Surface Wear Analysis of Spur Gear

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Abstract

Aim of this paper to investigate the wear of tooth surface of spur gear at wear load capacity and overcome the shortcomings of existing wear analysis of spur gear flank. This paper describes the impact/affect of surface wear on bending stress of spur gear flank. According to the standard gear geometric relationship a precise assembly model of the gear and pinion is established using CATIA and wear load capacity has been calculated theoretically. Then transient structural analysis has been carried out during its working process to find out the surface wear at wear load capacity using ANSYS.

Key Words: *Spur Gear, Surface Wear, Theoretical Wear Load Capacity, Archard's Wear, Bending Stress, ANSYS, Transient Structural*

1.INTRODUCTION

Generally, transfer of power plays an important role everywhere in small to large scale industries like a automobile, aerospace, and others machine equipments. Gear technology is used widely in all kinds' applications, where power needs to be transmitted from source to destination without major loss by engaging teeth. But during power transmission gear wear causes one of the major failures of gear tooth. Moreover, efficiency and strength decreases. Also creates severe noises and vibrations. Practically each spur gear tooth acts as a cantilever beam. Surface wear and bending are considered as the major failures of gear tooth. To avoid bending stress, the area of cross-section should be more and to avoid wear the temperature of contact region should be in permissible limit by keeping speed, pressure as parameters uniform.

1.1 Material used and Its Properties

Here, the gear and pinion are made of **Cast Iron,** as cast iron has the capability to resist wear. Corresponding mechanical properties of Cast Iron are chosen as,

Material Properties		
Properties	**Symbol**	**Value**
Young's Modulus	E	164 Gpa
Poisons ratio	γ	0.291
Ultimate Strength	σ_u	4.826e+8Pa
Yield Strength	σ_y	3.654e+8 Pa
BHN = Hardness Number	H	170
Allowable Endurance Limit Stress	σ_{es}	630 Mpa

2. Dimensions of Gear Pinion Assembly

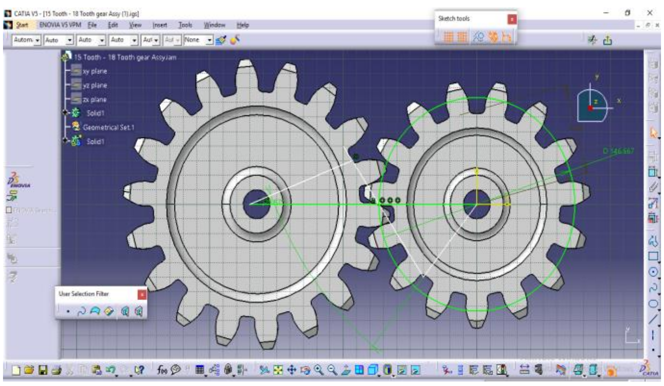

Figure -1

Here 3D model of a gear and pinion assembly was created by CATIA having dimension mentioned below.

2.1 Gear:

Addendum Circle radius = 100 mm

Base circle radius = 77.5 mm ~ 77 mm

2.2 Pinion:

Addendum Circle radius = 85 mm

Base circle radius = 62 mm

2.3 Tooth:

Face Width = b = 30mm

Tooth Thickness = t = 16.31mm

Depth = 23

Z_g = 18

Z_p = 15

α = Pressure Angle

3. Calculation

Input Torque = T = 30 N-m

Wear Load capacity = $F_w = K_s.Q_g.D_{pp}.b$

Where,

$$Q_g = \left[\frac{\sigma_{es}{}^2.\sin\alpha}{1.4}\right]\left[\frac{1}{E_1} + \frac{1}{E_2}\right]$$

Here, $E_1 = E_2$

$$K_s = \frac{2Z_g}{(Z_g + Z_p)}$$

D_{pp} = Pitch Circle diameter of Pinion

b = Face width of pinion tooth

By putting the corresponding value in above formula, we found,

Wear Load Capacity = F_w = 4137.61 N

4. Analysis and Results:

To analyze what actually happens during rotation of gears, the 3D model of gear and pinion assembly was imported to ANSYS workbench to perform transient structural analysis. Then theoretical wear load capacity calculated above was applied to the contact region.

Here it is noted that wear load capacity is the amount of load applied without premature wear.

