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Numerical Investigation of Anti-Reverse Differential Mechanism Employed In Manual Transmission

Dr.Mohan Kulkarni[1] Atharva Bagrao[2] Amey Tibile[3] Amey Chougule[4] Vaibhav Loyal[5]
[12345] *MIT ADT University Pune*

Abstract
To improve a vehicle's stability and performance in difficult terrain, an Anti-Reverse Differential Locking (ARDL) system must be designed and developed. This system uses an advanced locking mechanism to improve traction and prevent unwanted wheel spin. This study's main objective is to use static structural analysis to thoroughly analyze the locking mechanism. The researchers hope to assess the ARDL system's overall performance, stress distribution, and structural integrity under a range of operating circumstances and loads by using this analytical method. The analysis gathered from this will help to improve the Anti-Reverse Differential Locking system's resilience, functionality, and design. This will enhance the vehicles that use this cutting-edge technology's off-road capabilities and safety.

Keywords*: ARDL, Traction, Locking Mechanism & Structural Analysis.*

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I. Introduction

The design and development of an Anti-Reverse Differential Locking (ARDL) system is a critical endeavor in enhancing vehicle performance, particularly in off-road scenarios where traction and stability are paramount. The ARDL system serves to mitigate wheel spin and improve overall traction by incorporating a sophisticated locking mechanism. This study focuses on the comprehensive analysis of the locking mechanism using Static Structural analysis [4]. By employing advanced engineering analysis techniques, the research aims to evaluate the structural integrity and stress distribution of the ARDL system under various loads and operating conditions. The insights gained from this analysis will not only refine the design and optimize the functionality of the ARDL system but also contribute to advancing the understanding of its mechanical behavior, ensuring its reliability and effectiveness in challenging terrains.[3]

Nomenclature

Table 1 – Nomenclature

Sr No	Abbreviations	Full Name
1	PTC	Parametric Technology Corporation
2	LSD	Limited Slip Differential
3	DARM	Differential Anti-Reverse Mechanism
4	FEM	Finite Element Method
5	AGMA	American Gear Manufacturers Association
6	FEA	Finite Element Analysis
7	M_{pa}	Megapascal
8	P_a	Pascal
9	P	Power Of Motor
10	N	Rpm Of Motor Shaft
11	T_1	Torque transmitted
12	D_1	Diameter of small pulley
13	D_2	Diameter of big pulley
14	T_1`	Tension on the tight side
15	T_2`	Tension on the slack side
16	O_1, O_2	Center distance between two shafts or pulleys
17	τ	Permissible shear stress

18	θ_{P1}	Pitch Angle of Pinion
19	θ_{p2}	Pitch Angle of Gear
20	T_{ep}	Formative Number Of Teeth For Pinion
21	T_{eg}	Formative Number Of Teeth For Gear
22	W_T	Tangential Load
23	C_v	Velocity Factor
24	L	Slant Height of Pitch Cone
25	b	Face Width
26	a	Addendum
27	c	Clearance
28	d	Dedendum

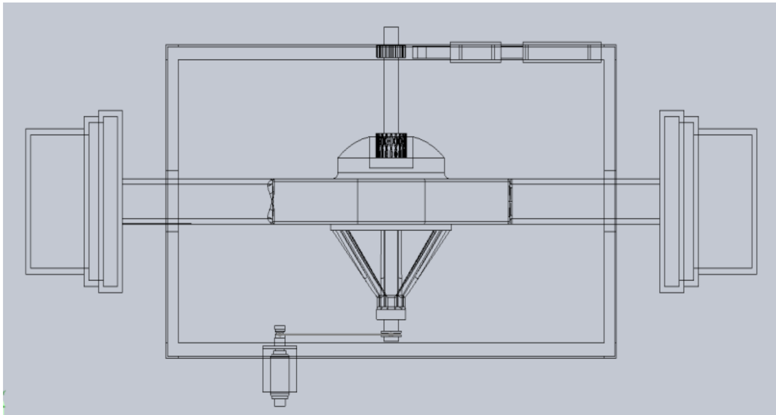


Figure 1: Schematic Diagram of the Project

II. Problem statement

To design and develop a differential system which prevents rolling back of vehicles on inclines. With the help of a locking mechanism, it will prevent the vehicle from rolling backwards on slope. & also to reduce the traction while in motion, the vehicle's differential, which permits the wheels to spin at various speeds when turning, may not be able to transmit torque between the wheels efficiently in the absence of an ARDL system. When one wheel loses traction, it may begin to spin uncontrollably while the other wheel gets very little power.

