Journal of Mines, Metals and Fuels, *71(12A) : 348-354; 2023. DOI: 10.18311/jmmf/2023/44017*

INFORMATICS JOURNALS

Print ISSN : 0022-2755

Journal of Mines, Metals and Fuels

Contents available at: *www.informaticsjournals.com/index.php/jmmf*

Finite Element Analyses of Non-Standard Spur Gears

P. Srinivasa[1*], C. Shashishekar[1] and G. Mallesh[2]

[1]Department of Mechanical Engineering, Siddaganga Institute of Technology, Tumkuru – 572103, Karnataka, India; psrinivasa87@gmail.com

[2]Department of Mechanical Engineering, Sri Jayachamarajendra College of Engineering, JSS S&T University, Mysuru – 570006, Karnataka, India

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 c 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 Matlab 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 42.14% by increasing the pressure angle from 200 to 400 without altering other gear parameters. Hence, the possibility of tooth break can be minimized with the use of non-standard spur gears.

Keywords: *ANSYS Software, Bending Stress, Finite Element Analysis, Non-Standard Gears, Profile Generation*

1.0 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]. The position of the shaft axes, peripheral velocity, gear type, and tooth position on the gear surface are often used to classify these gears. Spur gear, helical gear, bevel gear, worm and worm wheel, rack and pinion are extensively used to transmit motion and power for many engineering applications. Among these classification, Spur gears are widely utilized because they maintain constant speed and torque with the maximum efficiency and precision[2]. Currently, gears are prone to a different types of failures either during its production or while it's in operation. These failures are mainly due to backlash, undercutting and interference (Figure 1).

These issues are addressed by many researchers and suggested that interference can be avoided by using minimum number of teeth on a pinion, undercutting

*Author for correspondence

can be minimized by using a larger pressure angle and backlash can be avoided by increasing the addendum of mating gears or by modifying the gear tooth geometry are the established techniques in the sophisticated gear design process[3]. The nomenclature describing these types of gear modifications are quite complicated. Hence, another alteration rarely used to make the gears non-standard or asymmetric[4]. In symmetric gears, tooth profile is same on either side and it performs well in either direction. It is observed that gears are significantly loaded for a longer period in drive side when compared to the coast side[5]. Therefore, it is essential to make the gears non-standard or asymmetric with different pressure angles on drive and coast side as shown in Figure 2.

Numerous gear tooth forms have been developed by many researchers in the past. The involute profile is the most commonly used tooth form because of its better performances. However, due to the complexity involved in the gearing and its applications many investigators proposed their own mathematical models to generate an involute gear tooth profile. Kapelevich[6] developed an internal and external geometry of asymmetric spur gears using generated rack cutters to carry out vibration and bending stress analysis by single stage gear generator. Deng[7] developed an asymmetric spur gear tooth geometry using different rack-cutters for involute and fillet profile on driving side and coast side on the basis of conjugate action of the gear drives. Cavdar[8] developed a computer program to generate involute asymmetric gear tooth profile and to evaluate contact ratio and bending stress using finite element analysis. Mallesh[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 work equations developed by the above researchers are used to develop Matlab program to generate spur gear tooth profile to evaluate bending stresses in asymmetric spur gear tooth by changing the profile shift and pressure angle on the drive side. ISO 6336 and DIN 3990 standards are used to compare the solutions obtained from FEA.

2.0 Gear Tooth Profile Generation

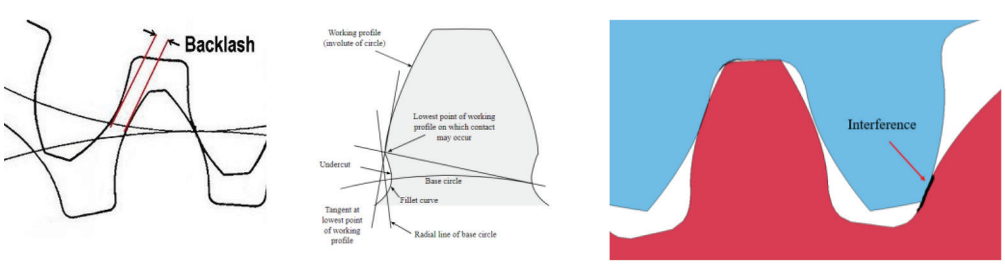
The study utilizes test gear parameters to design involute and fillet profiles for non-standard spur gears with a gear ratio of 1:1 given in the Table 1. The tooth involute and fillet profile equations[10,11] are utilized to generate the coordinates of gear tooth geometry.

