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Analysis of an Addendum Modified Fiber Reinforced Composite Spur Gear Pair to Eliminate Noise & Interference Defects

[1]Dr V. Balambica, [2]Ganesh Babu Loganathan, [3]Vishwa Deepak, [4]Dr M,Achudhan,

[1]Faculty of Mechatronics Engineering, BIHER, Chennai, Tamil Nadu, India.
[2]Assistant Professor, Department of Mechatronics, Faculty of Engineering, Tishk International University-Erbil, Kurdistan Region, Iraq.
[3,4]Faculty of Mechanical Engineering, BIHER, Chennai, Tamil Nadu, India.

ABSTRACT

Elimination of transmission errors and the problem of interference, is a requirement in less number of toothed gearboxes. A standard spur gear does not always solve this problem. Hence profile modification becomes necessary for optimum performance. While designing a gear, the choice of material is also important for determining the function and strength. Nowadays, reinforced gears are attracting attention due to its many advantages in transmission and weight reduction. Here in this paper, addressing the deficiencies of the existing involute profile for smaller pinions, the main focus was to model and analyse a profile corrected fiber reinforced polymer composite spur gear. In this process, the thermal heat capacity changes were recorded experimentally using an infra-red temperature camera. The results of deflection were analytically compared with modified steel, and polymer reinforced glass fibre spur gear tooth. The analysis presents an efficient finite element method to predict the normal stresses and its deformation effects for varying number of teeth. Finite element analysis software Ansys14.5 was used on the proposed models in determining the tooth contact analysis parameters. Addendum modified fiber reinforced spur gear tooth models revealed a much lesser deformation, when compared to the other two materials, thus predicting stronger stiffness.

KEYWORDS

Addendum modification, fibre reinforced, polymer composite, deformation analysis, infra-red temperature camera (FLIR-E8),bending stress.

INTRODUCTION

There are many methods to increase the load carrying capacity and reduce noise. Among them, one is addendum modification. Again in this there is positive addendum modification and negative addendum modification [12-35]. Negative modification is carried out when there is a requirement. Here in this paper positive modification was adopted to achieve the purpose. Polymeric gear reinforced with glass fibre is chosen where the power application is low,The major concerns in gear design are the errors due to transmission, noise occurrence, stress concentration and the dynamic load [36-45]. Profile modification is one such technique, where these can be reduced. The proposed gear models based on the calculated gear parameters were created using modeling software CATIA V5 [46-58].Further more deformational analysis was performed and FLIR E8 high resolution camera was used to detect the heat capacity of the gear tooth. In order to develop upon this, polymer glass fibre reinforced material PA6+ 30% glass is used for the study and analysis. For the tooth correction,an addendum modification coefficient was added to the pinion model [59-78]. And then the same value was subtracted for the gear wheel but of the opposite sign. Interference and Undercutting can be prevented using smaller number of teeth. No interference occurs, if the addendum does not extend inside the interference point.. the true involute profile shape was considered while assuming it to be a short cantilever during analysis. The elastic modulus was compared with the equation of the Halpin Cardo mixture for a homogeneous model described in reference [1]. This showed the fiber influence from which the computed modulus was validated. Jeong Su et al. [2] formulated a method to calculate the tooth profile modification amount for a wind-turbine gearbox and confirmed its reliability by evaluation through finite element analysis simulation. It was found that contact was evenly distributed over the tooth surfaces improving the edge contact than before the correction. In this way, the impact and noise of the gear pair was decreased due to profile modification. Yilmazer and Cansever [3] prepared glass fiber reinforced PA6 through injection molding with different screw speed and feed rate. It was found that the alteration of screw speed and feed rate significantly reduces the fiber length. Accordingly, the meachanical properties such as shear and tensile strength increases, Mallesh et al. [4] analysed that the defects can be eliminated by increasing the pressure angle and addendum of mating gears. The effect of glass fibre orientation had shown a good performance shortening the time period of the crack growth according to A.J Merteus [5]. P.B. Pawar et al. [6] recorded composite gears manufactured by stir casting offered improved properties over steel alloys and these could be used as better alternative for replacing any metal gears. Polymer gears have increased applications presenting a number of advantages unmatched by steel gears [79-84]. However, they have poor thermal resistance limiting to rotation transmission. Ghazali et al [7] that the failures in gears usually occurred because of the limitation of material, load carrying capacity and thermal properties. Ulrich Kissling[8]reported addendum modification resulted in having a strong influence in transmission errors, generating an algorithm which was used to find out the tooth stiffness in regards to the normal stress and deflection.It was proved that FEA was one such technique that could be used for predicting loads with respect to time acting on the gear tooth. Li X Y Wang et al.[9] proposed a study optimizing the corresponding parameters by orthogonal experiments improving stability of transmission by the method of tooth profile modification. It was proved experimentally that the root stresses were reduced extensively eliminating noise and vibration using corrected gears. Recently Bodzas [10] studied the analysis effects of the addendum modification coefficient for tooth contact analysis parameters which can reason for appropriate gear geometry of the given construction and working condition. Gears are made up of different materials for various uses. Glass fiber is being increasingly used as a reinforcing material in gears for their low cost and high strength. J.Cathelin et al. [11] suggested that adding fibers can significantly increase gear performance in high torque transmitting applications. Transmission error was reduced to an amount of 8 % and 3% reduction in the tooth root stresses.

