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ISSN:2348-2079 Volume-5 Issue-2

International Journal of Intellectual Advancements and Research in Engineering Computations

Thermal-stress analysis and gearbox testing of spur gear tooth with relief holes

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Abstract— This paper describes my attempt to investigate the effect of relief features of different size, location and number through stress and thermal analysis. The study is made through ANSYS using the finite element analysis (FEA) and Lewis Bending equation for calculation of forces involved in gear tooth. The result obtained from ANSYS is compared with the theoretical value of Lewis bending stress for the accuracy of the FEA model. Relief features are tried out with different positions and sizes in the involute spur gear tooth. Readings from ANSYS are tabulated and graphs are plotted. Comparison of graphs from stress and thermal analysis are then done to obtain the optimized result. The optimized position and size of the relief features for pinion is to be determined and its effects on stress and temperature distribution are tested in gearbox test rig.

A pair of gear teeth in action is generally subjected to two types of cyclic stresses: bending stresses inducing bending fatigue and contact stress causing contact fatigue. Both these types of stresses may not attain their maximum values at the same point of contact. However, combined action of both of them is the reason of failure of gear tooth leading to fracture at the root of a tooth under bending fatigue and surface failure, like pitting or flaking due to contact fatigue. However the fracture failure at the root due to bending stress and pitting and flaking of the surfaces due to contact stress cannot be fully avoided. These types of failures can be minimized by careful analysis of the problem during the design stage and creating proper tooth surface profile with proper manufacturing methods.

I. GEAR DESIGN

In the spur gearbox, EN353 Low alloy steel was used as driving gear and driven gears. The gears were carburized and quenched and oil annealed, measured using a Brinell hardness tester and was 546BHN at the spur gear surface.

A. *Selection of Gear*

Gear Material Composition:

- EN 353 Steel.(15NiCr1 Mo12)
- 0.17-0.22% Carbon.
- 1-1.4 % Manganese.
- 0.15-0.35 % Silicon.
- 1.0-1.3 % Chromium.

Gear Material Properties:

- Tensile Strength (σ_t) = 1820 N/mm^2.
- Yield Strength (σ_y) = 1270 N/mm^2
- Min. endurance limit stress (σ_{-1}) = 600 N/mm^2.
- Modulus of Elasticity (E) = 2.1x10^5 N/mm^2.
- Poisson Ratio (γ) = 0.3.
- Density = 7817x10^{-9} kg/mm^3.

B. Gear Specification

TABLE I
GEAR SPECIFICATIONS

Parameter	Pinion	Mating Gear
Profile	Involute	Involute
Pressure Angle	20 Degree	20 Degree
Pitch Circle Diameter	80 mm	120 mm
No of Teeth	32	48
Module	2.5	2.5
Addendum	2.5 mm	2.5 mm
Dedendum	3.125 mm Including Clearance	3.125 mm Including Clearance

Hardness	546 BHN	546 BHN
Face Width	30 mm	30 mm
Root Circle Diameter	74.55 mm	55.63 mm
Out Circle Diameter	85 mm	125 mm
Tooth Thickness	3.927 mm	3.927 mm

II. GEAR MODELLING

The steps followed in making gear tooth using CAD software are as shown in Fig 1.

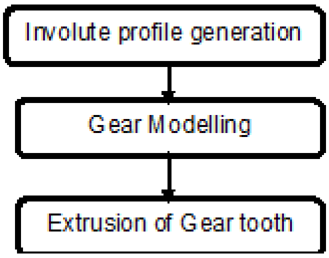

Fig. 1 Steps involved in Modelling of Gear

The spur gear of specified properties and specification has been modeled using the software Pro-E. Spur gear is designed using Unwin's construction of involute profile generation as shown in Fig.2 and an array of tooth is removed which shows a gear with single tooth as in Fig.3.