The Graphics User Interface (GUI) to generate involute and fillet points of gear tooth profile is as shown in Figure 3 along with various input parameters. The output of the program is a key points of single, three gear tooth segment and complete gear as shown in Figure 4.

2.1 Validation of Key Points Generated using Matlab

A set of gear tooth parameters listed in the Table 1 are used to calculate the coordinates of the involute in the present research work. Figure 5 shows the coordinates of the involute computed by Vedang singh[12] and Mallesh[9] for the same data, it is noticed that the asymmetric gear points generated from the program are matches with the results of Singh and Mallesh.

Further, key points generated using Matlab program are converted into APDL codes to generate lines, splines and area to create 2-Dimensional model of symmetric and non-standard gear tooth by varying pressure angle and profile shift as shown in Figure 6.



Backlash   Undercutting   Interference

Figure 1: Common defects in gears.

3.0 FEM of Spur Gear Tooth

In this stage, gear tooth geometry created using an APDL program is imported into the ANSYS software by *. iges file format and converted into a FE model using 8-noded quadrilateral elements with mapped mesh shown in Figure 7. It is observed that conducting the FEA for the entire body is time consuming and tedious and it is suggested by the Celik[13] that the use three and five gear tooth segments for bending stress analysis yields same results as that of whole body. Hence, in this research work only three gear tooth segments are used. A series of FE analyses were conducted on symmetric

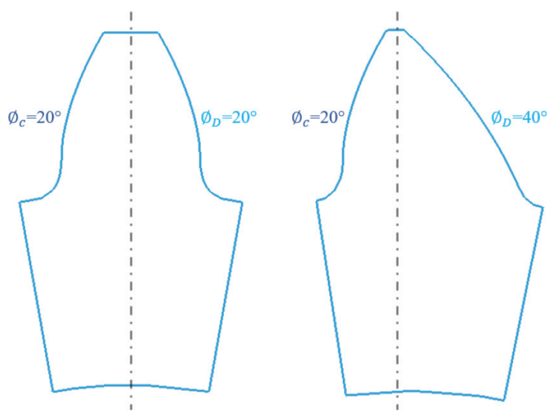


Figure 2: Symmetric and asymmetric spur gears.

Table 1: Test gear parameters

Gear type	Standard involute, full depth teeth
z_1 (Gear tooth on pinion)	23
Pressure angle	20^0
Module, M_n, (mm)	6
Addendum, a (mm)	1 M_n
Dedendum, b (mm)	1.25 M_n
Face width, mm	15

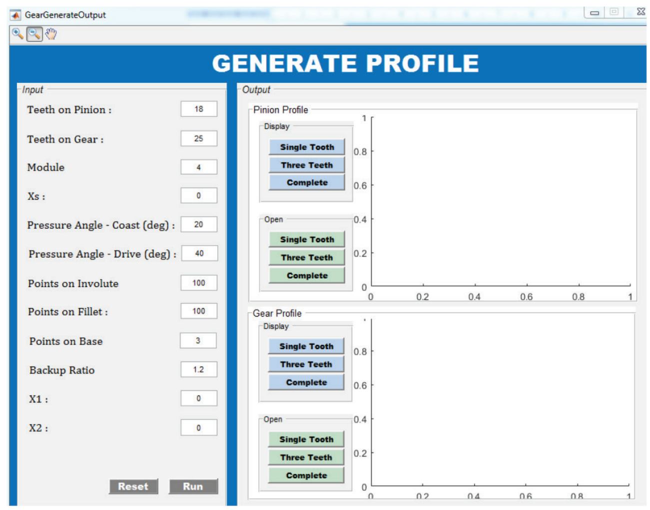


Figure 3: Matlab GUI for gear profile generation.

