Comparison of Bending Stress on Circular and Elliptical Profile Fillet of Helical Gear Using AGMA and ANSYS

Bhupendra Kumar Sahu[1], Mahesh Dewangan[2]

[1]PG Scholar, [2]Associate Professor, [12]Department of Mechanical Engineering

[12]Shri Shankaracharya Technical Camus, Shri Shankaracharya Group of Institution, Bhilai, Chhattisgarh, India

Abstract

Gear tends to play a very vital role in all industries for power transmission between shafts. It is a critical mechanical component in a mechanical power transmission system. A pair of helical gear generally subjected to two types of repeated load, one is bending load and another is surface load. Two types of stresses developed in helical gear pairs are: Bending stress (causing bending fatigue) and contact stress (causing contact fatigue). Both stresses act at a time at the same point of contact. If the both stresses attain their maximum value, the failure of gear occurs mainly in two distinct regions, which are tooth flank and the tooth root fillet. These types of tooth failure can be minimized or can be avoided by taking greater care during the design stage by avoiding the problem and creating a proper tooth surface profile with the better manufacturing method. This thesis reviews the methodology used to investigate bending strength of helical gear. Bending stress plays a vital role in gear design and manufacturing to reduce failure and cost. Magnitude of bending stress is controlled by the nominal bending stress and the stress concentration due to the geometrical shape. This work shows that the bending stress estimation for different shape of tooth root fillet. AGMA and ANSYS is used to estimate bending stresses at tooth root fillet and suggestion for best suited shape of tooth fillet.

Keywords: *Design; modelling; Helical gear; Tangential force; bending stress; root fillet; AGMA; ANSYS; FEA.*

I. Introduction

Gear is a mechanical element which has number of teeth which is used to transmit power or motion or torque from one shaft to another shaft. Shafts may be parallel or may be at an angle. According to the shaft position different types of gears are used.

Power transmission process is accomplished by successively engaging teeth. For transmitting power no intermediate link is required in gear, power is transmitted by direct contact of gear teeth. To transmit definite power or motion or torque from one disc to the other disc with any slipping between disc, projection and recesses on the disc are needed which can mesh with each other. The disc on which teeth are formed is known as gear wheels. As gear is a power transmitting element therefore gear must have sufficient strength to transmit power without any failure. Gear material as well as gear geometry highly influence the gear strength. This thesis is focused on the profile of gear teeth with most common used material to transmit high power with higher load carrying and respectively with lower weight. Failure of gear occurs when working stress exceeded the yielding/ultimate stress. It is observed that the major failure of gear is due to tooth bending fatigue, tooth-bending impact, and tooth wear. Failure of gear before life span can be avoided if greater care is taken during the design stage a sound gear design can be designed. Gear tooth generally subjected to cyclic and dynamic stress failure that are due to tooth bending fatigue, tooth-bending impact, and tooth wear. Gear tooth failures mainly occur in two distinct regions, the tooth flank, and the root fillet. The objective of this study is to estimate bending stress at root fillet using circular and elliptical root fillet and suggest best suited tooth root fillet for helical gear having better performance and strength.

II. Literature review

Gear analysis is the most important issue in the field of design of machine element. Many researches in this field is carried out by many authors for improving strength and performance of gear are given here are as following: C. Salamoun. [1] They have proved for helical gear that the coordinates of fillet and the coordinates of the involute tooth profile were found correct. In their research they compared the result computed of the same gear made by computer software. The computed profile and the plotted profiles were found alike. S. Vijayarangan. [2] It is proved that the behavior observed for orthotropic material of helical gears and adopted carbon steel material of helical gears are very similar to each other. The maximum normal stress in the direction of fiber in orthotropic material gears was found little more than the maximum stress in adopted carbon steel gear material. The maximum normal stresses in the other mutual perpendicular directions for orthotropic material gears are smaller than that of carbon steel gear. Chien-Fa Chen. [3] They have developed mathematical model for the modified helical gear with small number of teeth by tooth-profile shifting and basic geometry modification. Tooth undercutting condition for the involute profile gears has been developed using the developed mathematical models. The proposed methods by him and mathematical models developed by him for the modified helical gear can be very useful for design and manufacture of spur and helical gears with small number of teeth.

