BENDING STRESS ANALYSIS OF HELICAL GEARS USING FINITE ELEMENT METHOD

By

OBAKENG LEVY SIKO

Submitted to the Department of Mechanical Engineering
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Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

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Approved:

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Dr Saravanan Karuppanan

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK

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CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Obakeng Levy Siko

ABSTRACT

This report presents the methodology and the approach taken in the analysis of a helical gear failure. It also presents the effects of bending stress on the failed helical gear, and how this bending stress changes with the change in tangential loading, on the gear tooth. The analysis presented in this report is one tooth analysis. The helical gear that is being analysed in this report is obtained from a gearbox of bus, which was experiencing gear failure. The report also presents the final results obtained from the analytical calculations, as well as the final results from the ANSYS simulation. Although there are a lot of different causes for gear failure, the contents of this report and the study conducted, is limited to failure caused by bending stress in helical gears. The report also presents the theory of the bending stress, and how it is used in analytical analysis of a helical gear failure. From the Lewis bending stress equation, the effects of the tangential loading are investigated (analytically). The report also contains a brief discussion on the obtained results from the Lewis bending stress equation, and also the ANSYS simulation results. The finite element analysis (FEA) results showed that the bending stress at higher tangential loads is higher at root of the gear tooth.

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CHAPTER 1
INTRODUCTION

1.1 Project Background

Gears are used in many industrial applications to transmit force, or cause motion. They are used in various engineering applications, and they simplify even the most complicated jobs, for example; gears are used in transmission gearbox of heavy vehicles like truck and buses, to provide motion. Like most mechanical components they fail under certain loading, due to different modes of failure. There are many ways in which a gear can fail, and there are a lot of causes for that particular gear failure. In most cases tooth breakage is usually the end result of a failed gear, although there are other ways for the gear to fail. One of the major causes of failure in gears is increased bending stresses. Investigating more on gear failures and the causes thereof can provide solutions as to avoid these failures from occurring in future.

1.2 Problem Statement

A helical gear (15˚ helix angle), from a transmission gear box of a bus, from a worldwide-accredited bus company has a problem. It was reported by the chief technician from the company, that the fourth speed helical gear of the bus had been damaged. The company requested a failure analysis study on the causes of the helical gear failure. The focus in this report is on the effects of bending stress on the failure of the above mentioned helical gear. The bus has a six-speed gearbox and transmits 275 kW at 2200 rpm engine power [1].

1.3 Objectives

The objectives of this study are:

(i) To study the effects of bending stress, for a failed 15 degrees helical gear of a bus.

(ii) To study the effects of tangential loading on the bending stress values for the same 15 degrees helical gear.

1.4 Scope of Study

The scope of this report covers, bending stress analysis of a 3D model of a helical gear, using ANSYS workbench software. The 3D model was developed using CATIA, and imported to ANSYS workbench for finite element analysis (FEA). A linear analysis of the 3D helical gear model was carried out using ANSYS. An analytical approach using Lewis bending stress equation was also carried out for the same helical gear. The results of the ANSYS workbench and the results from the analytical approach were compared, and a conclusion was reached.

CHAPTER 2
LITERATURE STUDY

A gear is a machine element designed to transmit force and motion from one mechanical unit to another [2]. Gears are used in wide range of applications to transmit power. There are different types of gears namely; helical gears, spur gears, bevel gears and worm gears. Gears used in power transmissions may fail in several different ways such as tooth breakage caused by high bending stresses in the gear teeth, scoring of the gear tooth surface due to an inadequate lubricant film, or surface pitting caused by high surface contact stress [3]. Normally spur gears are used for low speed power transmission (20m/s) and helical gears are preferred for high speed power transmission [4]. Helical gears are preferred for use at higher speeds, because there is a high total contact ratio which reduces noise and dynamic loads [5].

Helical gears are the modified form of spur gears in which, all teeth are cut at a constant angle, known as the helix angle, to the axis of the gear [6]. Helical gears are employed to transmit power between two shafts parallel to axis [6]. There are two types of helical gears, namely; helical gears and double helical gears.

Although there are different kinds of gears, and there are also different modes of failure that occurs in gears, the study and the research in this report are limited to helical gears. The focus is on the effects of bending stresses on helical gear.

Helical gears can be arranged as crossed helical gears as shown in Figure 2.1, and they are usually used to transmit power between two non-parallel shafts. The pressure angle and normal pitch should be the same between two crossed helical gears, in order for the gears to operate perfectly. Crossed helical gears are usually used in applications that involve very light loads. If a cross helical gear is accurately designed, the gears would also be able to function perpendicularly to each other.

Figure 2.1: Cross helical gear [7]

Double helical gears are cylindrical in form and contain two sections of teeth as shown in Figure 2.2. Double helical gears are actually two crossed helical gears joined together. This kind of gear provides a much better advantage, as compared to crossed-helical gears. The advantage is that the gear has helix angle slanting in different directions, this reduces the load parallel to the shaft, unlike in crossed-helical gears. Double helical gears provide a greater strength at the point of contact of gear teeth. Since the point of contact is increased the sideway force does not have any effect.

