PARAMETRIC ANALYSIS OF NON-METALLIC HELICAL GEAR BY USING FEM

Harshkumar Prabhudas Bumtariya[1], Sanjaykumar Asondariya[2]

[1] *Student, Department of Mechanical Engineering, Vadodara Institute of Engineering, Gujarat, India*

[2] *Professor, Department of Mechanical Engineering, Vadodara Institute of Engineering, Gujarat, India*

ABSTRACT

Gears are one of the most critical components in mechanical power transmission systems. The bending and surface strength of the gear tooth are considered to be one of the main contributors for the failure of the gear in a gear set. Thus, analysis of stresses has become popular as an area of research on gears to minimize or to reduce the failures and for optimal design of gears. This paper investigates the characteristics of an involute helical gear system mainly focused on bending and contact stresses using analytical and finite element analysis. Primary objective of this paper is to observe the effects of polymers which gives more benefits than the metallic gears like noise reduction, less vibration, lower maintenance, lower lubrication, low cost and easy to manufacture for the optimization. Static finite element analysis is required of both the materials. In this paper optimization method is presented for the comparison of metallic and non-metallic gear using static FEA.Helix angle, face width, pressure angle and module are important geometrical parameters in determining the state of stresses during the design of gears. Thus, in this work a parametric study is conducted by varying the helix angle, face width, pressure angle and module to study their effect on the bending stress of helical gear.

Keyword: -*Helical gear, Finite element analysis, Delrin-100, Bending stress, Von-Mises stress*

1. INRODUCTION

Gearing is one of the most effective methods transmitting power and rotary motion fromthe source to its application with or without change of speed or direction. Gears willprevail as a critical machine element for transmitting power in future machines due totheir high degree of reliability and compactness. The rapid development of heavyindustries such as vehicle, shipbuilding and aircraft industries require advancedapplication of gear technology.

A gearbox consists of a set of gears, shafts and bearings that are mounted in an enclosedlubricated housing. They are available in a broad range of sizes, capacities and speedratios. Their function is to convert the input provided by the prime mover into an outputwith lower speed and corresponding higher torque. In this thesis, analysis of thecharacteristics of helical gears in a gearbox is studied using finite element analysis.

Helical gears are currently being used increasingly as a power transmitting gear owing totheir relatively smooth and silent operation, large load carrying capacity and higheroperating speed. Designing highly loaded helical gears for power transmission systemsthat are good in strength and low level in noise necessitate suitable analysis methods thatcan easily be put into practice and also give useful information on contact and bending stresses. The finite element method is proficient to supply this information but thetime required to generate proper model is a large amount. Therefore to reduce themodeling time a preprocessor method that build up the geometry required for finiteelement analysis may be used, such as Pro/Engineer.

The major cause of vibration and noise in a gear system is the transmission error betweenthe meshing gears. By definition transmission error is the difference between thetheoretical and the actual position between driving gear and the driven gear. It can bedefined also as the amount by which the ratio at a given point in a revolution departsfrom the correct ratio. For this reason, with prior knowledge of the operating conditionsof the gear set it is possible to design the gears with minimum vibration and noise.

Gear analysis can be performed using analytical methods which required a number ofassumptions and simplifications which aim at getting the maximum stress values only butgear analyses are multidisciplinary including calculations related to the tooth stresses andto failures like wear. In this thesis, an attempt will be made to analyze static contactand bending stresses to resist bending of helical gears, as both affect transmission error.

Due to the progress of computer technology many researchers tended to use numericalmethods to develop theoretical models to calculate the effect of whatever is studied.Numerical methods are capable of providing moretruthful solution since they requirevery less restrictive assumptions. However, the developed model and its solution methodmust be selected attentively to ensure that the results are more acceptable and itscomputational time is reasonable.

2. PARAMETRIC MODELLING OF HELICAL GEAR IN PRO/ENGINEER

First of all, setup the geometric input parameters. For this case following are the input parameters:

- Number of teeth
- Module
- Pressure angle
- Pitch diameter
- Face width
- Helix angle

Part parameters are the basic parameters defining the gear which are used as the input while modeling. These part parameters determine all the other parameters that define the gear tooth profile by using theTools→Parameters menu.

Tools→Relations menu is used to define relationship equations between parts.

