BENDING STRESS ANALYSIS OF SPUR GEAR BY USING MODIFIED LEWIS FORMULA

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ABSTRACT

Gears are one of the most important component in mechanical power transmission systems. The bending stress of the gear tooth is regarded as one of the key contributors for the failure of the gear in the gear set. Thus, analysis of stresses has become popular as an area of study on gears to minimize the chances of failures and also for the optimal design of gears. The analytical study is based on Modified Lewis formula. The project is basically concerned with ANALYSIS of SPUR gears which are actually used in all types of two wheelers. For doing analysis I taken the SPLENDOR of HERO HONDA company. I can determined stress in tooth of spur gear by using Modified Lewis equation.

Keyword: Bending stress, Spur gear, Modified Lewis theory

I. INTRODUCTION

Gears are used for a variety of applications. They have numerous applications starting from textile looms to aviation industries. They are the most frequent ways of transmitting power. They will change the rate of rotation of machinery shaft as well as the axis of rotation. For high speed machine, such as an motor vehicle transmission, they are the optimum medium for low energy loss, high accuracy and reliability. Their function is to convert input provided simply by prime mover into an output with lower velocity and corresponding higher speed. Toothed gears are being used to transmit the power with high velocity ratio. In this phase, they face large stress at the point of contact. A set of teeth for is generally subjected to two types of cyclic stresses:

i) Bending stresses inducing bending fatigue

ii) Contact stress causing contact fatigue.

Both equally these kind of stresses may well not achieve their maximum values at the exact same point of contact. Nevertheless, combined action of that they are all is the reason of failure of gear tooth leading to fracture at the bottom layer of a tooth underneath bending fatigue and surface area failure, due to call fatigue. When loads will be applied to the body, their surfaces deform elastically near to the point of speak to. Stresses developed by Normal force in a photo-elastic type of gear teeth are displayed in the Fig.1.1. The highest stresses can be found at the region where the lines are bunched nearest together. The greatest stress arises at two locations:

A. At contact point exactly where the force F work.

B. At the fillet region near to the base in the tooth. [1]

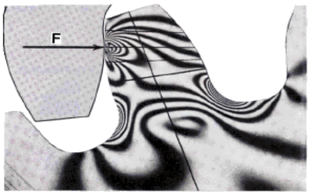

Fig 1.1 Photo-elastic Model of gear tooth [1]

II. LITERATURE REVIEW

V.Rajaprabakaran, Mr.R.Ashokraj The main objective of this study is to add different shaped slot to reduce stress attention. A finite factor type of Spur gear with a segment of three teeth is knowm for evaluation and stress concentration minimizing holes of numerous sizes will be introduced on gear tooth at various locations. Evaluation revealed that aero-fin designed hole introduced along the stress flow direction produced better results.

Toni Jabbour, Ghazi Asmar, present a method to calculate the distribution of the stress at the tooth origin along with the contact stress along each contact series of set of spur and helical gears. For the latter, the method is founded on the decomposition of the gear into an unlimited number of small field spur gears. The results received from this method include been further confirmed simply by finite factor calculations. To get spur gears, the location of the point of contact which causes to the critical tooth-root stress will depends on the contact ratio from the pair of gears which usually increases with the amount of teeth. The crucial contact stress is attanied at the radius from the pitch circle. For helical gears, the critical tooth-root stress is obtained in the event that the point of get in touch with situated at a range of 1.65 millimeter, while the contact stress is located at a radius equal to the radius of the pitch circle of the gear. Both stresses are obtained when the total length of lines of contact is minimal.

Miryam B. Sánchez, José I. Pedrero , Miguel Pleguezuelos, represent a non-uniform type of load distribution along the type of contact of spur and helical gears, obtained from the minimum amount of elastic potential criterion, offers has been used, combined with the equations of the linear elasticity, to judge the fatigue tooth-root stress. The critical value of the stress and the critical load conditions has been recevied and a complete evaluation of the tooth bending strength has been performed. As the load every unit of length in any point from the line of contact and any kind of position of the meshing cycle has been defined by a very easy analytic formula, a complete study of the location and the value of the tooth-root bending stress has been carried out. From this kind of study, a recommendation intended for the calculation of the bending load capacity of spur and helical gears is proposed. The research has been restricted to gears with transverse contact ratio between 1 and 2, with non-undercut tooth.

