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DESIGN AND ANALYSIS OF HELICAL GEAR IN SLAG PORT TRANSFER CAR GEAR BOX

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ABSTRACT: *Gears are one of the most critical components in mechanical power transmission systems. They are generally used to transmit power and torque. The efficiency of power transmission through gears is very high when compared to other kind of transmission. The major factors for the failure of gear tooth are contact stress and bending stress. The present investigation is carried on helical gear which is related to slag transfer in the steel manufacturing industry. By analysing the contact stresses for existing design set of gears, the most effected stress concentrated area of any set of gears in the slag port transfer car (SPTC) gear box is discerned. The re-design of the high stress concentrated gear set took place at constant Pressure angle (20˚) and different Helix angles (14, 15, 16, 17˚, 20˚, 25˚, 30˚) with constant (45mm) Face width. The theoretical approach is based on AGMA contact stress equation, to determine the contact stresses between two mating gears. A Three-dimensional solid model is generated by CREO 3.0 parametric which is powerful and modern solid modelling software. The numerical analysis was performed in ANSYS Workbench module. The results obtained from ANSYS values are compared with theoretical values are in close agreement. The present analysis is useful in quantifying the above said parameters that helps in safe and efficient design of the helical gear in SPTC Gearbox.*

Keywords: *Gearbox, helical gear, AGMA stress, CREO, ANSYS, pressure angle, helix angle.*

1.INTRODUCTION:

Gears play a prominent role in transmitting power between the shafts. The amount of power loss in the machine influences the efficiency of the machine in process. Gears are the machine elements which is used for transmitting torque as well as angular velocity. The design and manufacture of precision gears are made from high strength of materials. Now days, Gears used in many fields such as aerospace, automotive, marine and other related areas. The modified form of spur gear is the helical gear whose teeth is parallel to the axis and has line contact as shown fig no (2). The gear teeth should have sufficient strength, so that they will not fail under static and dynamic loading during normal running conditions. The gear teeth should have clear characteristics so that their life is satisfactory, the use of space and material should economical. The alignment of the gears and deflections of the Shafts must be considered, because they affect the Performance of the gears. Helical gears are currently being used increasingly as a power transmitting gear owing to their relatively smooth. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise.

2. LITERATURE REVIEW

Praveen M Kinge [1] carried stress analysis on gear box which is used in the sugar industry. ANSYS software was used to do stress analysis on gear box.by analysing the existing gearbox, the reason of failure is found as gear teeth edges due to wear of teeth. This is majorly due to high stress concentration on teeth edges.to relieve these stresses three modification has been carried out. the three modification are first one was edges of teeth tapered by an angle to 20, second was making groove in wheel, and final one creating holes at the roots of gear teeth. By analysing this, observed that existing gear box design was not in the safe limit stress. After modification gearbox life is increased by improving safe limit stress.

M Prashant [2] reviewed papers on design and analysis of helical gear using finite element method, ANSYS and AGMA standards. The problem statement and conclusion of each paper as follows below.
Paper1 about bending analysis on helical gear tooth using finite element method. For modelling Catia software used. The analysis carried out in ANSYS software. Analysis of bending stress of gear tooth done by Lewis equation. Hence concluded as at any point of time, only one pair of teeth was in contact and takes the total load.
Paper 2 investigates characteristics of involute gears for contact and bending stress using FEM and AGMA standards. For modelling and analysis CATIA and ANSYS used. For analytical calculations Lewis equation used. For contact stress analysis Hertz equation used.

Paper 3 on design, analysis and manufacturing of helical gear using AGMA standards.by the application of different materials on gear tooth, the reduction of weight and producing high accuracy obtained.
Paper 4 done analysis on helical gear. In this the design of helical gear with different pressure angle carried. For each pressure angle, respective analytical and numerical analysis carried using AGMA and ANSYS standards. Hence compared the two stress criterion values, observed that analytical values little higher than analytical values.

Rati Jain, Pratik Goyal [3] done analysis on spur gear box using different materials. These materials were 15NiCr1Mo15 and SCM415 used.by applying contact stress on this gear box using material data found different stress results. For FEM used as analysis technique and ANSYS as software.by observing the deformation profile, type of model has preferred.

