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STRESS ANALYSIS FOR SPUR GEARS USING SOLID WORKS SIMULATION

Ahmed kadhim zarzoor, Nabeel Almuramady * and Eman Shaker Hussein
Department Of Mechanical Engineering, University Of Al-Qadisiyah, Al-Qadisiyah, Iraq
*Corresponding author

ABSTRACT

Spur gear is one of the main essential elements used to transferring motiom and energy among the machine parts specially to transfer energy between two parallel axies. Bending, friction, wear and also sliding contact may be cause failure in spur gear. In this paper the bending features of an involute gear in case of the static load have been studied. The model has been made and examination using solidworks software which provides equivalent results to that of the AGMA. The AGMA method used in the theoritical calculation then compared with solidworks analysis. The outcomes received theoretically have been in great contract with these received from software.

Keywords: spur gears, bending stress, contact stress, AGMA method.

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1. INTRODUCTION

Meshing gears system is one of the most crucial mechanism in a rotary machines [19], and it considers as the best method for transmitting power in these type of machines because of their high level of consistency and conservativeness. There are a different types and sizes of starting from Nano/Micro-scale as these have been used in the Micro-Electro-Mechanical System (MEMS) [1,2,4] and reached to macroscale, which is the target of this study. Gears tooth have many effective properties, the important one is that the contact point is constantly moving along the face of the tooth; therefore, before the geometry can be determined, it should be understood how the gears be in contact for more information about contact geometry see [25,23,13,3].

The fast development in the industrial field, especially in the vessel and vehicle industry, have been pushed the designers to increse the using of gears in their designs more and more. It also been necessary to fill the gap in the information about how the stress distributed along the gear profile during the mesh operating [8,26]. The needs to a good transmission mechanism in machines are rapidly increase in the last decades. The features of gears design in the cars, motors and generators are usually required extra precise checking in to ensure the maximum safty [10]. one of the most impotant test after the manufacturing of gears is the distribution of stress, especially bending stresses, on the face of the teeth [22,16].

Bending stresses is a common phenomenon has been observed in the meshing gear teeth. Even with a small reduction in the stresses at tooth results in great increase in the life of a spur gear. For decades, improvement of gear design has been done by enhance their material, or sometimes by hardening the surfaces using heat treatment or improve the surface finishing. [9].

This study has been done on the meshing spur gear teeth with involute profile see Figure 1 by simulating the bending stress features of various types and the best one among them is proposed. A finite element model of spur gear is taken into consideration for evaluation and compared with the analytical method and stress relieving functions of diverse sizes are inserted on gear teeth at root place. the most effective size and area of the stress remedy capabilities for Spur gear are proposed, which assist in decreasing the fatigue failure in gears[5,26,17].

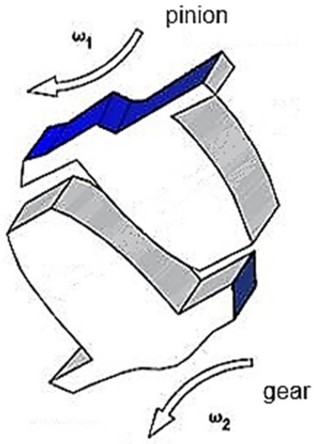


Figure 1 Meshing of spur gear teeth

The stress analyses in the spur gears in the earlier duration had been achieved using analytical techniques [14,12,11], which requisite several assumptions and generalizations. Usually, the stress analysis in gears have estimates associated to the tooth stress and to tribological phenomena such as friction, pitting and wear.

In this study, the bending stress analyses are accomplished, whereas trying to design spur gears to resist the failure due to bending of the teeth. with the growth of softwares to be greater and more robust, researchers have been learned to apply the arithmetic techniques and create a good theoretical models to calculate the impact of anything can be studied in detail [18,15]. A number of specific solutions could be provide by the arithmetical techniques because they are commonly need to a less assumptions. The strategies and the particular form of solution, despite of the fact that, have to be decided on carefully to ensure that the results are specific and that the computational time is realistic.