III. Objective

1. To offer a comprehensive assessment of the influence, functioning, and performance of anti-reverse differential gear systems in automobiles.
2. To provide light on the benefits, drawbacks, and potential areas for improvement of anti-reverse differential gear systems in order to better comprehend their function in vehicle safety and technology.
3. To create a model that is in functional condition, examine a frame structure, and design the model.
4. To conduct a Structural analysis on Locking mechanism and differential system for safety.
5. To modulate the system as per the designed condition and test.
6. To evaluate the impact these systems have on the vehicles overall dynamics, handling, and responsiveness.

IV. Literature Survey

R. Selvaraj To lock the differential, engage a mechanism that physically connects the two wheels on the same axle, ensuring equal torque distribution, typically through a manual or automatic locking mechanism. Consequently, direct power transmission reaches the axle and eventually the wheels. This significantly lessens the loss of power that may occur if the axle inadvertently transfers power to the differential, axle, and ultimately the wheels [4]. As a result, the technique lessens unnecessary power loss while the differential is being transmitted. We will solve some of the current mechanism's flaws in the suggested proposal [4].

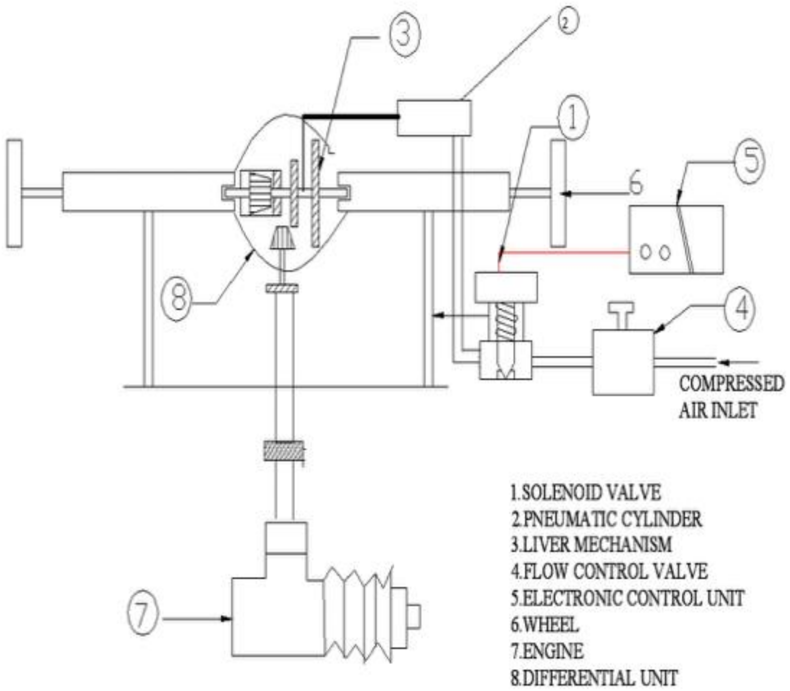


Figure 2 (a) : Automatic differential locking [4]

Mrunmay Raut, [5] Automobiles featuring anti-reverse differential gear systems are often constructed from robust, long-lasting materials that ensure long-term reliability even under extreme mechanical stress. Because they provide durability and wear resistance, hardened steel alloys are commonly employed in the building of these systems for the gears and shafts. Furthermore, materials that balance strength and weight considerations, such as aluminum or cast iron, are commonly used to make housing components [5]. The bearings in the system might be made of chrome steel or ceramic to reduce friction and lengthen their lifespan [5]. The materials chosen determine how resilient the anti-reverse differential gear system is under varying loads and situations, ensuring optimal performance and efficient power transmission over the life of the vehicle [5].

