REDESIGN OF FACE GEAR OF SPINNING MACHINE USING FINITE ELEMENT ANALYSIS

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Abstract : The face gear is used in the spinning machine to lift the spindle mechanism. The face gear are attach with the heart cam for spinning the yarn about spindle. During the working, the tangential force is acting on face gear, which may cause to break the teeth. The tangential load which is acting on the gear is from pinion. The face gear are working similar to the bevel gear and transmitting the power. The intersection axis of both the pinion and gear at right angle to each other. The failure of the gear is due to low strength of the gear material. The beam strength of the face gear is low as compare to the tangential load that why the face gear fail. By changing the material of gear we can prevent the gear from failure. The redesign involve to change the material or to chance the dimension. But in that case, by changing the dimension likes to change the module and thickness which may affect the working of machines. Similarly to change in the setup of machine, in that case it is desirable to chance the material of face gear with high strength material than the existing one. In this paper the redesign of face gear by analytically and FEA analysis are shown. The main cause of face gear failure is due to bending stress is more than the allowable stress of that gear material. So in this paper, we increase the beam strength of gear by using various gear material.

***Keywords**:* Face Gear, FEA, Bending Stress, Spinning Machine, Gear Material ,

INTRODUCTION

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 parameters for the failure of the gears. For optimal design of gears and to reduce their failures, the stresses are becoming increasingly important. All the analytical investigations would be carried out on the basis of Lewis stress formula and contact stress are calculated by using Hertz formula of contact stress.

Face gears typically have a straight or skewed tooth line and varying tooth profile in normal cross section at different radii from major to minor diameter. These face gears are engaged with spur or helical involutes pinions at intersecting or crossed axes.

About Hi- Spin Ring Spinning Frame Machine:- Texmaco –Hi-Spin Ring Spinning Frame is designed and manufactured to run at the highest obtainable spinning speed in the industry with ring and traveler combination .However ,to fully utilize the design and manufacturing potentials of this machine careful handling by the operator is essential.

Redesign:- Much more frequently ,engineering design is employed to improve an existing design .the task may be to redesign a component in a product that is failing in service or to redesign a component so as to reduce its cost of manufacture. Often redesign is accomplished without any change in the working principal or concept of original design. For example, the shape may be change to reduce a stress concentration or a new material substitute is reduce weight or cost . When redesign is achieved by changing some of the design parameter, its is often called variant design.

What is a Face Gear:-

This is a pseudo bevel gear that is limited to 90^0 intersection axis. The face gear is a circular disc with a ring of teeth cut in its side face, hence the name face gear. Tooth elements are tapered toward its center. The mate is spur or helical pinion .

Characteristic of Face Gear:-

1. The pinion of face gear is either a spur or a helical gear.
2. Face gear are mounted on intersecting shaft that are at right angle to each other.
3. The teeth of face gear can be straight or curved.

Face gear have the following advantages:-

1. They can be cut with spur gear cutter and gear shapers.
2. The axial position of the pinion is not critical as in case of bevel gear.

3. Reduce level of noise due to the very low level of transmission error.

Fig.No.1 – face gear

REVIEW OF LITERATURE

The previous research works addressing gear analysis published by some author is as follows,

Dr. V. B. Sondur et.al, have discussed about theoretical and finite element analysis of load carrying capacity of asymmetric involute spur gears. In this paper, they have presented a method for investigating the bending stress at the critical section of "Asymmetric Involute spur Gear". The gears with different pressure angle have been modeled by using CATIA software and analysis was carried out. The results obtained by theoretical method have been verified by using ANSYS. From their work they have proved that bending stress can be minimized up to 20% by increasing pressure angle from 20° to 35°. Thus from their work it is clear that FEA can be the best technique for designing and analyzing mechanical component. [1]

James W. Dalley et.al, have explained various experimental techniques which can be used for analysis including fracture mechanics, strain measurements with electrical strain gauges and strain gauge circuits, Moiré method, theory of photoelasticity and brittle coating methods in the book 'Experimental Stress Analysis', McGraw-Hill Inc.[2]