The focus in this project is on failures due to increased bending stresses. The analytical analysis in this report is carried out using Lewis Bending Stress equation for helical gears. The terms and the parameters in the equation are stated below.

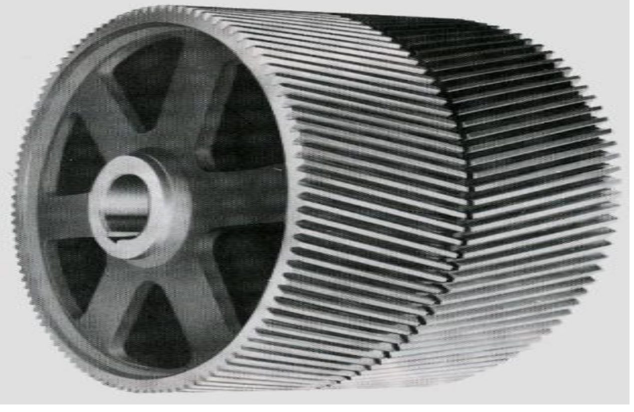

Figure 2.2: Double helical gear [8]

2.1 Force Analysis of a helical gear

Dealing with helical gears, there are four major forces we have to consider namely; total force (F), radial force (F_r), axial force (F_a) and the tangential load (F_t), refer to Figure 2.3 below to see the directions of the above mentioned forces.

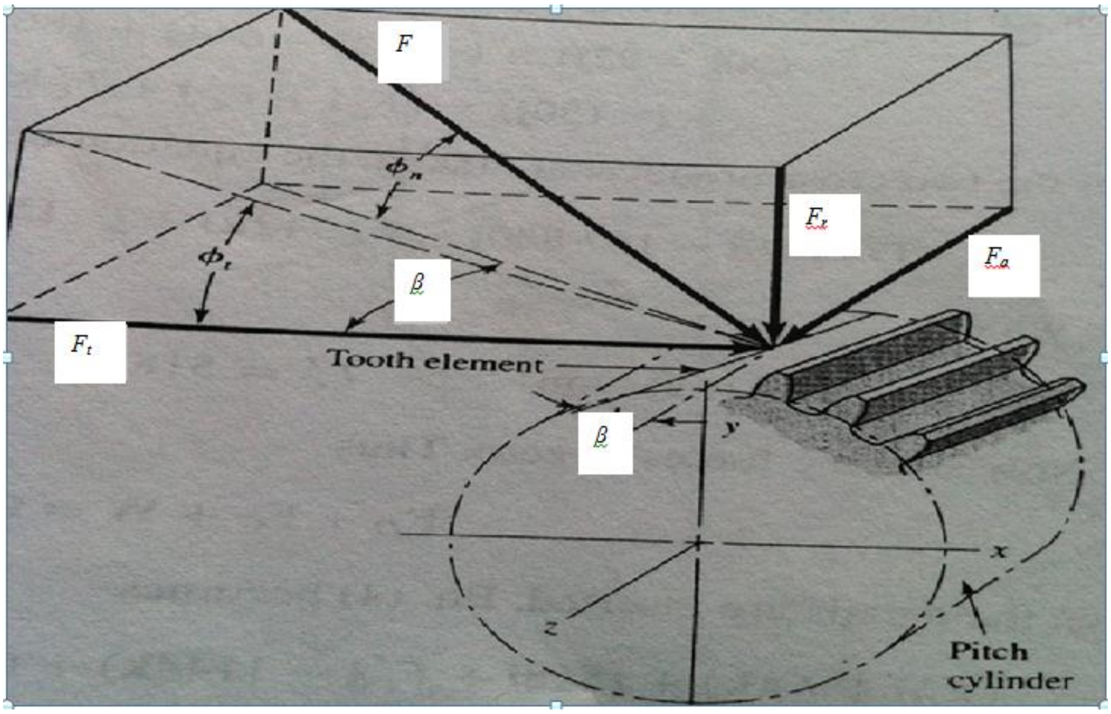

Figure 2.3: Forces on a helical gear [9]

2.2 Bending stress analysis

The Lewis bending stress equation for helical gears is used to determine the bending stress acting on the gear, for different tangential loads and different velocities.

Bending Stress (σ_b):

$$\sigma_b = \frac{F_t}{C_v b \pi Y}$$

where: F_t = the tangential loading (N)

b = face width (mm)

Y = Lewis factor

C_v = velocity factor

CHAPTER 3
METHODOLOGY

3.1 Methodology

Figure 3.1 shows the flow of the project.

