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INVESTIGATING AN INNOVATIVE APPROACH TO GENERATE ASYMMETRIC TOOTH PROFILES AND ANALYZING BENDING STRESS

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Abstract

Gears are the mechanical components and are predominate as the most effective means of power transmission in mining equipment which encompasses a wide range of machinery used in the extraction of minerals and resources from the earth. Currently, gears in mining equipment are prone to a different types of failures when they are subjected to impact, shock and fatigue loads. It is noticed from the literature that 34.4% of the failures are due to poor lubrication, 19.6% due to impurity, 17.7% due to fixing errors, 6.9% are due to excess load and 2.8% failures are due to handling errors. Further, more than 1,500 gear failures were examined throughout the investigation, and it was observed that tooth bending fatigue is the common mode, leading to tooth fracture at the root owing to bending stress. Bending stress developed at the tooth root can be minimized by modifying addendum of mating gears or the involute geometry. An additional alteration that is rarely used is to make the gear asymmetric or nonstandard. Therefore, in this research work attempts are made to generate non-standard spur gear tooth profile by varying pressure angle and profile shift using APDL program to estimate bending stress using FE software ANSYS. Bending stress calculated using the Lewis equation and FE analysis were compared to validate the FE procedure. It is noticed that bending stress was reduced to 38 % by increasing the pressure angle from 20^0 to 35^0 without altering other gear parameters. Therefore, the utilization of non-standard asymmetric spur gears offers a potential means to minimize the risk of tooth breakage.

Keywords: ANSYS APDL, Bending Stress, FEA, Asymmetric Spur Gears, Profile Generation.

1. INTRODUCTION

Gears are the most significant mechanical power transmission components in rotational machinery for industry. Gears may become the most efficient mechanism of transmitting power in mining equipment because to their high durability and compactness **[1]**. Gears are classified as Spur, Helical & Bevel gears and are extensively used to transmit power for many engineering applications. Spur gears find extensive use due to their ability to uphold a constant speed and torque while maximizing efficiency and precision **[2]**. At present, gears are susceptible to failures, either during the manufacturing process or while in service. These failures are mainly due to backlash, undercutting and interference **(Figure:1)**.

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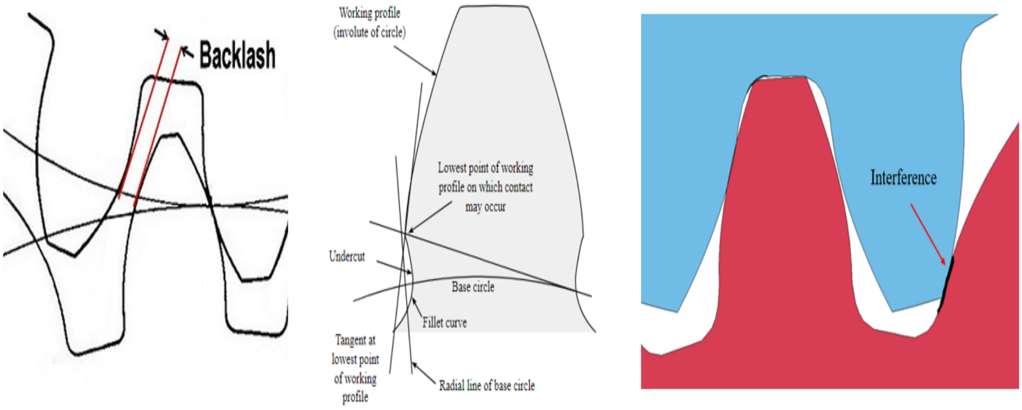

Figure 1: Common defects in gears

Many researchers have tackled these issues and proposed solutions, indicating that interference can be mitigated by minimizing the number of teeth on a pinion, reducing undercutting by employing a larger pressure angle, and preventing backlash by either increasing the addendum of mating gears or modifying the gear tooth geometry **[3]**. This research work introduces an additional modification, suggesting a change in the pressure angle on the drive side or make gears asymmetric **[4]**. In symmetric gears, the tooth profile is uniform on both sides. However, in asymmetric gears, there is a variation in the pressure angle between the coast and drive sides, as illustrated in **Figure 2**.

