ISSN 2395-1621

Determination of true bending stresses in spur gears by kiss soft, fem & experimental setup

#1Pradip B. Adake, #2Amol N. Patil

1pradipadake@gmail.com
2patil.amol2015@rediffmail.com

1ME Student of Mechanical Engineering, S.P. Pune University, Dr. D. Y. Patil School of Engineering, Lohegaon, Pune. India.
2Assistant Professor, Department of Mechanical Engineering, S.P. Pune University Dr. D. Y. Patil School of Engineering, Lohegaon, Pune. India.

ABSTRACT

This paper presents the work carried regarding the determination of true bending stresses in spur gears. The two dimensional model of gear teeth was prepared using 'KISSsoft software. The results of bending stress obtained by 'KISSsoft software' are in well agreement with those obtained by analytical method. And also The three dimensional model of gear teeth was prepared using 'Creo 2.0' software. IGES files were imported to 'ANSYS' software. 8 Node Brick 45 (SOLID45) elements were used for meshing. Loads were applied along the top face at equal distances. The results of bending stress obtained by 'ANSYS'& KISSsys KISSsoft are in well agreement with those obtained by analytical method. Keeping the load per unit thickness of the gear constant & the analysis is repeated for various tooth thicknesses of the gears. Also load variation is done to study the effect of load on bending stress & deformation.

After that we will measure bending stress on actual gears & compare with these results with theoretical results.

Keywords— Spur Gear, Bending Stress, FEA; KISSsys & KISSsoft; Bending Stress of Spur Gears

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Gear drive is used to transmit power from one shaft to another when distance between two shafts is very small. It maintains the constant velocity ratio without slip. Gears may be classified as spur gear, helical gear, bevel gear, rack and pinion type, worm and worm wheel etc. Spur gears i.e. gears with their teeth parallel to axis of the gear are widely used, as they can be manufactured easily & are capable of withstanding normal loads & are good for low speeds.

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined application of gear technology.

Gears analyses in the past were performed using analytical methods, which required a number of assumptions and simplifications. In general, gear analyses are multidisciplinary, including calculations related to the tooth stresses and to tribological failures such as like wear or scoring. In this thesis, static contact and bending stress analyses were performed, while trying to design spur gears to resist bending failure and pitting of the teeth, as both affect transmission error. As computers have become more and more powerful, people have tended to use numerical approaches to develop theoretical models to predict the effect of whatever are studied. This has improved gear analyses and computer simulations. Numerical methods can potentially provide more accurate solutions since they

normally require much less restrictive assumptions. The model and the solution methods, however, must be chosen carefully to ensure that the results are accurate and that the computational time is reasonable.

A. True Bending Stress:

It is well known from the literature that the true bending stresses at the tooth root of spur gears are quite different from the nominal values that are utilized for the calculation of load capacity, either by standards or usual design rules. No problems arise in using a load capacity rating when the simplified values are compared with the results of bending fatigue tests whose limits are calculated with the same schematic method. But the "true" stress at the tooth root has different trends and values, and the designer must be aware of this difference, especially for light gears with narrow ribs and rims.

In the case of tooth bending strength, a cantilever-beam model is generally used to compute the bending stress. With this approach, Lewis in 1892 first calculated the tooth root stress of spur gear teeth (W. Lewis, "Investigation of the Strength of Gear Teeth," Proceedings of Engineers Club, Philadelphia). This model is still the basis for standard calculation methods successfully used in gear design. However, the local stress state—the "true" stress—in the tooth root fillet may be different from the nominal values obtained by this method. In truth, the calculation of the maximum tensile stress at the tooth root is a three-dimensional problem: The plane strain or plane stress model can be used without approximations only in the case of infinite, or infinitesimal, face width.

In this work, an accurate FEM analysis has been done of the "true" stress at tooth root of spur gears in the function of the gear geometry. The obtained results confirm the importance of these differences.

B. Bending Stresses In Gears:-

There are several failure mechanisms for spur gears. Bending failure of the teeth is one of the main failure modes. The bending stresses in a spur gear are another interesting problem. When loads are too large, bending failure will occur. Bending failure in gears is predicted by comparing the calculated bending stress to experimentally-determined allowable fatigue values for the given material. This bending stress equation was derived from the Lewis formula. Wilfred Lewis (1892) was the first person to give the formula for bending stress in gear teeth using the bending of a cantilevered beam to simulate stresses acting on a gear tooth. The Lewis equation considers only static loading and does not take the dynamics of meshing teeth into account.

Different factors required for the calculation, can be obtained from the books on machine design. This analysis considers only the component of the tangential force acting on the tooth, and does not consider effects of the radial force, which will cause a compressive stress over the cross section on the root of the tooth. Suppose that the greatest stress occurs when the force is exerted at top of tooth, which is the worst case. When the load is at top of the tooth, usually there are at least two tooth pairs in contact. In fact, the maximum stress at the root of tooth occurs when the contact point moves near the pitch circle because there is only one tooth pair in contact and this teeth pairs carries the entire torque. When the load is moving at the top of the tooth, two teeth pairs share the whole load if the ratio is larger than one and less than two. If one tooth pair was considered to carry the whole load and it acts on the top of the tooth this is adequate for gear bending stress fatigue.

