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Design and Analysis of an Open Differential

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Abstract: A transmission or gearbox provides speed and torque conversions from a rotating power source to another device using gear ratios. The most common use is in motor vehicles, where the transmission adapts the output of the internal combustion engine to the drive wheels. Such engines need to operate at a relatively high rotational speed, which is inappropriate for starting, stopping, and slower travel. The transmission reduces the higher engine speed to the slower wheel speed, increasing torque in the process. We have designed a differential gearbox and tried to create the frictional contact between two mating gears. And we have performed the structural analysis on gear box by providing the torque to the assembly of crown gear and pinion gear, assembly of inner gears- spider gears and side gears and crown gear with the cage to attach spider gears. We have selected two kinds of alloy steel and have compared the factor of safety and structural analysis of the both.
Keywords: Differential, Bevel Gears, Ansys, Solidworks, SKF Bearing, FEA

I. INTRODUCTION

Differential is an integral part of all four wheelers. Wheels receive power from the engine via a drive shaft. The main function of the differential is to allow the wheels to turn at different rpm while receiving equal power from the engine. In the case of vehicle turning right, the left wheel has to travel more distance as compared to the right wheel this means the left wheel has to rotate at a higher speed. If these wheels were connected using a solid shaft, the wheels would have to slip to accomplish a turn. This is exactly where a differential comes. The ingenious mechanism in a differential allows the left and right wheel to turn at different rpm transferring power to both wheels. The power from the engine is transferred to the ring gear or crown gear through a pinion gear. Spider gears, connected to the crown gear, have combined rotation- it spins about its own axis and also rotates about the axis of the crown gear. Side gears are attached to the half shafts that are connected to the wheels. The average of the rotational speed of the two driving wheels equals the input rotational speed of the drive shaft. An increase in the speed of one wheel is balanced by a decrease in the speed of the other. Apart from allowing the wheels to run at different rpm, the differential has two more functions. First is speed reduction at pinion ring gear assembly, this results in torque multiplication. The other function is to turn the power flow direction by 90 degrees.

II. DESIGN PROCEDURE

A. Approach

The Design process initially starts with considering the number of teeth, module and reduction of bevel gear set. Furthermore, according to the input torque applied, the forces acting on the bevel gear set are to be determined. The major causes of failure for this bevel gear set are Bending and Pitting. In order to withstand these forces and prevent failure, the gear material should have a high ultimate tensile strength and high Brinell hardness number. Considering this fact two alloy steels, i.e., 20MnCr5 and 15Ni4Cr1 are to be chosen as the material for the gearbox because of their high strength. Later after performing static structural analysis using Ansys Workbench software package, the respective deformation and equivalent stresses for both the materials will be determined. Material Properties are given below:

Alloy Steel	Ultimate Tensile Strength	Tensile Yield Strength	BHN
15Ni4Cr1	1500 N/mm^2	385 N/mm^2	650
20MnCr5	1158 N/mm^2	420 N/mm^2	600

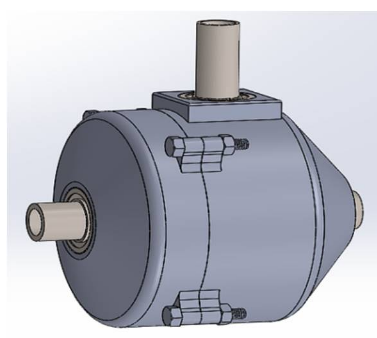

Fig.1 Open Differential with Casing

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B. Input Parameters

Input Power: -	5.8913 kW
Input Speed: -	4100 rpm
Input Torque: -	13.7321Nm
Reduction Ratio: -	3:1

III.DESIGN CALCULATION

A. For material: - 15Ni4Cr1

It is highly recommended that a separate excel sheet should be made to determine the fos(beam) and fos(wear) for various permutation and combination for different number of teeth and module. After various iterations the following gear dimensions were finalized and their respective factor of safeties are calculated below:

1) For Drive Pinion And Crown Gear

Module	3mm
Reduction	3:1
No. of teeth of Pinion (zp)	19
No. of teeth of Gear (zg)	57
PCD of Pinion (Dp)	57
PCD of Gear (Dg)	171

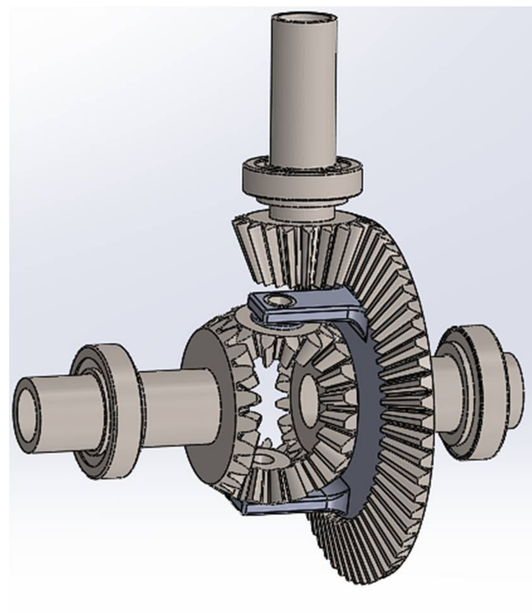

Fig.2 Construction of an Open Differential

$A_o = \sqrt{\left(\frac{Dp}{2}\right)^2 + \left(\frac{Dg}{2}\right)^2} = 90.125\text{mm}$

b = Ao/3 = 90.125/3= 30.04167 = 30mm

Since pinion is weaker, Design is based on pinion:

tan(γ)= zp/zg => tan(γ) = 1/3

$r_m = \frac{Dp}{2} - \frac{b*\sin(\gamma)}{2} = (28.75- 4.75)= 23.75$ mm

$P_t = \frac{M_t}{r_m} = \frac{13.732.1}{23.75} = 578.2$ N

For IS Grade 5 pitch error is given by : $e = \left(5 + 0.4\left(m + 0.25\sqrt{PCD}\right)\right)$

