Effect of Change of Spur Gear Tooth Parameter On Contact stress

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Abstract: *In this paper contact stress analysis of spur gear is done by using theoretical method using Herrzian contact theory and finite element analysis by using ANSYS Workbench 14.0. Parameter like face width is varied and trochoidal root fillet radius is replaced by circular root fillet radius. Thus proposed circular root fillet radius withstands higher contact and bending stress. Also by increasing the face width contact stress goes on decreasing.*

Keywords: Spur Gear, Face Width, Trochoidal Root Fillet, Circular Root Fillet, Contact Stress.

1. INTRODUCTION

Gear transmission systems play an important role in many industries. The knowledge and understanding of gear behaviour in mesh such as stress distribution, work condition and distortion is critical to monitoring and controlling the gear transmission system.

A pair of 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 due to contact fatigue. These types of failures can be minimized by careful analysis of the problem during the design stage and creating proper tooth surface profile, optimal teeth parameters with proper manufacturing methods.

One of the primary causes of gear tooth failure is the presence of large tensile stresses in the root fillet of loaded gear tooth. These stresses reduce the overall gear life and can result in catastrophic tooth failure under peak load conditions. Many attempts have been made by earlier investigators to relate the tooth failure and the tensile stresses observed in loaded gear, and found that maximum principle stress is the key factor, which governs the fatigue life of the spur gear. A small reduction in maximum principle stresses leads to increase in the fatigue life of the gears considerable. Therefore it is important to find out the method of reducing maximum principle stress in the gear there by increasing the life of gears.

Most of them are given solutions to the use of material with improved strength, hardening the surfaces selectively with heat treatment and carburization, and shot peening to improve the surface finish. Many efforts such as altering the pressure angle, using the asymmetric teeth, introducing stress relief feature and using the gear with high contact ratio have been made to improve the durability and strength of the gear.

2. HERTZIAN CONTACT THEORY

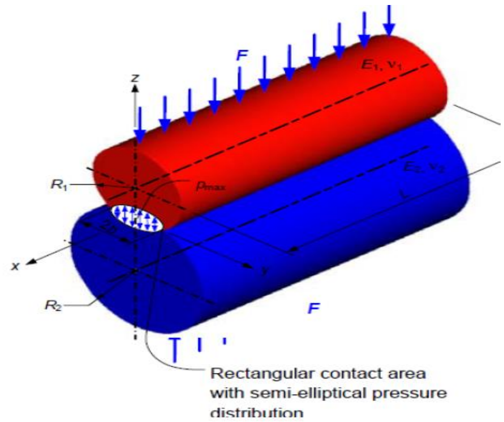


Figure-1: Contact between two cylinders

Fig. shows the contact between two cylinders with radius R1 and R2 with parallel axes. In contact between two cylinders, the force is linearly proportional to the indentation depth. The width of contact zone is 2b.

The half width b of the contact area of the two parallel cylinders is given as

$$b = \sqrt{\frac{4.F\left[\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right]}{\pi * L\left[\frac{1}{R_1}+\frac{1}{R_2}\right]}}$$

Where,

b = half width of deformation

$\mu_1{}^2, \mu_2{}^2 =$ poison's ratio of two cylinder

$E_1, E_2 =$ Modulus of elasticity of two cylinder

L = axial length of cylinders

$R_1, R_2 =$ radius of two cylinders

The maximum contact stress along the centre line of the contact area is

$$\sigma_c = \frac{2.F}{\pi.b.L}$$

Where,

$$F = \frac{f_t}{cos\emptyset}$$

At L = 9.5 mm F = 337.69 N

$$b = \sqrt{\frac{4*337.69\left[\frac{1-0.27^2}{200000}+\frac{1-0.27^2}{200000}\right]}{\pi*9.5*\left[\frac{1}{7.5}+\frac{1}{38.5}\right]}}$$

b = 0.0513

$$\sigma_c = \frac{2*337.69}{\pi*0.0513*9.5}$$

σ_c = 441.12 MPa

Table -1 Calculated results (Theoretical)

Sr.NO.	Face Width (mm)	Hertzian contact stress (MPa)
1.	9.5	441.12
2.	10	429.96
3.	10.5	419.55
4.	11	409.72
5.	11.5	401.16
6.	12	392.01

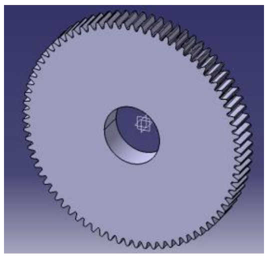
3. MODELING OF GEAR

Gear is modelled using CATIA V5R16. Specifications of gear given in following table :