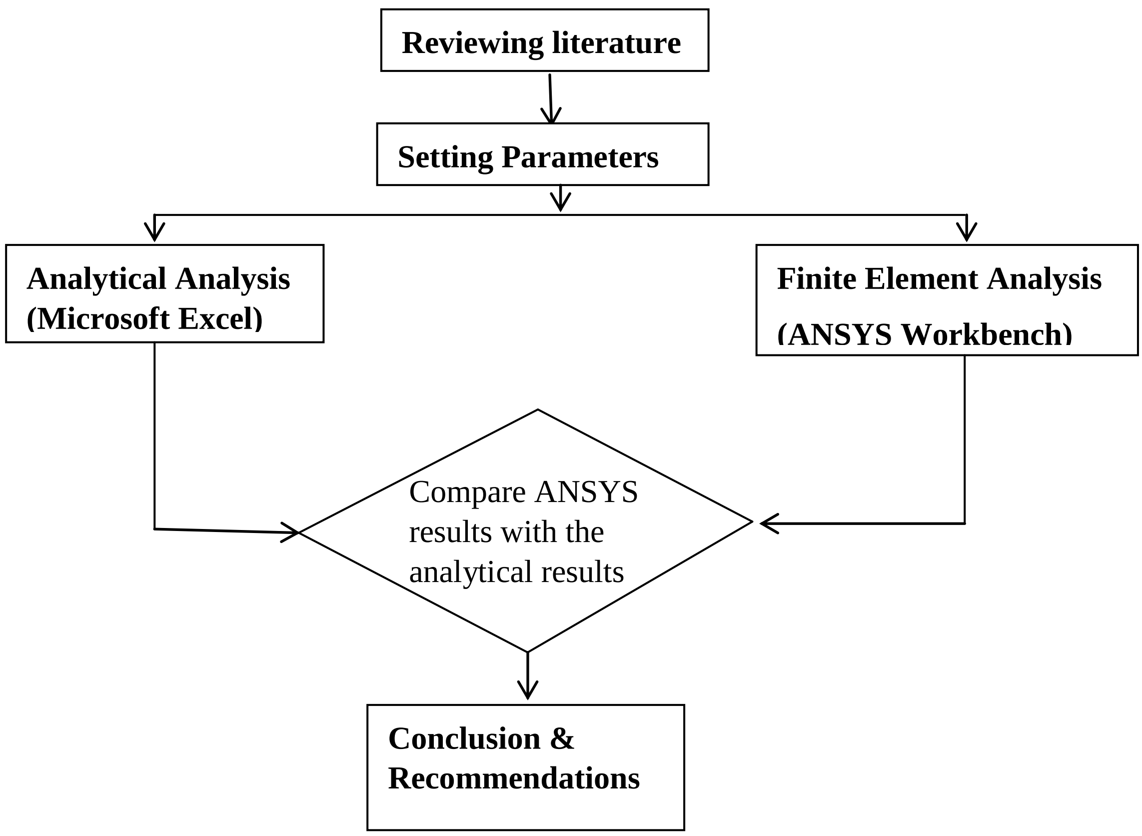

Figure 3.1: Methodology used during the project

3.2 Project Activities

(i) Reading and researching materials, related to helical gear.

(ii) Identifying the relevant mathematical model to be used in the analytical approach.

(iii) Model a helical gear (15˚ helix angle) in CATIA.

(iv) Solve for bending stresses using ANYS Workbench simulation.

3.3 Gantt chart and Key milestone

Table 3.1 shows FYP 1 Gantt Chart.

Table 3.1: FYP 1 Gantt chart

No	Detail/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Topic Selection / Proposal														
2	Preliminary Research Work														
3	Submission of Extended Proposal Report						★								
4	Proposal Defense; Oral Presentation							Mid semester break			★				
5	Project Work Continues														
6	Submission of Interim Draft Report													★	
7	Submission of Interim Report														★

★ Suggested Milestone

Process

Table 3.2 shows FYP 2 Gantt Chart.

Table 3.2: FYP 2 Gantt chart

No	Detail/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
2	Modeling and simulation							Mid semester break							
3	Submission of FYP 2 Progress Report								★						
4	Pre-SEDEX Poster Presentation											★			
5	Submission of Final Draft Report												★		
6	Submission of Dissertation (softbound)													★	
7	Submission of Interim Report														★

★ Suggested Milestone

Process

3.4 The programs and software used in this project are:

(i) CATIA

(ii) ANSYS Workbench

(iii) Microsoft Excel

(iv) Microsoft Word

CHAPTER 4
RESULTS AND DISCUSSION

4.1 Analytical Approach

The helical gear considered in this case study is a fourth speed gear [gear ratio, 1:1], having the properties shown in Table 4.1.

Table 4.1: Helical gear properties and parameters

Number of teeth	**36**
Helix angle (*β*)	15˚
Pressure angle, normal (θ_n)	20˚
Module, normal (m_n)	4 mm
Pitch circle diameter (*d*)	150 mm
Outer diameter (d_0)	158 mm
Power (*H*)	275 kW
Rotational Speed(*n*)	2200 rpm
Lewis factor (*Y*)	0.375
Yield Strength (σ_y)	833 MPa

In order to determine the bending stress, of the above stated helical gear, we will first determine all the parameters for the Lewis bending stress equation.

