ANALYSIS OF LOAD FACTORS AND MODES OF FAILURE ON SPUR GEAR

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Abstract - *Gears play major role in the transmission of mechanical power. The analysis of spur gear is carried out based on load distribution factor. The bending and contact stresses are considered as the cause of failure on spur gear. The detailed study on modes of failure is made. The load factor is calculated with various angular arrangements. The analysis is carried out with and without considering the external factors. The mathematical model is analyzed with Finite Element Analysis (FEA) and compared with the analytical procedure and the percentage error is calculated.*

A Finite Element procedure is developed in this work to determine the load distribution factor, of the American Gear Manufacturer's Association (AGMA) formula for a set of spur gear. At first, a spur gear with perfect involute is modeled using 3-D CAD software. The model is then assembled with shafts having 1, 2, and 3 degree misalignments. The generated 3-D model is then imported to ANSYS workbench to calculate the maximum bending and contact stresses using FEA. The result generated is then compared with the maximum bending stress results obtained for parallel shafts to estimate the Load Distribution Factor.

Key Words: **Finite Element Analysis, American Gear Manufacturer's Association, Spur Gear, Bending Stress, Contact Stress, Load Distribution Factor, Angular Arrangement.**

1. INTRODUCTION

Gear, Belt and Chain drives are often called as mechanical drives. A mechanical drive is defined as a mechanism, which is intended to transmit mechanical power over a certain distance, usually involving change in speed and torque. In general, mechanical drive is required between the prime mover, such as electric motor and the part of the operating machine. Mechanical drive is used on account of the following reasons:

1. The torque and speed of the machine are always different than that of electric motor or engine.

2. In certain machines, variable speeds are required for the operation, where as the prime mover runs at constant speed.

Although gear manufacturing has achieved lots of advancement during its evolution, however the failure of gear due to bending and contact stress still remained a challenge for designers and manufacturers until 1892. In 1892 the Philadelphia Engineers club first recognized Wilfred Lewis presentation of stresses on the gear tooth and it still serves as the basis to determine the gear stress.

The Lewis bending equation has a lot of draw backs which include

1. Load on gear tooth is dynamic and is influenced by pitch-line velocity.
2. The entire load is carried on single tooth.
3. The location of application of load is not true as the load is shared by the tooth.
4. The Stress concentration factor at tooth fillet is not considered.

In order to overcome all these factors AGMA (American Gear Manufactures Association) came out with several factors which influence bending stress on the gear tooth is used.

2. LITERATURE REVIEW

Ismail Ali Abdul Aziz et al. (2017) reviewed that the methodology used to investigate bending strength of spur gear; Finite Element Method (FEM), Numerical Calculation and Investigational Techniques were usually carried out in order to understand the bending strength of thin-rimmed spur gear. The most common method used to investigate bending strength is the numerical calculation. This method was used with several types of established equations and standards to predict gear failures. Next stage is to simulate the gear using Finite Element Method (FEM) in order to get the analysis of gear strength. The most important stage is to put the gears to physical experiment or testing facilities to determine and validate all data from the numerical calculation and FEM method.

Vivek Singh et al. (2013) presented the stress redistribution by introducing the stress relieving features in the stressed zone to the reduction of root fillet stress in spur gear. In this work circular stress relieving features are used and better results are obtained. A finite element model with a segment of three teeth is considered for analysis and stress relieving features of various diameters and shapes are introduced on gear teeth. Analysis revealed that circular stress relieving features at specific locations are beneficial.

N. D. Narayankar and K. S. Mangrulkar (2017) presented the contact stress analysis and bending stress analysis of spur gear by Analytical method. For contact stress analysis, Hertzian equation is used and for bending stress analysis of spur gear by Analytical method. For contact stress analysis Hertzian equation is used and for bending stress analysis Lewis equation is used. For calculating these contact stress and bending stress analysis of Spur gear both material of Pinion and Gear is made of Steel.

K Sivakumar et al. (2015) calculated the contact stress of a spur gear tooth pair for two different materials C-14 and Sintered Steel is going to be carried out. Thereafter Contact stress and bending stress at gear and pinion of both materials analyzed using ANSYS. The comparisons of contact stress results are to be studied.

