Analysis of exerted stresses of final drive in MF 285 tractor by theoretical and finite element methods

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Abstract: Gears is a rotating cylindrical wheel having tooth cut on it and which meshes with another toothed part to transmit the power or torque.In this investigation, at first, bending stress and contact stress between sun gear and planet gear tooth were determined using of Lewis and Hertzian equations. Afterward, a 3D model of final drives gear was investigated via finite element method (FEM). According to the obtained results, maximum of bending stress and contact stress occurred in Gear 1 and low status of helping gear, according to the results that given in the Table 3, maximum bending stress using of theoretical and FEM simulation methods were obtained 918.62 and 951.82 Mpa, respectively. Also, maximum contact stress in same status using of theoretical and FEM simulation methods were obtained 2952.71 and 2903.20 Mpa, respectively. The percentage difference between the theoretical and FEM bending stress results and contact stress results are of average 3.48% and 11%, respectively, which are still acceptable. As regards, some parameters are ignored in theoretical calculation such as: radial forces in Lewis equation and tangential forces in Hertzian equation, FEM simulation results are acceptable than theoretical results.

Keywords: Final drive, stress analysis, Lewis equation, Hertzian equation, bending stress, contact stress.

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1 Introduction

Gears is a rotating cylindrical wheel having tooth cut on it and which meshes with another toothed part to transmit the power or torque. Spur gear is the simplest type of gear having at tooth cut parallel to the axis of shaft on which the gear is mounted. Spur gears are used to transmit the power between parallel shafts. Spur gear gives 98%-99% operating efficiency (Karaveer et al., 2013). Due to globalization industries are facing competition. It be-comes more and necessary to consider alternative technology of manufacturing materials used for gears (Meenakshi et al., 2012). Gears are one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery (Sabu et al., 2014). Planetary gear train has the advantages of compact structure, large transmission torque, high efficiency and stable transmission. It is one of the basic structures of the gear reducers and is widely used in various mechanical settings (Li et al., 2013). Chen et al., (2007) reported an innovative design of planetary cam trains based on pure-rolling contact intended to overcome the drawbacks of gear trains, such as Coulomb friction and backlash. Sankar et al. (2010) have introduced Corrective measures are taken to avoid tooth damage by introducing profile modification in root fillet. During the recent past, significant progress in the field of contact analysis of gears has also been made, and finite element analysis (FEA) is gradually becoming established

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as an efficient tool in gear box design. Using the finite element analysis, which is a general and systematic computational procedure for approximately solving problems in physics and engineering, many contact problems, ranging from relatively simple ones to quite complicated ones, can be solved with high accuracy (Koisha and Doshi,2012). Vogel et al., (2002), presented a constructive approach for the approximation free tooth contact analysis of hypoid bevel gears. There are several stresses present in the teeth of rotating gears but out of all the stresses, root bending stress and surface contact stress calculation is the basic of stress analysis (Kumar Tiwari and Kumar Joshi, 2012). Theoretically, for the calculation of contact stress at the surface of mating teeth, Hertz equation is used and for determining bending stress at the root of meshing gears, Lewis formula is used. In detail study of the contact stress produced in the mating gears is the most important task in design of gears as it is the deciding parameter in finding the dimensions of gear (Gupta et al., 2012).

The purpose of this study is investigation of Final drive gears of MF 285 tractor for determination of bending and contact stresses of final drive gears using of finite element method (FEM) and theoretical equations and comparison between these methods.

2 Materials and methods

2.1 Theoretical method

In this study, Final drive of MF 285 Tractor has been investigated. The technical characteristics of MF 285 Tractor are given in Table 1.

Table 1 Technical characteristics of MF 285 tractor (Anonymous,2008)

Parameters	Value
Number of cylinders	4
Piston course, mm	127
Cylinder diameter, mm	101
Indicated revolution, r/min	2000
Maximum revolution, r/min	2200
Indicated engine power, Kw	52.94
Maximum torque, N·m	278
Revolution in maximum torque, r/min	1300

The value of input revolution and torque for sun gear of final drive in different status of gear box and helping gear are given in Table2, these values were determined by gears ratio.