Fatigue or yielding of a gear tooth due to excessive bending stresses is one important gear design considerations. In order to predict fatigue and yielding, the maximum stresses on the tensile and compressive sides of the tooth, respectively, are required. In the past, the bending stress sensitivity of a gear tooth has been calculated using photo elasticity or relatively coarse FEM meshes. However, with present computer developments we can make significant improvements for more accurate FEM simulations.

Work to be completed:-

- Analyze the bending stress in spur gears using Theoretical calculations for various gear thickness
  1. For Gear Face Width 20 mm
  2. For Gear Face Width 30 mm
  3. For Gear Face Width 40 mm
- Modeling of Gears in KISSsys
- Analyze the bending stress in spur gears using KISSsoft software for various gear thickness
  1. For Gear Face Width 20 mm
  2. For Gear Face Width 30 mm
  3. For Gear Face Width 40 mm

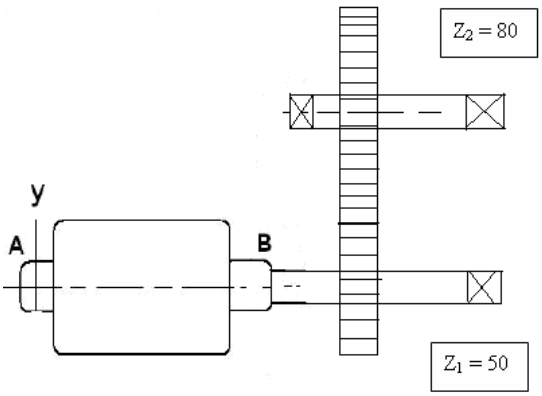
Proposed work:-

- Experimental setup for actual readings of Bending Stress
- Finite Element Analysis To study the effect of load on the Bending Stress & Deformation

The objective the work is:

- To study the variation in the bending stress, if any, along the thickness on the bending stress keeping load per unit thickness constant.
- To study the effect of load on the Bending Stress & Deformation
- The stress field at the spur gear tooth root will be analyzed using a KISSsys & KISSsoft Software. The study will be for thin-rimmed gear geometries.
- The study will examine the bending stress along the tooth width or thickness for a fixed geometry and for different values of face width. Linear force uniformly distributed along the face width will be applied.
- Also effect of load on true bending stress will be studied.
- Software KISSsys &KISSsoft will be used for calculation of Bending Stress.
- To analyze the bending stress in spur gears using actual practical setup.
- To analyze the bending stress in spur gears using FEM

II. GEARING ARRANGEMENT & THEOROTICAL CALCULATIONS



Speed	Rpm	Teeth	Nos.	P.C.D.
n1	750	Z1	50	d1 150
n2	397	Z2	80	d2 240

Parameters:
Motor Power (P) = 15 HP (8 Pole, Speed 750 Rpm)
Module (m) = 3 mm
No. of Teeth of Pinion (n1)= 50
Pitch Circle Diameter of Pinion (d1)= Module XNo. of Teeth = 3 X 50 = 150 mm
Face Width (b) = 20 mm
No. of Teeth of Gear (n2) = 80

Tooth Bending Stress (AGMA)
The bending stress is found from AGMA equation:

$$\sigma = \frac{F_t}{b.m.J} K_v K_0 K_m$$

Where,
Ft = Tangential force
b = Face width of the gear
m = Module of the gear
J = Spur gear geometry factor.
Kv = Velocity factor or dynamic factor
Ko = Overload factor
Km= Load distribution factor
Spur Gear Geometry Factor (J): This factor includes the Lewis form factor Y and also a stress concentration factor based on a tooth fillet radius. It also depends on the number teeth in the mating gear.
Velocity factor or dynamic factor (Kv): This factor indicates the severity of impact as successive pairs of teeth engage. This is a function of pitch line velocity and manufacturing accuracy.
It is given by inverse of Barth's equation & is as follows.

$$K_v = \left[\frac{78 + (200V)^{0.5}}{78}\right]^{0.5}$$

$$= \left[\frac{78 + (200 \text{ x } 5.89)^{0.5}}{78}\right]^{0.5}$$

$$= 1.2$$

Overload factor (Ko) : This factor reflects the degree of non-uniformity of driving and load torques.

