

Design and Weight optimization of Pinion by using FEA Method

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Abstract: A spur gear is generally subjected to bending stress which causes failure. However it is observed that performance of the spur gear is not satisfactory in certain applications and therefore it is required to explore some alternate materials to improve the performance of the spur gears. Composite materials provide adequate strength with weight reduction and they are emerging as a better alternative for replacing metallic gears. This work includes design and weight optimization of output pinion of dual rotor wind turbine. In Cad software modeling is done and for post processing we are used Ansys software. We are compared the analytical result with FEA result. For the design and weight reduction we are replaced existing material with composite material. We are analyzed bending stress, total Strain and weight for both existing material and composite material. From the analysis it is concluded that bending stress developed in pinion gear tooth subjected to fixed load is independent of material.

Key words: Dual rotor wind turbine, glass fiber, Finite Element Method, Design and Weight Optimization.

1. INTRODUCTION

Gears are mechanical components used for transmitting motion and torque from one shaft to another. Ever since invention of rotating machines, gears existed. During early states that around 2600 BC Chinese used gears to measure the speeds of chariots. In 250 B.C. Archimedes used a screw to drive toothed wheels which were used in engines of war. In 4th Century B.C., Aristotle used gears to stimulate astronomical ratios. Greek and Roman literature mention the extensive use of gears in clock of cathedrals and ecclesiastical buildings. During early centuries gears made of either wood or stone teeth set in wood. Later during metal ages Iron, Bronze or tin were used instead of stone. There is no standard procedure for gear manufacturing until 1835 when English inventor Whitworth patented the first gear hobbing process. Although gear manufacturing has achieved lots of advancements during its evolution, however the failure of gear due to bending and contact stress still remained challenge for designers and manufacturers until 1892.

In 1892 the Philadelphia Engineers Club first recognized Wilfred Lewis presentation of stresses on the gear tooth and it still serve as the basis to determine the gears stress. The Lewis bending equation has a lot of drawbacks which

include 1. Load on gear tooth is dynamic and is influenced by pitch line velocity 2. The entire load is carried on single tooth. 3. The equation of application of load is not true as the load is shared by the tooth. 4. The stress concentration factor at tooth fillet is not considered. Apart from this aspect for the application of gears in dual rotor wind turbine gear box, bending stress, total deformation and weight of gears are considered for the efficiency of wind turbine. As turbine is increased, in this work bending stress, total deformation and weight of gear is lesser final output of wind turbine is increased. As bending stress, total deformation and weight of gear is lesser final output of weight of pinion is analyzed. Analytical results of existing materials are compared with FEA results of composite material. For the existing material cast iron, plain carbon steel results are compared with software based results of glass fiber as composite material are calculated and compared. From this analysis we can say that Ansys software can be used for analysis comparison purposes.

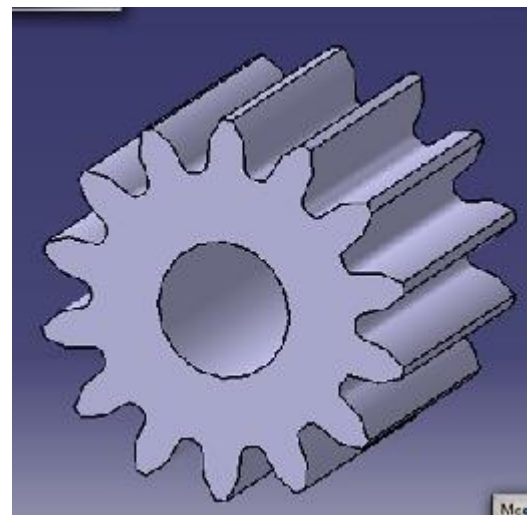


Fig. 1 Pinion of Dual rotor wind Turbine Gearbox.

2. LITERATURE REVIEW

Mrs. Shinde S.P., Mr. Nikam A.A. , Mr. Mulla T.S have performed work on Spur Gear by using Finite Element Analysis method. In this paper bending stress of gear is calculated theoretically i.e. conventional method and these results are compared with FEA results. This has improved gear analysis and computer simulations. Numerical

methods can potentially provide more accurate solutions since they normally required much less restrictive assumptions. They have created 3D model by Pro-E software can generate models of three dimensional gears easily. It is concluded that sets of gears having same module and pressure angle can be created and assembled together. By creating 3D deformable body of spur gear results are checked with theoretical calculation data ,simulation results have good agreement with theoretical results. This study provides future studies on contact stresses. From this study numerically obtained values of distributions are comparable with theoretical results.