Draw a circle centered on the sketch references for the extrusion profile and takes theextrusion depth equals to the thickness of gear. By using the Tools / Relations Menuwe define relations between the sketch dimensions and the part parameters. Fig.1shows the Tools/Relations Menu. After defining these relations, the circle should have
the diameter equals to the diameter of the addendum diameter of the gear blank. Fig.1 shows the parameter relations to obtain the addendum diameter and the Fig.2shows the gear blank with required addendum diameter and thickness. Extrudecommand is used to convert 2-D sketch into 3-D gear blank model.

To generate the datum curve by using equations we select datum curve generated by equation and taking cylindrical coordinate system. Select the equations from the Insert/ Model Datum/ Curve Menu. Take“PRT_CSYS_DEF” as a default coordinate system. At this point a Notepad window will pop up where we can enter all equations for the datum curve as shown in the Fig.3. The preview shows the involute curve over the gear blank after entering all parameters. The generated involute curve is shown in fig.4.

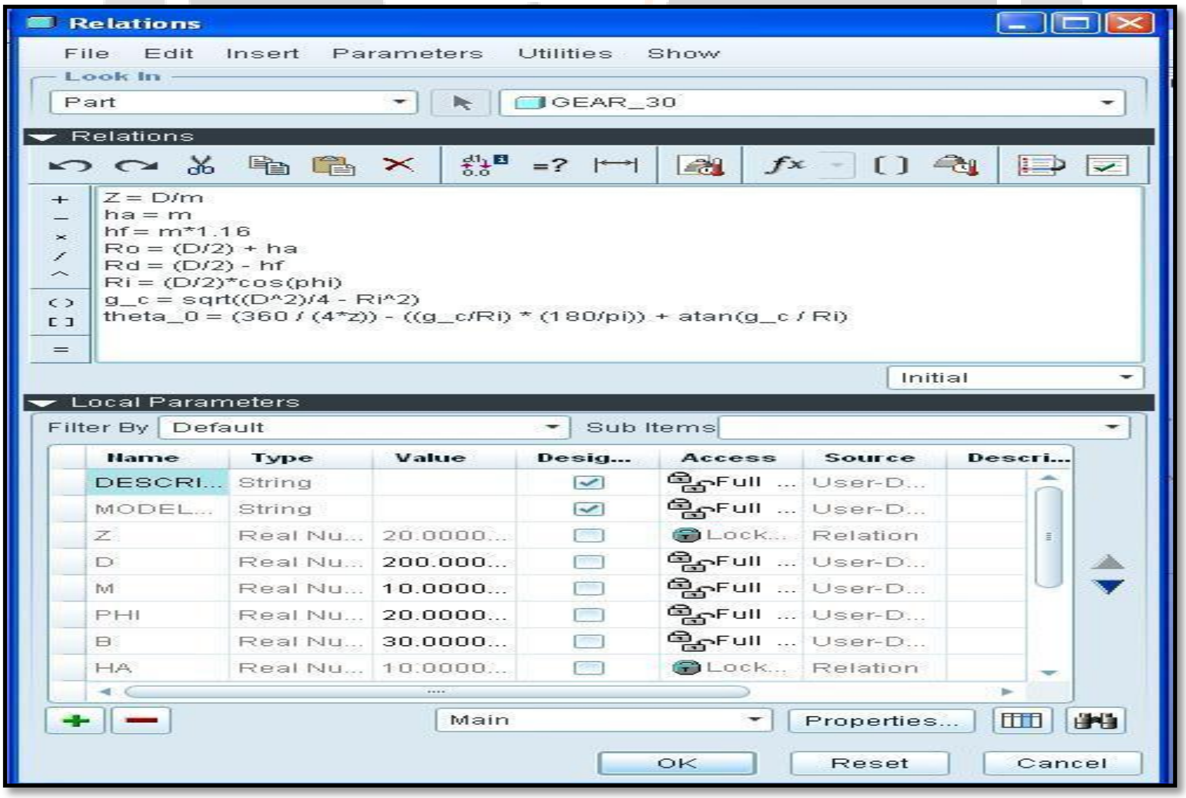

Fig -1 Parameter relation menu

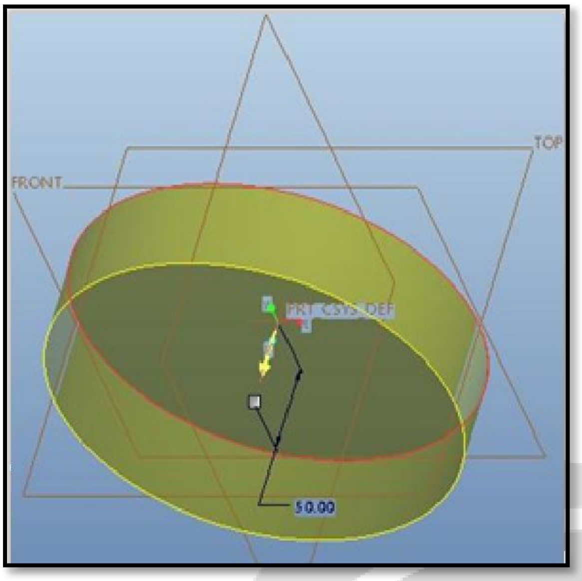

Fig -2 Gear blank

Fig -3 Equations to generate curve

Helix angle of the gear is generated by sketching a line using top plane. With the help of variable sweep section on this line, we can generate the profile of the gear as shown in fig.5. Here we relate the helix angle using Tools/Relation menu.

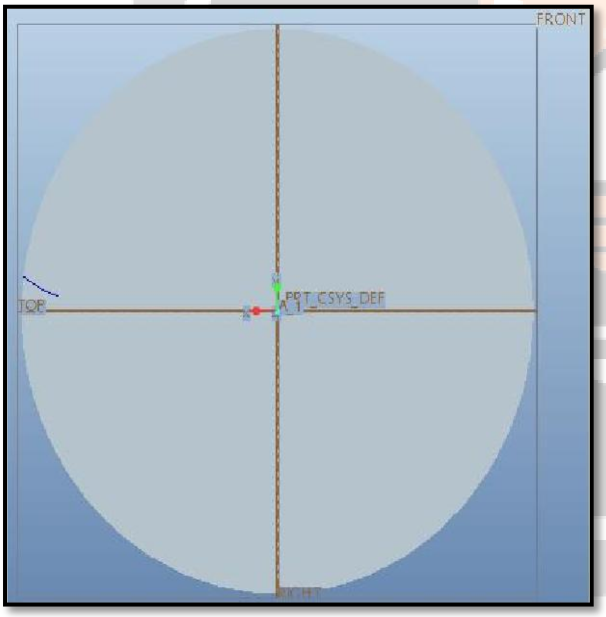

Fig -4 Curve generated on gear blank

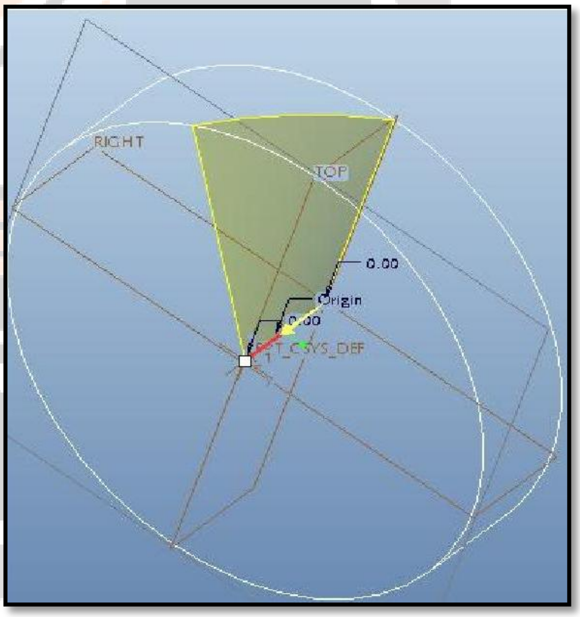

Fig -5 Helix angle for curve

Now, create the cross sectional area for tooth space using curve and this cross sectional area will be throughout guided by the helix angle shown in fig.7.

For latest variable sweep featureselect the pattern tool to make equal number ofteeth given in the table. Select following parameters in pattern dash board for pattern feature.

1) Pattern type: Axis pattern
2) Axis for pattern: A_1 at the centre of the gear
3) Number of copies: Equals to the number of teeth here 20
4) Included angle of the pattern: 360^o

After accepting we get shape of the gear as shown in fig.8.