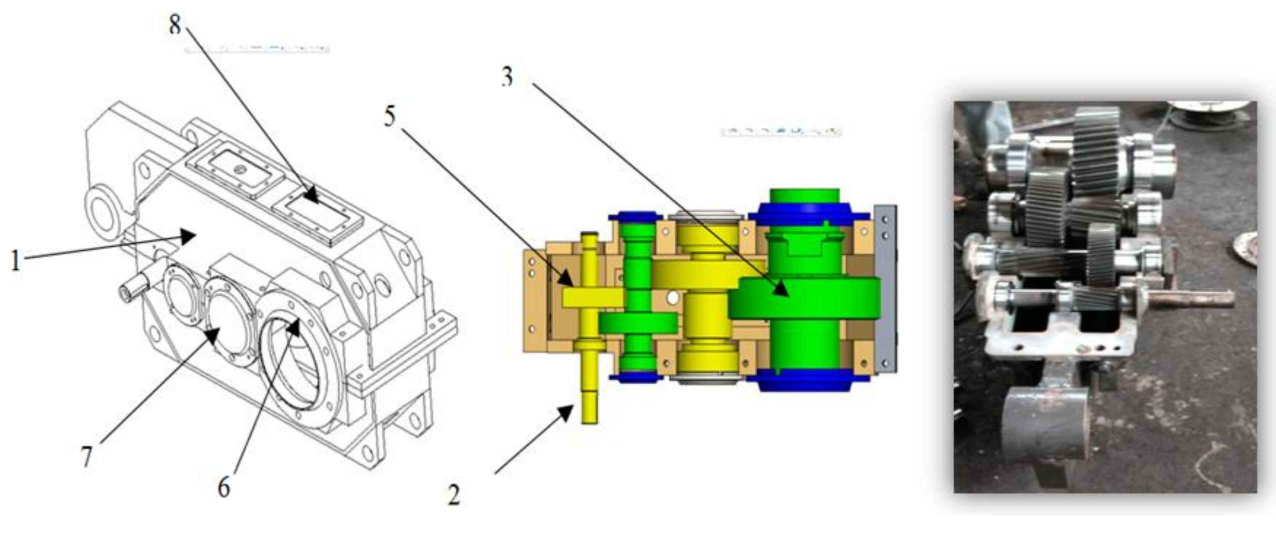
S Sai Anusha et al. [4] investigated contact stress analysis of helical gear by using AGMA and ANSYS This is carried out to make use of helical gear by analysing the contact stresses for different pressure angles, helix angle angles and face width. A 3D solid model prepared using ProE software and the numerical solution of stress is done by using ANSYS software. The results obtained is quantified with calculated AGMA stress under given parameters helps in safe and efficient design of helical gears.

3.SLAG PORT TRANSFER CAR GEARBOX

SPTC gearbox is used in steel melting shop in the steel manufacturing industry for the transfer of slag. This SPTC gearbox runs for 6 hours per day. The actual SPTC gearbox is shown in fig 1(c). The list of components of SPTC Gearbox as mentioned below fig no (1).

COMPONENTS IN SPTC GEARBOX

1. **Casing:** casing is the sub assembly which encloses mechanical components of gearbox.it imparts mechanical support for the operational components, a mechanical shield from the outside world for those internal components, and a fluid tight container to brace the lubricant that clean internal components. Basically, it is made of cast iron or aluminium alloy. For extreme hard conditions we use composite material as an alternative.
2. **Pinion shafts:** pinion shaft is used for transmitting power to the respecting mating gear.in SPTC gearbox, pinion shafts are located at four areas.one is situated at the input pinion shaft which gets power from the motor. Second and third are the intermediate shafts which are situated in between the first intermediate gear and second mediate gear. Final pinion shaft is also called output shaft which is mated with output gear as shown fig 1(b).



a) Isometric view b) top sectional view c) current working model

Fig no:1 SPTC Gearbox

3. **Helical gear:** It is crucial part of the SPTC gearbox as well as any gearbox. The helical in SPTC gear are three.one is positioned near the input pinion shaft. Second positioned at between the intermediate pinion shafts. Output gear is positioned along with the output pinion shaft.
4. **Bearing:** it is machine element which used for reduce friction between the pinion shaft and other moving parts to

get desired motion in the gearbox.

5. **Bearing block:** It is the supporting device which used for the controlling movement of the bearing.
6. **Through cover:** it is located near output shaft as shown fig 1(a).it protects from the outside dust and other damages.
7. **Blind cover:** it is also similar to the through cover but it is completely closed condition as shown in fig 1(a).
8. **Inspection cover:** it is also similar function what through and blind cover do, but it is located at the upper side of the casing as shown in the fig 1(a).

