Design and analysis of 4WD 2-stage reduction gearbox for mBAJA buggy using finite element method (FEM).

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Abstract - *When it comes to a student design competition for ATVs, Baja SAE India has always been a pioneer. Their racing events on various tracks for various events (such as braking, acceleration, and endurance) necessitates varied performance parameters, which must be met by the powertrain system in order to achieve greater vehicle performance. Things have altered with their most recent rules modifications, and the significance of a better and efficient engine system has grown.*

*They have incorporated the **4-WD** option in their most recent rulebook version. The purpose of this study is to discuss how to develop and optimize the **4-WD** powertrain system for mBAJA. The event is powered by a **Briggs and Stratton** engine, which is required of all teams*

***Key Words*: Single stage 4WD reduction gearbox, BAJA SAE INDIA, All-terrain vehicle (ATV), differential, gear design, simulation, CAD, FEM**

1. INTRODUCTION

Baja SAE Indian is a student design competition that compels the students to design and fabricate their own ATV which must comply with the rule book. The designing process starts together in five different departments but in this paper, we will focus entirely on powertrain system designing and its optimization. Engine specifications are given in table Ⅰ.

TABLE Ⅰ. ENGINE SPECIFICATION

Serial . no	Specification As per BAJA 2021 rule book	
	part	***Specification***
1	Engine power	10 HP
2	Engine torque	19.67 N-m
3	bore/ stroke	3.12”/2.44”
4	Displacement	305 cc
5	Compression ratio	8.1

From various events of this competition, three of them challenge the powertrain system the most.

- Acceleration test (it requires very high-power transmission at very high torque)
- Brakes test (this involves panic braking)
- Endurance race (this is the most demanding part of the event and requires continuous acceleration and braking).

Our gear train system consists of a 2-stage reduction gearbox, paired with Gazed GX9 CVT. The propeller shaft goes from idler gear to the front differential for transferring the power to the front wheels.

For the front, we decided to go with the locking differential as compared to the rear, where we decided only to have a solid shaft.

This is so because not having a differential, especially the locking one will affect the steering performance and increase the overall turning radius of the vehicle. The locking differential which we will be using is the Torsion differential.

2. MATERIAL

After looking into the various materials available for the easy fabrication of gears and considering our loading conditions. It has been decided AISI 4340 will be good for reduction gears and differential also.

AISI 4340 has excellent fatigue strength and can maintain its properties even at high temperature conditions along with good wear strength.

Material’s mechanical properties are given below in table Ⅱ

TABLE II. MECHANICAL PROPERTIES

Serial. no	Mechanical properties of AISI 4340	
	Mechanical properties	***Values***
1	Tensile Strength, Ultimate	1110 MPa
2	Tensile Strength, yield	720 MPa
3	Modulus of elasticity	205 GPa
4	Bulk modulus	140 GPa
5	Poisson's ratio	0.29
6	Shear modulus	80 GPa

3. METHODOLOGY

There is a stringent restriction on parameters that you cannot exceed, like braking distance, minimum acceleration, turning radius and top speed.

Two of which, top speed and minimum acceleration influence the design of the gear train system.

First, we calculated the maximum tyre force which is required to meet all the boundary constraints like acceleration and top speed within a given amount of engine power. That is the design of the gear system is optimized and does not go under any failure as our aim to design the unit which provides the output velocity @60kmph and able to climb 35-degree slope to provide both high speed and torque

The rest of the design procedure is in the following ways:

Calculation of Gear Ratios → Design of Gears → Calculation of Loads acting on the gears → Gear Materials Selection → simulations of gears (under various extreme loading conditions) → Calculation of factors depends upon the gears.