Torque (*T*):

$$T = \frac{H}{n}$$

$$= \frac{275 \times 10^3}{2\pi(\frac{2200}{60})}$$

$$= 1193.68 \text{ Nm}$$

Tangential load (F_t):

$$F_t = T \times \frac{2}{d}$$

$$= (1193.68) \times \frac{2}{0.15}$$

$$= 15915.47 \text{ N}$$

Transverse module (m_t):

$$m_t = \frac{m_n}{cos(\beta)}$$

$$= \frac{4 \text{ mm}}{\cos(15°)}$$

$$= 4.14 \text{ mm}$$

Transverse Pitch (P_t):

$$P_t = \pi m_t$$

$$= \pi(4.14 \text{ mm})$$

$$= 13.006 \text{ mm}$$

Axial Pitch (P_a):

$$P_a = \frac{P_t}{tan(\beta)}$$

$$= \frac{13.006 \text{ mm}}{\tan(15)}$$

$$= 48.5 \text{ mm}$$

Face width (b):

$$b = 1.15 P_a$$

$$= 1.15(48.5 \text{ mm})$$

$$= 55.8 \text{ mm}$$

Pitch line velocity (v)

$$v = \pi dn$$

$$= \frac{\pi(149.079)(2200 \text{ rpm})}{60}$$

$$= 17172.668 \text{ mm/s}$$

$$= 17.17 \text{ m/s}$$

Velocity factor (Cv):

$$C_v = \frac{5.55}{5.55\sqrt{v}}$$

$$= \frac{5.55}{5.55(\sqrt{17}.173)}$$

$$= 0.57$$

Bending Stress (σ_b)

$$\sigma_b = \frac{F_t}{C_v b\pi Y}$$

$$= \frac{15915.467N}{(0.57)(55.8\text{mm})(\pi)(0.375)}$$

$$= 424 \text{ MPa}$$

The values of the bending stress for different intervals were determined for same 15 degrees helical gear, using an increasing tangential load (F_t), and the values are shown in Table 4.2

Table 4.2: Analytical Results: Tangential loading and bending stress

Tangential Load (N)	**13915**	**14915**	**15915**	**16915**	**17915**
Bending Stress (MPa), Analytical	370.37	396.99	424	450.22	476.84

4.2 Finite Element approach (ANSYS WORKBENCH)

A 15 degree helical gear was modeled in CATIA, and it was then imported into ANSYS WORKBENCH. The helical gear was specified to be a fourth speed drive gear (1:1 gear ratio). The value of the tangential load (15915 N) obtained from the analytical approach is applied on the edge of one tooth of a helical gear having the same geometry as the one specified in the problem statement. Then the Equivalent Von-Mises stresses were obtained from the solver in ANSYS WORKBENCH as shown in Figure 4.1.