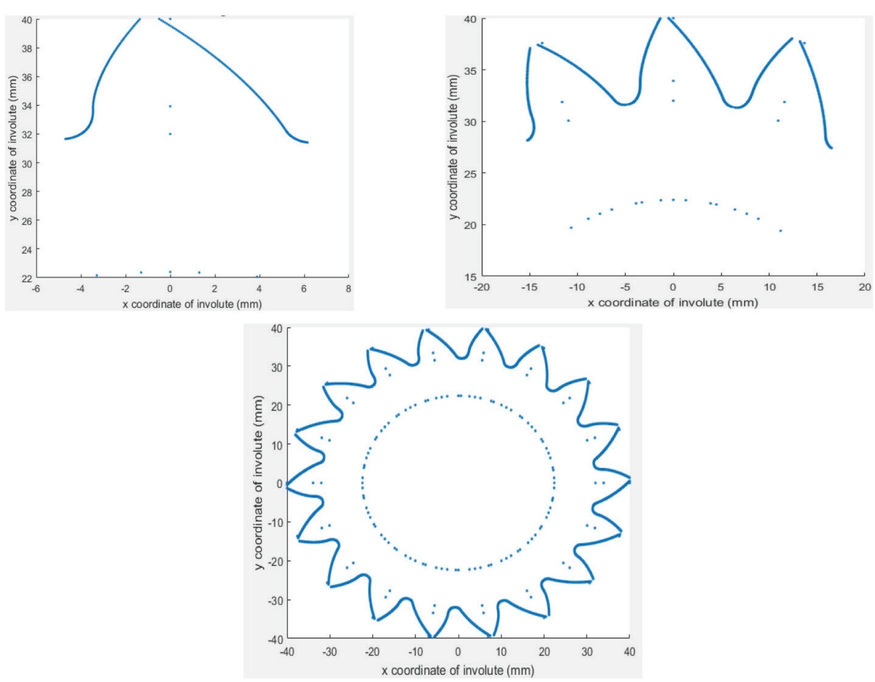


Figure 4: Coordinates of Non-standard gears.

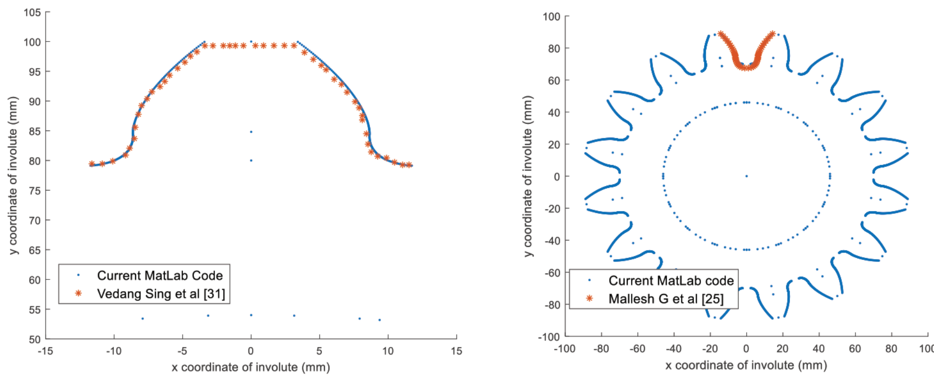
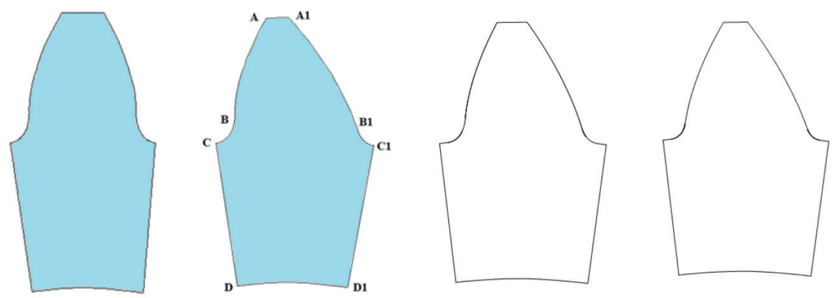


Figure 5: Comparison of involute and root fillet coordinates.



Symmetric gear Asymmetric gear Profile shifted Asymmetric spur gears

Figure 6: 2-D Gear tooth model for different pressure angle and profile shift.

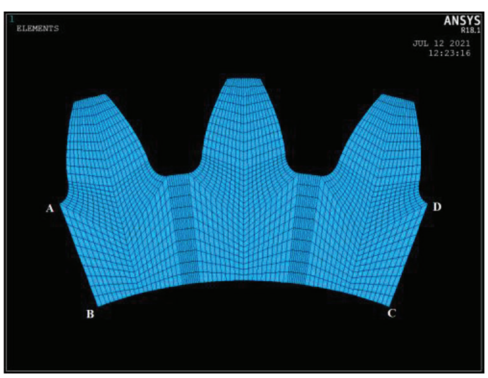


Figure 7: FEA model of spur gear tooth.

