Comparison of Bending Stress and Contact Stress of Helical Gear Transmission Using Finite Element Method

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Abstract— **This research is an investigation on the fracture of a helical gear transmission which was to adjust the engine to increase the horsepower. The gears were made of chromium steel JIS-SCr 420. The bending stress and contact stress of the gear tooth are considered to be one of the main contributors for the failure of the gear in a gearbox. Thus, the analysis of stress has become popular as an area of research on gears to minimize or reduce the failure and for optimal design of gears In this paper bending and contact stress are calculated by using analytical method as well as finite element analysis. Bending stress is calculated by using Lewis bending equation and contact stress by using AGMA contact stress equation. Finite element method software package is used to analyze the bending stress and the contact stress. Finally these two methods bending and contact stress results are compared with AGMA equation. The results showed that the gear failed with a fatigue fracture, which the load of contact stress over took the contact strength of the material (1,550 MPa). The summary analysis results can be accorded with the assumption of this research and as an information for a protection of any failure gearing.**

Index Terms—**Bending stress, Contact stress, FEM, Helical gear**

I. Introduction

In Thailand, the pickup car is an important component in driving the economy. It is used in agriculture and industry. Therefore, when there is a demand for more weighted cargo, the pickup has to adjust the engine to increase power all over. But in reality it will affect the gear-box, propeller shaft and differential, which are not designed for higher torque. For example, in this case, the pickup has to adjust the engine to increase the horsepower. The gear-box was not designed for the higher torque. This results in a fracture in the helical gear in the 5^{th} position during the change from gear of 4^{th} to 5^{th} position. The helical gears were made from chromium steel (JIS-SCr 420/AISI 5120) and extensively processed under heat treatment.

The design of gear is a complex process. Generally it needs large number of iterations and data sets. In many cases gear design is traditional and specified by different types of standards [1]. Involute shaped gears found to be almost everywhere because of the contact forces which act along a straight line. Many researchers have studied to behavior of helical gears under different conditions [2].

Most of the damage is caused by a fracture of the teeth reacting with contact stress and bending stress. This phenomenon is called as ‘pitting’ or the holes on the gear teeth which exposed stress on the surface due to the backlash (contact stress). Analysis of the bending stress and contact stress is used in several methods. Analysis of the bending stress by AGMA bending equation and contact stress by AGMA contact equation show difficult [3]. One of the best methods is finite element method which can transmit power between the shafts through gears. Gears are mostly used to transmit torque and angular velocity.

Finite element method can simulate the stress induced in the teeth flank, teeth fillet during meshing of gears. The involute profile of helical gear has been modeled and the simulation is carried out for the bending and contact stress and the same has been estimated. To estimate bending and contact stress, three- dimension models for different tangential transmitted loads are generated by solid works 2014 and simulation is done by finite element software packages. Analytical method of calculate bending stress and contact stress uses AGMA equation. It is important to develop appropriate models of contact element and to get equivalent result using finite element method and compare the result with standard AGMA stress [4].

This paper aims at identifying the cause of failure of a helical gear in a pickup car in Thailand in order to prevent or minimize the prolong service life of the component.

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II. Experimental Procedure

A. Three Dimension Modeling of Helical Gears

Parametric of helical gears let the characteristic parameters of a product. The main parameter that would describe the helical gear such as module, pressure angle, tooth thickness, number of teeth could be used as the parameters to define the gear as shown on Table I [5].

TABLE I. Dimensions of Helical Gear [5]

Description	Symbol	Values	Unit
Power	H	120	kW
Speed	n_p	3200	rpm
Number of teeth pinion gear	N_p	26	-
Number of teeth drive gear	N_G	46	-
Pinion gear diameter	d_p	64	mm
Drive gear diameter	d_G	109	mm
Normal module	m	2.25	mm
Pressure angle	ϕ	30°	degree
Helix angle	ψ	20°	degree
Pitch diameter	d	103.5	mm
Pitch	P	58.5	mm
Gear face width	b	23	mm
Elastic coefficient	Z_E	191	N/mm^2

Transmission gears were modeled according to original dimensions of gears then they were assembled by Solid works 2014 and simulation was done by Ansys 14.5. It can be seen in Fig. 1 their Three-dimension model.