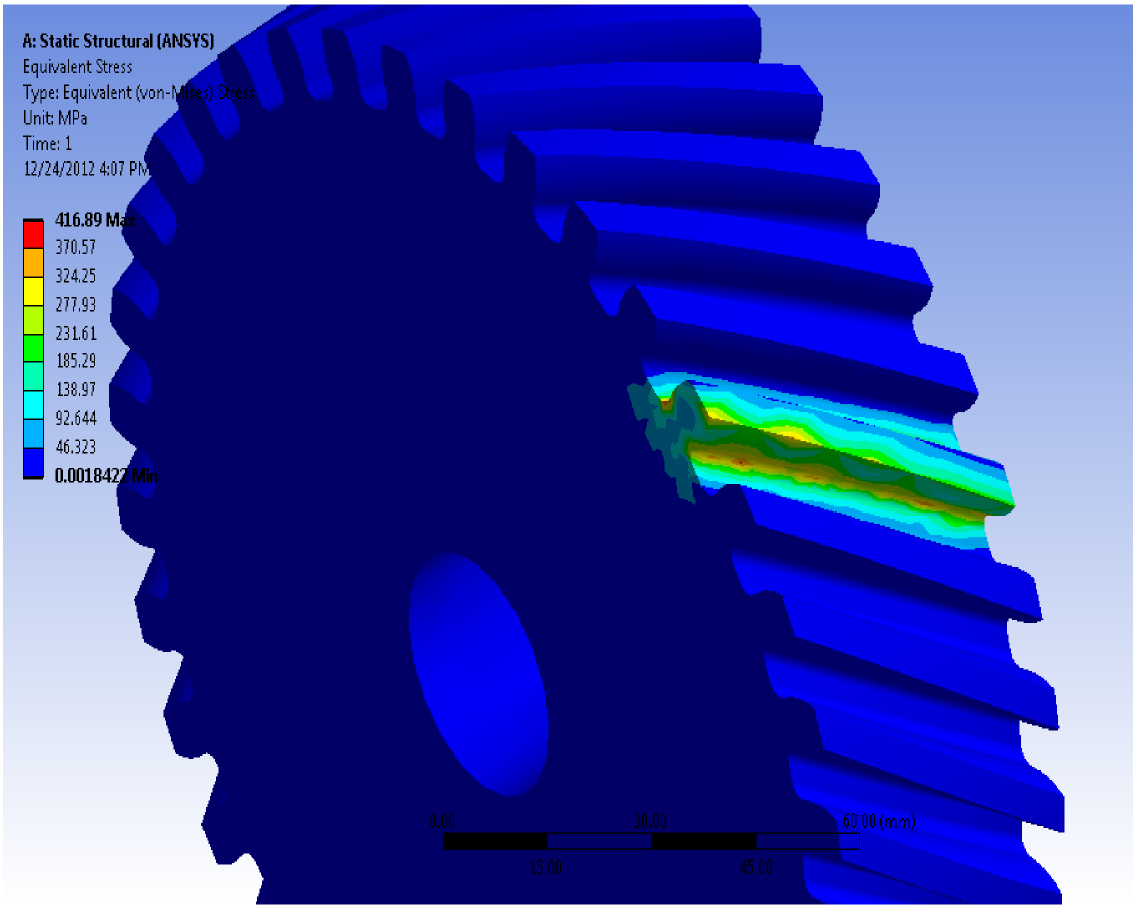

Figure 4.1: ANSYS WORKBENCH results

From the results obtained from the ANSYS simulation shown in Figure 4.1 one can also observe that the bending stress value is quite high at the root of the gear tooth, which will most likely be the point of failure at higher tangential loads.

Similar to the approach taken in the analytical approach, the tangential load was varied for different intervals, and the stress values were tabulated in Table 4.3

Table 4.3: ANSYS WORKBENCH Results: Tangential load and bending stress

Tangential Load (N)	**13915**	**14915**	**15915**	**16915**	**17915**
Bending Stress (MPa), FEA	364.5	390.7	416.89	443.09	469.29

The stress values obtained from the analytical approach, and the values obtained from the Finite Element Analysis (ANSYS WORKBENCH), were plotted on the same axis using Microsoft excel, as shown in Figure 4.2.

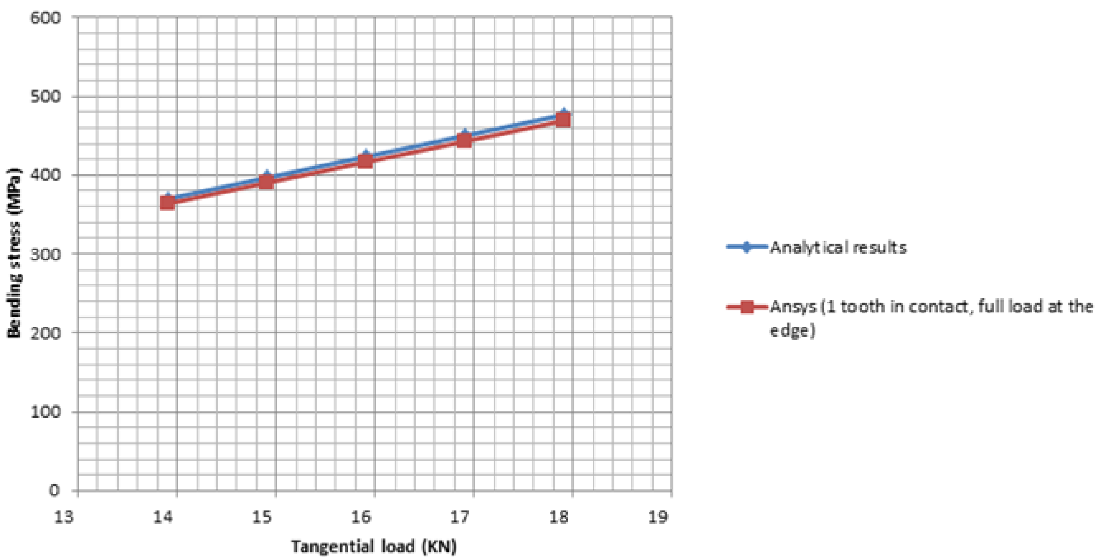

Figure 4.2: Bending Stress v. s tangential load: ANSYS results and Analytical results

From the graph in Figure 4.2, we can observe that, the relationship between the bending stress and the tangential loading is a directly proportional one. As the tangential loading increases, the bending stress on the gear tooth also increases. This is true for both the analytical and the ANSYS simulation results. The values obtained for the analytical approach are slightly higher than the values from the ANSYS simulation; this could be because of a number of reasons. The most probable cause of the difference in the ANSYS and analytical results is that, with the analytical approach using the Lewis bending stress equation, no correction factors were used, and in ANSYS they were used. . From the maximum stress value and the yield stress of the material, it can be concluded that at the tangential load of 15915N, the gear does not fail due to bending stresses.