M. Keerthi, K. Sandya, K. Srinivas [2] In this project work is done on composite material spur gear, values of deformation and vonmises stresses are calculated for both composite material and existing material. Static analysis is performed to determine the deformation and Von-misses stresses. Modal analysis is performed to determine the natural frequencies and mode shapes. The results were validated with theoretical calculations by Lewis equation. It was concluded that the stress values are calculated for composite materials is approximately same as compared to the structural steel, gray cast iron and aluminum alloy. So from these analysis results, we conclude that, the stress induced, deformation and weight of the composite spur gear is almost same as compared to the structural steel spur gear, gray cast iron spur gear and aluminum alloy spur gear. So, Composite materials are capable of using in automobile vehicle gear boxes instead of existing cast steel gears with better results.

Mr. Kishor N. Naik, Prof. Dhananjay Dolas, [3] The main objective of this paper is to analyze the bending stresses occur on the gear tooth profile of gear used in gear box of special purpose machine also effect on bending stress by variation of the gear parameters. Face width and root radius are taken gear parameters, how stress redistribution are taken place by varying this parameter studied. The stresses are calculated with the help of the FEA this result are compared with the stresses calculated by Lewis equation. For this work parametric modeling is done using Pro-E 5.0 and for analysis ANSYS 12.0 workbench is used. This work helpful to conclude Effect of bending stress on gear tooth profile by variation of gear parameters also give the comparison of FEM method with analytical calculation. Effect of bending stress on gear tooth profile by variation of gear parameters also give the comparison of FEM method with analytical calculation.

Vivek Karaveer, Ashish Mogrekar and T. Preman Reynold Joseph, [4] This paper presents the stress analysis of mating teeth of spur gear to find maximum contact stress in the gear teeth. The results obtained from Finite Element Analysis (FEA) are compared with theoretical Hertzian equation values. For the analysis, steel and grey cast iron are used as the materials of spur gear. The spur gears are sketched, modeled and assembled in ANSYS Design

Modeler. As Finite Element Method (FEM) is the easy and accurate technique for stress analysis, FEA is done in finite element software ANSYS 14.5. Also deformation for steel and grey cast iron is obtained as efficiency of the gear depends on its deformation. The results show that the difference between maximum contact stresses obtained from Hertz equation and Finite Element Analysis is very less and it is acceptable.

S.Mahendran, K.M.Eazhil, L.Senthil Kumar, [5] work has been carried out on spur gear to study the weight reduction and stress distribution for cast steel and composite materials. In this project analysis of both cast steel and composite materials are analyzed in the application of gear box used in automobile vehicles. From these analysis we got values stress, deformation, and weight of composite materials are less as compared to cast steel spur gear.

Pradeep Kumar Singh, Manwendra Gautam, Gangasagar and Shyam BihariLal, [6] A pair of spur gear tooth in action is generally subjected to two types of cyclic stress: contact stress and bending stress including bending fatigue. Both stress may not attain their maximum values at the same point of contact fatigue. These types of failure can be minimized by analysis of the problem during the design stage and creating proper tooth surface profile with proper manufacturing methods. In this paper using ansys work bench software, bending stress, contact stress and static load on the tooth of spur gear drive is found.

Devikant Baviskar, [7] This work presents the design and Analysis of polymer PEK spur gear and Comparison of results of PEK with Metallic Cast Iron under limited loading conditions. Application of Plastic gear reduces the weight and noise vibration. Analytical Method is used to calculate Tooth load with help of Lewis equation & dynamic tooth load with help of Buckingham equation. Gear profile modeling is done by using SOLIDWORKS 2015.

3. OBJECTIVES

- ° To propose a composite material for spur gear which will have increased strength and durability as compared to conventional material.
- ° To prove the conventional methods of analysis are comparable with FEA methods.
- ° To reduce the weight of gear in application of dual rotor wind turbine for improvement of turbine efficiency.