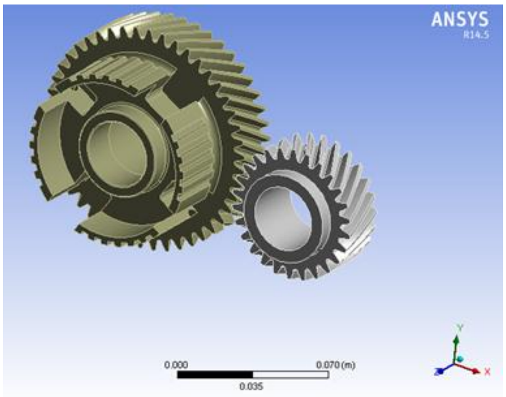

Figure 1. Three-dimension models of helical gear.

Stress analysis, we assumed that gears are working in maximum load condition. In the pickup car operation with transmission, required power was taken as 120 kW and revolution was 3200 rpm. According to a pickup car power and transmission ratios, moment of force have been accounted.

III. Results and Discussions

A. Bending Stress Analysis

The AGMA equation for bending stresses given by

$$\sigma_b = \frac{W^t K_o K_v K_s K_H K_B}{b m_t Y_J} \tag{1}$$

where W^t is tangential transmitted load = 10.60 kN

K_o is overload factor = 1.25

K_s is size factor = 1

K_H is the load distribution factor = 1.16

K_v is dynamic factor = 1.62

K_B is the rim-thickness factor = 1

b is the gear face width = 23 mm

m_t is the transverse metric module = 2.60 mm

Y_J is the geometry factor for bending strength = 0.48

$$\sigma_b = \frac{10.60\times10^3\times1.25\times1.62\times1\times1.16\times1}{23\times2.60\times0.48}$$

$$= 867.45 \text{ N/mm}^2$$

Results from the bending stress analysis showed that (σ_b) is 867.45 MPa. From Schmid et al. the stress at base on the results from the analysis, it was found to be higher than the bending strength for the gear material is 450 MPa [10]. The calculated bending stress due to normal operation was 1.92 times higher than the allowable bending strength. The presence of the surface therefore cannot be, on its own, responsible for the fatigue crack [6].

Analyses bending stress using finite element method.

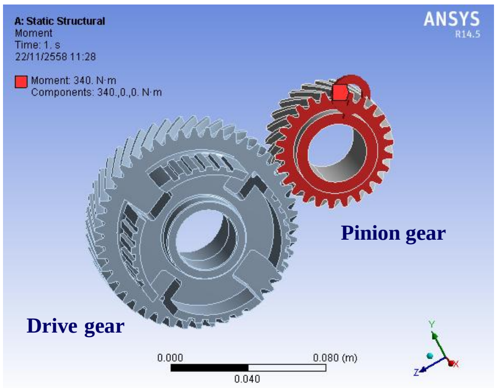

Figure 2. Fixed from drive gear and set the torque into pinion gear according to 340 N.m.

The structural analysis of the helical gear tooth model is carried out using the finite element analysis in Ansys 14.5. Boundary condition was applied to fix from drive gear and the torque was set into pinion gear according to 340 N.m following the Fig. 2. The mesh was generated with tetrahedral elements. Fig. 3 shows the generated mesh operation models have 41327 total elements and 72325 total nodes were selected for the mesh control. By applying the analysis over the tooth which is facing the load we get the stress distribution in numerical [7].

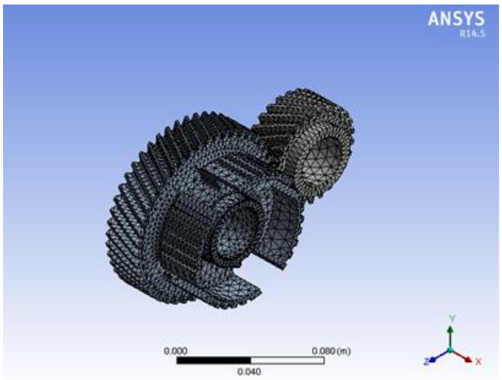

Figure 3. Meshed of helical gear model.

