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Research Article

Design and Analysis of Custom Manually Locking Open Differential

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ABSTRACT

This paper is written on research of utilizing the capabilities of an open and a lock differential into one differential for an SAEINDIA All-Terrain Vehicle (ATV). A differential is essential for handling tough terrain because it directs power to wheels with better traction. The custom-made design for the differential is aimed for enhancing the off-road capabilities in Lock mode while maintaining on-road performance with the same differential. In off-road settings, traction is a considerable difficulty, especially on low friction surfaces like mud or gravel. For addressing to this problem, a custom locking differential was designed, having the flexibility of an open differential with the traction control of a locked one. Open, limited-slip, and locking and locking differentials are the traditional differentials and all of them have limitations. Where cost is considered open differentials are used but they struggle in off road characteristics, Limited slip type differentials solve the problem of power loss but are complex and expensive to manufacture, locked types provide traction but steering becomes difficult on paved roads. This innovative differential design incorporates a simple locking system that may be triggered manually or electronically, allowing for adaptable on- and off-road application. It strikes a mix between cost, maintenance, and performance, with easy switching between open and locked modes for increased stability and manoeuvrability. This idea attempts to improve the versatility of vehicles as they navigate through different terrains

1. Introduction

A four-wheeled vehicle designed especially for off-road conditions is called an all-terrain vehicle (ATV). ATVs are used for a variety of tasks, such as hunting, military operations, and leisure [1-4]. The capacity to navigate across difficult and uneven terrain is a crucial requirement for ATVs. In this situation, the differential is essential because it distributes power to the traction-equipped wheels, enabling the ATV to traverse challenging terrain.

The most prevalent kind of differential is an open differential, which is typically found on ATVs. Open differentials, however, may not provide adequate traction when one or more wheels lose their grip, which can be troublesome in difficult situations.

A locked differential, on the other hand, assures equal power distribution to both wheels, which improves traction, but it can make driving on flat surfaces difficult due to the necessity for varied wheel speeds during turns [5-7]. A special open differential with a manual locking capability was created for an ATV in order to get around these restrictions. This technique maintains good drivability on normal road surfaces while improving off-road performance and handling

2. Open Differential

A ring gear is driven by a pinion gear in a differential system. The ring gear is fastened to a pin-supported cage-like construction that contains spider gears [8-9]. The output side gears, which eventually transmit power to the wheels of the vehicle, are driven in large part by these spider gears. Because it enables the wheels to spin at varying speeds, a differential is crucial, particularly when negotiating corners. Because the outer wheel must travel a greater radius during a turn than the inner wheel, smooth handling necessitates a faster rotating speed [10-13]. Tires would slip in the absence of a differential, putting lateral stress on the wheel assembly and perhaps resulting in component failure.

One of the differential's primary roles is to control the power distribution between the left and right wheels, preventing skidding during turns. However, one disadvantage of a conventional differential is that it sends more power to the wheel with less traction [14-16]. This might cause the lower-traction wheel to slip, spinning quicker and losing more grip, which can be dangerous on uneven or slippery surfaces like as mud, snow, or gravel.

In addition to its fundamental job in regulating wheel speed during turns, the differential performs numerous additional critical functions:

It decreases the rotational speed conveyed by the gearbox before providing power to the rear axles, essentially altering the speed to meet the vehicle's requirements.

The differential reverses the direction of power rotation by 90 degrees, transforming longitudinal rotation from the drivetrain into transverse rotation suitable for the wheels. This adjustment improves power delivery to the wheels [17-18]. When the vehicle moves in a straight line, the differential distributes power evenly to both rear driving axles, resulting in balanced propulsion.

During turns, the differential changes the power split, allowing the outer wheel to revolve faster than the inner wheel to compensate for the longer distance it must cover. This feature is essential for smooth and controlled handling.

It increases tractive effort, letting the vehicle to maintain grip under difficult situations [19-21]. The differential is crucial in all-wheel drive (AWD) and four-wheel drive

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(4WD) systems because it distributes power over many wheels, maintains traction, and prevents wheel slip.