Table No 1: Technical Details of SPTC Gearbox

S no	Description	Detail unit
1	Type	Vertical ,helical with forced lubrication system
2	Input speed N1	1465 RPM
3	Output speed N2	22.93 RPM
4	Ratio	63.90:1
5	Rated power output	22KW

HELICAL GEAR AND ITS PARAMETERS

In helical gears, the helix angle varies from a range of 7 to 30 degrees. These gears transmit more power and can be utilized at larger speeds and noiseless operations. For the mating of pinion and gear the helix angle must be same. The helical parameters are calculated on the basis on ref [11] text book by G Maitra. To proceed the calculation of helical gear parameters, need normal module and no of the teeth on each gear and pinion. The calibrated values of the SPTC helical gears are tabulated in the table no (2) as below.

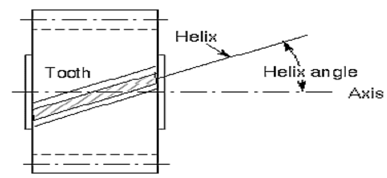


Fig no: 2 Helix angle

Table No: 2. Helical Gear Parameters Calibration of Each Gear Set in SPTC Gearbox

S No	DESCRIPTION	STAGE 1		STAGE 2		STAGE 3	
		PINION	GEAR	PINION	GEAR	PINION	GEAR
1	Normal Module	3		4		7	
2	Number Of Teeth	16	64	16	71	15	54
3	Helix Angle	16.26		14.835		14.983	
4	Pressure Angle	20		20		20	
5	Reference Diameter	50.00	200.00	66.21	293.79	108.70	391.30
6	Tip Circle Diameter	56.00	206.00	77.41	298.59	127.60	400.40
7	Root Circle Diameter	42.50	192.50	59.41	280.59	96.10	368.90
8	Clearance	0.75		1		0.35	
9	Centre Distance	125		180		250	

MATERIAL PROPERTIES OF HELICAL GEAR

The material properties of the above mentioned gears is same for all gears. The properties of the gear are tabulated in the below table no (3)

Table no 3. Material properties of helical gear in SPTC gear box

S no	Description	Unit
1	Material Type	ALLOY STEEL-15Ni5Cr4Mo1
2	Working temperature	850-1150
3	Tensile strength	1350 MPa
4	Yield strength	720 MPa
5	Density	7850 kg/m3
6	Young's modulus	210 GPa
7	Poisson ratio	0.3

4. AGMA STRESS CALCULATION

AGMA (American Gear Manufacturing Association) stress theory is a standardized calculation methodology of gear tooth design for the tooth bending stress and contact stress. From this theory, for calculation of contact stress which depends upon on the tangential force acting on the tooth. For the calculation of tangential force, we need to carry out the force analysis which is depends on the parameters like normal module, helix angle, gear teeth, speed of the gear and power acting on the pinion shaft. The sample calculations of stage 1 gear set are presented below.

FORCE ANALYSIS

1.Transverse modulus

$m_t = \frac{m_n}{\cos\beta}$

2.Pitch circle radius in mm

PCR = $\frac{(m_t \, X \, Z_G)}{2}$

Z_G = Gear teeth

3.Torque in Nm

T= $\frac{60P}{2\pi N}$

P = Power in KW

N = Speed in RPM

4.Tangential force (F_t) in N

$F_t = \frac{T}{PCR}$

SAMPLE CALCULATIONS

$m_n = 3$

$m_t = \frac{m_n}{\cos\beta} = \frac{3}{\cos 16.26} = 3.125$

$Z_G = 64$

PCR = $\frac{(m_t \, X \, Z_G)}{2} = \frac{3.125 \, X \, 64}{2} = 100$

P = 22 KW N= 366.25 RPM

T= $\frac{60P}{2\pi N} = \frac{60 \, X \, 22 \, X \, 1000}{2\pi \, X \, 366.25} = 573.6$ Nm

$F_t = \frac{T}{PCR} = \frac{573.6 \, X \, 1000}{100} = 5736$ N

All the calibrated values of the gear set stage 1,2 and 3 are presented in the table no (4) as follows.

