Comparison of Contact Stresses in Involute and Cycloidal Profile Spur Gear Tooth

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Abstract— This study investigates the comparison of contact stresses in involute and cycloidal profile spur gear tooth for different modules by means of Finite Element Analysis. The mathematical models of the gear pair has been generated based on the theory of gearing and the generation mechanism. Models are made in Creo parametric software tool and the analysis has been done in ANSYS Workbench. The contact stresses between the mating teeth corresponding to different modules for both involute and cycloidal profile spur gear teeth obtained from FEA are compared with that obtained by the basic Hertz theory for contact stress and it is observed that the FEA results are validating with the Hertz equation. Also, it is found that the contact stresses for involute pinion as well as involute gear are much lower than cycloidal pinion and gear. So, it is concluded that as the module increases, the contact stress for both the profile decreases and for the same module, involute profile spur gear is stronger than cycloidal profile spur gear in consideration of contact fatigue failure.

Key words: Finite Element Analysis, Hertz Theory for Contact Stress, Contact Fatigue Failure

I. Introduction

In gears, pitting is a surface fatigue failure that results from repetitions of high contact stress. Stress analysis for gear teeth is regarded as a limiting factor for the designers. Stress analysis focuses on the determination of the region of stress concentration where failure or fracture may be initiated. Contact or Hertzian fatigue failure mode is one of the common modes of gear failure. When loads are applied to the bodies, their surfaces deform elastically near the point of contact. The failure of the gear tooth attributable to corroding happens when the contact stress between two meshing teeth exceeds the surface endurance strength of the materials.

Contact stress in mating gears depends upon the geometry of pairing teeth, that isn't constant throughout the contact. The teeth geometry additionally depends on gear module and pressure angle. In this investigation contact stress calculations are analyzed with different module.

II. Literature review

Chauhan [1] reviewed the effect of important parameters on bending and surface durability of gears. It was found that the results obtained by such experiments are not parallel to estimations based on the non-uniform load distribution on gear teeth. In order to create coherence among the results obtained by AGMA and ISO, a number of modifications factors have been applied in deriving mathematical relations for gears. Latest methods such as minimum elastic potential energy method have been introduced to overcome drawbacks of these mathematical relations.

Marimuthu and Muthuveerappan [2] calculated gear stress values in terms of non-dimensional stress. The influence of gear drive parameters such as drive and coast side pressure angles, top-land thickness coefficients, contact ratio, coefficient of asymmetry, gear ratio and teeth number on load carrying capacity has been studied extensively on non-dimensional fillet stress and maximum contact stress and compared with that of the conventionally designed gears.

Patil et al [3] evaluated the contact stresses among the helical gear pairs, under static conditions, by using a 3D finite element method. The helical gear pairs on which the analysis was carried are 0, 5, 15, 25-degree helical gear sets. The FE results have been further compared with the analytical calculations. The analytical calculations are based upon Hertz and AGMA equations, which are modified to include helix angle. The contact stress results have shown a decreasing trend, with increase in helix angle.

Qin and Guan [4] investigated the surface and sub-surface stresses of gear teeth using hertz theory and the finite element method. The number of loading cycles required for fatigue crack initiation was predicted using the Smith Watson topper method based on the multi axial fatigue mechanism. It was concluded that the life of the teeth is introduced to decrease the stresses on these points and improve their initiation fatigue life.

Roy et al [5] calculated the contact stresses developed in a mating spur gear which has involute teeth. A pair of spur gears were taken from a lathe gear box and proceeded forward to calculate contact stresses on their teeth. Contact failure in gears was predicted by comparing the calculated Hertz contact stress to experimentally determined allowable values for the given material. Analytical contact stresses were calculated for different module and these results were compared with the results obtained in modeling analysis in ANSYS.