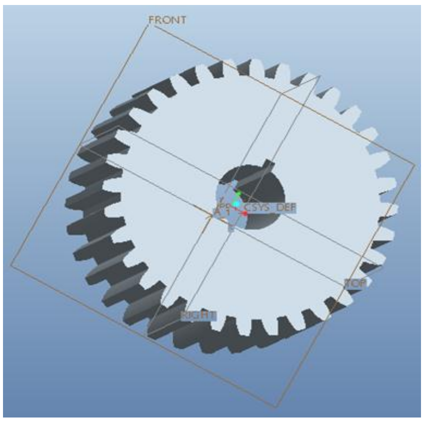

Fig. 2 Unwin's Constructed Spur gear

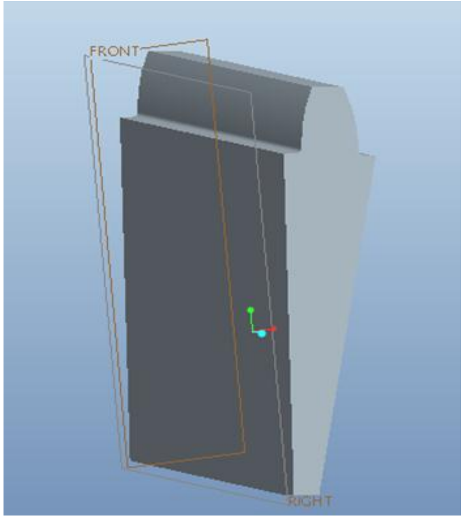

Fig. 3 Extruded Gear tooth

A. Gear Strength and FOS

For larger face width, lesser will be the static and dynamic stresses and gear tooth will be stronger. For the given module of 2.5 mm, face width will be 25 mm. But, actual face width of 30 mm is used, So, the gear is sufficiently strong for our consideration. Calculations and analysis should be within the yield stress and allowable working stress limit under the Factor of Safety (FOS)

Factor of Safety = 1.25
Ultimate Tensile strength = 1820 N/mm^2
Yield strength = 1270 N/mm^2
Allowable working stress = (1270 / FOS)
= 850 N/mm^2

B. Lewis Bending Equation

It is given by,

$$\sigma_w = F_t \times P_d / b \times Y \quad (1)$$

- F_t - tangential load
- P_d - Diametral Pitch
- b - Face width
- y – Form factor

Assumptions are as follows,

- The effect of radial component (F_r) which induces compressive stresses is neglected.
- It is assumed that the tangential component (F_t) is uniformly distributed over the face width of the gear.
- It is assumed that at any time only one pair of teeth is in contact and takes total load.

C. Force Calculation

By Lewis equation,

$$F_t = \sigma_w \times b \times (3.14) \times m \times y \quad (2)$$

Allowable stress, $\sigma_w = \sigma_t/3$ = 1820/3 = 606.66 N/mm^2

Face width, b = 30 mm

module, m = 2.5

Form Factor, $y = 0.154 - (0.912/z) = 0.154 - (0.912/32)$

$= 0.1255$

Tangential force, $F_t = 606.66 \times 30 \times (3.14) \times 2.5 \times 0.1255$

$= 17939.08$ N

$F_t = 550$ N/mm facewidth.

This tangential force value can be applied at the mating nodes in the Ansys stress analysis.

III. FINITE ELEMENT ANALYSIS OF GEAR TOOTH

Initially, the finite element models and solution methods needed for the accurate calculation of two dimensional spur gear bending stresses and temperature distribution were determined. Then, the temperature distribution and bending stresses were obtained using ANSYS 9.0 and compared to the results obtained from with and without hole features. The procedure of both stress and thermal analysis are as follows,

- Start → All programs → ANSYS 9.0 → Mechanical ADPL (ANSYS).
- File Menu → Import → Modeled Gear tooth.
- ANSYS Main Menu→ Preferences → Structural
  Pre-processor→ Element Type→ Add → Solid →Tetra 10node 92.

Fig.4 shows the library of element types and the highlighted one is selected for analysis.

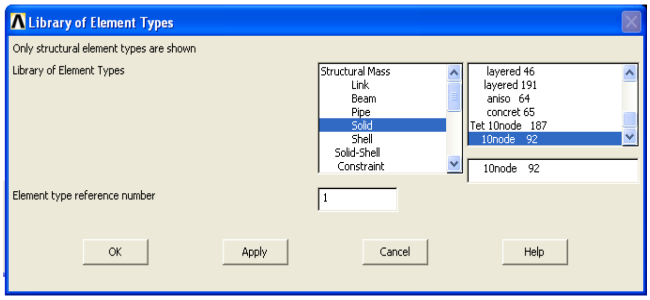

Fig. 4 Element Type

- Material Properties → Material Model→ values of Density and Young's modulus are given.
- Modelling → Create → Keypoints (4.06, 39.79, 0), (4.06, 39.79, 30), (8.13, 39.17, 30) and (8.13, 39.17, 30)
- Workplane Menu→ Align workplane → keypoints. Pick keypoints.
- Modelling →Operate →Boolean →Divide →Volume →Volume by workplane.

Gear tooth is imported from Pro-E and workplane is created so that it forms a platform for applying force on gear tooth as shown in Fig.5.

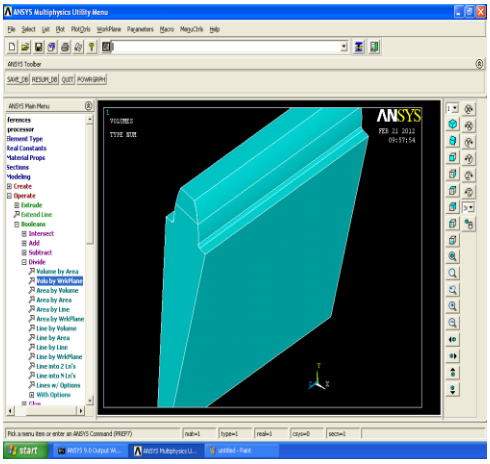

Fig. 5 Gear Tooth in Ansys

Meshing determines the no. of elements and element size required for the analysis. It is performed by Mesh tool as shown in Fig 6. Meshing can be done by choosing the option Ansys Main Menu → Meshing → Mesh tool. Then element attribute is given as global and size of the mesh is given as smart size of 6. Then mesh option is selected over the whole volume of the single gear tooth model. Smart size – 6 is selected for finite mesh and less processing time. The meshed gear tooth is shown in Fig 6.