From the above research papers and published literature study, it was seen how metallic gears were nowadays being replaced using FRP materials with a much-improved gear performance. Also work was done on profile corrected gears using standard gear materials or if the materials varied from usual than the gears used for analysis was standard. Hence here in this paper, an attempt was made to perform static deformation analysis on an addendum modified fiber reinforced polymer composite spur gear to reduce the problems of interference and errors due to transmission furthermore studying the thermal capabilities [85-86].

The profile corrected gear was modeled, fabricated through injection moulding and analysed with all the improvements as discussed above. Positive correction causes an increased addendum with a net affect to eliminate the non-involute portion of the pinion and gear avoiding the problem of interference and undercutting. The objective was to analyse both the root stresses as well as deformation for an addendum modified fiber reinforced polymer composite spur gear making a comparison with modified steel and a polymer tooth. This was to improve the gear performance and the stiffness of the gear tooth further reducing the deformation and weight of the spur gear for different applications like railways, wind turbine gearboxes, aircrafts etc.

NOMENCLATURE:

i =Transmission ratio
P = Power in KW
m = module in mm
x =addendum modification factor
Z = Number of Teeth
Dp= Pitch Circle diameter in mm
Da= Addendum circle diameter in mm
Df = Root circle diameter in mm
h = tooth depth in mm
t = Tooth thickness in mm
Rf = root circle radius in mm
Rb = Base circle radius in mm
φ1 = Pressure angle at R in degrees
ba= Distance from pitch line of hob to the point of tangency of rounded corner with straight line form in mm
R = Pitch Radius of the gear in mm
M_t = Transmitted Torque in Nm
[M_t] = Design Torque in Nm
k = Load concentration factor
k_d= Dynamic load factor
a = Centre distance in mm
b= face width in mm
y=form factor
[σ_c]= Induced Contact stress in N/mm^2
[σ_b] = Induced Bending stress in N/mm^2
σ_c= Design Composite stress in N/mm^2
σ_f= fiber stress in N/mm^2
σ_m = Matrix stress in N/mm^2
$E_{(cl)}$= Elastic modulus longitudinal direction in N/mm^2
$E_{(ct)}$ = Elastic modulus transverse direction in N/mm^2
E_f= Elastic modulus of fiber in N/mm^2
E_m = Elastic modulus of matrix in N/mm^2
V_{et}= Composite volume in mm^3
V_f= Fiber volume in mm^3
V_m= Matrix volume in mm^3

i =Transmission ratio
n1 = Pinion speed in RPM
ρ= Density in g/cc
E= Young's Modulus in MPa
υ = Poisson ratio

3. METHODOLOGY

3.1 *Input Parameters and Design*

All the standard parameters used to construct the spur gear and the corrected gears for different addendum modification factors are as shown in Table 1.

m= 10mm Number of teeth on pinion (Z_1) = 20

Power P = 22 KW n_1= 900 rpm $\varphi_1 = 20^0$ Life=10,000hrs

Table 1 - Dimension Details as per Theoretical Design

Description	**Pinion Details**		
Type of Gear	**Standard**	**Profile corrected**	
m	10		
Z	20	14	16
X	0	0.43	0.5
Dp	200	140	160
Da	220	168.6	190
Df	175	123.6	145
h	22.5		
t	15.71	18.838	19.347

3.2. *Standard Gear Stresses*

The formulae is with reference to [1].