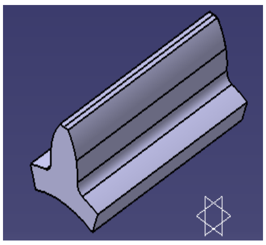
Table- 2. Specification of gear

Input parameters	Value
No. of teeth on gear	15
No. Of teeth on pinion	77
Module (m)	1 mm
Pressure angle (Φ)	20
Helix angle (ψ)	0
P.C.D. of gear	15 mm
P.C.D. of pinion	77 mm
Thickness	9.5 mm
Tooth root fillet	Trochoidal and Circular

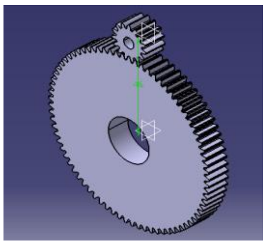
Modeling of 15 no. of teeth (full gear)



Modeling of 77 no. of teeth (full gear)



Trochoidal root fillet (single tooth)



Assembly of gears

4. FINITE ELEMENT ANALYSIS

FEM is the easy technique as compared to the theoretical methods to find out stress developed in a component or body. Therefore Finite Element Method is widely used for the stress analysis of gears. In this dissertation work finite element analysis is carried out in ANSYS Workbench 14.0 to determine the bending and contact stress for 20MnCr5 steel gears.

A assembled gear model is considered for finite element analysis. Gear material strength is a major consideration for the operational loading and environment. In this work 20M_nCR5 (Case Hardened Steel) is used for analysis. ANSYS version 14.0 software is used for analysis. The gear tooth is meshed in 3-D solid 186 with fine mesh. Solid 186 is a structural 3D 20 node solid element. It has 3 degree of freedom in X, Y, Z direction (Translation). It supports plasticity, creep, stress and large deflection.

Force components for 15 teeth :

Power (P) = 375 watt

Speed (N) = 1500 rpm

Torque (T) = 2380 N-mm

Table- 3. Material properties

Parameter	Value
Density	7700 Kg/m3
Young's modulus	200000 MPa
Poisons ratio	0.27

Meshing of model is shown in following fig.

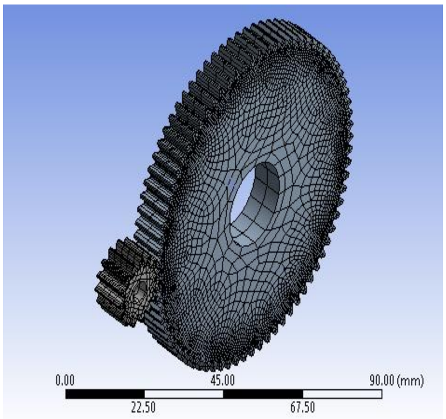


Figure- 2. Meshing of model

Boundary condition of model is shown in following fig.

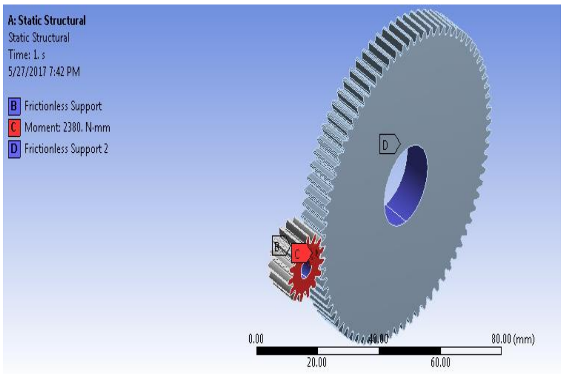


Figure- 3. Boundary condition and moment

5. RESULTS AND DISCUSSION

Contact stress analysis was carried out by varying face width and trochoidal root fillet radius is replaced by circular root fillet radius. Contact stress values are presented in table.

The result shows that contact stress decreases with increase of face width. There is reduction in contact stress value when trochoidal root fillet radius is replaced by circular root fillet radius.

Contact stress analysis by varying the face width is shown in figure.