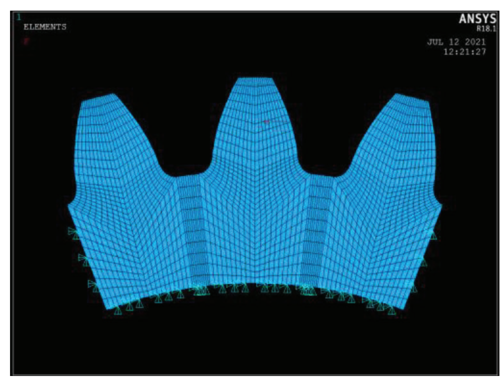


Figure 8: Boundary conditions for FE analyses.

and non-standard spur gears by varying the pressure angle and profile shift. A tangential tooth load is applied at Highest Point of Single Tooth Contact (HPSTC) and two radial lines AB, CD and the rim surface BC are fixed as shown in Figure 8.

Each element's node has two degrees of freedom, allowing it to move in both the X and Y, plane stress condition with 5 mm thickness is used to carry out FE analysis[14]. Each model has 18,000 elements, 54,901 nodes, 600 constrained degrees of freedom and 1,09,202 active degrees of freedom. Sparse solver was used to evaluate the bending stress.

4.0 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, the critical section thickness of a gear tooth is a crucial factor to consider. The relationship between critical section thickness and pressure angle is represented in Figure 9. As the pressure angle on the driving side rises, the tooth thickness at the critical section increases, increasing load bearing capacity.

4.1.2 Position of HPSTC

Table 2 illustrates the variation of HPSTC position with varying pressure angle. It is noted that increasing pressure angle increases load angle and position of the load shifting towards the tooth, weaning the stress concentration from the root fillet. Hence, the horizontal component of the load at HPSTC increases.

4.1.3 Tooth Thickness at The Addendum Circle

By increasing pressure angle on the driving side, the tooth thickness on the addendum circle depreciates. Studies on a 6 mm module gear tooth segment revealed that when the matching tooth thickness on the addendum circle increases, an increase in the pressure angle is restricted to 1.2[8] i.e., The studies on 4 mm module gear tooth segments with varying number of teeth supplemented the above interpretation as shown in Figure 10.

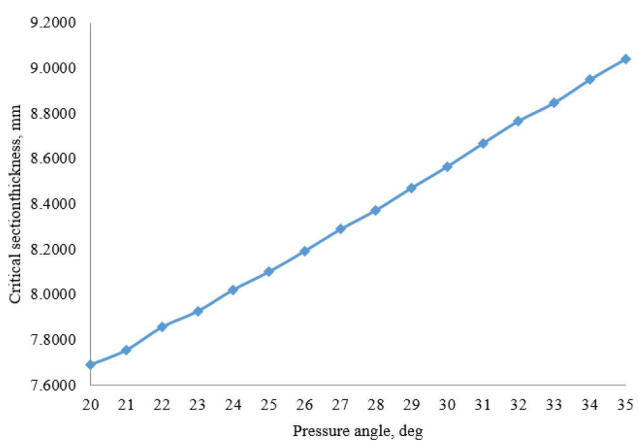


Figure 9: Variation of critical section thickness.

Table 2: HPSTC position of asymmetric spur gear tooth

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

4.1.4 Profile Shift

As the pressure angle on the driving side rises, the tooth thickness on the addendum circle decreases for gears with increasing profile shift Figure 11. The tooth thickness on the addendum circle also reduces drastically in gears with increasing profile shift and pressure angle. However, the increase in critical section thickness is observed for these gears. When the pressure angle on the drive side was increased up to 45^0 in case of 4 mm module zero profile shift gears, it is observed that the involute profiles of the gear tooth intersect representing the flipping of tooth in real case scenario. Hence, the pressure angle modification in profile shifted gears is limited to values lesser than 45^0, depending on the profile shift (Figure 12). It is noticed from the results that the gear tooth profile is influenced

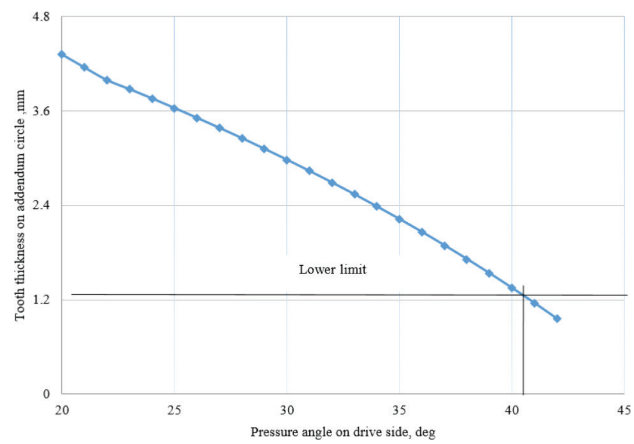


Figure 10: Tooth thickness at the addendum.