Hwang et al [6] calculated the stress generated in the meshing gear teeth in finite element analysis. Variation of contact stress at different contact position was investigated. Comparison was made between variation of contact stress during rotation and contact stress at the lowest point of single tooth contact (LPSTC) and the AGMA equation for contact stress. The conclusion of this analysis was that the design that considers contact stress is more severe than the AGMA standard. The values obtained by finite element method were lower than contact fatigue strength of material. Thus, they ensured appropriate safety and strength.

Tiwari and Joshi [7] evaluated the contact stress and bending stress of mating involute spur gear teeth. FEM software has been used to perform meshing simulation. It was observed that the theoretical results obtained by Lewis formula and Hertz equation and results found by AGMA/ ANSI equations are comparable with Finite Element Analysis of spur gear.

Ali Raad Hassan [8] calculated the contact stress between two spur gear teeth in different contact positions. A platform has been established to plot a pair of teeth in contact. Each case was represented a sequence position of contact between these two teeth. The platform gives graphic results for the profiles of these teeth in each position and location of contact during rotation. Finite element models were made for these cases and stress analysis was done. The results were presented and finite element analysis results were compared with theoretical calculations.

III. GEAR SPECIFICATIONS

Modeling of involute and cycloid gear arrangement with different module has been done as per literature data [7]. The main dimensions and parameters for standard 20° full depth involute spur gear are,

- No. of tooth on Gear = 50
- No. of tooth on Pinion = 18
- Module of the gears = 2.5 mm
- Face width = 30 mm
- Pinion speed = 1425 rpm
- Young's modulus = 2.1×10^5 (MPa)
- Poisson's ratio = 0.3

IV. MATERIAL SELECTION

In this investigation, the grade 1 steel has been considered as material for both gear and pinion due to its non-shrinking characteristic, general purpose tool steel with good abrasion resistance, toughness, and machinability. It is extremely stable with minimal deformation after hardening and tempering. Maximum attainable Rockwell hardness is C57-C62. Melting point is 2800° F.

V. RESULTS AND DISCUSSION

A finite element analysis has been carried out to investigate the contact stresses induced in both involute and cycloidal spur gear tooth. The models have been generated in Creo parametric software and analyzed in ANSYS software tool. The contact stress results for module of 2 mm for both pinion and gear of involute and cycloidal profile has been shown below from figure A to D which was performed in ANSYS structural analysis with same boundary domain conditions.