There are many types of failure occur in the meshing spur gears, but the two major kinds are the failure through bending and failure due to contact stresses at the gear tooth surface. The contact stress or pitting stress between the two contacting gears teeth can be calculated by using the Hertzian contact equation see Figure 2, and is relative to the square root of the applied tooth load (AGMA 2001-D04). The bending stress is predicted with the aid of assuming the gear tooth as a cantilevered beam, by means of a pass section of face width by tooth thickness. The spur gear tooth load is immediately proportional to bending stress. typically, bending failure will occur when the stress on the tooth is superior than or equal to the yield strength of the gear tooth material.

2. THEORETICAL CALCULATION OF GEARS CONTACT

Hertz line contact theory has been used to calculate the pressure distribution and it is represented in Figure 2, where $p(x)$ is the pressure. It has been determined at each single point along the contact zone between the meshing gear teeth surfaces. Moreover, p_o has been referred to the maximum pressure in the contact; L, in this specific case, is the thickness of the gear tooth; and, R_1 and R_2 are the radii of the curvature of the pinion and gear teeth, respectively. However, R_1 and R_2 depend on the time moment of their contact process, while $R(t)$ is the radius of relative curvature and it is time dependent [6].

$$\frac{1}{R} = (\frac{1}{R_1} + \frac{1}{R_2}) \tag{1}$$

where R is the radius of relative curvature.

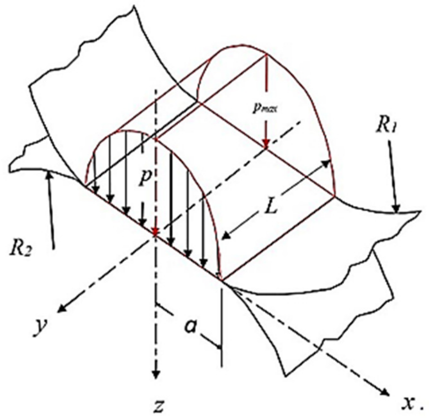


Figure 2 the pressure distribution according to Hertiz line contact

$$p(x) = p_o \sqrt{1 - \frac{x^2}{a^2}} \tag{2}$$

The contact area at each position along the path of contact can be found by determine the half-length of the contact (a) and multiplying it by the face width of the tooth by (L), as shown in Figure 2:

$$A = 2aL \tag{3}$$

$$a = \sqrt{\frac{4PR}{\pi E^*}} \tag{4}$$

Where p is the load has been applied, R is the radius of curvature and E^* is the modulus of contact and it can be found by

$$\frac{1}{E^*} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \tag{5}$$

The maximum pressure calculated at each point of contact according to the equation of p_o:

$$p_o = \frac{2p}{\pi a} \tag{6}$$

The dimensions of the meshing two gears in this study have been assumed to be as following the pitch diameter for both gear is (100mm), pressure angle α is (20°), the module is equal to (5) and the addendum also equal to (5mm) and the dedendum is (6.25mm), so the number of teeth N will be equal to (20). The shaft diameter is (30mm) and the face width of the tooth is (30mm). The power transmitted through the contact will be (6 kw) and the driving speed is equal to (1200 r.p.m.) and finally the Young's modulus is (200GPa) and Poison ratio is (0.3). Therefore, by applying Equation (2) it can found the pressure distribution along the tooth face as shown in Figure 3.

