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Design and Manufacturing of a 4 Wheel Drive Gearbox of an off-road Vehicle (Atv)

Ayush Munot, Adarsh Bhavsar, Vaishnavi Wadhi, Y.A Kadam
Student, Department of Mechanical Engineering P.E.S's Modern College of Engineering Pune, Maharashtra, India
Assistant Professor, Department of Mechanical Engineering P.E.S's Modern College of Engineering Pune, Maharashtra, India

ABSTRACT
The purpose of this project is to conceptualize, design, and manufacture an advanced Four-Wheel Drive (4WD) system for an All-Terrain Vehicle (ATV), enhancing its off-road performance and maneuverability. The ATV is intended for recreational and utility purposes in diverse terrains, including rough trails, mud, sand, and rocky surfaces.
The proposed 4WD system aims to optimize traction, stability, and control by distributing power to all four wheels, thereby improving the vehicle's overall off-road capabilities. The design process involves a comprehensive study of existing ATV drivetrain technologies, a detailed analysis of terrain requirements, and the incorporation of innovative engineering solutions.
Key components of the project include the development of a robust transfer case, differential mechanism, and drive shafts to seamlessly transfer power from the engine to each wheel. Emphasis is placed on creating a system that adapts to varying terrain conditions, ensuring optimal torque distribution for improved traction without compromising on efficiency.
Computer-aided design (CAD) tools are employed for the creation of 3D models, allowing for virtual prototyping and simulations to validate the system's performance under different scenarios. Finite Element Analysis (FEA) is utilized to assess structural integrity and durability, ensuring the 4WD system can withstand the harsh conditions associated with off-road use.
The manufacturing process involves the selection of high-quality materials and precision machining techniques to achieve the desired strength-to-weight ratio and durability. Prototypes are rigorously tested in controlled environments to validate the design's functionality and performance.
The successful implementation of the 4WD system is expected to result in an ATV with superior off-road capabilities, offering users a versatile and reliable vehicle for various recreational and utility applications. This project contributes to the advancement of ATV technology, showcasing innovative solutions for enhancing off-road mobility and expanding the range of activities achievable with these vehicles.

Keywords: **Research Paper, Technical Writing, Science, Engineering and Technology**

I. INTRODUCTION

The design and manufacturing of a Four-Wheel Drive (4WD) system for an All-Terrain Vehicle (ATV) represents a pivotal project aimed at elevating off-road mobility to new heights. The significance of 4WD technology in off-road vehicles cannot be overstated, as it plays a fundamental role in enhancing traction, stability, and maneuverability across diverse terrains.

This project undertakes a comprehensive exploration of the engineering intricacies involved in creating a cutting-edge 4WD system, from conceptualization to prototyping and validation.

Off-road enthusiasts and utility operators demand vehicles that can navigate challenging terrains with finesse. The conventional Two-Wheel Drive (2WD) systems often fall short in providing the necessary traction on uneven surfaces, limiting the ATV's overall off-road potential.

The implementation of a 4WD system aims to address these limitations by distributing power to all four wheels, thereby improving grip and control.

1.1 Objectives:

The primary objective of this project is to design and manufacture a 4WD system that not only meets but exceeds the expectations of off-road enthusiasts and utility users. Key objectives include optimizing power distribution, ensuring adaptability to varying terrains, enhancing vehicle stability, and incorporating intelligent control systems for dynamic performance adjustments.

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2. KEY COMPONENTS OF THE 4WD SYSTEM:

2.1 Transfer Case: The transfer case serves as the core component responsible for directing power from the engine to the front and rear axles. Its design focuses on efficiency, durability, and the seamless transition between 2WD and 4WD modes.

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2.2 Gears and Gear Train: The gearbox incorporates a complex arrangement of gears and a gear train to transmit power from the engine to the wheels. These gears are meticulously designed to provide the necessary torque and speed ratios required for various driving conditions. Gear ratios can be adjusted to optimize performance for tasks such as climbing steep inclines, traversing rough terrain, or achieving higher speeds on flat surfaces.

2.3 Differential Mechanism: The differential is engineered to manage torque distribution between the front and rear wheels. A limited-slip differential is integrated to accommodate varying wheel speeds, preventing wheel slippage and optimizing traction.