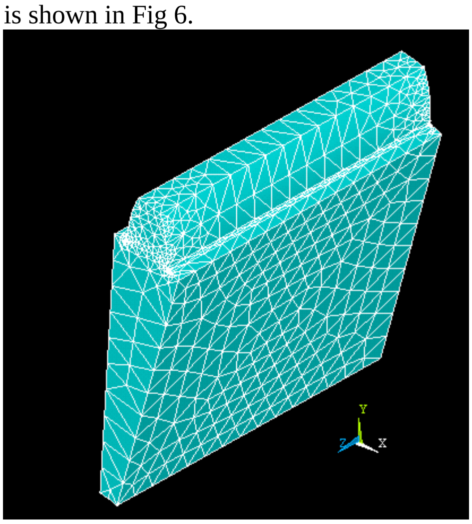

Fig. 6 Meshed Gear tooth

A. Stress Analysis

The load acting during the entire period of engagement is not uniform. A new stage of contact is affected when a single pair of teeth is in contact near the pitch circle. This is called a single tooth contact. Hence, loads are applied along the line of action in the pitch circle.

The axial and radial components of this load are

F_x= F_t/(No of nodes along pitch line)× cos(11.72)
= 566.615 N
F_y= F_t/(No of nodes along pitch line)× sin(11.72)
= 117.546 N

Using "Plot Lines" option, gear tooth is displayed with lines. From the lines plotted, the particular line is selected in one phase of gear tooth. Using "Plot Nodes" option, nodes are displayed as shown in Fig 7.

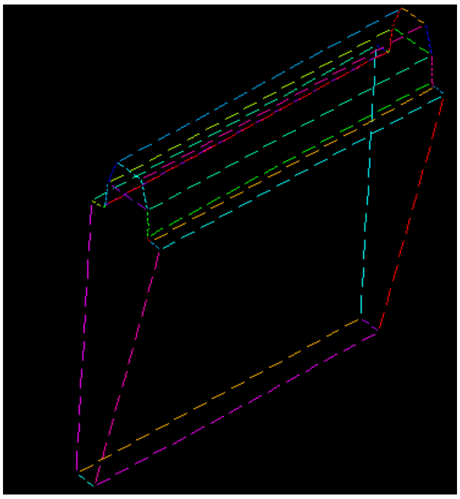

Fig. 7 Gear Tooth with Plot lines

FEA model in Figure.8 presents a gear tooth profile that is limited from the sides and bottom by a constrained border with stationary nodes. All other nodes on the tooth profile and inside the tooth contour are movable. The fillet portion of the tooth profile (where maximum bending stress is expected) has equally spaced nodes with higher density (number of nodes per unit of profile length) than the rest of the tooth profile. The nodes on the involute profiles and the top land are located to have higher density close to the fillets and lower density in the top part of the tooth.

The tooth load distribution problem is considered to define a value, a set of application point coordinates, and the direction of the force resulting in maximum bending stress. The friction effect at the contact point has been ignored. The load application point typically does not exactly match with a tooth profile node. It is replaced by a pair of forces i.e., x and y components.

Higher the number of nodes, larger will be the processing time. Then, Force components are applied in each of these nodes as shown in Fig 8. Total no. of equations determine the processing time. Higher the mesh size and element size, higher will be the total no. of equations and higher the processing time.

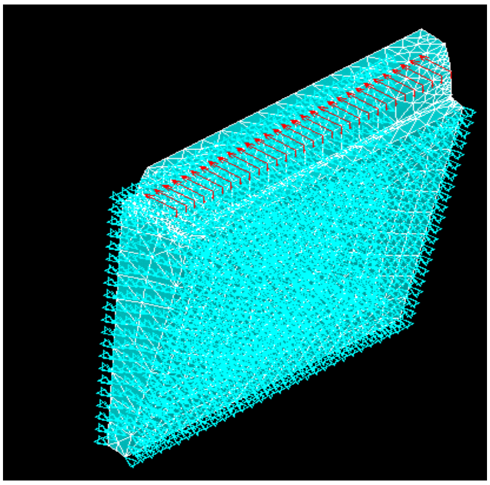

Fig. 8 Applied Forces on Tooth nodes

- When the option Solve → Current L.S is selected, then the Ansys begins to process the solution.
- First, Sparse solver dialog box appears on the screen as soon as the process begins
- Then, the sparse solver calculates the number of elements and the number of nodes in the single gear tooth model.
- In each element and node, the solver processes the process.
- In the next step, the sparse solver dialog box shows the number of the total equation involved in the problem and the number of the equation that has been processing.
- When the total number of equation has been solved, it is indicated by the Status command dialog box.
- After a time span of about 5 minutes, processing is over and the dialog box displays a note "Solution is done" as shown in Fig 9.
- Solution Menu →Solve →Current L.S. →Solution is done.
- Solution →Stress in Y component and Von Mises stress and DOF solution.

