Analysis and Optimization of Contact Stresses in Spur Gear using Finite Element Analysis

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ABSTRACT

The gears are the most common means to transmit the power. Spur Gears are used in many machine tools to transmit the power like Lathe machines, Drilling Machines etc. Since the Spur gears are subjected to high loads in such machine tools, results in high contact stresses between mating gear tooth. As a result of that, gear tooth failures occur, soit is necessary to reduce it. The purpose of this dissertation work is to identify the magnitude of the stresses for a Specified design and to optimize the weight of spur gear. Analysis and Optimization of spur gear is done by using Finite Element tool. Results are compared with Hertz contact stress. By changing the current conventional design parameters we can achieve better performance of the gear under contact pressure as well as in equivalent stress while reducing the weight of the gear. It can be clearly observed that 30.01 % of Equivalent stress, 10.21% of contact pressure is reduced& 12.51% of total mass is reduced when compared to baseline gear assembly with modified design.

KEYWORDS: *Spur Gear, Contact Stress and Fatigue Life & Finite Element Analysis*

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1. INTRODUCTION

It is assumed that, as this small area of contact forms, points that come into contact are points on the two surfaces that originally were equal distances from the tangent plane. There are two types of pitting-Initial and destructive Pitting. The initial pitting is a localized phenomenon, characterized by small pits at high spots. Initial pitting is caused by the errorsin the tooth profile, surface irregularities and misalignment. The Destructive pitting is a surface failure, which occurs when the load on the gear tooth exceeds the surface endurance strength of the material. Destructive pitting depends upon the magnitude of the Hertz's contact stress and number of stress cycles. This type of failure is characterized by pits, which continue to grow resulting in complete destruction of the tooth surface. It is common practice to design the gear teeth on the basis of calculations and excessive tests which are used to validate calculations. With advancement of technology now we can go for third method of verification of the design by the means of FEA analysis. Many techniques are available to simulate the contact stresses but correct one should be selected and methodology should be finalized to reduce the contact stresses in the conventionally designed gear pairs and advancements in the design on the basis of the same is needed.

1.1 LITERATURE REVIEW

Dr. Ali Raad Hassan," Contact Stress Analysis of Spur Gear Teeth Pair", World Academy of Science, Engineering and Technology 58, 2009, The maximum stress result obtained from AGMA stress calculation method was 587MPa while the maximum contact stress obtained from the finite element contact analysis was 595MPa.

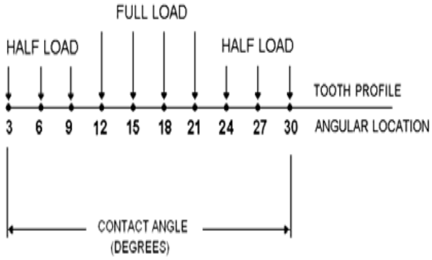

Fig-1: Contact Stress Analysis of Spur gear Teeth Pair

Dhavale A.S. Abhay Utpat ,"Study of Stress Relief Features at Root of Teeth of Spur Gear ", International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622, Vol. 3, Issue 3, May-Jun 2013, pp.895-899 2013

Table-1 Equivalent Stress Vs Total Deformation

Name	Cx (mm)	Cy (mm)	Dia 1 (mm)	Equivalent Stress Maximum (N/mm2)	Total Deformation Maximum (mm)
1	15.39	14.65	8	8.8249693	0.005628490
2	9.39	13.86	8	6.9156129	0.005132248
3	11.26	13.11	8	7.9884064	0.005274934
4	11.46	13.16	8	7.3715043	0.005272500
5	11.62	13.22	8	7.5745565	0.005272489
6	11.83	13.32	8	7.4616923	0.005281491

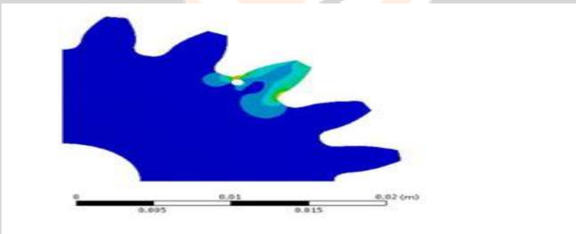

Fig-2: Equivalent Stress Vs Total Deformation

Mr.Bharat Gupta, Mr.Abhishek choubey ," CONTACT STRESS ANALYSIS OF SPUR GEAR " , International Journal of Engineering Research & Technology (IJERT), ISSN: 2278-0181, Vol. 1 Issue 4, June -2012.