2.4 Input and Output Shafts: The input shaft connects the gearbox to the engine's crankshaft, transmitting power into the gearbox. The output shafts extend from the gearbox to the front and rear axles, delivering power to the wheels. These shafts are essential for ensuring a direct and efficient transfer of torque, contributing to the vehicle's overall responsiveness and drivability.

2.5 Drive Shafts: Precision-engineered drive shafts transmit power from the differential to each wheel, ensuring a direct and efficient power transfer while accommodating suspension movements.

3. COMPUTER-AIDED DESIGN (CAD) AND SIMULATION:

Advanced CAD tools are employed to create detailed 3D models of the 4WD system. Simulation techniques, including Finite Element Analysis (FEA), are utilized to assess the structural integrity and performance of the components under different loads and terrain conditions.

3.1 Adaptive Control System: Intelligent control systems, driven by data from sensors such as accelerometers and wheel speed sensors, enable real-time adjustments to the 4WD system. This adaptive control enhances traction, stability, and responsiveness, contributing to an optimal off-road driving experience.

3.2 Manufacturing Process: The manufacturing process involves selecting high-strength materials and utilizing precision machining techniques. Prototypes undergo rigorous testing to validate the design's functionality, structural integrity, and durability, ensuring the final product meets the demands of off-road environments.

3.3 Sealing and Durability: Given the demanding nature of off-road driving, 4WD gearboxes are designed with robust sealing mechanisms to prevent the ingress of water, mud, and other contaminants. Additionally, they undergo rigorous testing to ensure durability under extreme conditions, including temperature variations, shocks, and vibrations.

4. PROBLEM STATEMENT.

Design a robust, efficient 4wd powertrain for a 10bhp BAJA ATV, including gearbox, differential and propeller shaft. The design should have optimized power transfer, adaptability to off-road terrain, durability and acceleration.

ENGINE RPM: - 3800RPM
ENGINE TORQUE: - 18.69 N/mm
TARGET SPEED: - 58KM/HR

4.1 SCOPE:

• Sustainable Consideration: Explore opportunities to make the transmission system more energy efficient.

• Continual Improvement: Consider opportunities for future enhancement and refinement to the 4WD transmission system. Stay updated with technological advancement in the field

• Lightweight Material: Integrating lightweight material like advanced composites, aluminium alloys, into the transmission system can reduce overall vehicle weight leading to full efficiency and performance.

4.2 Methodology:

Designing a 4WD gearbox for an off-roading vehicle involves defining specific requirements, selecting an appropriate transmission type with suitable gear ratios for diverse terrains, and incorporating a transfer case for varied gearing options. Material choice is crucial for durability, considering the challenging conditions off-road vehicles face. Implementing effective sealing, ensuring torque handling capacity, and conducting rigorous testing and simulation are vital steps. A well-designed cooling system is necessary for heat management during extended off-road use. Considerations should include weight, cost, and prioritizing maintenance and serviceability to ensure a robust and practical gearbox design.

5. CALCULATIONS: -

Engine Rpm = 3800 rpm

Consider the target speed = 56km/hr = 15.56m/s

Dimension of wheel = 23x7-10

Wheel end RPM = $\frac{60*V}{2\pi r} = \frac{60}{2*3.14*292*10^{-3}}$ x15.56

=508.7 $\approx \mathbf{509\ rpm}$

$\therefore \boldsymbol{wheel\ end\ RPM = N = 509\ rpm}$

- Continuous Variable Transmission (CVT)

Low end Ratio:- 3.9

High End Ratio:- 0.9

- Gear box Ratio :- $\frac{\frac{engine\,rpm}{High\,end\,ratio}}{Wheel\ end\ rpm} = \frac{\frac{3800}{0.9}}{509} = 8.29$

$\therefore$ $\boldsymbol{Gear\ box\ Ratio = 8.29}$

No. of Stages of Gearbox = 2

By taking square root,

Stage 1 = 2.879

Stage 2 = 2.879

Maximum Engine Torque = 18.69 Nm

=18690 Nmm

Final Drive Ratio = Low end ratio x Gearbox Ratio

= 3.9 x 8.29

= 32.331

Torque on input shaft (T_{in}) = 18690 x 3.9

[Engine Torque x Low end ratio] = 72891 Nmm

$\therefore$ $\boldsymbol{Tin = 72891\ Nmm}$

- Material Used: - 20MnCr5

  20MnCr5 steel is commonly used in the production of gears due to its ability to be case-hardened and its excellent wear resistance. This makes it well-suited for gears that are exposed to high levels of wear and tear.