4. MATERIAL SELECTION

4.1 Cast Iron

Table 1. Material properties for Cast Iron

PROPERTIES	VALUES
ELASTIC MODULUS,E	1.2 x 10 ⁵ MPa
POISONS RATIO,μ	0.3
YIELD STRESS	250 MPa
ULTIMATE TENSILE STRESS	320 MPa
DENSITY,ρ	7850 kg/m ³

4.2 Cast Iron

Table 2. Material properties for Carbon Steel

PROPERTIES	VALUES
ELASTIC MODULUS,E	1.65x10 ⁵ MPa
POISONS RATIO,μ	0.26
YIELD STRESS	250 MPa
ULTIMATE TENSILE STRESS	320-350 MPa
DENSITY,ρ	7200 kg/m ³

4.1 Cast Iron

Table 3. Material properties for Glass Fiber

PROPERTIES	VALUES
ELASTIC MODULUS,E	1.9X10 ⁵ MPa
YIELD STRESS	N/A
ULTIMATE TENSILE STRESS	1600 MPa
DENSITY,ρ	1600 kg/m ³

5. ANALYTICAL ANALYSIS

5.1 Specifications of Dual Rotor Wind Turbine: [literature]

Power (P) = 4309.65 W
 Speed (n_g) = 195 RPM
 Gear Ratio(i) = 5
 $i = n_p / n_g$
 $5 = 195 / n_g$
 Speed (n_p) = 975 RPM
 Power (P) = 2π nt / 60
 $4309.65 = (2π \times 975 \times t) / 60$
 Torque (t) = (4309.65 x 60) / (2π x 975)
 t = 42200 N-mm

Step-1) Deciding the weaker between Pinion & Gear since same material is used for the Pinion & Gear, Pinion is weaker than the Gear.

Assume no. of teeth on pinion Z_p = 14 (Practically) φ = 20°
 From design data book Lewis form factor for the 14 teeth
 Y_p = 0.276
 Also no. of teeth on Gear ,
 $i = Z_g / Z_p$
 $5 = Z_g / 14$

Z_p = 70

Step-2) Estimation of module based on beam strength

Assume pitch line velocity v = 5 m/s
 Velocity factor C_v = 3/3+v = 3/3+5 = 3/8
 From design data book service factor C_s = 1.5
 Module (m) = 60*10⁶ / π { (KW* C_s* f_s) / Z_g*n_p C_v (b/m)*(S_{ut}/3)* Y_p }
 Also factor of safety f_s=2, b= 10m, S_{ut} = 1600 N/mm²
 Module (m) = 60*10⁶ / π { (4.30965* 1.5* 2) / 975 *14 (3/8)*10*(1600/3)*0.276 }
 Module (m) = 3.2001 mm

Step 3) Selection of module

From design data book, standard value of the module is 3.5 mm
 d = Pitch circle dia. of pinion = No. of teeth x module = 14 x 3.5 = 49 mm
 Face width b = 10m = 10*3.5 = 35 mm
 t = P_t * (d/2)
 P_t = t / (d/2)
 P_t = 42200 / 24.5
P_t = 1722.44 N
 Where 'P_t' is the Tangential load
 Using Lewis equation
 Tangential load

$$P_t = m b \sigma_b Y$$

$$1722.44 = 3.5 \times 54 \times \sigma_b \times 0.276$$

$$\sigma_b = 1722.44 / 0.276 \times 3.5 \times 54$$

$$\sigma_b = 33 \text{ N/mm}^2$$

The Maximum allowable stress as per the design of the desired Pinion gear is,
= 33N/mm² (Induced Stress)

Where σ_b = permissible bending stress
 Y' is the Lewis form factor

'b' is the face width of the gear
 'd' is the pitch circle diameter of the gear

Allowable stress of Cast iron (high grade) =Ultimate tensile strength / 2
 = 320 / 3
 = 106.67 N/mm²
33 N/mm² < 106.67 N/mm²

Allowable stress of Plain Carbon Steel =Ultimate tensile strength /1.5
 = 460/3
 = 153.34 N/mm²
33N/mm² < 153.34 N/mm²

Allowable stress of Glass Fiber =Ultimate tensile strength / 3
 = 1600/3
 = 533.34 N/mm²
33N/mm² < 533.34 N/mm²

So the design is safe

Note that the bending stress developed in a Pinion gear tooth subjected to a fixed load is independent of the material properties, while the deflection depends on the material property E. Hence the bending stress is same whether it is made of a Cast Iron, Plain Carbon Steel and Glass Fiber, but the strain would be different.