Gitin M. Maitra, V.B. Bhandari, PSG College of Technology, Norton, P. C. Gope, M. F. Spotts et.al, have explained all the details of every type of gear including geometry, gear related parameters, force calculations, deflections, effect of heat generation, stress concentration, design criterion, load rating and efficiency of gears, friction in worm gears, material selection and strength rating of worm gears in their respective books.[4]

Prashant Patil et.al, have discussed about 3D Photoelastic and Finite Element Analysis of helical gear. They have discussed an industrial problem which uses spreading machine to spread bagasse. This spreading machine has Positive Infinite Variable (PIV) gearbox which contains helical gears. In working condition, helical pinion fails due to load coming on the teeth. It seemed that the failure was due to stress concentration and bending stresses at tooth root of gear. The calculation of maximum tensile stress at tooth root was a three dimensional problem. Thus they have analyzed the stress pattern by using 3D Photo elasticity techniques. Also they have verified obtained results with FEA. They have found out that the failure of helical gear of PIV gear box may be due to improper alignment or due to improper heat treatment process during teeth hardening.[4]

Bensely et.al In this paper failure investigation of crown wheel and pinion has been done. A fractured gear was subjected to detailed analysis using standard metallurgical techniques to identify the cause for failure. The study concludes that the failure is due to the compromise made in raw material composition by the manufacturer, which is evident by the presence of high manganese content and non-existence of nickel and molybdenum. This resulted in high core hardness (458 HV) leading to premature failure of the pinion.[5]

From the literatures survey it has been conclude that the failures of gears is due to tooth load coming on the gears is more than the beam strength of the tooth. The failure of the teeth of gears are due to bending failure. Some other paper reports , technical articles ,text book ,design data book and catalogs is also reviewed. Title and author details are mention in literature cited.

DESIGN OF SPIRAL FACE GEAR

Problem Formulation

The face gear and pinion is used in spinning machine gearbox. During operation it was observed that the face gear fail due to load coming on the teeth. The failure starts at the central thickness of tooth and continues up to the root of the tooth. The failure occurs once within operational period of about 90 days. So the industry has replace face gear which is not cost effective. The material of face gear is cast iron and pinion is forged carbon steel SAE 1045.

In order to study failure analysis of gears in industries, the present work is planned. Gears generally fail when the working stress exceeds the maximum permissible stress. The gears generally fail when tooth stress exceeds the safe limit. Therefore, it is essential to determine the strength of that gear tooth is subjected under a tangential load. Analysis of the gears strength is carried out so that tooth can be prevented from failure.

The design of the face gear for power transmission in spinning machine play an important role. This gear is moving the bobbins in upward and down ward direction. This is give data of spinning machine and required parameter to fulfill the design requirement.

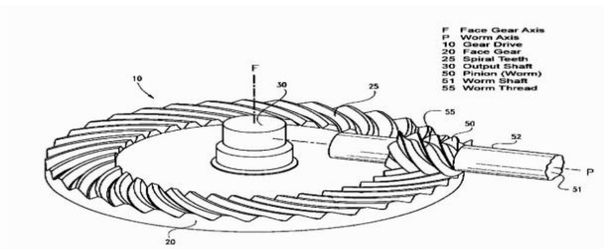

Fig.No.2.The image of Failure of face gear

Design of Gear Teeth

The process of designing gear teeth is somewhat arbitrary in that the specific application in which the gear will be used determines many of the key design parameters.This is the analytical method of finding beam strength of gear.

The following are the input parameter to design the face gear.

Motor power = P=15kW =20 HP.
No. of teeth on pinion = t_p = 10
No. of teeth on gear = t_g =43
RPM of pinion = N_p = 20
RPM of gear= N_g = 15
V.R.= N_p/N_g = 20/15 = 1.333
Module = m =5.5

Step No.1 - Pitch diameter of gear & pinion

$D_g = m \times t_g$
$= 5.5 \times 43$
$= 236.5$ mm
$D_p = m \times t_p$
$=5.5 \times 10$
$= 55$ mm