TRANSMISSION SPECIFICATION

TABLE III. TRANSMISSION SPECIFICATION

Serial . no	Specification As per BAJA 2021 rule book	
	Transmission specification	***Values***
1	Engine power (P_E)	10 HP/7.5KW
2	Engine torque	19.67 N-m
3	Maximum engine speed	3800 RPM
4	Wheel diameter	0.5588 m
5	Weight of vehicle (M)	250 kg
6	maximum speed of the vehicle	60 kmph
7	CVT ratio [max, min]	[3.9, 0.9]
8	Efficiency of the transmission (η_t)	90%
9	Rolling resistance (μ)	0.15

Anticipating the worst-case scenarios, the value of Rolling Resistance is taken as 0.15 and efficiency is taken as 90%. these values are validated by various other research paper.

4. CALCULATION

A. 2-stage reduction Gear calculation

Calculation of gear ratios

considering the technical specification as per requirements

1. rolling resistance (F_r)= $\mu \times M \times g \cos 35 = 301$ N

2. grade resistance (F_g)=$M \times g \times \sin 35 = 1405.26$ N

3. drag resistance (F_d)= $\rho \times A \times C_d \times V^2 = 65$ N

where C_d=0.4, ρ=1.225 kg/m³, A=1.5 m², V=60 km/h

thus, total resistance(R)=$F_r+F_g+F_d$

Tractive force ≥ total Resistance

$(19.66 \times .085 \times 3.9 \times GR)/(r_{dynamic}) \geq 1803.76$

$(19.66 \times .90 \times 3.9 \times GR)/(.2682) \geq 1803.76$

GR ≥7.0072

considering the gradability and top speed final GR is chosen to be the value of 7.11

since the gear train system is designed for 2 stage reduction. so, reduction at each stage will be= $(7{,}11)^{1/2}$

so, the reduction at each stage is = 2.66

Calculation of minimum number of teeth, module, FOS, face width for 1st stage reduction

b = $S_{ut}/3$ = 585.33 N/mm2

$b \times Y_p < b \times Y_g$ (prioritizing pinion since it's the weakest)

Z_p:14 (see reference [3])

Z_g= 14×2.66 =38

D_p = 14M

D_g = 38M (where M is module)

M_T : torque on the Pinion

M_T =(60 × 1000000 ×P_E×η_T)/ [2× π× (3800 / 3.9)]N-mm

M_T = 66178.39 N-mm

C_S = 1.5 (value taken from data handbook of VB.BHANDARI)

C_V ={ 6 / (6+V) }

V_p = { (π × D_P× N)/ (60×1000) } m/s

V_P= (π × 14 M × 974)/60000 m/s

V_P= 0.714M m/s

C_v=(6 / (6+ .714M))

P_T = (2M_T)/D_P N

P_T = (2×66178.39)/14M N

Let Face width (W_F) = 8M

F_b = M × F × b × Yp

F_b = M× 8M× 585.33× 0.279

F_b = 1306.456 M^2 N

F_w= D_p × F× Q×K

Let Fwidth= 8M mm

Q = [(2 ×Z_g)/(Z_g+Z_p)]= [(2×38)/(38+14)]= 1.4615

K = 0.16× (BHN/100)^2

K=3.8416

F_w = 14M×8M×1.4615×3.8416= 628.82 M^2

As F_w < F_b (So We have to Design the gear against wear load)

P_{eff} = (C_s/C_v)×Pt

P_{eff} = [1.5/{6 / (6+.714M)}]×{ (2×66178.39)/14M}]

F_w = FOS_{wear}× P_{eff}

628.8M^2=1.25×[1.5/{6/(6+.714M)}]×{(2×66178.39)/14}]

(assuming FOS_{wear}=1.25, since it will be enough for wear load.)