The differential's components are designed to work together to maximise vehicle performance. Allowing for speed differential between wheels reduces wear and tear, improves the driving experience, and increases the vehicle's capacity to handle a variety of terrains.

This ability is particularly beneficial in off-road situations in which traction varies greatly between surfaces. Without a differential, managing tight turns, rugged terrain, or treacherous weather would be much more difficult, with a higher danger of wheel slippage and mechanical damage. The differential's design shown in fig 1, 2 and construction are thus critical to providing both stability and versatility, combining the necessity for regulated wheel movement with efficient power transmission.

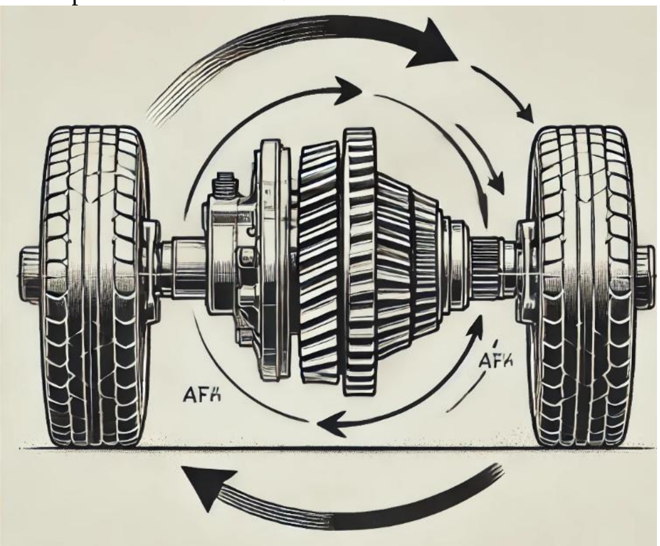

Figure1. A differential attached to the wheels

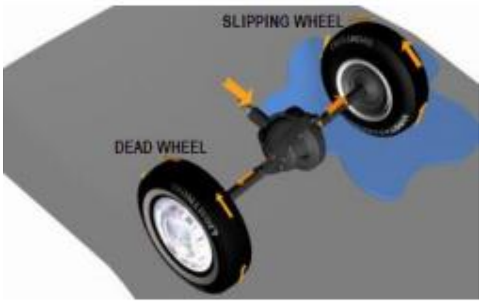

Figure2. An open differential with its problemm on slippery surface

3. Manual Locking Mechanism

The locking mechanism in the differential allows it to transition between an open and locked state. This mechanism is manually operated shown in fig 3, enabling the user to engage or disengage it while driving. When the locking mechanism is activated, it evenly distributes power to both wheels, improving traction in challenging conditions such as rough or slippery terrains. In contrast, when the lock is disengaged, the differential functions as an open differential, allowing for better drivability on smooth, paved surfaces. This adaptability provides the vehicle with the ability to perform optimally in various driving environments, enhancing both off-road performance and on-road handling [22-24].

A locking differential is designed to overcome the limitations of a standard open differential by connecting both wheels on an axle to rotate as if they were part of a single shaft. This means that both wheels rotate together, regardless of their individual traction conditions. When the differential is unlocked, it operates like a traditional open differential, allowing each wheel to rotate at different speeds, which is essential for turning. In an open differential, both wheels receive the same torque, even if one wheel is stationary while the other is spinning.

However, a locked differential forces both wheels to rotate at the same speed under most conditions, ensuring equal power distribution to each wheel, even when traction is uneven. This provides more consistent rotational force, although it may result in unequal torque on each axle. Some automatic locking differentials feature a mechanism that allows the differential to act like an open one during turns, unlocking when one wheel rotates faster (as in cornering) and re-locking when both wheels are moving at the same speed.

The locking differential shown in fig 4 offers significant traction advantages over an open differential, especially when wheel traction varies. Manual locking systems allow the driver to control when to engage or disengage the lock, making it ideal for experienced drivers. It should only be used at low speeds, and the lock must be disengaged on smooth terrain. This mechanism is particularly effective in four-wheel drive systems, helping to free vehicles stuck in off-road conditions like mud, significantly improving control and performance in challenging environments.