S_{ut} = 980 N/mm^2 S_{yt} = 750N/mm^2

BHN = 248

Where,

S_{ut} = Ultimate tensile strength

S_{yt} = Yield strength

Consider module of spur gear (m) = 2mm &

No. of Teeth on Pinion (Z_p) = 20

$\therefore$ No. of teeth on gears (Z_g) = Z_p x Stage 1 ratio

= 20 x 2.8979

= 57.58 ≈ 58

$\therefore$ Z_g = 58

Let,

Pressure Angle = $\emptyset$ = 20^0 --- ($\boldsymbol{V.B.Bhandari}$)

- Lewis form factor

$Y = 0.484 - \frac{2.87}{zp} = 0.3405$

$\therefore$ Y=0.3405

- Beam Strength $(\boldsymbol{Sb}) = m * b * \sigma b * Y$

Where, m=module

B=face width=10m

$\boldsymbol{Sb}$= S_{ut}/3, Permissible Bending Stress N/mm^2

Y = Lewis form factor

$\therefore$ $\boldsymbol{\delta b = 4449.2\ N}$

$Q = \frac{zg*2}{zg+zp} = 1.4871$

$K = 0.16 \text{ x}[\frac{BHN}{100}]^2 = 0.9840$

- Wear strength

S_w = b x Q x dp x K ------[dp = PCD of pinion = z_p * m]

S_w = 1170.64512 N

For Input Shaft :-

$P_t = \frac{2Tin}{zp*m} = \frac{2*72891}{20*2} = \boldsymbol{3644.55\ N}$

Where, P_t=Tangential Component

$P_r = P_t * \tan\emptyset$

= 3644.55 x tan20

= 1326.507 N

Where, P_r = Radial Component

$\therefore$ For Stage 1 [For spur gear]

Z_p = 20

Z_g = 58

m = 2

Similarly, for Stage 2

Z_p = 20

Z_g = 58

m = 2

As the ratio is same for stage 1 and stage 2

Now,

Torque on Intermediate Shaft (T_{int})

= 72891 x 2.879

=209853.1789 Nmm

$P_t = \frac{2*Tint}{zp*m} = \frac{2*209853.189}{20*2} = \boldsymbol{70492.6594\ N}$

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$P_r = P_t$ x tan(20) = 38190.157

Bevel Gear Calculation :-

Consider module =3mm

$Z_p = 16$

Going by the ratio of 1.25

$Z_g = 20$

Material used is same

$S_{ut} = 980\ N/mm^2$ $S_{yt} = 750 N/mm^2$

$\boldsymbol{\gamma p = tan^{-1}[\frac{zp}{zg}] = tan^{-1}[\frac{16}{20}] = 38.65°}$ [Pitch cone angle on pinion]

$\boldsymbol{\gamma p = tan^{-1}\left[\frac{zp}{zg}\right] = tan^{-1}\left[\frac{20}{16}\right] = 51.34°}$ [Pitch cone angle on gear]

$Z_p' = \frac{zp}{cos\gamma p} = 20.48$ ----[virtual number of teeth on pinion]

$Z_g' = \frac{zg}{cos\gamma g} = 32.01$------ [virtual number of teeth on gear]

$Y_p' = 0484 - \frac{2.87}{zp'} = 0.3438$ --- [Lewis form factor on Virtual number of teeth]

$d_p = m \times z_p = 48$ $d_g = m \times z_g = 60$

$P_t = \frac{Tp}{Rpm}$ where T_p = Torque on Int (Pinion) R_{pm} Mean Radius of Pinion.