Using the Lewis bending stress equation and the value of the material`s yield strength (σ_y=833 MPa) we can determine the maximum tangential load ($F_{t\ (max)}$) the gear tooth can withstand. This can be done by substituting the yield strength value in to the Lewis bending stress equation as shown in the calculation below.

$$\sigma_b = \frac{F_t}{C_v b \pi Y}$$

$$\begin{aligned} F_{t\ (\mathrm{max})} &= (833) \times (0.57)(55.8\mathrm{mm})(\pi)(0.375) \\ &= 312112.98\ \mathrm{N} \\ &= 31.2\ \mathrm{k\ N} \end{aligned}$$

From the value of the maximum tangential load obtained above, we can conclude that at tangential loads higher than ($F_{t\ (max)}$ =31.2 kN) the gear will fail due to bending stresses.

CHAPTER 5
CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

From the stress values obtained from both the analytical approach and finite element analysis (FEA), it is evident that the gear does not fail because of bending stresses. The reason for this is that, at the specified conditions (F_t= 15915 N), the value of the bending stress is 424 MPa, which is less than the yield strength of the material (833 MPa). Another important conclusion one can draw from the graph plotted in Figure 4.2. is that as the tangential loading on the edge of the helical gear tooth is increased, the bending stress is also increased. From the above mentioned observations, the final conclusion is that the specified helical gear did not fail by bending stresses.

5.2 RECOMMENDATIONS

In order to obtain better results of what truly caused the helical gear in the bus to fail, the company has to run a full failure analysis, instead of just bending stress analysis, because it is most likely that the helical gear failed due to fatigue bending. Fatigue takes time to occur, even if the same load is applied on the tooth over time.

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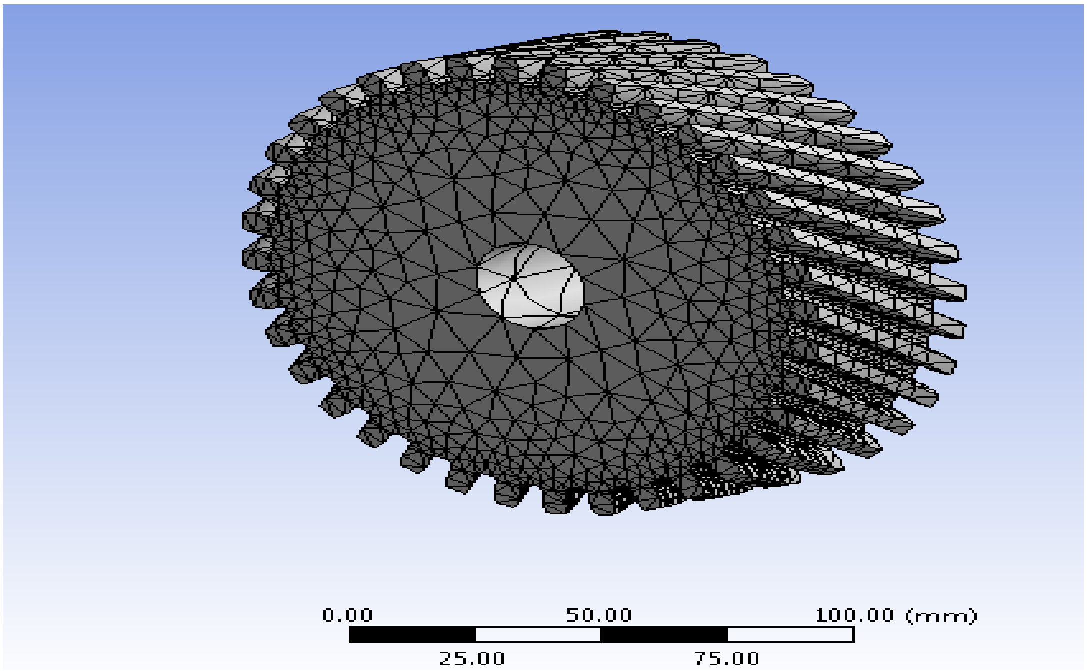
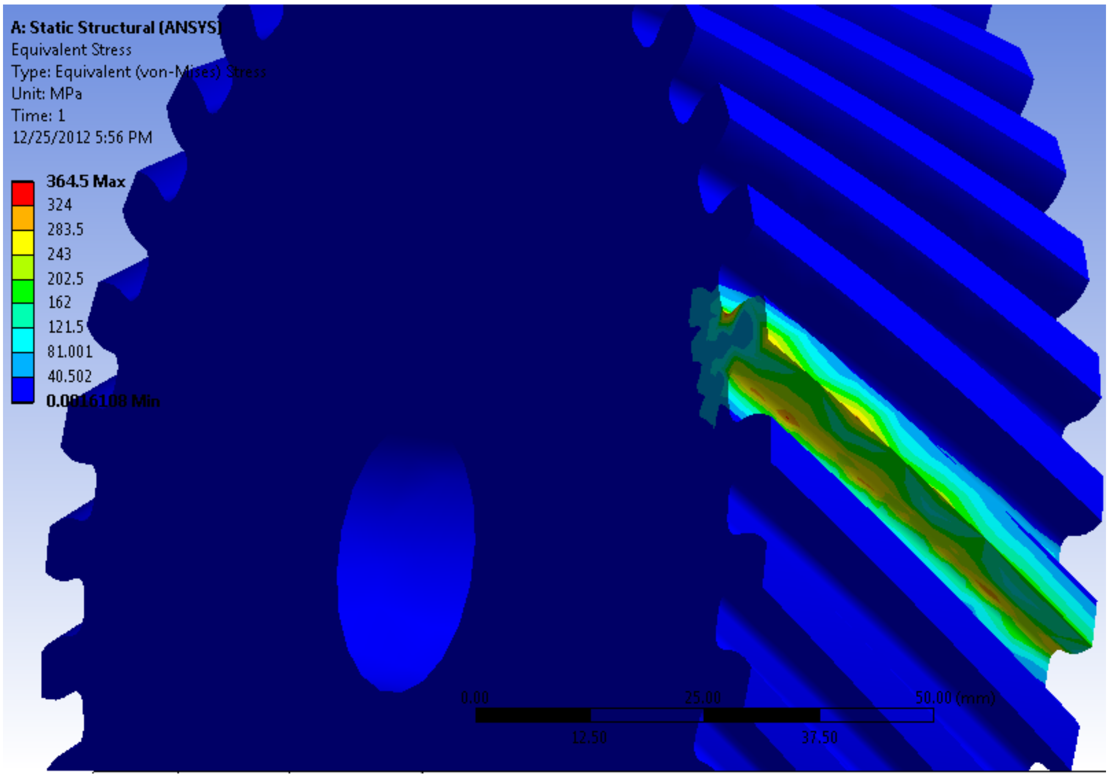
APPENDIX A