Table no: 4. Force analysis on SPTC gear box gears sets

S No	Stage 1	Stage 2	Stage 3
Transverse modulus	3.125	4.1379	7.246
PCR	100 mm	146.895 mm	195.65 mm
Torque	573.6 Nm	2545.399 Nm	9163 Nm
Tangential force	5736 N	1732 N	46898.5 N

AGMA CONTACT STRESS FORMULAE:

$$\sigma_C = C_P \sqrt{\left(\frac{F_t}{bdI}\right)\left(\frac{\cos\beta}{0.95\,CR}\right) K_V K_O (0.93 K_M)}$$

σ_C = Contact Stress in MPa

C_P = Elastic Coeff. Factor in MPa

F_t = Tangential force in N

B = Face Width in mm

D = Outer Diameter of Gear in mm

I = Geometry Factor

β = Helix Angle in Degrees

CR= Contact Ratio

K_V = Velocity Factor

K_O = Overload Factor

K_M = Load Distribution Factor

SAMPLE CALCULATIONS

For alloy steel C_P = 191.58 N/mm2

b = 45 mm

d = 206 mm

I = 0.37

$$CR = \left(\frac{\sqrt{(r_1+a)^2 - {r_{b1}}^2} + \sqrt{(r_2+a)^2 - {r_{b2}}^2} - (r_1+r_2)sin\alpha}{\pi m_n cos\alpha} \right)$$

Where

r_1 = 25 ; r_2 = 100 ; a= 3 ; m_n=3

α = 20; r_{b1} = 21.25; r_{b2}= 96.25

$$CR = \left(\frac{\sqrt{(25+3)^2 - 21.25^2} + \sqrt{(100+3)^2 - 96.25^2} - (25+100)sin20}{\pi X\ 3X\ cos20} \right) = 1.373$$

$\sigma_C = C_P \sqrt{\left(\frac{F_t}{bdI}\right)\left(\frac{\cos\beta}{0.95\ CR}\right) K_V K_O (0.93 K_M)}$, where F_t = 5736 N ;K_V = 1.5 ;K_O = 1 ;K_M = 1.2

$$\sigma_C = 191.58 \sqrt{\left(\frac{5736}{45X\ 206X0.37}\right)\left(\frac{\cos 16.26}{0.95\ X\ 1.373}\right) 1.5\ X\ 1(0.93\ X\ 1.2)} = 260.8 \text{ N/mm2}$$

The calculated valued for the contact stresses by the AGMA for the existing design is given in the table no. 5 as follows,

Table no:5. AGMA contact stress values for existing design

Stage no	Module	Helix angle	Pressure angle	Torque Nm	Tangential force N	Contact ratio	Contact stress N/mm2
1	3	16.26	20	573.6	5736	1.372	260.8
2	4	14.835	20	2545	17328	1.375	397.415
3	7	14.983	20	9163	46898	1.75	1097.88

5.CONTACT STRESS ANALYSIS (NUMERICAL APPROACH)

MODELLING OF SPTC GEAR BOX AND ITS PARTS

The modelling of SPTC gearbox done is using CREO 3.0 parametric software. The dimensions of each component is collected using reverse engineering from the work site. Reverse engineering is done by the usage of basic instruments like Vernier calliper, steel rule, protractor, tape and other instruments. This processing of collecting data is known as sketching. This sketching data is converted into 3D objects using CREO software as shown in below figures.