$R_{pm} = 19.94$ mm , $T_p = 209853.189$ Nmm

$P_t = 10524.23$N

Separating Force = 3830.50 N

$P_a = 2392.38$ N ------------------------(Axial Component)

$P_r = 2991.53$ N ------------------------(Radial Component)

$P_a = 10524.23$ N ----------(Tangential Component)

RPM = $\frac{dp}{2} - \frac{b*sin*\gamma p}{2}$

$P_r = P_t \times \tan \emptyset \times \cos \gamma p$ N

$P_a = P_t \times \tan \emptyset \times \sin \gamma p$ N

Consider $\emptyset$ = 20 ° --------[Pressure angle] -------------------[V.B Bhandari]

$\boldsymbol{\gamma p = pitch\ angle\ of\ Bevel\ pinion}$

Pitch Cone Distance

$$A_o = \left(\sqrt{\frac{dp}{2} + \frac{dg}{2}}\right)^2$$

$$= \left(\sqrt{\frac{48}{2} + \frac{60}{2}}\right)^2$$

$A_o = 38.41$ mm

$\boldsymbol{b = \frac{Ao}{3}}$ or 10m

=12.80 or 30

Considering smaller value

B = 12.80 ≈ 13mm

$F_b = \boldsymbol{\sigma bp * b * m * Yp'\left[\frac{1-6}{Ao}\right]}$ ------------------

$\boldsymbol{\sigma b = \left(\frac{Sut}{3}\right)}$

$F_b = 2897.58$N

- Wear Strength: -

Ratio Factor (Q') = $\frac{2*zg'}{zg+zp'} = \frac{2*32.01}{32.01+20.48} = 1.21$

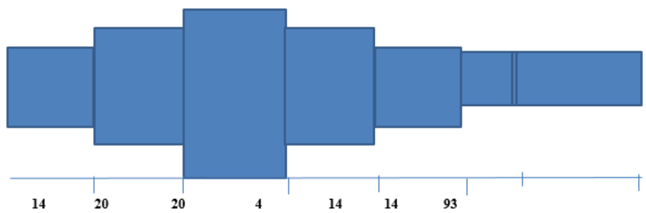
Load Stress Factor (K)

$= 0.16\left(\frac{BHN}{100}\right)^2 = \boldsymbol{6.9840\ N/mm^2}$

$F_w = \frac{0.75\ dp*b*\theta'*k}{cos\gamma p}$

$\boldsymbol{\therefore Fw = 713.49\ N}$

5.1 INPUT SHAFT :-



*(Dimensions are in mm)

$P_t = 3644.55$ N $P_r = 1326.50$ N

Total Length (L) = 14+20+20+4+14+14+93

=179 mm

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Material used :- 20MnCr5

S_{ut} = 980 N/mm^2 S_{yt} = 750N/mm^2

K_b = 2.0 [Combined shock and fatigue factor applied to bending]

K_t = 1.5 [Combined shock and fatigue factor applied to torsional moment]

Possible Shear Strength

0.3 x Syt = 225 Mpa

0.18 x S_{ut} = 0.18 x 980 = 176.4 Mpa

Choosing the lower value,

$\tau max = 0.75 * 176.4$ = 132.3 N/mm^2

Force Analysis

- For vertical plane - For horizontal plane

R_a = 480.287 N R_a = 1319.578 N

R_b = 846.22 N R_b = 2324.972 N

M_{max} = 17771 Nmm M_{max} = 48824 Nmm

Resultant

$R_a = \sqrt{480.287^2} + \sqrt{1319.578^2}$ = 1404.2655 N

$R_b = \sqrt{846.22^2} + \sqrt{2324.972^2}$ = 2474.1833 N

$M_{max} = \sqrt{17771^2} + \sqrt{48824}^2$ = 51957.592 N

Moment Torsional: -

$$M_t = \frac{Pt*Dp}{2} = \frac{P3644.55*20*2}{2}$$

M_t = 72891 Nmm

- Diameter of Shaft

$$d^3 = \frac{16}{\pi * \tau max}\ x \sqrt{(kb * mb)^2 + (kb * mb)^2}$$

$$= \frac{16}{\pi * 132.3}\ x \sqrt{(2 * 57957.542)^2 + (1.5 * 72891)^2}$$

d^3 =5806.681

$\therefore D = 17.97 \approx 20\ mm$

$\therefore Diameter\ of\ Input\ shaft = 20mm$

5.2 Design of Key

- Intermediate Shaft :

Torque on Intermediate shaft = M_t = 209853 Nmm.

