Analysis of a Fatigue Crack in Spur Gear

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Abstract- *Gears are commonly used for transmitting power and motion. An effective gear design balances strength, durability, reliability, size, weight and cost. The engineering structures may fail due to crack, which depends on the design of gear, material used and also on operating conditions in which it operates. This failure of gear can be avoided by analyzing and understanding the manner in which crack originates and propagates. It is necessary to develop design guidelines to prevent failure modes considering gear tooth fracture, by studying the crack propagation path in a spur gear. In variety of gear tooth geometry the crack propagation paths are predicted at various crack initiation location. The life cycle of spur gear can be determined by analyzing the fatigue crack behavior in spur gear tooth. Designing of simple and less costly spur gear test rig .The three-dimensional model of spur gear is modelled in SOLIDWORKS 2016 and for analyzing it is imported into ANSYS 16.0software.*

Keywords- Spur gear, Crack, Tooth bending,fatigue, Stress intensity factor

I. INTRODUCTION

A gear is a rotatingmachine part having cut teeth which mesh with another toothed part to transmit the motion and to transmit the power. Spur gear, Helical gear, Double helical gear, bevel gear, spiral bevel gear, worm gear, rack and pinion are different types of gears. Gears are commonly used mechanical components in power transmission and are frequently responsible for gearbox failures. Gears can fail in many different ways. Each type of failure leaves characteristic clues on gear teeth, and detailed examination often yields enough information to establish the cause of failure. wear, fracture, plastic flow, fatigue failure.Fatigue is the most common failure in gearing.Tooth bending fatigue and surface contact fatigue are two of the most common modes of fatigue failure in gears. Toothbending fatigue results in progressive damage to gear teeth and ultimately leads to complete failure of the gear. Bending fatigue failures occur when the stress at the root of the gear tooth exceeds the capability of the gear material. This can be due to excessive loads, incorrect heat treatment, inclusions in the steel, or a notch in the root of the tooth. The initial crack is located at the point of the largest stresses in a gear tooth root. The complete bending fatigue failure of mechanical elements is mainly divided into two parts namely crack initiation and crack propagation period.

II. PROBLEM DEFINATION

The purpose of this is to analyse the crack behaviour in spur gear tooth.

III. LITERATURE REVIEW

[1]studied the crack propagation in the tooth foot of a spur gear by using Linear Elastic Fracture Mechanics (LEFM) and the Finite Element Method (FEM). The tooth foot crack propagation is a function of Stress Intensity Factors(SIF) that play a very crucial role in the life span of the gear. The study estimates the stress intensity factors and monitors their variations on the tooth foot according to crack depth, crack propagation angle, and the crack position. An appropriate methodology for predicting the crack propagation path is applied by considering gear tooth behaviour in bending fatigue.

[2] studied analytically and experimentally the effect of gear rim thickness on crack propagation life. The Fracture Analysis Code (FRANC) computer program was used to simulate crack propagation. Various fatigue crack growth models were used to estimate crack propagation life based on the calculated stress intensity factors(SIF). Experimental tests were performed in a gear fatigue rig to validate predicted crack propagation results. Test gears were installed with special crack propagation gages in the tooth fillet region to measure bending fatigue crack growth. Good correlation between predicted and measured crack growth was achieved when the fatigue crack closure concept was introduced into the analysis. As the gear rim thickness decreased, the compressive cyclic stress in the gear tooth fillet region increased.

[3] studied the crack propagation in the tooth foot of a spur gear of milling machine gear box by the Finite Element Method(FEM). In this method Zen-crack software is used to crack detection. The tooth foot crack propagation is a function of Stress Intensity Factors (SIF) that plays a very crucial role in the life span of the gear. An appropriate methodology for predicting the crack propagation path is applied by considering gear tooth behaviour in bending fatigue. The results are used to prevent catastrophic rim fracture failure modes from occurring in critical components.

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[4]determined the complete service life of mechanical elements N can then be determined from the number of stress cycles N_i required for fatigue crack initiation and the number of stress cycles N_P required for a crack to propagate from the initial to the critical crack length and failure as, $N=N_i + N_P$. Also he has discussed about various causes of breakage failure, various factors affecting the fatigue strength and also about methods to improve the fatigue life.

IV. DESIGN CALCULATIONS

Spur gear test rig design for input shaft speed 1440 RPM and power 15 KW. No. teeth on pinion are 14 and that on gear are 72. Load application factor is 1.5 and load distribution factor is 1.2.