Arunkumar A. In order to keep the car from sliding backwards when it is on a mountainous route, [6] has been studied. It has been established that the front axle is stopped from moving by the ratchet and pawl system. The ratchet and pawl mechanism, highlighted in studies by J. A. Kennedy, L. L. Howell, et al., serves as an effective solution for mechanical safety systems, particularly in scenarios where traditional components cannot be accommodated within a compact design envelope. This mechanism has been successfully implemented in an anti-roll back device designed for vehicles, specifically tested using the front axle assembly. The functionality of this mechanism relies on the interaction between the pawl and the ratchet, where the pawl is securely installed within its frame and engages with the ratchet on the front drive shaft. When activated, typically by a lever, the pawl makes contact with the ratchet, preventing the vehicle from rolling backward on inclines or hills. This method ensures stability and safety during hill road maneuvers, offering a reliable solution to mitigate the risk of unintended movement. The simplicity and effectiveness of the ratchet and pawl mechanism make it a practical choice for applications where space constraints or design limitations necessitate compact and efficient safety solutions. Additionally, its mechanical nature ensures reliability even in challenging environments, making it a valuable component in automotive safety systems aiming to enhance vehicle stability and control [6].
The restrictions of the mechanism are outlined, along with the mechanism and its modeling. The outcomes of the testing of three scaled prototypes are shown.[6].

Patil, Utkarsh A.[8] This paper discusses several differentials that address problems arising from the limitations of a basic differential. Wheel torques in front-wheel drive can affect the way the car moves and how the steering feels.[8] Losses over the differential are caused by torque steer, which is demonstrated by variations in the torques of the left and right shafts. A basic differential restricts acceleration, particularly when turning and acceleration are coupled, by allowing the inside wheel to spin freely under high torque. One wheel getting trapped is a possibility on slick tracks [8].

Figure 2(b): Differential Meshing [8]

V. Theoretical Analysis -

5.1 Torque calculation on pulleys

Motor of Power = P = 17 watt (Wiper Motor)

Therefore, shaft transmitted power,

$P = \frac{2\pi NT}{60}$……………………………………..……….(1) [9]

Where, N → Rpm of motor shaft = 24

T_1 → Transmitted Torque

$17 = \frac{(2\square * 24 * T)}{60 * 10^3}$

$T_1 = 6.76 \times 10^3$ N-mm

Therefore,

Diameter of small pulley ($D_1 = 50$ mm)

Diameter of big pulley ($D_2 = 150$ mm)

Ratio = R = 50 : 150 = 1:3

Torque on pulley = 3 * T_1

= 3 * 6.76 * 10^3

$T_2 = 20.280 * 10^3$ N-mm

T_2 = Force * radius……………………………..…….(2) [9]

$20.280 * 10^3 = F * 75$

F = 270N

F = 270/9.81

F = 27.5 Kg

5.2 Design of v- belt -

As we know maximum torque on shaft

$T_{max} = T = 20.280 * 10^3$ N-mm

Where ,

T_1` = Tension in tight side

T_2` = Tension in slack side

O_1, O_2 = center distance between two shafts or pulleys

$\sin \propto = \frac{\square 1 - \square 2}{\square 1 \square 2}$ …………………………………………….(3) [9]

$\sin \alpha = \frac{(75 - 25)}{305}$

$\sin \alpha = 0.1639$

$\alpha = 9.4353°$

TO FIND θ

$\theta = \frac{(180-2\alpha)*\pi}{180}$……………………………………..………..(4) [9]

$= \frac{(180-2*9.4353)*\pi}{180}$

$\theta = 2.8122$ rad

we know that,

$\frac{T_1`}{T_2`} = e^{(\mu\theta \operatorname{cosec} \beta)}$……………………………………………..….(5) [9]

$\frac{T_1`}{T_2`} = e^{(0.25*2.8122*\operatorname{cosec} 20)}$

$T_1` = 7.81137 * T_2`$

We have,

$T = (T_1` - T_2`) * R$………………………………………….(6) [9]

$20.280 * 10^3 = (7.81137 * T_2` - T_2`) * 25$

$T_2` = 119.09$ N

$T_1` = 7.81137 * T_2`$

$= 7.81137 * 119.09$

$T_1` = 930.31$ N

Hence we find out tension in tight side = $T_1` = 930.31$ N

5.3 Length of belt calculation :-

Shaft pulley Radius,`

$r_1 = \frac{d_1}{2} = \frac{150}{2} = 75$mm

Radius of pulley on motor shaft

$r_2 = \frac{d_2}{2} = \frac{50}{2} = 25$mm

Center distance between two pulley = 305 mm

We know length of belt

$L = \frac{\pi(r_2 + r_1) + 2 * O_1O_2 + (r_2 - r_1)^2}{2}$…………..………………(7) [9]