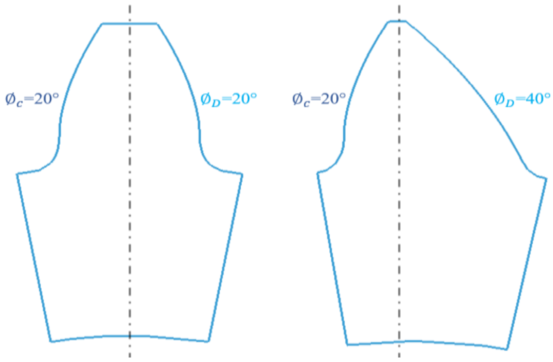

Figure 2: Symmetric and Asymmetric spur gears

It is evident from the literatures that numerous researchers in the past have opted for the involute profile due to its superior performance. Many investigators have put forth their own mathematical models to generate an involute gear tooth profile. **Kapelevich et al. [6]** developed the internal and external geometry of asymmetric spur gears using specially designed rack cutters and are used to conduct vibration and bending stress analyses through a single-stage gear generator. An asymmetric spur gear tooth geometry of an involute and fillet profile on drive and coast side were developed by **Deng et al. [7]** based on the conjugate action of the gear drives by using different rack-cutter parameters.

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Cavdar et al. [8] developed a computer program to generate involute profiles on an asymmetric gear to evaluate contact ratio and bending stress using FEA. **Mallesh et al. [9]** developed a C-Program to generate involute and fillet profiles of symmetric and asymmetric gears to examine the effect of bending stress at critical section by varying the drive side pressure angles and positive profile shift. In the present research work equations developed by the above researchers are utilized to develop APDL (Ansys Parametric Design Language) program to generate spur gear tooth profile to evaluate bending stress in asymmetric gears by changing the profile shift and pressure angle on the drive side. Further, FEA solutions obtained from the simulations are compared with the well-established ISO 6336 and DIN 3990 standards.

2. GEAR TOOTH PROFILE GENERATION

The involute and fillet profiles of the gears developed in this study are derived from tested gears with a gear ratio of 1:1, and the parameters of the test gears are given in **Table: 1.** Profile equations found in the reference **[10, 11]** are used to develop APDL program to generate the coordinates of gear tooth geometry.

Table 1: Test Gear Parameters

Gear type	Standard involute, full depth teeth
Number of teeth on Pinion, z_1	23
Pressure angle, ϕ	20^0
Module, M_n, (mm)	6
Addendum, mm $a = \alpha\ M_n$	$1\ M_n$
Dedendum, mm $b = \beta\ M_n$	$1.25\ M_n$
Face width, mm	15
Tip radius, $r_c = \gamma\ M_n$	$\gamma = 0.25$
Addendum modification coefficient	$X = \frac{e_s}{M_n}$
Cutter offset	e_s

2.1 APDL Program for Gear Tooth Profile Generation

```
/prep7
*SET, pi,3.1415926
*SET, x1,0
*SET, fi, pi/9
*SET, w,180/35
*SET, ti, pi/w
*SET, g1,0.25
*SET, br,1.25
*SET, a1,1
*SET, n,23
```

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```
*SET, m,6
*SET, nu,200
*SET, nu2,120
*SET, p, nu+1
*SET, r,2*nu+nu2
*SET, s,2*(nu+nu2)
*SET, u, -(pi/4+(a1-g1) *tan(fi)+g1/cos(fi))
*SET, v, g1-a1
*SET, thmin, (u+(v+x1)/tan(fi)) *2/n
*SET, thmax, ((2+n+2*x1) **2-(n*cos(fi)) **2) **0.5/(n*cos(fi)) -(1+2*x1/n) *tan(fi)-pi/(2*n)
*SET, inc, (thmax-thmin)/nu
*do, i,1, nu
*SET, th, thmax-inc*(i-1)
*SET, x, (n*m/2) *(sin(th)-((th+pi/(2*n)) *cos(fi)+(2*x1*sin(fi))/n) *cos(th+fi))
*SET, y, (n*m/2) *(cos(th)+((th+pi/(2*n)) *cos(fi)+(2*x1*sin(fi))/n) *sin(th+fi))
k, npt, x, y,
*cfopen, spline, mac
*cfwrite, FLST,3, nu,3
*do, i_,1, nu
*cfwrite, FITEM,3, i_
*enddo
*cfwrite, BSPLIN, P51X
*cfclose
*enddo
spl_ine.mac
*SET, thmax2,2*u/n
*SET, inc1,abs(thmax2-thmin)/nu2
*do, i,1, nu2
*SET, th, thmin+inc1*(i-1)
*SET, labc, (1+4*(((v+x1)/(2*u-n*th)) **2)) **0.5
*SET, pq, (g1/labc) +(u-n*th/2)
*SET, qp,2*(g1/labc) *(v+x1)/(2*u-n*th) +v+(n/2) +x1
```

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```
*SET, x, m*(pq*cos(th)+qp*sin(th))
*SET, y, m*(-pq*sin(th)+qp*cos(th))
k, npt, x, y,
*cfopen, spl_ine, mac
*cfwrite, FLST,3, (nu2+1),3
*do, i_, (nu), (nu+nu2)
*cfwrite, FITEM,3, i_
*enddo
*cfwrite, BSPLIN, P51X
*cfclose
*enddo
spl_ine.mac
*SET, u, -(pi/4+(a1-g1) *tan(ti)+g1/cos(ti))
*SET, v, g1-a1
*SET, thmin, (u+(v+x1)/tan(ti)) *2/n
*SET, thmax, ((2+n+2*x1) **2-(n*cos(ti)) **2) **0.5/(n*cos(ti)) -(1+2*x1/n) *tan(ti)-pi/(2*n)
*SET, inc, (thmax-thmin)/nu
*do, i,1, nu
*SET, th, thmax-inc*(i-1)
*SET, x, -(n*m/2) *(sin(th)-((th+pi/(2*n)) *cos(ti)+(2*x1*sin(ti))/n) *cos(th+ti))
*SET, y, (n*m/2) *(cos(th)+((th+pi/(2*n)) *cos(ti)+(2*x1*sin(ti))/n) *sin(th+ti))
k, npt, x, y,
*cfopen, spl_ine, mac
*cfwrite, FLST,3, nu,3
*do, i_, (nu+nu2+1), (2*nu+nu2)
*cfwrite, FITEM,3, i_
*enddo
*cfwrite, BSPLIN, P51X
*cfclose
*enddo
spl_ine.mac
*SET, thmax2,2*u/n
```