Table 2 Revolution and torque to the sun gear in different status of gear box and helping gear

Gear status	Helping gear	Sun revolution, r/min	Sun torque, N.m
Gear 1	High	118.18	3058
Gear 1	Low	37.63	9602.12
Gear 2	High	173.33	2085
Gear 2	Low	55.20	6546.9
Gear 3	High	237.22	1523.44
Gear 3	Low	75.54	4783.6
Gear 4	High	317.84	1137.02
Gear 4	Low	101.22	3570.24

2.2 Gear tooth bending stress using Lewis equation

Bending stress evaluation in modern gear design is generally based on the Lewis equation. This equation, applied with the stress concentration factor Kf, defines the bending stress geometry factor J for traditionally designed standard or close-to standard gears (Kapalevich and Shekhtman,2002). For determination of bending stress at gear root, Equations (1), (2) and (3) were used.In Equation (1) the gear root were investigated as a cantilevered beam, as shown in Figure 1 (Budynas and Nisbett,2008).

$$\sigma = \frac{W^t P}{FY} \quad (1)$$

$$Y = \frac{2XP}{3} \quad (2)$$

$$l = \frac{t^2}{4X} \quad (3)$$

Where: σ is bending stress, Mpa; $W^t$is tangential force, N; P is diametrical pitch, 1/mm; F is face width, mm; Y is Lewis form factor, dimensionless; l is tooth height, mm;X in Figure 2 ,mm; t is thickness of tooth, mm.

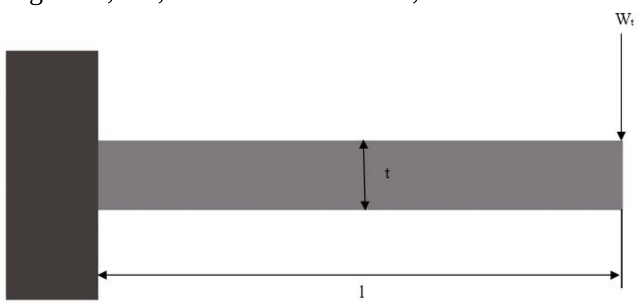


Figure1 Loads and length dimensions used in cantilevered beam by Lewis

Hence, considering worst load condition in this work, the Y Lewis factor for a planet gear with 12 teeth, full depth profile, and 25 degree pressure angle is 0.245. Figure 2 shows the tooth gear with applied load approximately near to the pitch diameter of the tooth surface and their dimensions used in determining bending tooth stress.

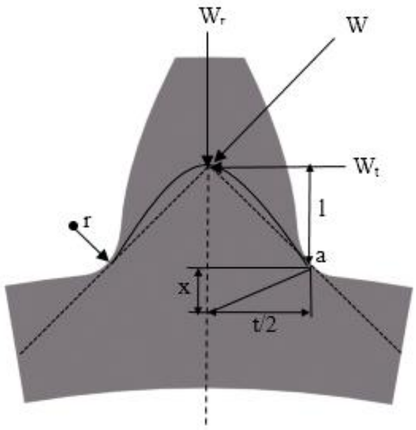


Figure 2 Loads and length dimensions used in determining tooth bending stress

The Lewis equation is based on following assumptions (Ooi et al, 2012):

(1) The effect of radial load is ignored.

(2) The effect of stress concentration at the root fillet is ignored.

(3) It is assumed that at any time only one pair of teeth is in contact.

(4) It considers static loading and does not take the dynamics of meshing teeth into account.

(5) The Lewis form factors with various numbers of teeth only assume a pressure angle of 20 and a full-depth involute.