Figure3. CAD model of the Manual Locking open differential

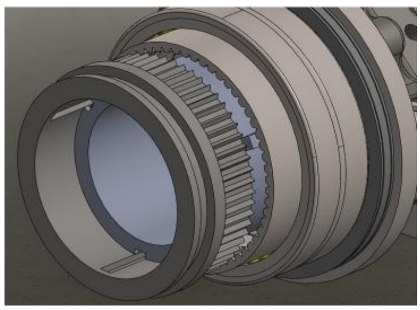

Figure4. Locking disc for mechanism

4. Design Consideration

The design of the custom differential was guided by several key considerations:

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Improved Off-Road Capabilities: The differential needed to deliver superior traction on challenging terrains by enabling equal power distribution to both wheels when necessary, ensuring the ATV can handle rough and uneven surfaces effectively.

Enhanced Handling: It was essential that the differential not compromise the ATV's handling and maneuverability on smooth surfaces, allowing for precise control and stability in a variety of conditions.

Drivability: The differential had to maintain smooth drivability during turns on regular terrain, preventing any handling issues that could arise from the locking mechanism when not engaged.

Ergonomics: The differential's design had to allow for easy integration into the ATV model, fitting seamlessly within the existing structure while maintaining ergonomic standards for ease of use and maintenance.

5. Material selection for the differential

5.1 Factor of Material Selection

Gear materials are generally categorized into metallic and non-metallic groups. Given the considerations of material availability, manufacturing methods, and application demands, metallic materials emerged as the more viable option for this design. The goal of material selection is to identify a material with the optimal combination of physical properties to fulfill the system's functional requirements. In selecting the most suitable material, specific factors are prioritized to align with the design's needs. Here, the primary factors are determined based on the parameters necessary to calculate gear strength and ensure surface durability, both critical for reliable performance in the intended application.

4.2 Characteristics

The material to be selected must be capable of withstanding the various failure modes that might occur in the system such as:

- Bending Fatigue

Gear tooth failure is primarily caused by cyclic bending stresses that concentrate at the root of the gear tooth. This stress concentration initiates a crack at the surface of the root fillet due to fatigue. The crack then spreads across the gear tooth, perpendicular to the root fillet surface. The failure process is typically divided into three stages: crack nucleation (the formation of the initial crack), crack propagation (the expansion of the crack during repeated load cycles), and fracture (the full failure of the gear tooth).

- Pitting Failure

Metal-to-metal contact or insufficient lubricant film thickness can result in surface cracks that eventually grow into pits. Gear material inclusions can cause internal cracks by acting as stress concentrators.

Pits form when these internal cracks extend to the surface, causing small portions of material to separate and erode. This deterioration compromises the gear's integrity, affecting its performance and lifespan.

- Wear

Wear occurs when material is gradually removed from the teeth of mating gears due to continuous abrasive action. This wear can result from several factors, including inadequate lubricating film thickness, the loss of a hardened surface layer on surface-hardened gears, or contaminants and inclusions in the lubricating oil. Persistent wear at the tooth root weakens the gear structure, eventually leading to fracture and potential gear failure.

6. Calculation For The Differential

6.1 Input paramenters of the engine of ATV

- Torque: 19.6N, 3800 RPM
- Gearbox Ratio: 3:1
- Pressure angle of the Gear: 20
- Power from the engine:7.46kW
- Rpm of the pinion: 1267