ANSYS RESULTS

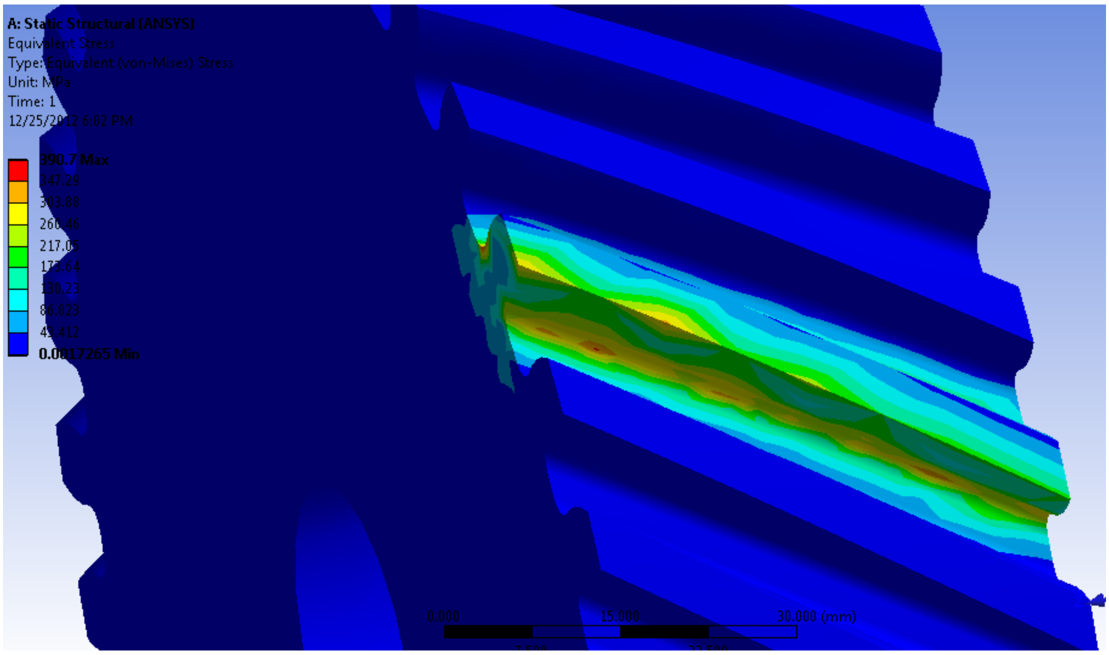
A.1. Helical gear mesh



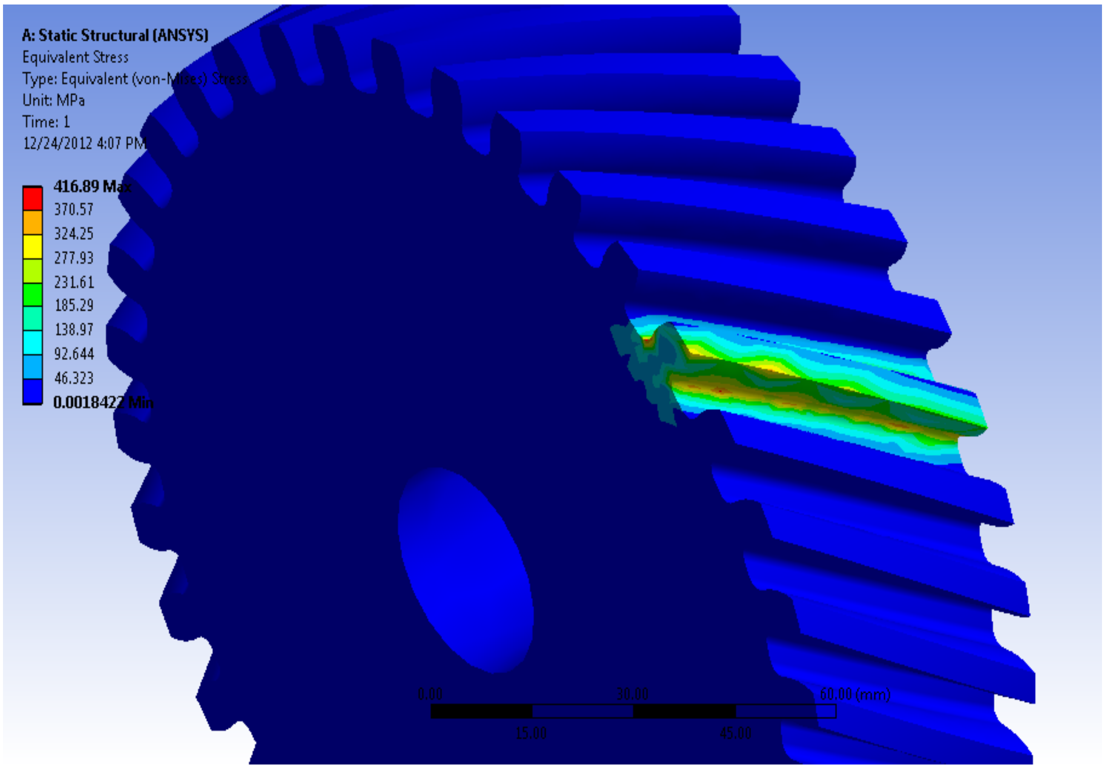
A.2. Helical gear (Tangential load = 13915N)



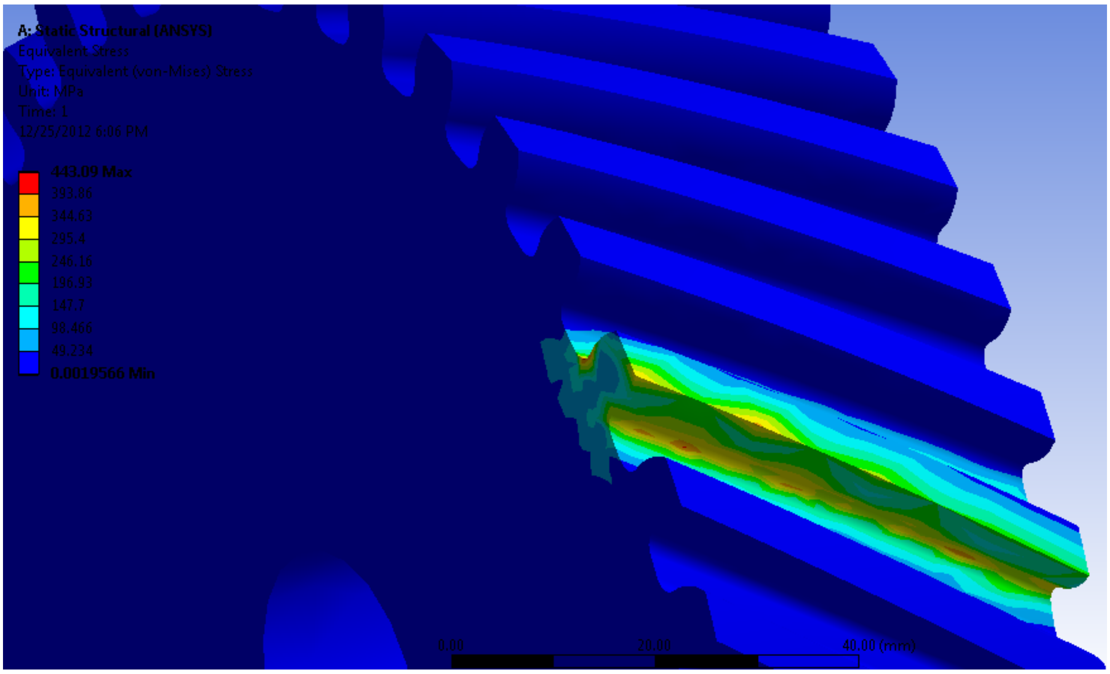
A.3. Helical gear (Tangential load = 14915N)



A.4. Helical gear (Tangential load = 15915N)



A.5. Helical gear (Tangential load = 16915N)



A.6. Helical gear (Tangential load = 17915N)