2.3 Gear tooth contact stress using Hertzian equation

In addition to considering the critical bending stress in gears, analysis of gear tooth contact stress is equally important because excessive contact stress may cause failure such as pitting, scoring, and scuffing of surfaces (Dudley,2002). The Hertzian contact stress of gear teeth is based on the analysis of two cylinders under a radial load. It is assumed in the gear model that the radii of cylinders are the radii of curvature of the involute tooth forms of the mating teeth at the band of contact as shown in Figure 3.

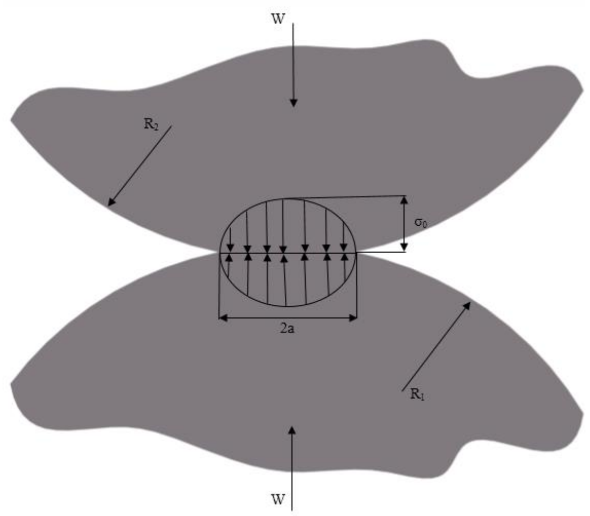


Figure3 Hertzian model of the two cylinders in contact under normal load

The Hertzian theory assumes an elliptic stress distribution, as seen in the Figure 3. The maximum stress is in the middle and determined by Equation (4) (Budynas and Nisbett,2008).

$$\sigma c = \sqrt{\frac{W\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}{F\pi\left[\frac{1-\nu_1{}^2}{E_1}+\frac{1-\nu_2{}^2}{E_2}\right]}} \qquad (4)$$

Where *W* is the normal load, E_1 and E_2 are the modulus of elasticity of the pinion and gear, respectively, υ_1 and υ_2 are Poisson's ratios of the pinion and gear, respectively, and *F* is the face width of pinion, R_1 and R_2 are the respective radii of the involute curve at the contact point, as shown in Figure 4. However, the pitch radius of the pinion and gear denoted as r_{b1} and r_{b2} respectively, can be related to the gear involute radius as $R_1= r_{b1} \sin \varphi$and $R_2= r_{b2} \sin \varphi$

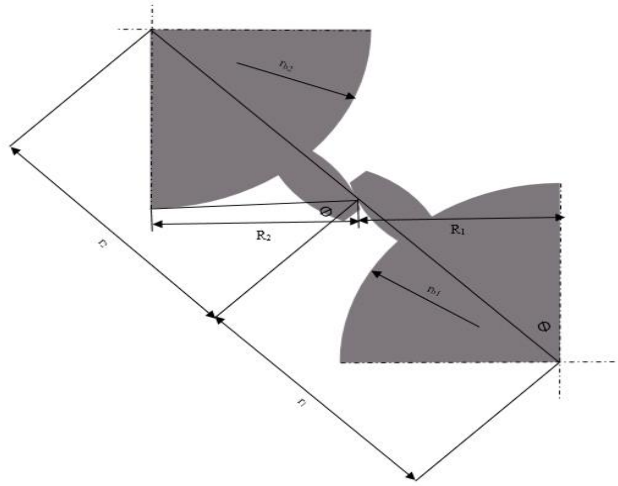


Figure 4 Two involute teeth in contact

In the Hertz contact stress equation, a few assumptions are made, such as pure bending of short beam, elliptic distribution of stresses at tooth contact, and friction between the gear contacting surfaces is not accounted in the stress equation. A question therefore arises concerning their accuracy (Zahavi,1991).

