Stress Analysis of Herringbone Gear by Using FEA

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Abstract — **Herringbone gears are extensively used in numerous engineering applications including gearboxes. Premature failures of such gears could lead to many serious consequences such as process downtime and late delivery which are critically important in this day and age of intense competition. This paper reports the results of an investigation into the premature failure of a herringbone gear used in a gearbox of a steel millin Wardha. The gear failed after about 9000 hrs.of service which was much shorter than the normal service life of 20,000 – 30,000 hrs. Itwas concluded the herringbone gear subsequently failed due to random fracture initiated by surface and subsurface damages resulting fromexcessive bending stress.Structural analysis on herringbone gear used in rolling mill gearbox has been carried out.The stress generated on gear tooth have been analysed theoretically as well as by using Finite Element analysis.Finally the results obtained by theoretical analysis and finite element analysis are compared to check the correctness.**

Keywords: **herringbone gear failure, bending stress, finite element analysis**

I. INTRODUCTION

A gear is a rotating machine part having cut teeth, which mesh with another toothed part in order to transmit torque. Two or more gears working in tandem are called a transmission and can produce a mechanical advantage through a gear ratio and thus may be considered a simple machine. Geared devices can change the speed, magnitude, and direction of a power source. The most common situation is for a gear to mesh with another gear however a gear can also mesh with a nonrotating toothed part, called a rack, thereby producing translation instead of rotation. An advantage of gears is that the teeth of a gear prevent slipping. Herringbone gears are widely used in numerous engineering applications including gearboxes. Gearboxes are key Components of most heavy duty machines and are extensively used in steel industry. Failure of gears not only results in replacement cost but also in process downtime. This could have a drastic consequences on productivity and, more importantly, on delivery which could possibly result in permanent loss of customers. For example, in this case the 'downtime' was 3 days and 4,200 metric tonne's of steel output was lost before the failed herringbone gear could be replaced.

There are two basic modes of gear tooth failure-breakage of tooth due to static and dynamic loads and the surface destruction. The complete breakage of tooth can be avoided by adjusting the parameters in the gear design, such as the module and the face width, so that the beam strength of the gear tooth is more than the static and dynamic loads. The surface destruction and tooth wear is classified according to the basis of their primary causes.the principal types of gear tooth wear are Abrasive wear, corrosive wear, initial pitting, destructive pitting, scoring and fatigue breakage.

This paper aims at identifying the cause of failure of a herringbone gear in a gearbox (used in a steel rolling milling, Wardha).The knowledge gained from investigation would help prevent or minimize their occurrence of similar failures in the future.

II. A. PROBLEM IDENTIFICATION

By visual inspection it is observed that the failure occurred in the gearbox is random fracture and the pitting failure. Pitting is a surface fatigue failure of the gear tooth. It occurs due to repeated loading of tooth surface and the contact stress exceeding the surface fatigue strength of the Material. Material in the fatigue region gets removed and a pit is formed.

Figure 1. Location of failure of gear tooth

Figure 2. Location of failure of herringbone gear

B. OBJECTIVE

To find out the bending stresses induced on the teeth of gears which causes tooth failures theoretically and by using finite element analysis.

III. DESIGN OF HERRINGBONE GEAR

Number of teeth on pinion Z1 =32
Number of teeth on Gear Z2 =120
Speed of pinion N1 = 340 rpm

$$\text{Speed of gear } N2 = \frac{Z1}{Z2} \times N1 = \frac{32}{120} \times 340 = 90 \text{ rpm}$$

Helix angle = 35^0
Normal pressure angle $\phi n = 20^0$
Power, P = 375 KW

Angular Velocity of pinion,

$$\omega_1 = \frac{2\pi N}{60} = \frac{2\pi \times 340}{60} = 35.60 \text{ rad/sec}$$

Angular Velocity of gear,

$$\omega_2 = \frac{2\pi N2}{60} = \frac{2\pi \times 90}{60} = 9.42 \text{ rad/sec}$$

$$\text{Speed ratio} = I = \frac{N1}{N2} = \frac{340}{90} = 3.77$$

Torque,

$$T_1 = \frac{1000 \times P}{w1} = \frac{1000 \times 375}{35.60} = 10533.70 \text{ Nm}$$

$$T_2 = \frac{1000 \times P}{w2} = \frac{1000 \times 375}{9.42} = 39808.91 \text{ Nm}$$

The double helical gear is considered as two single helical gears coupled together sharing the torque equally. Torque on each half is

T_1 = 10533.7/2 = 5266.85 Nm
= 5266853 N-mm.

T_2 = 39808.91/2
= 19904.458 Nm
= 19904458 N-mm.