Table-2: Stress Vs No of Modules

Sr. No.	Module (mm)	$P_{p(ANSYS)}$ (MPa)	$P_{p(Hertzian\ stress)}$ (MPa)	Differences [%]
1	2	1733.7	1724.13	0.5
2	3	800.6	791.02	1.19
3	4	468.64	465.56	0.65
4	5	255.43	257.34	0.74
5	7	129.83	129.61	0.16
6	8	102.41	102.85	0.43
7	9	53.457	52.39	1.97

Nidal H. Abu-Hamdeh, Mohammad A. Alharty " A Study on the Influence of using Stress Relieving Feature on Reducing the Root Fillet Stress in Spur Gear ", Proceedings of the 2014 International Conference on Mathematical Methods, Mathematical Models and Simulation in Science and Engineering, ISBN: 978-1-61804-219-4

2. METHODODLOGY

In this, grey cast iron is used as the spur gear materials. The material properties of grey cast iron are given in Table 3.

Table-3: Material Property of Grey Cast Iron

Material Property	Symbol	Value	Unit
Density	P	7100	Kg/m^3
Poisson Ratio	Θ	0.26	-
Young's Modulus	E	1.65E+05	MPa
Tensile Yield	S_{yt}	250	MPa
Tensile Ultimate	S_{ut}	350	MPa

Table 4: Dimensions of Spur Gear

Dimension	Symbol	Gear	Pinion	Unit
No. of Teeth	Z	100	56	-
Pitch Circle Dia	D	160	87.8	mm
Pressure Angle	Φ	14.5	14.5	Degree(^{0})
Addendum Radius	R_A	80.95	46.2	mm
Dedendum Radius	R_D	77.5	42.5	mm
Face Width	B	15.3	16.2	mm
Shaft Radius	R_S	22.2	22.2	mm
Module	M	1.6	1.57	mm

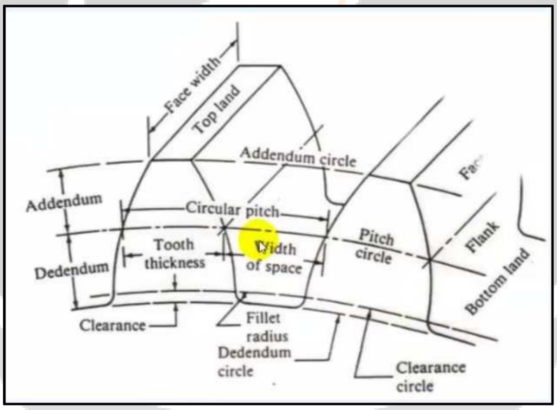

Fig-3: Nomenclature of the Gear Tooth Profile

2.1 Design Procedure

Pitch circle radius = (Module × Number of teeth's)/2

Addendum radius = Radius of pitch circle + Module

Deddendum radius = Radius of pitch circle – 1.25× Module

Clearance radius = Deddendum radius + 1.25× Module

Table-5: Pinion and Gear Specifications

	Pinion	Gear
Module (m)	1.6mm	1.6mm
Number of Teeth's	56	100
Addendum Circle Radius (Ra)	46.4mm	81.6mm
Dedendum Circle Radius (Rd)	42.8mm	78mm
Pitch Circle Radius (Rp)	44.8mm	80mm
Clearance Circle Radius (Rb)	43.2mm	78.4mm
Involute Angle	1.6 Degree	0.9 Degree
Face Width	20mm	20mm