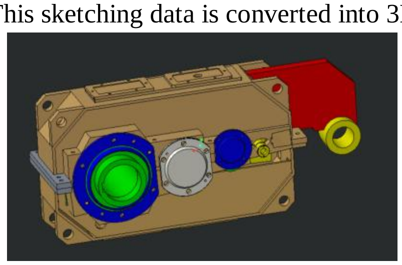


Fig no:3 assemblies of SPTC gearbox

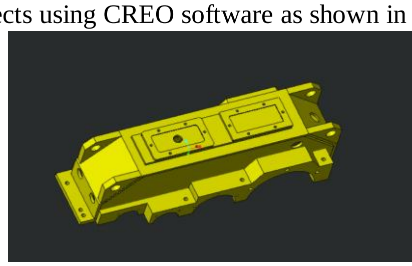


Fig no: 4 Casing subassembly

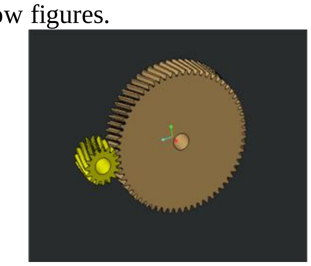


Fig no:5 stage 1 gears set

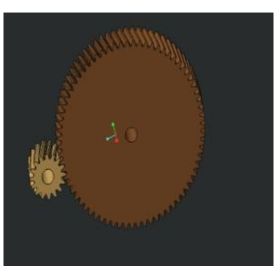


Fig no: 6 stage 2 gears set

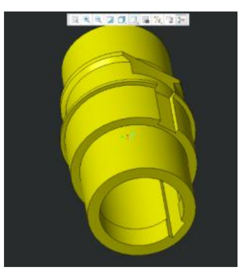


Fig no: 7 output shaft

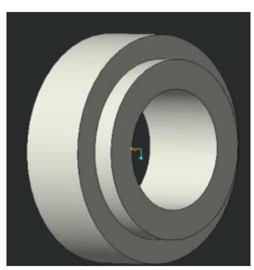


Fig no:8 Bearing

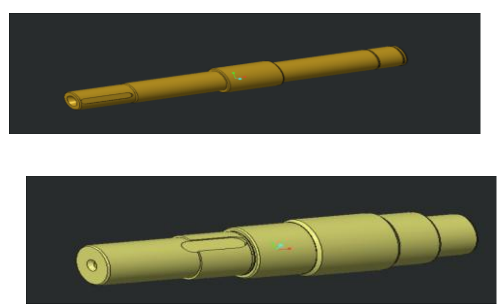


Fig no:9 input and Intermediate pinion shaft

In order to do contact analysis on the helical gear assembly, a developed model of helical gear assembly set is imported to ANSYS workbench in static structural module in the format of IGES.The material of helical gear is added as mentioned in the table no (3). To definite DOF of the helical gear set, model is meshed as shown in the figure no (10). To simplify the model from the Indefinite constraints to definite constraints like frictionless support between the pinion and gear as shown in the fig.no(11).

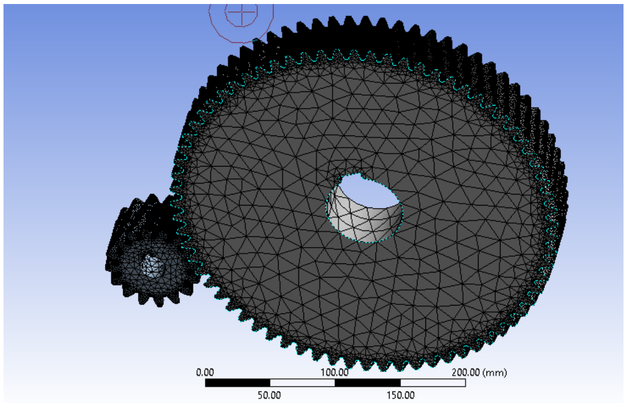


Fig no:10 meshing of gear set

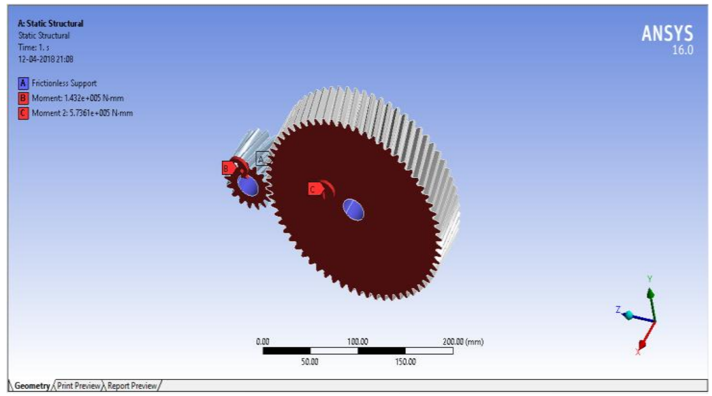


Fig no:11 Boundary conditions for stage 1

Torque applied on the both gear and pinion as moment of 573.4 Nm and 143.4 Nm respectively. For post-processing of model, Von-mises stress as maximum stress is developed on the helical gear tooth. The contact stress developed on the stage 1 gear is 258.64 N/mm2 as shown in fig.no (12). similarly, for the stage 2 and stage 3 gear set has calculated with the same boundary conditions and respective torque as mentioned in the table no (5). The results obtained from the load conditions 415.19 N/mm2 and 1068.9 N/mm2 respectively as shown in the fig.no (13) and fig no (14).