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```
*SET, inc1, abs(thmax2-thmin)/nu2
*do, i,1, nu2
*SET, th, thmin+inc1*(i-1)
*SET, labc, (1+4*(((v+x1)/(2*u-n*th)) **2)) **0.5
*SET, pq, (g1/labc) +(u-n*th/2)
*SET, qp,2*(g1/labc) *(v+x1)/(2*u-n*th) +v+(n/2) +x1
*SET, x, -m*(pq*cos(th)+qp*sin(th))
*SET, y, m*(-pq*sin(th)+qp*cos(th))
k, npt, x, y,
*cfopen, spl_ine, mac
*cfwrite, FLST,3, (nu2+1),3
*do, i_, (r), (s)
*cfwrite, FITEM,3, i_
*enddo
*cfwrite, BSPLIN, P51X
*cfclose
*enddo
spl_ine.mac
k, npt,0,0
*SET, pt0, (n*m/2) *(cos(thmax)+((thmax+pi/(2*n)) *cos(fi)+(2*x1*sin(fi))/n)
*sin(thmax+fi))
*SET, pt2, m*(-pq*sin(thmax2) +qp*cos(thmax2))
*SET, ptd, pt2-br*(pt0-pt2)
*SET, radmin,((pi/2)-(pi/n))
*SET, radmax, ((pi/2) +(pi/n))
*SET, incr,(radmax-radmin)/3
*do, ra,radmin,(radmax)
*SET, radi,radmin+incr*(ra+4/3)
*SET, x, ptd*cos(radi)
*SET, y, ptd*sin(radi)
k, npt, x, y
*enddo
*SET, pt0, (n*m/2) *(cos(thmax)+((thmax+pi/(2*n)) *cos(fi)+(2*x1*sin(fi))/n)
```

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```
*sin(thmax+fi))
*SET, pt2, m*(-pq*sin(thmax2) +qp*cos(thmax2))
*SET, ptd,pt2-br*(pt0-pt2)
*SET, radmin, ((pi/2)-(pi/n))
*SET, radmax, ((pi/2) +(pi/n))
*SET, incr, (radmax-radmin)/3
*do, ra, radmin, (radmax)
*SET, radi, radmin+incr*(ra+4/3)
*SET, x, -ptd*cos(radi)
*SET, y, ptd*sin(radi)
k, npt, x, y
*enddo
*SET, radii, (radmax+radmin)/2
*SET, x, ptd*cos(radii)
*SET, y, ptd*sin(radii)
k, npt, x, y
l,1, (nu+nu2+1)
l, (2*(nu+nu2) +2), (nu+nu2)
l, (2*(nu+nu2) +3), (2*(nu+nu2))
BSPLIN, (2*(nu+nu2) +2), (2*(nu+nu2) +4), (2*(nu+nu2) +3)
al,1,2,3,4,5,6,7,8
```

finish

The developed APDL codes are input into the ANSYS software's input window to create key points, splines, lines, and the area corresponding to a single gear tooth. Further, single gear tooth model is imported into a solid edge ST6 CAD software to create three gear tooth and whole gear geometry of symmetric and asymmetric gears as shown in **Figure 3**. In addition, the coordinates of involute and fillet profiles developed using the APDL program are compared to assess the consistency of APDL with published results as shown in **Figure 4** and found that they are in good agreement **[9]**. Later, 2-D and 3-D profiles of symmetric and asymmetric spur gears are used to conduct Finite Element Analysis (FEA) to estimate the bending stress.