2.4 3D modeling of final drive of MF285 tractor

In this study, first a 3D model of final drive of MF285 Tractor was created using of real model and by Geartrax 2013 and SOLIDWORKS 2013 software, Figure 5. As shown in the Figure 5 other component such as: carrier and input shaft for reducing the analyzing time and errors has been removed. Afterward Final drive was analyzed by ANSYS WORKBENCH 11.

Figure 5 Isometric view of final drive

2.5 Finite element method for determination of bending stress

First of all, static structural analysis has been used and mechanical mesh was used, the mesh size was in "fine" mode. The mesh statistic is included of 89842 nodes and 16558 element as shown in Figure 6.

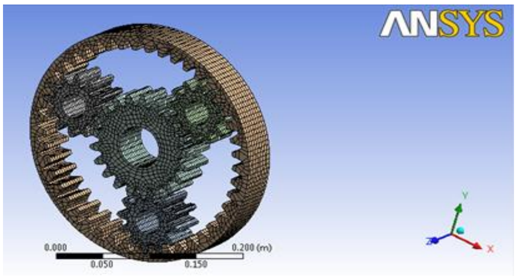


Figure 6 Mesh model of the final drive of MF285 tractor for FE simulation

In this analysis, surrounding gear and planet gears was fixed and cylindrical support was used for sun gear. The contact between the gears surfaces was selected of "No separation" type and input torque is applied on the sun gear. All of steps are shown in Figure 7.

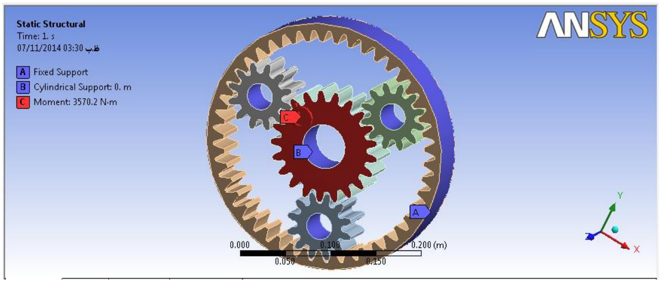
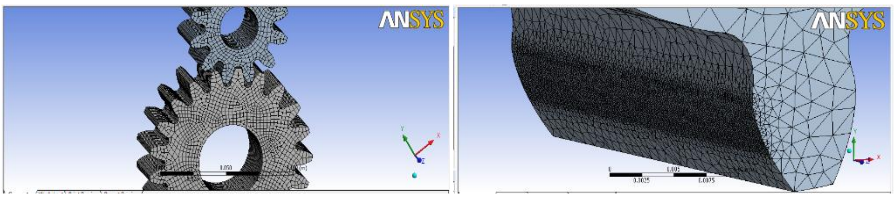


Figure 7 Boundary condition settings of the final drive of MF285 tractor for FE bending Stress analysis

First of all, static structural analysis has been used, as mentioned before, it is assumed that at any time only one pair of teeth is in contact in Lewis and Hertzian equations. In order to, for contact stress determination between the sun gear and planet gear, "Tetrahedrons" element was used as shown in Figure 8-a. For increase the accuracy in the analysis, the contact faces between two teeth has been refinement by "face sizing" command as shown in Figure 8-b. The mesh statistic is included of 136836 nodes and 92572 element.



(a) (b)

(b)

Figure 8 (a) Mesh model of the gear train for FE simulation, (b) Finer mesh elements at the contacting gear tooth surface of the output gear

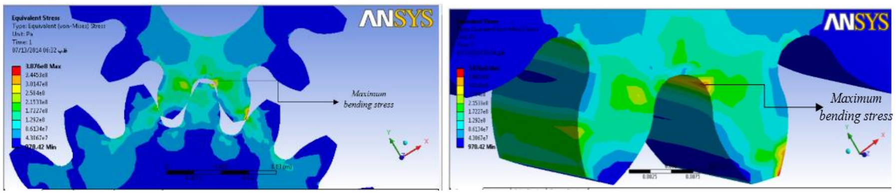
3 Results and discussion

Bending and contact stresses of the final drive in four status of gear box and two status of helping gear were determined using of theoretical and finite element methods calculation and indicated in the Table 3.