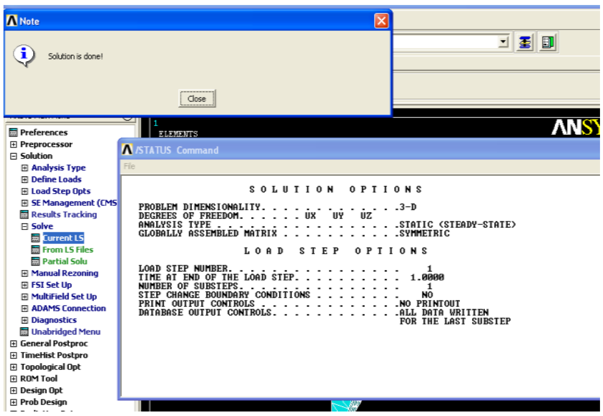

Fig. 9 Solution Status command

B. Thermal Analysis

Thermal properties of the material are,

- Thermal conductivity, k = 51.9 W/mK
- Specific Heat, C = 446 J/kgK
- Density = 7817 kg/m^3
- Heat Flux, q = 3.549 W/mm^2
- Convection heat transfer coefficient
  h = 25 W/m^{2}K
- Heat coefficient of grease used
  = 200 W/m^{2}k
- Efficiency = OP/IP = 90%
- Input power = 7.5 HP
- Losses = 10%
  0.9 = 1 – (Heat gen./input power)
- Heat generated = 0.1 (I.P)
  = 0.75 HP
  = 0.56 kW

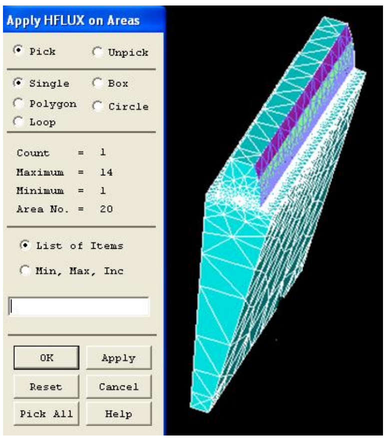

Fig. 10 Meshed Gear tooth showing heat flux area

Fig.10 shows the meshed gear tooth after applying the boundary condition of the hear flux on the surface area.

IV. FEA ANALYSIS OF RELIEF HOLES AT DIFFERENT SIZES AND POSITIONS

A. Stress Analysis

The procedure followed for the stress analysis of standard gear tooth is same for the modified gear tooth with holes.In this section, stress analysis of modified gear tooth have been analysed by the varying the size of the holes and varying the position of the hole.

1) Variation of Hole Size in Base Circle Diameter: Sizes of holes are varied in Base Circle Diameter as shown in Fig 11 with their corresponding stress values at the root of gear tooth are shown in Table II and plotted as graph for analysing the gear tooth with hole at the base circle diameter by varying the hole sizes.

TABLE II
STRESSES W.R.T DIFFERENT HOLE SIZES AT BCD

Diameter (mm)	Y - Component of stress (N/mm^2)	von Misses stress (N/mm^2)	% of stress reduction
0.4	351.708	409.193	31.57 %
0.5	358.946	413.673	30.82 %
0.6	365.918	420.365	29.7 %
0.7	408.899	430.527	28 %
0.8	427.83	467.814	21.76 %
1	450.851	474.837	20.6 %
1.2	457.923	497.405	16.81 %
1.5	471.344	516.649	13.6 %
1.8	595.64	642.367	No stress reduction

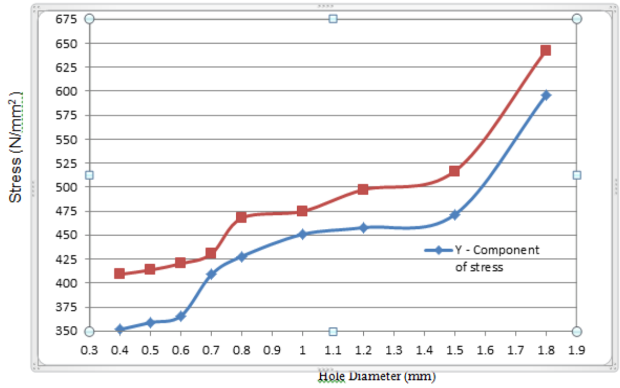

Fig. 11 Stress distribution of various hole sizes at BCD

Within the allowable bending stress limit (597.97 N/mm^2), the size of holes that are permitted are Base Circle Diameter 0.4 to 1.6 mm dia .