5.2 Calculations of bending stress and strain at loading Condition.

5.2.1 Calculations for Cast Iron Pinion Gear

- Bending Stress at $P_t=1722.44 \text{ N}$

$$P_t = m b \sigma_b Y$$

$$1722.44 = 3.5 \times 35 \times \sigma_b \times 0.276$$

$$\sigma_b = 1722.44 / 0.276 \times 3.5 \times 35$$

$$\sigma_b = 50.94 \text{ N/mm}^2 \dots\dots\dots \text{bending stress}$$

-Strain at $P_t=1722.44 \text{ N}$

$$\sigma_b = E \times \epsilon$$

$$50.94 = 1.2 \times 10^5 \times \epsilon$$

$$\epsilon = 0.42 \times 10^{-3} \dots\dots\dots \text{strain}$$

5.2.2 Calculations for Plain Carbon Steel Gear

- Bending Stress at $P_t=1722.44 \text{ N}$

$$P_t = m b \sigma_b Y$$

$$1722.44 = 3.5 \times 35 \times \sigma_b \times 0.276$$

$$\sigma_b = 1722.44 / 0.276 \times 3.5 \times 35$$

$$\sigma_b = 50.94 \text{ N/mm}^2 \dots\dots\dots \text{bending stress}$$

-Strain at $P_t=1722.44 \text{ N}$

$$\sigma_b = E \times \epsilon$$

$$50.94 = 2.1 \times 10^5 \times \epsilon$$

$$\epsilon = 0.267 \times 10^{-3} \dots\dots\dots \text{strain}$$

5.2.3 Calculations for Glass Fiber Pinion Gear

- Bending Stress at $P_t=1722.44 \text{ N}$

$$P_t = m b \sigma_b Y$$

$$1722.44 = 3.5 \times 54 \times \sigma_b \times 0.276$$

$$\sigma_b = 1722.44 / 0.276 \times 3.5 \times 54$$

$$\sigma_b = 50.94 \text{ N/mm}^2 \dots\dots\dots \text{bending stress}$$

-Strain at $P_t=1722.44 \text{ N}$

$$\sigma_b = E \times \epsilon$$

$$50.94 = 1.9 \times 10^5 \times \epsilon$$

$$\epsilon = 0.268 \times 10^{-3} \dots\dots\dots \text{strain}$$

5.2.4 Evaluation of Mass

5.2.4.1 For Cast Iron Material

$$m = \text{density (}\rho\text{)} \times \text{volume (v)}$$

$$m = 7850 \times 6.67 \times 10^{-5}$$

$$m = 0.524 \text{ Kg}$$

5.2.4.2 For Plain Carbon Steel Material

$$m = \text{density (}\rho\text{)} \times \text{volume (v)}$$

$$m = 7200 \times 6.65 \times 10^{-5}$$

$$m = 0.4752 \text{ Kg}$$

5.2.4.3 For Plain Glass Fiber Material

$$m = \text{density (}\rho\text{)} \times \text{volume (v)}$$

$$m = 1600 \times 6.67 \times 10^{-5}$$

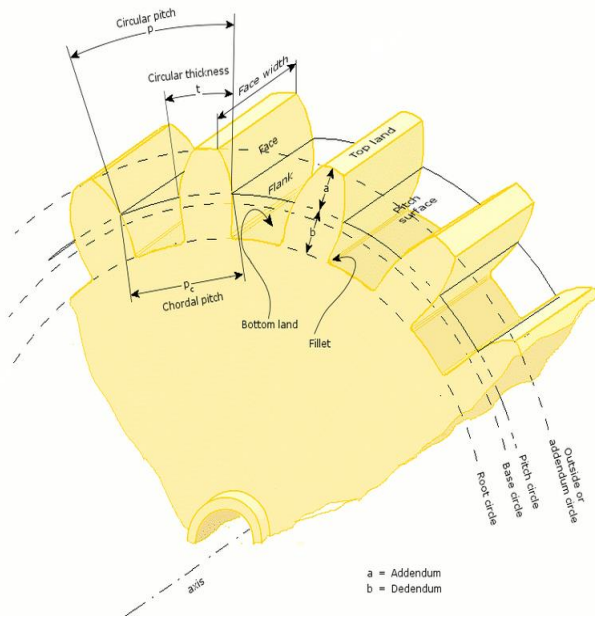
$$m = 0.107 \text{ Kg}$$

6. SIMULATION WORK

6.1 Cad Model generation:

Dimensions are required for calculating of boundary conditions. Hence its CAD model is necessary. Dimensions are drawn by design procedure for gears. CAD model then is made by the commands in CATIA of Pad, pocket, fillet, and geometrical selections in part design module. Parametric generation of drawings will help to get the dimensions useful in forces calculations in static loading conditions on gears.