Design torque $[Mt] = Mt \text{ x } k \text{ x} kd$ (1)

Transmitted torque $Mt = \frac{60\text{x}P}{2\pi N}$ (2)

$Mt = \frac{60x22x10^3}{2x\pi x900}$= 233.427 Nm (2.1)

[Mt]= 233.427*x 1.3*(Initially k x kd=1.3*)* (2.2)

[Mt] =303.46 Nm (2.3)

Induced contact stress $[\sigma c] = 0.74x\left(\frac{i+1}{a}\right)\left[\left[\frac{i+1}{ib}\right]xE[Mt]\right]^{1/2}$
= 315N/mm^2 (3)

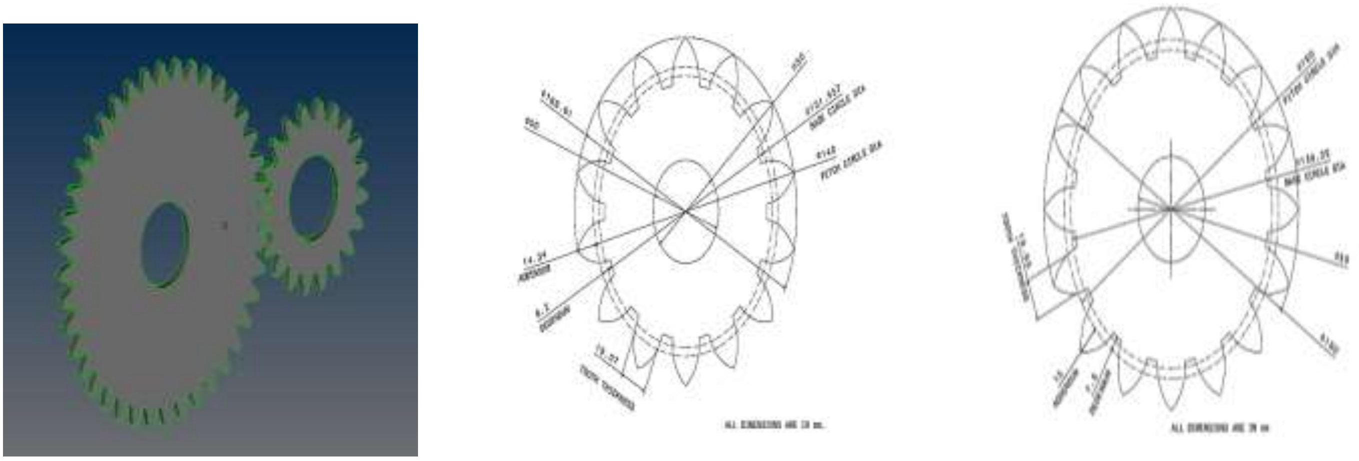
Induced bending stress $[\sigma b] = [\frac{[(i+1)x[Mt]]}{axmxbxy}]$ (4)
= 10.564 N/mm^2

3.3. *Correction Factor for corrected gears based on number of teeth:*

Case I: $i = 1.5$ and $z1 = 14$ $x = 0.5$

Case II : $i = 1.5$ and $z1 = 16$ $x = 0.43$

The profile of the involute full spur gear tooth was generated using CATIA. The single tooth profile was then selected meshed using hyper mesh software tool and imported through IGES file conversion to the ANSYS file. The gear teeth at the fillet region are discretized into very small elements comparatively with other regions of gear teeth. Here element types were considered as 8 node 82 plane elements.



(a) (b) (c)

Figure 1 - (a). 2-D model of Standard Gear Pair, x1=0, m=10, z1=20**(b).**2-D model of corrected pinion, x1=0.43, m=10, z1=14, $\varphi 1 = \mathbf{20^0}$**(c).**2-D model of corrected pinion, x1=0.5, m=10, z1= 16, $\varphi 1 = 20^0$