The AGMA bending stress equation,

$$\sigma_b = \frac{Ft}{bm_n\ j} k_v k_0 (0.93\ k_m) \ldots\ldots\ldots\ldots(1)$$

Where,
F_t – Tangential load
b – Face Width
m_n – Normal module
J – Load multiplier
Kv – velocity factor
Ko – Overload Factor
Km – Load Distribution Factor

Parameters and values for calculating the bending stress
We have PCD of gear is 41'' = 1041.4mm.

Module m = PCD/no. of teeth
= 1041.4/120
=8.678mm
Normal Module m_n = m x Cos 35
m_n = 7.11 mm
Considering standered value on higher side
= 8 mm
Assuming b = $1.2p_a$ = 1.2 p /tan ψ
= 1.2 x $3.833m_n$/ $\tan 35^o$ = $6.569m_n$

$F_t = 2T_2/ d_2 = 2T_1 / mZ_2 = 2T_2\cos\psi / m_n Z_2$ = (2 x 19904458 x cos35) / (m_n x120) = 271746.29 / m_n

Load Multiplier, J = 0.5505
Velocity factor, K_v = 1.2
Overload Factor, K_o =2.0
Load Distribution Factor, K_m =1.7

Using equation (1) and the above values the calculated bending stress is

$$\sigma_b = 556.90 \text{ MPa}$$

Tensile yield strength of material 17 Mn 1Cr 95 is 1100 MPa.

Considering FOS = 2
Allowable stress of material is 550MPa.

IV. SOLID MODELLING AND ANALYSIS

A solid modelling is done with Pro E 5.0 and then meshing and Analysis is done with ANSYS Workbench 12.0.