For square key dimension of the key for shaft diameter (3038)mm

B x h = 10 x 8

Material selected is 10C4

$\tau c = 113.34\ N/mm^2$

$\tau = 56.67\ N/mm^2$

$$\therefore length\ of\ key = l = \frac{2mt}{\tau * db} = \frac{2 * 209853}{56.67 * 30 * 10} = 24.68mm$$

$$l = \frac{4mt}{\rho c * dh} = \frac{4 * 209853}{113.34 * 30 * 8} = 30.85mm$$

$\therefore l\ must\ be\ atleast\ 31\ mm$

For Output Shaft

Torque on output shaft =M_t=604166 Nmm

Diameter of Shaft at gear location = 35 mm

Hence Sq. key dimension of shaft =b x h=16 x10

$$length\ of\ key = l = \frac{2mt}{\tau * db} = \frac{2 * 604166}{56.67 * 55 * 16} = 38.07\ mm$$

$$l = \frac{4mt}{\tau * dh} = \frac{4 * 604166}{113.34 * 35 * 10} = 60.92\ mm$$

As face width of our gear is 20mm and length of the key is approx. 61 mm

$\therefore we\ will\ be\ using\ 3\ keys\ which\ means\ each\ key\ will\ be\ 20.3\ mm.$

$\therefore length\ of\ the\ key\ for\ output\ shaft = 21\ mm$

5.3 Bearing Selection :-

- **For Input Shaft:**

From SKF Catalogue, available bearings (DGBR) for bore diameter 20 mm are : 6004,6204,6304,6404

Selection Procedure for B1 :

As the gear is spur gear, there will be no axial forces acting on the bearings.

Hence, P_e = x * F_r --------- where x = 1

∴ P_e = 1 * 1404.2655 ------------ $F_r = R_a$

= 1404.265 N

$L_{10} = \frac{60*n*L_{10h}}{10^6}$ ------------- [L_{10h} range (12000-20000 h] ....V.B Bhandari

$= \frac{60*974.35*16000}{10^6}$ ----------- [n = Input shaft RPM]

= 935.376 rev

Now, C = P_e * $(L_{10})^{1/P}$ ---------- [P = 3 for ball bearing]

$= 1404.265 * (935.376)^{1/3}$

$= 13733.39$ N = 13.73 kN

From SKF Catalogue , Bearing no. 6304 has C = 16.8 kN. Thus , selecting 6304 for bearing B1

∴ B1 = 6304

- **Selection Procedure for B2 :**

Hence, $P_e = x * F_r$ --------- where x = 1

∴ $P_e = 1 * 2474.18$ ------------ $F_r = R_a$

$= 2474.18$ N

$L_{10} = 935.376$ rev .... From previous calculation

$C = P_e * (L_{10})^{1/P}$ ---------- [P = 3 for ball bearing]

$= 2474.18 * (935.376)^{1/3}$

$= 24196.91$ N = 24.196 kN

From SKF Catalogue, Bearing no. 6404 has C = 30.7 kN. Thus , selecting 6404 for bearing B2

∴ B2 = 6404

For Output Shaft :-

From SKF Catalogue, available bearings (DGBR) for bore diameter 35 mm are : N 207 ECM, N 307 ECM, NJ 2307 ECP

- **Selection Procedure for B1 :**

As the gear is spur gear, there will be no axial forces acting on the bearings.

Hence, $P_e = x * F_r$ --------- where x = 1

∴ $P_e = 1 * 6723.01$ ------------ $F_r = R_a$

$= 6723.01$ N

$$L_{10} = \frac{60 * n * L_{10h}}{10^6}$$ ------------- [L_{10h} range (12000-20000 h)] ....V.B Bhandari

$$= \frac{60 * 509 * 16000}{10^6}$$ ----------- [n = Input shaft RPM]

$= 488.64$ rev

Now, $C = P_e * (L_{10})^{1/P}$ ---------- [P = 10/3 for ball bearing]

$= 6723.01 * (488.64)^{0.3}$

$= 43078.48$ N = 43.078 kN

From SKF Catalogue , Bearing no. N 207 ECM has C = 56 kN. Thus , selecting N 207 ECM for bearing B1

