International Journal of Mechanical Engineering and Robotics Research

ISSN 2278 – 0149 www.ijmerr.com
Vol. 3, No. 4, October 2014

Research Paper

EFFECT OF PRESSURE ANGLE AND BACK UP RATIO ON ASYMMETRIC SPUR GEAR TOOTH BENDING STRESS

Sharatchandrai R Karagi[1]* and Prashant S Patil[1]

*Corresponding Author: Sharatchandrai R Karagi, ✉ sharatrk072@gmail.com

Gearing is one of the most critical components in the mechanical power transmission system and in most industrial rotating machinery. It is possible that gears will be the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In 18th century saw an explosion in metal gearing. Gear design and manufacturing rapidly developed in the 19th century. Now a day, the gear design has become a highly complicated and comprehensive subject. An asymmetric spur gear tooth means that the two profiles (sides) of a gear tooth are functionally different for many gears. The workload on one profile is significantly higher and is applied for longer periods of time than for the opposite one. The design of the asymmetric tooth shape reflects this functional difference. The main objective of the present work is to estimate the stress across the critical section for different backup ratios and the results obtained. Developed programme is used to create a finite element model for asymmetric spur gear tooth to study the effect of bending stress at the critical section for different backup ratios. To study the effect of above parameter ANSYS was used. The rim thickness and the pressure angle was varied and the location and magnitude of the maximum bending stresses were reported and results obtained and compared.

Keywords: Rim thickness, Pressure angle, Asymmetric spur gear tooth, Finite element analysis, Bending stress

INTRODUCTION

Gears have long been widely used in machines of all kinds with increasing requirements in recent times with smaller and lighter designs. Gearing is one of the critical components in a mechanical power transmission system and most industrial rotating machinery. It is possible that gears will be the most effective means of transmitting power in future machines due to their high degree of reliability and compactness.

[1] Department of Mechanical Engineering, BLDEA's College of Engineering & Technology, Bijapur, Karnataka, India.

It is possible that gears will be the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In 18th century saw an explosion in metal gearing. Gear design and manufacturing rapidly developed in the 19th century. Now a day, the gear design has become a highly complicated and comprehensive subject.

Thin rim gears find application in high-power, lightweight aircraft transmissions. Bending stresses in thin rim spur gear tooth fillets and root areas differ from the stresses in solid gears due to rim deformations. Rim thickness is a significant design parameter for these gears. The rim thickness factor is used in the situations in which a gear is made with a rim and spokes rather than a solid disc.

Presently, gears are suffered by backlash undercut and interference. Interference is a serious defect in the involute system of gearing when no of teeth less than the minimum required number of teeth and can be avoided by undercutting the tooth. One major cause of gear failure is fracture at the base of gear tooth due to bending fatigue. The bending strength is influenced by gear size, described by the daimetral pitch; the shape of the tooth, described by the number of teeth on the gear; the location of the highest load, described by the number of teeth on the mating gear. Rim deflections increase the bending stress in tooth fillet and root area. Therefore in aircraft applications rim thickness and allowable stress are optimized to achieve light weight.

In this paper a single teeth gear segment with 25 number of teeth and with module 6, asymmetric spur gear for different pressure angles on drive side and coast side of the gear with different back up ratios were studied. To study the effect of above parameters ANSYS was used. Back up ratios and pressure angles are varied and location and magnitude of maximum bending stress were reported.

ASYMMETRIC SPUR GEAR TOOTH

The two profiles (sides) of a gear tooth are functionally different for many gears. The workload on one profile is significantly higher and is applied for longer periods of time than for the opposite one. The design of the asymmetric tooth shape reflects this functional difference.