Table-6: Baseline Design Parameters

Gear & PinionThickness (mm)	**Gear & PinionFillet Radius (mm)**
4	**0.6**

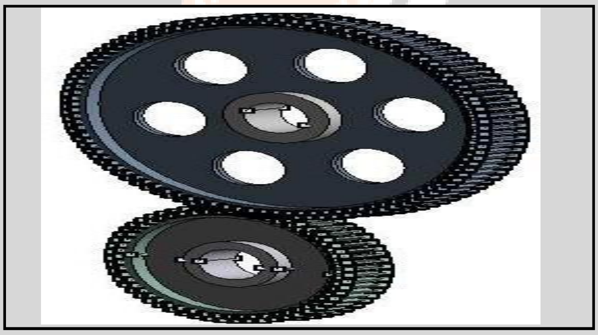

Fig- 4: Baseline Geometry of Model

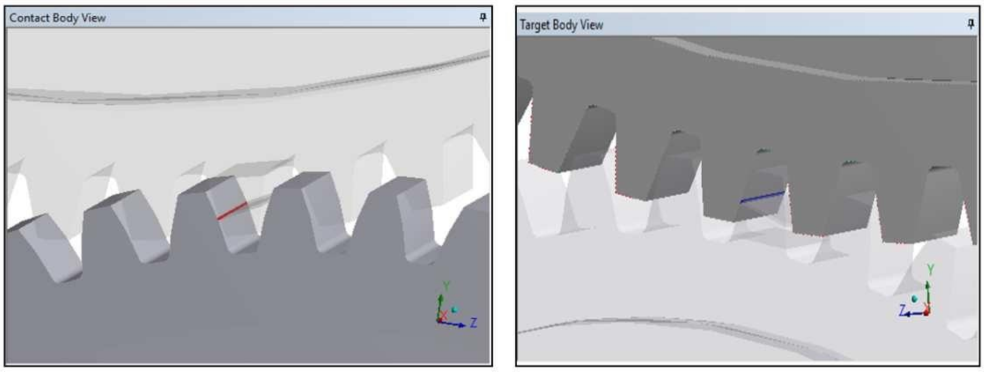

Fig- 5: Contact of Gears Baseline

The assembly is meshed with tetrahedron higher order elements with 1207498 nodes &492052 elements.

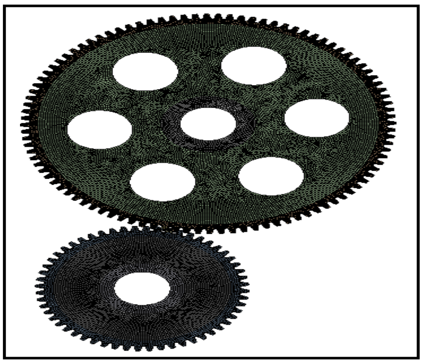

Fig- 6: Meshing of Gears Baseline

The shaft mounting region of the gear is fixed & shaft mounting region of the pinion is defined with frictionless support which is free to rotate. The moment of 5936.5 N-mm is applied at the shaft mounting region of pinion in clockwisedirection w.r.t x axis as shown in the figure below.

Fig-7: Boundary Conditions Baseline and Loading Base line

The maximum equivalent stress is found to be 32.289MPa at the contact region & minimum at other than contact Region.