Design of spur gear set

N_p=1440 RPM, P=15 KW,Z_p=14,Z_g=72,K_a=1.5,K_m=1.2, C=11500 x e N/mm,e=40 x 10^{-3}mm.

Material for both pinion and gear is Mild Steel.(S_{ut}=460 N/mm^2,S_{yt}=250 N/mm^2)

Hence, Bending stress, $\sigma_b = \frac{S_{ut}}{3}$ = 460 N/mm^2.

Lewi's Form factor,$Y_p = 0.484 - \frac{2.87}{Z_p}$

As material is same Pinion will be weaker than gear hence designing for pinion.

Now, Bending Strength,

$F_b = \sigma_{bp} \times b \times m \times Y_p$ = 427.79 m^2 N. (b = 10m)

Wear Strength,$F_w = d_p \times c \times Q \times K$ where, $Q = \frac{2Z_g}{Z_g+Z_p}$=1.6744 ,$k = 0.16\left[\frac{BHN}{100}\right]^2$=1.96

Hence, F_w=459.455m^2 N. F_b <F_w ,So designing will be according to beam strength.

Effective Load, $F_{eff} = \frac{K_a \times K_m \times f_t}{K_v}$, Where $K_v = \frac{6}{6+v}$ = $\frac{6}{6+1.00557m}$ ($V = \frac{\pi \times d_p \times N_p}{60.000}$ =1.0557 m).

Hence, $F_{eff} = \frac{25575.447}{m} + 4500$

$F_b = N_f \times F_{eff}$ Solving we get, m=5.63mm. Hence, m=6 as per standard module.

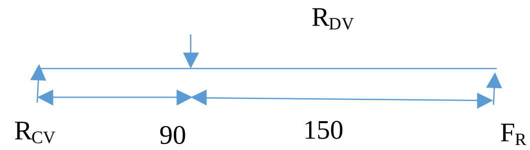
Dimensions for pinion and gear, m=6 mm, b =60 mm, d_p=84 mm, d_g=432 mm, h_a=1*m=6 mm, h_f=1.25*m=7.5 mm,

C.D. = $\frac{d_p+d_g}{2}$ =298 mm.

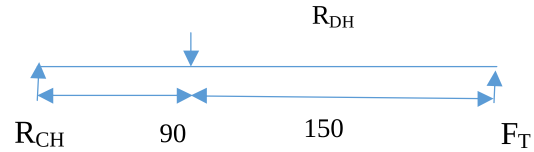
Design of pinion shaft

$\tau_{allowable}$= 0.18 x S_{ut} =82.8 N/mm^2 ,$\tau_{allowable}$= 0.3 x S_{yt} =75 N/mm^2

Shaft is weaker for yielding.so $\tau_{allowable}$ = 75 N/mm^2.



F_R=861.74 N,R_{DV}= 2297.973 ↓R_{CV}= 1436.233 ↑



F_T=2368.047 N,R_{DH} = 6314.097 ↓R_{CH}= 3946.83 ↑

hence, R_C=4200.044 N, R_D=6720.041 N.

Now, Bending moment for vertical load is 129261 N-mm and that for horizontal load is 355214.7 N-mm. Hence total bending moment will be 378002.5 N-mm. Calculated torque, T=99.4598 N-m. So, equivalent torque will be 390865.952 N-mm.

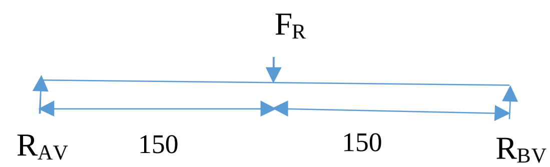
Now $T_e = \frac{\pi}{16} \times e \times d^3$, diameter of shaft, d=45 mm.

Bearing selection

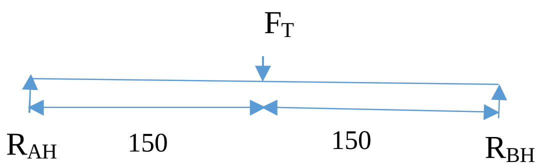
F_r=R_D=6720.041 N, $P_e = (X \times V \times f_r)$= 6720.041 N, assuming the life of bearing as 10,000 hrs.L_{10}=894 million revolution. Hence Dynamic capacity, C will be 63.84 N.