0.213×M^3 - 0.714×M -6=0

M = 3.40

For Checking the beam Stress

F_b = P_{eff} * $FOS_{bending}$

1306.456 * 3.4^2 = 6050 * $FOS_{bending}$

$FOS_{bending}$ = 15998.5/6050

$FOS_{bending}$ =2.49 (for FOS_{wear}=1.25)

Module(M)=3.4

Zp=14,Zg=38

Dp= 14*3.5=45.5mm

Dg = 38*3.5= 123.5mm

W_F = 8* 3.5= 28mm

TABLE IV. 1STT STAGE REDUCTION CALCULATION RESULT

Serial. no	1st stage reduction result	
	Specification	***value***
1	Pinion diameter (D_p)	45.5 mm
2	Idler diameter (D_g)	123.5 mm
3	Face width (W_F)	28 mm
4	Number of teeths of pinion (Z_p)	14
5	Number of teeths of idler gear (Z_g)	38
6	Module (M)	3.4
7	$FOS_{bending}$	2.49
8	FOS_{wear}	1.25

Calculation of minimum number of teeth, module, FOS, face width for 2nd stage reduction by performing the similar calculation methodology for 1st stage reduction, the second stage reduction calculation is shown in table V

TABLE V. 2ND STAGE REDUCTION CALCULATION RESULT

Serial . no	2nd stage reduction result	
	Specification	***value***
1	Pinion diameter (D_p)	59.5 mm
2	Idler diameter (D_g)	172.25 mm
3	Face width (W_F)	34 mm
4	Number of teeths of pinion (Z_p)	15
5	Number of teeths of idler gear (Z_g)	41
6	Module (M)	4.25
7	$FOS_{bending}$	1.83
8	FOS_{wear}	1.25

Torsion differential calculation

Torsion differential consists of various gears like spur gear, worm gear and worm wheel, all three works together to provide the traction in all type's road conditions.

Specification values of all these gears are calculated by using the same calculation methodology as used in calculation of specification of gears at 1st stage reduction.

Design specification of the spur gear

Design of the spur gear is finalized by using the same methodology as used for the calculation of the 1st stage reduction, and the design specification values are given in table VI.

TABLE VI. SPUR GEAR DESIGN CALCULATION RESULT

Serial . no	spur gear design calculation	
	Specification	***value***
1	Module (M)	2
2	Pressure angle (α)	20°
3	Number of teeth (z)	20
4	Center distance (a)	40 mm
5	Pitch circle diameter (d)	40 mm
6	Base diameter (d_b)	35 mm
7	Addendum (h_a)	2 mm
8	Dedendum (h_b)	2.5 mm
9	Tooth depth (h)	4.5 mm
10	Tip diameter (d_a)	44 mm
11	Root diameter (d_r)	35 mm
12	Circular pitch (P_c)	6.28 mm
13	Diametral pitch (P_a)	0.5 mm
14	Tooth thickness (T_h)	3.14 mm
15	Fillet radius (r)	0.8 mm
16	Working depth (WD)	4mm

Design specifications of the worm gear are given in table VII and off worm wheel are given in table VIII

TABLE VII. WORM GEAR DESIGN CALCULATION RESULT

Serial . no	Design specification of worm gear	
	Specification	***value***
1	Module (M)	2
2	Pressure angle (α)	20°
3	Number of teeth (z)	20
4	Pitch circle diameter (d)	40 mm
5	Addendum (h_a)	2.39 mm
6	Lead angle	45 mm
7	Face width	30 mm
8	Helix angle	45 mm

A.

Table VIII. WORM WHEEL DESIGN CALCULATION RESULT

Serial . no	Design specification of worm gear	
	Specification	***value***
1	Module (M)	2
2	Pressure angle (α)	20°
3	Number of teeth (z)	10
4	Pitch circle diameter (d)	20 mm
5	Addendum (h_a)	2.39 mm
6	Lead angle	45 mm
7	Face width	55 mm
8	Helix angle	45 mm