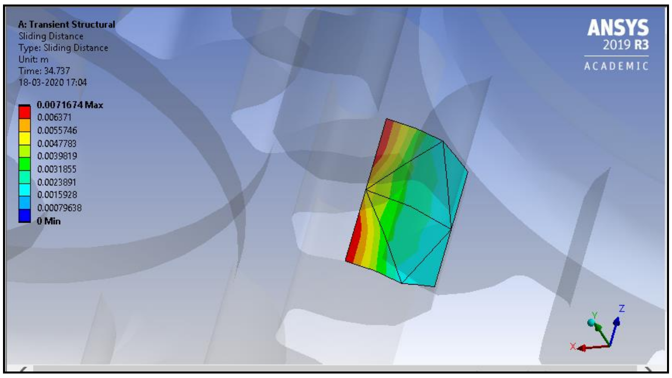

Figure -2

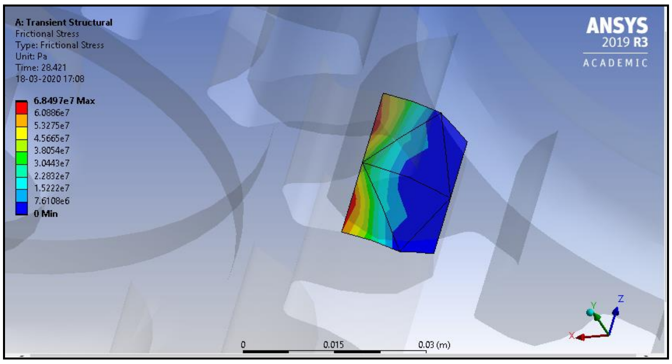

Figure -3

It was found that during the end of engagement followed by starting of disengagement, the contact surface experience sliding also variation of pressure in that region. The sliding speed and contact pressure is maximum at the end of contact, i.e.: near the flank of tooth. The sliding distance and variation of pressure is shown in fig -2 & 3, respectively. Putting the corresponding values of sliding distance in Archard's wear equation the wear volume at different points near tooth flank along the direction of face width is shown in fig 4.

Archard's Wear equation is given by:

$$Q = \frac{K.W.L}{H}$$, Where,

Q = Total volume of Wear debris produced

K = Wear constant

W = Total Normal Load

L = Sliding distance

H = Hardness

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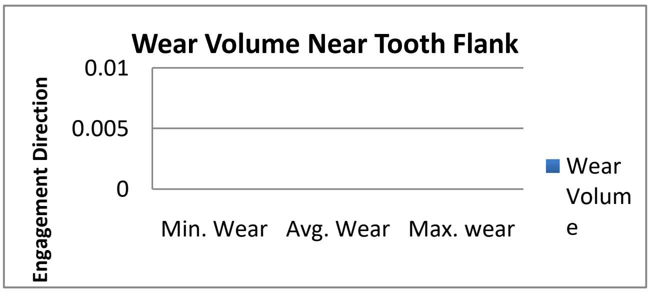

Chart -1: Wear Volume in mm^3

Also observed that in the same region near flank of tooth stickiness is high, this tries to remove material from contact region. At this region due to high temp and high friction stress as well, there is a chance of removal of the material.

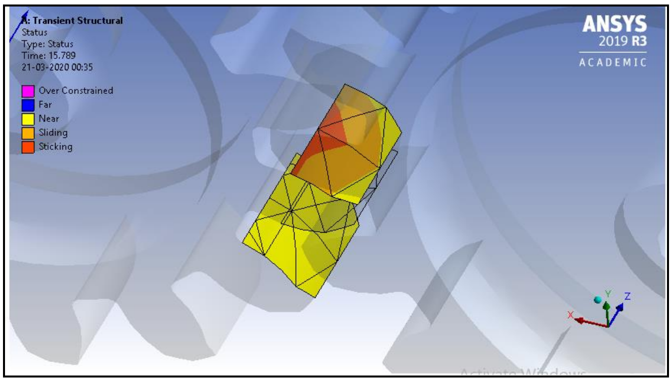

Figure -4

Now the above observation can be related to bending stress of a gear tooth. We know that, Bending Stress is given by,

$$\sigma_b = \frac{6F_t h}{bt^2}$$

As the gears are designed in such a manner that particular tooth of gear does not come in contact with the same tooth of pinion for next no. of cycles. Hence different tooth surface comes in contact. Gradually thickness of tooth is decreased. So from the bending stress relation, we can easily observe that with the reduction of gear thickness bending stress will be increased.

3. CONCLUSIONS

- Surface wear is considerably found at tooth flank near root, which increases the value of bending stress relatively high at wear load capacity.
- Wear at flank can result higher stress concentration factor due to irregularity.

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