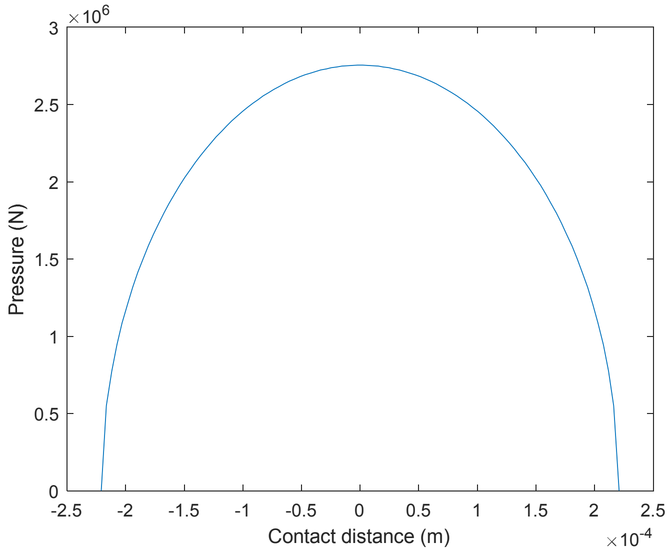


Figure 3 The pressure distribution along the tooth face in the contact zone

Where a is the length of the half contact area and it is equal to $(2.2076 \times 10^{-4} m)$; therefore, the total contact length is $(4.4152 \times 10^{-4} m)$. In the contact region, p_o has been determined to be $(2.755 \times 10^{+6} N)$ according to Equation (6). Also, during the simulation, the sliding and traction motion effects have then been added to the calculation of the meshing between gear teeth. These calculations have been taken in the single tooth contact (STC).

Then, the next step is to find the FE version and solution techniques required for the precise computation of 2nd spur gear bending stresses are determined. Afterwards, the bending stresses calculated using FEA turned into evaluated to the consequences obtained from present techniques of AGMA (American gear production association) [20,21].

Now let's take AGMA bending stress modified formula

$$\sigma_b = f \times k_v \times k_o k_m / (L \times m \times j)$$

Module$m = \frac{pitch\ diameter}{number\ of\ teeth} = \frac{100}{20} = 5$

TorqueT $= \frac{power}{velocity} = \frac{60 \times p}{2 \times \pi \times N} = 47.75$ Nm

Tangential force$f = \frac{2 \times torque}{pitch} = \frac{2 \times 47.75}{0.1} = 955$N

Overload factor$K_o = 1$ (for uniform)

Load distribution factor$K_m = 1$

Dynamic factor$K_v = 1.25$

Geometry factor$j = 0.335$ for $\emptyset = 20°$

$$\sigma_b = \frac{955 \times 1.25 \times 1 \times 1}{30 \times 5 \times 0.335} = 23.76 N/mm^2$$

Then, spur gears will be analysed to see the various results from the static analyses by using solidworks software.

3. MATERIALS AND BOUNDARY CONDITIONS

The mechanical properties of the gear material can be explained as following: the Young's Modulus (E) is equal to (200GPa) and the Poisson's Ratio is (0.3) also the ultimate tensile strength is equal to (1962 MPa) and finally, the yield strength is (1500 MPa). Finite element technique is the smooth approach compared to the theoretical techniques to calculate the stress developed in tooth of gears. consequently FEM is extensively used for the stress analysis of mating gears. FE analysis is completed in solidworks software program to determine the maximum touch stresses for forjed steel. also the deformation is found out for each the gears.
In finite element, the boundary conditions play the vital role in the calculation. Finite elements has been used here to calculate and check the stresses that have been generated inside the spur gear with keeping in the mind the assumptions that have been made. For simplicity purpose, the boundary condition of the spur gear is fixed at the shaft bore and a tangential force of 955N has implemented on the gear tooth. This force has been taken at the tip of the tooth in the double tooth contact (DTC) zone.The boundary and loading condition is shown in Figure 4 and 5**.**
The mesh in the gear model has been achieved by using of solid mesh with the variety of elements of 249122 and number of nodes of 361809 with high mesh quality

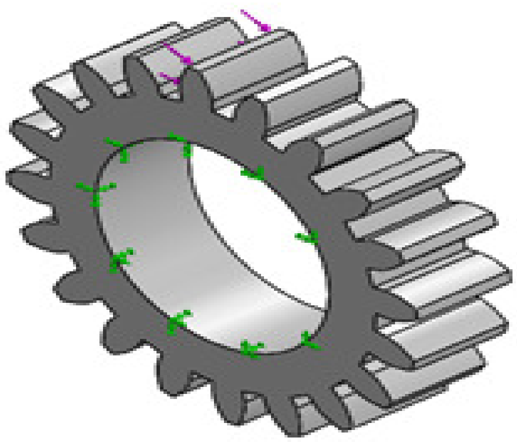


Figure 4 Force and boundary condition

3.1. MESHING

The element size that has used is 4.82996mm.