1 Bending Stress for Face width of Gear 20 mm
The Actual Bending Stress is found from AGMA equation:

$$\sigma = \frac{F_t}{b.m.J} K_v K_0 K_m$$

Where,

$$\sigma = \frac{1900}{20 \text{ x } 3 \text{ x } 0.457} \text{ x } 1.2 \text{ x } 1 \text{ x } 1.3$$

$$= 107.912 \text{ MPa}$$

But, in case of the Finite Element Based Software 'ANSYS', the inputs given to the software are only: Young's Modulus & Poisson's ratio.
So the values obtained from software will not have any consideration of Velocity factor or dynamic factor, Overload factor, Load distribution factor.
The value of stress obtained from software will be corresponding to

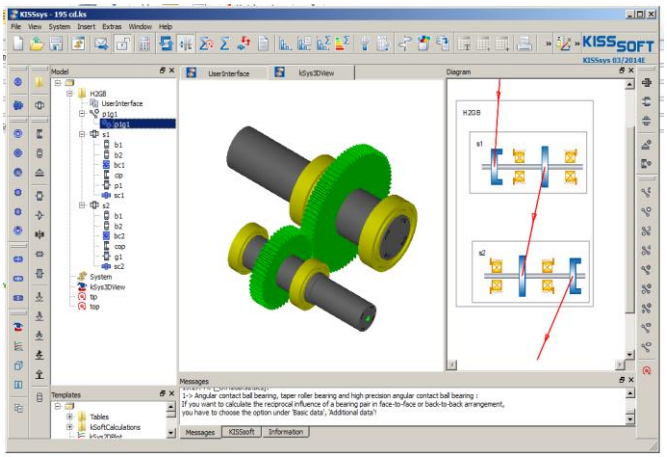
$$\sigma = \frac{F_t}{b.m.J}.$$

So,
σ = 69.175 MPa

III. BENDING STRESS CALCULATIONS USING KISSSOFT

The dissertation work deals with 'Determination of True Bending Stress in spur gear.

Bending Stresses Results Using KISSsoft:



Bending Stresses for Gear with Face Width = 20 mm-
The following result shows the true bending stresses at the root of the teeth for a gear with 20 mm thickness. Using KISSsoft software we find out results is

Output:

KISSsoft - Release 10-2008
KISSsoft evaluation

File
Name : PRADIP
Changed by: PRADIP on: 18.11.2014 at: 20:11:04

CALCULATION OF A CYLINDRICAL SPUR GEAR PAIR

Drawing or article number:
Gear 1: 0.000.0
Gear 2: 0.000.0

Calculation-method AGMA 2101-D04 (Metric Edition)

------- GEAR 1 -------- GEAR 2 --

Transmitted power (kW, hp, ft*lb/s) [P] 11.190, 15.006, 8253
Speed (1/min) [n] 750.0 468.8
Torque (Nm, ft*lb) [T] 142.5, 105.1 228.0, 168.1
Overload factor [Ko] 1.00
Required service life [H] 40000.00
Gear driving (+) / driven (-) + -
Gearbox type: Standard gearbox with closed housing

1. TOOTH GEOMETRY AND MATERIAL

(Geometry calculation according ISO 21771)
------- GEAR 1 -------- GEAR 2 --
Centre distance (in, mm) [a] 7.6772, 195.000
Centre distance tolerance ISO 286 Measure js7
Normal Diametral Pitch (1/in)[Pnd] 8.46667
Transverse Diametral Pitch (1/in)[Pd] 8.46667
Normal module (in, mm) [mn] 0.11811, 3.0000
Pressure angle at normal section (°) [alfn] 20.0000
Helix angle at reference diameter (°) [beta] 0.0000
Number of teeth [z] 50 80
Facewidth (mm) [b] 20.00 20.00
Helix Spur gear
Accuracy grade [Q-AGMA2015] A6 A6
Inside diameter (mm[di] 0.000.00
Inside diameter of rim (mm)[dbi] 0.00 0.00

Material
Gear 1: 18CrNiMo7-6, Case-carburized steel, case-hardened ISO 6336-5 Figure 9/10 (MQ), core strength >=25HRC Jominy J=12mm<HRC28
Gear 2: 18CrNiMo7-6, Case-carburized steel, case-hardenedISO 6336-5 Figure 9/10 (MQ), core strength >=25HRC Jominy J=12mm>=HRC28

------- GEAR 1 -------- GEAR 2 --
Surface hardness HRC 61 HRC 61
(lb/in²), (N/mm²) (lb/in²), (N/mm²)
Allowable bending stress number [sat] 62366, 430.00 66717, 460.00
Allowable contact stressnumber [sac] 217557,1500.00 217557,1500.00
Tensile strength (N/mm²) [Rm] 1200.00 1200.00
Yield point (N/mm²) [Rp] 850.00 850.00
Youngs modulus (N/mm²) [E] 206000.00 206000.00
Poisson's ratio [ny] 0.30 0.30
Average roughness, Ra, tooth flank (µm) [RAH] 0.60 0.60
Mean roughness height, Rz, flank (µm) [RZH] 4.80 4.80
Mean roughness height, Rz, root (µm) [RZF] 20.00 20.00

Tool or reference profile of gear 1:
Reference Profil 1.25 / 0.38 / 1.0 ISO 53.2 Profil A
Addendum factor [haP*] 1.000
Dedendum coefficient [hfP*] 1.250
Tip radius factor [rhoaP*] 0.000
Root radius factor [rhofP*] 0.380
Tip form height coefficient [hFaP*] 0.000
Protuberance height factor [hprP*] 0.000
Protuberance angle [alfprP] 0.000
Ramp angle [alfKP] 0.000
Not topping