$= \frac{\pi(25+ 75) + 2 * 305 + (25 - 75)^2}{2}$

L = 924.15 mm = 36.45 inch

5.4 Design of bevel gear shaft

Torque transmitted by shaft,

$T = \frac{\pi}{16} * \tau * (D`)^3$ ……………………………………….…(8) [9]

Selecting permissible shear stress (τ) design data book.

$\tau = 70$ N/mm^2

whereas, $20.28 \times 10^3 = \frac{\pi}{16} * 70 * (D`)^3$

D` = 12 mm
selecting the diameter of shaft = 12 mm
Using factor of safety (FOS) for shaft = 1.6
D` actual for shaft = 12 * 1.6 = 19.2 mm = 20 mm (according to std)

5.5 Pitch angle for pinion

$\theta_{P1} = \square\square\square^{-1}\left(\frac{1}{\square.\square}\right)$…………………………….……………...(9) [9]

$= \square\square\square^{-1}\left(\frac{\square\square}{\square\square}\right)$

$= \square\square\square^{-1}\left(\frac{20}{40}\right)$

$\theta_{P1} = 26.56°$

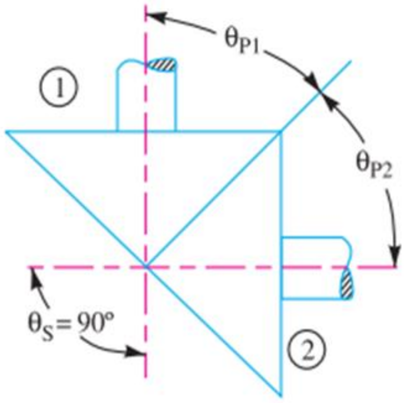


Figure 3 : Schematic Bevel Gear [12]

5.6 Pitch angle for gear

$\theta_{p2} = \theta_s - \theta_{P1}$………………………………………..……(10) [9]

$= 90 - 26.56$

$\theta_{p2} = 63.44°$

5.7 Formative teeth number for pinion

$T_{ep} = T_P * \sec\theta_{P1} = 20 * 26.56 = 531.2$ ………………..(11) [9]

$y`_P = 0.124 - \frac{0.684}{\square\square\square}$

$= 0.124 - \frac{0.684}{531.2}$

$y`_P = 0.12$

$\sigma_{op} * y`_P = 85 * 0.12 = 10.20$……………………………(12) [9]

5.8 FORMATIVE TEETH NUMBER FOR GEARS

$T_{eg} = T_G * \sec\theta_{"p2"} = 40 \times 63.44 = 2537.6$…………..…(13) [9]

$y`_G = 0.124 - \frac{0.684}{\square\square\square}$

$= 0.124 - \frac{0.684}{2537.6}$

$y`_G = 0.12$

$\sigma_{og} * y`_G = 85 * 0.12 = 10.20$……………………...………(14) [9]

Since the product of $\sigma_{og} * y`_G$ & $\sigma_{op} * y`_P$ is the same so design should be based on gear which is the same for pinion.

5.9 BEVEL GEARS DESIGN

A pair of teeth of bevel gears mounted, which are interesting at right angles, consists of 20 teeth on both the pinion and 40 teeth on gears.
Lewis Equation For Strength of Gears -

$W_T = (\sigma_o * C_v) b * \pi * m * y` * \left(\frac{\square-\square}{\square}\right)$………………………………..(15) [9]

$\sigma_{og} = \sigma_{op}$ = Allowable static stress = 85 N /mm^2

1. Tangential Load (W_T) -

$W_T = \frac{2\square}{\square\square}$……………………………………….………………(16) [9]

$= \frac{2\square}{\square * \square\square}$

$= \frac{2*19266}{\square*40}$

$W_T = \frac{963.3}{\square}N$

2. Velocity Factor (C_v) –

$C_v=\frac{6}{6+\square}$ ……………………………………………….………(17) [9]