6.2 Design of Pinion gear



Design of output Pinion gear:

Number of teeth's (z) = 14

Module (m) = 3.5 mm

Pressure angle standard (α) = 20°

Pitch circle diameter (D) = z*m
 = 14*3.5
 = 49 mm

Base circle diameter (D_b) = D cos α
 = 49*cos (20°)
 = 46.045 mm

Outer diameter (D_o) = (z+2)*m = (14+2)*3.5 = 56mm

Diametral pitch (P_d) = $\frac{\text{Number of teeth } s+2}{\text{Outer diameter}}$
 = $\frac{N+2}{D_o}$
 = $\frac{14+2}{56}$
 = 0.286 mm

Circular pitch (C) = m * 3.142 = m * 3.142
 = 3.5 * 3.142 = 10.997 mm

Addendum (a) = $\frac{1}{\text{Diametral pitch}} = \frac{1}{P_d}$
 = $\frac{1}{0.286}$
 = 3.5 mm

Dedendum (b) = $\frac{1.157}{\text{Diametral pitch}}$
 = $\frac{1.157}{P_d}$
 = $\frac{1.157}{0.286}$
 = 4.04 mm

Total depth (d_t) = $\frac{2.157}{\text{Diametral pitch}}$
 = $\frac{2.157}{P_d}$
 = $\frac{2.157}{0.286}$
 = 7.542 mm

Working depth (d_w) = $\frac{2}{\text{Diametral pitch}}$
 = $\frac{2}{P_d}$
 = $\frac{2}{0.286}$
 = 6.993 mm

Thickness of tooth (T) = $\frac{1.5708}{\text{Diametral pitch}}$
 = $\frac{1.5708}{P_d}$
 = $\frac{1.5708}{0.286}$
 = 5.492 mm

Face width (F) = $\frac{8}{\text{Diametral pitch}} = \frac{8}{P_d} = \frac{8}{0.286} = 28$
 mm

Face width (F) = $\frac{16}{\text{Diametral pitch}} = \frac{16}{P_d} = \frac{16}{0.286} = 56$
 mm

Face width should be in between the range 28 and 56.

Face width (F) = 4 * 3.142 * m
 = 4 * 3.142 * 3.5
 = 43.988 mm

Clearance (C) = $\frac{0.2}{\text{Diametral pitch}} = \frac{0.2}{P_d}$
 = $\frac{0.2}{0.286}$
 = 0.7 mm

Root diameter = $\frac{\text{Number of teeth } s-2}{\text{Diametral pitch}} = \frac{N-2}{P_d}$
 = $\frac{14-2}{0.286}$
 = 42 mm

6.3 Parametric Modeling of Spur Gear

Parameters

- No of teeth (N) = 14
- Module (m) = 3.5 mm
- Pitch Circle radius R_p = m* N/2
- Clearance Circle radius R_b = 0.94 * R_p
- Addendum Circle radius R_a = R_p + m
- Dedendum Circle Radius R_d = R_p - 1.25 * m
- Pressure angle (α) = 20 degrees.....Angle1
- Angle 2 = 90/N = 6.42 deg.
- Fillet radius = 0.39 * module

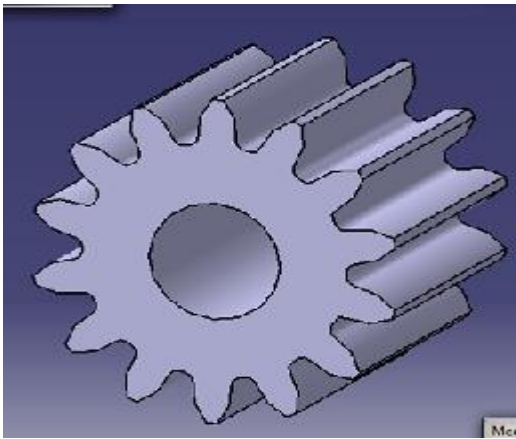


Fig.2 CAD Model

Max. Bending stress value for C.I. Pinion gear is 53.17 N/mm² which is less than max. Allowable stress hence design is safe.