∴ B1 = N 207 ECM

- **Selection Procedure for B2 :**

Hence, $P_e = x * F_r$ --------- where x = 1

∴ $P_e = 1 * 12342.319$ ------------ $F_r = R_a$

$= 12342.319$ N

$L_{10} = 488.64$ rev .... From previous calculation

$C = P_e * (L_{10})^{1/P}$ ---------- [P = 10/3 for ball bearing]

$= 12342.319 * (488.64)^{0.3}$

$= 79084.88$ N = 79084.88 kN

From SKF Catalogue, Bearing no. NJ 2307 ECP has C = 106 kN. Thus , selecting NJ 2307 ECP for bearing B2

∴ B2 = NJ 2307 ECP

6. DESIGN AND ANALYSIS

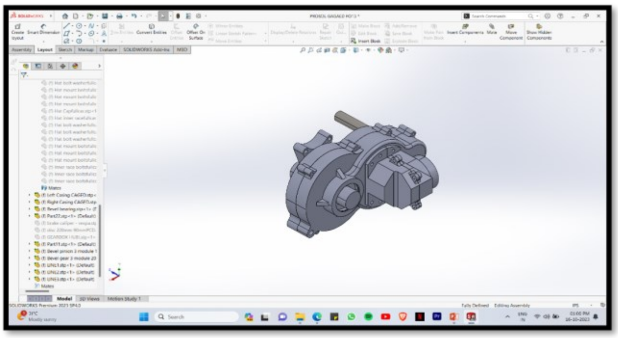

Figure 2: - Part assembly of gearbox

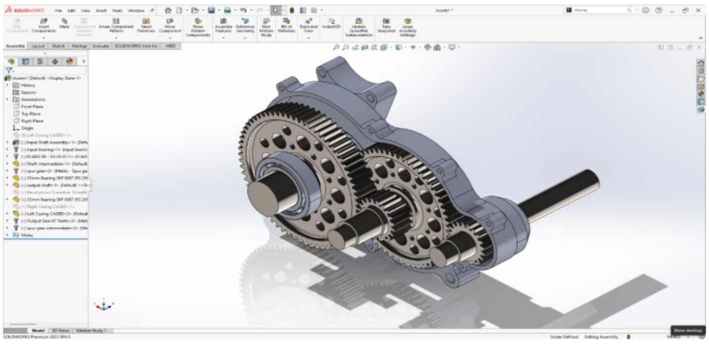

Fig 3 :- Part assembly of gearbox with bevel gear

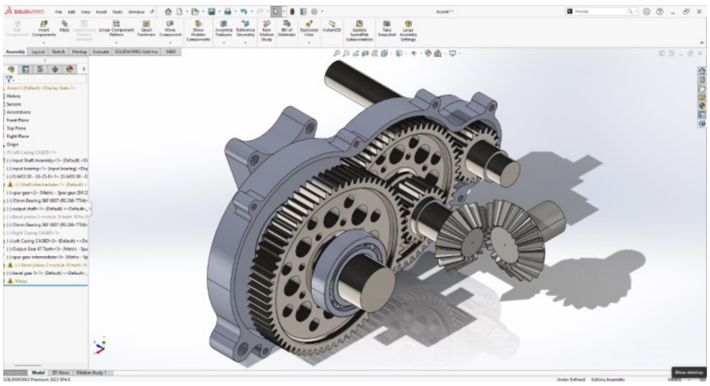

Fig 4:- Stress concentration of Spur gear

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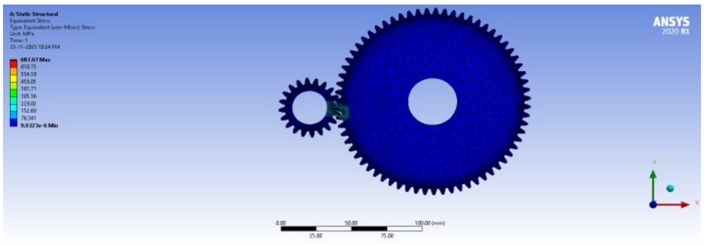


