Influence of Stresses Ina Modified Non-Metallic Spur Gear Pair

J. Manikandan and Bala Murali

Abstract--- *A gear is a rotating machine part having cut teeth, or cogs, which mesh with another toothed part in order to transmit torque. In order to avoid undercutting and interference, addendum modification of the gear tooth is carried out in this paper, a standard and a profile corrected carried out spur gear pair is modeled. The material used is plastic (Nylon-66).The gear ratio assumed was 1.5 with a module of 10 and correction factor taken to be 0.5. The dimension of the models was arrived at by theoretical calculations. In this paper, a model part is done with PRO-E for the modeling of the spur gear for both standard and profile corrected tooth. The finite element model of the gear tooth is imported to an analysis software ANSYS to study the bending effects for different modules. Results were obtained from the comparisons made for the bending stress variations for both the standard plastic spur gear tooth and the profile corrected plastic spur gear. The results of the study for the addendum modified tooth showed a decrease in bending stresses for the wider tooth for the same loading*

Keywords--- *Spur Gear, Bending, Contact, Stress Analysis.*

I. INTRODUCTION

Gears operate in pairs, when pinion is the driver, it results in step down drive in which the output speed decreases and the torque increases. On the other hand, when the gear is the driver, it results in step up drive in which the output speed increases and the torque decreases. Much research had been carried out from the yester years till today on gears. There were varied difference of opinion regarding gear tooth failures and there still exists a need for much more research on the behavior of surfaces in a pair of gear in contact. In the year 1950,Dr. W.A.Tuplin (7) concentrated on bending stress. Bending stress was a possible factor in failure of a gear. If the maximum stress in a gear tooth is less than the fatigue limit for the material, the tooth should not fail even after prolonged running. In 1957 Niemann and Rettig(5) tested a number of steel gears. The deflection of the tip was measured under both static and dynamic condition. In 1953 Strauch (6) in his paper presented that there was continuous error due to vibrations in uncorrected gears. Winter tested steel gears. The results of the bending stresses were obtained by the assumption of load application on the tooth tip teeth. In 1893 Lewis (9) presented in his paper, the Lewis formula for bending stress. J. D. Andrews (14) in his paper investigated the fillet stresses predicted in both external and internal forms of spur gear teeth using Finite Element Method. Sorincananau (17) in his paper investigated in his paper 2D versus 3D analysis for stresses in the root region of the teeth. M. Beghini (18) in their paper proposed a simple method to reduce the transmission error for a given spur gear by means of profile modification parameters. Ravichandrapatchigollaet.alin his paper emphasizes the results of Finite element analysis and the effect of rim thickness on gear tooth bending stress. C.V. Spitas & Spitas (19) in their paper made a comparison of the bending

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strength between the circular fillet and the trochoidal fillet, in which the use of trochoidal fillet proved to be advantageous.IvanaAtanasovsk(20) discusses in detail using Finite element method models of involute spur gears for monitoring the stiffness and the base pitch deviation influence on the load distribution and the load capacity.Dr. I.G.H Van Melick(21) in his paper investigates steel and plastic gear transmission using numerical finite element and analytical methods to study the influence of stiffness of the gear material on the bending of the gear teeth. The change in the load sharing also changes the stresses. The Costopoulous et.al (22) proposed several tooth alternative design for increasing the load capacity. Buljanovic. K et.al (23) presented linear tip profile modification and compared the gear tooth root stresses of the profile modified one with a standard spur gear tooth. Ali Raad Hassan (24) had presented in his paper, the results of contact analysis between two different spur gear teeth considered in different contact position. The results were compared theoretically. Dr. Eng. Ulrich Kissling's(25) approach uses an algorithm that includes conventional method for calculating tooth stiffness in regards to bending stress and shearing deformation. The results were then compared with FEM. He pointed out that the profile correction has an influence in reducing transmission error. Shanmugasundaram Shankar et.al(26)describes in their paper, the effect of profile modification in the root fillet region with the use of a circular fillet for increase of the tooth strength of a spur gear using CAD.

II. DESIGN

Gear Parameters used

Gear Type: - Standard involute full - Depth Teeth.