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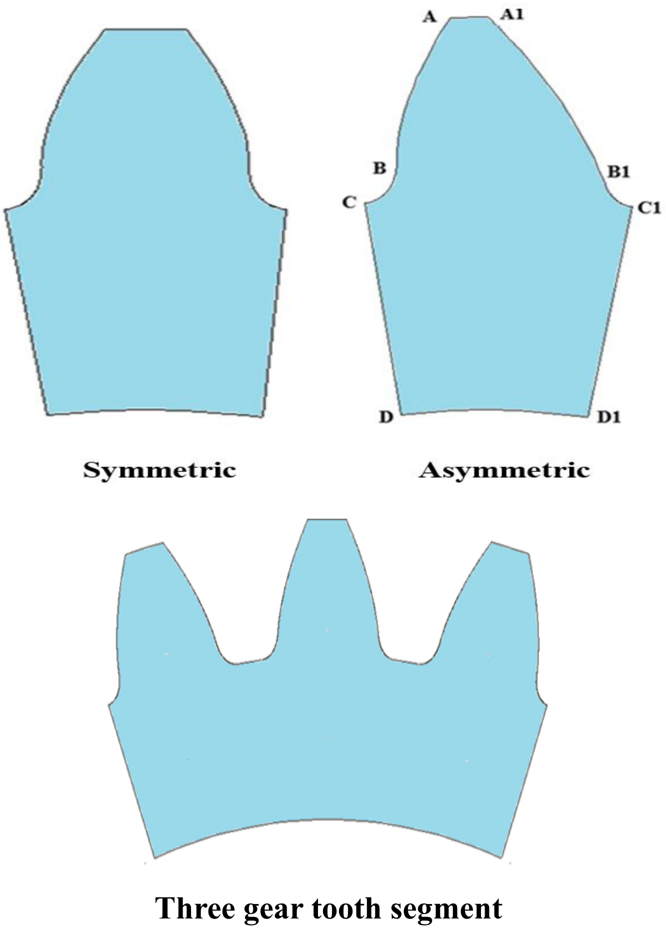

Figure 3: Profiles of gear tooth

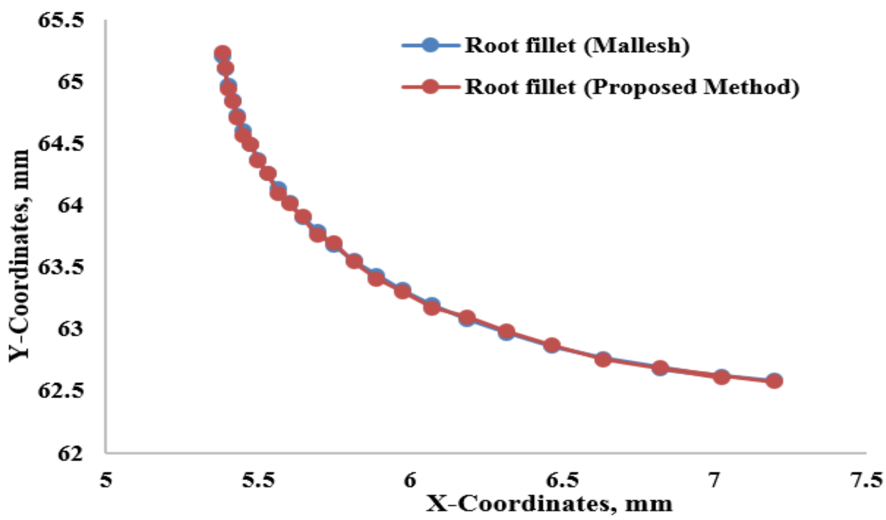

Figure 4: Comparison of Coordinates of root fillet

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3. FEM OF SPUR GEAR TOOTH

The gear tooth geometry (***. iges file**) created using a solid edge ST6 CAD software, is imported into the ANSYS software. Subsequently, the geometric model is converted into a FE model using 8-noded quadrilateral elements with mapped mesh shown in **Figure 5**. As evident from the prior references, performing an analysis for the entire gear body is both tedious and time-consuming. To address this challenge, **Celik et al. [13]** proposed to use three or five gear tooth segments for bending stress analysis which yields same results as that of whole body. Hence, in this research work only three gear tooth segments are used to estimate the bending stress in non-standard gears. Further, a series of FEA were conducted by varying the pressure angle on drive side. A tangential tooth load is applied at Highest Point of Single Tooth Contact (HPSTC) and radial lines AB, CD and the rim surface BC are fixed as shown in **Figure 5.** For the Finite Element (FE) analysis, a plane stress with a thickness of 5 mm is utilized, and each node is assigned two degrees of freedom. The FE analysis is conducted using a Sparse solver **[14]**. Each model comprises 18,000 elements, 54,901 nodes, 600 constrained degrees of freedom, and 1,09,202 active degrees of freedom