$e = e_p + e_g = \left(5 + 0.4\left(3 + 0.25\sqrt{57}\right)\right) + \left(5 + 0.4\left(3 + 0.25\sqrt{171}\right)\right)$

e= 6.95+7.5 = 14.45 μm = 0.014 mm

c=11400

2) Dynamic Load is calculated by using Buckingham Load equation as follows:

$P_d = \frac{21v(Ceb+Pt)}{21v+\sqrt{Ceb+Pt}} = \frac{21*12.23*(11400*0.014*30+578.2)}{21*12.23+\sqrt{11400*0.014*30+578.2}} = \frac{1378201.146}{330.08} = 4175.35$ N

3) Hence the Effective is calculated by Considering the Service factor to be 1.25 :

$P_{eff} = C_s*P_t + P_d = (1.25) * 578.2 + 4175.35 = \underline{4898.1}$

4) Thus, the Beam strength is given by:

$S_b = m * b * \sigma b * Y * \left(1 - \frac{b}{Ao}\right)$; [Y is Lewis form factor for virtual no. of teeth]

$v = \frac{\pi*57*4100}{60000}$ = 12.23 m/s [Pitch line Velocity]

Virtual number of teeth , z' = $\frac{zp}{\cos(\gamma)} = \frac{19*\sqrt{10}}{3}$ = 20.028 => $Y = \pi * \left(0.154 - \frac{0.912}{z'}\right)$ = 0.34075

$S_b = 3 * 30 * 500 * 0.34075 * \left(\frac{2}{3}\right)$ = <u>10222.5</u> N

The factor of safety for beam strength is given by:

Fos (beam) = $\frac{Sb}{Peff} = \frac{10222.5}{4898.1}$ = **<u>2.087</u>**

5) The Wear Strength is given by:

$S_w = \left(\frac{0.75*b*Q*Dp*K}{\cos(\gamma)}\right)$

$Q = \frac{2*Zg}{Zg+Zp\tan(\gamma)} = \frac{2*57}{57+19/3}$ = 1.8 [Ratio factor]

$K = 0.16 * \left(\frac{BHN}{100}\right)^2 = 0.16 * \left(\frac{650}{100}\right)^2$ = 6.76 [Load stress factor]

$S_w = \left(\frac{0.75*30*1.8*57*6.76}{\frac{3}{\sqrt{10}}}\right)$ = <u>16450</u> N

Fos (wear) = $\frac{16450}{4898.1}$ = **<u>3.358</u>**

6) For Spider and Side Gear

Module	4mm
Reduction	1.6:1
No. of teeth of Pinion (zp)	12
No. of teeth of Gear (zg)	20
PCD of Pinion (Dp)	48
PCD of Gear (Dg)	80

$A_o = \sqrt{\left(\frac{Dp}{2}\right)^2 + \left(\frac{Dg}{2}\right)^2}$ = 46.64

b = Ao/3 = 46.64/3= 15.5mm

Since the side gear transmits the torque, design is based on the side gear:

tan(Γ)= zg/zp => tan(Γ) = 5/3

$r_m = \frac{Dp}{2} - \frac{b*\sin(\Gamma)}{2}$ = (40-6.646)= 33.35 mm

$P_t = \frac{M_t}{r_m} = \frac{41196.3}{33.35}$ = 1235.27 N

Pa=Pt ∗ tanα ∗ sinΓ = 385.6 N

Pr= Pt ∗ tanα ∗ cosΓ = 231.35 N

For IS Grade 5 pitch error is given by : $e = \left(5 + 0.4\left(m + 0.25\sqrt{PCD}\right)\right)$

$e = e_p + e_g = \left(5 + 0.4\left(4 + 0.25\sqrt{48}\right)\right) + \left(5 + 0.4\left(3 + 0.25\sqrt{80}\right)\right)$

e= 6.892+7.09 = 13.98 μm = 14 μm =0.014 mm

C=11400

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7) *Dynamic Load is calculated by using Buckingham Load equation as follows:*

$P_d = \frac{21v(Ceb+Pt)}{21v+\sqrt{Ceb+Pt}} = \frac{21*5.72*(11400*0.014*15.5+1235.27)}{21*5.72+\sqrt{11400*0.014*15.5+1235.27}} = \frac{445533.4884}{181.02} = 2461.24$ N

8) *Hence the Effective is calculated by Considering the Service factor to be 1.25 :*

$P_{eff} = C_s*P_t + P_d = (1.25)*1235.27 + 2461.24 = 4005.32$ N

9) *Thus, the Beam strength is given by:*

$S_b = m*b*\sigma b*Y*\left(1-\frac{b}{Ao}\right)$; [Y is Lewis form factor for virtual no. of teeth]