Input parameters for standard spur gear pair

For Gear ratio 'i' = 1:1.5

Module 'm'= 10,

Pressure angle 'α' = 20^0,

No. Of teeth on pinion'$Z_{1'}$= 14, No. Of teeth on gear Z_2 = 21, Speed n_1 = 1000 (assumed)'

a =175 mm

n_2 = 666.67 rpm

Circular pitch 'p'=31.415 mm

Pitch circle diameter 'd'

Pinion 'd_1' = 140 mm

Gear 'd_2'= 210 mm

Thickness of the teeth 'S' = 15.70 mm

Outer circle diameter 'd_a'

Pinion = 160 mm

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Gear = 230 mm

Base circle diameter of the gear 'd_b'

Pinion 'd_{b1}' = = 131.55 mm

Gear 'd_{b2}' = 197.33 mm

Root circle diameter 'd_{f1}'

Pinion, 'd_{f1}' =116.86 mm

Gear 'd_{f1}'= 186.86 mm

Face width 'b'= 94.24 mm

Input parameters for corrected spur gear pair

For Gear ratio 'i' = 1:1.5

Module 'm' = 10

Pressure angle 'α' = 20^0

No. Of teeth on pinion 'z_1'= 14

No. Of teeth on gear 'z_2 = 21

Correction factor for Pinion 'X_1' = 0.5

Correction factor for Gear 'X_2' = -0.5

Speed n_1 = 1000 (assumed)

Speed ratio 'I' =

$$\frac{1000}{n2} = \frac{21}{14}$$

n_2 = 666.67 rpm

Circular pitch 'p'=31.415 mm

Pitch circle diameter 'd'

Pinion 'd_1' =140 mm

Gear 'd_2' =210 mm

Thickness of the teeth 'S'

Pinion 'S_1' = 19.34 mm

Gear 'S_2'=12.06 mm

Outside diameter of the gear 'd_a'

Pinion 'd_{a1}' =170 mm

Gear ‘d_{a2}’ = 220 mm

Base circle diameter of the gear ‘d_b’

Pinion ‘d_{b1}’ = 140 cos20^0= 131.55 mm

Gear ‘d_{b2}’ = 210 cos20^0 = 197.335 mm

Root diameter of gear ‘d_r’