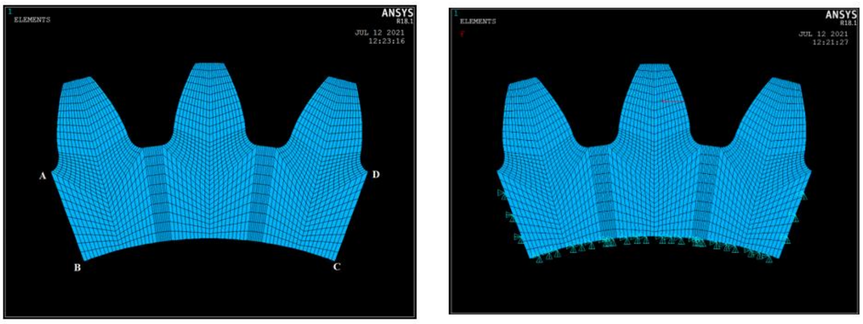

Figure 5: FEA model with boundary conditions

4. RESULTS AND DISCUSSIONS

4.1. Effect of Pressure Angle Modification on Spur Gear Tooth

The effect of changing the pressure angle on spur gear tooth performance, as well as its relationship to other factors, are explored in the following sections.

4.1.1 Critical Section Thickness

Computing bending stress, in gears, the critical section thickness of a gear tooth plays a crucial role in analyzing the gear's structural integrity. The relationship between the critical section thickness and pressure angle is depicted in **Figure 6**. It is evident that as the pressure angle on the driving side increases, the tooth thickness at the critical section also increases. This increase in critical section thickness contributes to enhancing the gear's load-bearing capacity.

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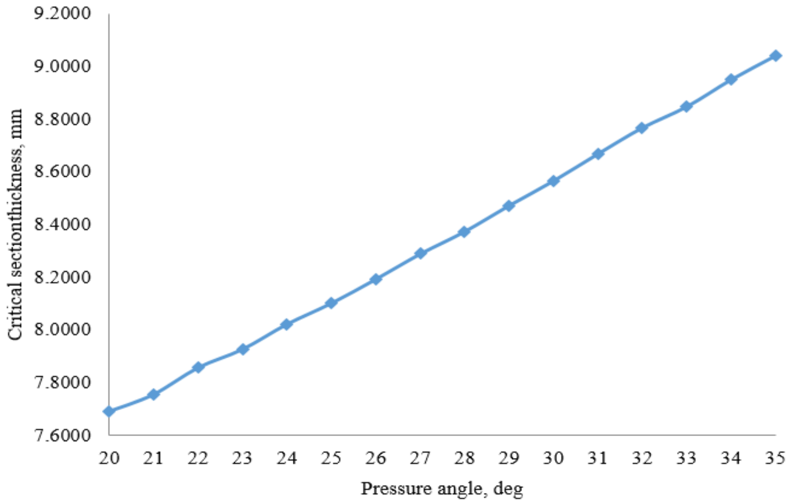

Figure 6: Variation of critical section thickness

4.1.2 Position of HPSTC

Table.2 depicts the changes in the Highest Point of Single Tooth Contact (HPSTC) in gears, which represents the position on the gear tooth profile where the contact between two mating gears achieves its maximum height during meshing. Observations from the table reveal that an increase in pressure angle corresponds to an increase in load angle, causing the load to shift towards the tooth tip. This shift mitigates stress concentration at the root fillet. Consequently, the horizontal component of the load at HPSTC increases with the rise in load angle.

Table 2: HPSTC Position in asymmetric spur gears

Test No.	Pressure angle Coast /Drive side	Distance between HPSTC and gear tooth center (mm)
1.	$20^0/20^0$	75.806
2.	$20^0/25^0$	76.784
3.	$20^0/30^0$	77.611
4.	$20^0/35^0$	78.212

4.1.3 Tooth Thickness at The Addendum Circle

Increasing the pressure angle on the driving side leads to a decrease in tooth thickness on the addendum circle, and this reduction is constrained to 0.2 times the module i.e. $0.2M_n$**[8]**. For a 6 mm module gear, the tooth thickness at the addendum is 1.2 mm. The pressure angle on the drive side can be increased until the tooth thickness at the addendum reaches 1.2 mm. Further, observations from **Figure 7** indicate that for a 4 mm module asymmetric spur gear tooth, the pressure angle on the drive side can be increased up to 40^0. However, a further increase in the pressure angle on the drive side reveals that the involute profiles of the gear tooth intersect, signifying a flipping of the tooth in a real-case scenario as shown in **Figure 8**.