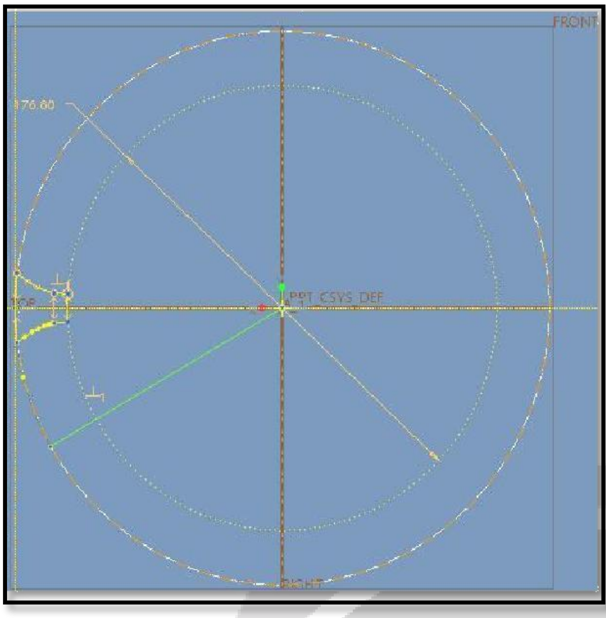

Fig -6 cross sectional area for tooth space

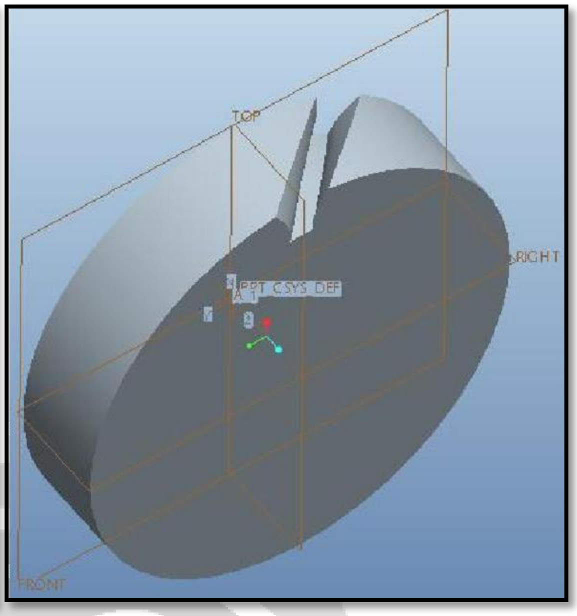

Fig -7 Gear blank with tooth space

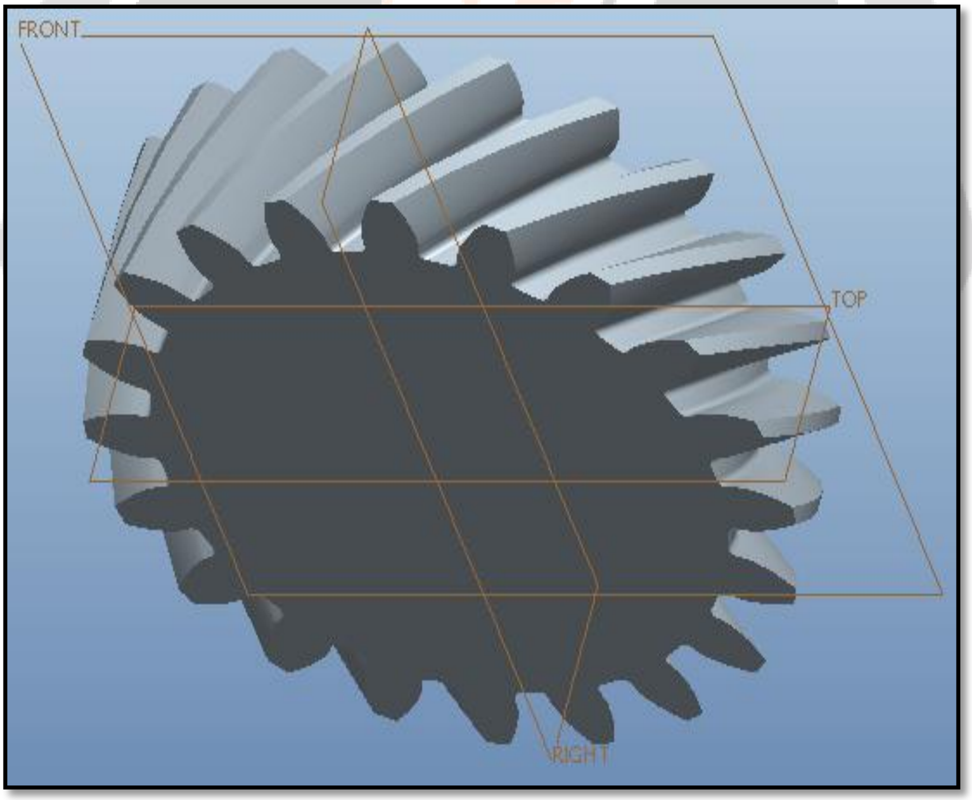

Fig -8 Complete helical gear model

3. COMPOSITIONS

Standard Delrin and Delrin P are available in several basic melt flow rates, i.e., Delrin100, 500, 900, and 1700 grades. These differ primarily in melt viscosity, with 100 being the most viscous and 1700 the most fluid. Delrin products are also available as specialty grades with additives that allow for enhanced UV stability, faster cycling, lower friction and wear, and toughness.

Among all the different grades of delrin compositionsselect delrin100 for thisanalysis. Different mechanical properties of the delrin100 are as shown below. All theproperties given below are at 23^{O}C.

Mechanical Property	Numerical value
Tensile Stress	69MPa
Shear Stress	66Mpa
Compressive Stress	36Mpa
Yield Stress	99Mpa
Modulus of Elasticity	2900Mpa
Poisson’s Ratio	0.35

Table -1 Delrin100 mechanical property

4. EFFECTS OF PARAMETERS ON HELICAL GEAR

To vary different parameters and checking the effects on the helical gear. It is natural that in metal gears deformations are very low, so in case of metal gears it is very hard to analyze the effect of those deformations. In non metal gears there will be large deformation on tooth compared to metal gears. So the effect of deformation is also higher in case of non metal gears.
AGMA conditions used for mathematical modeling and Pro/Mechanica for the numerical calculations. Here performed only static analysis and will obtain the Von Mises stress by varying helix angle, pressure angle, face width and module.