$v = \frac{\pi*80*1366.67}{60000} = 5.72$ m/s

$z' = \frac{zp}{\cos(\gamma)} = \frac{20*5.83}{3} = 38.86 \Rightarrow Y = \pi*\left(0.154-\frac{0.912}{z'}\right) = 0.41$

$S_b = 4*15.5*500*0.41*\left(\frac{2}{3}\right) = \underline{8419.34}$ N

The factor of safety for beam strength is given by:

$fos = \frac{Sb}{Peff} = \frac{8473.34}{4005.32} = \underline{\mathbf{2.11}}$

10) *The wear Strength is given by:*

$S_w = \left(\frac{0.7*b*Q*Dp*K}{\cos(\Gamma)}\right) = \left(\frac{0.75*15.5*1.47*80*6.76}{\frac{3}{5.83}}\right) = \underline{17960\ N}$

$Q = \frac{2*Zg}{Zg+Zp\tan(\Gamma)} = \frac{2*20}{20+12*3/5} = 1.47$ [Ratio factor]

$K = 0.16*\left(\frac{BHN}{100}\right)^2 = 0.16*\left(\frac{650}{100}\right)^2 = 6.76$ [Load stress factor]

$Fos\ (wear) = \frac{17960}{4005.32} = \underline{\mathbf{4.48}}$

B. *For Material 20MnCr5*

After various iterations the following gear dimensions were finalized.

1) *For Drive pinion and Crown Gear*

Module	4mm
Reduction	3:1
No. of teeth of Pinion (zp)	15
No. of teeth of Gear (zg)	45
PCD of Pinion (Dp)	60
PCD of Gear (Dg)	180

Similarly, after doing calculation as shown in A., we can conclude that:

b = 31.62mm, Pt = 549.284 N, P_{eff} = 5399.75 N, Sb = 9530.5 N and S_w = 12294.94 N

- ⇨ Fos (beam) = **1.764991**
- ⇨ Fos (wear) = **2.276**

2) *For Side and Spider Gear*

Module	4mm
Reduction	1.6:1
No. of teeth of Pinion (zp)	14
No. of teeth of Gear (zg)	23
PCD of Pinion (Dp)	56
PCD of Gear (Dg)	92

Similarly, after doing calculation as shown in A., we can conclude that:

b = 17.95mm, Pt = 588.5 N, P_{eff} = 3672.9 N, Sb = 5157.88 N and S_w = 8215.6 N

- ⇨ Fos (beam) = **1.4043**
- ⇨ Fos (wear) = **2.236**

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3) Factor Of Safety For Both Materials Are Summarised Below

Material	Bevel Pinion		Inner bevel Set	
	Fos(beam)	Fos(wear)	Fos(beam)	Fos(wear)
20MnCr5	1.764991	2.276	1.4043	2.236
15Ni4Cr1	2.087	3.358	2.11	4.48

4) Selection Of Bearing Bevel Drive Shafts

The arrangement of the inner bevel set is such that the radial and the tangential forces in vertical and horizontals planes respectively cancels each other out and thereby doubling the axial forces.

Total axial force (P)= 2*Pa

P=2*385.6 => P = 771.2 N

Consider the L_{10} as 100 million revolution and load factor as 1.4

C=$P * (\mathrm{L}10)^{\frac{1}{3}} * Lf$ => C= $771.2 * (100)^{\frac{1}{3}} * 1.4$

C=5011.43N

5) Selecting Single Row Tapered Roller Bearing Form Skf Catalogue

Designation	d (mm)	D (mm)	T (mm)	Dynamic load rating, C (kN)	Static load rating, Co (kN)
32007X	35	62	18	52.3	54

32007X is selected.

IV. ANALYSIS

FEA was done using the Ansys Workbench software package for two materials 20MnCr5 and 5Ni4Cr1. Static Structural analysis was performed. In case of Bevel pinion and Crown assembly, the Crown was fixed and a torque of 13.7321Nm was applied to the bevel pinion after giving it frictionless support. On the other hand, in the case of Inner bevel gear set, both of the spider gears were fixed and a torque of 41.1963 Nm was applied on both of the side gears after giving them frictionless support. For Crown with cage assembly, the structural integrity of cage was check by fixing the holes of the cage and applying a torque of 41.1963 Nm after giving frictionless support to the Crown gear. The stress and deformation for both is given below.