5. CAD DESIGN AND FEM RESULT

Software's used for CAD and FEM are SolidWorks 2020 and ANSYS 2019

- ***CAD and FEA of 2-stage reduction gear***

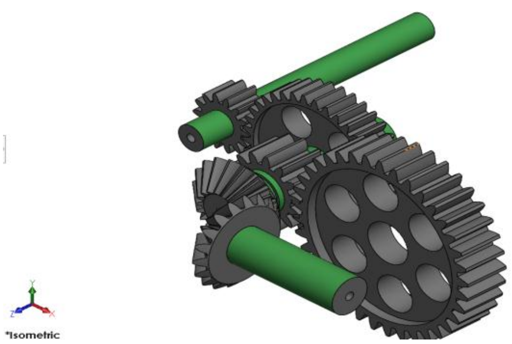

Figure 1; CAD model of 2-stage reduction 4WD gear system

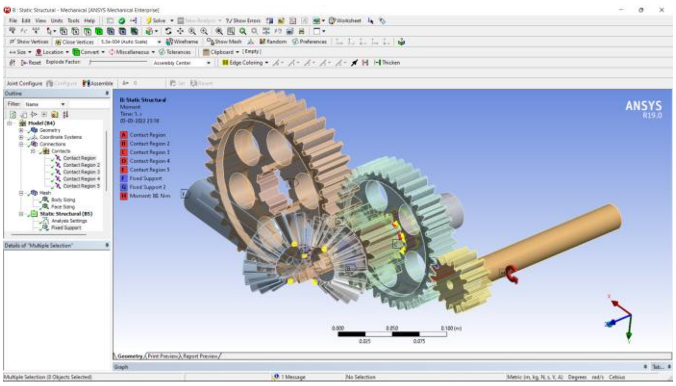

Figure 2; contact region and boundary conditions applied for the simulation of 2 stage 4WD gear reduction system

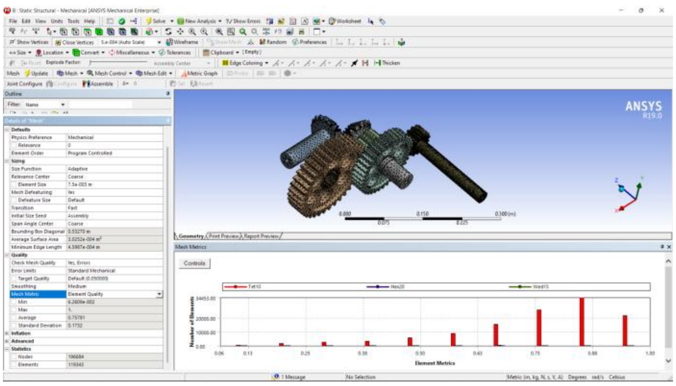

Figure 3

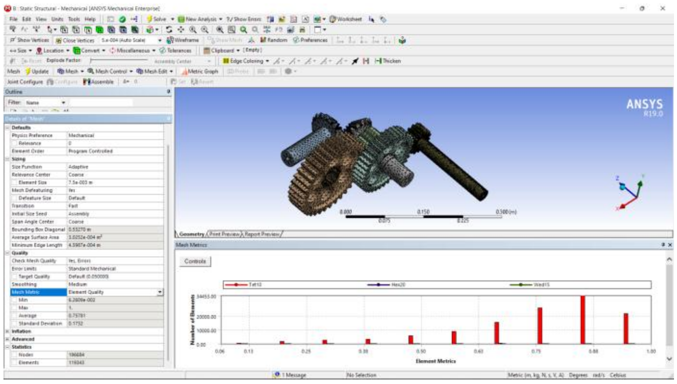

Figure 4

Figure 3 and 4 ;showing number of elements and element quality and skewness of tetrahedral, hexahedral and wed15 elements

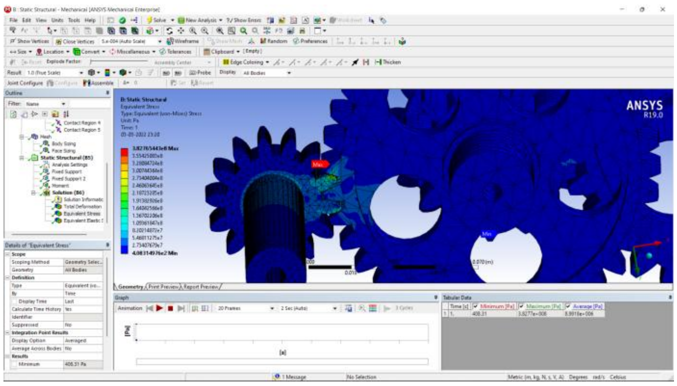

Figure 5; showing the maximum and minimum equivalent von-mises stress on the gear-train system

