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Design and Development of Open Differential for Transmission System of Quad Bike

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Abstract – *A differential is an important torque transmitting device in most of the rear wheel drive vehicles. An ATV is a vehicle which is used to ride in off-road terrains; hence continuous application of traction is a vital factor which showcases its performative aspect. The paper describes designing and manufacturing an open differential for an off road ATV (Quad Bike) so that the vehicle maneuvers sharp corners without losing traction to the driving wheels.*

Key Words: **Open differential, traction difference, ANSYS, CREO, gear, pinion, centre pin.**

1. INTRODUCTION

A differential is a gear train with three shafts that has the property that the rotational speed of one shaft is the average of the speeds of the others, or a fixed multiple of that average. In automobiles, the differential allows the outer drive wheel to rotate faster than the inner drive wheel during a turn. This is necessary when the vehicle turns, making the wheel that is travelling around the outside of the turning curve roll farther and faster than the other. 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. The system described in the paper uses a final reduction of 1:1, meaning it is not driven by a pinion and a crown gear rather than a sprocket which is mechanically coupled to the roataing cage or differential casing of the differential.

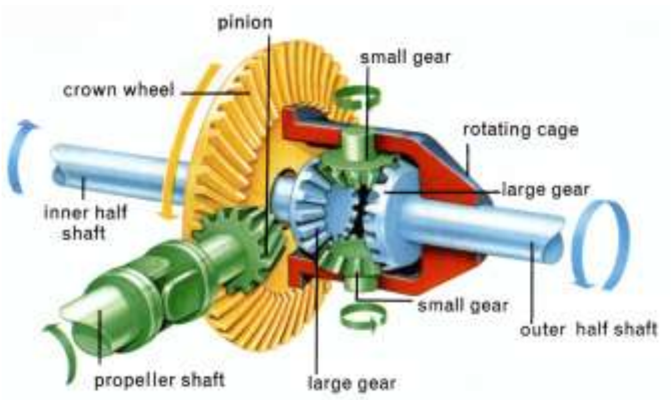

Fig -1: Conventional open differential

2. ANALYTICAL CALCULATIONS

2.1 Primary Calculations for Gear Reduction

Input Data:

Turning radius of vehicle: - 3m
Rear Track width: - 40”
Rear tire diameter: - 23”

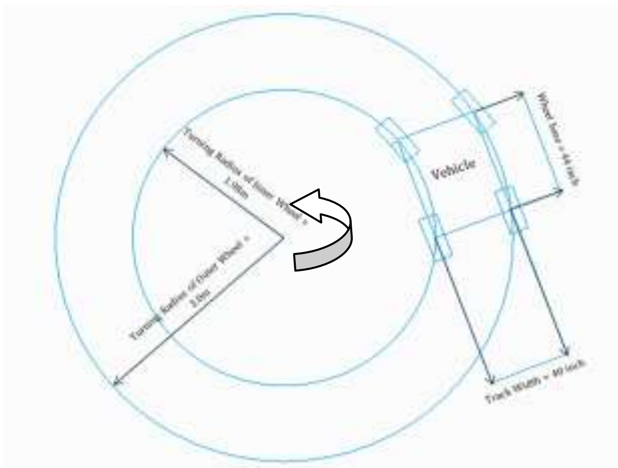

Fig -2: Vehicle moving in circular motion

A) Circumference of wheel
= 2*π*R
= 2*3.142*0.292
= 1.835m

B) Circumference of outer wheel trajectory
= π*D2
= 3.142*6
= 18.85m

C) Circumference of inner wheel trajectory
= π*D1
= 3.142*3.96
= 12.44m

D) Ratio (C/A)
= 12.44/1.835
= 6.79

Hence inner wheel needs 6.79 rotations to complete one revolution of the circular trajectory.

E) Ratio(B/A) = 18.85/1.835=10.27,

Hence outer wheel needs 10.27 rotations to complete one revolution of the circular trajectory.

F) Ratio(D/E) = 10.27/6.79 = 1.53,
Hence outer wheel rotates 1.53 times faster than inner wheel.
Reduction between spider gears and side gears
= 1.53
= **1.8** (considering available modules.)