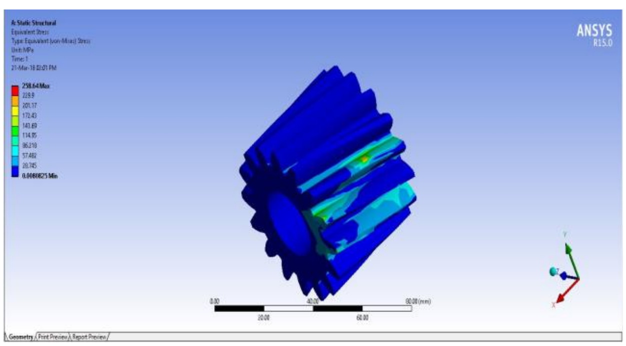


Fig no: 12. stage 1 contact stress 258.64 N/mm2

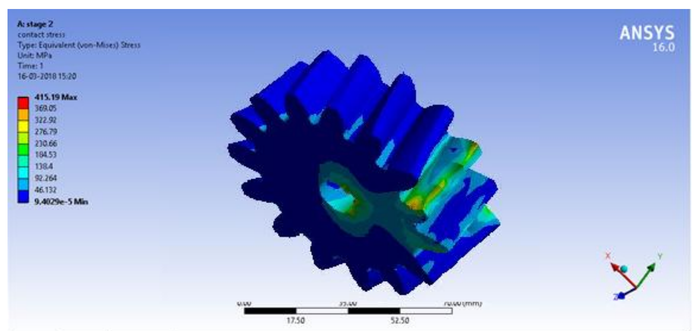


Fig no: 13 stage 2 contact stress 415.19 N/mm2

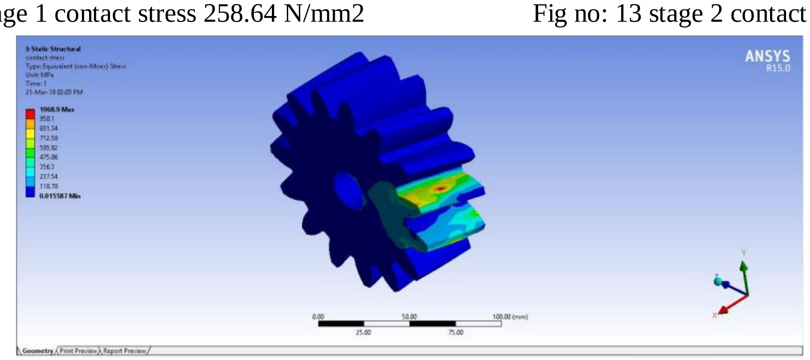


Fig no: 14 stage 3 contact stress 1068.9 N/mm2

DISCERNING ABOUT PROBLEM: The contact stress at stage 3 assembly which is 2nd pinion shaft and output gear exceed limit of yield strength 720 MPa other than cases stage 1 and Stage 2. So, by decreasing stress concentration on the stage 3 will make a safe design of the helical gear.

6.REDESIGN OF HELICAL GEAR

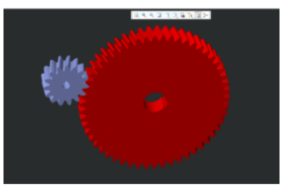
The redesign of helical gear of stage 3 gear set as mentioned above section. To reduce the stress concentration on the stage 3 by making constant module and pressure with varying helix angle. The analytical calculations are carried out similar to the above mentioned procedure and obtained values are shown in the table no (6).

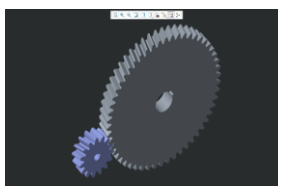
Table no:6. AGMA contact stress values for Redesign of helical Gears set stage 3

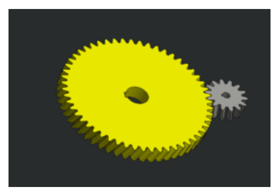
S No	Module	Helix angle	Pressure angle	Torque Nm	Tangential force N	Contact Ratio	Contact stress N/mm2
1	7	14	20	9163	47004.21	1.243	293.05
2	7	15	20	9163	46809.71	1.609	325.932
3	7	16	20	9163	46616.81	1.603	432.53
4	7	17	20	9163	46362.11	1.638	544.638
5	7	20	20	9163	45553.13	1.688	614.643
6	7	25	20	9163	43959.91	1.794	1034.772
7	7	30	20	9163	42001.35	1.936	1890.327