4. Finite Element Modeling:

The geometrical models of the gears were created in CATIA v5 as seen in Figure.1 (a,b,c) using dimensions obtained from theoretical design calculations. The tooth form includes the root fillet connecting the involute profile. It depends on the tool that will be used during manufacture. Hence a trochoidal shaped fillet is generated at the root. The geometrical conditions are as shown below;

$$Rf = \sqrt{(R\sin\varphi 1-(ba/\sin \varphi 1)2 + Rb2} \quad (5)$$

By applying a positive correction and performing the addendum profile shift with increased thickness, a 10-toothed gear can also obtain the strength of a 200-toothed gear. In positive correction, the two teeth of the mating pair receive an equal correction factor, but of opposite sign. It should be seen that the tooth tip does not sharpen much. This happens when the correction applied exceeds the limit. The pinion teeth are weaker than the gear tooth when both are made up of the same material. Hence in this paper, positive So correction was used, which tends to minimize the susceptibility of damage, making the tooth stronger. The maximum allowable positive correction values for tooth which are commonly in use [2]is 0.43 for 14 number of teeth and 0.5 for 16 number of teeth. The same was used to model the corrected tooth profile.

$$X1+X2=0 \quad (6)$$

(a) x = 0 (b) x = 0z = 10 (c) x = 0.5 z = 10

Figure- 2 - Standard Gear Tooth Profile Vs Addendum modified Profile

Figure- 2(a**)** depicts a standard gear tooth (x=0) for a 10 and 20 number of teeth. In Figure- 2(b) it is clearly seen how deep the root radius looks. This results in high stress concentration. As the number of teeth increases the fillet region is reduced and so the stress concentration. But when there is a requirement of a smaller number of teeth such as 10, 9, 8 etc. in gearboxes, using a standard tooth becomes difficult for continuous and efficient power transmission. Figure- 2(c), when compared to Figure- 2(b) shows relief to a greater extent thus providing low stress concentration.

4.1*Polymer Reinforced FRP Material and Its Elastic Behavior*:

Nowadays we come across many structural materials with a combination of plastic and glass fiber due to their low cost [28].Usually for structural applications, unidirectional layers are used to laminate a composite.Laminates which are unidirectional have higher strength and stiffness[30]. The role of polymer, should be in such a way that it should bind the fiberstogether, adequately improving mechanical behavior. This is because,they have high strength weight ratios [32], enhance stiffness, corrosion resistant, weather resistant and possess sound insulation properties. The orientation affects the mechanical properties in a polymer matrix, the fiber modulus is much greater than the polymer modulus. Here an assumption of Iso-Stress components was made. The volume of the polymer matrix (V_m)was assumed 70% and the glassfiber(V_f) as30%.

$$\sigma c = \sigma f = \sigma m \qquad (7)$$

The tensile load is acting along the fiber direction.Perfect bonding is assumed between the fibers and matrix. The strain or deformation of the entire composite $\in_c$is

$$\in_c = \in_m V_m + \in_f V_f \qquad (8)$$

The modulus of elasticity of a continuous and aligned fibrous composite in the direction of alignment is

$$= E_m V_m + E_f V_f \qquad (9)$$

$= E_{cl} = 2400(1 - 0.3)(0.7) + (80000x0.3)$ =25176Mpa (9.1)

Equation of transverse modulas for polymer composite (unidirectional) is,

$$\frac{1}{E_{ct}} = \frac{(1-V_f)}{E_m} + \frac{V_f}{E_f} \qquad (10)$$

=270095.2367 Mpa (10.1)

Eeq= 2x$E_{cl} x E_{ct} / (E_{cl} + E_{ct})$(11)

=46058.788 MPa (11.1)

Table 2 - Material properties

Properties	Steel	PA6	Glass Fibre	PA6+ 30% Glass Fibre
Young's modulus (MPa)	2100	2400	80000	46058.7
Poisson's ratio	0.3	0.39	0.23	0.35
Density (g/cc)	7.75-8.07	1.13-1.15	0.041-0.042	1.17-1.62

5. Results and Discussion:

The Static stress analysis part was divided into;

i. Stress and deformation analysis an addendum modified steel & PA6 gear tooth and
ii. Stress and deformation analysis for an addendum modified FRP (PA6 +30 % Glass fiber) gear tooth.