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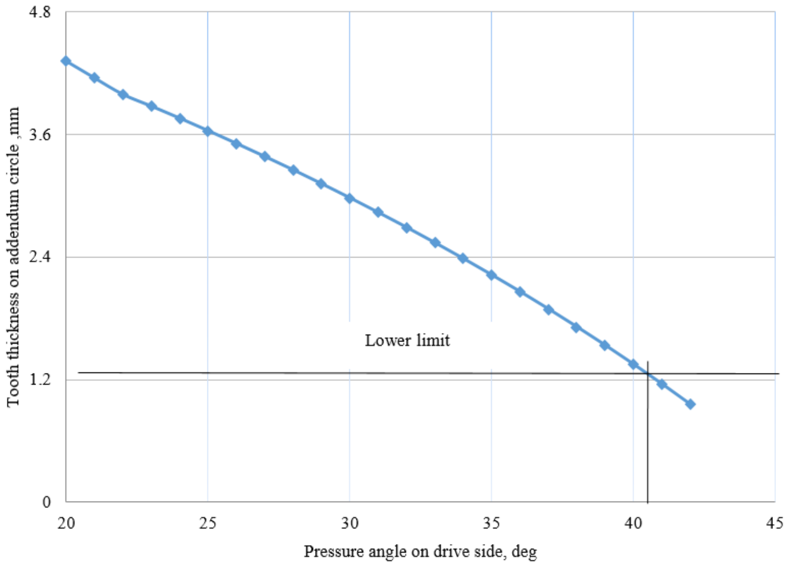

Figure 7: Tooth thickness at the addendum

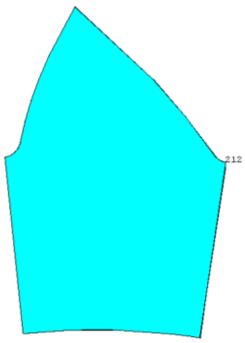

Figure 8: Pointed tooth

4.1.4 Profile Shift

Profile shift in gears refers to the intentional modification of the tooth profile by shifting it either closer to the center or away from the center of the gear. This adjustment is made to achieve specific design goals related to load distribution, gear strength, and noise reduction. Positive profile shift is often used to enhance the load-carrying capacity of the gear and improve its strength. It is noticed from the **Figure 9** that as the pressure angle on the driving side rises, the tooth thickness at the critical section increases with the use of positive profile shifted gears by improving the structural integrity of gears. Further, it is noticed that the gear tooth profile is influenced by change in the pressure angle, module and number of teeth.

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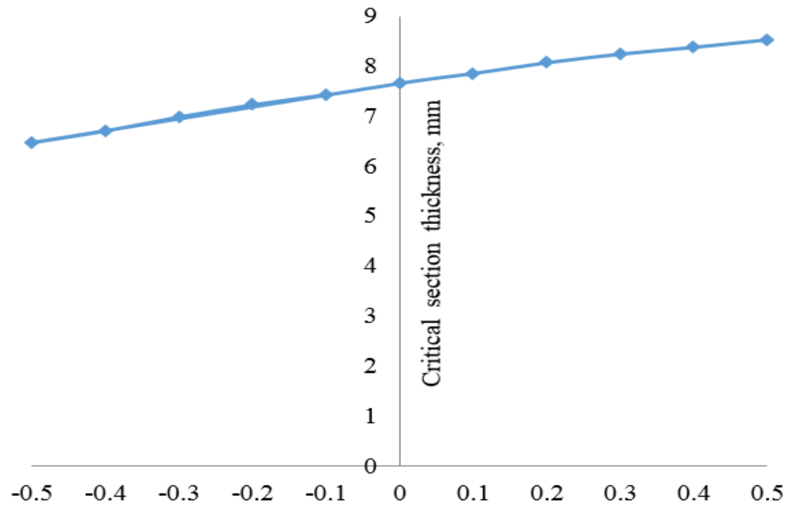

Figure 9: Effect of profile shift on Critical thickness

4.2. Gear Tooth Stresses

Stresses in gears are predominantly influenced by tooth geometry, load, and the pressure angle on the drive side. It has been observed that there is a significant decrease in bending stress at the critical point. As depicted in **Figure 10** and detailed in **Table 3**, the bending stress in both symmetric and asymmetric spur gears was examined. The results indicate that increasing the pressure angle on drive side from 20^0 to 40^0, without altering other gear parameters, led to a notable reduction in bending stress, amounting to 42.14%.