2.2 Material Selection for Gears

Table-1: Material Selection Chart

Material Properties	20MnCr5 Steel	EN 8 Steel	EN19 Steel
Tensile yield Strength (Mpa)	750	433	470
Tensile ultimate Strength (Mpa)	1000	650	745
Density (kg/m3)	7850	7800	7800

Hence considering more tensile strength and more ultimate strength with optimum material cost, 20MnCr5 steel was finalized as gear material.

2.3 Gear Teeth Calculations

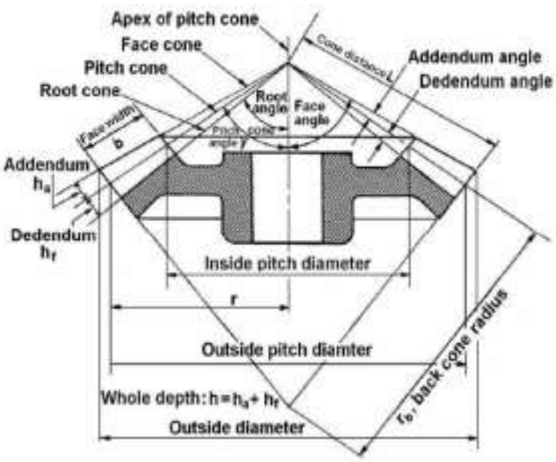

Fig-3: Bevel gear terminology

Assuming,
Pitch circle diameter of gear (Dg) = 72mm
No. of teeth on gear (Zg) = 18,
Pitch circle diameter of pinion (Dp) = 40mm
No. of teeth on pinion (Zp) = 10
Reduction = 1.8, Module = 4mm,
Addendum (ha)= 4mm, dedendum (hf)= 5mm (According to stub involute profile)

According to design data book,
Material used = 20MnCr5,
Syt = 750Mpa
Sut = 1000Mpa.
Case hardness = 42HRc (Material to be case hardened for pinion only.)
Pitch cone angles are,
$\gamma_p = \tan^{-1}$ (zp/zg)ASD
$\gamma_p = \tan^{-1}(10/18)$
=29.05°
γ_g =60.95°
Beam strength (σ_b) = sut/3
=1000/3
(σ_b) =333.33N/mm^2
Virtual number of teeth on pinion (Z`p)
= Zp/cos (γ_p)
=10/cos(29.05)
Z`p = 11.439
Similarly,
Virtual number of teeth on gear (Z`g) = 37.069
The pinion is designed considering bending,
Dp = m*Zp
= 4*10
Dp = 40mm
Similarly,
Dg = 72mm
Pitch cone distance (Ao) =$\sqrt{}$(dp/2)^2+(dg/2)^2
=$\sqrt{}$40/2)^+(72/2)^
Ao = 41.182mm

Lewis form factor (Y`p) = 0.55-(2.64/z`)
= 0.55-(2.64/11.439)
∴ (z` is 20 for stub involute profile)
Y`p = 0.3192
Beam strength (Fb) = σ_b *b*m*Y`p[1-(b/Ao)]
=333.33*13.727*4Z*0.3192[1-(13.727/41.182)]
Fb=3894.768N
Ratio factor (Q`) = 2*Z`g/ (Z`p+Z`g)
= 2*37.069/ (11.439+37.069)
Q`= 1.5283
Load stress factor (K) = 0.16[BHN/100]^2
= 0.16[400/100]^2
K = 2.56N/mm^2
Wear strength (Fw)= 0.75*dp*b*Q*K/[cos(γ_p)]
=0.75*40*13.727*1.5283*2.56/0.8742
Fw=1843.047N
As,Fb>Fw; Gear pair is designed for pitting.
V=π*dp*np/(60*1000)
=π*40*1080/(60*1000)
V=2.2619m/s
Tangential force (Ft) = P/V
= 12000/2.2619
Ft = 5305.165N
Velocity factor(Kv) = 6/(6+V)
= 6/(6+2.2619)
Kv = 0.72622
Effective load (Feff) = Ka*Km*Ft/Kv
= 1.25*1*5305.165/0.72622
Feff = 9131.40N
Hence, to avoid pitting failure we consider factor of safety of 1.75