Step No.2 - Pitch angle Υ of gear and pinion

$\tan \Upsilon_p = D_p / D_g$
$= 55/236.5$
$=0.2325$
Υ_p = Pitch angle of pinion = $\tan^{-1} 0.235$
$=13.10^0$
$\tan \Upsilon_g = D_g / D_p$
$= 236.5/55$
$=4.3$
Υ_g = Pitch angle of gear = $\tan^{-1} 4.3$
$=76.9^0$

Step No.3 - Pitch line velocity

$V_p = \frac{\pi . Dp . Np}{60\times1000} = \frac{\pi\times55\times 20}{60\times1000}$ 0.05759 m/s

Step No.4 – The power of motor is = 15 kW and RPM is 1440.This power is transmitted by whole gear train at various RPM. So we are taking the ratio of power input to the power output at pinion . The pinion RPM is 20

The power required to drive the pinion is given by = $\frac{15\times20}{1440}$ = 0.2083 kW

P_d = design power

$P_d = P_p \times K_l$

Where K_l = load factor,for steady load and continuous duty.

Pd = 0.2083×1.25=0.26041 kW

Tooth load = $F_t = P_d/V_p$ = 260.41/0.05759

= 4521.90 N

Radial load =F_r

$F_r = \frac{Ft}{Cos\ \Psi}\times (\tan \phi \times \cos \Upsilon_p - \sin \Psi \times \sin \Upsilon_p)$

Where Ψ is spiral angle = 35^0

Where ϕ = pressure angle = 20^0

$= \frac{4521.9}{Cos\ 35}$ (tan 20 × cos 76.9– sin 35 × sin 76.9)

= - 2628.48 N

Axial load = F_a =

$F_a = \frac{Ft}{Cos\ \Psi}\times (\tan \phi \times \sin\Upsilon_p + \sin \Psi \times \cos \Upsilon_p)$

$= \frac{4521.9}{Cos\ 35}$ × (tan 20 × sin 76.9+ sin 35 × cos 76.9)

= 2674.53 N

Step No.5 –Beam strength, F_B, N

$F_B = S_0 \times C_V \times b\times Y \times m\times(1-b/L)$

Where ,

F_B = Beam Strength (N)

S_0 = Basic stress of material

$S_0 = S_{yt}/1.5$ (N/mm²)

C_V = Velocity factor

b = Face width of gear

Y = Lewis form factor based on virtual no. of teeth

m = Module

L = Cone distance

(1-b/L) = Bevel factor

t_f = Formative teeth

$t_{f(p)} = t_p / \cos \Upsilon_p$

= 10/ cos 13.1

=10.26

$t_{f(g)} = t_g / \cos \Upsilon_g$

= 43/cos 76.9

= 189.71

Y= Lewis form factor for involutes gear, for 20^0 full depth type of profile

$Y_p = 0.485 - (2.87/ t_{f(p)})$

= 0.485 – (2.87/10.26)

= 0.2052

$Y_g = 0.485 - (2.87/ t_{f(g)})$

= 0.485 – (2.87/189.71)

= 0.4698

Cone distance = L = $\sqrt{(\frac{Dp}{2})^2 + (\frac{Dg}{2})^2}$

$= \sqrt{(\frac{55}{2})^2 + (\frac{236.5}{2})^2}$

= 121.40 mm

The material of gear is cast iron, having basic stress = S_0=56 MPa

The material of pinion is forged carbon steel having S_0 = 245 MPa

$(S_0Y)_g$=56 ×0.4698

= 26.30 MPa

$(S_0Y)_p$=245 × 0.2052

= 50.27 MPa

$(S_0Y)_g < (S_0Y)_p$ Hence pinion is strong and gear is weak.

C_V = Velocity factor =3/(3+V_P)---commercial cut and velocity 2.5 - 5m/s.....DDB page no. 166

C_V = 3/(3+V_P)

= 3/ (3+0.05759)

= 0.9811

b = 46mm

The beam strength of gear is given by,

$F_B = S_0 \times C_V \times b\times Y \times m\times(1- (b /L))$

=56 × 0.9811 × 46 × 0.4698 × 5.5 × (1-(46/121.4))

= 4055.9 N

$F_B < F_t$ i.e. 4055.9 < 4521.9 N.

Hence, gear is fail.