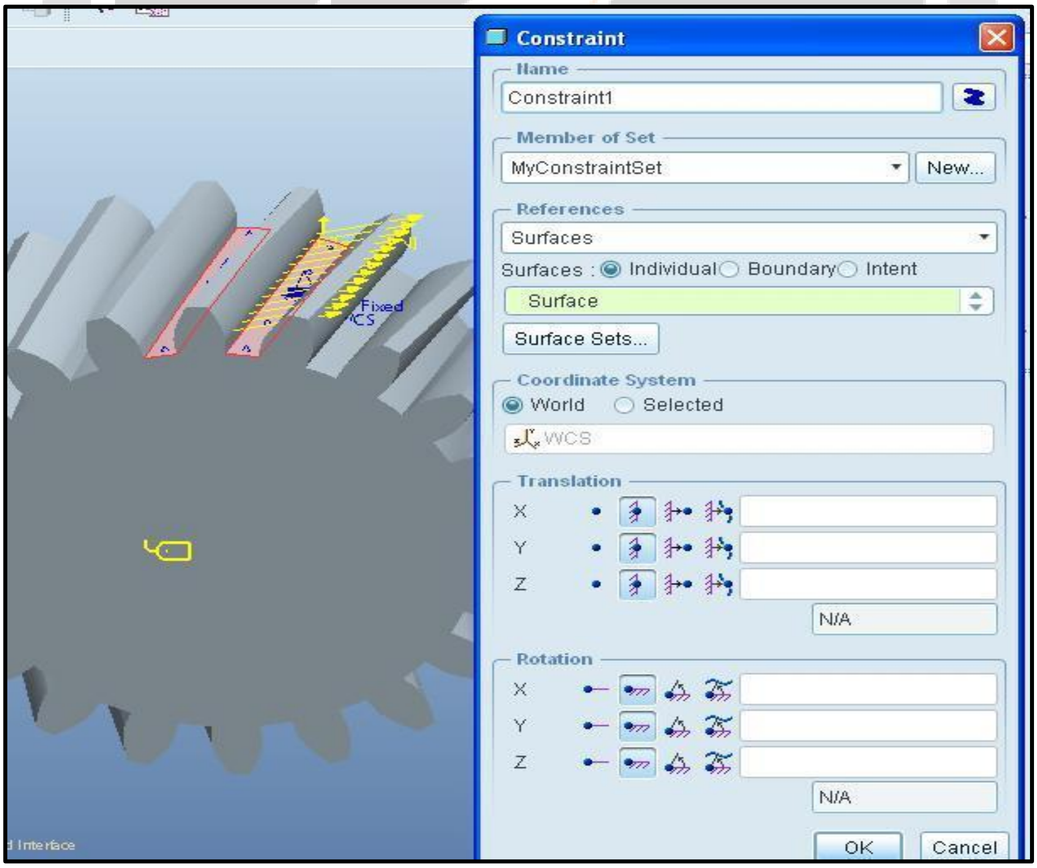

Fig -9 Gear with boundary conditions

The boundary conditions of the gear are as shown in Fig. 9. Treat gear tooth as the cantilever beam so all the degrees of freedom of both side of the gear tooth i.e. the tooth spaces are fixed. The load is taken on the pitch circle diameter of the gear.

The meshing of the gear is as shown in the figure 10. Element quality check menu isshown along the meshed gear which checks the quality of each and every element. Ihave set some parameters like aspect ratio, maximum and minimum edge angles etc. to generate sufficient numbers of element in each model. Near around 60,000 elements are generated for each model and I have generated only tetrahedral elements.