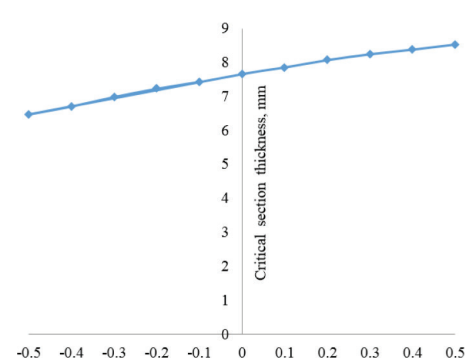


Figure 11: Critical section thickness with profile shift.

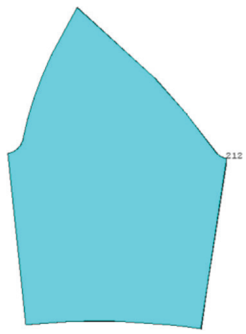


Figure 12: Pointed tooth.

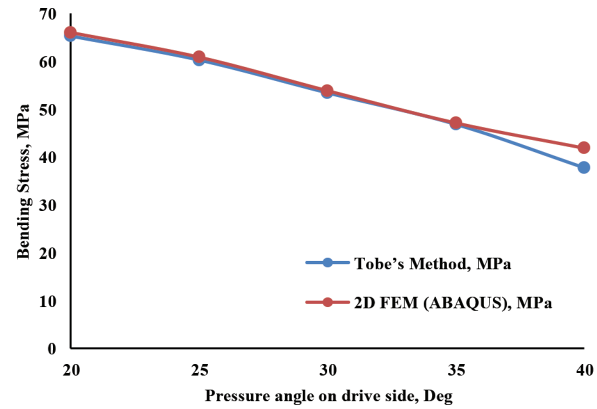


Figure 13: Bending stress at the tooth root.

by change in the pressure angle, module and number of teeth.

4.2 Gear Tooth Stresses

Stresses in gears are entirely dependent on tooth geometry, load and pressure angle on drive side it is noticed that a substantial drop in bending stress at the critical. Figure 13 and Table 3 shows the bending stress in symmetric and asymmetric spur gears and found that bending stress was reduced by 42.14 % by increasing the pressure angle from 20^0 to 40^0 without altering other gear parameters.

Table 3: Static bending stress

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

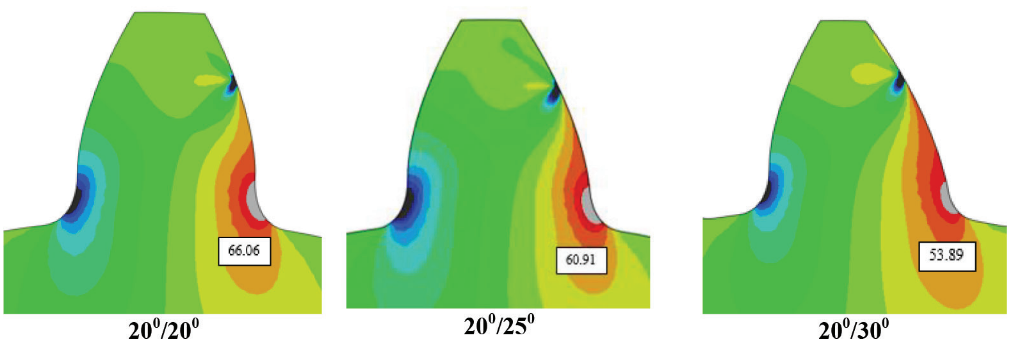


Figure 14: Bending stress contours.

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 14.

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 15 that bending stresses decrease with increases pressure angle and profile shift for a given pressure angle.

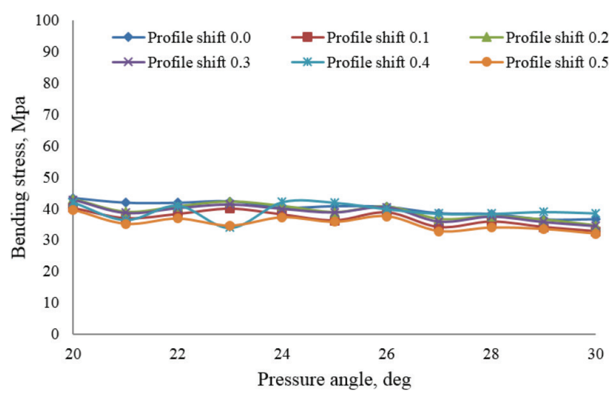


Figure 15: Bending stress for different profile shifts.

5.0 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 mining gears bending stress reduced to 32% without changing the load and material

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