Th. Costopoulos. [4] In this research concept of the asymmetric half-involute gear teeth was introduced. The geometry of the gears was investigated and the generation was done using the Theory of Gearing. The Finite Element Analysis results obtained for a wide range of tooth numbers that indicates decrease in the maximum pinion tooth root fillet bending stresses up to 28% as compared to standard designs, which results in an increase of the load carrying capacity of gear. In many machinery areas like automotive power transmissions, locomotives, and all power transmitting units, where we have to save volume and weight, also these are the primarily need for gear manufacturing, in their research positive shifting of the pinion tooth profiles has been done, resulting in an increase of the pressure angle. They also discussed the advantage of the use of the asymmetric design are: It can improve bending strength of gear; it also increases pitting & scoring resistance without changing the size or the number of teeth in the gearbox. H. Endo. [5] The concept of the differential diagnosis (DD) technique was developed based on the understanding of the meshing behaviour of gears. They proposed that differential diagnosis is possible by using the difference in the Design of the echoing fault impulses caused by spalls and tooth fillet cracks. Miryam B. [6] A model of non-uniform load distribution along the line of contact of spur and helical gear teeth obtained from the minimum elastic potential energy criterion, has been applied to the determination of the critical tooth-root stress, whose value is determinant for evaluating the load capacity of the gear set. A.Y Gidado. [7]The bending stress is estimated using different face width and suggested that the results obtained from ANSYS when compared with the AGMA procedure for various face width, results shows that there is a little variation with a higher difference in percentage of bending stress is 4.70%. From the results obtained they concluded that ANSYS can also be used for predicting the values of bending stress at any required face width which is much easier to use to solve complex design problems like gears.

S. Jyothirmai. [8] An attempt had been made to compare the performance of various helical gear systems for a given set of specification using AGMA standards as well as a FEA approach. The developed Finite Element Analysis model was validated against the analytical approach. The model was found to be very close to the analytical approach. Further stress analysis was carried out using Finite Element Analysis. They found that the overall performance of crossed helical gear was found to be the best in terms of stress as well as tooth strength at low speeds and low loads. R. Prabhu Sekar. [9] Compared to conventional design, in the direct designed symmetric helical gear, the LSR value is lesser at the double pair contact region. Due to increase in gear ratio, the maximum values of ultimate stress decrease marginally due to loading at the critical line. The values of ultimate stresses decrease with higher values of top land thickness. On increasing the number of pinion teeth, there is only a marginal reduction in the maximum value of ultimate strength due to loading at the critical loading line. Toni Jabbour. [10] A method to calculate the distribution of the stress at the tooth root and of the contact stress along each contact line of a pair of spur and helical gears. The method is based on the decomposition of the gear into an infinite number of small spur gears. The results obtained from this method have been further confirmed by finite element calculations. Zaigang Chen. [11] Two calculation methods of the gear tooth fillet-foundation stiffness had been discussed in the presence of tooth root crack. This calculation method proposed based on the traditional gear tooth fillet-foundation stiffness calculation model for gears. They compared proposed calculation model with the traditional calculation model, results shows that the proposed models are capable of taking tooth root crack into account in the gear tooth fillet-foundation stiffness calculation.

All the above wok are focused on improving the strength and performance of the gear by reduction of weight, wear, vibration and noise. Among all above mentioned work most of the work had been focused on tooth bending stress. The studies were performed mostly using involute and asymmetric gear tooth profiles of helical gear.

III. Methodology

Based upon formula discussed in problem formulation and using same material properties with following gear parameters shown in Table 4.1 tangential force and bending stress are calculated and shown in Table 5.2.

Table 4.1: Helical gear parameters used in this work

S. No.	Parameter	Description	Value
1.	Z	No. of teeth	30
2.	m	Normal Module	3, 4, 5, 6, 8, 10mm
3.	pcd	Pitch circle diameter	90, 120, 150, 180, 240, 300mm
4.	theta	Normal Pressure angle	20^o
5.	phi	Helix angle	15^o
6.	f	Face width	100mm
7.	P	Power	30 kW
8.	N	Speed	1200 rpm

The design of gears for power transmission for any particular application is a function of (Khurmi and Gupta 2009): The expected transmitted power, the driving gear's speed, the driven gear's speed or speed ratio and the centre distances. In this paper we designed the helical gear according to bending strength condition and the tooth bending stress equation for helical gear teeth is given as:

$$\boldsymbol{\sigma_b = \frac{F_t}{bmj}.C_v.K_o.(0.93K_d)} \text{ -------------------------- } \mathbf{(1)}$$