Govind T Sarkar et al. (2013) investigated finite element model for monitoring the stresses induced of tooth flank, tooth fillet during meshing of gears. The involute profile of helical gear has been modeled and the simulation is carried out for the bending and contact stresses and the same have been estimated. To estimate bending and contact stresses, 3D models for different helical angle, face width are generated by modeling software and simulation is done by finite element software packages. Analytical method of calculating gear bending stresses uses AGMA bending equation and for contact stress AGMA contact equations are used. It is important to develop appropriate models of contact element and to get equivalent result using Ansys and compare the result with standard AGMA stress.

Patil Amol Shivaji and Prof. Dr.S.B.Zope (2015) analyzed the one of failure types in gears. Scuffing is often characterized as a lubrication failure frequently accompanied by a sudden increase in friction and the instantaneous temperature at the contact zone. A pair of spur gear 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 fatigue. These types of failures can be minimized by careful analysis of the problem during the design stage and creating proper tooth surface profile with proper manufacturing methods. In general, gear analysis is multidisciplinary, including calculations related to the tooth stresses and to tribological failures such as wear or scoring.

3. DESIGN OF GEAR AND CYLINDER

3.1 Gear

The involute spur gear used for the current analysis has the following specifications

1. No. of Teeth of 36
2. Diametrical Pitch of 10'
3. Pressure Angle of 20^{o}

The gear is generated using Solidworks is

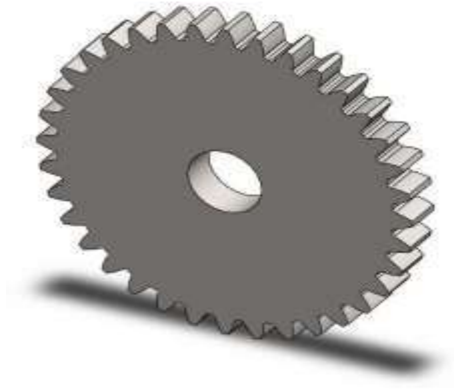

Fig -1 Exploded View of Gear

3.2 Cylinder

The current design has the two cylinders with 1 inch diameter and height equal to that of face width of the involute gear. The following are the specifications of the cylinder

1. Diameter is 1'
2. Height is 0.4'

The cylinders generated using Solidworks is

Fig -2 Cylinders

4. BENDING STRESS AND CONTACT STRESS WITHOUT FACTORS

4.1 Bending Stress

1. Analytical Calculation of Bending Stress Lewis bending stress from equation

$$\sigma_b = \frac{W_t . P_d}{b . y}$$

$$\sigma_{Lewis} = \frac{93.93 * 10}{0.4 * 0.38}$$

$$\sigma_{Lewis} = 6181.57psi$$

2. Analysis of Bending Stress with FEA with Ansys

Finite Element analysis for any imported 3-D model is performed in three main steps

1. Pre-Processing

2. Solution

3. Post-Processing

For the current analysis average aspect ratio is obtained as 1.85 by setting the mesh relevance to fine and smoothing to medium and span angle center to coarse in Discretization.

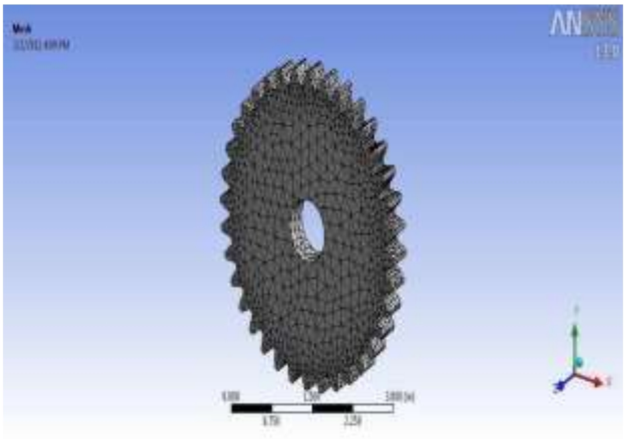

Fig -3 Final Mesh of Gear

Table -1: Force Components

Co-ordinate System	Force (lbs.)
X- Component	-93.96 lbs. (Ramped)
Y- Component	-34.20 lbs. (Ramped)
Z- Component	0 lbs.