Fig 5:- Total deformation of Spur gear

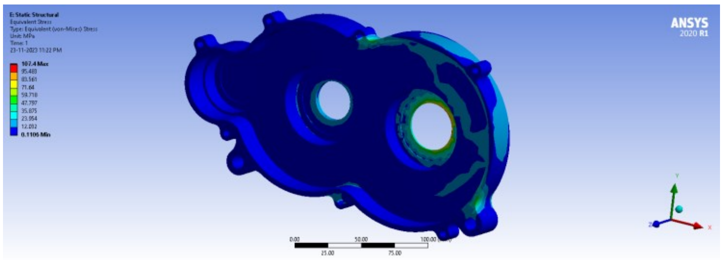


Fig 10 :- Differential assembly

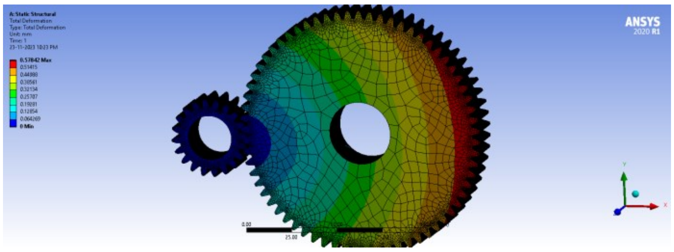


Fig 6:- Stress concentration of Output shaft

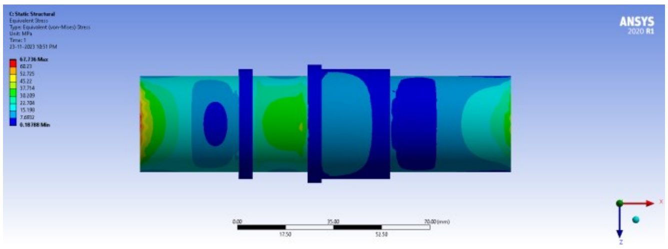


Fig 7 :- Total deformation of Output shaft

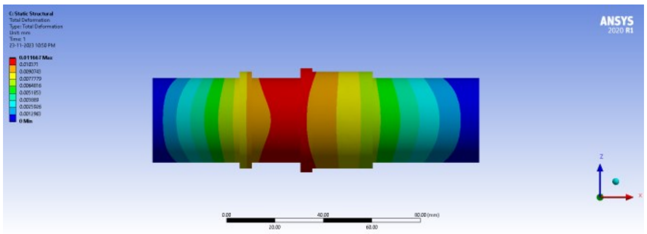


Fig 8 :- Total deformation of Intermediate shaft

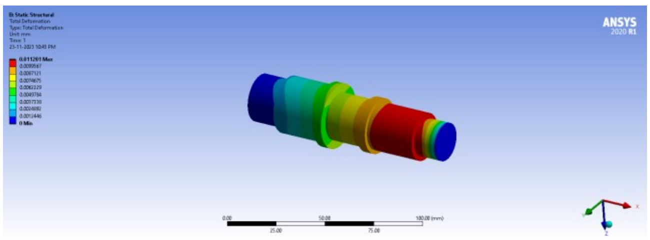
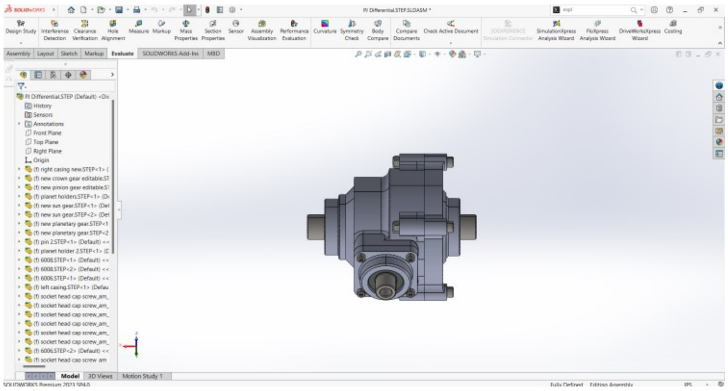


Fig 9:- Stress deformation of Gearbox Casing



7. CONCLUSION :-

At this stage we are done with the designing, modelling and analysis of 4WD gearbox of an off road vehicle. We are planning to start our manufacturing phase in the last week of December. We have set January 30th as the deadline of our manufacturing phase and are planning our activities accordingly.

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