Introduction of constant 0.93 with the mounting factor reflects slightly lower sensitivity of helical gears to mounting conditions. All the calculations are carried out on the basis of equation (1) recommended by AGMA. For modeling of gear different parameters used by A.Y Gidado for validation of bending stress are:

Force transmitted (F_t) = 1591.53 N
Face width (b) = 100 mm
Normal module (m) = 10 mm
Geometry factor (J) = 0.56
Velocity Factor (C_v) = 1.34
Overload factor (K_o) = 1.25
Load distribution factor (K_d) = 1.3

The materials properties are used for gear are: Density (ρ) = 1.6 kg/m^3, Modulus of elasticity (E) = 200 GPa, Poissson's ratio (m) = 0.3. Therefore, the torque (T) is calculated by equation (2) and tangential force (F_t) is calculated by equation (3)

$$\mathbf{T = \frac{P\times60\times1000}{2\pi N}} \text{ ---------------------------- } \mathbf{(2)}$$

$$\mathbf{F_t = \frac{2\times T\times1000}{pitch\ circle\ diameter}} \text{ --------------------------- } \mathbf{(3)}$$

Values of bending stress for module = 10 mm, face width = 100 mm and pitch circle diameter (pcd) = 300 mm as calculated by A.Y Gidado is

$$\sigma_b = \frac{1591.53}{100\times10\times0.56} \times 1.34 \times 1.25.(0.93 \times 1.3) = 5.755 \text{ MPa}$$

4.1 Modeling of helical gear

In this work the helical gear model was designed in Creo Parametric 2.0 design modeler. Creo Parametric is a suite of programs, which are basically used in designing and manufacturing range of products. This work basically deals with the solid modeling feature of Creo Parametric. In the case of these analyses five different gears were modeled in Creo Parametric by varying the module. In Creo Parametric the ‘Relation’ and ‘Equation’ itself gives the idea about relating the feature with the help of equations. The procedure used to model the gear of 30 number of teeth with the combination of the all above mentioned parameters in the Creo Parametric 2.0, are carry out using Tools parameter/relation menu, which is as shown in Figure 4.1.

First step is to define the basic parameters on which the model has dependencies. This can be done by going to Tools / Parameters menu, and inserting the basic gear parameters and then go to Tools/relation to relate these parameters using equations. Relations are used to express dependencies between the dimensions of a feature. Figure 4.1 and 4.4 showing the Tools parameter/Relation menu and 3D modeled of a helical gear from Creo Parametric screen respectively.

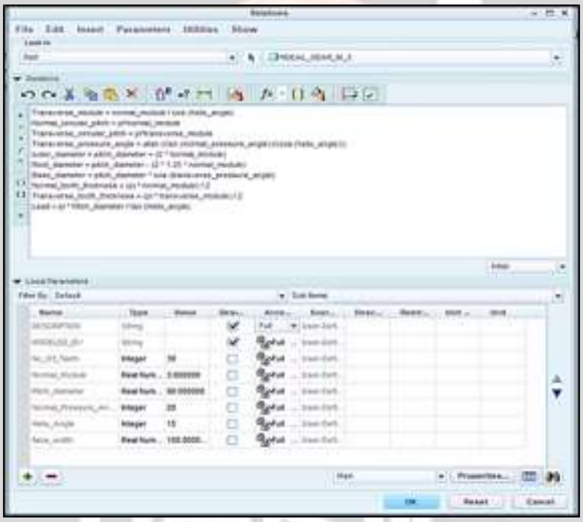
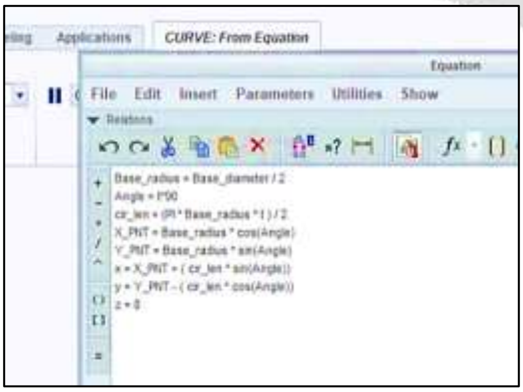
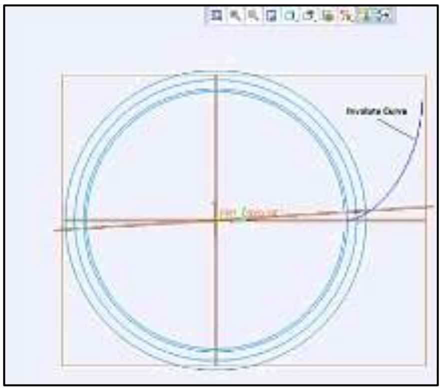