2) Variation of Hole Size in Pitch Circle Diameter: Sizes of holes are varied in Pitch Circle Diameter as shown in Fig 12 with their corresponding stress values at the root of gear tooth are shown in Table III and plotted as graph for analysing the gear tooth with hole at the pitch circle diameter by varying the hole sizes.

TABLE III
STRESSES W.R.T DIFFERENT HOLE SIZES AT PCD

Diameter (mm)	Y - Component of stress (N/mm^2)	von Misses stress (N/mm^2)	% of stress reduction
0.4	383.469	418.199	30 %
0.5	380.667	421.178	29.5 %
0.6	480.813	515.216	13.8 %
0.7	477.595	569.668	4.73 %
0.8	491.459	582.419	2.6 %
1	495.142	583.12	2.48 %
1.2	496.897	592.387	0.93 %
1.5	516.625	597.33	0.934 %
1.8	523.258	620.521	No stress reduction

TABLE IV
STRESSES W.R.T DIFFERENT HOLE SIZES AT RCD

Diameter (mm)	Y - Component of stress (N/mm^2)	von Misses stress (N/mm^2)	% of stress reduction
0.4	405.019	424.247	29.05 %
0.5	410.425	428.452	28.35 %
0.6	411.037	437.062	26.9 %
0.7	414.688	440.005	26.41 %
0.8	425.693	452.902	24.26 %
1	466.578	509.703	14.76 %
1.2	474.62	527.133	11.85 %
1.5	476.501	548.216	8.32 %
1.8	479.44	585.451	2.09 %

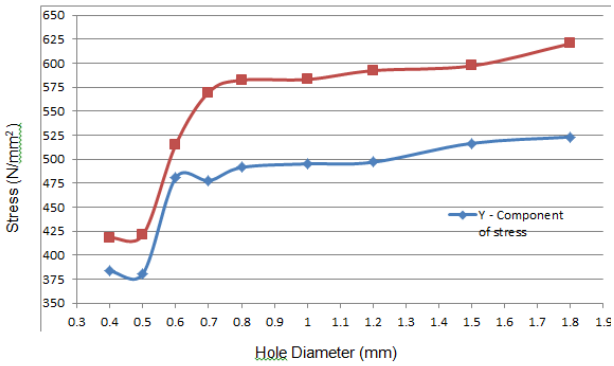

Fig. 12 Stress distribution of various hole sizes at PCD

Within the allowable bending stress limit (597.97 N/mm^2), the size of holes that are permitted are Pitch Circle Diameter 0.4 to 1.4 mm dia.

3) Variation of Hole Size in Root Circle Diameter: Sizes of holes are varied in Root Circle Diameter as shown in Fig 13 with their corresponding stress values at the root of gear tooth are shown in Table IV and plotted as graph for analysing the gear tooth with hole at the root circle diameter by varying the hole sizes.

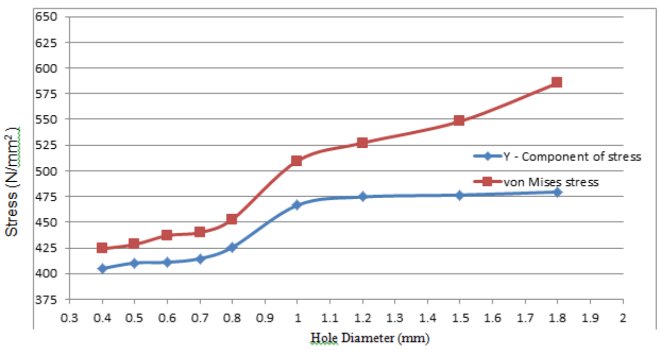

Fig. 13 Stress distribution of various hole sizes at RCD

Within the allowable bending stress limit (597.97 N/mm^2), the size of holes that are permitted are Root Circle Diameter 0.4 to 1.8 mm dia.

B. Thermal Analysis

The procedure followed for the thermal analysis of standard gear tooth is same for the modified gear tooth with holes. In this section, thermal analysis of modified gear tooth have been analysed by the varying the size of the holes and varying the position of the hole.

1) Variation of Hole Size in Base Circle Diameter: Sizes of holes are varied in Base Circle Diameter and the corresponding temperature readings are taken along the thickness of gear tooth for each 0.4 mm and shown in table V.