Hence selected bearing is 6409.So rated, C= 76.10 KN which is greater than calculated so selection of bearing is proper.

Design of gear shaft



F_R=861.74 N,R_{AV}= 430.87 N ,↑R_{BV}= 430.87 N ↑



F_T=2368.047 N,R_{DH} = 1184.0235 ,R_{CH}= 1184.0235 N
hence, R_A=1260 N, R_B=1260 N.

Now, Bending moment for vertical load is 64630.550 N-mm and that for horizontal load is 177607.275 N-mm. Hence total bending moment will be 189001.193 N-mm. Calculated torque,

T=511503.12 N-mm. So, equivalent torque will be 545304.4 N-mm.

Now $T_e = \frac{\pi}{16} \times e \times d^3$, diameter of shaft, d=45 mm.

Bearing selection

F_R=R_A=1260 N, $P_e = (X \times V \times f_r)$= 1260 N, assuming the life of bearing as 10,000 hrs.L_{10}=894 million revolution. Hence Dynamic capacity, C will be 12.138 N.

Hence selected bearing is 6009.So rated, C= 16.3 KN which is greater than calculated so selection of bearing is proper.

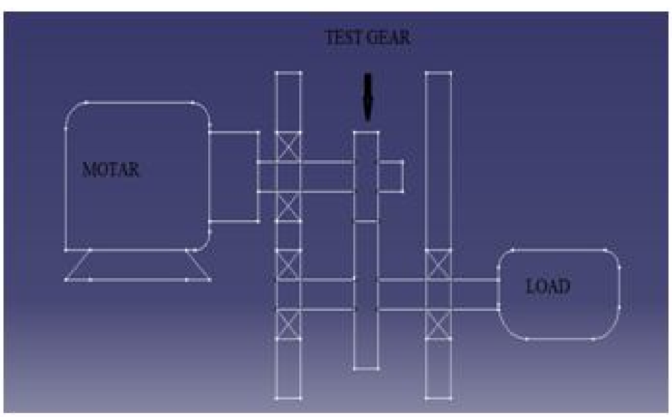


Fig 1-Line diagram of experimental setup

Experimental setup

Figure 1 shows line diagram of experimental setup.Variable speed motor will be used. A pinion gear which is nothing but the test gear will be mounted on motor shaft. The crack will initially be given at the root of gear tooth. Load will be given on second shaft. Two plates are used to hold the bearings.While working gear will be checked that is it will be removed from shaft and the propagated crack will be measured using fluorescent test.and using those readings calculations of fatigue life of gear will be calculated.

V .DESIGN OF SPUR GEAR

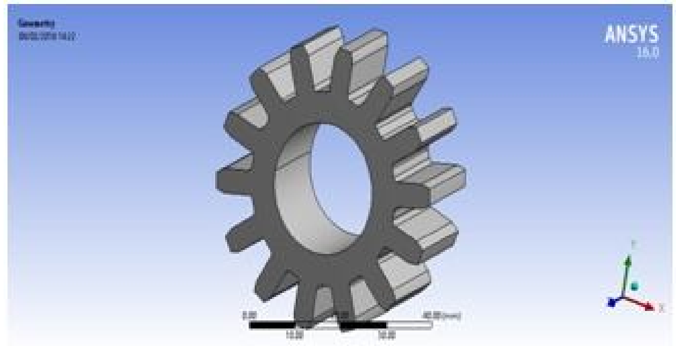


Fig. 2-spur gear

VI. MATERIAL PROPERTIES AND CALCULATION

Table 1-material properties and its value

Sr. No.	Property	Value
1.	Density	7870 Kg/m^3
2.	Young's Modulus	2E11Pa
3.	Poisson's ratio	0.30
4.	Yield strength	370 MPa
5.	Ultimate strength	440 MPa

VII. STATIC ANALYSIS OF SPUR GEAR

A. Solidworks Modelling

The spur gear is modelled using a Solidworks 2016 software.The spur gear is drawn in sketch first according to dimensions and then extruded.Crack is drawn initial at 80^o and rim thickness ratio is 0.8 and crack length is 0.5 mm. The Solidworksmodel is shown in Fig. 2.