At first the full model of the spur gear was created as seen in Figure- 3. One tooth was cut and taken up for the problem of spur gear tooth stress distribution and deformational analysis with a back-up ratio of 0.5. Earlier according to texts [3] & [4], the shape of the tooth was not an involute profile, but being assumed as a rectangular shaped cantilever. This was a classical approach according to the cantilever beam theory. The equations of deflection and stresses were thus formulated. The true involute profile shape of the gear tooth was taken into consideration for the analysis. Figure 4 shows the gear meshing, surface to surface contact was carried out and the element type used was 8 node plane 82 with solid quadrilateral mesh. A fine mesh was done for accuracy. The gear teeth under meshing condition have imperfections due to their involute shape and as there is no uniform spacing, the teeth deflect under loading. One major cause of gear failure is also the fracture at the base of the gear tooth. Though perfect determination of the amount of static loading stresses is not possible, approximate values can be derived. The boundary condition is such that the bottom of the tooth is assumed fixed (Figure .5(a)) and the other two sides (Figure. 5(b)) of the cut tooth was assumed symmetrical the loading was done at the pitch point as shown in Figure. 6 for a standard gear.

Under static state stress condition, standard gear for the steel material was analyzed initially in finding the bending and contact stresses. Figure. (7,9) shows the Ansys results for the contact and bending stresses. These were validated with the theoretical results which were in agreement except with a 14% and 3% variation respectively. Figure. 8 shows the deformation of standard structural steel to be around 0.0020916 mm for a 20 number of toothed gears. Whereas in Figure. 10 & 12, for a 14 &16 toothed steel profile corrected gear, the deformation was obviously seen reduced to more than 50% for steel which in turn improves the stiffness properties Comparing Fig. 10 and Figure. 14, the deformation is seen to be increased more in an addendum modified polymer gear tooth provided the number of teeth remaining the same. The load is applied at a distance of 15.5mm from the tip reaching the pitch point as shown in Figure. 16. The results of deformation are more for a profile corrected PA6 gear as Figure. 14 when compared to an FRP corrected gear in Figure. 17 for the same number of teeth

From Figure. 9 and 11 the bending stress results were seen for standard steel and corrected steel gear. It shows a vast difference in relation to the bending stresses. A small reduction in the bending stress has much greater influence in the relief of stress concentration at the root region. It can be seen from Figure.13, 15 and 20 that, when the gear has same number of teeth, there are no variations found in the bending stresses for an addendum modified tooth of three different materials. But Figure. 11, 13 and 18, &20 depicts how when tooth number is increased in a profile corrected gear,the bending stress increases with a decrease in deflection. This can be noticed in Figure.10,12 and 18,19.

5.1- ***Stress and Deformation Analysis report: Material Standard Steel, Addendum modified Steel & PA6.***