Table 3 Comparison of maximum bending and contact stress obtained from Lewis and Hertzian equations and ANSYS WORKBENCH 11

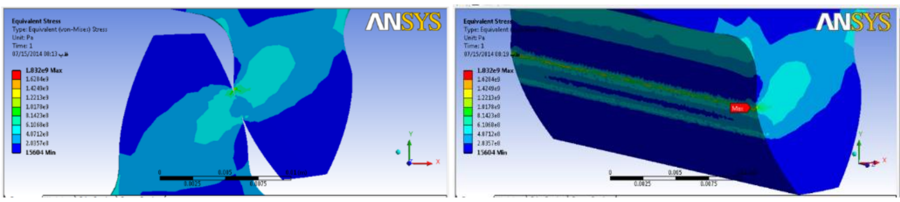
Gear status	Helping gear	σ (Lewis), Mpa	σ(ANSYS), Mpa	σc(Hertzian), Mpa	σc(ANSYS), Mpa
Gear 1	High	303.13	292.55	1570.65	1666.34
Gear 1	Low	951.82	918.62	2903.20	2952.71
Gear 2	High	206.68	199.46	1249.10	1375.95
Gear 2	Low	648.97	626.33	2267.40	2438.13
Gear 3	High	151.01	145.74	912.67	1176.16
Gear 3	Low	474.18	457.64	1976.80	2084.10
Gear 4	High	112.71	108.77	681.17	1016.12
Gear 4	Low	353.9	341.56	1595.50	1800.49

Bending and contact stresses distribution are shown in the Figure 9-a, b and Figure 10-a, b, respectively.



(a) (b)

Figure9 (a) Simulation results shows the maximum bending stress at the root fillet of the input gear;(b) Detailed view of FEM bending stress distribution of the planet gear



(a) (b)

Figure 10 (a) FEM stress distribution of the two contacting gear teeth in side view; (b) Detailed view of FEM contact stress distribution of the planet gear

According to The Lewis equation and Figure 9, the maximum bending stress occurred at the gear root. The gear tooth bending stress results calculated from the 3D FEM model of the planet gear was compared to gear tooth bending stress results calculated using the Lewis equation. The gear tooth bending stress was calculated using both methods with respect to the increased torque load, Figure 11.

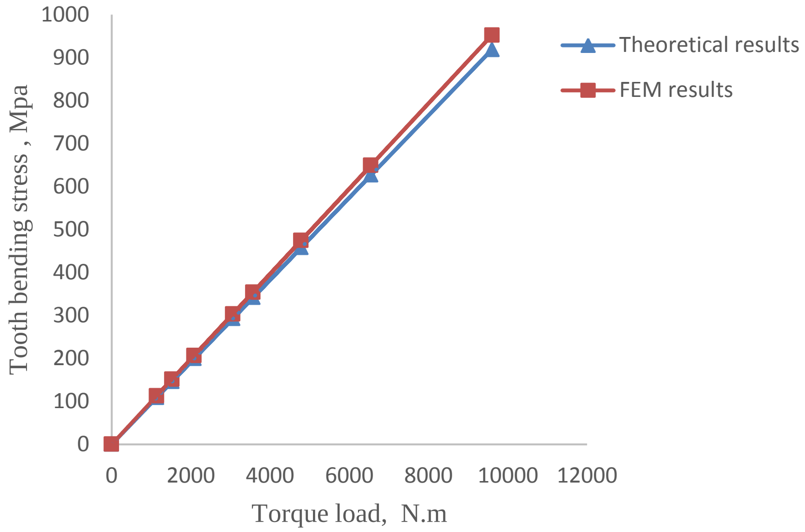


Figure11 Lewis theoretical bending stress results and the FEM simulation bending stress results

Figure11shows the comparison between the theoretical and FEM simulation results for planet gear tooth bending stress. The relationship between the bending stress against increasing torque load using of theoretical and FEM simulation methods are σ= 0.0957 T and σ= 0.0991T+0.0002 with R_2=1, respectively. However, the

FEM stress results are slightly higher than the one calculated from the results calculated using the Lewis formula. This is because FEM takes into account the radial load component of the resultant force exerted from the torque load, which causes higher stress results. The percentage difference between the theoretical and FEM bending stress results is of average 3.48%, which is still acceptable. Therefore, FEM simulation results are near more than theoretical results to the real results.