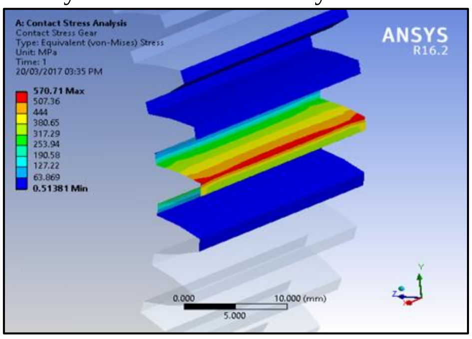

Fig. 1: Contact stress of involute gear

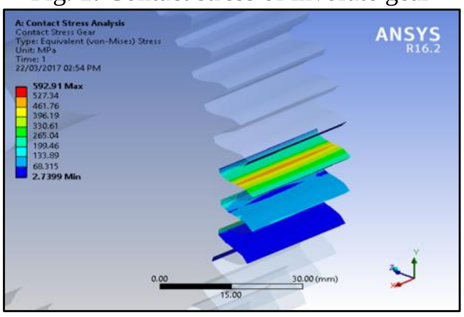

Fig. 2: Contact stress of cycloidal gear

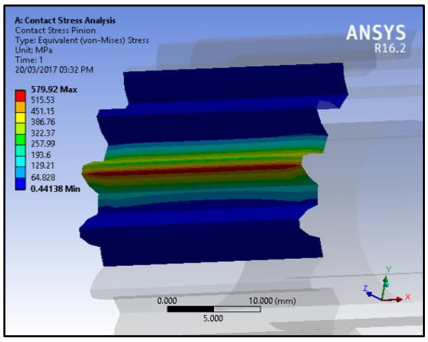

Fig. 3: Contact stress of involute pinion

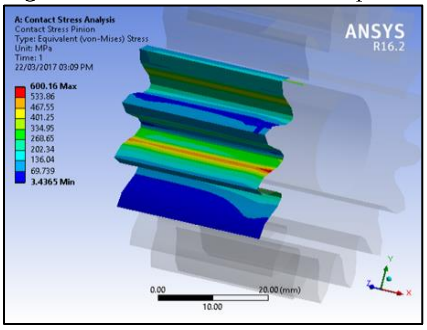

Fig. 4: Contact stress of cycloid pinion

The above results have been tabulated and a comparison is made between involute and cycloidal profile spur gear and pinion teeth which are shown below

Module, mm	Contact Stress, MPa			% Error
	Hertz Equation	Literature	FEA	
2.5 Gear	562.27	567.75	562.12	1%
2.5 Pinion	567.11	571.62	570.33	0.3%

Table 1: Contact stress results for involute gear from literature [7] and FEA

Table 1 indicates that the percentage difference between literature [7] and FEA is 1% for gear and 0.3% for pinion which is under consideration hence the results are validated.

Module, mm	Contact Stress, MPa		
	Involute Pinion	Cycloid Pinion	% Difference
2	579.92	600.16	3.49
2.5	570.33	595.57	4.43
3	564.06	582.56	3.28
4	557.33	575.69	3.29
5	547.67	565.11	3.18

Table 2: Comparison of contact stress in involute and cycloid pinion teeth

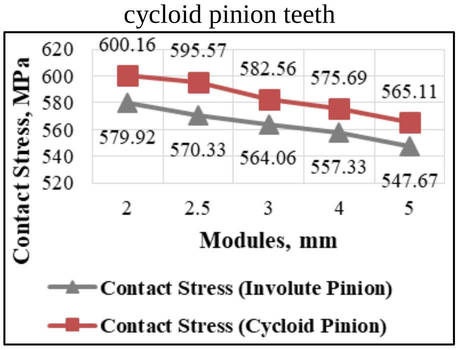

Fig. 5: Variation of contact stress for different modules for involute and cycloid pinion teeth

The variation of contact stress induced in the pinion with different modules is shown in figure 5. It is observed that the contact stress in involute pinion teeth is lower than the cycloidal pinion teeth for the same module. With the increase in module from 2mm to 5mm, the contact stress decreases for both involute and cycloidal profile spur gear.

Module, mm	Contact Stress, MPa		
	Involute Gear	Cycloid Gear	% Difference
2	570.71	592.91	3.89
2.5	562.12	584.95	4.06
3	556.04	576.98	3.77
4	549.22	569.01	3.60
5	538.13	551.03	2.40

Table 6: Comparison of Contact stress in involute and cycloid gear teeth

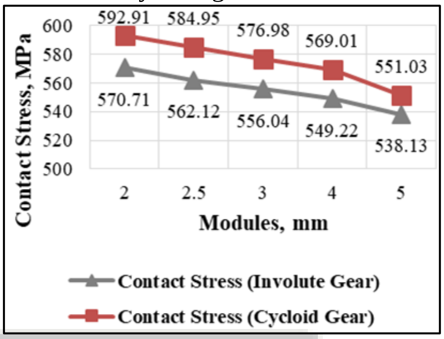

Fig. 7: Variation of contact stress for different modules in involute and cycloid gear teeth

The variation of contact stress induced in the pinion with different modules is shown in figure 7. It is observed that the contact stress in involute pinion teeth is lower than the cycloidal pinion teeth for the same module. With the increase in module from 2mm to 5mm, the contact stress decreases for both involute and cycloidal profile spur gear.

VI. Conclusion

Results presented in this paper for contact stress from finite element analysis are compared from the literature review and Hertz equation and it is found that a slight variation is obtained which is under consideration. Further comparison is made between involute and cycloidal pinion and gear teeth and it is concluded that for any value of module, the contact stress for involute spur gear teeth is less than cycloidal spur gear teeth, also the contact stress values for any profile spur gear decreases with the increasing module.

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