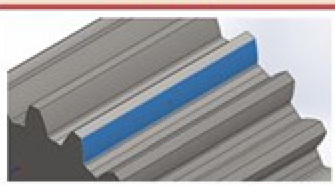
Load name	Load Image	Load Details
Force-1		Entities: 1 face(s) Type: Apply normal force Value: 955 N

Figure 5 Loads and fixtures

This mesh has been used as it is high-quality and gives least quantity of elements with appropriate outcomes, so the calculation time is reduced. The meshing model has been illustrated as shown in Figure 6.

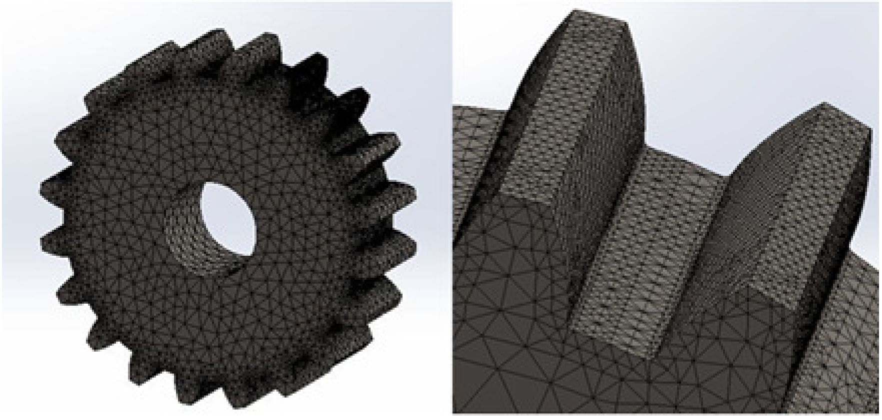


Figure 6 The meshing model

4. RESULT AND DISCUSSION

Five spur gears have been modeled in the solid work with the same parameters, but in different face width. The value of the face width has been ranged from 30mm to 70mm with 10mm increments. The analysis of the bending stresses of the model has been done statically with all the preprocessing steps, in which the initial effects of the static analysis are shown in Figures 7 and 8.

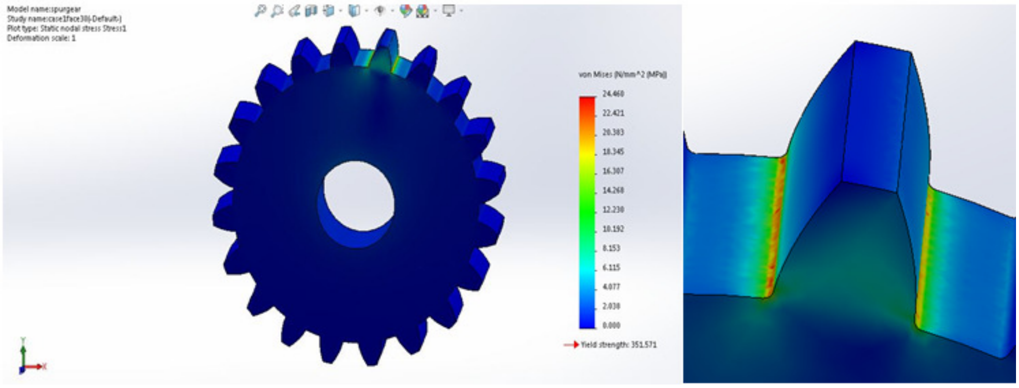


Figure 7 bending stree with 30 mm face width

In Figure 7, the results of the static analysis for the spur gear has taken for the face width of 30mm and the bending stress that has been obtained is 23.76 MPa. While, in Figure 8, The static analysis of the gear has been taken for the face width of 40mm and the bending stress that has been obtained is 17.589 MPa.