The beam strength of pinion is given by

$F_{BP} = S_0 \times C_V \times b\times Y_p \times m\times(1- b /L)$

= 245 × 0.9811 × 46 × 0.2052× 5.5 × (1-(46/121.4))

F_{BP} = 7750.5 N

The following table show that the material and their basic stress for gear

Table No.1 - Material and their basic stress

Sr.No.	Material	S_0; MPa
1	Cast iron, medium grade	70
2	Cast iron, high grade	105
3	Cast iron ,0.20% carbon, untreated	140

In order to increase the beam strength the suitable material is select, which is having basic stress more than the existing material stress of gear.

[1] The material is selected cast iron medium grade

S_0 = 70MPa

For that material, calculate the beam strength

$F_B = S_0 \times C_V \times b \times Y \times m \times (1- b/ L)$

$=70 \times 0.9811 \times 46 \times 0.4698 \times 5.5 \times (1-(46/121.4))$

$= 5069.8$ N.

Here $F_B > F_t$ i.e. the material selected is suitable for the given application.

[2] The second choice of material is Cast iron high grade

S_0 = 105MPa

For that material, calculate the beam strength

$F_B = S_0 \times C_V \times b \times Y \times m \times (1- b/ L)$

$=105 \times 0.9811 \times 46 \times 0.4698 \times 5.5 \times (1-(46/121.4))$

$= 7604.8$ N.

Here $F_B > F_t$ i.e. the material selected is suitable for the given application

[3] The third choice of material is Cast steel,0.20% carbon , untreated

S_0 = 140MPa

For that material, calculate the beam strength

$F_B = S_0 \times C_V \times b \times Y \times m \times (1- b/ L)$

$=140 \times 0.9811 \times 46 \times 0.4698 \times 5.5 \times (1-(46/121.4))$

$= 10139.7$ N.

Here $F_B > F_t$ i.e. the material selected is suitable for the given application.

We have three choice but the beam strength for Cast steel, 0.20 % carbon ,untreated is more than the pinion. So it may fail the pinion.

Table No. 2 – Beam strength and Tooth load

Sr.No.	Material	S_0; MPa	Beam Strength F_B of gear	F_t
1	Cast Iron	56	4055.9	4521.90
2	Cast iron, medium grade	70	5069. 8	4521.90
3	Cast iron, high grade	105	7604.8	4521.90
4	Cast iron ,0.20% carbon, untreated	140	10139.7	4521.90

We have two choice from above material i.e., Cast iron ,medium grade and Cast iron high grade. The strength of cast iron high grade material is more than cast iron medium grade, so material selected is cast iron high grade.
.

Step no.6- Dynamic load calculation, F_d, by using Buckingham's equation

$$F_d = F_t + \frac{21Vp(Ceb+Ft)}{21Vp+\sqrt{Ceb+Ft}}$$

Where, F_d = Dynamic load (N)

F_t = Tooth load (N)

V_P=Pitch line velocity (m/s)

C = Deformation Factor

e = Sum of the error between two meshing teeth (mm)

$$C = \text{Deformation Factor} = \frac{a}{(\frac{1}{Ep}+\frac{1}{Eg})}$$

Where,

a = Constant depend on the form of teeth

a = 0.111 for 20^0 full depth

E_p = 203 GPa

E_g = 77 GPa

$$C = \frac{0.111}{(\frac{1}{203\times10^3}+\frac{1}{77\times10^3})} = 6196.57 \text{ N/mm}^2$$

e = 0.06 ---- from the DDB on page 165, the graph show the module verses probable error.

$$F_d = F_t + \frac{21Vp(Ceb+Ft)}{21Vp+\sqrt{Ceb+Ft}} = 4521.9 + \frac{21\times0.05759(6196.5\times0.06\times46+4521.9)}{21\times0.05759+\sqrt{6196.5\times0.06\times46+4521.9}} = 4698.29 \text{ N}$$