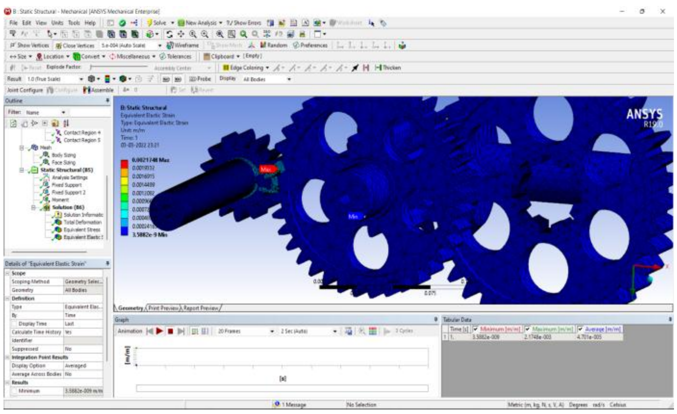

Figure 6; showing the maximum and minimum equivalent elastic strain on gear-train system

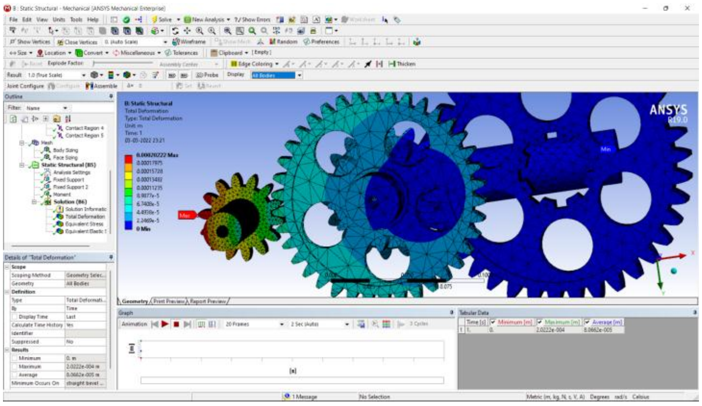

Figure 7; showing the maximum and minimum total deformation when engine torque is applied

physics preference	**mechanical**
element size and type	**7.5e-003 m (tetrahedral, hexahedral, and Wed15)**
number of nodes	**196684**
number of elements	**119343**

Table Ⅸ showing the number of elements, element size and type of elements used during the simulation

value	***maximum***	***minimum***
equivalent von-mises stress	**3.8277e+008 [Pa]**	**408.31 [Pa]**
total deformation	**2.022e-004 [m]**	**0 [m]**
equivalent strain	**2.1748e-003**	**3.5882e-009**

Table Ⅹ showing the simulation result of 2 stage 4-WD reduction gear train system.

- ***CAD and FEA for torsen differential***

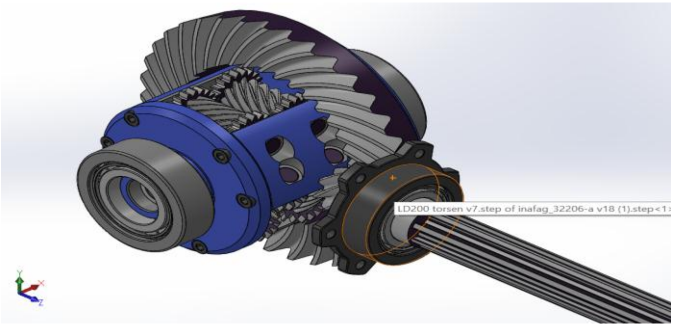

Figure 8 CAD model of torsion differential.