Peripheral Velocity (v) -

$v = \frac{\square*\square\square*\square\square}{60}$……………………………………………….…..(18) [9]

$= \frac{\square*\square*\square\square*\square\square}{60}$

$= \frac{\square*\square*40*8}{60}$

v = 16.7551m mm/sec

v = 0.0167m m/sec

Now, $C_v = \frac{6}{6+0.0167\square}$

3. Slant Height of Pitch Cone (L) –

$L = \frac{\square\square}{2*\square\square\square\ \square\square2}$……………………………………………..(19) [9]

$= \frac{\square*\square\square}{2*\square\square\square\ \square\square2}$

$= \frac{\square*40}{2*\square\square\square\ 63.44°}$

L = 22.3596m mm

4. Face Width (b) –

$b = \frac{\square}{3}$

$= \frac{22.3596\square}{3}$

b = 7.4532m mm

Hence we find out the face width = b = 7.4532m mm

$W_T =(\sigma_o * C_v) b * \pi * m * y` * \left(\frac{\square-\square}{\square}\right)$ [9]

$\frac{963.3}{\square} = \left(85 * \frac{6}{6+0.0167\square}\right)*7.4532m * \pi * m * 0.12 * \left(\frac{22.3596\square - 7.4532\square}{22.3596\square}\right)$

$m = 4.2788 \approx 4$

5.10 BEVEL GEAR PROPORTION

Addendum = a = 1 * m = 1 * 4 = 4 mm……………………..……. (20) [9]

Dedendum = d = 1.2 * m = 1.2 * 4 = 4.8 mm………………..…… (21) [9]

Clearance = c = 0.2 * m = 0.2 * 4 = 0.8 mm…………………..……(22) [9]

Working depth = w = 1.5708 * m = 1.5708 * 4 = 6.28 mm ……….(23) [9]

5.11 MEAN RADIUS (Rm) –

$Rm = \left(\square - \frac{\square}{2}\right) \sin\theta_p$ ……………………………...……. (24) [9]

$= \left(89.4384 - \frac{29.8128}{2}\right) \sin 45°$

Rm = 52.7020 mm

5.12 INDUCED TANGENTIAL FORCE (W_T) –

$W_T = \frac{\square}{\square\square}$ ……………………………………..…… (25) [9]

$= \frac{19266}{52.7020}$

$W_T = 365.5648$ N

5.13 BEVEL GEAR'S CAPACITY TO BEAR TANGENTIAL FORCE –

$W_T =(\sigma_o * C_v) b * \pi * m * y` * \left(\frac{\square-\square}{\square}\right)$ [9]

Putting all values for m=4

W_T = 2443.48 = 2444 N

The design is safe because the force bearing capacity of gear is more than the applied force.

VI. L -SECTION DESIGN -

Material - Mild Steel

Horizontal section is subjected to the bending stress

Stress => $\frac{□}{□} = \frac{□□}{□}$

- Minimum moment of inertia , considering the angle of 30 * 30 * 3mm size
- The channel is subjected to the axial compressive load
- The maximum bending moment will occur at the channel of the section

$M = W * \frac{□}{4}$ (simply supported beam) …………..……..(26) [9]

$= 500 * \frac{930}{4}$

M = 116250 N-mm

We know, $\sigma_b = \frac{□}{□}$………………………………………………(27) [9]

Where, Z = Section Modulus

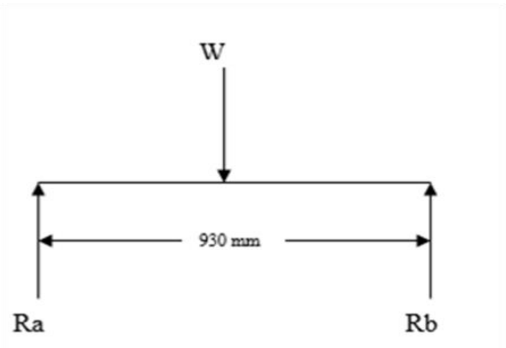


Figure 4 : Reaction Force on L Section[14]