Step no.7 - Limiting wearing strength, F_w, N

$$F_w = \frac{KbDpQ}{\cos Yp}$$

Where,

K = Load stress factor

b = Face width

D_p = Pitch diameter of pinion

Q = Ratio factor

$$Q=\frac{2\ tf(g)}{tf(p)+tf(g)}$$ ---------- DDB page 180

$$=\frac{2\ \times189.71}{10.26+189.71} =1.8973$$

$$K = \sigma^2_{es} \times \sin \phi \times (\frac{1}{EP}+\frac{1}{EG})$$

Where, σ_{es} = Surface endurance limit of gear pair in N/mm^2 = 630 N/mm^2
ϕ = Pressure angle = 20^0
E_P = Modulus of elasticity of Pinion = 203 GPa
E_g = Modulus of elasticity of Gear= 77 GPa

Therefore K = 1.7369 N/mm^2

$$F_w = \frac{1.7369 \times 46 \times 55 \times 1.8973}{\cos(13.1)} = 8560.18 \text{ N}$$

Since $F_w > F_d$ Hence it is safe for continuous duty.

Step No.8 - Bending stress calculation , σ_b

$$\sigma_b = M \times y \ / \ I$$

Where,
σ_b = Bending stress in N/mm^2
M =Maximum bending moment, Ft × h
h = Height of tooth
y = Half of the thickness of tooth (t) at radical section = t / 2
I = moment of inertia about the centre line of tooth = bt^3 /12

$$\sigma_b = \frac{Ft \times h}{bt3\ /12} \times t / 2 = 6\ Ft \times h \ / \ bt^2$$

$$\sigma_b = 6 \times 4521.9 \times 11 / (46 \times 8.6^2)$$

$$= 87.72 \ N/mm^2$$

$\sigma_b > S_0$ i.e., 87.72 N/mm^2 > 56 N/mm^2

Hence the gear tooth are fail due to bending stress develop is more than the allowable stress of gear material i.e., grey cast iron.

Similarly the $\sigma_b > S_0$ i.e., 87.72 N/mm^2 > 70 N/mm^2 for cast iron medium grade, hence gear fail due to bending stress .

By changing the the material i.e., high grade cast iron having allowable stress 105 N/mm^2,which is more than the stress develop in gear tooth.
$\sigma_b < S_0$ i.e., 87.72 N/mm^2 < 105 N/mm^2. Hence the teeth have more strength than bending strength.

Step No.9 – Hertz Contact stress calculation , σ_c

$$\sigma_c = C_p \sqrt{\frac{Ft \times Ko \times Kv \times Km \times Cf}{Dg \times b \times J}}$$

Where,
σ_c =Hertz contact stress in N/mm^2
F_t = Tooth load = 4521.9 N
K_o = Overload factor = 1.25
K_v = Dynamic Factor = 1.25
K_m = Load distribution factor = 1.8
C_f = Surface condition factor = 1
D_g = Pitch diameter of gear = 236.5 mm
b = face width = 46 mm
J = Geometric factor = 0.35
C_p = Elastic constant

$$C_p = \sqrt{\frac{3 \times EG}{4\pi(1-\mu 2)}}$$

Where ,
E_g = Modulus of elasticity of Gear= 77 GPa
μ = Poisson's ratio = 0.271 (Cast Iron)
Therefore,
C_p = 140.85 N/mm^2

$$\sigma_c = C_p \sqrt{\frac{4521.9 \times 1.25 \times 1.25 \times 1.8 \times 1}{236.5 \times 46 \times 0.35}}$$

$$= 140.85 \times 1.82$$

$$= 256.35 \ N/mm^2$$

Standerd proportion for gear
1) Addendum = 1m = 1× 5.5 = 5.5 mm
2) Dedendum = 1.2 m = 1.2 × 5.5= 6.66 mm
3) Clearance = 0.2m = 0.2 × 5.5 =1.1 mm
4) Working depth = 2m = 2 × 5.5 = 11 mm
5) Thickness = 1.57 m = 1.57 × 5.5 =8.63 mm
6) Outside diameter = Pitch diameter + 2 a × cos ϒg = 236.5 + 2 × 5.5 × cos 76.9 = 239 mm
7) Inside diameter = 239 – (2 × 46) = 147 mm