Tool or reference profile of gear 2 :
Reference Profil 1.25 / 0.38 / 1.0 ISO 53.2 Profil A
Addendum factor [haP*] 1.000
Dedendum coefficient [hfP*] 1.250
Tip radius factor [rhoaP*] 0.000
Root radius factor [rhofP*] 0.380
Tip form height coefficient [hFaP*] 0.000
Protuberance height factor [hprP*] 0.000
Protuberance angle [alfprP] 0.000
Ramp angle [alfKP] 0.000
Not topping

Sum of reference profile gears:
Dedendum reference profile (module) [hfP*] 1.250 1.250
Tooth root radius Refer. Profile (module)[rofP*] 0.380 0.380
Addendum Reference profile (module) [haP*] 1.000 1.000
Protuberance height (module) [hprP*] 0.000 0.000
Protuberance angle (°)[alfprP] 0.000 0.000
Buckling root flank height (module) [hFaP*] 0.000 0.000
Buckling root flank angle (°) [alfKP] 0.000 0.000

Type of profile modification: No
Tip relief (µm) [Ca] 2.00 2.00

Type of lubrication oil bath lubrication
Type of oil Oil: ISO-VG 320
Lubricant base Mineral-oil base
Kinem.viscosity oil at 40 °C (mm²/s) [nu40] 320.00
Kinem.viscosity oil at 100 °C (mm²/s) [nu100] 22.00
FZG-Test A/8.3/90 (ISO14653-1) [FZGtestA] 12
Specific density at 15 °C (kg/dm³) [roOil] 0.900
Oil temperature (°C) [TS] 70.000
ambient temperature (°C) [TU] 20.000

------- GEAR 1 -------- GEAR 2 --
Overall transmission ratio [itot] -1.600
Gear ratio [u] 1.600
Transverse module (mm)[mt] 3.000
Pressure angle at Pitch circle (°)[alft] 20.000
Working transverse pressure angle (°) [alfwt] 20.000
[alfwt.e/i] 20.019 / 19.981
Working pressure angle at normal section (°) [alfwn] 20.000
Helix angle at operating pitch diameter (°)[betaw] 0.000
Base helix angle (°) [betab] 0.000
Reference centre distance (mm) [ad] 195.000
Sum of the Addendum modification [Summexi] 0.000
Profile shift coefficient [x] 0.0844 -0.0844
Tooth thickness (Arc) (module) [sn*] 1.6322 1.5093

Modification of tip diam. (mm) [k] 0.000 0.000
Reference diameter (mm) [d] 150.000 240.000
Base diameter (mm) [dB] 140.954 225.526
Tip diameter (mm) [da] 156.507 245.493
(mm) [da.e/i] 156.507 / 156.497 245.493 / 245.483
Tip diameter allowances (mm) [Ada.e/i] 0.000 / -0.010 0.000 / -0.010
Tip chamfer/ tip rounding (mm) [hK] 0.000 0.000
Tip form circle (mm) [dFa] 156.507245.493
(mm) [dFa.e/i] 156.507 / 156.497 245.493 / 245.483
Operating pitch diameter (mm) [dw] 150.000 240.000
(mm) [dw.e/i] 150.018 / 149.982 240.028 / 239.972
Root diameter (mm) [df] 143.007 231.993
Generating Profile shift coefficient [xE.e/i] 0.0409 / 0.0180 -0.1279 / -0.1508
Manufactured root diameter with xE (mm)
[df.e/i] 142.746 / 142.608 231.732 / 231.595
Theoretical tip clearence (mm) [c] 0.750 0.750
Effective tip clearence (mm) [c.e/i] 0.977 / 0.858 0.977 / 0.858
Active root diameter (mm) [dNf] 145.580 234.809
(mm) [dNf.e/i] 145.620 / 145.547 234.853 / 234.772
Root form diameter (mm) [dFf] 145.293 234.177
(mm)[dFf.e/i] 145.110 / 145.015 233.973 / 233.866
Reserve (dNf-dFf)/2 (mm) [cF.e/i] 0.303 / 0.219 0.494 / 0.400
Addendum (mm) [ha] 3.253 2.747
(mm) [ha.e/i] 3.253 / 3.248 2.747 / 2.742
Dedendum (mm) [hf] 3.497 4.003
(mm) [hf.e/i] 3.627 / 3.696 4.134 / 4.202
Roll angle at dFa (°) [xsi_dFa.e/i] 27.648 / 27.639 24.638 / 24.632
Roll angle to dNa (°) [xsi_dNa.e/i] 27.648 / 27.639 24.638 / 24.632
Roll angle to dNf (°)[xsi_dNf.e/i] 14.865 / 14.745 16.648 / 16.574
Roll angle at dFf (°) [xsi_dFf.e/i] 14.015 / 13.852 15.827 / 15.725
Tooth depth (mm) [H] 6.750 6.750
Virtual gear no. of teeth[zn] 50.000 80.000
Normal Tooth thickness at Tip cyl.(mm) [sa] 2.283 2.422
(mm) [san.e/i] 2.189 / 2.132 2.329 / 2.274
Normal Tooth space as Tip cylinder (mm) [efn] 2.421 2.324
(mm) [efn.e/i] 2.464 / 2.487 2.351 / 2.366
Max. sliding speed at tip (m/s) [vga] 1.067 0.950
Specific sliding at the tip [zetaa] 0.399 0.399
Specific sliding at the root [zetaf] -0.665 -0.665
Sliding factor on tip [Kga] 0.181 0.161
Sliding factor on root [Kgf] -0.161 -0.181
Pitch (mm) [pt] 9.425
Base pitch (mm) [pbt] 8.856
Transverse pitch on contact-path (mm) [pet] 8.856
Length of path of contact (mm) [ga, e/i] 15.804 (15.871 / 15.713)
Length T1-A, T2-A (mm) [T1A, T2A] 18.204(18.137/18.284) 48.490(48.490/48.477)
Length T1-B (mm) [T1B, T2B] 25.152(25.152/25.141) 41.542(41.475/41.621)
Length T1-C (mm) [T1C, T2C] 25.652(25.626/25.677) 41.042(41.001/41.084)
Length T1-D (mm) [T1D, T2D] 27.061(26.993/27.141) 39.633(39.633/39.621)
Length T1-E (mm) [T1E, T2E] 34.008(34.008/33.997) 32.686(32.618/32.764)
Length T1-T2 (mm) [T1T2] 66.694 (66.627 / 66.761)
Diameter of single contact point B (mm)
[d-B] 149.661(149.661/149.654) 240.343(240.297/240.398)
Diameter of single contact point D (mm)
[d-D] 150.987(150.939/151.045)