A. Crown and Bevel Pinion

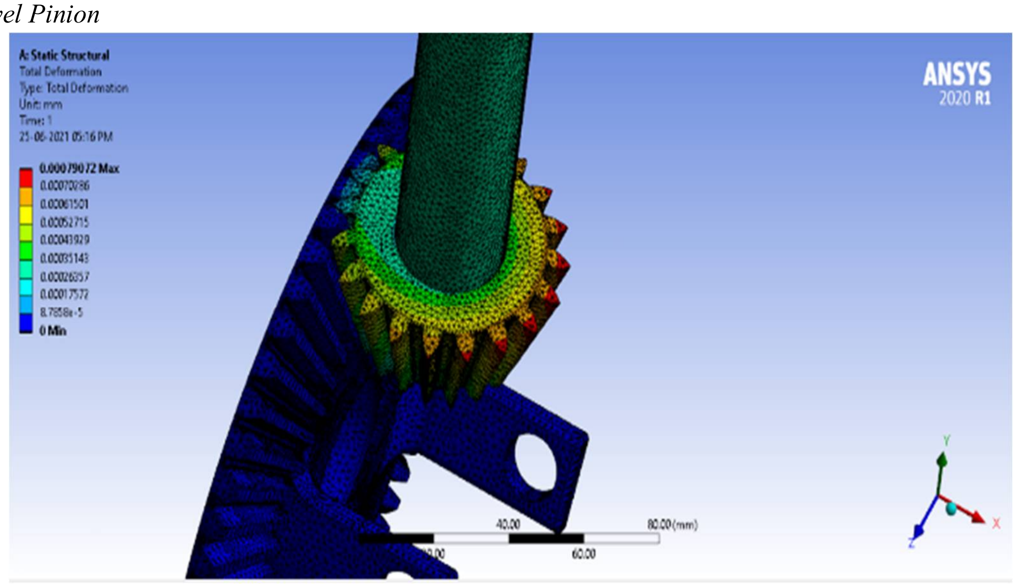

Fig.3 Total Deformation of Bevel pinion and Crown assembly with 15Ni4Cr1

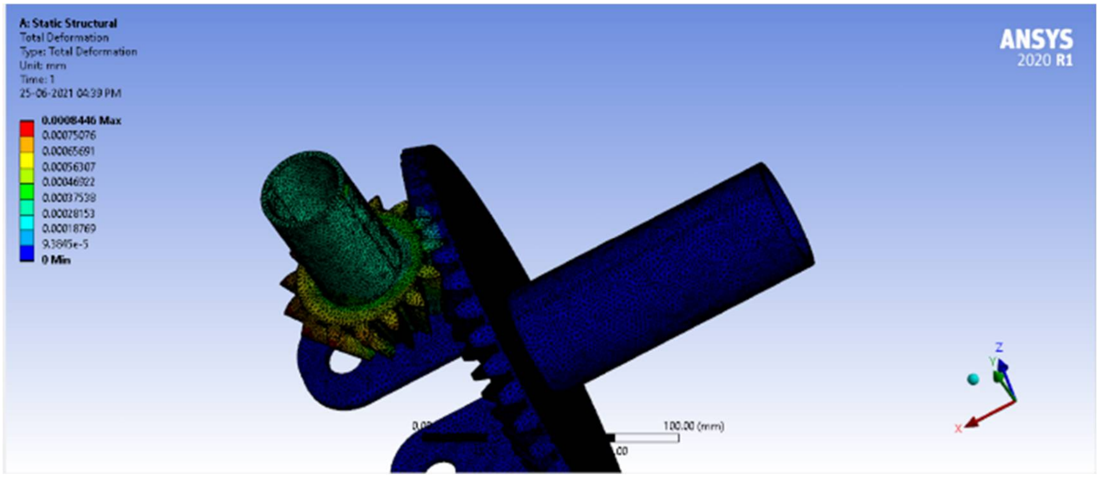

Fig.4 Total Deformation of Bevel pinion and Crown assembly with 20MnC5

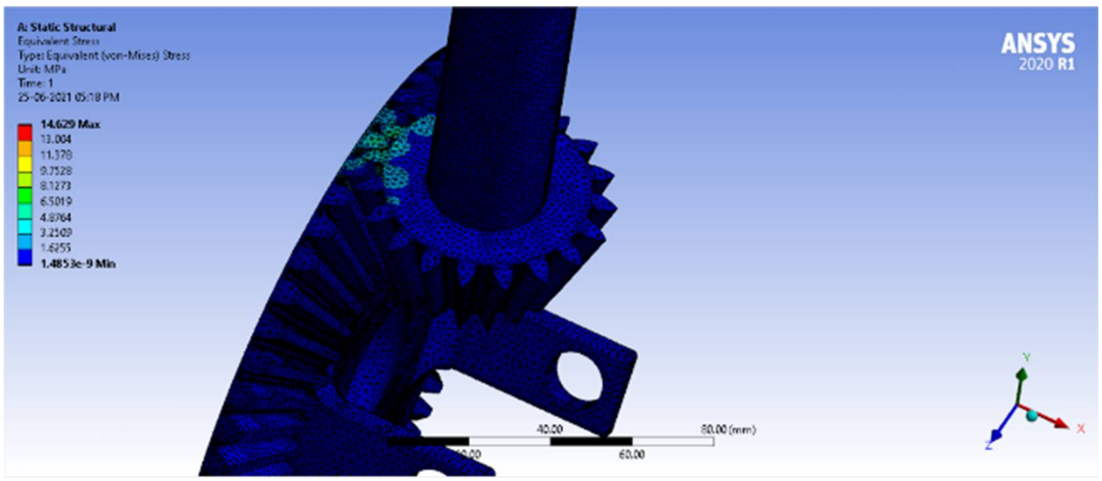

Fig.5 Equivalent Stress induced in Bevel pinion and Crown assembly with 15Ni4Cr1

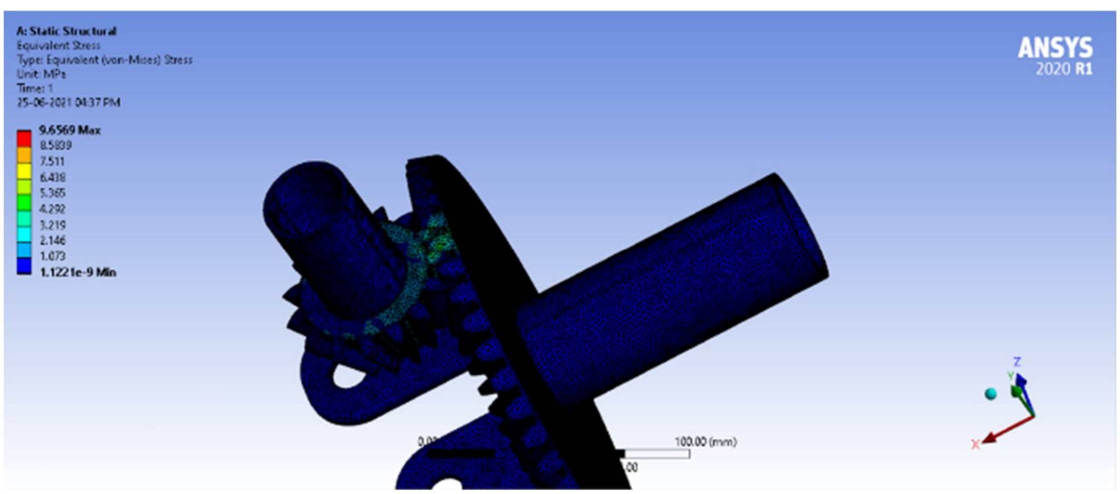

Fig.6 Equivalent Stress induced in Bevel pinion and Crown assembly with 20MnC5

Type	15Ni4Cr1	20MnCr5
Deformation (mm)	0.00079072	0.0008446
Von-Mises stress (Mpa)	14.629	9.6569