Figure 1: Asymmetric Spur Gear

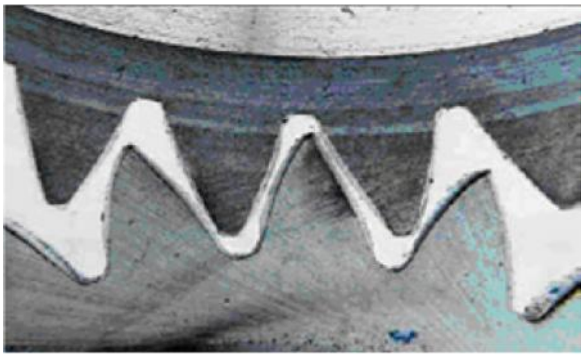
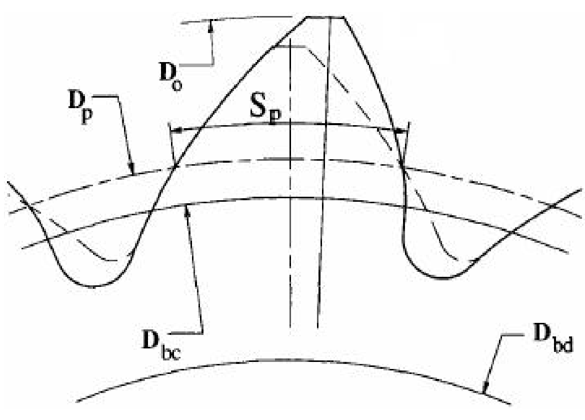


Figure 2: Asymmetric Spur Gear with Different Base Circles



The difference between symmetric and asymmetric tooth is defined by two involutes of two different base circles Dbd and Dbc. The common base tooth thickness does not exist in the asymmetric tooth. The circular distance (tooth thickness) Sp between involute profiles is defined at some reference circle diameter Dp that should be bigger than the largest base diameter.

Asymmetric gears simultaneously allow an increase in the transverse contact ratio and operating pressure angle beyond the conventional gear limits. Asymmetric gear profiles also make it possible to manage tooth stiffness and load sharing while keeping a desirable pressure angle and contact ratio on the drive profiles by changing the coast side profiles. This provides higher load capacity and lower noise and vibration levels compared with conventional symmetric gears.

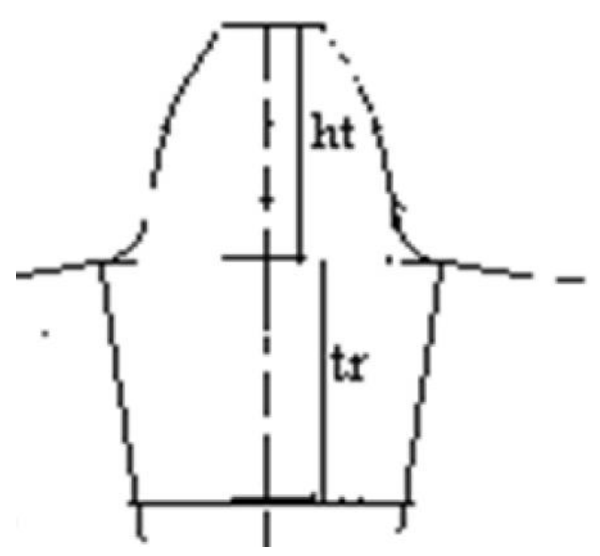
The design intent of asymmetric gear teeth is to improve the performance of the primary contacting profile. The opposite profile is typically unloaded or lightly loaded during relatively short work periods. The degree of asymmetry and drive profile selection for these gears depends on the application.

RIM THICKNESS AND BACKUP RATIO

Thin rim gears find application in high-power, lightweight aircraft transmissions and used in an airplane to meet the lightweight need. In recent years thin rim gear has been a wider application in common machines such as in prototype cars for weight and space reduction. But stress calculation of this gear has not solved so far because this gear has very complicated thin rim structure. In Figure shows a gear segment. The radial distance, between the tip circle and root circle is termed as tooth height (*ht*) and distance between the root circle and inner surface of the rim is called the rim thickness (*tr*). The angle subtended at the g-ear centre by the two radial lines joining lowest point on the root fillet to the gear centre is termed as tooth root angle. The ratio of tooth height (*ht*) to the rim thickness (*tr*) is called backup ratio.

$$' = \frac{tr}{ht}$$

Figure 3: Spur Gear Tooth with Rim Thickness and Tooth Height



PARAMETERS USED FOR INVOLUTE GEAR TOOTH PROFILE GENERATION

With the emergence of computers, engineering modeling and analysis is getting more dependent on computers day by day. Here different parameters listed in table are used to generate gear tooth profile using C Programming.