Solution

The location of loads on gear is

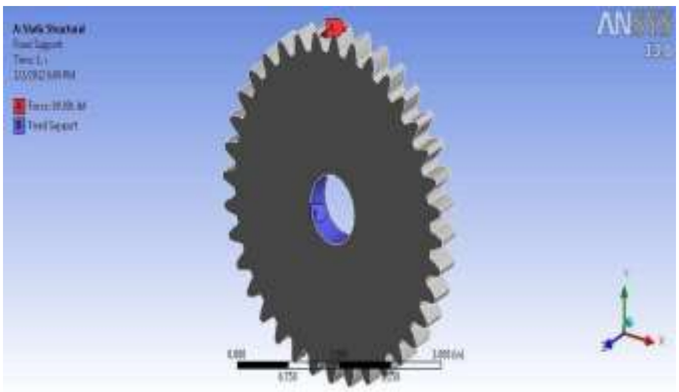

Fig -4 Location of Loads

Post Processing

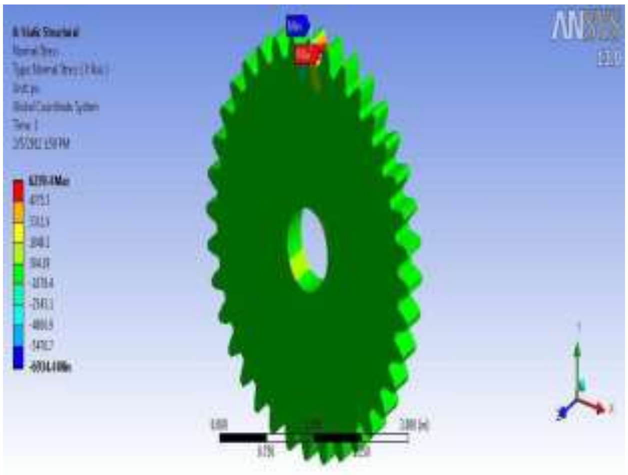

Fig -5 Bending Stress

4.2 Contact Stress

1. Analytical Calculation for Maximum Pressure on Cylinder

$$P_{max} = \frac{2F}{qLC}$$

where,

1. F is force acting on the Cylinders
2. C is half width of the ellipse
3. L is length of Cylinder
4. P_{max} is the maximum pressure generated

The following are the specifications and other factors used for calculation of the hertz contact stress

- Modulus of Elasticity E: 30E6
- Poisson's Ratio µ:0.28
- Load 500lb
- Length L:0.4
- Diameter d: 1

From above parameters the contact stress are given by the equation (3.10)

$$(\sigma_C)_{Hertz} = 161087.99 \ psi$$

2. Analysis of Cylinder by Ansys

For current analysis, an alloy with Poisson's ratio of 0.28 and Young's Modulus of 30E6 psi is used. A mesh with an aspect ratio of 2.00 is required.

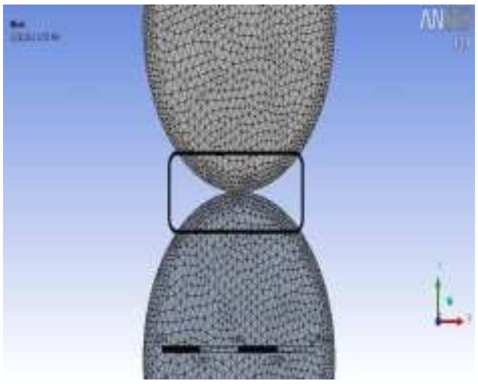

Fig -6 Fine Mesh of Cylinder

Post – Processing

To obtain Contact pressures a contact tool must be used. The contact surfaces on which the pressures to be determined are selected and then evaluated for contact pressures.