B. Caged Crown

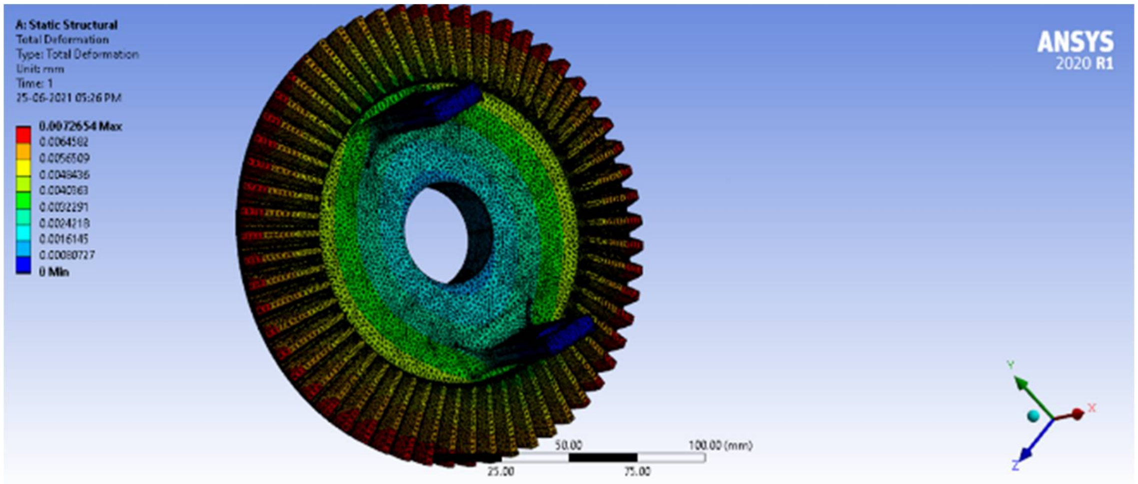

Fig.7 Total Deformation of Caged Crown assembly with 15Ni4Cr1

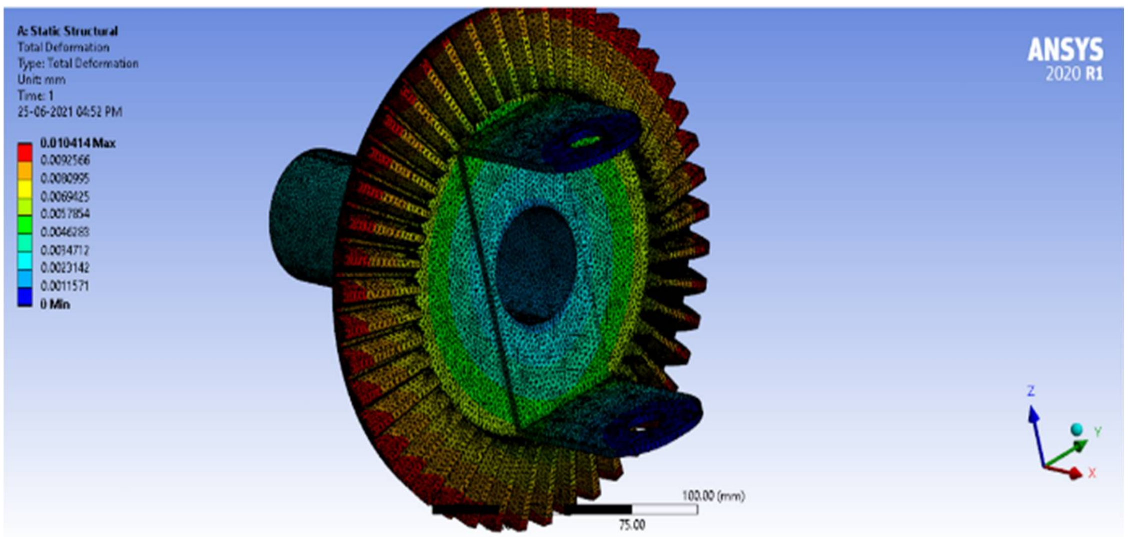

Fig.8 Total Deformation of Caged Crown assembly with 20MnC5

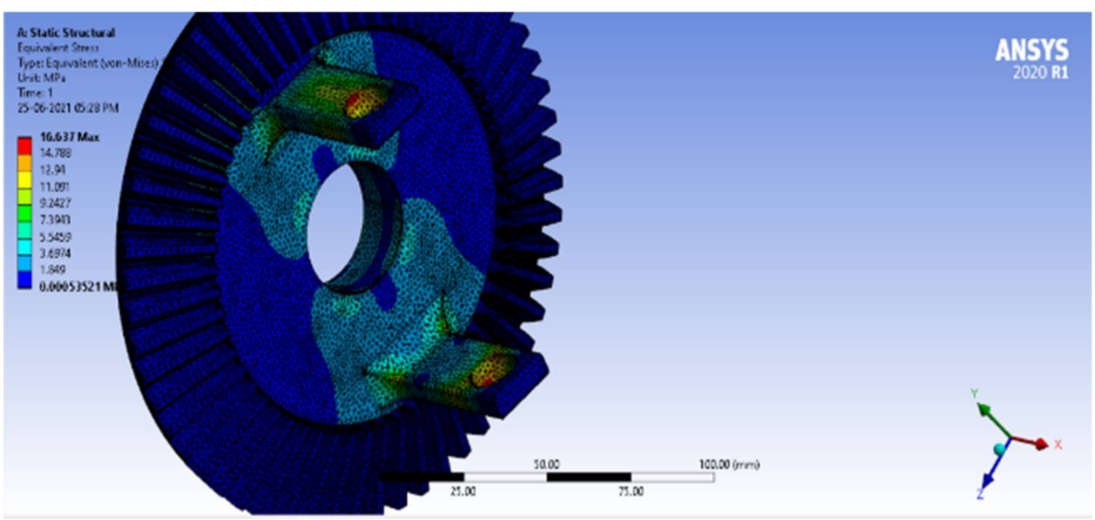