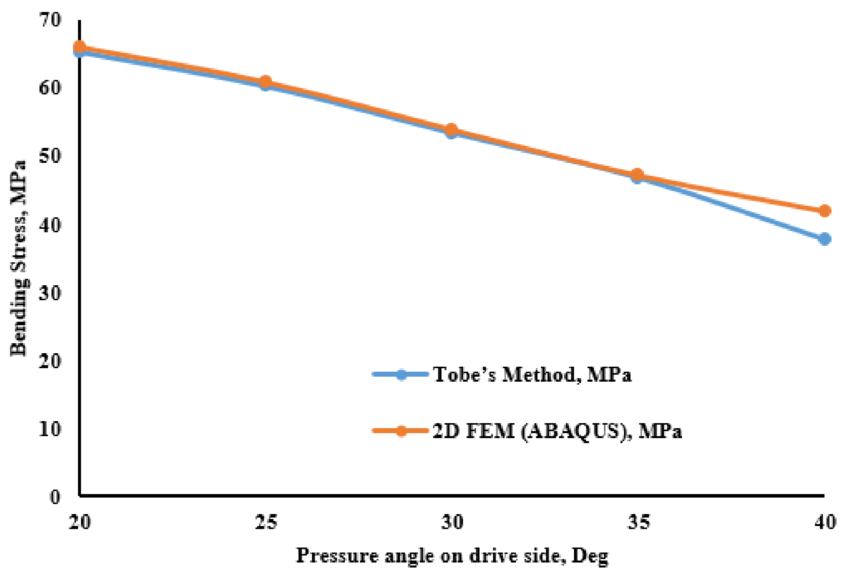

Figure 10: Comparison of bending stress

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Table 3: Static Bending for spur gear tooth

Pressure angle, Deg	Tobe's Method, MPa	2D FEM, MPa
20^0-20^0	65.32	66.06
20^0-25^0	60.31	60.91
20^0-30^0	53.45	53.89
20^0-35^0	46.86	47.14
20^0-40^0	37.79	41.89

It is evident from AGMA and Hofer's theory that maximum and minimum bending stresses are concentrated at the tooth root of a symmetric gears and non-standard gears as shown in **Figure 11**.

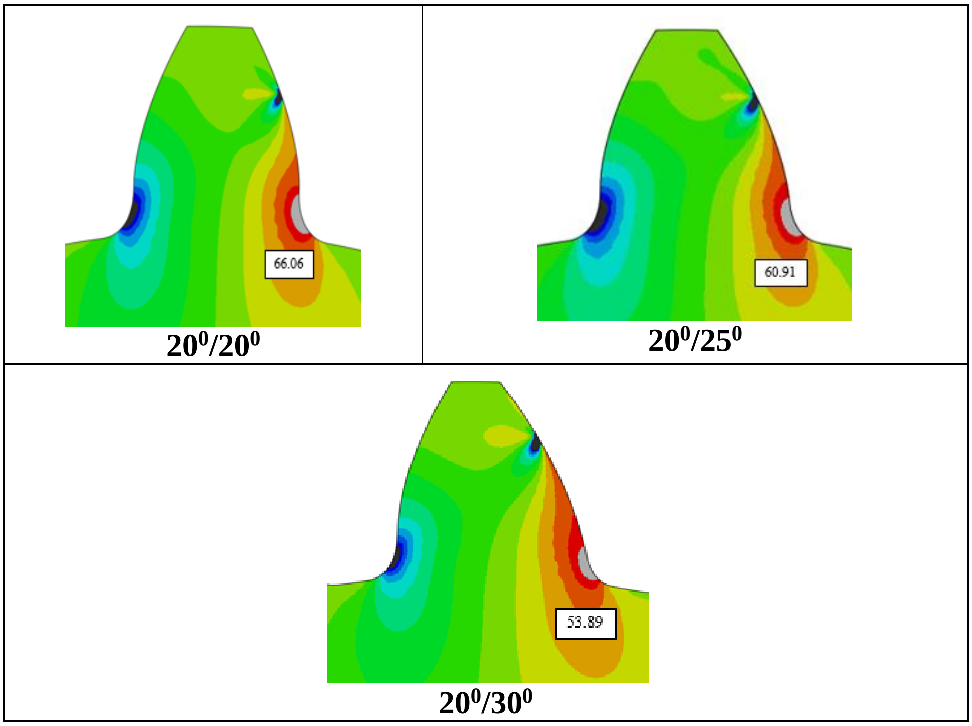

Figure 11: Bending Stress Contours

4.3. Effect Profile Shift on Gear Tooth Stresses

Profile shift is a widely adopted technique in gear design that allows for achieving nonstandard shaft distances, enhancing load-bearing capacity, and preventing undercut in gears with a minimal number of teeth. It is noticed from the **Figure 12** that bending stresses decrease with increases pressure angle and profile shift for a given pressure angle.