6.2 Ratio Selection

Tractive Effort: The amount of force exerted on the ground to propel the vehicle. It is given by:

$$TE = T/r$$

Rolling resistance: The minimal amount of frictional resistance offered by the plane ground that tires prevent them from moving forward. It is given by:

$$fr = \mu N$$

Gradient Resistance: The rolling resistance offered to the vehicle by the inclined plane. It is given by:

$$fg = mg\mu cos\alpha$$

Top speed: Trying to accommodate for the above the resistances, the gear box need to be increased which comes at the expense of decreased top speed of the vehicle. It is given by:

$$S = (2\pi * rpm * r)/(60 * CVT * GR)$$

S= (2π*3800*0.266) / (60*3.75*7.2) = **55.1km/h**

Gear Ratio Taken - 7.2

6.3 Calculation for bevel gears

- *Crown and pinion gear-*

1) Pitch angle(ϒ)= tan−1 $zp\ zg$ = **22.58**
2) σb = Sut/3=**366 N/mm2**
3) lewis form factor(Yp) = 0.487- (2.87/zp) = **0.282**

Module estimation for crown and pinion gear-

Torque transmissible(Mt)= $(60 * 106 * KW)/2\pi np$ = 56254.0532 Nmm

Dp= m*zp= 15m b=10mm

Dz= m*zg= 36m

- Tangential load(Pt)= 2*Mt/Dp= 7500.54/m

We have, b/Ao= 1/3

- Pitch line velocity(v)= $(\pi * Dp * np)/(60 * 103\ )$= 3.71 m/s
- Cv=0.74, Cs=1
- Beam strength(Sb)= m*b*σb*Yp*[1-(b/Ao)]= 688.08m2 N
- Effective load(Peff)= (Cs*Pt)/Cv= 10135.86/m

Also, Sb=Peff*Fs

m=2.805

m=3.5

Module taken is 3.5 for crown and pinion gear.

Dp= 15m= 15*3.5= 52.5mm

Dz=36m= 36*3.5= 126mm

b= 17mm

Cone distance(Ao)= $\sqrt{(\frac{Dp}{2})^2 + (\frac{Dg}{2})^2}$ = **68.25**

Beam strength(Sb) = m*b*σb*Yp*[1-(b/Ao)]= **4605.83N**

Tangential load(Pt)= 2*Mt/Dp= **2143.011 N**

Pitch line velocity(v)= $(\pi * Dp * np)/(60 * 10^3)$ = **3.48 m/s**

Dynamic load(Pd)= $(21v * (C * e * b + Pt))/(21v + \sqrt{C * e * b + Pt})$ =**2372.309 N**

Effectice load(eff)= Cs*Pt+Pd= **4515.32 N**

FOS on bending= Sb/Peff= **1.0**

Table 1 . Bevel Gear Calculation for crown pinion gears

CONSTANTS					
no. of teeth of pinion	**z p**	**15**	**Lewis Factor of pinion (Y p)**		0.295667
no. of teeth of gear	**z g**	**36**	**Diameter of pinion (D p)**		52.5
Module	**M**	**3.5**	**Diameter of Gear (D g)**		126
Material property	**S ut**	1100	**Cone Distance (A not)**		68.25
Material property	**Sigma g**	366.6667	**Beam Strength (S b)**		4843.753
face width	**b**	**17**	**Ratio Factor (Q)**		1.704545
pitch angle	**gama**	**22.58**	**Material Constant (k)**		2.56
input RPM	**n p**	1267	**Wear Strength (S w)**		3174.901
deformation factor	**C**	11400	**Mt**		56254.05
error acc to case3	**e**	0.0125	**Tangential Load (P t)**		2143.012
Service Factor	**C s**	1	**Pitch line Velocity (v)**		3.481083
Power of Engine	**KW**	7.46	**Dynamic Load(P d)**		2372.562
BHN	**BHN**	400	**Effective Load (P eff)**		4515.573
Pressure angle	**Alpha**	20	**FOS OF BENDING**		1.072677
			FOS OF WEAR		0.7031
			Axial Load (P a)		295.607
sin (gama)		0.38	**Radial Load (P r)**		715.6801
Tan (gama)		0.416			
cos (gama)		0.92			
Numerator for Pd		333751.4			
Denominator for pd		140.6713			
tan (alpha)		0.363			