TABLE V

TEMPERATURE DISTRIBUTION W.R.T DIFFERENT HOLE SIZES AT BCD

Diameter of holes (mm)	Distance (dx) along thickness of gear tooth									
0.2	44.61	46.32	48.03	49.74	51.45	53.16	54.87	56.58	58.29	[illegible]
0.25	44.59	46.30	48.01	49.73	51.44	53.15	54.87	56.58	58.29	[illegible]
0.3	44.57	46.28	48	49.71	51.41	53.14	54.86	56.57	58.29	[illegible]
0.35	44.55	46.26	47.98	49.7	51.41	53.13	54.85	56.57	58.28	[illegible]
0.4	44.51	46.23	47.95	49.67	51.39	53.11	54.84	56.56	58.28	[illegible]
0.45	44.47	46.2	47.92	49.65	51.37	53.1	54.82	56.55	58.27	[illegible]
0.5	44.43	46.16	47.89	49.62	51.35	53.08	54.81	56.54	58.27	[illegible]
0.6	44.32	46.06	47.8	49.55	51.29	53.03	54.77	56.52	58.26	[illegible]
0.75	44.11	45.87	47.64	49.41	51.17	52.94	54.7	56.47	58.23	60
0.9	43.85	45.64	47.44	49.23	51.03	52.82	54.62	56.41	58.21	60

2) Variation of Hole Size in Pitch Circle Diameter: Sizes of holes are varied in Pitch Circle Diameter and the corresponding temperature readings are taken along the thickness of gear tooth for each 0.4 mm and shown in table VI.

TABLE VI

TEMPERATURE DISTRIBUTION W.R.T DIFFERENT HOLE SIZES AT PCD

Diameter of holes (mm)	Distance (dx) along thickness of gear tooth									
0.2	44.487	46.211	47.934	49.658	51.382	53.105	54.829	56.553	58.276	60
0.25	44.403	46.136	47.869	49.602	51.335	53.068	54.801	56.534	58.267	60
0.3	44.3	46.044	47.789	49.533	51.278	53.022	54.767	56.511	58.256	60
0.35	44.187	45.944	47.701	49.458	51.215	52.972	54.729	56.486	58.243	60
0.4	44.045	45.818	47.591	49.364	51.136	52.909	54.602	56.455	58.227	60
0.45	43.888	45.679	47.469	46.259	51.049	52.839	54.629	56.42	58.21	60
0.5	43.708	45.518	47.328	49.138	50.949	52.759	54.569	56.379	58.19[illegible]	60
0.6	43.295	45.151	47.007	48.863	50.72	52.576	54.432	56.288	58.144	60
0.75	42.526	44.468	46.409	48.351	50.292	52.234	54.175	56.117	58.058	60
0.9	41.534	43.586	45.637	47.689	49.741	51.793	53.845	55.896	57.948	60

3) Variation of Hole Size in Root Circle Diameter: Sizes of holes are varied in Root Circle Diameter and the corresponding temperature readings are taken along the thickness of gear tooth for each 0.4 mm and shown in table VII.

TABLE VII

TEMPERATURE DISTRIBUTION W.R.T DIFFERENT HOLE SIZES AT RCD

Diameter of holes (mm)	Distance (dx) along thickness of gear tooth									
0.2	44.632	46.339	48.047	49.754	51.462	53.17	54.877	56.585	58.292	60
0.25	44.606	46.317	48.027	49.738	51.448	53.158	54.869	56.579	58.29	60
0.3	44.59	46.302	48.014	49.727	51.439	53.151	53.863	56.576	58.288	60
0.35	44.566	46.281	47.996	49.711	51.426	53.14	54.855	56.57	58.285	60
0.4	44.545	46.262	47.955	49.676	51.396	53.117	54.848	56.566	58.283	60
0.45	44.514	46.234	47.955	49.676	51.396	53.117	54.838	56.599	58.279	60
0.5	44.48	46.204	47.929	49.653	51.378	53.102	54.827	56.551	58.276	60
0.6	44.41	46.142	47.875	49.607	51.339	53.071	54.803	56.536	58.268	60
0.75	44.248	45.998	47.748	49.498	51.249	52.999	54.749	56.499	58.25	60
0.9	44.046	45.819	47.592	49.364	51.137	52.909	54.682	56.455	58.227	60

C. Inference

It is obvious that as the size of hole increases, the temperature within the gear tooth reduces. Significant change in temperature is not seen when considering small element or small region in a gear tooth. But, considering overall gear tooth, significant decrease in temperature is seen.

V. OPTIMIZATION OF SIZE AND POSITION OF RELIEF HOLE

Constraints to be considered for optimization are as follows:

- ➢ Maximum stress relief
- ➢ Maximum Temperature relief

From results of stress and thermal analysis, maximum stress relief is obtained for small diameter holes while maximum temperature relief is obtained for large diameter holes. So, an optimized size and position of hole has to select satisfying both the analysis. Comparison of von misses stresses are made to estimate the optimized size and position of hole in gear tooth as shown in Fig 14.

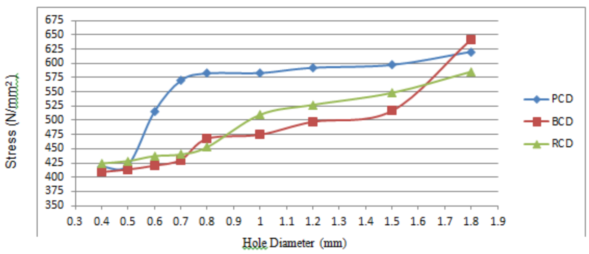

Fig. 14 Comparison of Von mises stresses

The optimized size and position of hole

- Position - BCD
- Size - 1.2 mm

Optimization of the fillet profile allows reducing the maximum bending stress in the gear tooth root area by 16.81%. The effect on stress and temperature distribution due to optimized hole is shown in Fig 15.