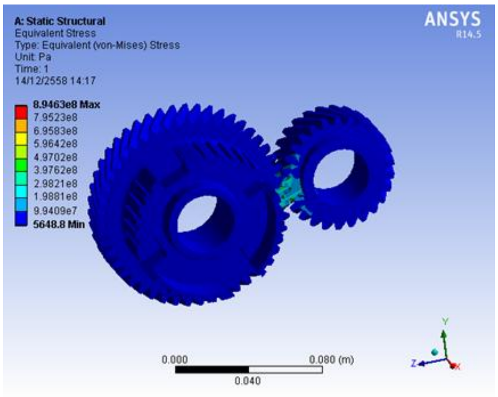

Figure 4. Von Mises stress (bending stresses of W^t 10.60 kN).

TABLE II. COMPARISON OF VALUES OF THE BENDING STRESSES

Torque (N.m)	Tangential transmitted load W^t (kN)	Bending Stress AGMA σ_b (MPa)	Von Mises ANSYS σ_b (MPa)
340	10.60	867.45	894.63
375	11.64	964.32	1048.41
405	12.56	1068.32	1132.38
430	13.38	1128.75	1202.24
455	14.12	1205.44	1272.16
475	14.80	1278.45	1328.62
490	15.41	1338.93	1369.93
515	15.97	1403.72	1439.87
535	16.48	1456.87	1495.75
545	16.96	1516.44	1523.76

Finite element method results for bending, the helical gear assembly was imported in Ansys 14.5 and the boundary conditions were applied to the gear model. In helical gear only Ansys 14.5 analysis was performed because of the helical profile of its teeth. In Fig. 4, it shows the stress distribution plot along the tooth from minimum to maximum of tangential transmitted load is 10.60 kN [8].

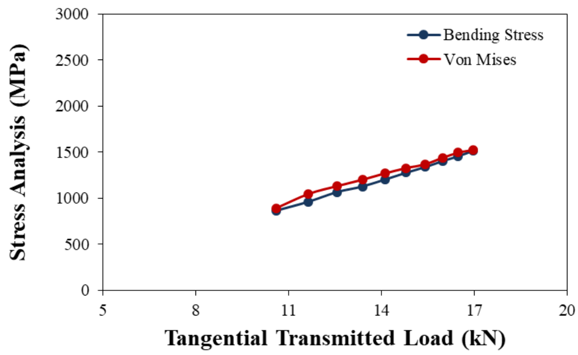

Figure 5. Graph between AGMA bending stress and Ansys results.

Effect of the minimum to maximum bending stresses obtained are show in Table II, there is a variation in the maximum bending stress with the change from tangential transmitted load [9].

The maximum bending stresses obtained are shown in the Fig. 5. It is evident in these that the bending stress given by AGMA and the finite element are very similar. The bending stress value increased with the increase of torque which is in close agreement with values obtained in the AGMA formula. The Chromium steel SCr 420 has the bending strength of 450 MPa [10].

B. *Contact Stress Analysis*

AGMA Contact Stress Equations

The fundamental equation for pitting resistance (contact stress) is

$$\sigma_c = Z_E \sqrt{\frac{W^t K_o K_v K_s K_H Z_R}{d_{wl} b Z_I}} \tag{2}$$

where Z_E is elastic coefficient = 191 N/mm^2

W^t is tangential transmitted load = 10.60 kN

K_o is overload factor = 1.25

K_s is size factor = 1

K_H is the load distribution factor = 1.16

K_v is dynamic factor = 1.62

Z_R is Surface condition factor = 1

d_{wl} is the pitch diameter of the pinion = 67.55 mm

Z_I is geometry factor = 0.19

b is the gear face width = 23 mm.

$$\sigma_c = 191\sqrt{\frac{10.60\times10^3\times1.25\times1.62\times1\times1.16\times1}{67.55\times23\times0.19}}$$

$$= 1753.53 \text{ N/mm}^2$$

Results from the contact stress analysis showed that (σ_c) is 1753.53 MPa. From the stress at base on the results from analysis, it was found to be higher than the contact strength for the gear material is 1550 MPa [10]. The calculated contact stress due to normal operation was 1.13 times higher than the allowable contact strength. The presence of the surface therefore cannot be, on its own, responsible for the fatigue crack.

Analyses contact stress using finite element method.