Fig.9 Equivalent Stress induced in Bevel Caged Crown assembly with 15Ni4Cr1

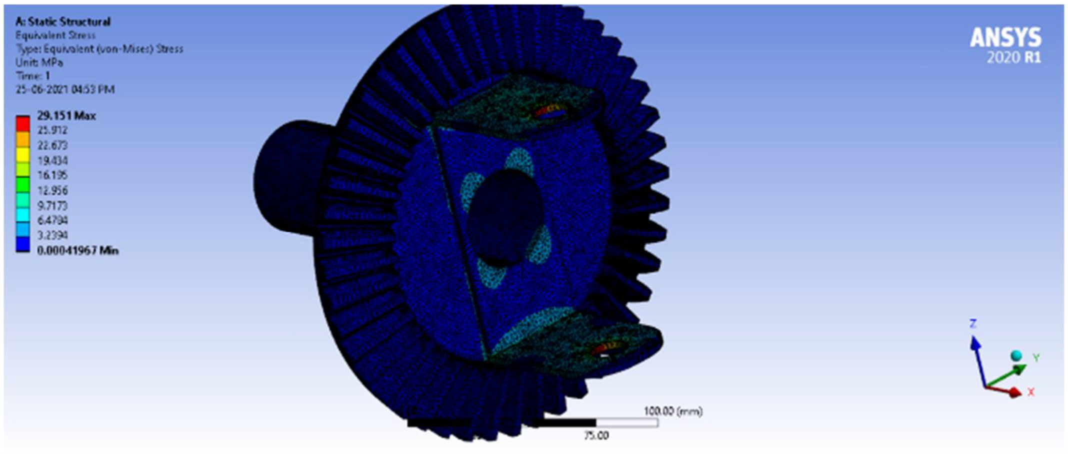

Fig.10 Equivalent Stress induced in Bevel Caged Crown assembly with 20MnC5

Type	15Ni4Cr1	20MnCr5
Deformation (mm)	0.0072654	0.010414
Von-Mises Stress (Mpa)	16.637	29.151