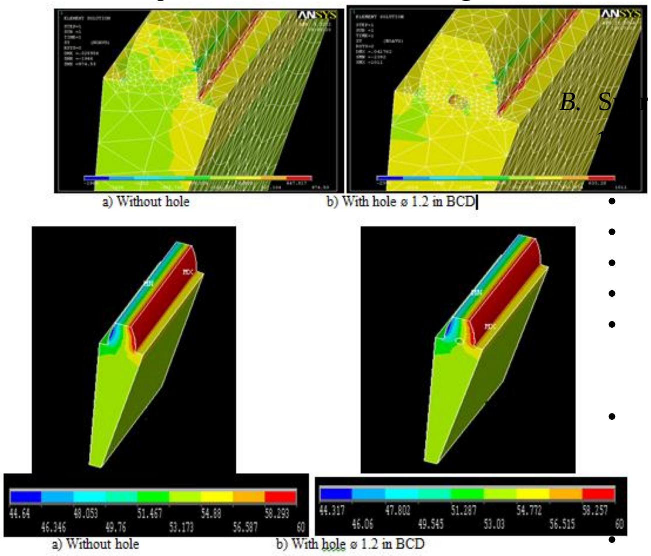

Fig. 15 Comparison of Stress and Temperature Distribution in optimized relief hole

The function of cooling holes

- decreases the surface temperature by emitting heat from the tooth to the external environment via convection.
- the service life is extended
- the thermal damage that occurs on the gear tooth surface is reduced.

The bending stress reduction leads to:

- Size and weight reduction
- Higher load application
- Cost reduction (less expensive materials, heat treatment, etc.)
- Noise and vibration reduction, increased efficiency.

VI. MANUFACTURING AND TESTING OF MODIFIED GEAR TOOTH WITH OPTIMIZED RELIEF HOLES

A. EDM Drilling of Relief Holes

Electric Discharge Machining is a slow machining process used to make holes in spur gear tooth. The process is done at A1 EDM Drilling, Ganapathy, Coimbatore. Machining cost varies according to the size of hole. Machining cost varies according to the size of hole as shown in Table VIII.

TABLE VIII
COST OF DRILLING HOLE

Diameter	Cost/mm
0.5	6
0.8	4
1	2.5
1.3	2.5
1.5	2.5

2	3

B. Spur Gearbox Testing of Modified Gear Tooth

Test rig components: The components of the test rig are as follows,

- Control panel
- Motor - 5.5 kW
- Fenner Coupling (Motor and dynamometer)
- Dynamometer
- Gearbox

Eddy current Dynamometer

- Make - Dynaspedz Integrated Systems Pvt. Ltd
- Model - 635000
- Type - P34 635000
- Br. Torque - 5.0 kg.m (at 1500 rpm)
- Poles - 48
- Coil - 85 V (short time) – 35 V

Gearbox

- Make - Flender
- Type - Spur Gearbox
- Reduction ratio - 1.5:1

Induction Motor

- No. of phase - 3
- Power - 5.5 kW
- Current - 11.5 A
- Frequency - 50 Hz
- Voltage - 415 V
- Speed - 1440 rpm

2) Temperature Testing: Temperature testing of standard gear tooth and modified gear tooth with optimized relief holes has been carried out in Spur Gearbox test rig. Temperature measurement arrangement is shown in the Fig. 16. Tests have been conducted at various load conditions as shown in the table IX, table X and table XI.

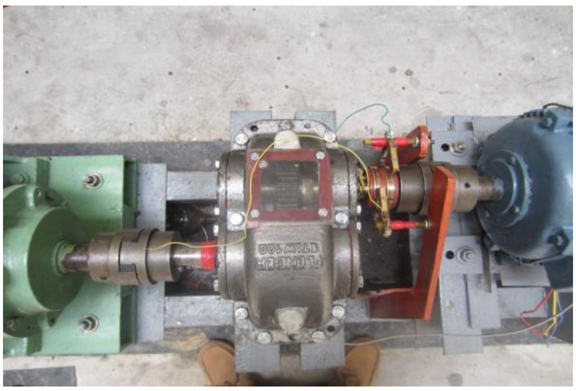

Fig. 16 Temperature Measurement arrangement

TABLE IX
TEMPERATURE AT NO LOAD CONDITION

Time (min)	Standard Spur gear tooth (°C)	Spur gear tooth with cooling holes (°C)	Temperature difference between standard & Modified gears (°C)
5	36.4	34.4	2
10	38.8	36.5	2.3
15	41.5	39.6	1.9
20	43.8	42.1	1.7
25	45.9	44.4	1.5
30	47.9	46.5	1.4
35	50.2	48.9	1.3
40	51.8	50.4	1.4
45	53.6	52.3	1.3
50	54.5	53.0	1.5
55	55.4	53.8	1.6
60	56.4	54.6	1.8