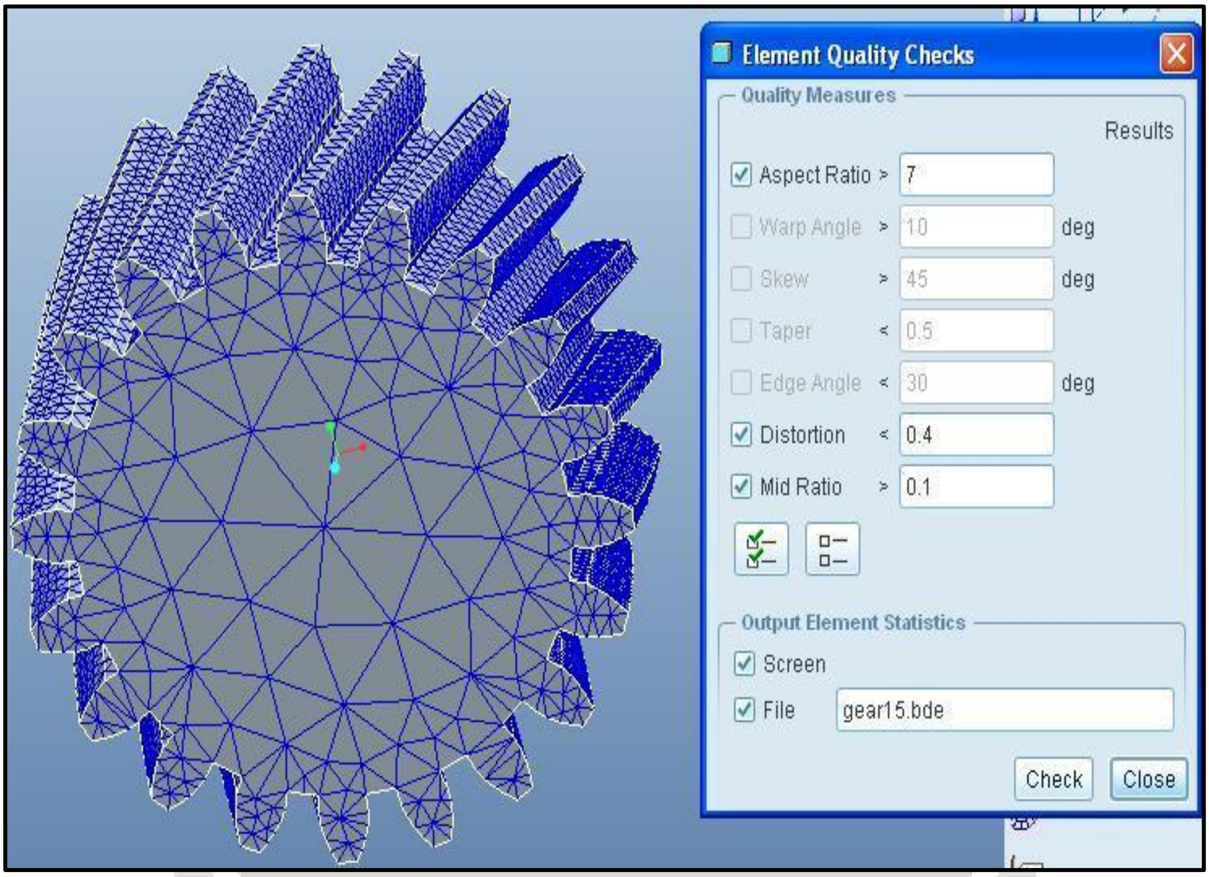

Fig -10 Gear in meshing condition and Element quality check menu

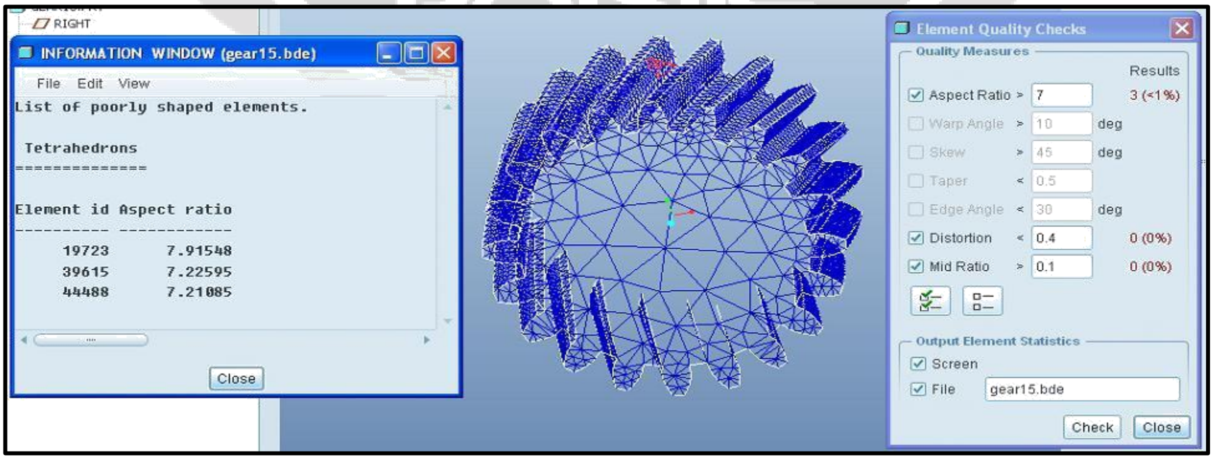

Fig -11 Poorly shaped elements menu

Now after checking the quality of the mesh the elements which poorly shaped and not satisfy the preliminary conditions are also shown in the following figure. Here only three elements are there which are poorly shaped which have element number 19723, 39615 and 44488.

The result of Pro/Mechanica software is as shown in the figure given below.

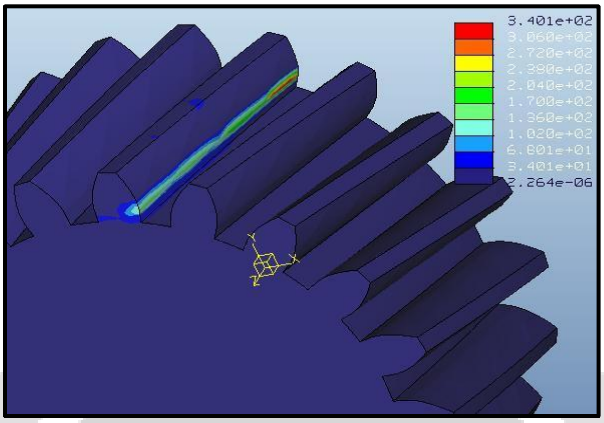

Fig -12 Von Mises stress on the gear tooth

The maximum stress is generated on the pitch circle diameter of the gear. Now check the effect of the different parameters on the tooth.