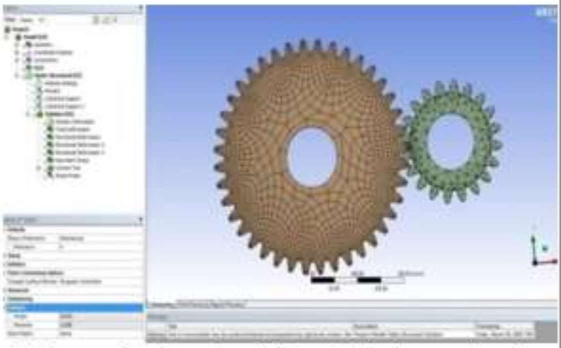


Figure. 3 -Standard Gear Pair in mesh z1 z1=20,z2=40, X=0

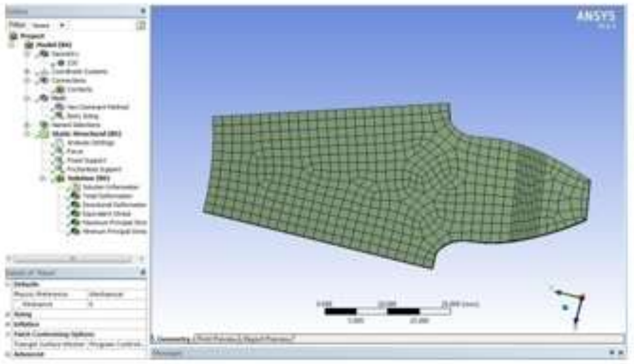


Figure. 4 - Meshed Model of a standard spur gear

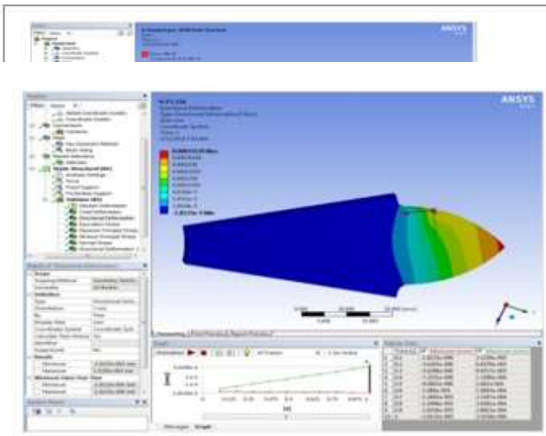


Figure.12-CorrectedSteel–Total deformation=0.00033239 mm;Z=16 X=0.5

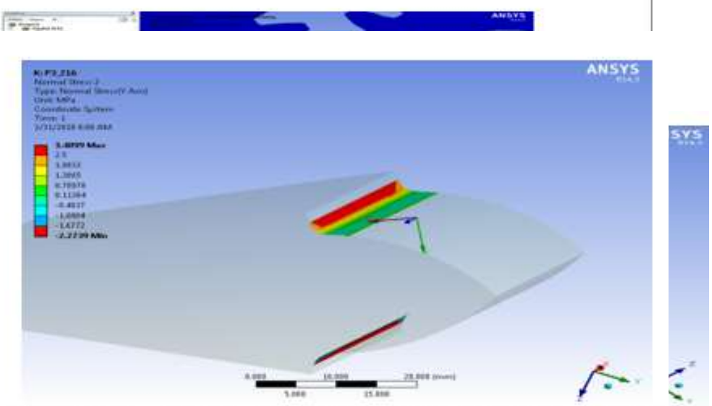


Figure13 - Corrected Steel Bending stress= 3.489 Mpa;Z=16,X=0.5

5.2-Stress and

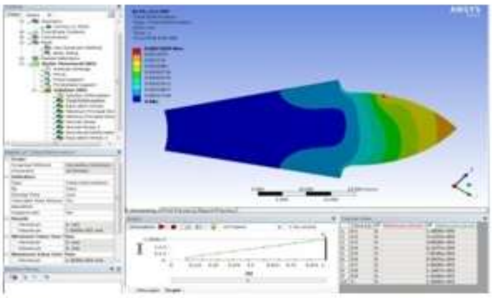


Figure.14- Corrected PA6 - Total deformation =0.0015609m ; Z=14, X=0.43

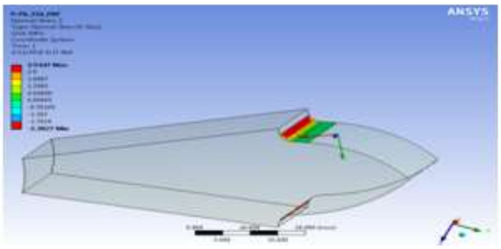


Figure.15-Corrected PA6 -Bending stress= 3.5447 Mpa ; Z=16 X=0.5

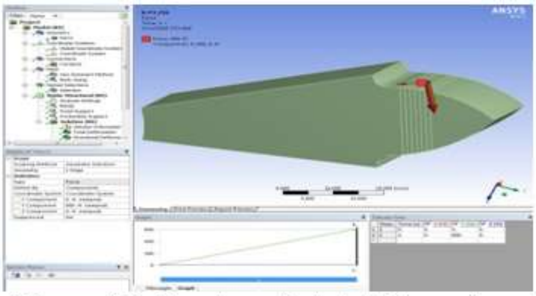


Figure.16 - Load applied at 15.5mm from tip for a corrected FRP spur gear tooth Z=16