Fig.4.1: Tools parameter and relation menu in Creo Parametric 2.0

The most complicated part in any gear modeling is the involute profile of its teeth. There are number of ways of creating involute profile of a helical gear as shown in Fig4.2 (B). The curve equation for defined the involute curve is shown in Fig4.2 (A):

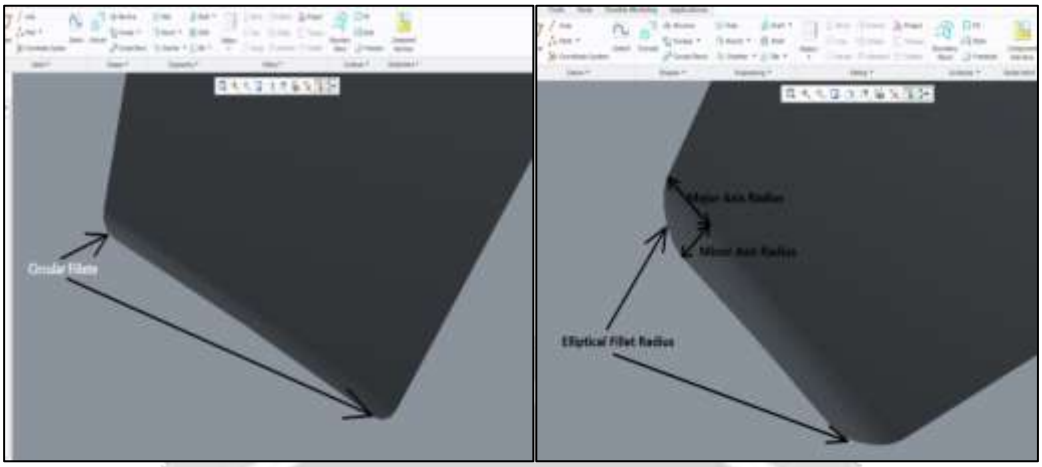




(A) **(B)**

Fig.4.1: (A) The curve equation for defined the involute curve, (B) Involute curve defined in Creo Parametric 2.0 using curve equation

Now modelling is done in Creo 2.0 using Circular and Elliptical fillet as shown in fig. 4.5 and 4.6.



(A) **(B)**

Fig. Error! No text of specified style in document.**.2: (A) Circular fillet radius applied on the all helical gear, (B) Elliptical fillet radius applied on the all Table 5.2: Value of bending stress on gear calculated from formula**

Table 4.2 Value of bending stress on gear calculated from formula

S. No.	Module (mm)	Face width (mm)	Tangential Force (N)	Bending Stress (MPa)
1	3	100	5305.11	**63.948**
2	4	100	3978.83	**35.971**
3	5	100	3183.06	**23.021**
4	6	100	2652.56	**15.987**
5	8	100	1989.41	**8.993**
6	10	100	1591.53	**5.755**

IV. Results and Discussions

Based upon literature Gidado et al [10] value of bending stress calculated in present work using ANSYS is validated with the literature Gidado et al [10]. Results are shown in the below Table 5.1, it shows that stress obtained are very close in magnitude. So, the value of bending stress for face width 100 mm and module 10 mm is taken for present work.