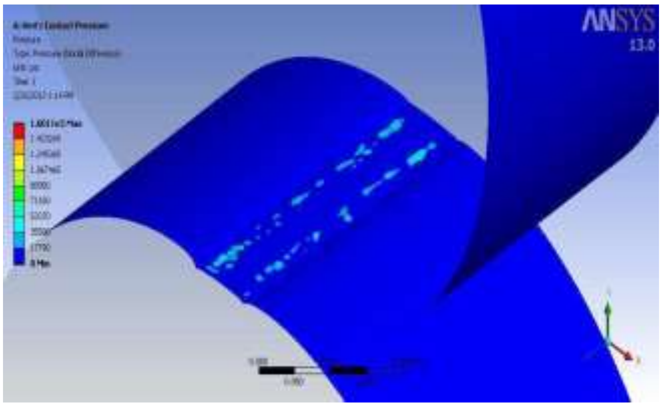

Fig -7 Contact Stress

5. DESIGN AND MODELING OF GEAR WITH BEARING

5.1Design of Assembly

Bearing Specifications

- Bore: 0.75
- Outer Diameter: 1.6250
- Thickness: 0.3
- Number of Balls: 10

Bearing Holder Specifications

Bearing holder is designed based on bearing specifications. The diameter of the outer race of the bearing is equal to the inner diameter of the bearing holder and thickness of the bearing holder is equal to thickness of bearing.

Spur Gear Specifications

- Number of Teeth N: 36
- Diametrical Pitch P_d : 10
- Diametrical Pitch P_d : 10
- Pressure Angle φ:20
- Face Width b: 0.4

The design of gear with bearing with Solid Works

Fig -8 Exploded View

5.2 Analysis with Ansys

Grey Cast Iron is used for this Analysis. The main material properties like the Young's modulus, Poisson's Ratio and Density that are required to perform a static analysis are defined in this section.

The geometry is then configured by adding the contacts and joints to the imported model. The current assembly is configured with two different contacts and two different joints.