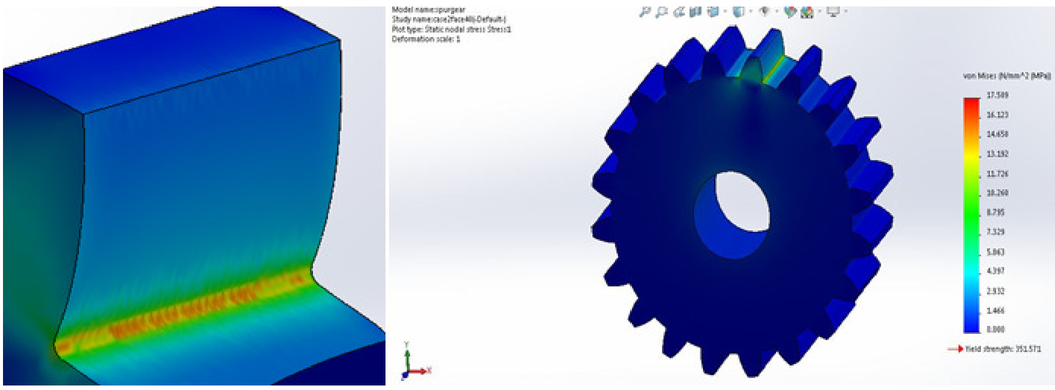


Figure 8 bending stress with 40 mm face width

Five models in the solid work have been done with the same conditions, but with the variety of the face width. Table 1 shows the bending stress for different face width. The value of the face width has been shown for the 30mm to 70mm with 10mm increments. The theoretical results have been taken similar as to the ref [7], which have been compared with ANSYS software. While in this study this work has been done in SolidWork software and we have a better match with the theoretical results

Table 1 Bending stress for different face width

case	Face width L(mm)	Bending stress (MPa)		Percentage different (%)
		Theoretical	simulation	
1	30	23.76	24.466	2.97
2	40	17.82	17.589	1.29
3	50	14.25	14.346	0.673
4	60	11.88	12.559	5.715
5	70	10.18	10.204	0.235

While, Figure 9 shows the bending stress versus face width and the value of the face width has been taken as the 30mm to 70mm with 10mm increments. The results obtained have been showed incredible match between the theoretical and the simulation results.

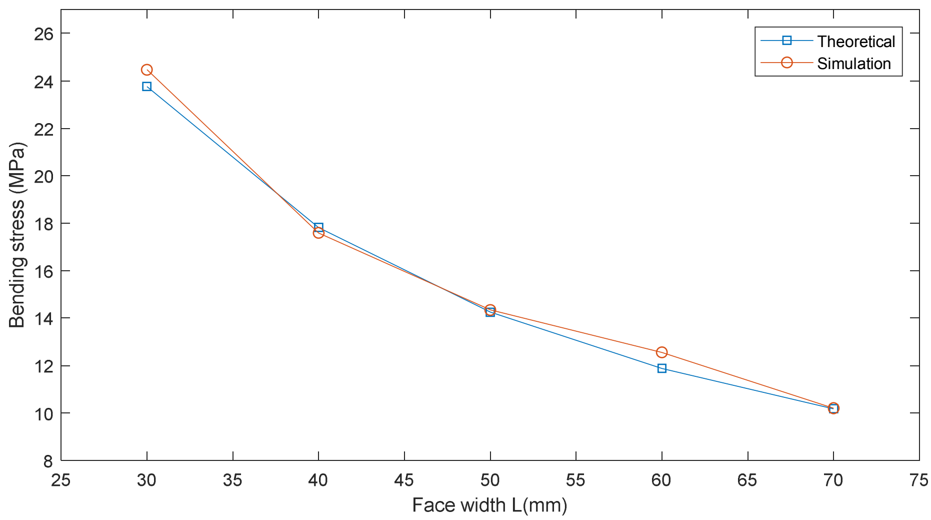


Figure 9 bending stress versus face width

Now we are going to check the value of bending stress by changing the values of module and calculate the stress using AGMA analytical apprach also compare these values with solidwork simulation.