Table Error! No text of specified style in document.**.1: Validation of present work and existing work as per literature Gidado et al [10]**

S. No.	Face Width (mm)	Gidado et al [10] (AGMA)	Gidado et al [10] (FEA)	Present (ANSYS)
1	80	7.1941	7.0256	7.1222
2	85	6.7709	6.5818	6.6355
3	90	6.3948	6.4873	6.3409
4	95	6.0582	6.1673	5.9478
5	100	5.7553	5.4856	5.7185

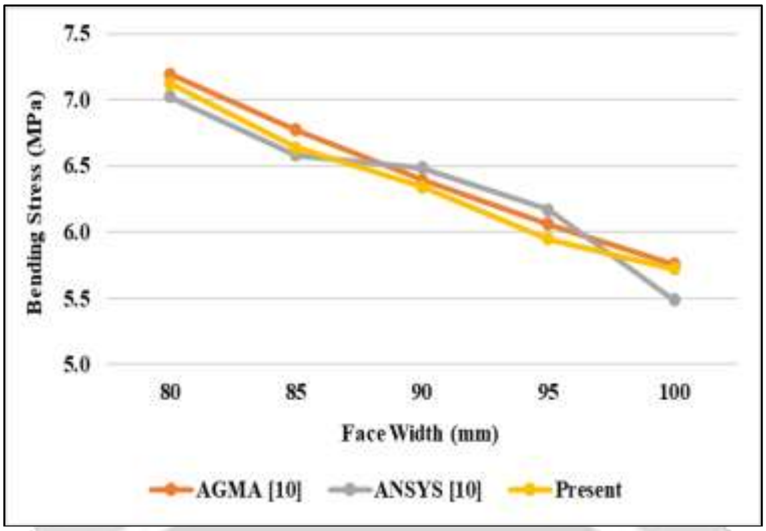


Figure Error! No text of specified style in document.**.3: Validation graph of literature work [10] and present work for different face width.**

Gear model prepared for analysis using materials properties [10] and taking face width 100 mm for varying module 3, 4, 5, 6, 8, 10 mm values of bending stress and deformation are calculated using ANSYS. The Table 5.4 and 5.5 shown below is results of bending stress and deflection on helical gear obtained from finite element analysis (FEA) for circular tooth root fillet and elliptical tooth root fillet applied in helical gear with different modules and face width 100 mm. Percentage difference of bending stress and deformation between circular fillet and elliptical fillet also shown in Table 5.2 and 5.3. Validation graph is shown in figure 5.1.

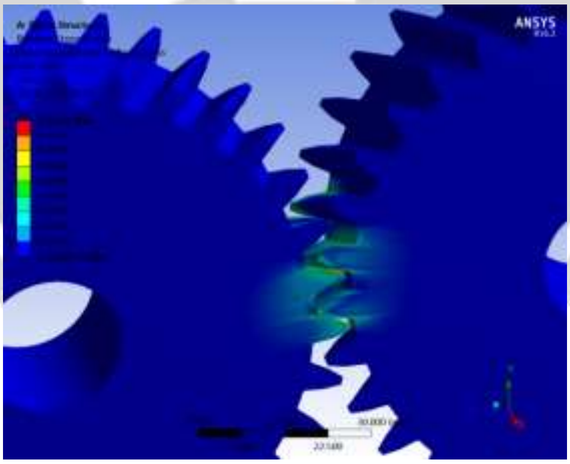


Fig. 5.2: Validation program done in ANSYS Workbench

Table Error! No text of specified style in document.**.2 Comparison of bending stress results obtained from finite element analysis method**

S. No.	Module (mm)	Present (Circular) (MPa)	Present (Elliptical) (MPa)	% Difference
1	3	7.1222	6.5651	7.82%
2	4	6.9442	6.1782	11.03%
3	5	6.8217	5.9727	12.45%
4	6	6.5374	5.5889	14.51%

5	8	6.0226	4.7814	20.61%
6	10	5.6421	4.1745	26.01%

Table 5.4 indicates that the percentage difference in bending stress for circular fillet and elliptical fillet for different modules are in increasing order on increasing order of modules for same face width. Also, it shows that the bending stress is reduced in case of elliptical root fillet as compared to circular root fillet. The graphical representation of bending stress with respect to module is shown in figure 5.3. From this figure it is observed that graph for elliptical fillet is clearly apart from graph for circular fillet. Both graphs do not intersect anywhere. Also, the gap between points (indicating bending stresses) is increasing for increasing in module. This shows that the elliptical shape root fillet has better strength than the circular shape root fillet.