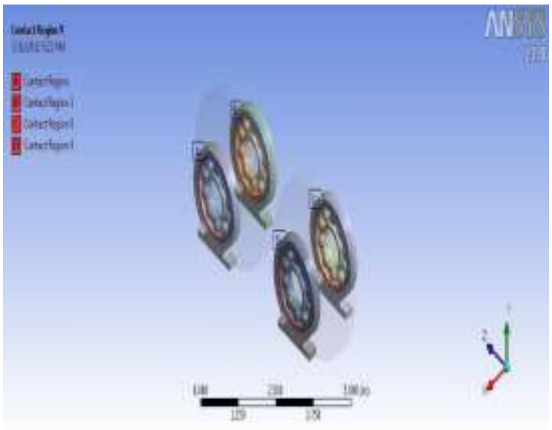

Fig -9 Bonded Contacts

The location of the No-separation contact is displayed in the following figure

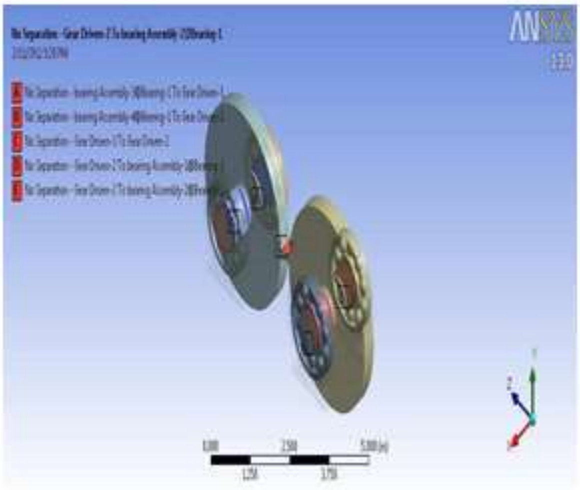

Fig -10 No Separation Contacts

Location of Types of Joints

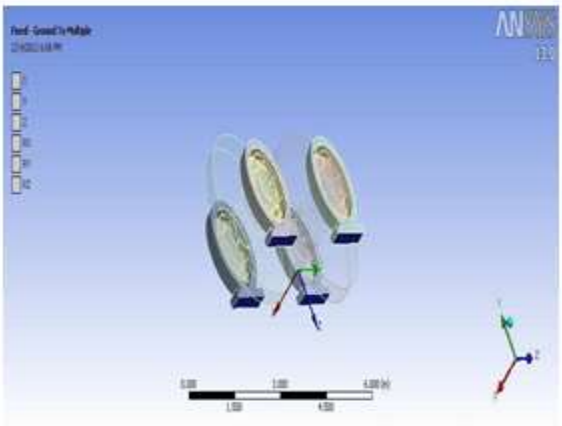

Fig -11 Location of Fixed Joints

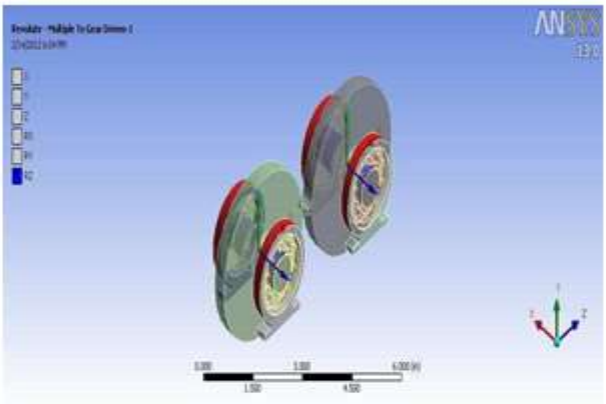

Fig -12 Location of Revolute Joints

Discretization

Meshing for contact analysis is complex and requires more refined meshing tools for accurate solution. For the current analysis a fine mesh is used with smoothing and initial span angle kept medium. Initial seed size should be kept part since we need the mesh to be distinct and separate from each part. The following image displays the generated mesh.

Fig -13 Element View of Final Mesh

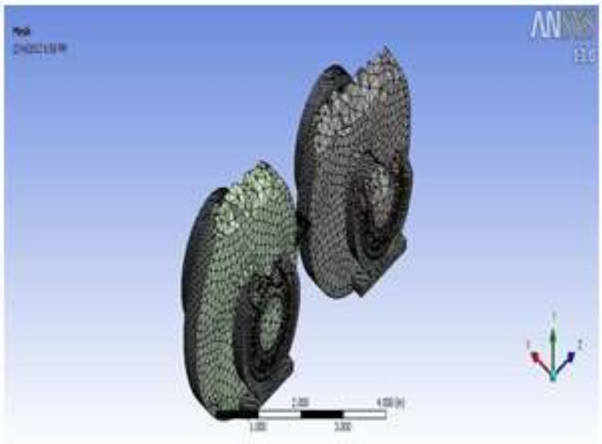

Fig -14 Tetrahedral Mesh Elements

5.3 Bending Stress

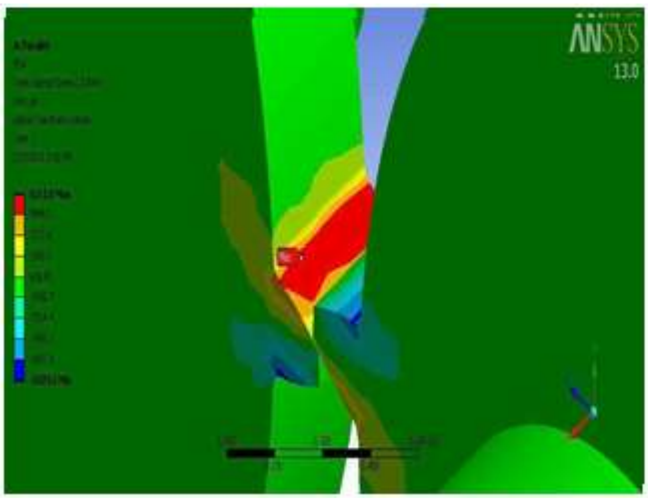

Fig -15 Stress Distribution on Z- Axis

5.4 Contact Stress

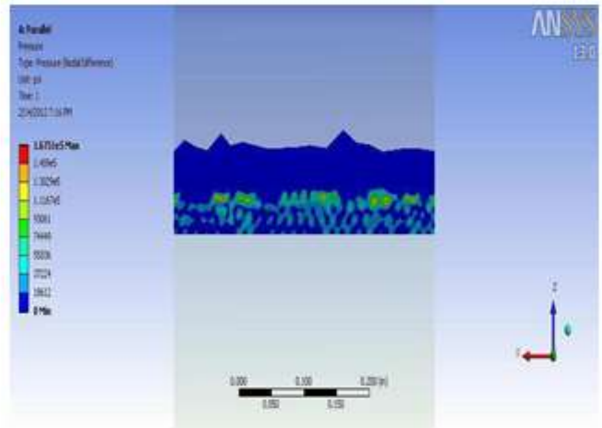

Fig -16 Contact Stress Analysis

6. LOAD DISTRIBUTION FACTOR(K_n)

Determination of load distribution factor depends on various factors which include the design of gear as well as design of shaft, bearings, housing and structure on which gear drive is mounted. The main objective of the load distribution factor is to reflect the non-uniform load distribution across the line of contact. The load distribution factor can either be defined by AGMA2001/2101or can be defined directly which is given by