∴Fw = Nf * Feff
= 1.75*9131.40
Fw = 15979.958N
Mean radius of bevel pinion (r_{mp})
=(dp/2)-[b*sin(γ_p)/2]
= (40/2)-[13.727*sin(29.05)/2]
r_{mp} = 16.6672 mm
Similarly,
Mean radius of bevel gear (r_{mg})=29.99mm
According to IS grade 6, pitch error is given by the equation,
e = 8.0+0.63[m+0.25*$\sqrt{rm}$]
For pinion, ep =8.0+0.63[m+0.25*$\sqrt{rmp}$]
=8.0+0.63[4+0.25*$\sqrt{2*16.6672}$
ep = 11.429µm
Similarly,
For gear,eg =11.739µm
Pitch error on meshing teeth © = ep + eg
=11.426+11.72
e=23.168*10^-3mm
Ftmax = Ka*Km*Ft
= 1.25*1.* 5305.165
Ftmax = 6631.456N
For steel pinion and steel gear deformation factor is given by,
C = 11000*e
= 11000*30*10^-3
C = 330N/mm
By using Buckingham's equation, dynamic loading is given by,
Fd = 21*V*(bC+Ftmax)/[21V+$\sqrt{bc}$+Ftmax]
=21*2.2619*(13.727*330+6631.456)/[21*2.2619+$\sqrt{}$13.727*330+6631.456]
Fd = 3461.79N
Feff = Ka*Km*Ft+Fd
= 1.25*1*5305.165+3247.176N
Feff = 10093.25N
Nf = Fw/Feff
=23997.64/12392.13
Nf= 1.583

∴ Hence, design is safe

2.4 Bearing Calculations

Inner Bearings :

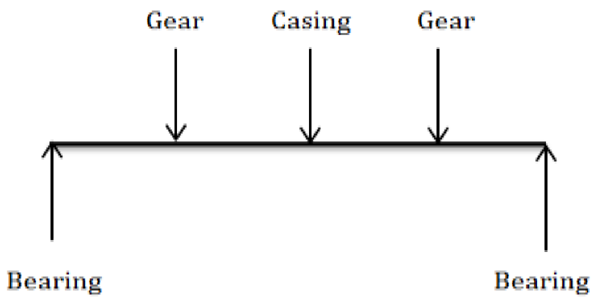

Fig-4: Free body diagram of differential

Forces on gear:-
F_t =p/v
=12*10^3/2.26
F_t =5305.16N
Radial forces on gear:-
F_{rp} = F_t*tan (ϕ)*cos(γ_p)
=5305.16*tan(20)*cos(29.05)
F_{rp} =1688N
F_{rg}=F_t*tan(ϕ)cos(γ_g)
=5305.16*tan(20*cos(60.95)
F_{rg}=937.602N
Axial forces on gear:-
F_{ag}=Ftm*tan(ϕ)*sin(γ_p)
=5305.16*tan(20)*sin(29.05)
F_{ap}=937.602N
F_{ag}=Ftm*tan(ϕ)*sin(γ_p)
=5305.16*tan(20)*sin(60.95)
F_{ag}=1688N
=5305.16*2.26
Resultant load on bearing:

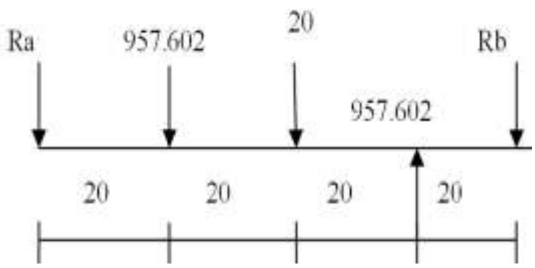

Fig -5: Vertical loading diagram

Moment about A:
(R_{bv}*80)+(957.602*20)+(20*40)-(957.602*60)=0
R_{bv} = 468.801N
Similarly moment about B gives;
R_{av} = -448.801N