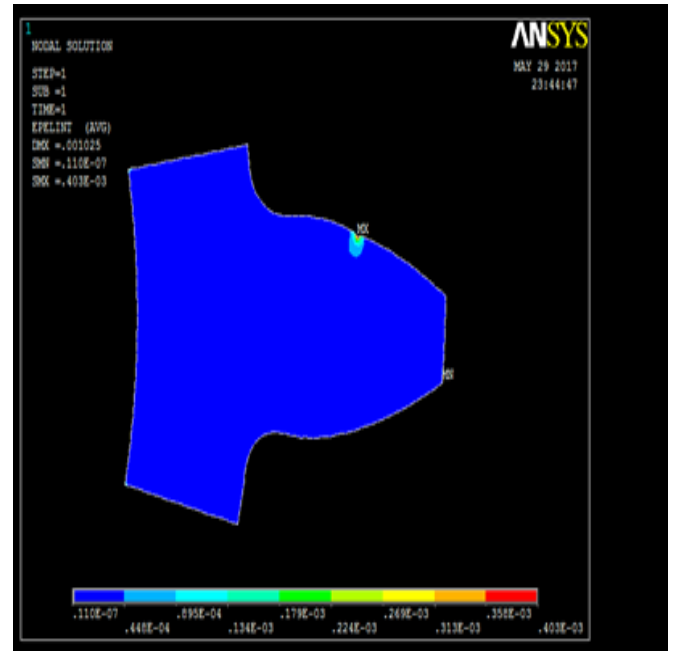


Fig.5 Strain 0.43×10^{-3} at load of 1722.44 N (Cast Iron)
From fig. value of strain for C.I. is 0.43×10^{-3}

6.4 Processing (Solution Stage)

Apply Boundary Conditions

Constrains : Define Structural Displacement Zero

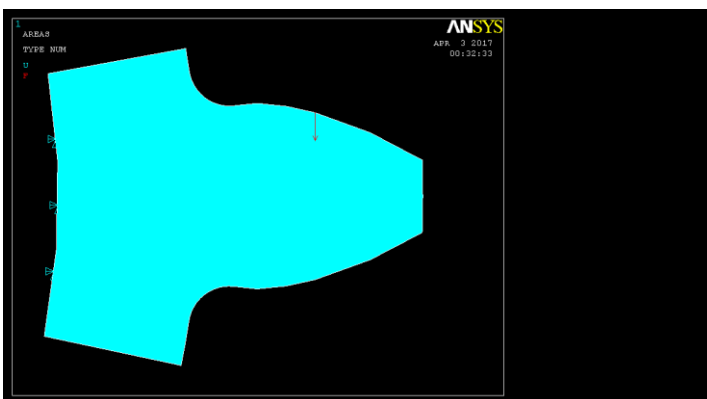


Fig.3 Constraints

A. Stress, Deformation and Weight Results for Cast Iron

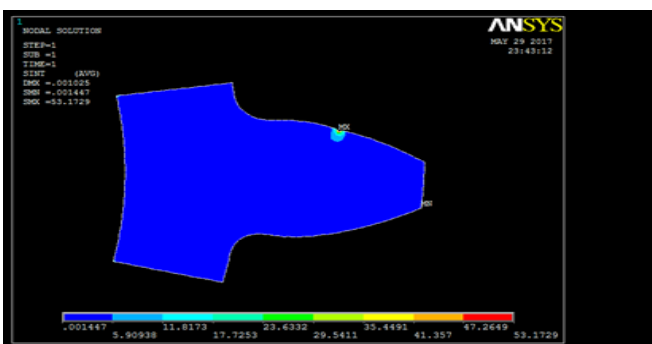


Fig. 4 Bending Stress 53.17 N/mm² at load 1722.44 N (Cast Iron)