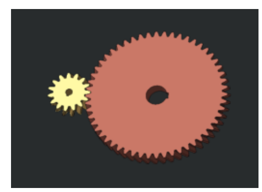
MODELLING OF HELICAL GEAR WITH DIFFERENT HELIX ANGLE:

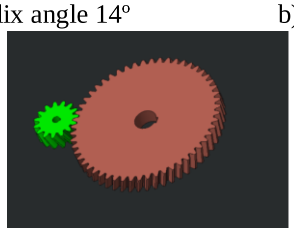
For numerical analysis, redesign of helical gear set is modelled in the CREO with various helical angle is shown in fig no (15). To know the range of helix angle for the redesign, gears were designed with various helix angle assembled in CREO. Then this modelled assemblies are imported to the ANSYS but from the helix angle below models are not meshed properly. Hence the design of helix angle above 14 degrees is suitable and applied to the respective modelled parts.

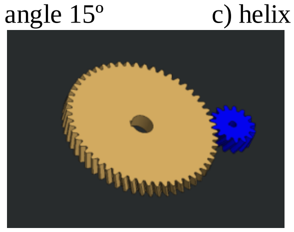

a) helix angle 14º


b) helix angle 15º


c) helix angle 16º


d) helix angle 17º


e) helix angle 20º


f) helix angle 25º

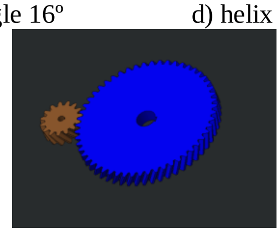

g) helix angle 30

Fig no: 15 Helical Gear with Different Helix Angle

After meshing, the post processing to present results of maximum stresses obtained on gear tooth in a gear set are represented as shown in the figures as below.at helix angle 14, the maximum contact stress is obtained is 260.8 N/mm2.for the 15,16,17,25 and 30 helix angles the contact stresses obtained are 397.415 ,427.82,535.08,600.32,1011.52 and 1862.4 N/mm2 respectively.