Pinion ‘d_{r1}’ =125 mm

Gear ‘d_{r2}’ =175 mm

Face width ‘b’ =94.247 mm

Total depth ‘h’= 22.5 mm

Stresses and Load Calculation

Modulus of elasticity (E) = 5.5x103 N/mm^2

Poisson’s ratio μ= 0.420

Yield stress σ_y= 110 N/mm^2

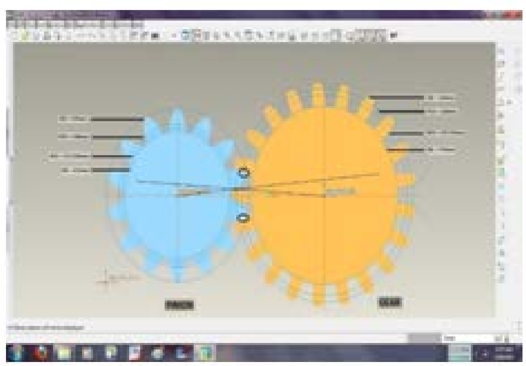

Fig.1: Meshing model of a corrected spur gear

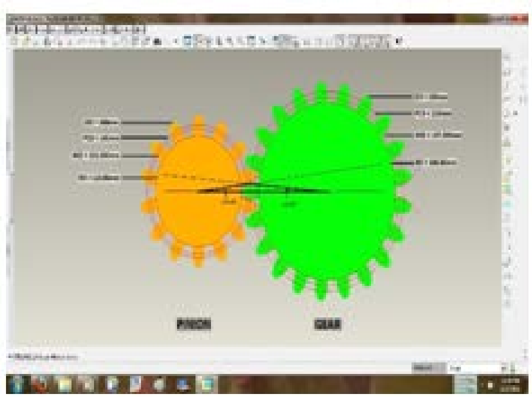

Fig.2: Meshed model of standard spur gear pair

Pair at 0^0 with case 4 & 5 point of contact

BHN= <350

Tensile strength $\sigma_{u=}$ 170 N/mm^2

Design bending stress σb = 85 N/mm2

Induced bending stress σ_b = 8.07*10^3 N/mm^2

Design contact stressσ_c = 180 N/mm^2

Induced contact stressσ_c = 48.067 N/mm^2