The analysis in ANSYS Software

Table3- Gear Specification

Specification	Gear
Number of teeth	43
Module	5.5
Pitch diameter	236.5
Base circle diameter	239
Pressure angle	20^0
Tooth thickness	8.63
Addendum	5.5
Dedendum	6.66

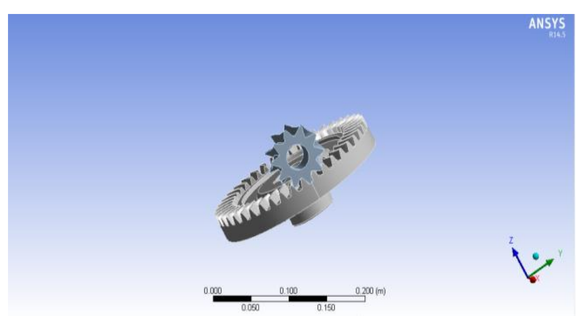

Fig.No. 3 – Solid modeling

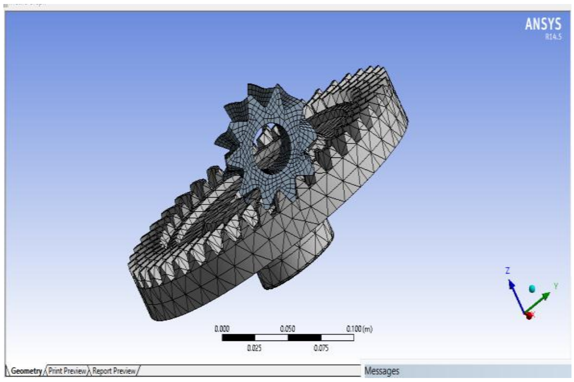

Fig. No.4 - Meshing

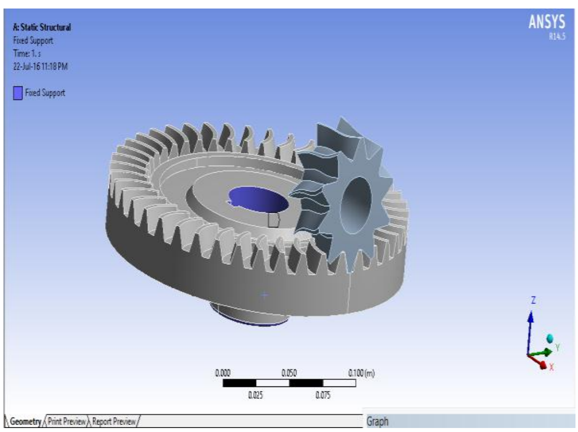

Fig. No.5 - Fixed Support

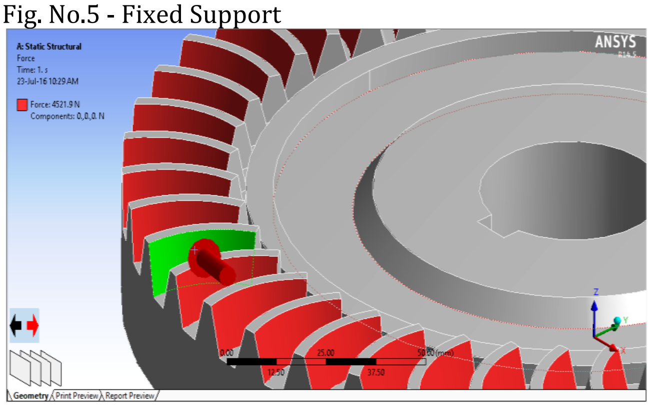

Fig.No.6 – Tangential load applied on tooth

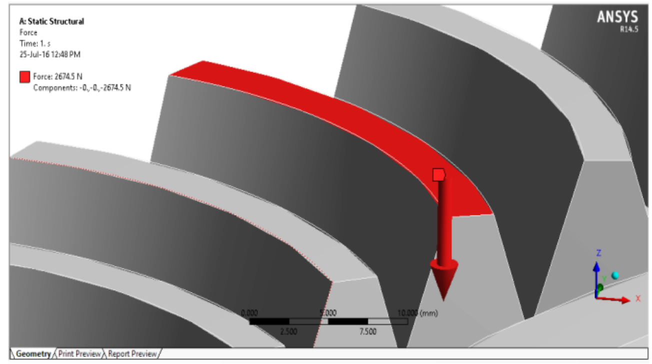

Fig.No.7- Axial load applied on tooth

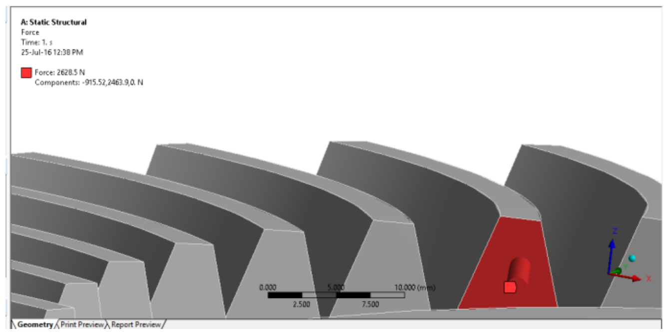

Fig.No.8. –Radial load applied on tooth

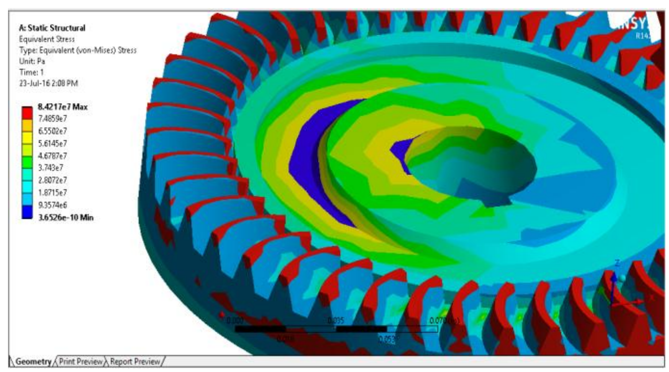

Fig No.9-Bending stress

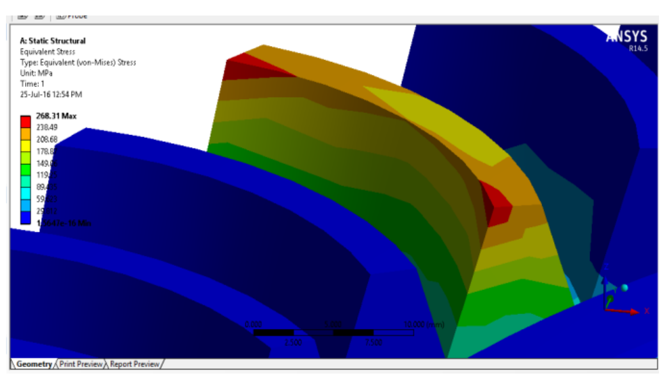

Fig.No.10- Contact Stress

RESULT AND DISCUSSION

Result:-In this project work the result is of failure of face gear is given. The analytical and ANSYS result are compare.

Table No.4

Sr.No.	Analytically Calculated Stress in N/mm^2	Von-Mises Stress in Pa	% Error
Bending Stress	87.72 N/mm^2	89.46 N/mm^2 (8.9466 e^7) Pa	1.94 %
Contact stress	256.35 N/mm^2	262.9 N/mm^2 (26.297 e^7 pa)	2.49 %

Discussion:- This project work addresses live problem of failure of face gear used in Texmaco Hi-Spin Spinning Machine at Indira Sahakari Sutgirni ,Sewagram. As per the visual and load consideration the type of face gear failure is bending fatigue. This types of failure caused by repeated loading, starts as a crack that grows until the fracture. This fatigue failure occurs in the tooth.

CONCLUTION

In this project work ,the failure of face gear is due to the tooth load . The another load i.e. axial load and radial load is less as compare to tooth load. Due to this tooth load, the stress develop in tooth is more than the basic stress of gear, that why gear fail. Due to gear fail, the machine is in break down stage, So the production is hamper. To avoid breakdown it is very important to chance the material of gear. Here in this project an analytical calculation and analysis in ANSYS of face gear is done. This show that by changing the material with new material is good for desirable application in spinning machine.

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