Equations used to generate spur gear tooth with back up ratio

$$\bar{r}(\text{–}) = \begin{Bmatrix} x(\text{–}) \\ y(\text{–}) \end{Bmatrix}$$

$$X(\text{–}) =$$

$$N\frac{M_n}{2}\left\{\sin\text{–} - \left[\left(\text{–} + \frac{\text{®}}{2N}\right)\cos\text{¤} + \left(\frac{2x}{N}\right)\sin\text{¤}\right]\cos(\text{¤} + \text{–})\right\}$$

$$Y(\text{–}) =$$

$$N\frac{M_n}{2}\left\{\cos\text{–} - \left[\left(\text{–} + \frac{\text{®}}{2N}\right)\cos\text{¤} + \left(\frac{2x}{N}\right)\sin\text{¤}\right]\sin(\text{¤} + \text{–})\right\}$$

$$\text{–}_{\min} \leq \text{–} \leq \text{–}_{\max}$$

$$\text{–}_{\min} - \frac{2}{N}\left[U + (V + X)\cot\text{¤}\right]$$

$$\text{–}_{\max} = \frac{1}{N\cos\text{¤}} \times \sqrt{(2 + N + 2X)^2 - N(\cos\text{¤})^2}$$

$$-\left(1 + \frac{2x}{n}\right)\tan\text{¤} - \frac{\text{®}}{2N}$$

$$U = \frac{\text{®}}{4} + (\sim - \text{¥})\tan\text{¤} + \frac{\text{¥}}{\cos\text{¤}}$$

$$V = \text{¥} - \sim$$

$$\bar{r}(\text{–}) = \begin{Bmatrix} x(\text{–}) \\ y(\text{–}) \end{Bmatrix}$$

$$X(\text{–}) = M_n(P\cos\text{–} + Q\sin\text{–})$$

$$Y(\text{–}) = M_n(-P\sin\text{–} + Q\cos\text{–})$$

$$\text{–}_{\min} \leq \text{–} \leq \text{–}_{\max}$$

$$\text{–}_{\min} = \frac{2}{N}\left[U + (V + X)\cot\text{¤}\right]$$

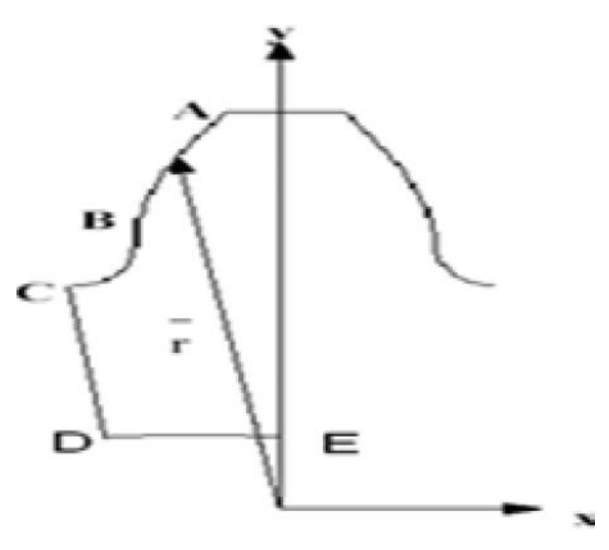
$$\text{–}_{\max} = \frac{2U}{N}$$

$$P = \frac{\text{¥}}{l + \left(u - \frac{n\text{–}}{2}\right)}$$

$$Q = \frac{2\text{¥}}{L} \times \left(\frac{v + k}{2u - n\text{–}}\right) + V + \frac{N}{2} + X$$

$$m_B = \frac{tr}{ht}$$

Figure 4: Generated Involute Profile and Fillet Radius



FINITE ELEMENT ANALYSIS METHOD

Finite element analysis has been carried out for different sets of asymmetric gears as listed in Talbe 1, subjected to load at highest point of single tooth of contact. Key points