Tangential Load (F_t) = 9.553 N

Normal Load (F_n) = 10.144 N

Radial Load (F_r) =3.469 N

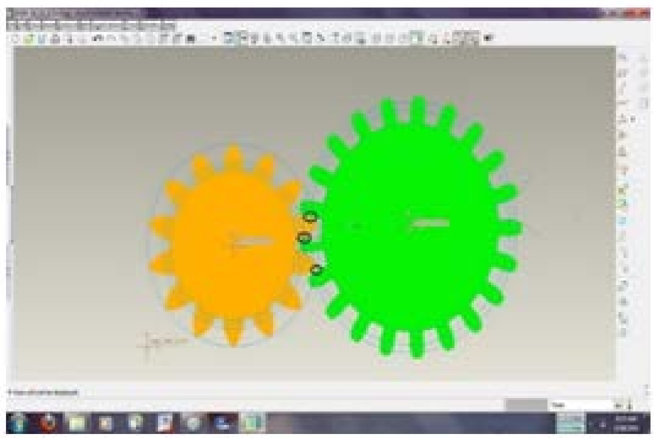

Fig.3: Meshed Model of Corrected Spur Gear

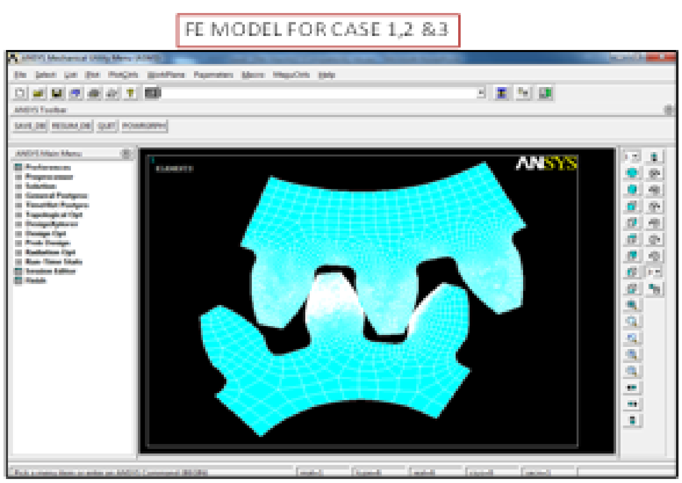

Fig.4: FEA model for case 1,2&3 With 1,2& 3 point of contact

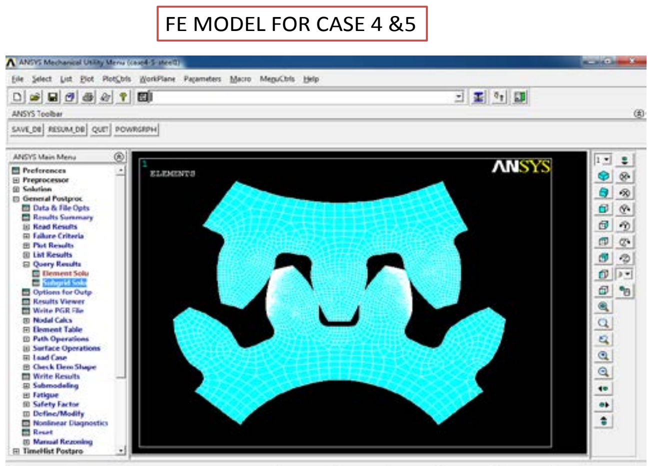

Fig.5: FEA model for case 4& 5

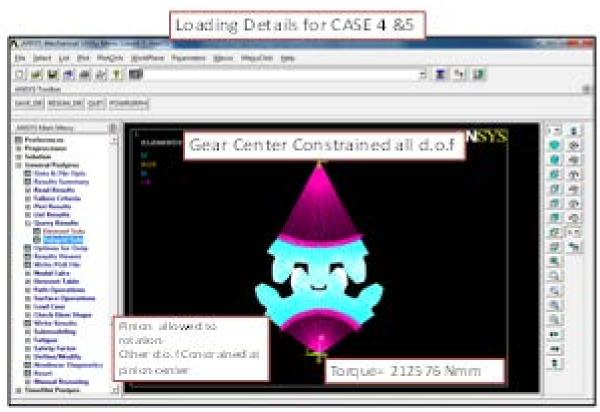

Fig.6: Boundary condition for case 1, 2, & 3

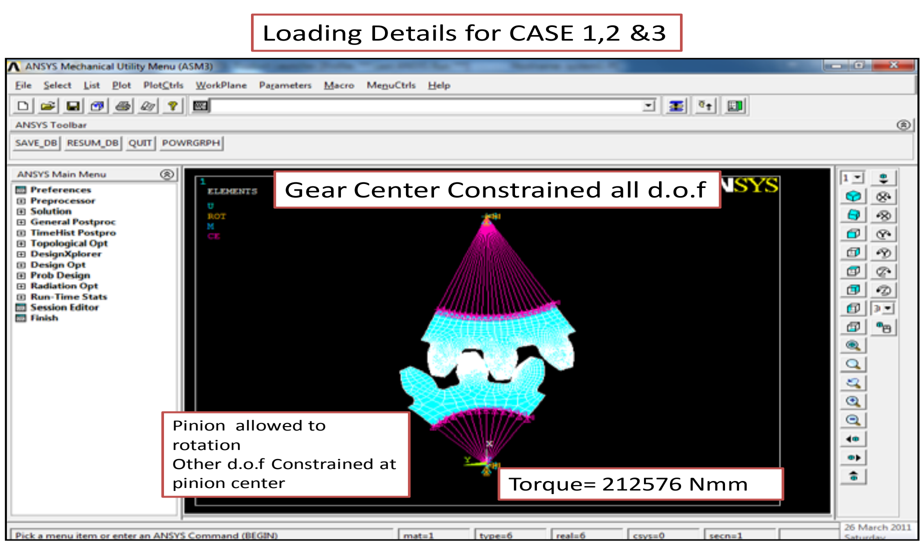

Fig.7: Boundary condition for case 1, 2, & 3 Standard Gear Pair

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Bending Stress & Contact Stress Analysis