- *For side and spider gear*

Circumference of wheel= π*D =1.66 m

Circumference of outer wheel= 2*π*r= 11.304 m

Circumference of inner wheel= 2*π*r= 9.04 m

Inner wheel ratio= (3)/(1)= 9.04/1.66

Outer wheel ratio= (2)/(1)= 11.304/1.66

- Ratio= (5)/(4) = **1.25**

1) Pitch angle(ϒ)= $tan^{-1}\frac{zp}{zg}$
2) σb= Sut/3
3) lewis form factor(Yp)= 0.487- (2.87/zp)

Module Estimation

4) Torque transmissible(Mt)= $(60 * 10^6 * KW)/2\pi np$

Dp= m*zp b

Dz= m*zg

5) Tangential load(Pt)= 2*Mt/Dp
6) We have, b/Ao= 1/3
7) Pitch line velocity(v)= $(\pi * Dp * np)/(60 * 10^3)$
8) Cv=0.74, Cs=1
9) Beam strength(Sb)= m*b*σb*Yp*[1-(b/Ao)]
10) Effective load(Peff)= (Cs*Pt)/Cv
11) Also, Sb=Peff*Fs

m=3

Dp= 12m=36mm

Dg=15m=45mm

b= 17mm

12) Cone distance(Ao)= $\sqrt{(\frac{Dp}{2})^2 + (\frac{Dg}{2})^2}$
13) Beam strength(Sb) = m*b*σb*Yp*[1-(b/Ao)]
14) Tangential load(Pt)= 2*Mt/Dp
15) Pitch line velocity(v)= $(\pi * Dp * np)/(60 * 10^3)$
16) Dynamic load(Pd)= $(21v * (C * e * b + Pt))/(21v + \sqrt{C * e * b + Pt})$
17) Effectice load(eff)= Cs*Pt+Pd
18) FOS on bending= Sb/Peff

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Table 2. Bevel Gear Calculation

CONSTANTS						
no. of teeth of pinion	**z p**	12		**Lewis Factor of pinion (Y p)**		0.247833
no. of teeth of gear	**z g**	15		**Diameter of pinion (D p)**		36
20	**M**	3		**Diameter of Gear (D g)**		45
Material property	**S ut**	1100		**Cone Distance (A not)**		28.81406
Material property	**Sigma g**	366.6667		**Beam Strength (S b)**		1900.186
face width	**B**	17		**Ratio Factor (Q)**		1.515152
pitch angle	**Gama**	21.8		**Material Constant (k)**		2.56
input RPM	**n p**	1267		**Wear Strength (S w)**		1935.178
deformation factor	**C**	11400		**Mt**		56254.05
error acc to case3	**E**	0.0125		**Tangential Load (P t)**		3125.225
Service Factor	**C s**	1		**Pitch line Velocity (v)**		2.387028
Power of Engine	**KW**	7.46		**Dynamic Load(P d)**		2231.704
BHN	**BHN**	400		**Effective Load (P eff)**		5356.929
Pressure angle	**Alpha**	20		**FOS OF BENDING**		0.354715
				FOS OF WEAR		0.361248
				Axial Load (P a)		419.749
sin (gama)		0.37		**Radial Load (P r)**		1043.7
Tan (gama)		0.4				
cos (gama)		0.92				
Numerator for Pd		278094.1				
Denominator for pd		124.6106				
tan (alpha)		0.363				

Table 3. Gear ratio of bevel gear pairs

Gear	Teeth	Module	Gear Ratio
Diff. Pinion	15	3.5	2.4:1
Crown Gear	36	3.5	
Side Gear	15	3	1.25:1
Spider Gear	12	3	

6.4 Bearing Selection

- *For Pinion Gear*

Pt=2009.073 N
Paxial = Pt*tan(α)*cos(γ)= 670.95 N
Pradial = Pt* tan(α)*sin(γ)= 270 N
L10 = (60*1267*L10H)/1000000= 15.204 million rev
Axial

$$C=P*(L_{10})^{1/3} = \mathbf{1662.17\ N}$$

Radial

$$C= P*(L_{10})^{1/3} = \mathbf{668.88\ N}$$

Bearing = AXK 3047 (Thrust bearing for radial load)
Bearing= HK 3016 (for axial load)
ID=30mm
OD= 37mm
WIDTH= 16mm