Table 1: **Parameters Used for Analysis**

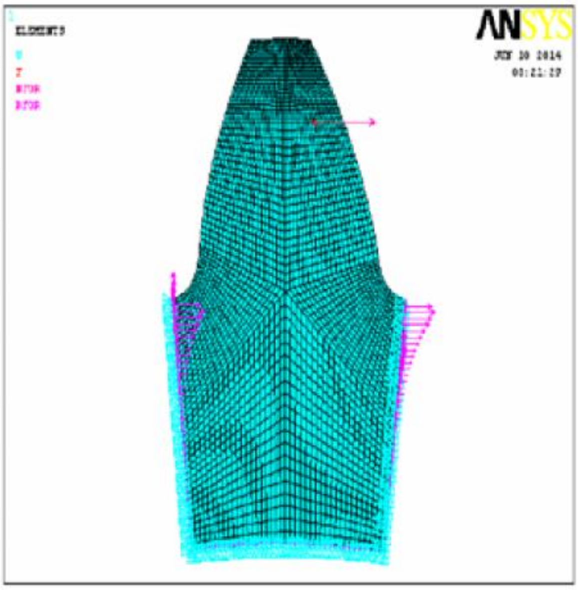
Parameter	Notation	Values Taken
Pressure angle	~	20°, 25°, 30°
Backup ratio		0.6, 0.8, 1.0, 1.2, 1.4, 1.6
No. of teeth	*z*	25 and 47
Profile shift factor	*sf*	0
Module	*m*	6 mm

are generated using c programme for generating involute gear tooth and is used for generating the model and is shown in below Figure 5.

The PLANE182 (plane stress) element is chosen for analyses and material property of steel is used for gear. The first investigation involved a two dimensional plane stress analysis for 6mm module and 20º pressure angles on both sides of the gear and 25 teeth and with back up ratio 0.6. The gear tooth is loaded at HPSTC.and analysis is done.

Similar investigations are done for different pressure angles and back up ratio.

Figure 5: Finite Element Analysis of Single Teeth Gear Segment Subjected to Load and Boundary Conditions

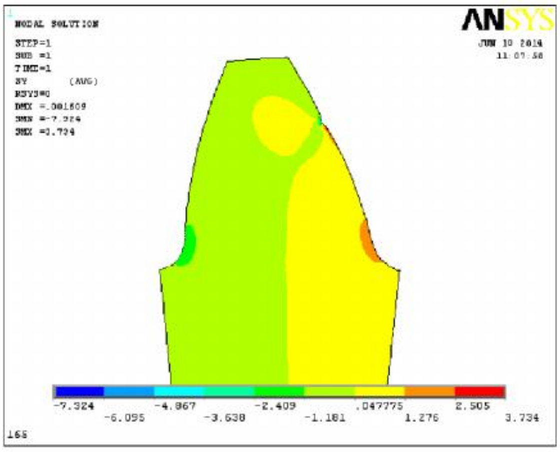


RESULTS

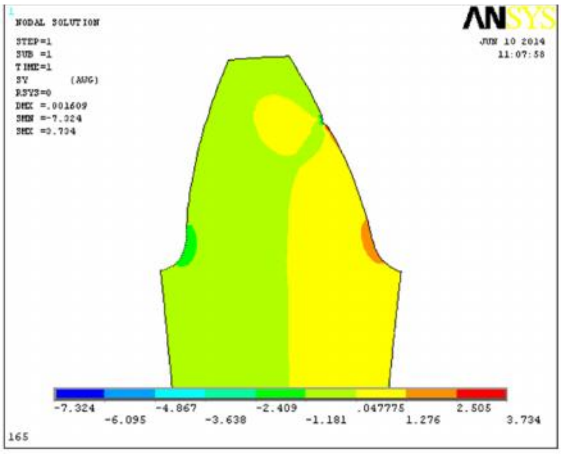
Results mainly contain bending stress at critical section and von mises stress and displacements