239.051(239.051/239.042)
Addendum contact ratio [eps] 0.944(0.947/ 0.939) 0.841(0.846/ 0.835)
Minimal length of contact line (mm) [Lmin] 20.000

Transverse contact ratio [eps_a] 1.784
Transverse contact ratio, effective [eps_a.e/m/i] 1.792 / 1.783 / 1.774
Overlap ratio [eps_b] 0.000
Total contact ratio [eps_g] 1.784
Total contact ratio, effective [eps_g.e/m/i] 1.792 / 1.783 / 1.774

2. FACTORS OF GENERAL INFLUENCE

------- GEAR 1 -------- GEAR 2 --
Calculated with the operating pitch diameter:
Nominal circumferential force (N) [Ftw] 1899.7
Axial force (N) [Faw] 0.0
Radial force (N) [Frw] 691.4
Net face width of narrowest member (in)[F] 0.79 (20.00 mm)
Nominal force at operating pitch dia. (lb)
[Wt] 426.86 (1899.67 N)
Pitch line velocity (ft/min) [vt] 1159.54 (5.89 m/s)
Gear unit type: Commercial enclosed gear unit
Mesh alignment factor [Cma] 0.139
Mounting procedure: Contact improved by adjusting at assembly
Mesh alignment correction factor [Ce] 0.800
Gearing: without longitudinaleflanc correction
Load distribution factor introduced:
Face load distribution factor [Cmf] 1.000
Load distribution factor [Km] 1.000
Transmission accuracy level number [Av] 7
Dynamic factor [Kv] 1.145
Number of load cycles (in mio.) [NL] 1800.000 1125.000

3. TOOTH ROOT STRENGTH

------- GEAR 1 -------- GEAR 2 --
Rim thickness factor [KB] 1.00 1.00
Size factor [KS] 1.00 1.00
Limiting Variation in action (in/10000) [LimVarAc] 2.00
Load sharing:
0 = No (Loaded at tip) 1 = Yes (Loaded at HPSTC) 0
Calc. as helical gear (0) / as LACR (1) 0 0
Load angle (°) [phinL] 24.97 22.73
Calculation of factor Y following AGMA 908
(in) , (mm) (in) , (mm)
Height of Lewis parabola [hF] 0.215, 5.47 0.214, 5.44
Tooth thickness at critical section [sF] 0.244, 6.20 0.247, 6.28
Radius at curvature of fillet curve [roF] 0.048, 1.22 0.048, 1.21
Helical factor [Ch] 1.00
Helix angle factor[Kpsi] 1.00
Tooth form factor Y [Y] 0.444 0.446
Stress correction factor [Kf] 1.530 1.545
Helical overlap factor [Cpsi] 1.00
Load sharing ratio [mN] 1.00
Bending strength geometry factor J [J] 0.290 0.289
(lb/in²), (N/mm²) (lb/in²), (N/mm²)
Bending stress number [st] 18117, 124.91 18204, 125.51
Stress cycle factor [YN] 0.928 0.936
(for general applications)
(lb/in²), (N/mm²) (lb/in²), (N/mm²)
Allowable bending stress number [sat] 62366, 430.00 66717, 460.00
Temperature factor [KT] 1.00 1.00
Reliability factor [KR] 1.00
Reverse loading factor [-] 1.000 1.000
Effective allow. b.s.n. [sateff] 57863, 398.95 62420, 430.37
Bending strength power rating (hp) [Pat] 47.93(35.74 kW) 51.45(38.37 kW)
(Calculated with SFmin = 1.0)
Unit load [UL] 4590.5, 31.650
Allowable unit load [Uat] 14661.6, 101.088 15740.1, 108.524
Safety factor (foot) [sateff/st] 3.19 3.43
Required safety factor [SFmin] 1.00 1.00
Transmittable power [Patu/SFmin] 47.93(35.74 kW) 51.45(38.37 kW)