4.1EffectonTooth Strength by Varying Helix Angle

The value of the AGMA calculations, Pro/Mechanica calculation and difference between them is shown is the following table.

Helix angle	AGMA stress (N/mm^2)	Pro/Mechanica stress (N/mm^2)	Differance (%)
3	346.33	359.2	3.5829621
6	344.70	356.6	3.3370723
9	341.29	347.6	1.8153049
12	337.74	340.4	0.7814336
15	332.90	340.1	2.1170244
18	327.09	342.1	4.3876067
21	321.39	319.8	0.9718574
24	313.74	319.2	1.7105263
27	308.51	315.7	2.2774786
30	302.16	307.2	1.6406252
33	295.97	306.6	3.4670581

Table 4.1 Von-Mises stress obtained by varying helix angle

4.2 EffectonTooth Strength by Varying Face Width

The value of the AGMA calculations, Pro/Mechanica calculation and difference between them is shown is the following table.

Face width (mm)	AGMA stress (N/mm^2)	Pro/Mechanica stress (N/mm^2)	Difference (%)
42	367.28	381.7	3.777836
44	350.58	355.6	1.4116985
46	335.34	350.3	4.2706252
48	321.37	337.6	4.8074645
50	308.51	325.7	5.2778631
52	296.65	310.9	4.5834674
54	285.66	274.8	3.9519650
56	275.46	278.6	1.1270639

58	265.96	274.3	3.0404666
60	257.09	259.7	1.0050058

Table 4.2 Von-Mises stress obtained by varying face width

4.3EffectonTooth Strength by Varying Pressure Angle

The value of the AGMA calculations, Pro/Mechanica calculation and difference between them is shown is the following table.

Pressure angle (degree)	AGMA stress (N/mm^2)	Pro/Mechanica stress (N/mm^2)	Difference (%)
14.5	352.7	366.4	3.73908297
16	350.6	348.3	-0.6603503
17	348.4	339.9	-2.5007355
18	346.5	333.7	-3.8357806
19	344.4	339.8	-1.3537375
20	342.3	343.3	0.29129045
21	340.1	340.6	0.14679977
22	337.8	340.1	0.67627168
23	335.4	338.8	1.00354191
24	332.8	333.7	0.26970333
25	330.2	328.0	-1.6

Table 4.3 Von-Mises stress obtained by varying Pressure angle

4.4EffectonTooth Strength by Varying Module

The value of the AGMA calculations, Pro/Mechanica calculation and difference between them is shown is the following table.

Pitch diameter (mm)	Module (mm)	AGMA stress (N/mm^2)	Pro/Mechanica stress (N/mm^2)	Difference (%)
100	5	332.90	342.1	2.689272143
120	6	277.23	286.4	3.201815642
140	7	237.86	234.9	-1.26011069
160	8	208.01	205.2	-1.36939571
180	9	185.03	194.3	4.770972723
200	10	166.45	160.0	-4.031250
220	11	151.26	147.4	-2.61872456
240	12	138.73	128.2	-8.21372855
260	13	128.02	129.4	1.066460587
280	14	118.93	116.1	-2.43755383
300	15	110.97	110.0	-0.88181818

Table 4.4 Von-Mises stress obtained by varying Module

5. CONCLUSION

Analytical method of gear analysis uses a number of assumptions and simplificationsand it is intended to determine the maximum stress values. In this paper, numericalapproach has used for predicting the bending stresses of involute helical gear. Aparametric study is also made by varying the helix angle, face width, pressure angleand module to investigate their effect on the bending stress of helical gears.

Duringthis research work it was found that the behaviour of non-metallic gears are differentcompared to the metal gears. Due to less modulus of elasticity the tooth of gearundergoes large deformations.

It is concludedthat,
From table 4.1 that as the helix angle increases the stresses on the toothdecreases. But increase in helix angle also increases the axial thrust on the gear.
From table 4.2 it is shown that as the face width increases the stress on the toothdecreases.
From table 4.3 it is shown that as the pressure angle increases the stress on the toothdecreases.

From table 4.4 it is shown that as the module increases the stress on the tooth decreases.

The maximum difference between the AGMA stress value and Pro/Mechanica stressvalue is within 5%.

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