Fig-7: Equivalent Stress Baseline

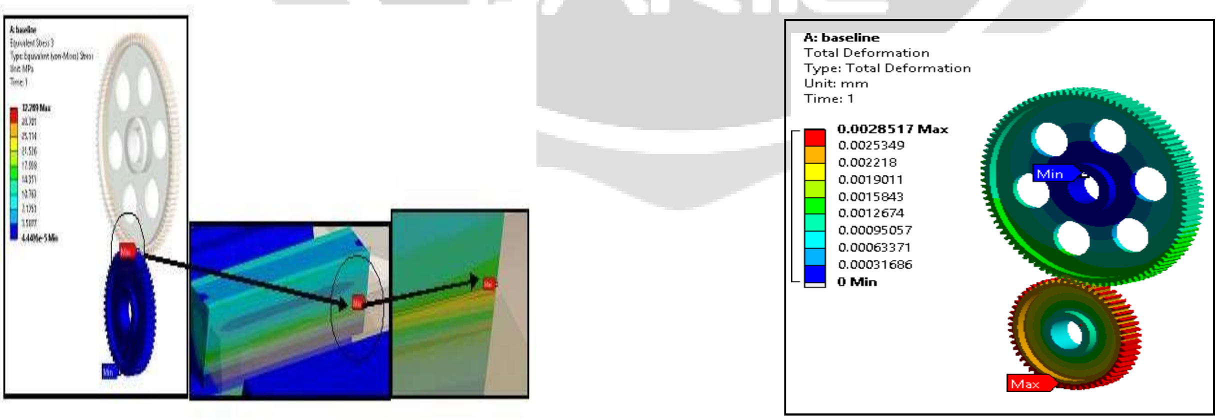

Fig-8: Equivalent Stress Details Baseline

Table-7: Iterations against No. of Holes and Thickness

Iteration No.	Hole Diameter (mm)	No. of Holes on Gear & Pinion	Thickness (mm)	Fillet Radius (mm)
1	12	6	4	0.45
2	12	3	3	0.6
3	12	2	2	0.75
4	10	6	3	0.75
5	10	3	2	0.45
6	10	2	4	0.6
7	8	6	2	0.6
8	8	3	4	0.75
9	8	2	3	0.75

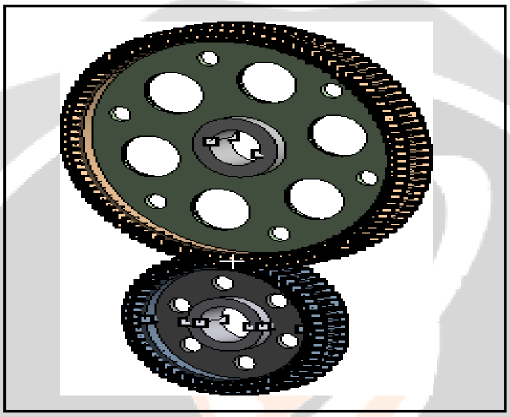

Fig-9: Geometry of Gears Iteration 4

The maximum equivalent stress is found to be 22.596MPa at the contact region & minimum at other than contact region.

3. RESULT AND DISCUSSION

Table-8: Analytical Results

Sr. No.	Analytical Result	Analysis Result	After Optimization
1	**162.85 N/mm^2**	**160.50 N/mm^2**	**131.50 N/mm^2**

1. From above result we can conclude that Maximum contact stress is 160.50 N/mm^2 and after optimization stress is 131.50 N/mm^2 so it is observed that reduction in stress.
2. Initially weight of Gear is 749gm and After Topology weight of gear is 638gm so there is 111gm reduction

Table-9: Observation

Parameter	Deflection determined by FEA Method (MM)	Deflection determined by Experimentation(MM)	Percent variation in results (Analysis vs. Experiment)
Deflection	**0.010**	**0.012**	**8.33%**

All the analysis results are listed in the table below. From those results final design of the gears will be selected

Table 10: Result Summary of all the Iterations in FEA

Iteration	Equivalent Stress (MPa)	Total Deformation (mm)	Pressure (MPa)	Mass of Gear (g)	Mass of Pinion (g)	Total Mass (g)
Baseline	32.239	0.00285	233.33	748.42	346.26	1094.68
1	39.078	0.00302	276.05	726.62	324.67	1051.29
2	33.206	0.0034	246.25	651.41	311.57	962.98
3	24.935	0.00415	236.63	567.53	289.68	857.21
4	22.596	0.00331	213.69	648.92	308.81	957.73
5	27.76	0.00424	308.1	566.28	288.9	855.18
6	32.425	0.00287	238.35	743.48	341.32	1084.8
7	28.817	0.00425	286.06	565.73	288.15	853.88
8	21.64	0.00278	206.81	744.3	341.32	1086.12
9	38.64	0.00332	280.25	656.55	316.91	973.46