VishwjeetV.Ambade,A.V.Vanalkar,P.R.Gajbhiye. This paper presented analysis of Bending stress and get in touch with stress of Involute spur gear teeth in meshing. There are several verities of stresses present in loaded and rotating gear teeth. Bending stress and contact stress (Hertz stress) calculation is the simple stress analysis. It is usually difficult to get rigth answer on gear teeth stress by implying important stress equation, such while Lewis formula for bending stress and Hertz equation for contact stress. Numerous research methods just like

Theoretical, Numerical and Experimental include been done throughout the years. This paper displays the theoretical and numerical method of to calculate bending and contact stress.

Mahesh. Badithe1, Srimanthula Srikanth2, JithendraBodapalliIn the paper Static analysis of a 3-D model got been performed by utilizing ANSYS 10.0. Analysis uncovered that aero-fin shaped gap introduced along the stress flow direction yielded better results. Finally Stress reducing feature using condition of shape of aero-fin can be used in the path of stress circulation which helped to regulate stress flow by redistributing the lines of force. This also yielded better results in comparison with elliptical and circular holes.

III. PROBLEM STATEMENT

When two gear mesh with each other to transmit the load, the teeth of each gear under bending action. The bending stress will be maximum at the root of the tooth. Due to the periodical effect of load, fatigue cracks may occur near the tooth base, which create ultimate failure of the toothSo to avoid fatigue failure of the gear, the stresses should be reduced at maximum stress concentrated location. Aim of this work is stress analysis res of spur gear.

IV. OBJECTIVE

In spite of the numerous of investigations devoted to gear research and analysis there still remains to be developed, a general numerical approach will be capable of to predicting the effects of variations in gear geometry, contact as well as bending stresses, torsional mesh stiffness and transmission errors. The main focus of the current research as developed here is:

1. To calculate the bending stress of spur gear by using Modified Lewis formula.

V. METHODOLOGY

Work will be carried out in the following step.

1. Analytical Approach.

5.1 Theoretical Calculation

Dimension of Spur Gear

From the Specifications of two wheeler Hero Honda Splendor.

We know that

$P = 5.51*10^3$ w.

b = Face width = 11mm.

We have gear ratio for 4th pair, 3rd pair & 2nd pair. These are,

4th pair

$Z_p = 24$ $Z_g = 23$

$G4 = 0.958$

3rd Pair

$Z_p = 21$ $Z_g = 26$

$G3 = 1.238$

2nd Pair

$Z_p = 17$ $Z_g = 29$

$G2 = 1.70$

By measuring the different parameters of pinion and gear of 4th pair we have following parameters.

□**For Pinion**

No. of teeth on pinion Zp =24

Outer diameter of pinion O.D. = 45.5mm

□**For Gear**

No of teeth on gear Zg =23

Outer diameter of gear O.D. =43.7mm

Shaft diameter of input shaft =21mm.

Shaft diameter of output shaft =18mm.

From above given data we can find out module of gear

For Pinion	For Gear
O.D = (Zp+2)*m	O.D = (Zg+2)*m
45.5 = (24+2)*m	43.7 = (23+2)*m
m = 1.75	m = 1.75

Module of gears & pinions is

m=1.75

We have taken is as m = 1.75

from above given data we can find out different parameters and dimensions of gear pairs.

4th pair pinion

Zp = 24

1) O.D = 45.5

2) Pitch circle diameter

Pdp = m*2 = 1.75*24 = 42mm.

3) Base circle diameter

d_b = Pdcos ϕ = 42cos20 = 39.46mm

4) Root circle diameter

d_f = (Zp-2.5)m = (24-2.5)*1.75 = 37.625mm

5) Addendum h_a = m = 1.75mm

6) Dedendumh_f= 1.25m = hf = 1.25*1.75=2.18mm

7) Total Working Depth

$h_ft = h_a1+h_f1$= 1.75+2.18 = 3.9375mm

4th pair gear

Zg = 23

1) O.D.= 43.7mm

2)Pitch Circle Dia

Pdg = m*Zg = 1.75*23 = 40.25mm

3)Base Circle Dia

d_b= Pdcos ϕ = 40.25*cos20 = 37.82mm

4)Root Circle Dia

d_f = (Zg-2.5)*m = (23-2.5)*1.75 = 35.875mm

Speed in rpm

1) Gear

The maximum velocity taken for 4th pair is 70km/hr.