Figure 6: Maximum Bending Stress for Different Pressure Angle and Back Up Ratios

For BP 0.6



For BP 0.8



For BP 1.0

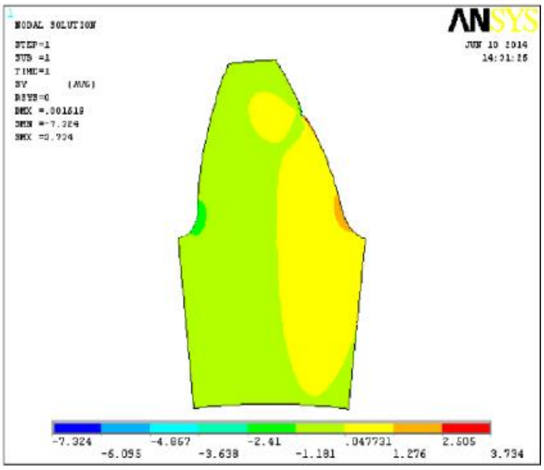
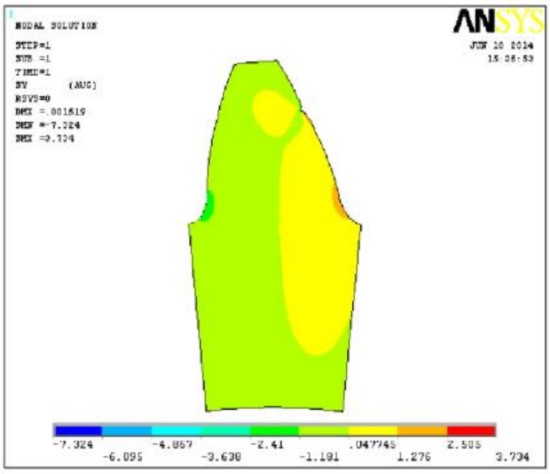
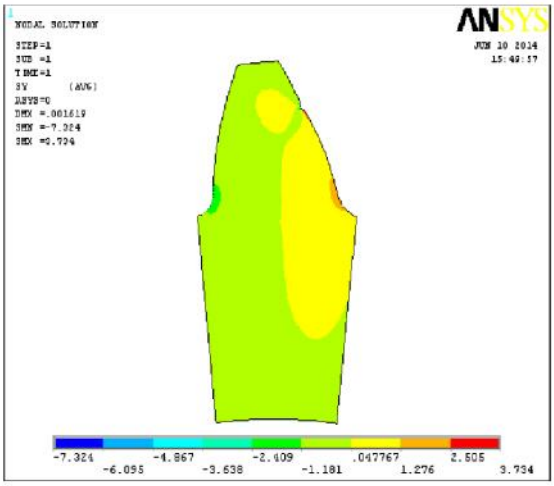


Figure 6 (Cont.)