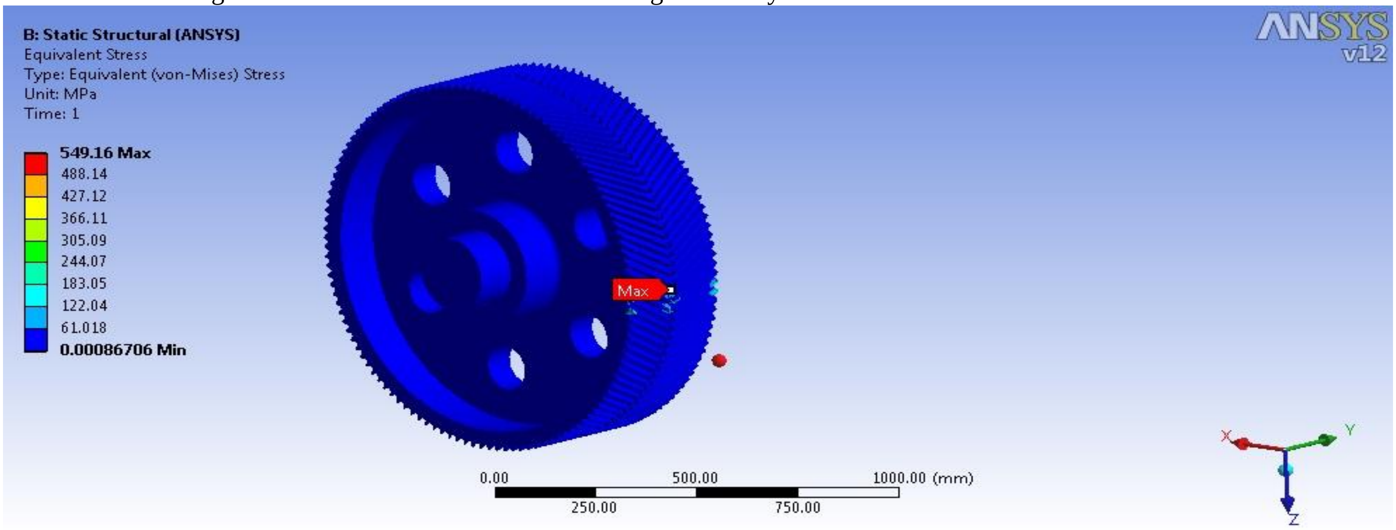

Figure. 3 Ansys result for material 17 Mn 1Cr 95

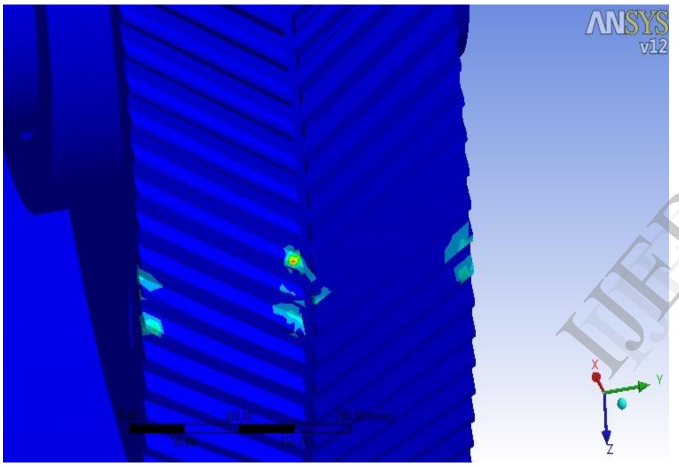

Figure 4. Stress plot

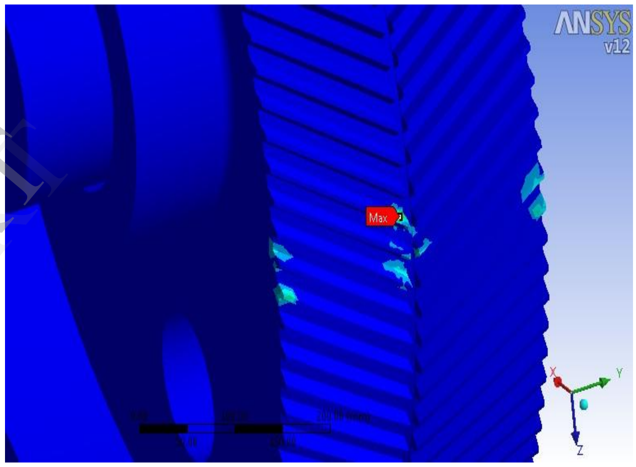

Figure 5. Maximum principal stress plot

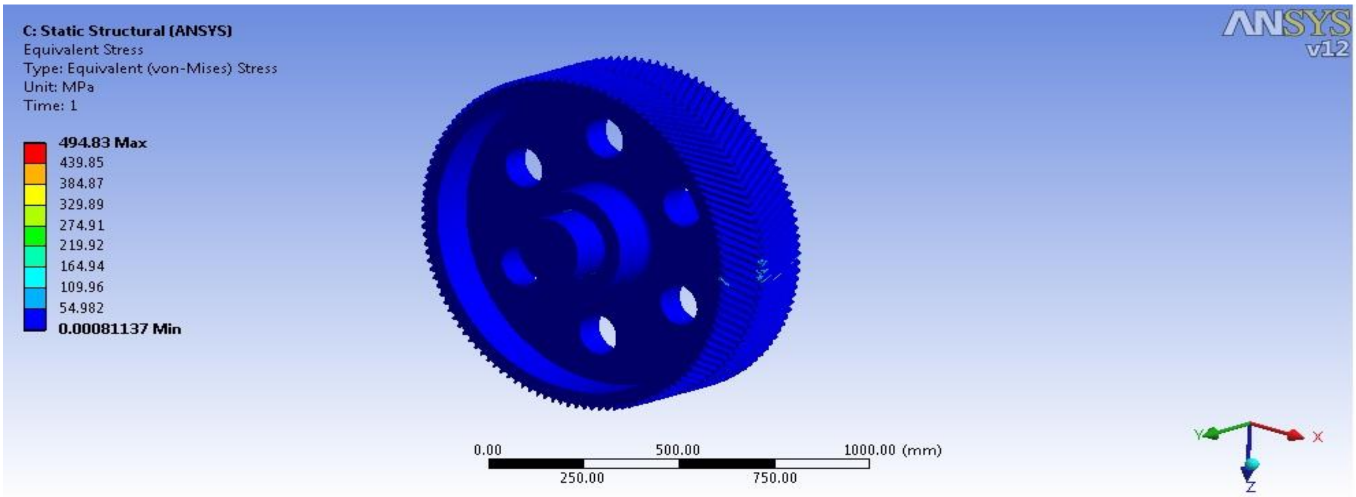

Figure 6. Ansys result for material 15 Ni 4Cr 1

V. RESULT AND DISCUSSIONS

Theoretical design is carried out using standard design formulae as per AGMA procedure and analysis carried out using ANSYS .The following table .1 shows the comparison the theoretical design values with ANSYS values.

Table 1:- Comparison of theoretical and ANSYS design values

SR. NO	MATERIAL	CALCULATED STRESS (MPa)	ANSYS RESULT (MPa)	ALLOWABLE STRESS (MPa)
1	17 Mn 1Cr 95 (magnesium alloy)	556.90	594.16	550
2	15 Ni 4Cr 1 (Nickel Chrome Alloy)	556.90	494.83	750

Since from calculated stress for Magnesium alloy are exceeded than the allowable limit so, the reason for failure of material are bending stress.

VI. CONCLUSIONS AND RECOMMENDATION

1. The failure of herringbone gear was caused by excessive bending stress on the surface of gear teeth. The calculated bending stress is higher than the allowable stress of gear material.

2. It is found that Allowable stress for the gear material (magnesium alloy) is less than the calculated bending stress. Therefore the material having higher value of allowable stress is to be used for manufacturing of herringbone gear.

3. From the results, it is observed that the bending stresses of Nickel alloy are less than that of the magnesium alloy.

4. Hence Nickel alloy is best suited for herringbone gears used in Rolling mill gearbox.

5. The Fracture starts from pitting area at the surface of gear tooth. The pitting occurred as a result of excessive stress.

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