$$K_n = 1.0 + C_{pf} + C_{ma}$$

where,

C_{pf} is proportion factor

C_{ma} is Mesh Alignment factor

6.1 Calculation of Bending Stress with Various Alignments

In the present chapter the variation of the bending stress with various angular misalignments parallel to the plain of action is studied. At first FE analysis for maximum bending stress is performed for parallel shafts and compared it to the maximum bending stress for 1, 2 and 3 degree

Table -2 Maximum Bending Stress for various angular alignments

Angular Alignment	Maximum Bending Stress (psi)
Parallel	6315.8
1°	6550
2°	7002.3
3°	7214.8

6.3 Calculation of AGMA Load Distribution Factor

The load distribution factor can be defined or can be calculated from the empirical method of AGMA 2001/2101.

In the present study the load distribution factor for parallel gear is defined as one and the stress distribution factor for the various angular alignments can be calculated as

$$\sigma_{FEA} = K_n * \sigma_{AGMA} * X$$

$$K_n = \frac{\sigma_{FEA}}{\sigma_{AGMA} * X}$$

Table -3 Load Distribution factors for various Alignments

Alignment	Load Distribution Factor
1°	1.03
2°	1.11
3°	1.14

7. FAILURE ON GEARS

7.1 NATURE OF FAILURE

The failure conditions can determine when and how to conduct an analysis. For example, if the gears are damaged but still able to function, the company may decide to continue their operation and monitor the rate at which damage progresses. In this case, samples of the lubricant should be collected for analysis, the reservoir drained and flushed, and the lubricant replaced. If gearbox reliability is crucial to the application, the gears should be examined by magnetic particle inspection to ensure that they have no cracks. The monitoring phase will consist of periodically checking the gears for damage by visual inspection and by measuring sound and vibration.

7.2 TEST TO PREDICT FAILURE ON GEAR

The non destructive tests, which aid in detecting material or manufacturing defects and provide rating information, include:

- Surface hardness and roughness.
- Magnetic particle inspection.
- Acid etch inspection.
- Gear tooth accuracy inspection.

The destructive tests to evaluate material and heat treatment. These tests include:

- Micro hardness survey.
- Micro structural determination using various acids etches.

- Determination of grain size.
- Determination of nonmetallic inclusions.
- SEM microscopy to study fracture surfaces.

8. CONCLUSION

Bending and Contact Stress is considered as one of the failure cause of gears. In this thesis, Lewis equation and Hertzian Equation are considered to analyze for Bending and Contact Stress respectively.

Another cause of failure is uneven load distribution on gear tooth, which is being calculated with the three different angular misalignments.

The AGMA formula is used for analytical calculation which is being compared with the FEA solution.

The stresses are calculated with and without using the external factors.

Bending stress

On comparing both the values the FEA value is deviated by 0.9% from AGMA bending stress value, without affect of external factors.

On comparing both the values the FEA value is deviated by 2% from AGMA bending stress value, with affect of external factors.

Contact Stress

On comparing both the values the FEA value is deviated by 0.6% from AGMA bending stress value, without affect of external factors.

On comparing both the values the FEA value is deviated by 4% from AGMA bending stress value, with affect of external factors.

From shaft misalignment, the angles 1, 2 and 3 degrees are considered; the load distribution factors and maximum bending stress are calculated. The results obtained are 1.03, 1.11, 1.14 and 5315.8 psi, 6550 psi, 7002.3 psi, 7214.8psi.

From these data obtained once the alignment of shaft changes the chance of failure on gear increases due to the large load distribution and stress concentration on gear tooth.

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