(Note: Materials with HB > 400: Yield strength not checked.)

4. SAFETY AGAINST PITTING (TOOTH FLANK)

------- GEAR 1 -------- GEAR 2 --
(lb^.5/in), (N^.5/mm)
Elastic coefficient [Cp] 2285.3, 189.81
Size factor [Ks] 1.000 1.000
Load sharing ratio [mN] 1.000
Helical overlap factor [Cpsi] 1.000
Geometry factor I [I] 0.098

(lb/in²), (N/mm²)
Contact stress number [sc] 74813, 515.82
Stress cycle factor [ZN] 0.887 0.897
(for general applications)
Surface condition factor [Cf] 1.00 1.00
Hardness ratio factor [CH] 1.00 1.00
Temperature factor [KT] 1.00 1.00
Reliability factor [KR] 1.00
(lb/in²), (N/mm²) (lb/in²), (N/mm²)
Allowable contact stress number [sac] 217557,1500.00 217557,1500.00
Effective allow. c.s.n. (lb/in²) [saceff] 193068,1331.15 195166,1345.62
Pitting resistance power rating (hp) [Pacu] 99.94(74.52 kW) 102.12(76.15 kW)
(Calculated with SHmin = 1.0)
Contact load factor (lb/in²) (N/mm²) [K] 149.2, 1.029
Allowable contact load factor (lb/in) [Kac] 107462.5,740.928 109811.1,757.121

Safety factor (flanc) [saceff/sc] 2.58 2.61
Required safety factor [SHmin] 1.00 1.00
Transmittable power (hp) [Pacu/SHmin^2] 99.94(74.52 kW) 102.12(76.15 kW)

SERVICE FACTORS:
Service factor for tooth root [KSF] 3.19 3.43
Service factor for pitting [CSF] 6.66 6.81
Service factor for gear set [SF] 3.19

5. STRENGTH AGAINST SCUFFING

Results from AGMA 925 (Details see in the specific calculation sheet)
Probability of wear (%)[Pwear] 6.873
Probability of scuffing (%) [Pscuff] low (<= 5%)

6. TOOTH THICKNESS DIMENSIONS

------- GEAR 1 -------- GEAR 2 --
Tooth thickness tolerance DIN3967 cd25 DIN3967 cd25
Tooth thickness allowance (normal section) (mm) [As.e/i] -0.095 / -0.145 -0.095 / -0.145

No of teeth over which to measure [k] 6.000 9.000
Base tangent length ('span') (no backlash) (mm) [Wk] 50.984 78.467
Actual base tangent length ('span') (mm) [Wk.e/i] 50.895 / 50.848 78.378 / 78.331
Diameter of contact point (mm) [dMWk.m] 149.853 238.750

Theor.ball/roller diameter (mm) [DM] 5.113 5.031
Actual ball/roller diameter (mm) [DMeff] 5.250 5.250
Theor.dim. centre to ball (mm) [MrK] 78.991 123.547
Actual dimension centre to ball (mm) [MrK.e/i] 78.875 / 78.813 123.423 / 123.357
Diameter of contact point (mm) [dMMr.m] 150.517 239.701
Theor.dimension over two balls (mm) [MdK] 157.982 247.094
Actual dimension over balls (mm) [MdK.e/i] 157.750 / 157.626 246.846 / 246.715
Actual dimension over rolls (mm) [MdR.e/i] 157.750 / 157.626 246.846 / 246.715

Chordal tooth thickness (no backlash) (mm) ['sn] 4.896 4.528
Actual chordal tooth thickness (mm) ['sn.e/i] 4.801 / 4.751 4.433 / 4.383
Chordal height from da.m (mm) [ha] 3.291 2.766
Tooth thickness (Arc) (mm) [sn] 4.897 4.528
(mm) [sn.e/i] 4.802 / 4.752 4.433 / 4.383