The gear tooth contact stress was calculated using both methods with respect to the increased torque load, Figure 12.

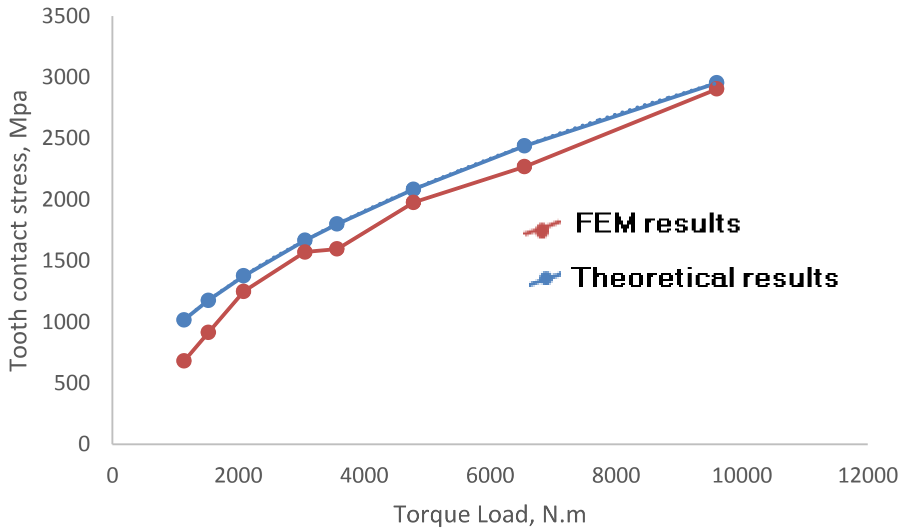


Figure 12 Hertzian theoretical contact stress results and the FEM simulation contact stress results

Figure12 shows the comparison between the theoretical and FEM simulation results for planet gear tooth contact stress. The relationship between the contact stress against increasing torque load using of theoretical and FEM simulation methods are σc= 30.139T0.5 and σc= 7.9072T0.649 with R_2=1 and R_2=0.97, respectively. The percentage difference between the theoretical and FEM contact stress results is of average 11%, which is still acceptable. There is a difference in the results calculated between both methods because the Hertzian equation does not consider the tangential force, which contributes to frictional force on the gear tooth surface. Therefore, FEM simulation results are near more than theoretical results to the real results.

4 Conclusions

According to the obtained results, maximum of bending stress and contact stress occurred in Gear 1 and low status of helping gear which is the worth operating conditions in tractor.Besides, the planet gear has overall higher root bending stress and contact stress compared to the root bending stress at the sun gear.The percentage difference between the theoretical and FEM bending stress results and contact stress results are of average 3.48% and 11%, respectively, which are still acceptable.The relationship between the bending stress against increasing torque load using of theoretical and FEM simulation methods are σ= 0.0957 T and σ= 0.0991T+0.0002 with R^2=1, respectively.The relationship between the contact stress against increasing torque load using of theoretical and FEM simulation methods are σc= 30.139$T^{0.5}$ and σc= 7.9072$T^{0.649}$ with R^2=1 and R^2=0.97, respectively.According to the results, failure points on final drive gear most happen on planet gear. As regards, some parameters are ignored in theoretical calculation such as: radial forces in Lewis equation and tangential forces in Hertzian equation, FEM simulation results are acceptable than theoretical results.

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