$$V = \frac{\pi d g n g}{60*1000}$$

$$\frac{70*1000}{3600} = \frac{\pi*40.25*ng}{60*1000}$$

ng = 9226.37rpm

From gear ratio

$$G4 = \frac{np}{ng}$$

$$0.958 = \frac{np}{9226.37}$$

np = 8841.93rpm

5.1.2 Tooth thickness at base circle and Pitch circle calculation

ϕ = 20°

Involute ϕ = 0.01490

4th Pair Pinion

At Base

Sb = rb2α

$\alpha = \frac{\pi}{2*Zp}$ + inv ϕ

$\alpha = \frac{\pi}{2*24}$+ 0.01490=0.08034

Sb = 19.73*2*0.080316 = 3.17mm

At Reference (Pitch)

S = d(α-invϕ) = 42(0.080316-0.01490) = 2.74mm

$\alpha = \frac{\pi}{2*23}$+ 0.01490 = 0.08319

4th Pair Gear

At Base

Sb = rb*2α

Sb = 18.91*2*0.08319 = 3.17mm

At Reference

S = d(α-invφ) = 40.25(0.0319-0.01490) = 2.74mm

Table 5.1.1 Dimension of gear pair

SR.No.		4^{th} PAIR		3^{rd} PAIR		2^{nd} PAIR	
		Pinion	Gear	Pinion	Gear	Pinion	Gear
1	No. of Teeth	24	23	21	26	18	29
2	Outer Diameter	45.5	43.7	40.25	49	35	54.25
3	Pitch Circle Diameter	42	40.25	36.75	45.5	31.5	50.75
4	Base Diameter	39.46	37.82	34.53	42.75	29.6	47.68
5	Root Diameter	37.62	35.87	32.37	41.12	27.12	46.37
6	Thickness At Base	3.17	3.17	3.09	3.21	3.17	3.29
7	Thickness At Pitch Circle	2.74	2.74	2.74	2.74	2.74	2.74

5.1.3 Bending Stress Calculations By Using Modified Lewis Equation.

$\sigma b= \frac{M\,Y}{I} = \frac{Ft*L*t/2}{bt3/2} = \frac{6Ft*L}{bt2} = \frac{Ft^{\wedge}*Y}{meccos\emptyset}$

Ft = tangential load

Ft*= tangential load per width

b = width

t = thickness of tooth at base

L = length of teeth

e_{c} = contact ratio

Y= Lewis form factor

4th Pair Gear

$Ft \frac{P}{V}$

$Ft = \frac{P * 60 * 10^3}{\pi * Pdg * ng}$

$= \frac{5.51 * 10^3 * 60 * 10^3}{\pi * 40.25 * 9228.94}$

Ft = 283.37N

$e_c = \frac{\text{Length of Arc}}{\pi m}$

Length of Arc = $\frac{\text{Length of path contact}}{\cos\emptyset}$

Length of Arc = $\frac{8.24}{\cos 20}$ = 1.59

$Y = \frac{3\cos\emptyset x}{2y^\wedge 2} = \frac{3 * 20 * 70875}{2 * 4.47^\wedge 2} = 0.55$

$\sigma b = \frac{Ft^\wedge * Y}{mec\cos\emptyset} = \frac{25.26 * 0.55}{1.75 * 1.59 * 4.47^\wedge 2}$

$= 5.41\ N/mm^2$

Table 5.1.2 Tangential Force and Bending Stress of gear pair

SR.No.			4^{th} PAIR		3^{rd} PAIR		2^{nd} PAIR	
			Pinion	Gear	Pinion	Gear	Pinion	Gear
1		Tangential Force(N)	287.37	287.37	661.25	661.20	751.95	793.44
2		Bending Stress (N/mm^2)	5.41	5.41	12.64	12.64	14.32	15.17

VI. CONCLUSION

From table we conclude that,

1. Torque decreases with increase in speed.
2. Bending stress increases with increase in tangential face.

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