For BP 1.2



For BP 1.4



For BP 1.6

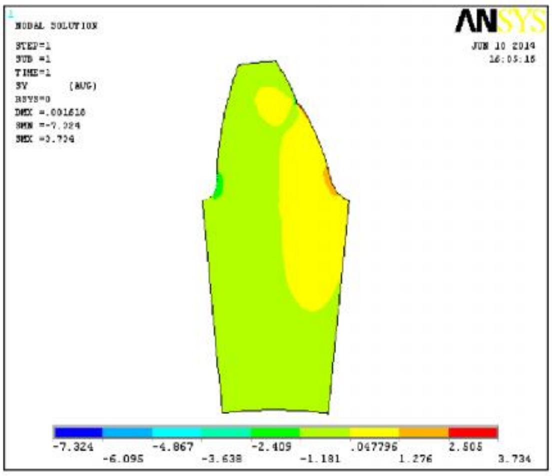


Figure 7: Bending Stress at the Critical Section for Different Back Up Ratio

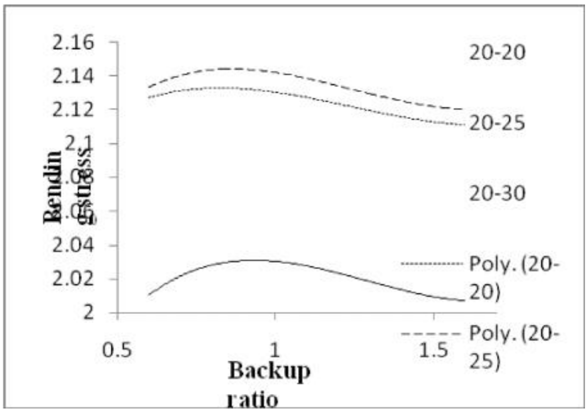
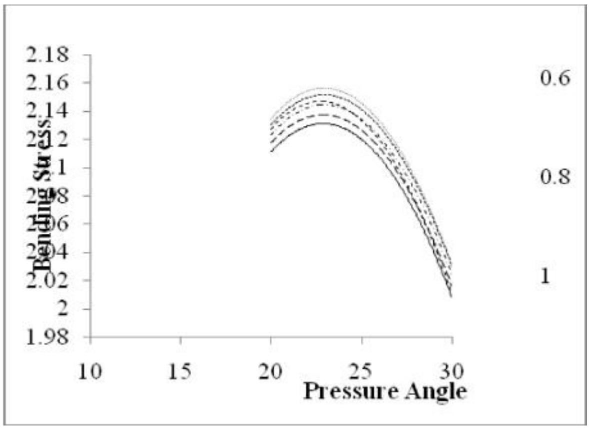


Figure 8: Bending Stress at the Critical Section for Different Pressure Angles



CONCLUSION

From the above results following conclusions are made

- The bending stress obtained at the critical section is minimum for back up ratio 1.6 in asymmetric spur gear tooth.
- For back up ratio 0.6 and 0.7 there is increase in the deflection takes place at gear tooth root which indicates there is a chance of rim fracture.
- The bending stress is maximum for back up ratio more than 1.6.

- These types of gears find application in light weight high power aircraft transmission.

REFERENCES

1. Bibel G D, Reddy S K, Saavage M and Handschuh R E (1991), "Effect of Rim Thickness on Spur Gear Bending Stress", NASA Technical Memorandum.
2. Chang S H, Huston R L and Coy J J (1983), "A Finite Element Stress Analysis of Spur Gears Including Fillet Radii and Rim Thickness Effects", *ASME, Journal of Mechanisms, Transmissions, and Automation in Design*, Vol. 105, pp. 327-330.
3. Chong Tae Hyong, Katayam N and Kubo A (1984), "Tooth Fillet Stresses of Gear with Thin Rim (5th Report, Tooth Fillet and Root Stresses of Internal Spur Gear Supported by Pinned Coupling)", *Bulletin of JSME*, Vol. 27, pp. 815-822.
4. David G Lewicki (1995), "Crack Propagation Studies to Determine Begin or Catastrophic Failure Modes for Aerospace Thin-Rim Gears", May, Department of Mechanical and Aerospace Engineering, Case Western Reserve University.
5. David G Lewicki and Roberto Ballarini (1996), "Effect of Rim Thickness on Gear Crack", NASA Technical Memorandum.
6. David G Lewicki and Roberto Ballarini (1997), "Gear Crack Propagation Investigations", *International Journal of Fatigue*, Vol. 19, No. 10, p. 731.
7. Kramberger J, Sraml M, Potrc I and Flasker J (2004), "Numerical Calculation of Bending Fatigue Life of Thin-Rim Spur Gears", *Engineering Fracture Mechanics*, Vol. 71, Nos. 4-6, pp. 647-656.