Axial Distance Without Backlash (mm) [aControl.e/i] 194.739 /194.602
Backlash free centre-distance, Tolerances (mm)jta] -0.261 / -0.398
Centre distance deviation (mm) [Aa.e/i] 0.023 / -0.023
Circumferential backlash from Aa (mm) [jt_Aa.e/i] 0.017 / -0.017
Radial clearance (mm) [jr] 0.421 / 0.238
Circumferential backlash (transverse section) (mm) [jt] 0.307 / 0.173
Normal backlash (mm) [jn] 0.288 / 0.163

7. TOLERANCES

------- GEAR 1 -------- GEAR 2 --
Following AGMA 2000-A88:
Accuracy grade [Q-AGMA2000] 11 11
Pitch Variation Allowable (µm) [VpA] 7.90 8.60
Runout Radial Tolerance (µm) [VrT] 29.00 33.00
Profile Tolerance (µm) [VphiT] 11.00 11.00
Tooth Alignment Tolerance (µm) [VpsiT] 6.60 6.60
Composite Tolerance, Tooth-to-Tooth (µm) [VqT] 14.00 14.00
Composite Tolerance, Total (µm) [VcqT] 44.00 47.00
(AGMA <-> ISO: VpA<->fpb, VrT<-> Fr, VpsiT<-> Fb, VqT<-> fi", VcqT<-> Fi")

Following AGMA 2015-1-1A01:
Accuracy grade [Q-AGMA2015] A6 A6
Single normal pitch deviation (µm) [fptT] 9.50 9.50
Total cumulative pitch deviation (µm) [FpT] 36.00 40.00
Profile deviation (µm) [ffaT] 10.00 11.00
Profile angular deviation (µm) [fHaT] 8.00 8.50
Profile total deviation (µm) [FaT] 13.00 14.00
Helix form deviation (µm) [ffbT] 8.50 8.50
Helix slope deviation (µm) [fHbT] 8.50 8.50
Tooth helix deviation (µm) [FbT] 12.00 12.00
Tooth-to-tooth tangential composite deviation (µm) [fisT] 3.60 4.00
Total tangential composite deviation (µm) [FisT] 40.00 44.00
(AGMA <-> ISO: fptT<->fpt, FpT<->Fp, fisT<-> fi', FisT<-> Fi', FaT<-> Fa)
(: fHaT<->fHa, ffaT<->ffa, FbT<-> Fb, fHbT<->fHb, ffbT<->ffb)

Tolerance for alignment of axes (recommendation acc. ISO/TR 10064, Quality 6)
Maximum value for deviation error of axis (µm) [fSigbet] 12.12
Maximum value for inclination error of axes (µm) [fSigdel] 24.23

8. ADDITIONAL DATA

Singular tooth stiffness (N/mm/µm) [c'] 14.447
Meshing spring stiffness (N/mm/µm) [cg] 22.947
Maximal possible centre distance (eps_a=1.0) [aMAX] 197.484
Medium coef.of friction (acc. Niemann) [mum] 0.042
Wear sliding coef. byNiemann [zetw] 0.713
Power loss from gear load (kW) [PVZ] 0.039
(Meshing efficiency (%) [etaz] 99.653)
Weight (g) [Mass] 3012.64 7412.45
Inertia (System referenced to wheel 1):
calculation without consideration of the exact tooth shape
single gears ((da+df)/2...di) (kgm²) [TraeghMom] 0.00770 0.04981
System ((da+df)/2...di) (kgm²) [TraeghMom] 0.02715

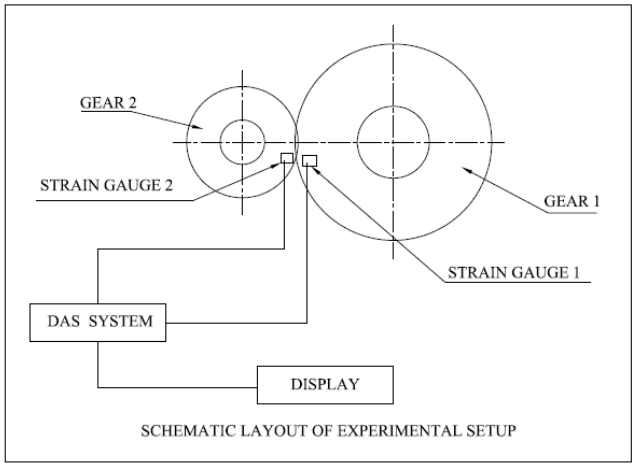
9. MANUFACTURING

Data not available.

REMARKS:
- Specifications with [.e/i] imply: Maximum [e] and Minimul value [i] with consideration of all tolerances
Specifications with [.m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. The maximum and the minimum backlash respective to the max.and min. tolerances are indicated. The calculation is done for the pitch diameter..
- Material factor Yst (analog to ISO6336):
sateff = sat*Yst*KL/KT/KR*Kwb/SF (SF = 1.0)
LACR = Spur gear or helical gear with eps.b< 1.0
PSTC = Point of Single Tooth Contact

IV. EXPERIMENTAL ANALYSIS

Proposed Experimental Setup:



Proposed FEM Analysis:-
Plan of Execution FEM Analysis:-

- Three dimensional models of gear teeth will prepare in 'Creo 2.0' software.
- IGES files will import in 'ANSYS' software 8 Node Brick 45 (SOLID45) elements will be use for meshing.
- Loads will be apply along the top face at equal distances.
- The results of bending stress& Deformation will be obtain by 'ANSYS'.