Contact stress was studied in the same manner as bending stress was calculated. In this research contact stress was obtained at the contact region. The Fig. 6 shows the stress distribution plot along the tooth. The modeled helical gear is analyzed to study the effect of tangential transmitted load (W^t). The minimum to maximum contact stresses obtained are shown in Table III, there is a variation in the maximum contact stresses with the change from tangential transmitted load.

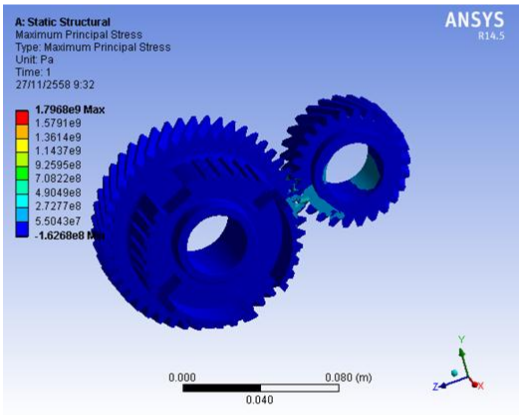

Figure 6. Principal stress (Contact stresses of W_t 10.60 kN).

TABLE III. COMPARISON OF VALUES FOR THE CONTACT STRESSES

Torque (N.m)	Tangential transmitted load W^t (kN)	Contact Stress σ_c AGMA (MPa)	Principal Stress σ_c ANSYS (MPa)
80	2.56	834.73	955.76
150	4.71	1139.66	1198.12
210	6.52	1349.55	1395.85
260	8.07	1511.01	1578.23
305	9.42	1642.82	1718.30
340	10.60	1753.53	1796.80
375	11.64	1848.85	1908.47
405	12.56	1932.20	1977.54
430	13.38	2000.27	2089.65
455	14.12	2067.11	2178.91

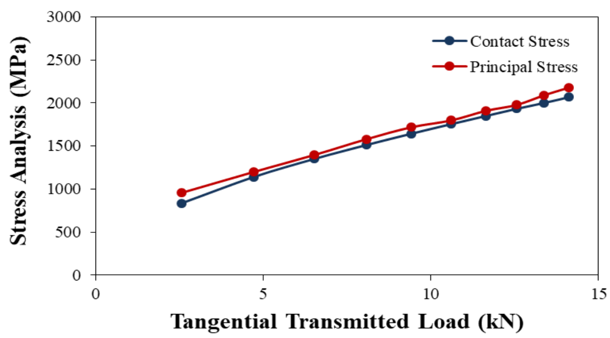

Figure 7. Graph between AGMA contact stress and Ansys results

The maximum contact stress obtained is shown in the Fig. 7. It can be seen in this the contact stress given by AGMA and the finite element are very similar. Stress starts from contact stress on the surface of a gear tooth and bending stress is then extended to the central axis of the gear which is the contact stress as a result of final fracture when torque higher than contact strength of 1550 MPa [10]. The conclusion is that if the torque exceeds contact strength, the gear will fracture.

IV. CONCLUSION

This study was conducted on a failed helical gear used in an automobile. The failed gear material was JIS-SCr 420 steel. The following conclusion can be drawn from the investigation:

1. The results of FEM analysis were compared with AGMA [6] almost similar to theoretical/empirical results.
2. Stress starts from the contact stress on the surface of a gear tooth, and bending stress is then extended to the central axis of the gear, which is the contact stress, as a result of a final fracture when the torque is higher than the contact strength of 1550 MPa. This will benefit the gear design (in such cases, it will allow the car to increase engine power, and the damage can be seen when it occurs as it affects the transmission, propeller shaft, and differential).
3. The failure of the gear was caused by excessive contact stress on the surface of the gear teeth. The calculated contact stress and the finite element are 1.13 times higher than the contact strength of gear material (1550 MPa) [10], if the torque is higher, it would make the contact stress higher as well which is the cause of fracture of the gear. This information is useful for gear design.

V. THER RECOMMENDATION

1. Finite element method of investigation and study can be conducted on the whole gearbox with all elements in the system including gear casing and bearing.
2. Finite element methods of investigation and study can be conducted on the analysis of bending and contact stress for all types of gears such as a spur gear, bevel and worm gears.
3. The failed gear of gearbox under this investigation failed by fatigue fracture.
4. In this case if it is required to increase the torque, the engineer will increase the size and face width of the gear or change the material of the gear to provide more torque.

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