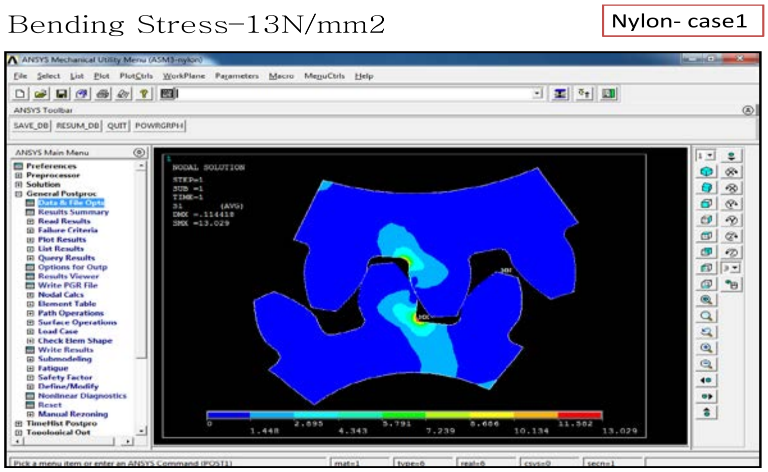

Fig.8: Bending stress case 1

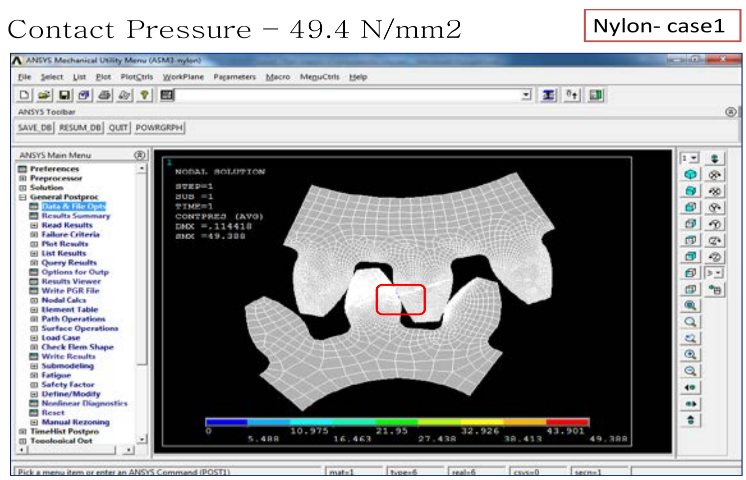

Fig.9: Contact stress case 1

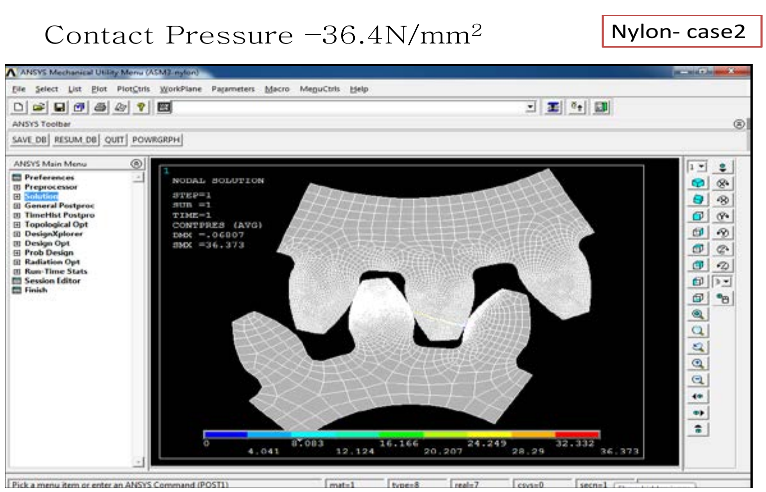

Fig.10: Contact stress case 2

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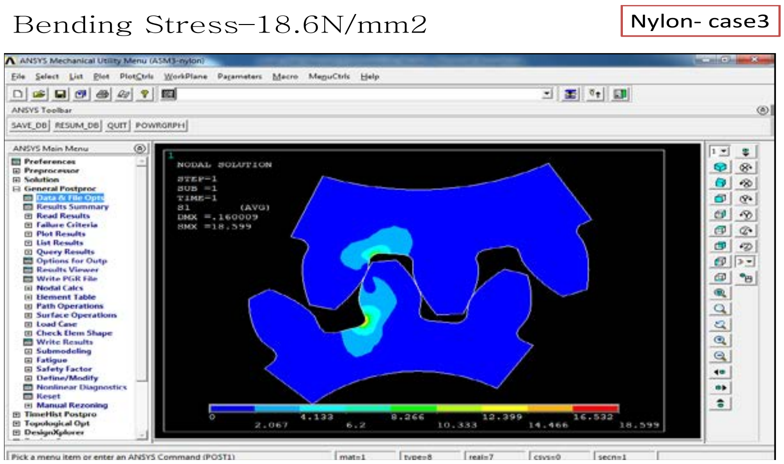