TABLE X
TEMPERATURE AT 10% (5 NM) LOAD CONDITION

Time (min)	Standard Spur gear tooth (°C)	Spur gear tooth with cooling holes (°C)	Temperature difference between standard & Modified gears (°C)
5	38.3	36.4	1.9
10	41.7	39.8	1.9
15	44.3	42.5	1.8
20	47.5	45.8	1.7
25	49.9	48.2	1.7
30	51.7	50.4	1.3
35	53.8	52.1	1.7
40	55.7	54.2	1.5
45	58.1	56.7	1.4
50	59.9	58.5	1.4
55	61.3	59.8	1.5
60	62.8	61.2	1.6

TABLE XI
TEMPERATURE AT 20% (10 NM) LOAD CONDITION

Time (min)	Standard Spur gear tooth (°C)	Spur gear tooth with cooling holes (°C)	Temperature difference between standard & Modified gears (°C)
5	40.9	38.9	2
10	43.6	41.7	1.9
15	47.2	45.5	1.8
20	49.9	48.1	1.8
25	53.2	51.5	1.8
30	55.1	53.4	1.7
35	57.6	56.0	1.6
40	59.9	58.6	1.3
45	62.3	60.7	1.6
50	63.9	62.5	1.4
55	65.2	63.8	1.4
60	66.4	64.8	1.6

It is evident that there is a notable temperature difference between the standard and modified gear tooth optimised relief holes.

VII. CONCLUSION

Finite Element Analysis (FEA) is used for analysis in ANSYS. Theories and facts related to gears, gearboxes, failures of gears are studied. Theoretical and analytical values are compared, verified and proceeded through ANSYS. An optimized size and position of hole is estimated through stress and thermal analysis. Optimization of the fillet profile allows reducing the maximum bending stress in the gear tooth root area by 16.81%.Holes are made through EDM in each gear tooth of spur gear. The modified spur gear is fitted in the Spur Gear Box Test

Rig and temperature measurement is taken. From those readings temperature difference between the standard and modified gear tooth optimised relief holes has been noted.

Scope of further work

- Experiments are to be conducted in the modified spur gear that involves surface roughness test, wear identification test etc., after running the gear for certain no. of cycles.
- Results obtained from these experiments are compared with the experimental results of standard spur gear without hole.
- Testing of vibration and fatigue are to be done with this modified spur gear using the test rig in the future.
- Other types of gears like helical, bevel, worm etc. can also be studied and different shapes of hole can also be studied for relieving stress.

ACKNOWLEDGEMENT

We take immense privilege to express our sincere thanks to our principal Dr. N.Kuppusamy who has been bastion of moral strength and source of innocent encouragement to us. We are highly indebted and graceful to our beloved professor, head and dean of the department Mr.M.Naveenkumar for his heartfelt support throughout this project work. We express our gracious gratitude to our beloved supervisor Mr.T.E.Narentharan for his guidance and support throughout our project. We also wish to thank all the teaching and non- teaching staff members of our department, our friends and well-wishers who are all behind the success of our project.

REFERENCES

[1] Alban L.E, "*Systematic Analysis of Gear Failures*", American Society for Metals, 1985

[2] Alexander Kapelevich, "*Geometry and design of involute spur gears with asymmetric teeth*", Mechanism and Machine Theory 35 (2000) 117-130

[3] ANSI/AGMA "*Gear Nomenclature, Definitions of Terms with Symbols*", 1012-F90, American Gear Manufacturers Association, 1990

[4] Autar K Kaw "*Mechanics of Composite Materials*", Second Edition, 2006.

[5] Boresi A.P and Sidebottom O.M, "*Advanced Mechanics of Materials*", 4th ed. John Wiley & Sons, 1985

[6] Boston Gear, Quincy, MA

[7] Callister, W., 1999, "*Materials Science and Engineering: An Introduction*", John Wiley and Sons Inc., New York

[8] Davis J.R, "*Gear Materials, Properties, and Manufacture*", ASM International, Edition 1, 2005.

[9] Dudley D.W, "*Handbook of Practical Gear Design*", McGraw Hill Book Company, 1984

[10] Ebubekir Atan, "*On the prediction of the design criteria for modification of contact stresses due to thermal stresses in the gear mesh*", Izmir 35430, Turkey, Tribology International 38 (2005) 227–233

[11] Gayatri Kansala, Raob P.N, Atreya S.K, "*Study: temperature and residual stress in an injection moulded gear*", Journal of Materials Processing Technology 108 (2001) 328-337

[12] Hayrettin Duzcukoglu, "*Study on development of polyamide gears for improvement of load-carrying capacity*", Tribology International 42 (2009) 1146–1153