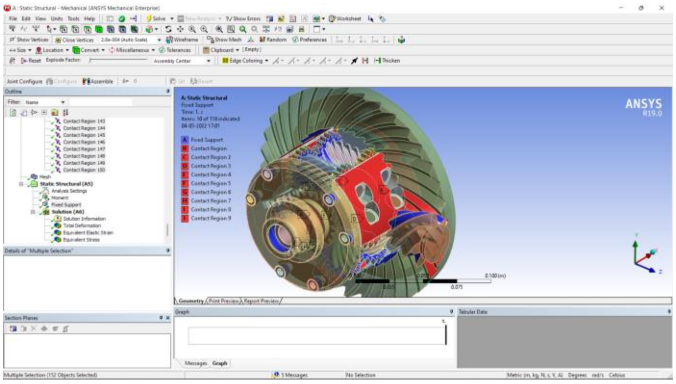

Figure 9; showing the contact regions between the different parts of torsen differential.

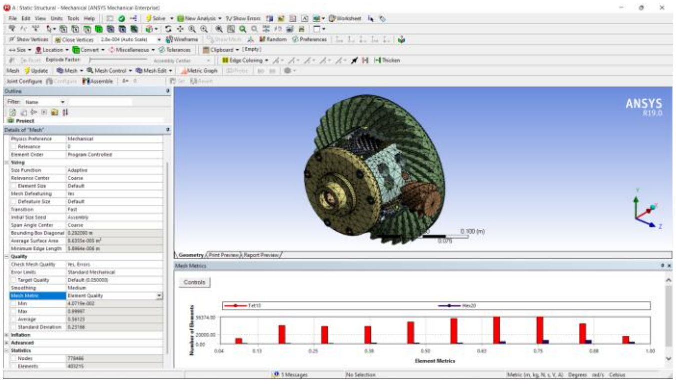

Figure 10;

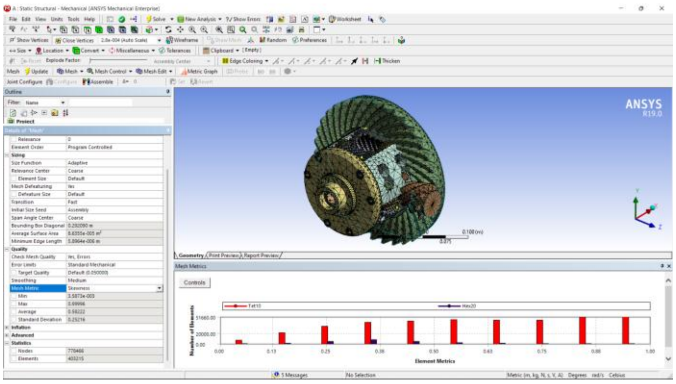

Figure 11.

(figure 10 and 11; showing number of elements and element quality and skewness of tetrahedral, and hexahedral elements)

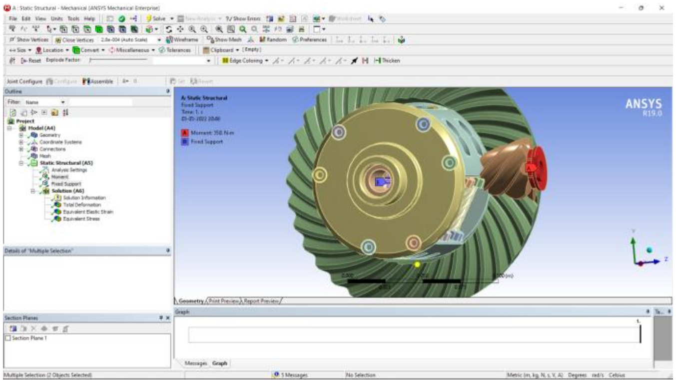

Figure 12; showing the boundary condition applied during the time of simulation.