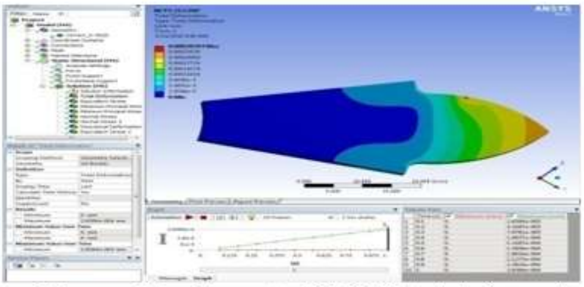


Figure.17 –Corrected FRP-Total deformation = 0.000265mm;Z = 14, X=0.43

Deformation Analysis report: Material Fiber Reinforced Polymer Composite

Figure.18 – Corr 2.941

–Total deformation m; Z=16

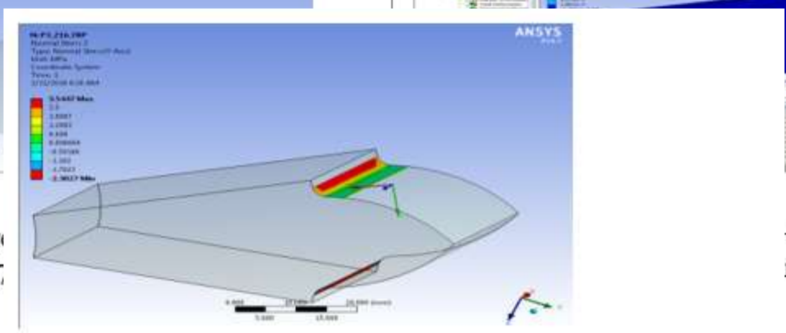


Figure.20- Corrected FRP-Bending Stress= 3.5447MPa ; Z=16

Table 3 and 4 exhibits the results of the total deformation and bending stresses induced. Bending stresses are so much important as failure can occur due to extensive loading. The total deformation decreases as the number of teeth in a profile modified gear is increased for an FRP material.

Table 3 -Total Deformation Results

MATERIAL(Teeth number)	TOTAL DEFORMATION inmm
Steel (14)	3.410E-04
Steel (16)	3.3289E-04

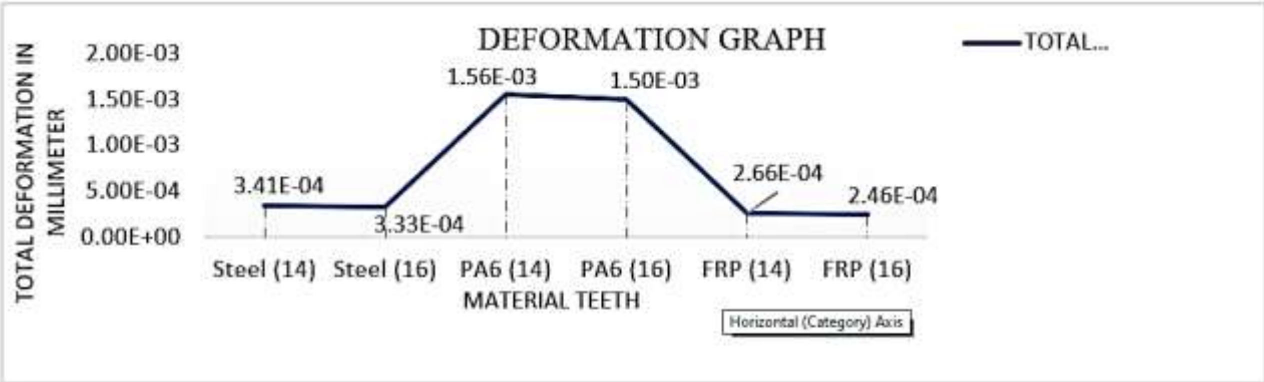
Table 4-Bending Stress Results

MATERIAL (Teeth Number)	BENDING STRESS in Mpa
Steel (14)	2.8911
Steel (16)	3.4899

PA6(14)	1.5609E-03
PA6(16)	1.5008E-03
FRP (14)	2.659E-04
FRP (16)	2.4569E-04

PA6(14)	2.9417
PA6 (16)	3.5447
FRP (14)	2.811
FRP (16)	3.5447

Graphical results show bending stress distributions and deformations to compare an addendum modified composite gear with steel and polymer gears. From Figure. 21,the deformation tends to decrease as the number of teeth increases.Also, deformation for the addendum modified fiber reinforced composite material was less when compared to the other two materials.Figure.22 represents the results of bending stresses vs the number of teeth for three different materials.It is clear from the graph that the bending stresses do not vary as long as there is a change in geometry. Material does not have any influence on the bending stresses.