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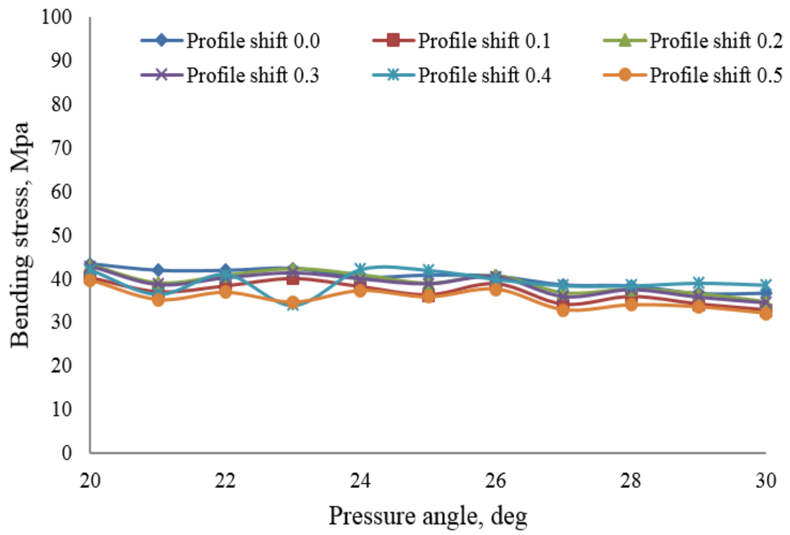

Figure 12: Bending stress for different profile shifts

5. CONCLUSIONS

- An algorithm was developed and implemented in ANSYS APDL to generate gear tooth profile.
- FEA of symmetric and symmetric spur gear tooth were done by FEA software ANSYS 11.0.
- Increasing in pressure angle bending stress reduced to 42.41%
- Using the positive profile shift of 0.25 and $20^0/30^0$ asymmetric gears for power transmission in mining gears reduces the bending stress by 12%.
- Combined effect of pressure angle and profile shift in gears bending stress reduced to 32% without changing the load and material

References

1) UMEZAWA K. Recent Trends in Gearing Technology. JSME international journal. Ser. 3, Vibration, control engineering, engineering for industry. 1988 Jun 15;31(2):357-62.

2) Osakue EE. Simplified Spur Gear Design. In ASME International Mechanical Engineering Congress and Exposition 2016 Nov 11 (Vol. 50657, p. V011T15A018). American Society of Mechanical Engineers.

3) Becker WT, Shipley RJ. Failure analysis and prevention. (No Title). 2002.

4) Colbourne JR. The geometry of involute gears. Springer Science & Business Media; 2012 Dec 6.

5) Richard GB. Shigley's Mechanical Engineering Design. McGraw-Hill Education; 2019.

6) Kapelevich A. Geometry and design of involute spur gears with asymmetric teeth. Mechanism and Machine theory. 2000 Jan 1;35(1):117-30.

7) Deng X, Hua L, Han X. Research on the design and modification of asymmetric spur gear. Mathematical Problems in Engineering

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ISSN 1533-9211

DOI: 10.5281/zenodo.10604810

8) Cavdar K, Karpat F, Babalik F C. Computer aided analysis of bending strength of involute spur gears with asymmetric profile.

9) Mallesh G, Math VB, Ashwij PS, Shanbhag R. Effect of tooth profile modification in asymmetric spur gear tooth bending stress by finite element analysis. In14th National Conference on Machines and Mechanisms (NaCoMM09), NIT, Durgapur, India 2009 Dec 17 (pp. 17-18).

10) Litvin FL, Fuentes A. Gear geometry and applied theory. Cambridge university press; 2004 Sep 6.

11) Litvin FL. Development of gear technology and theory of gearing. National Aeronautics and Space Administration, Lewis Research Center; 1997.

12) Singh, Vedang, and S. Senthilvelan. "Computer aided design of asymmetric gear." stress 11 (2007): 12

13) Celik M. Comparison of three teeth and whole body models in spur gear analysis. Mechanism and machine theory. 1999 Nov 1;34(8):1227-35.

14) Reddy JN. An Introduction to Nonlinear Finite Element Analysis Second Edition: with applications to heat transfer, fluid mechanics, and solid mechanics. OUP Oxford; 2014 Oct 24.

15) Reddy JN. Introduction to the finite element method. McGraw-Hill Education; 2019.

16) Dhillon, B. S., and O. C. Anude. "Mining equipment reliability: A review." Microelectronics Reliability 32, no. 8 (1992): 1137-1156.

17) Haque, Nawshad, Anthony Hughes, Seng Lim, and Chris Vernon. "Rare earth elements: Overview of mining, mineralogy, uses, sustainability and environmental impact." Resources 3, no. 4 (2014): 614-635.