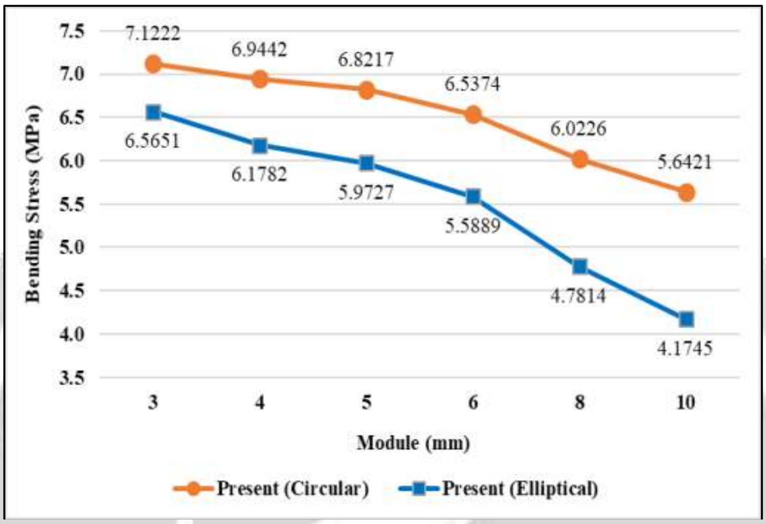


Fig. Error! No text of specified style in document.**.3 Graphical comparison and representation of bending stress due to circular and elliptical root fillet design**

Table Error! No text of specified style in document.**.3 Comparison of total deformation results obtained from finite element analysis method**

S. No.	Module (mm)	Present (Circular) (MPa)	Present (Elliptical) (MPa)	% Difference
1	3	0.0010031	0.0009619	4.11%
2	4	0.0009875	0.0008906	9.81%
3	5	0.0009619	0.0008593	10.67%
4	6	0.0009363	0.0007880	15.84%
5	8	0.0009107	0.0007167	21.30%
6	10	0.0008851	0.0006854	22.56%

Table 5.5 indicates that the percentage difference in total deformation for circular fillet and elliptical fillet for different modules are in increasing order on increasing order of modules for same face width. Also, it shows that the total deformation is reduced in case of elliptical root fillet as compared to circular root fillet. The graphical representation of total deformation with respect to module is shown in figure 5.4. From this figure it is observed that graph for elliptical fillet is clearly apart from graph for circular fillet. Both graphs do not intersect anywhere. Also, the gap between points (indicating bending stresses) is increasing for increasing in module. This shows that the elliptical shape root fillet has better strength than the circular shape root fillet.

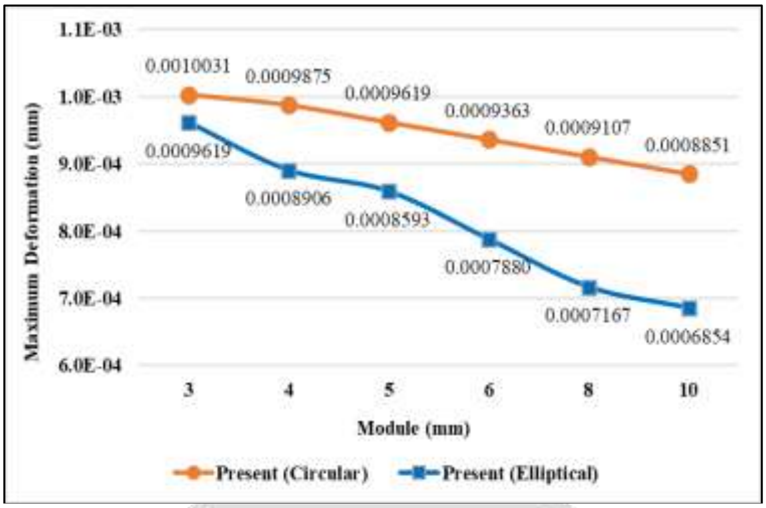


Fig. 5.4 Graphical comparison and representation of maximum deformation due to circular and elliptical root fillet design

V. Conclusion

The investigation result infers that the gear tooth stress and deflection in the elliptical root fillet design is less when compared to the circular root fillet design. Further, there is an appreciable reduction in the bending stress and the deformation for elliptical root fillet design in comparison to that of circular root fillet design. From the foregoing analysis it is found that the elliptical root fillet design is more apt for taken number of teeth and whatever may be the gear speed. Finite element analysis or ANSYS result indicate that the gears made of elliptical root fillet yield better strength (reduced bending and total deformation) thereby improving the fatigue life of the gear material.

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