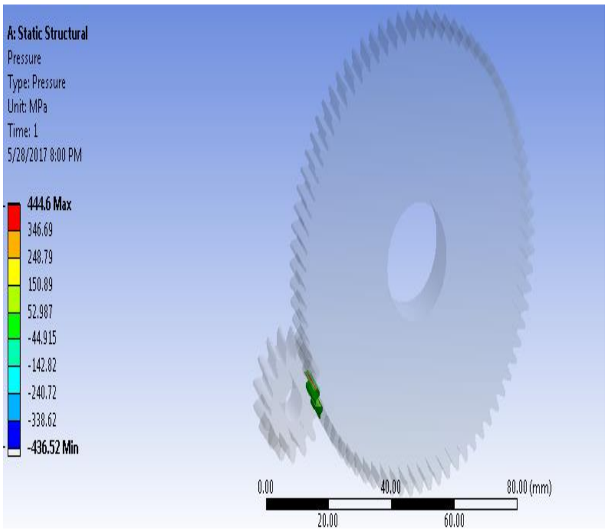


Figure- 4. Contact stress at 9.5 mm face width

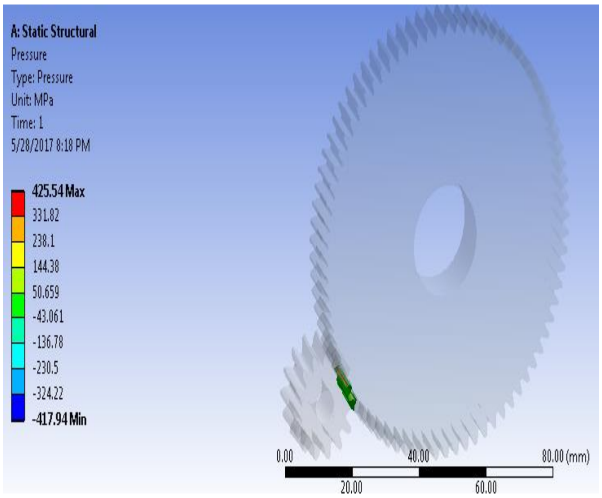


Figure-5 Contact stress at 10 mm face width

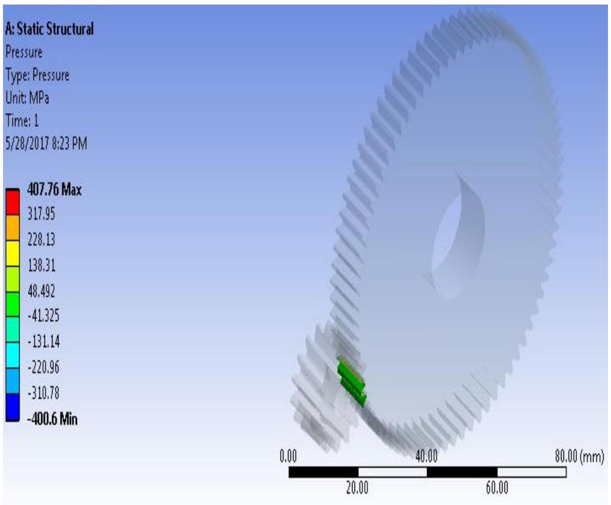


Figure- 6. Contact stress at 10.5 mm face width

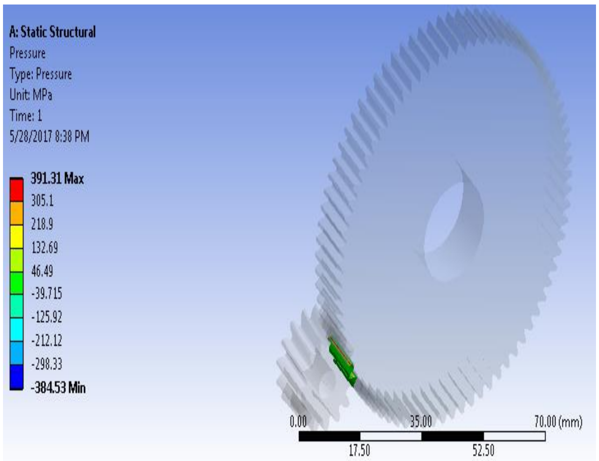


Figure -7. Contact stress at 11 mm face width

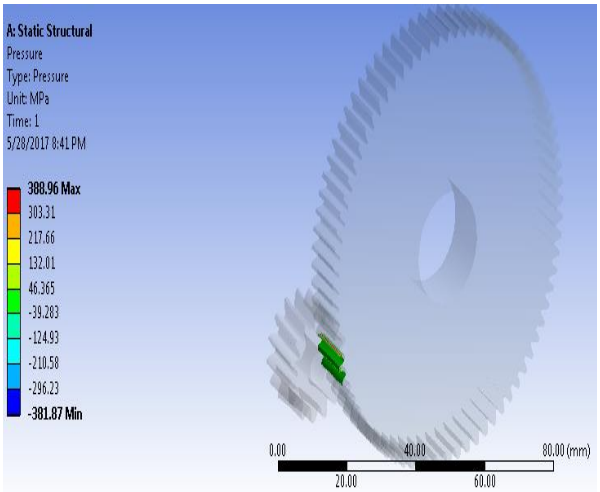


Figure- 8. Contact stress at 11.5 mm face width

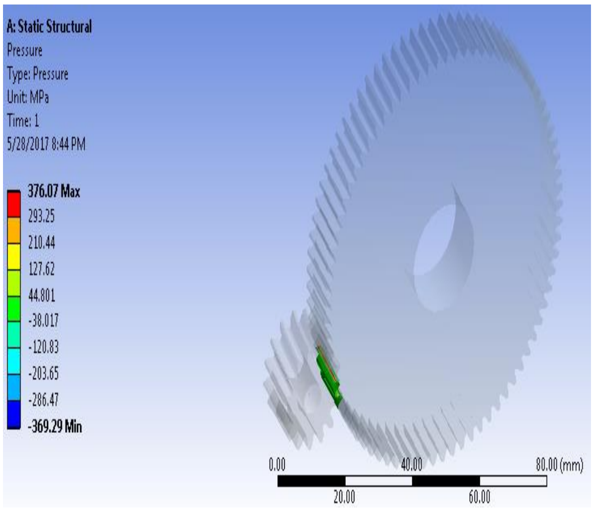


Figure- 9. Contact stress at 12 mm face width

5.1 COMPARISON OF CONTACT STRESS:

For trochoidal root fillet:

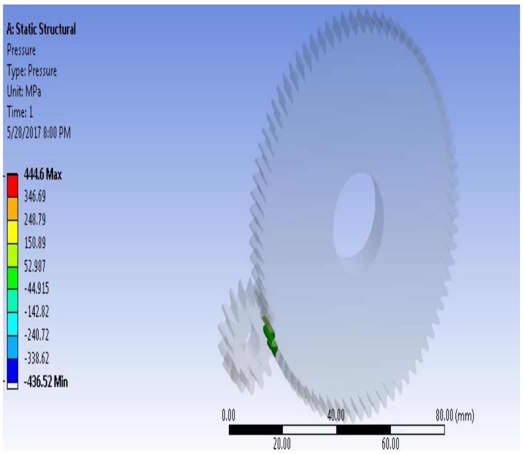


Figure -10 Contact stress (Trochoidal)

For Circular root fillet radius :

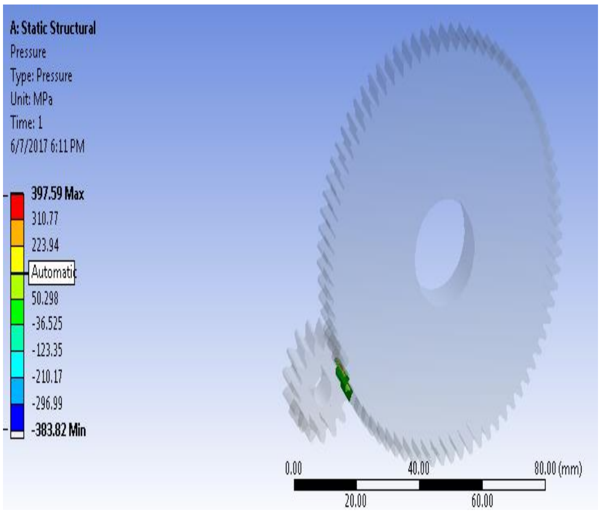


Figure -11 Contact stress (Circular)

Table-4: Contact stress results (Face width)

Sr. No.	Face width (mm)	Contact stress by ANSYS (MPa)
1.	9.5	444.6
2.	10	425.54
3.	10.5	407.76
4.	11	391.31
5.	11.5	388.96
6.	12	376.07

Table-5: Contact stress results (Root fillet)

Contact stress (MPa)		% Reduction
Trochoidal	Circular	10.57
444.6	397.59	

6. CONCLUSION

The effect of proposed circular fillet design on the contact stress induced in spur gear was investigated in comparison with standard trochoidal circular root fillet design. Also the effect of varying face width was studied. From the results it concludes that contact stress decreases with increase of face width. There is reduction in contact stress value for circular root fillet design compared to contact stress value in trochoidal root fillet design.

From the results it found that 10.57 % reduction in contact stress when circular root fillet design is used instead of standard trochoidal root fillet design for existing gear.

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