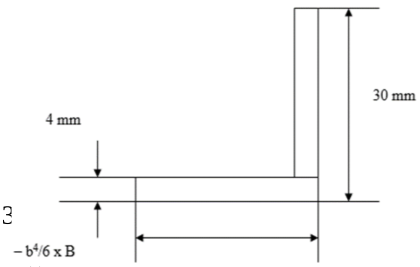
$Z = \frac{□^3}{6} - \frac{□^4}{6} * B$…………………………….…………..(28) [9]

$= \frac{30^3}{6} - \frac{26^4}{6} * 3$

Z = 1961mm³

Checking bending stress induced in L section

Figure 3.3



σ_b induced = $\frac{□}{□}$

σ_b induced $= \frac{116250}{1961} = 59.3$ N / mm^2

σ_b = Permissible bending stress = 160 N/mm²

The design is safe because the induced stress is less than the allowable stress

σ_b induced $< \sigma_b$ allowable

Hence our design is safe.

VII. Materials

Table 2 : Part name and Material of differential

Sr No	Part Name	Material
1	Differential Omni Car	AL Casting
2	Spiral Bevel	Alloy Steel
3	Bearing	Chrome Steel
4	Ratchet Mechanism	Stainless Steel
5	Solenoid 12v	Mild Steel
6	Battery 6v	Dry
7	DC Gear Motor 12v	Carbon And Graphite
8	Pulley	Anodized Aluminium
9	Motor Base	Mild Steel
10	Switch Box And Wiring	Standard
11	Angle Frame	Mild Steel
12	Belt	Rubber
13	Nut Bolt Washer	Carbon Steel

VIII. Numerical Analysis

Solid works software were used to make design of 3D modal of a differential locking system. First select the part modelling system in solid works select the plane on which u want to create the sketch. After selecting sketch draw lines joining each other as per the profile. Using 3D convertor tools extrude the profile to the required length as per symmetry considered. With the detailed part design of each part, let's move on the assemble modelling and align each sub parts as one assembled product and draft the design for the fabrication purpose. The software used is Solid works 2023 version for preparing the model of the setup. [5]