- *For Side Gear*

Pt=7471.05 N
Paxial = Pt*tan(α)*cos(γ)=211.35 N
Pradial = Pt*tan(α)*sin(γ)=1681.35 N
$L_{10} = 60*n*L_{10H}/10^6$ = 3.09 million rev
Axial

$$C=P*(L_{10})^{1/3} = 3050.86\ N$$

Radial

$$C= P*(L_{10})^{1/3} =2421.28\ N$$

Bearing left side
Axial load= NK 80/25
OD= 95mm
ID=80mm
WIDTH=25mm
Radia load= AXK 65.9

- *Bearing right side*

Axial load= RNA 4911
OD=8mm
ID=63mm
WIDTH=25mm
Radial load=AXK 0105

6.5 For Locking Disc

d=63mm
Power transmitted P=7.45KW
n=RPM=530
FOS=3
Syt=460 N/mm2
Permissible compressive and shear stress
Syc=Syt=460 N/mm2
σc=Syc/F(s)=153.33 N/mm2
Say=0.5*Syt=230N/mm2
T=Ssy/fos= 76.67 N/mm2
Torque transmitted by shaft
Mt= (60*10^6*P)/2πn=134298.72 Nom

Width, b=5mm
3 keys of height
l=2*Mt/T*d*b=11.12mm
l=15mm

7. CAD Model of the Differential


Figure.5 Front View

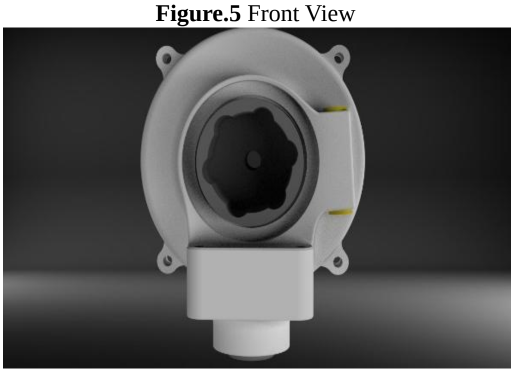


Figure 6. Side View

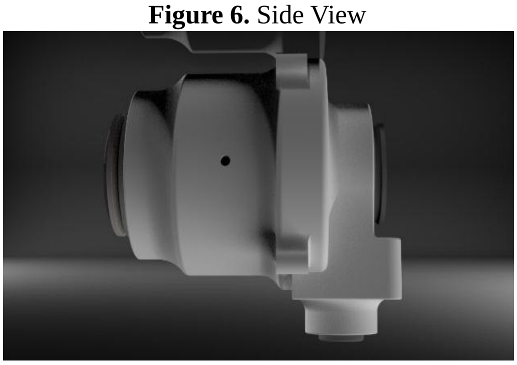


Figure7. Top View

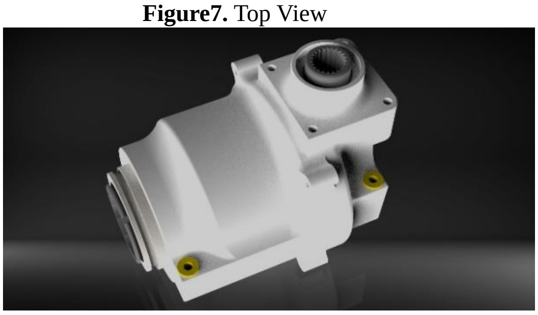


Figure 8. Isometric view

8. CAE Analysis of the CAD model

Once the unit was designed using the calculated gear parameters, finite element analysis (fea) was conducted to evaluate the assembly's integrity. Since each component in the system supports and reinforces the others, analyzing individual parts separately was not ideal. Therefore, the entire assembly was assessed as a single unit. The analysis included static structural, fatigue, and explicit dynamic simulations, all performed using ansys workbench to ensure the reliability and durability of the design.