V.RESULT & DISCUSSION

Effect of load on Maximum Bending Stress
The following graph shows the relationship between the load applied & the maximum bending stress.

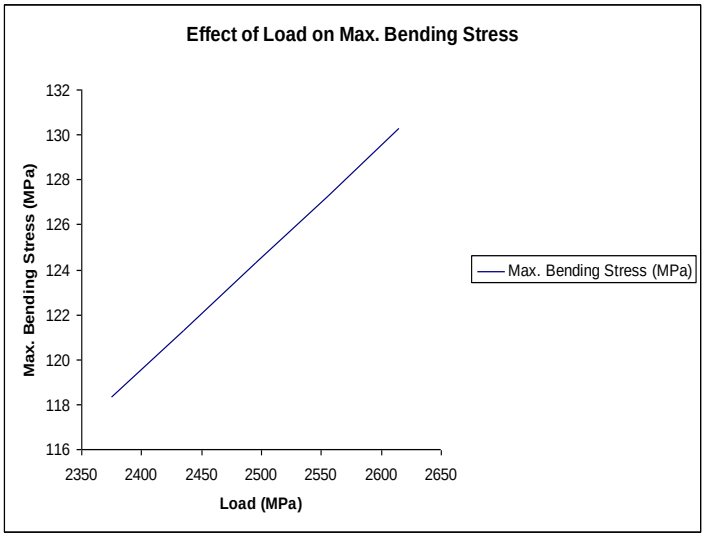


Figure 5.1 Effect of load on Maximum Bending Stress
From the above graph it is seen that as load increases, the maximum banding stress increases. The straight line relationship exists between the two.

Face width in mm	Bending Stress in N/mm^2		
	Theoretical	Using KISSsoft	
	Gear 1 & 2	Gear 1	Gear 2
20	107.912	124.91	125.51
30	71.941	83.27	83.68
40	53.956	62.46	62.76

Table 6.1 Comparison with Theoretical & KISSsoft results

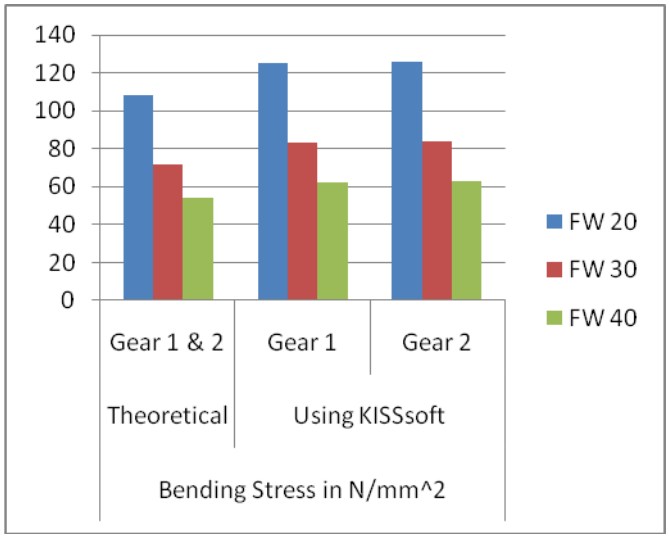


Figure 6.2Comparison with Theoretical & KISSsoft results

VI.CONCLUSION

The spur gears are analyzed for true bending stress.
The conclusions are as below:

- The bending stress is not constant throughout the thickness of the gear.
- The variation in the bending stress is considerable.
- If load per unit thickness is kept constant, Maximum Stress are nearly same for all thicknesses.
- As the loads increases, the bending stress increases. The bending stress are directly proportional to the applied load.
- Bending stress obtained by KISSsoft Software is nearly same as that obtained by Theoretical Analysis.

REFRENCES

1. Zeping Wei, "Stresses & Deformation in Involute Spur Gears by Finite Element Method", Department of Mechanical Engineering, University of Saskatchewan, Saskatoon, Saskatchewan,-Thesis, 2004
2. Lecture Notes, "Spur Gear Design" IIT Madras, India
3. EdoardoConrado and PiermariaDavoli, "The True Bending Stress In Spur Gears", Gear Technology, August 2007
4. Anil Patil, "Analysis of Bending Stress at Root of the Spur Gear by Finite Element Method" Shantiniketan Polytechnic, Maharashtra
5. oseph Edward Shigley, Charles R. Mischke"Mechanical Engineering Design", McGraw-Hill Book Company, New York, 2000, Sixth Edition
6. Ming-Hsiung Chen "Bending Stress in Gear teeth for Variable Surface Pressure Distribution" June 1990.
7. R Prabhushekar "Effect of backup ratio & cutter tip radius on uniform bending strength design of spur gears" 2014