IX. Software Model

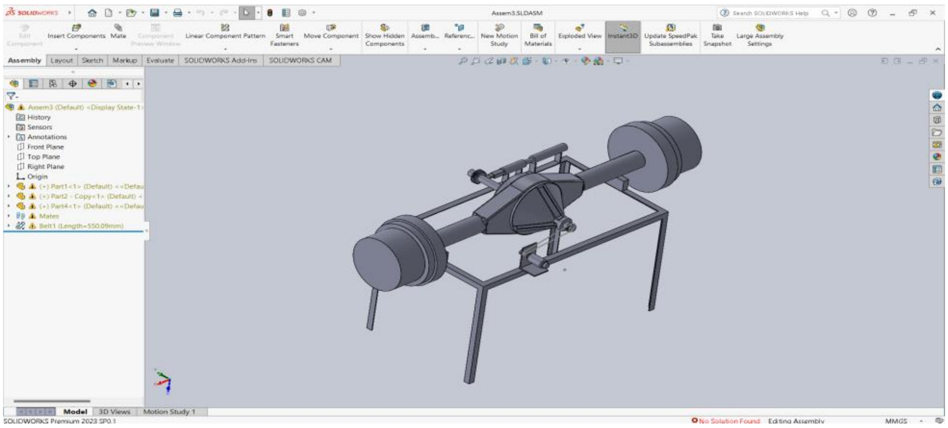


Figure 5 : 3D Model of Anti-Reverse Differential Mechanism

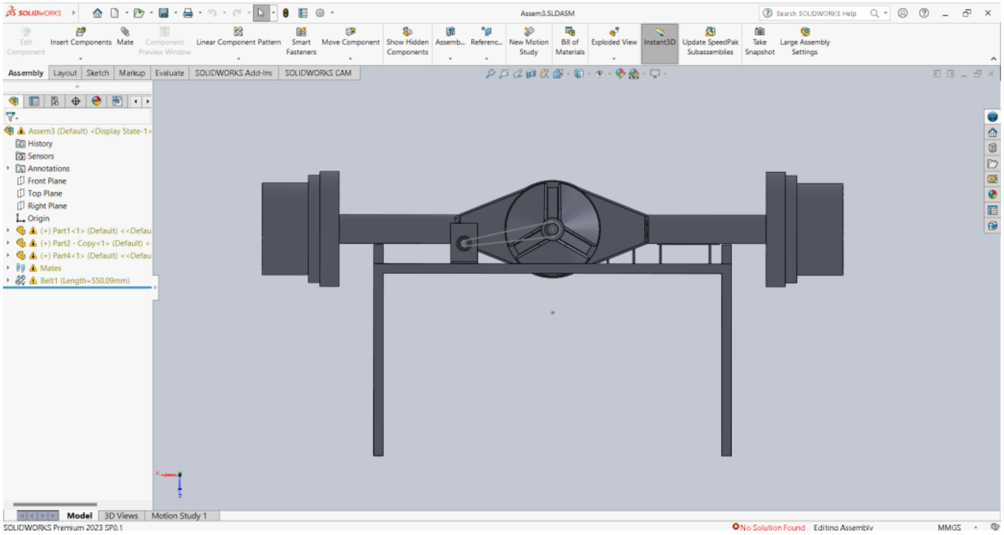


Figure 6 : Front View of Anti-Reverse Differential Mechanism

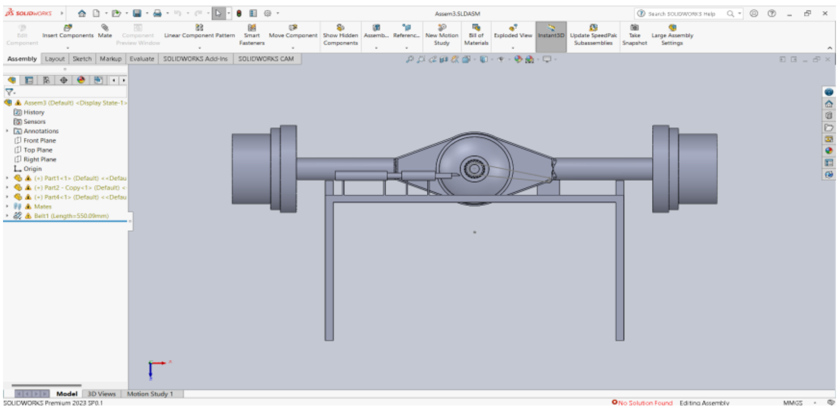


Figure 7 : Rear View of Anti-Reverse Differential Mechanism

X. FEA ANALYSIS of Locking Mechanism

10.1 Geometry

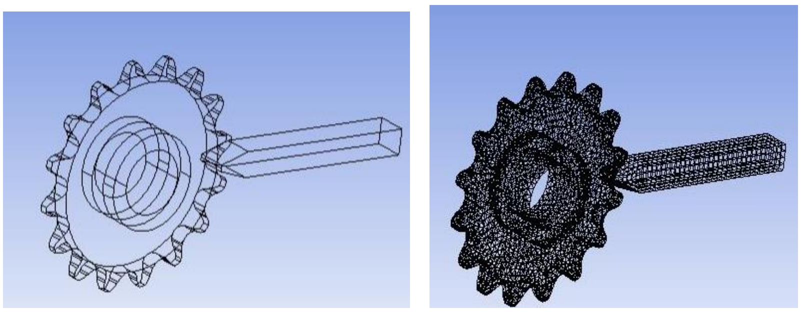


Figure 4.4 : Geometry Importation in the Ansys Modal & Mesh Configurations

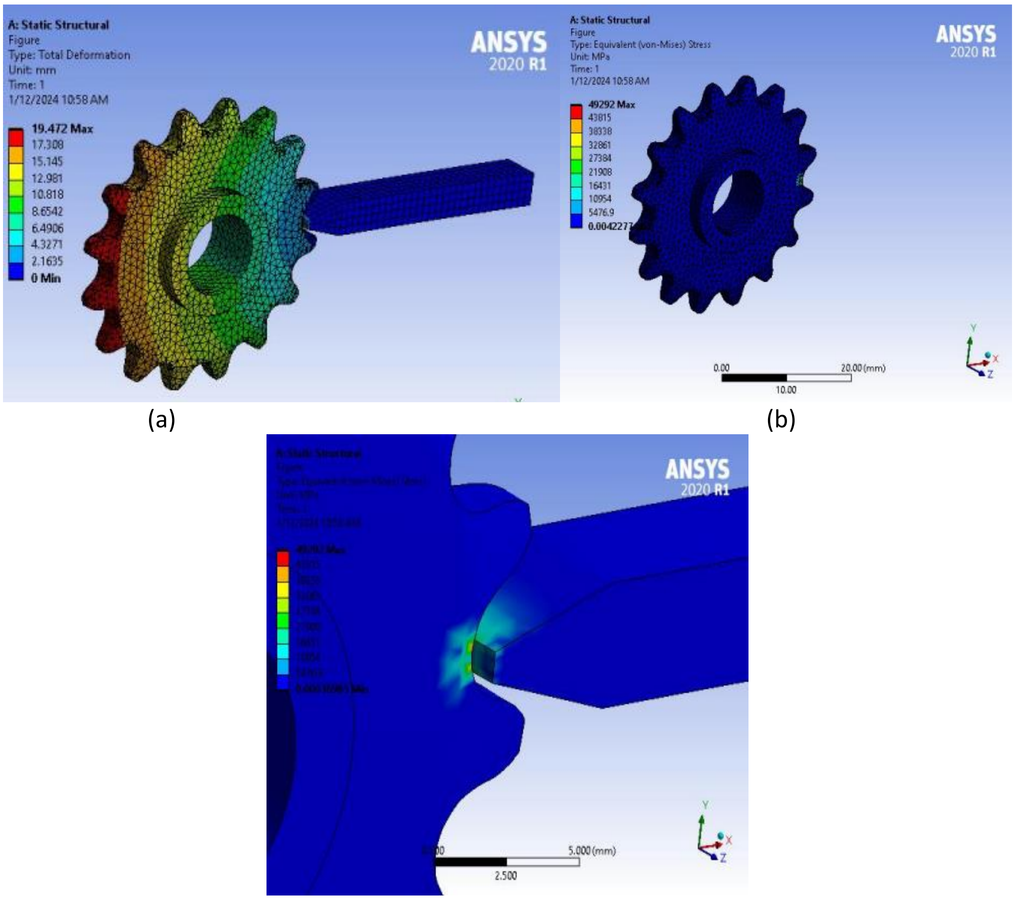
10.2 Number of Nodes and Elements generated

- Nodes 43804
- Elements 25293
- Defined Size = 1 mm
- Shape = Tet Mesh

10.3 Boundary conditions

- Momentum = 20280 N.mm
- Locking system = Fixed Support

10.4 Results Defamation & Stress on locking system



(a) (b)

(c)

Figure 8 : a) Deformation b & c) Stress Concentration factors

In our project, Finite Element Analysis (FEA) played a crucial role in ensuring the integrity and safety of the sprocket design. Initially, theoretical calculations were employed to estimate the torque that the sprocket could withstand. However, to validate these calculations and ensure the reliability of our design, we conducted a detailed analysis of the sprocket using ANSYS software. By inputting the calculated torque into the FEA model, we simulated the behavior of the sprocket under this specific load condition. The results of this analysis provided valuable insights into the structural response of the sprocket, allowing us to assess its safety and performance. Upon testing the sprocket under the calculated torque condition, we confirmed that it remained within acceptable safety margins, experiencing neither failure nor excessive deformation. This validation process not only affirmed the accuracy of our theoretical calculations but also provided assurance regarding the suitability of the sprocket design for its intended application. Through the combination of theoretical calculations and FEA, we were able to confidently conclude that the sprocket design is robust and capable of withstanding the expected torque, ensuring its safe and reliable operation in real-world conditions.

Fabricated Images

Figure 9 : Differential Gears

Figure 10 : Frame & Linear Actuator for locking purpose with locking bar

Figure 11 : Assembly of Anti-reverse differential locking system

XI. Conclusion

- The proposed Anti-reverse Differential Locking system has been successfully designed using Solid works software and static structural analysis of locking mechanisms has been solved for the deformation and stress concentration factors.
- The locking of the differential provides minimum traction by reducing the slip of wheels.

● In this proposed paper a detailed Design Construction, Structural Integrity and Fabrication process has been explained to provide optimized safety to the passengers.
● To increase a machine's dependability, effectiveness, and safety, the anti-reverse differential mechanism is necessary.
● It prolongs the life and improves the performance of equipment by avoiding undesired rearward movement.

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