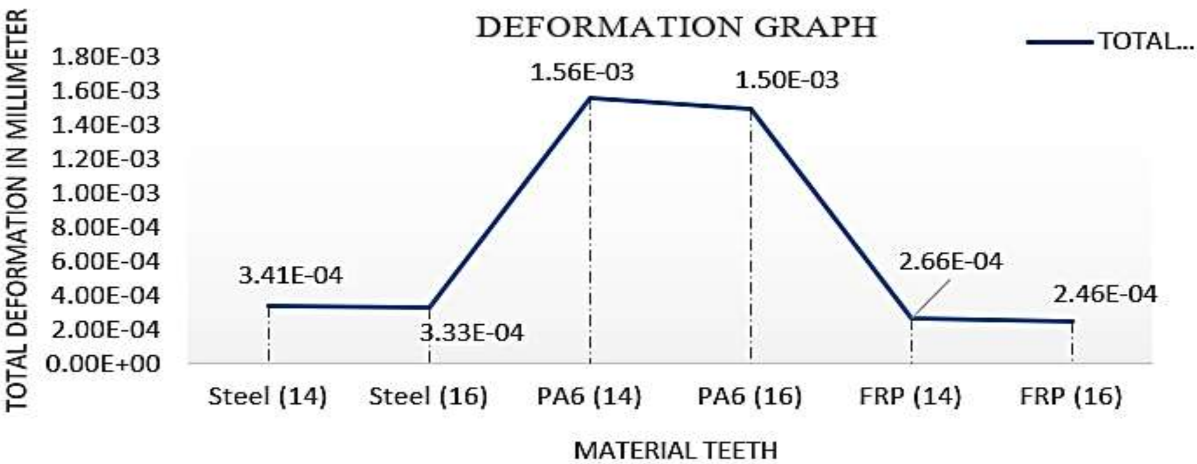


Figure. 21 - Deformation Graph

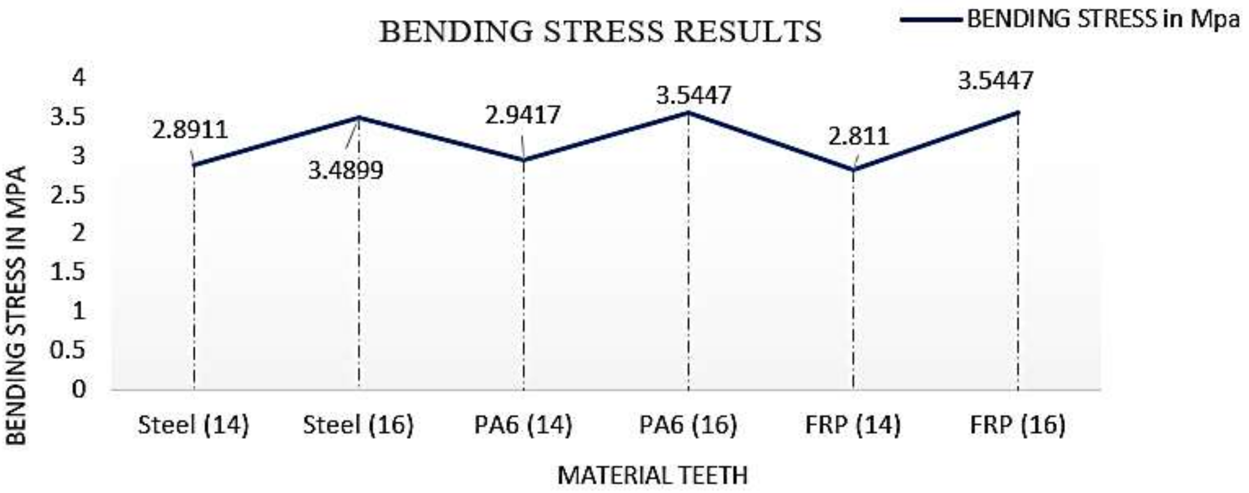


Figure. 22 -

Bending Stress Results

6. CONCLUSION

This analytical study presents an efficient finite element method to predict the bending stresses and deformation for an addendum modified fibre reinforced polymer gear.Regarding the bending stresses,no change occurred for the same geometry, as the geometry for any structural model does not have anyinfluence on the material type. Addendum modified FRP spur gear tooth models---compared to steel and PA6 revealed a much lesser deformation predicting stronger stiffness.

According to gear box manufacturers, improving only the accuracy of the gear does not reduce noise in the transmission unit. For compact gear boxes, corrected gears are found to be more efficient with high strength than standard gears. Thesegears extensively eliminates noise and interference defects.The work done was proved satisfactory both analytically and numerically. Hence, this FRP material composition can be a good alternative in manufacturing addendum modified gears for use in gearboxes and other gear application units. Figure. 22 - Bending Stress Results

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