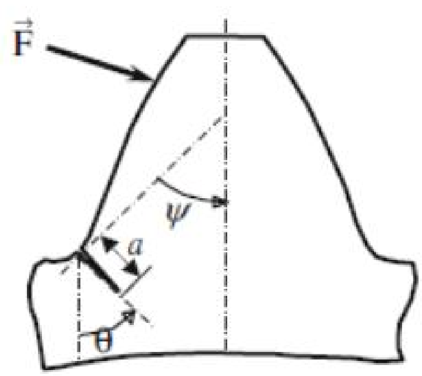


Fig 3-sketch of spur gear

B. Meshing

The ANSYS workbench uses a finite element method to discretize the model into finite elements. Finer the size accurate will be the answer. But at the same time the software needs more memory and time for processing. Thus optimum meshing is used.

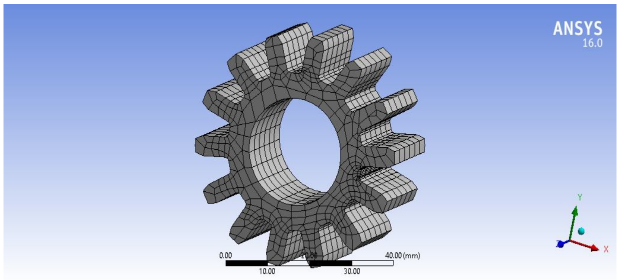


Fig. 4-meshing of spur gear

C. Analysis setup

A force 4200 N is applied on model at flange of spur gear tooth and the inner surface is kept fixed.

D. Deformation

The maximum deformation is 0.039 mm.
The minimum deformation is 0.0043 mm.

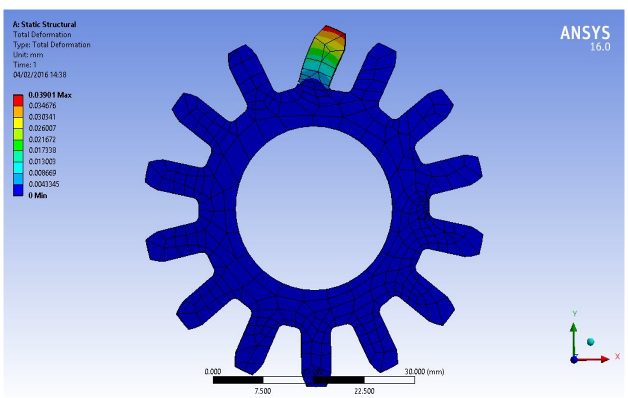


Fig 5-deformation of spur gear

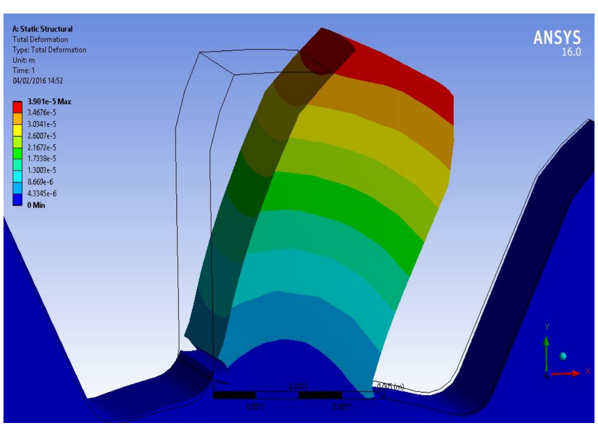


Fig 6-crack initiation of spur gear

E. Principle stresses

The maximum stress is 5.5487e8 Pa.
The minimum stress is 46.499 Pa.

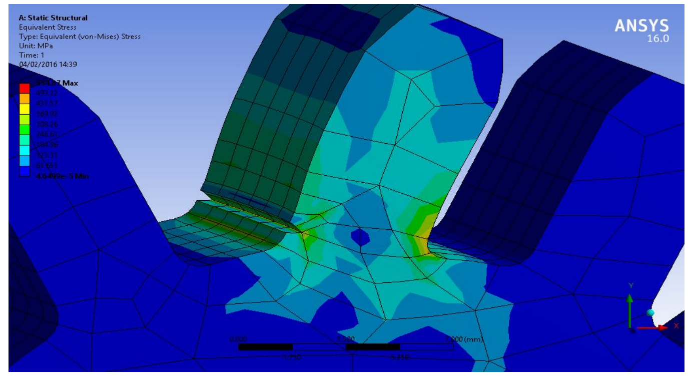


Fig 7-crack opening of spur gear

VIII. CONCLUSION

The Spur gear is analysed under static condition,And from the above analysis it can be concluded that as crack opening has started andit is failure of Mode 1.

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