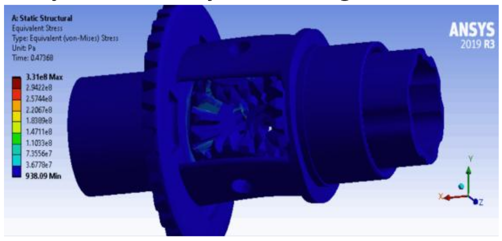


Figure.8. Simulation of Spider and Side gear setup in locked mechanism

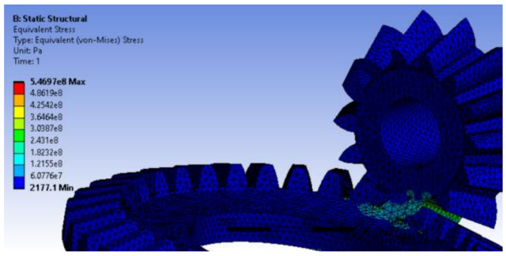


Figure 9. Simulation of Ring and Pinion gear setup

Table 8. Analytical study of bevel gear pairs

Gear Combination	Module	Face width (mm)	Deformation (mm)	Stress (von Mises) MPa	Factor of Safety
Crown - Pinion	3.5	17	0.038	546	1.42
Spider - Side	3	15	0.022	311	2.28