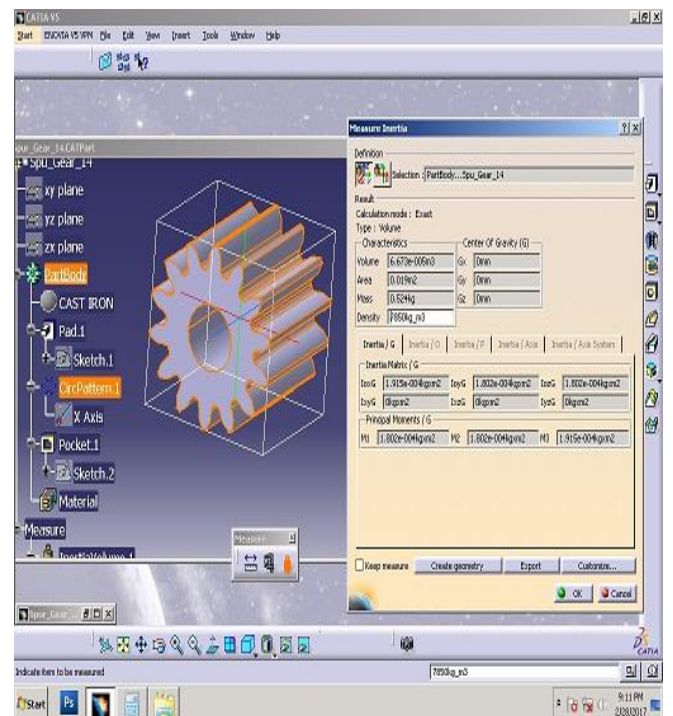


Fig.6 Weight of Pinion Gear in Catia software (Cast Iron) from fig. value of weight is 0.524 Kg.

B . Stress,Strain and Weight Results for Carbon Steel

C. Stress,Deformation and Weight Results for Glass Fiber

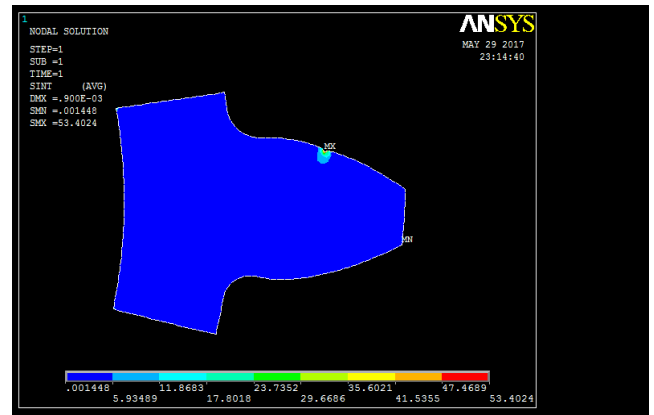
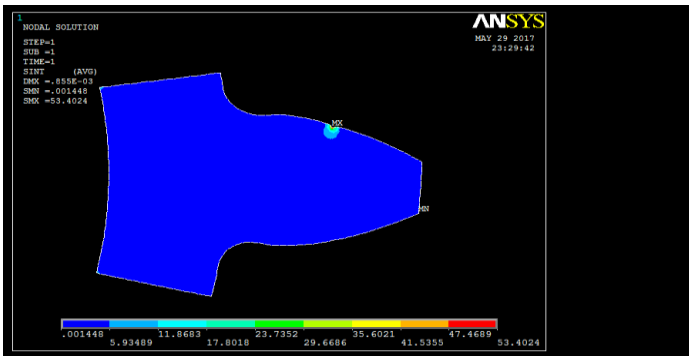


Fig. 7 Bending Stress 53.40 N/mm² at load 1722.44N (Carbon Steel)

Fig.10 Bending Stress 53.40 N/mm² at load 1722.44N (Glass Fiber)

Max. Bending stress value for Carbon Steel Pinion gear is 53.40 N/mm² which is less than max. Allowable stress hence design is safe.

Max. Bending stress value for Glass Fiber Pinion gear is 53.40 N/mm² which is less than max. Allowable stress hence design is safe.

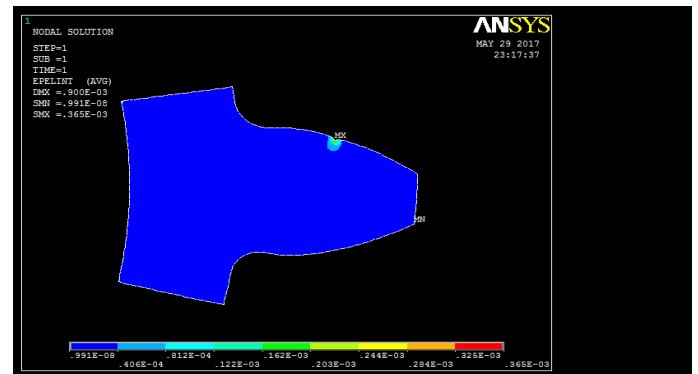
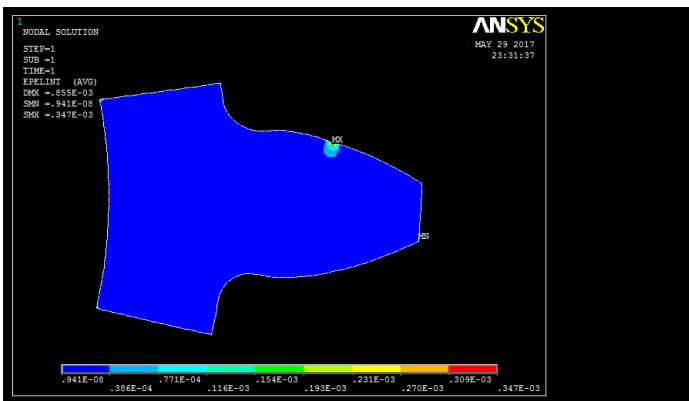