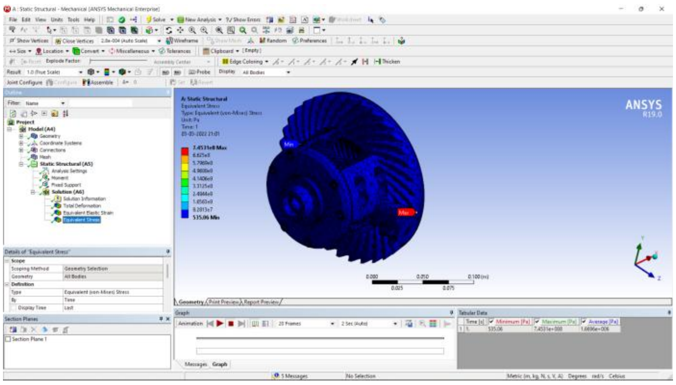

Figure 13; showing the maximum and minimum von-mises stress developed during the time of simulation.

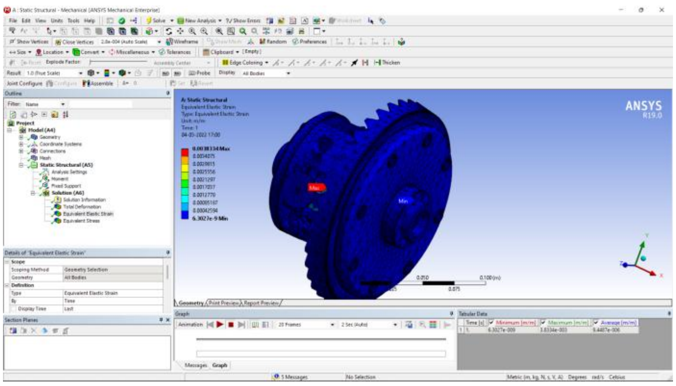

Figure 14; showing the maximum and minimum elastic strain developed during simulation as per our loading condition

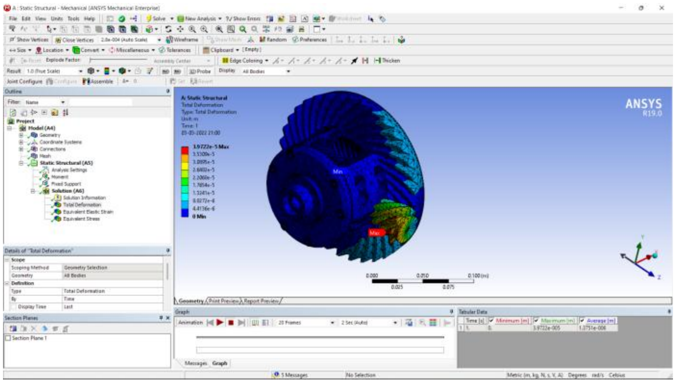

Figure 15; showing the maximum and minimum deformation produced during the time of loading.

physics preference	**mechanical**
element size and type	**program controlled (*due to low processing power)**
number of nodes	**778466**
number of elements	**403215**

Table XI showing the number of elements, element size and type of elements used during the simulation

value	***maximum***	***minimum***
equivalent von-mises stress	**7.4531e+008 [Pa]**	**535.06 [Pa]**
total deformation	**3.9722e-005 [m]**	**0 [m]**
equivalent strain	**3.8334e-003**	**6.3027e-009**

Table XII showing the simulation result of 2 stage 4-WD reduction gear train system.

TABLE XII I . SIMULATION RESULT SHOWING THE COMPARISON BETWEEN HAND CALCULATION AND FEA RESULT

Serial no.	**FOS calculation by hand calculation and by FEA**		
	part	***$FOS_{hand\ calculation}$***	***$FOS_{simulation}$***
1	2-stage reduction gear	1.85	1.82
2	torsen differential	1.63	1.52

6. CONCLUSIONS

The gearbox has been meticulously developed using the most up-to-date calculations and analyses. The ultimate gear ratio is 7.11, with the first stage at 2.66 and the second stage at 2.66. AISI 4340 is the material chosen for the designing of the gears. All of the gear analysis was completed and the results were reported. The gearbox is employed in a variety of applications.

All parts that can endure harsh circumstances in an ATV for about 2 years with regular maintenance and the calculation for gear ratio, minimum number of teeth, bearing forces for advanced helical gears, and analysis. Are done on all the parts to withstand all the extreme condition for events like BAJA SAE

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