C. Spider and Side Gear

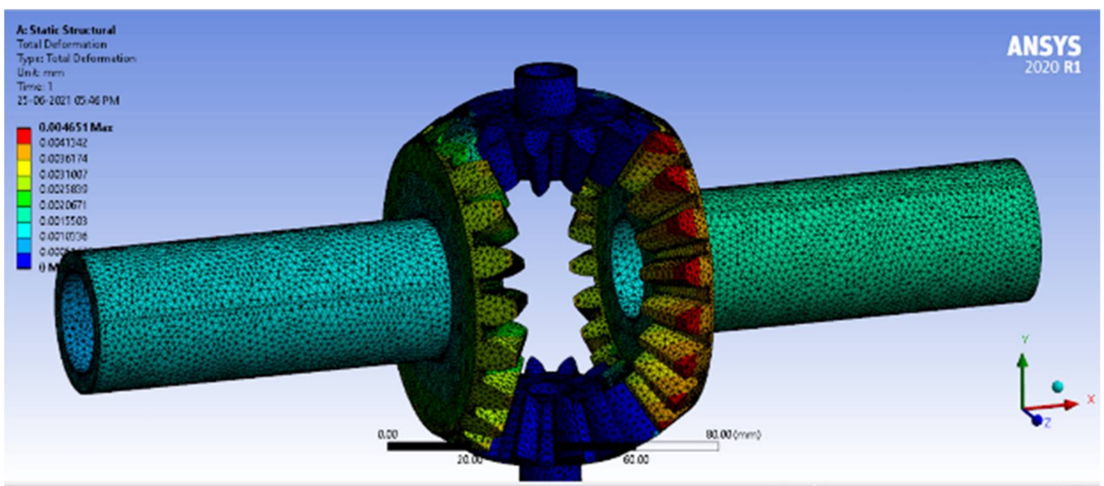

Fig.11 Total Deformation of Inner Bevel Set with 15Ni4Cr1

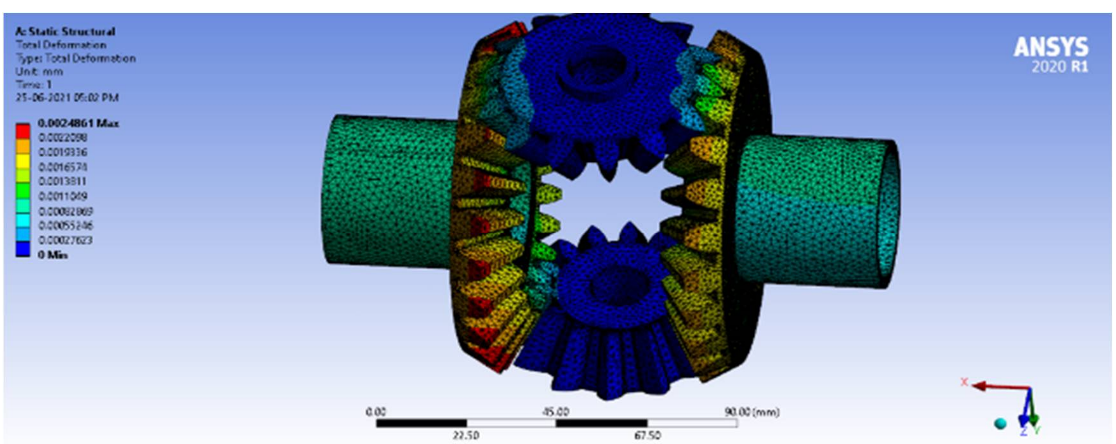

Fig.12 Total Deformation of Inner Bevel Set with 20MnC5

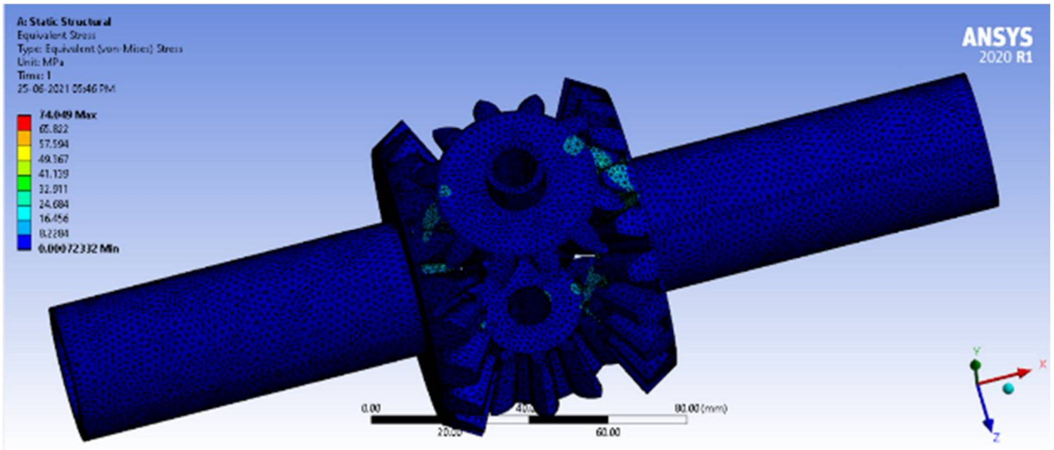

Fig.13 Equivalent Stress induced in Inner Bevel Set with 15Ni4Cr1

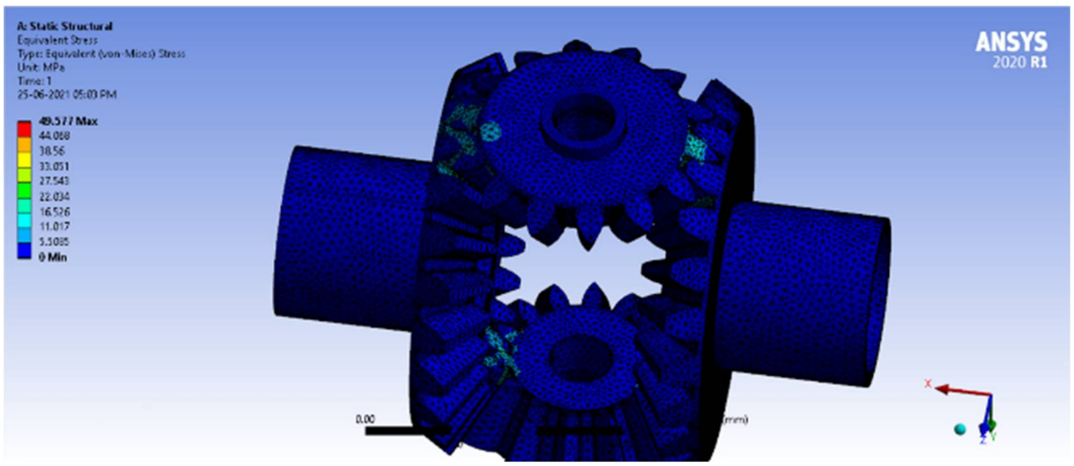

Fig.14 Equivalent Stress induced in Inner Bevel Set with 20MnC5

Type	15Ni4Cr1	20MnCr5
Deformation (mm)	0.004651	0.0024861
Von-Mises stress (Mpa)	74.049	49.577