Fig.11: Bending stress case 3

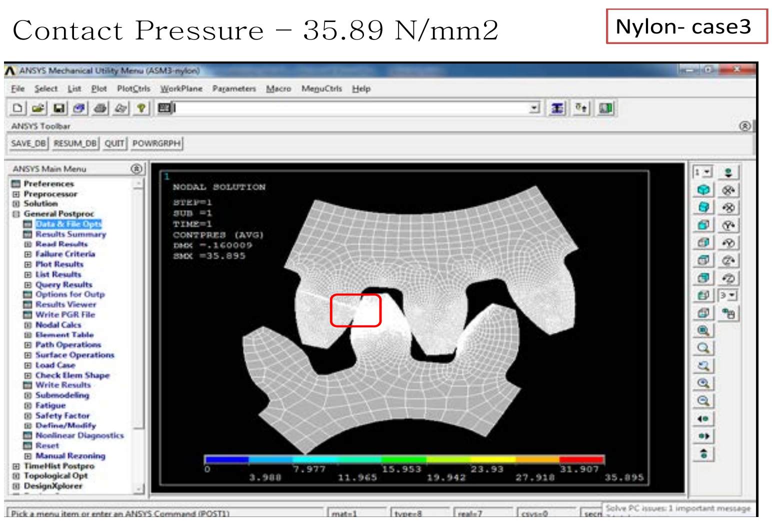

Fig.12: Contact stress case 3

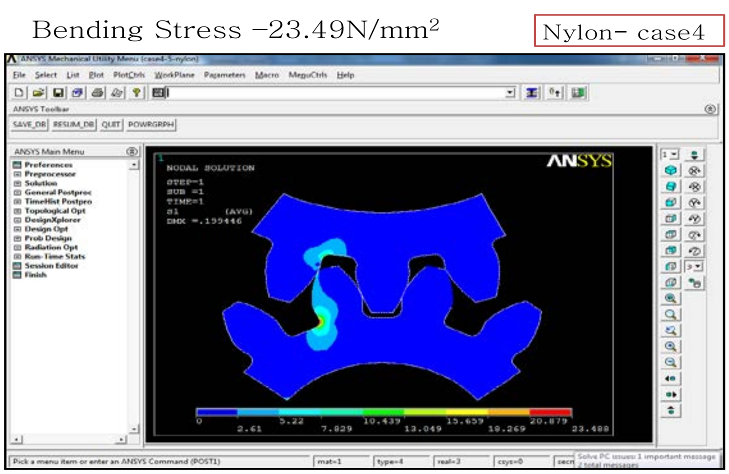

Fig.13: Bending stress case 4

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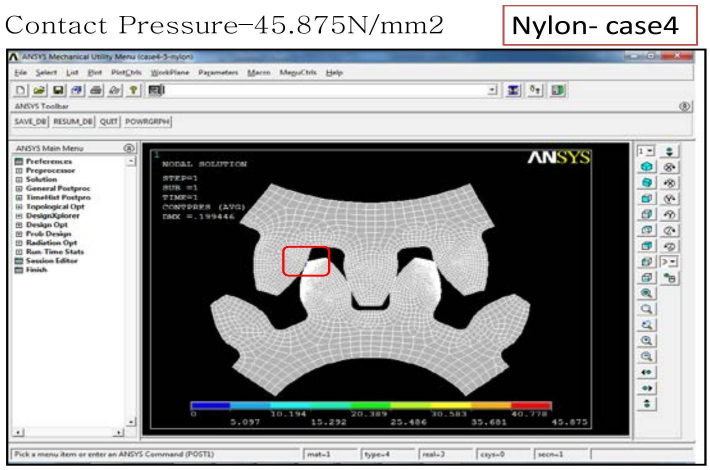

Fig.14: Contact stress case 4

Corrected Gear Pair Bending Stress & Contact Stress Analysis

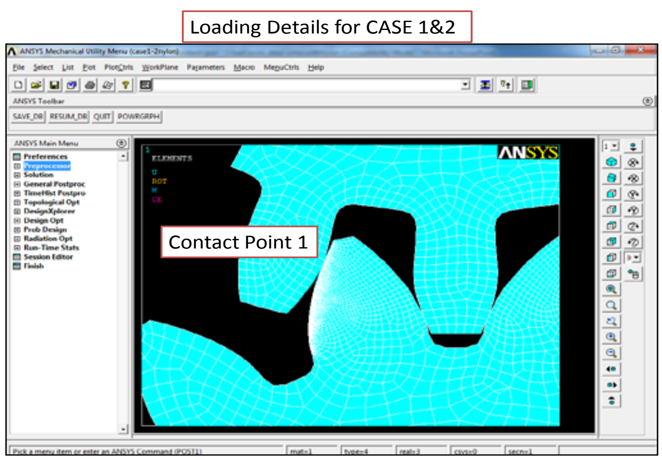

Fig.15: Contact point 1

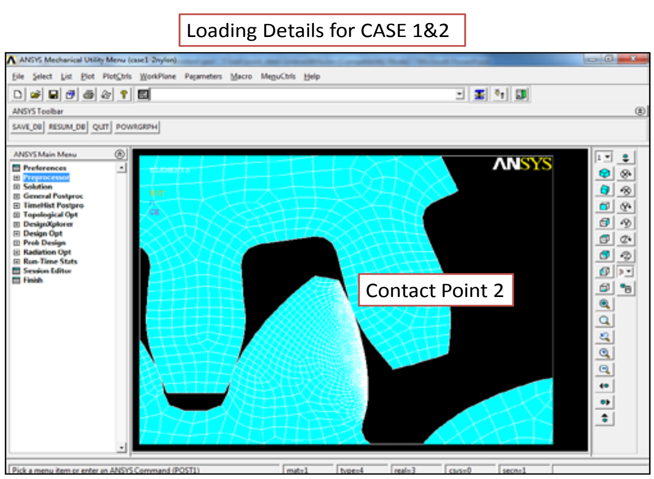

Fig.16: Contact point 2

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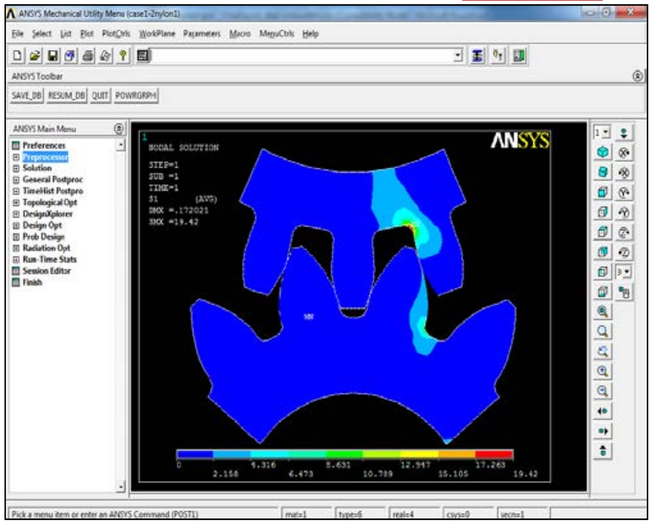
Bending Stress–19.42N/mm^2 Nylon- case2