<u>ANSYS RESULTS OF REDESIGN</u>

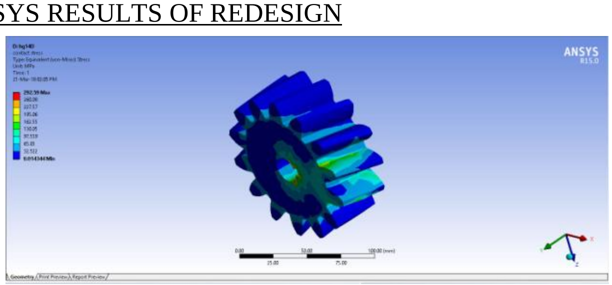


Fig no:16 contact stress 260.8 N/mm2 @ β=14º

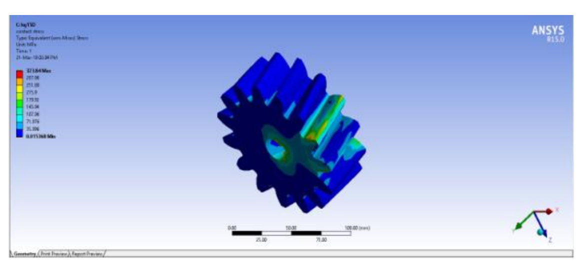


Fig no: 17 contact stress 397.415 N/mm2@ β=15º

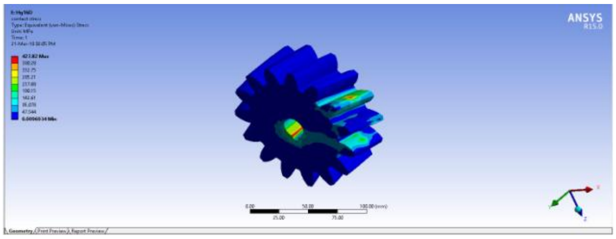


Fig no: 18 contact stress 427.82 N/mm2 @ β=16º

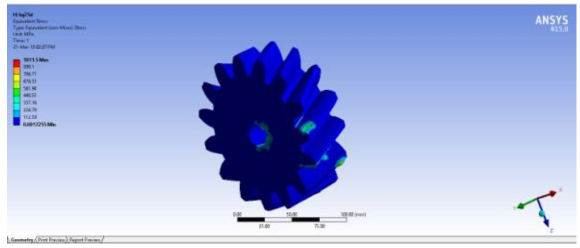


Fig no: 19 contact stress 535.05 N/mm2 @ β=17º

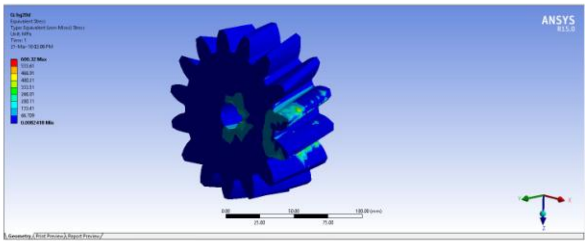


Fig no: 20 contact stress 600.32 N/mm2 @ β=20º

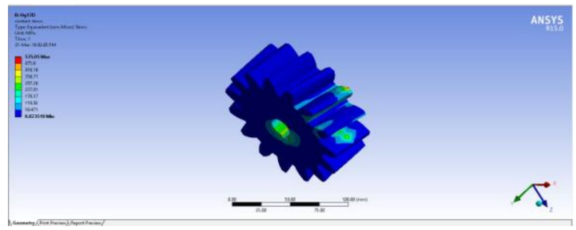


Fig no:21 contact stress 1011.52 N/mm2@ β=25º

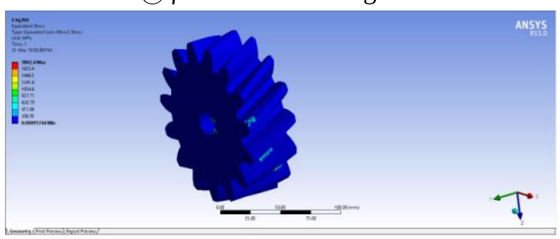


Fig no: 22 contact stress 1862.4 N/mm2@ β=30º

7.RESULTS

RESULT OF EXISTING DESIGN

The comparison of results both numerically and analytically obtained from existing design for SPTC Gearbox of each gear set of helical gear are presented in the table no (7).

Table no:7. Comparison of contact stress values for existing design

Stage No	Helix angle	AGMA contact stress N/mm2	ANSYS contact stress N/mm2
1	16.26	260.8	258.64
2	14.835	397.415	415.19
3	14.983	1097.88	1068.9

From the above results, for stage 1 and stage 2 contact stress both AGMA and ANSYS are nearer and within the yield stress limits. But for stage 3 contact stress both AGMA and ANSYS are exceeds the yield stress limit 720 MPa. So, the redesign of helical gear set done on at stage 3 and observed that, these stresses are high due to point contact between the gear teeth due to decimal angles in helix angle between the mating gear set.

RESULT OF RE DESIGN

The comparison of results both numerically and analytically obtained from re-design for SPTC Gearbox of each gear set of helical gear are presented in the table no (8). This redesign of helical gear is done with constant module because gear diameter and teeth other parameters are depending upon the module. By varying helix angle, contact stresses are obtained.

Table no:8. Comparison of contact stress values for Redesign design

S no	Helix angle	AGMA contact stress N/mm2	ANSYS contact stress N/mm2
1	14	293.050	260.80
2	15	325.932	397.42
3	16	432.53	427.82

4	17	544.638	535.05
5	20	614.643	600.32
6	25	1034.772	1011.52
7	30	1890.327	1862.40

From above results, contact stresses of helix angle between 14 to 20 degrees are within the limits and these are nearer computed values for both AGMA and ANSYS methodologies. But for the helix angle 25 and 30, the stresses were too high and exceeding the yield stress limit, so those helix angle are not safe. From helix angle 14 to 20 degrees, helix angle 14 degrees produces less stress comparatively.so it is the best angle redesign the helical gear set at stage 3 in SPTC gearbox to increase the life of the gearbox

8.CONCLUSION

Modelling and Analysis is done in CREO Parametric 3.0 and ANSYS Workbench respectively. Pressure angle, Helix angle and face widths are the important parameters in determining the state of stress during the design of gears. By observing the analysis results, the stress values obtained are less than their yield stress. It can be concluded that the design is safe under working conditions. The design at stage 1 and 2 are within in the safe under working conditions and it is seen at stage 3 that the stresses were more than the yield stress. So, the re design of helical gear at the stage 3 with constant module, face width, constant pressure angle 20 and varying helix angles (14,15,16,17,20,25, and 30) is done in CREO. The stresses for the remodelled gears were calculated according to the AGMA theory and the numerical analysis is done at different helix angles with constant face width. It is seen that the contact stress at 14 degrees is comparably less at the same module and working conditions of the SPTC gear box.

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