V. RESULTS

From the above tables it is evident that the deformation in each case is of order 10^{-3} and 10^{-4} mm, which is quite negligible and hence can be ignored. The Equivalent stresses incurred is far less than the permissible stress. Thus, the Design is Safe.

VI. CONCLUSION

In this project, a differential gearbox is modelled in Solidworks 18 and structural analysis is done by using ANSYS Workbench 18.0. Presently used material for differential gearbox is cast iron. In this project, we have tried to calculate the factor of safety, deformation and equivalent stresses of two material namely- 15Ni4Cr1 and 20MnCr5. After comparing the results, it is clear that both the materials have almost similar deformation and Von-Mises Stress and are very much suitable for manufacturing a differential gear box, hence both the materials are quite safe. And by comparing the Theoretical Calculations between materials, magnesium alloy is more advantageous than other materials due to its less weight and high strength.

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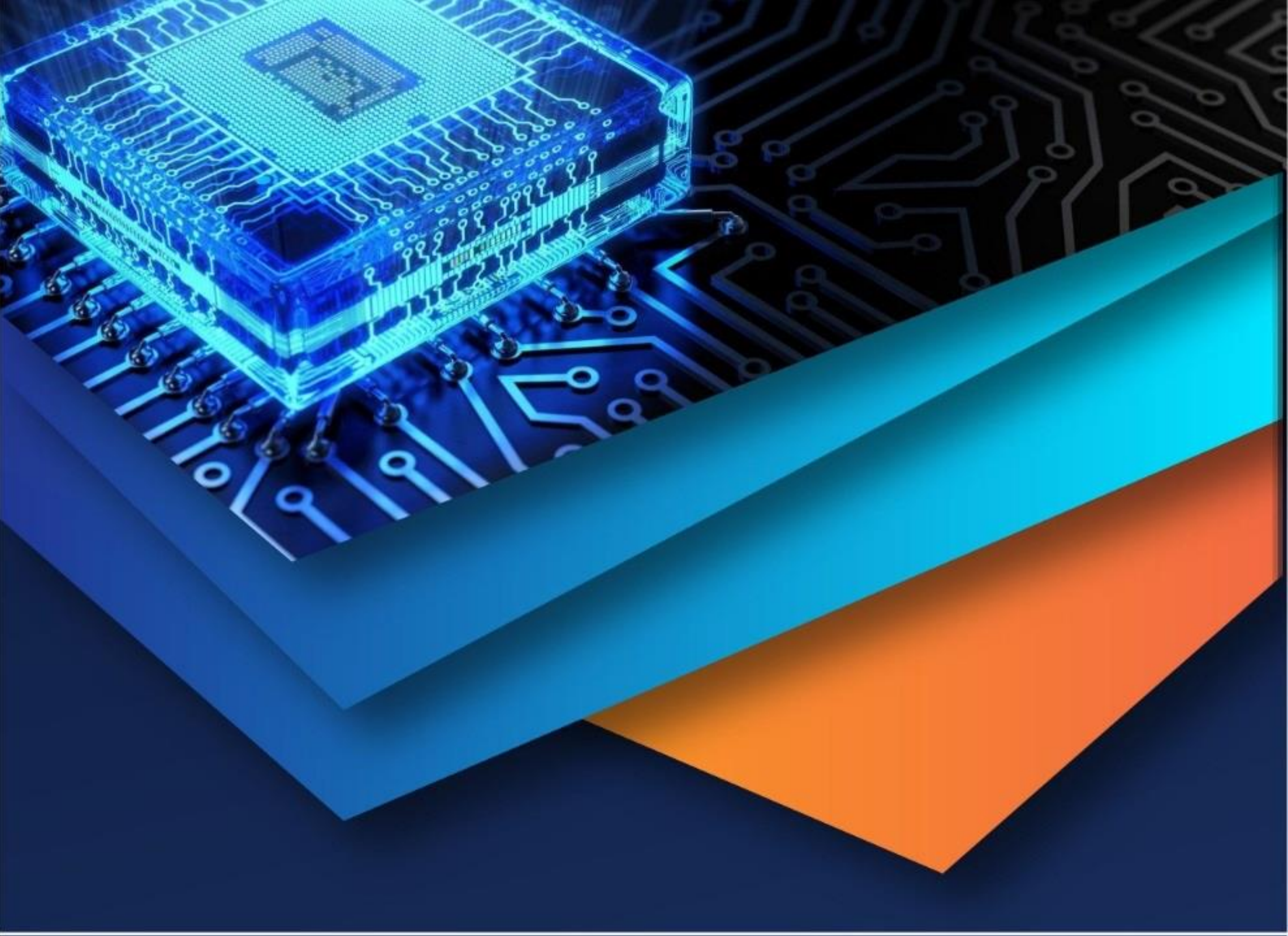

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