8.1. Analysis of case

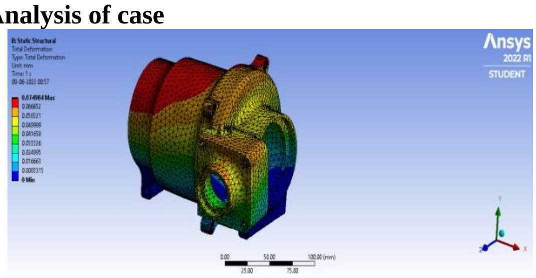


Figure 10. Deformation

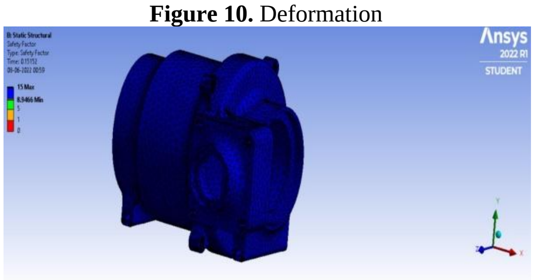


Figure 11. FOS

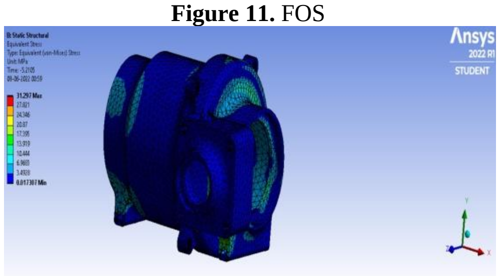


Figure12. Stress

8.2 Analysis of Cage

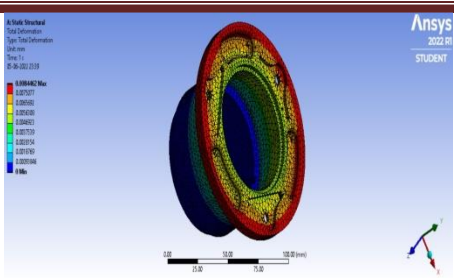


Figure 13. Deformation

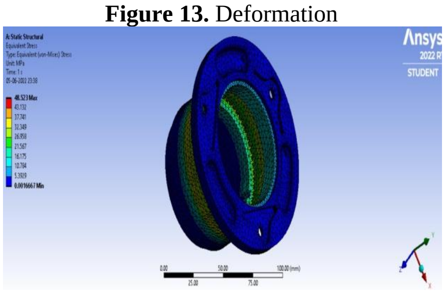


Figure.14 Stress

8.3 Analysis Of Locking Mechanism

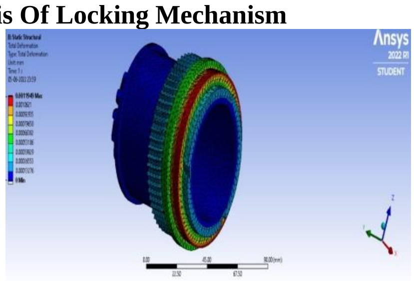


Figure 15. Deformation

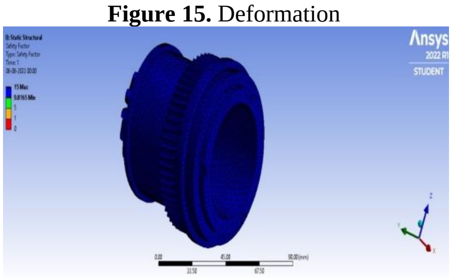


Figure16. FOS

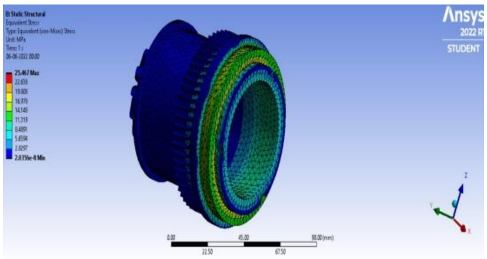


Figure 17. Stress

8.4 Analysis of Side - Spider Gear

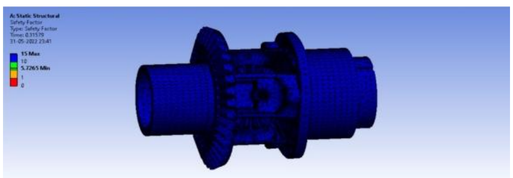


Figure18. FOS

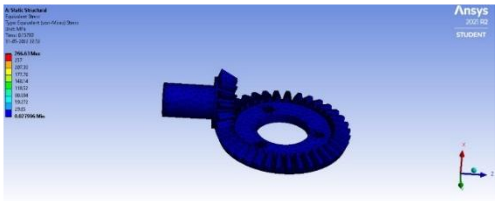


Figure 19. Stress

8.5 Analysis of Crown-Pinion Gear

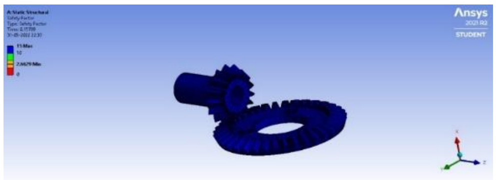
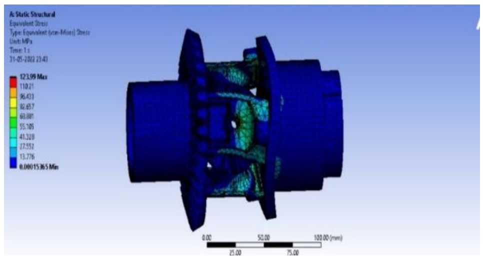


Figure 20. FOS


.**Figure 21.** Stress

9. Conclusion

The differential assembly meets the design specifications, demonstrating durability in rough terrains and challenging operating conditions. No traction issues were detected, and the vehicle handled sharp turns without losing grip, confirming that the differential's design objectives for the ATV's transmission system were successfully achieved. This success lays the foundation for further research into the system. Based on comparisons between theoretical and numerical analysis of tangential load, beam strength, and effective load on gear teeth—using Lewis Form Factor, module estimation, CAD models, and ANSYS simulations—the following conclusions are made:

- Material Selection: The differential's materials are EN24 for the pillars and cage, EN36 for the bevel gears, and Al 6061 for the casing. The calculations for tangential load, beam strength, and effective load are all within safe limits, with a safety factor more than one for each.

- Performance Evaluation: An ATV with the modified differential was tested over various terrains and situations to compare its performance to that of a regular setup. The tests proved that the unique differential improves off-road performance while preserving smooth driving on ordinary ground.

- Feasibility: This study examines the viability of a custom open differential with a manual locking mechanism for ATVs. The design improves off-road performance, handling, and manoeuvrability in challenging terrains. It may also be easily installed into existing ATV models, making it an affordable option for off-road enthusiasts.

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