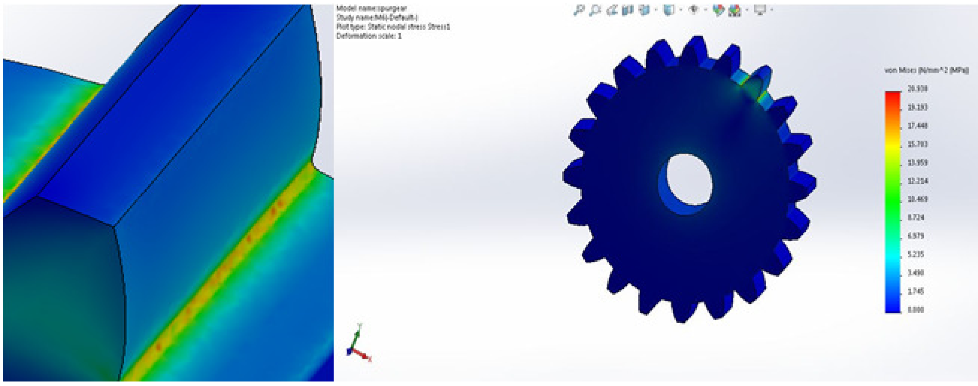


Figure 9 bending stress with module m=6

Then, four models of the solid work have been done with the same conditions, but with the variety of the module. Table 2 shows the bending stress for different modules. The value of the modules has been shown for the 5, 6, 7 and 8 respectively.

Table 2 Bending stress for different modules

case	Module	Bending stress (MPa)		Percentage different (%)
		Theoretical	simulation	
1	5	23.76	24.466	2.97
2	6	19.796	20.938	5.7
3	7	16.968	17.779	4.77
4	8	14.847	15.431	3.9

While, Figure 11 has shown the bending stress versus modules and the value of the modules has been taken as 5, 6, 7 and 8 respectively. Also, the results that have been obtained showed a good match between the theoretical and the simulation results.

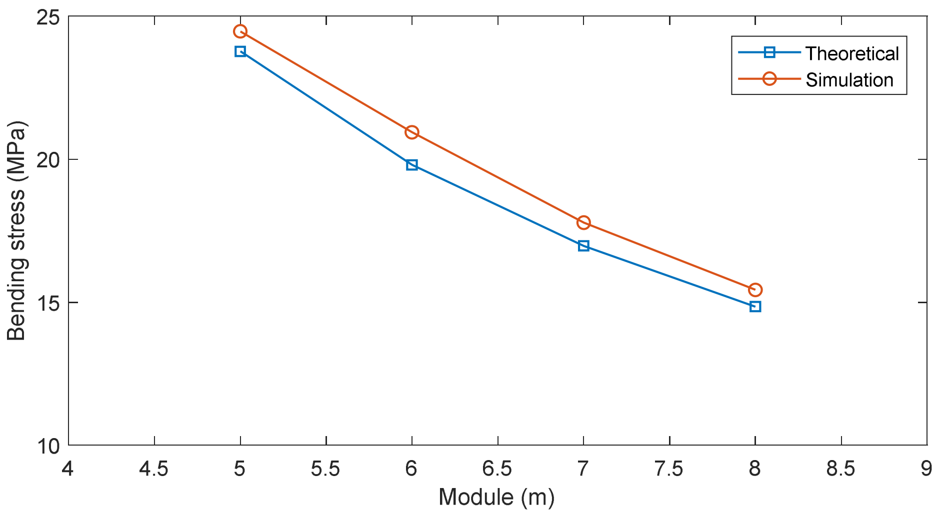


Figure 10 bending stress versus module

5. CONCLUSION

Bending, friction, wear and sliding contact is the main problems that are causing failure in gears. Finite element method has been used to find a real model of the geared set and also, to show the stress analysis inside the pair of gears. The development off finite element analysis model of the spur gear assembly has been done to simulate the contact stresses and calculate the bending stresses with taken into account the extra considerable function in the design of gears. Hertz line contact theory has been used to calculate the pressure distribution and it shows that the Hertz theory is a perfect method to calculate the stresses in the contact zone. Also, that AGMA method is used for calculating the bending stress in the meshing pair of gears. Theoretically, the final results have been acquired by using AGMA method similar with finite element analysis of spur gear. As a result, primarily based in this finding if the contact stress minimization in the primary situation and if the big energy is to be transmitted then spur gears with higher model is desired. hence, we conclude that analysis software can be used for different analyzing purposes.

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