Bending Stress–26N/mm^2 Nylon- case4

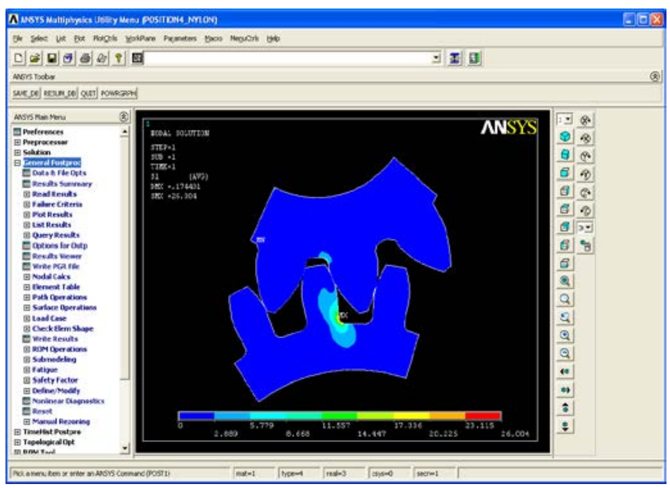
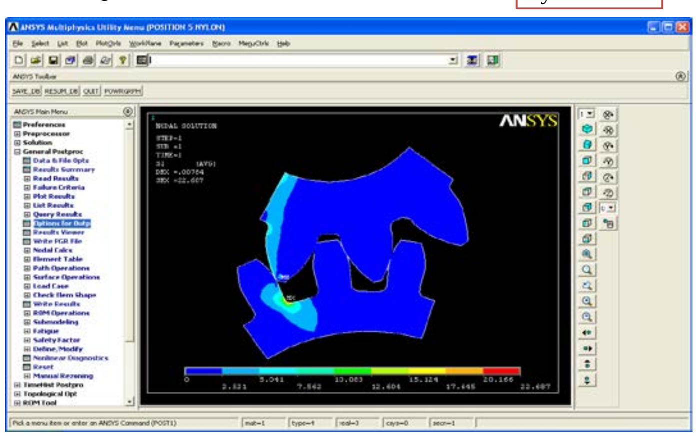


Fig.17: Bending stress case 1 & case 2

Bending Stress–22.68 N/mm^2 Nylon- case5



Contact Pressure – 40.9 N/mm^2

Nylon- case5

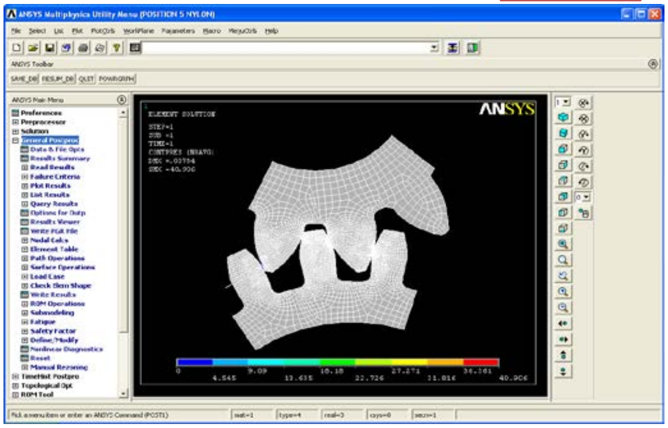


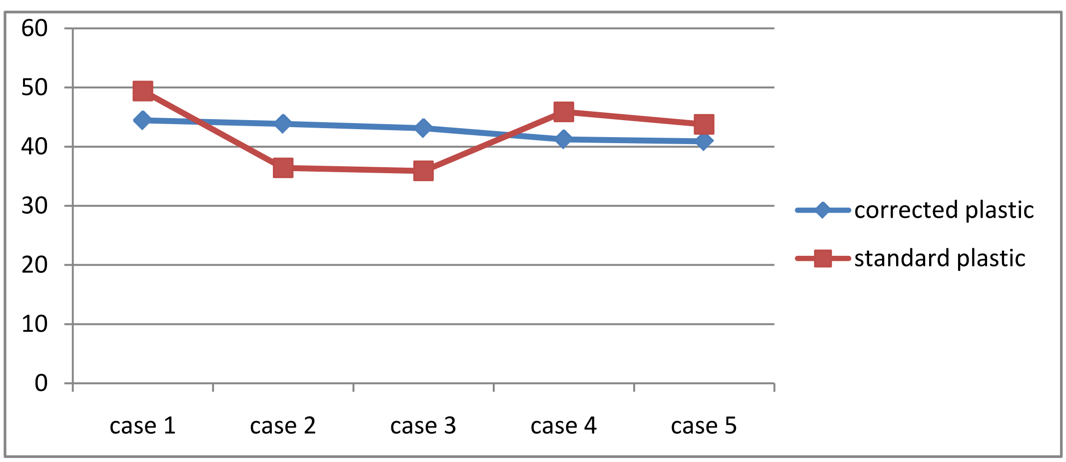
Fig.18: Bending stress & contact stress case 5

