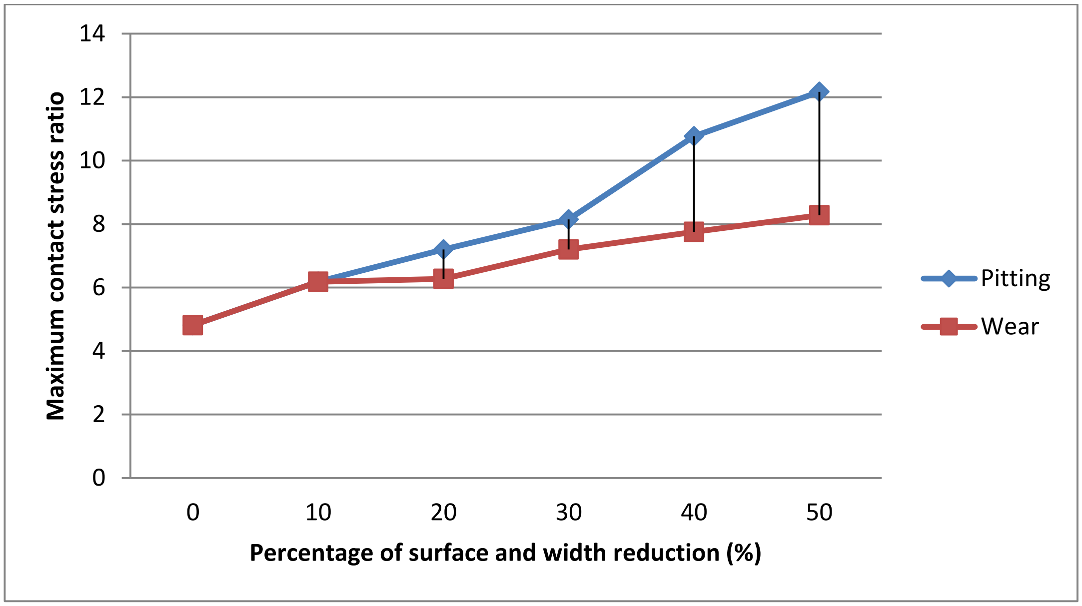


Figure 4.5: Comparison of maximum contact stress difference between pitting and wear

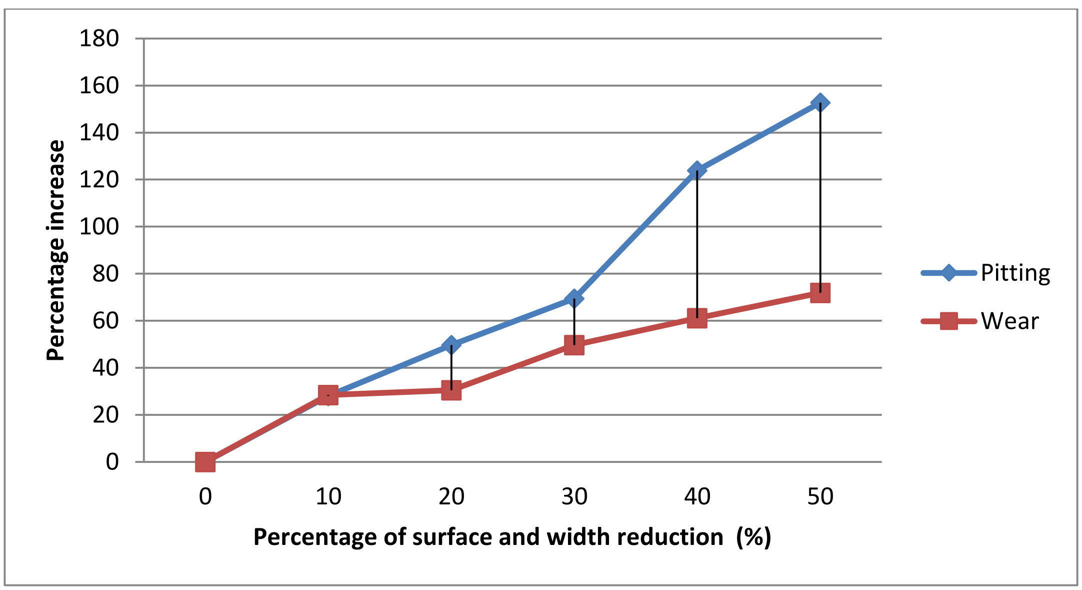


Figure 4.6: Comparison of percentage difference between pitting and wear

From Figure 4.5, it was determined that pitting experiences more contact stress to wear. In other word, the probability of pitting tooth to failure was higher than worn out tooth.

Finite element method had proved that pitting failure give riskier condition to the gear mechanism. From this finding, another research on preventing gear failure can be conducted. The priority will be focused on pitting problem first than the wear in order to save time and cost.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

As a conclusion, it is important to investigate the effects of gear failure modes to the gear mechanism. Different defects will affect the gear mechanism in different ways. In the case of contact stresses analysis, it is believed that, every failure mode will give different amount of contact stress. By using finite element method, the riskiest failure mode can be determined. In this study, it was concluded that the pitting tooth experienced more contact stresses compared to worn-out tooth. From the finding, a new research can be done on how to prevent the faults from occurring. The main focus will be on the one that gave the most severe damage to the system, which is the pitting. By preventing or slowing the progress of this failure mode, gear mechanism can keep functioning at longer lifespan. Focusing on the major fault will save time and cost. There are many other failures than pitting and wear. They are cracks, burrs, spalling, scuffing and erosion. It is recommended for the others to carry out simulation for other types of fault. Variation of failure modes will give better data and contain more information. Besides, by ranking them according to its maximum contact stress level, a priority list can be established. This will tell us which of the failure modes need to be taken care first. By considering other parameters, the analysis result will be more precise, accurate and comprehensive.

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