Fig. 8 Strain 0.347×10^{-3} at load 1722.44N (Carbon Steel) From fig. value of strain for Carbon steel is 0.347×10^{-3}

Fig.11 Strain 0.365×10^{-3} at load 1722.44N (Glass Fiber) From fig. value of strain for C.I. is 0.365×10^{-3}

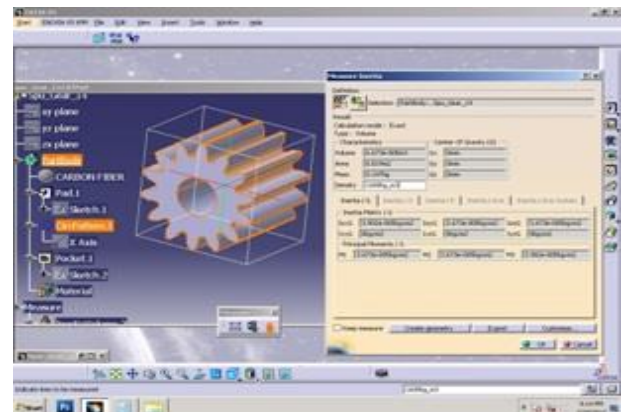
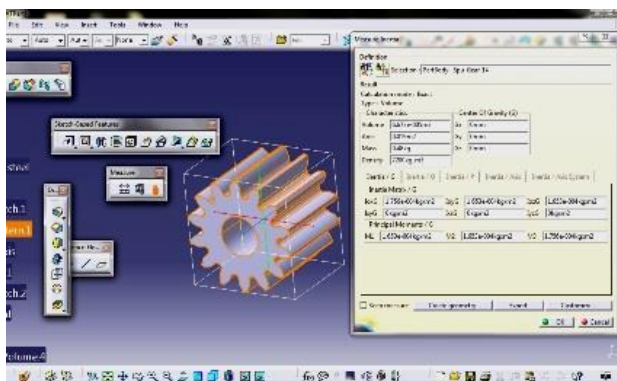


Fig.9 Weight of Pinion Gear in Catia Software (Carbon Steel) from fig. value of weight is 0.48 Kg

Fig.12 Weight of Pinion Gear in Catia Software (Glass Fiber) from fig. value of weight is 0.107 Kg.

7. RESULT VALIDATION

Table 4. Comparison of FEA and Analytical Result

Sr. No.	Material	FEA Max. Bending Stress	Analytical Stress
1	Existing Cast Iron	53.17 MPa	50.94 MPa
2	Existing Carbon Steel	53.40 MPa	50.94 MPa
3	Glass Fiber	53.40 MPa	50.94 MPa

Table 5. Comparison of FEA and Analytical Result

Sr. No.	Material	FEA Strain	Analytical Strain
1	Existing Cast Iron	0.43×10^{-3}	0.42×10^{-3}
2	Existing Carbon Steel	0.267×10^{-3}	0.347×10^{-3}
3	Glass Fiber	0.365×10^{-3}	0.271×10^{-3}

Table 6. Comparison of FEA and Analytical Result

Sr. No.	Material	Catia Software Weight	Analytical Weight
1	Existing Cast Iron	0.524 Kg	0.524 Kg
2	Existing Carbon Steel	0.48 Kg	0.475 Kg
3	Glass Fiber	0.107 Kg	0.107 Kg

8. CONCLUSION

- The objective of current work is to replace existing material with glass fiber composite material for pinion gear for gear box of dual rotor wind turbine.
- For design and weight optimization analytical and finite element method is applied for determining bending stress, strain.
- The obtained analytical results are compared with FEA results and found both results are comparable.
- Ansys software can be used for analysis purposes.
- From these analysis value of strain for glass fiber is less as compare to existing material for same loading conditions.

- Calculated value of bending stress is independent on material.
- Weight optimization is achieved drastically with glass fiber in place of existing material.

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