III. ANALYTICAL RESULT AND GRAPHICAL ILLUSTRATION

Table 1

Point of contact	Plastic material	Contact stress N/mm^2	Bending Stress N/mm^2
Case 1	Standard	49	13
	Corrected	44.45	17.75
Case 2	Standard	36.4	14.75
	Corrected	43.86	19.42
Case 3	Standard	35.89	18.6
	Corrected	43.12	20.54
Case 4	Standard	45.875	23.49
	Corrected	41.23	26
Case 5	Standard	43.76	23.36
	Corrected	40.9	22.68

Graph I:- Comparison of contact stress results between standard plastic and corrected plastic gear pair



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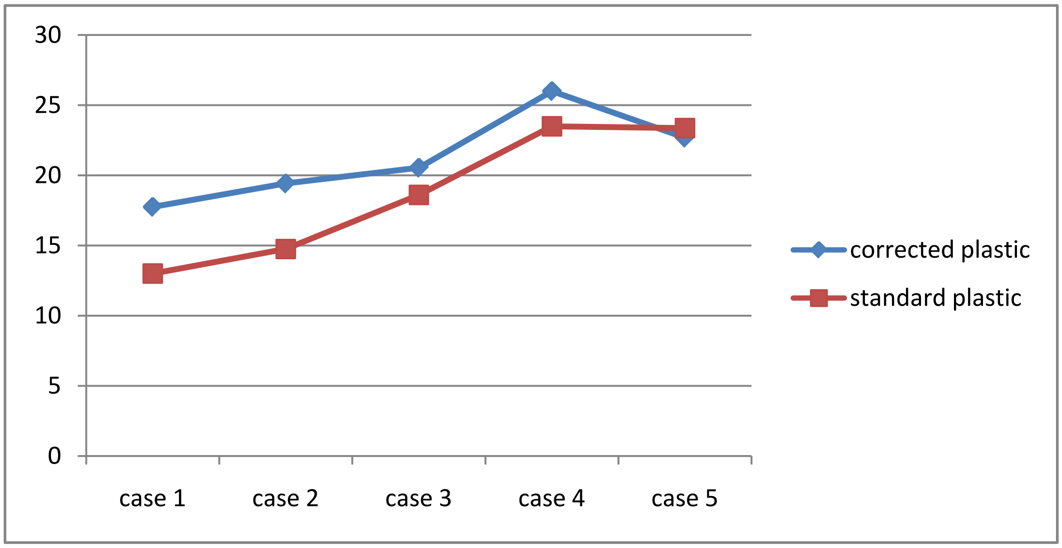
Graph II:- Comparison of Bending Stress Results between Standard Plastic And Corrected Plastic Gear Pair



IV. CONCLUSION

In this paper, the stress analysis of the standard and profile corrected plastic gear pair was done. The geometry of the gear was modeled in modeling software Pro-E. The meshing was carried out by HYPERMESH software, and the analysis was completed using ANSYS10software.It was seen from the results that the theoretical value of the bending stress does not agree with the analytical result for a standard gear. It was found to be high, but was within the permissible bending stress limit for a plastic material. Hence it is agreed that the profile corrected results obtained are also true. The Bending stresses for a standard and profile corrected tooth was found to be high. This is due to the fact of the material taken into consideration. Though the bending stress depends upon geometry, but due to load sharing, deflection is more for plastic material it has a tendency to bend more for the calculated torque. The torque can thus be revised for suitable application. The maximum contact stress occurs in case 4, which is the pitchpoint. Here the influence of young's modulus plays a role. Under static contact analysis, the contact stresses were found to be within limits calculated theoretically but the results may vary severely for dynamic loading.

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