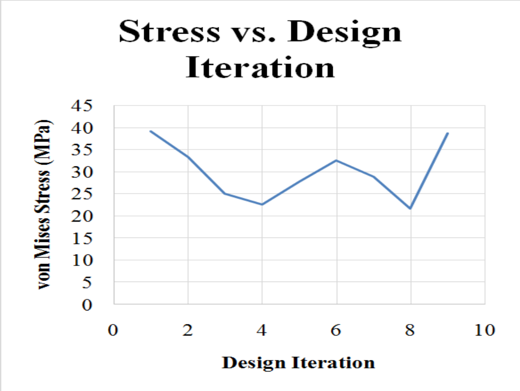

Figure 10: Maximum Von Mises Stress Vs Design Iteration

Iteration wise von-mises stress plot shows that iteration 8 has lowest value of von mises stress 21.6 MPa in all the iterations. While iteration 4 shows the second lowest 22.6 MPa.

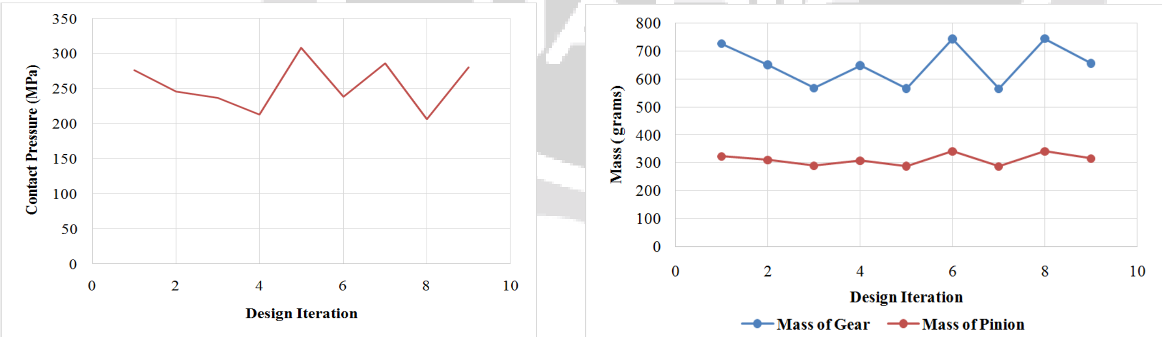

Figure 11: Design Iteration Vs Contact Pressure and Mass of Gears Vs Design Iterations

Similar pattern to von Mises stress is followed by contact pressure plot it shows the lowest value of 206 MPa while 213 MPa in iteration 4 is observed.

Table-11: Comparison of Baseline and Optimized Gears

Gear Assembly	Equivalent Stress (MPa)	Contact Pressure (MPa)	Mass of Gear (gm)	Mass of Pinion (gm)	Total Mass (gm)
Baseline	32.289	238	748.42	346.26	1094.68
Optimized	22.596	213.69	648.92	308.81	957.73
Percentage reduction	30.01%	10.21%	13.29%	10.81%	12.51%

It can be clearly observed that 30.01% of Equivalent stress, 10.21% of contact pressure is reduced& 12.51 % of total mass is reduced when compared to baseline gear assembly with modified design. Creating holes and reducing the thickness of the gear has huge impact on the weight of the gears while fillet radius has the large impact on the contact stresses of the gear. By changing the current conventional design parameters we can achieve better performance of the gearunder contact pressure as well as in equivalent stress while reducing the weight of the gear. FEA results of contact pressureare in conformance with the theoretical calculations. Optimized model results are in conformance with the testing results. Strain values of the FEA and practical testing are matching with the error of 7.96 % which is acceptable as less than 10 % error.

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