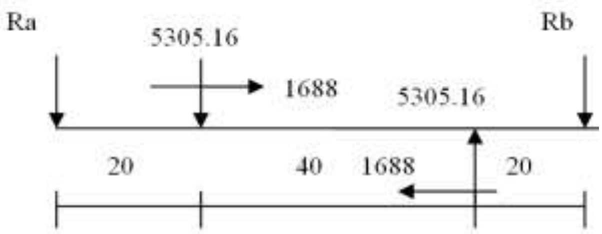

Fig -6: Horizontal loading diagram

Moment about B:
(R_{ah}*80)+(5305.16*60)-(1688*36)-(5305.16*20)+(1688*36)=0
R_{ah} = -2652.58N
R_{ah} = 103.782652.58N
Resultant load on bearing:
R_a= 2690.279 N
R_b=2693.68 KN

L_{h10}=12000
n = 600

$L_{10}= (L_{h10}*60*n)/10^{\wedge 6}$
$L_{10} = 432$ million revolutions
Effective dynamic load is given by:-
Pe=V*Fa*Ka
Pe=1*2693.68*1.2
Pe=3232.416
$(L_{10})^{\wedge(1/3)} = C/P_e$
$432^{\wedge(1/3)} = C/3232.416$
C = 24.43 kN

Dynamic load rating = 24.43kN
Hence from bearing catalogue, according to dynamic load rating bearing number is selected.
Bearing No = 6011(Ball Bearing)

Outer Mounting Bearings :-

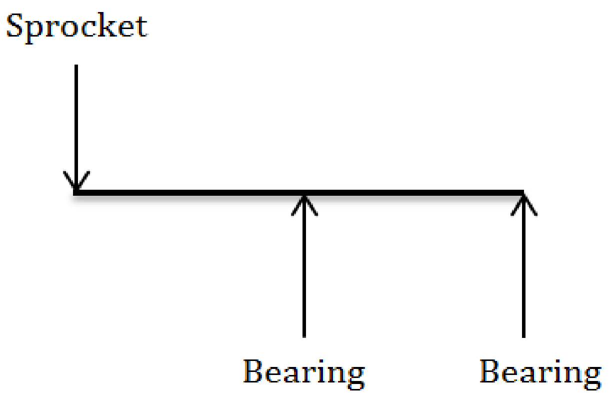

Fig -7: Load cases for outer bearings

Vertical forces:

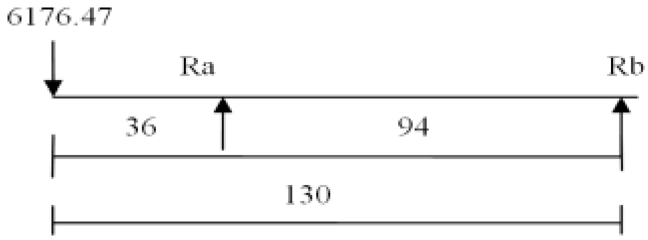

Fig -8: Vertical loading diagram

Moment about B:
$(-R_{av}*94)+(6176.47*130)=0$
$R_{av}=8541.926N$
Summation of forces:
$R_{av}+R_{bv}=6176.47$
$R_{bv}=-2365.456N$
Horizontal forces:

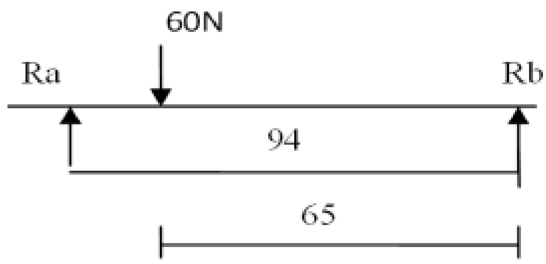

Fig -9: Horizontal loading diagram

Moment about B:
$(R_{ah}*94)-(60*65)=0$
$R_{ah}= 41.489N$
Summation of forces:
$R_{ah}+R_{hb}-60=0$
$R_{bh} = 18.51$ N.
Resultant load on bearing:
$R_a=\sqrt{(RAV^2+RAH^2)}$
$=\sqrt{(8541.926)^2+(41.489)^2}$
$R_a=8542.026N$
$R_b=\sqrt{(RBV^2+RBH^2)}$
$=\sqrt{(-2365.456)^2+(18.51)^2}$
$R_b= 2365.52N$
$L_{h10} = 12000$ hours
n=600
$L_{10}= (L_{h10}*60*n)/10^{\wedge 6}$
$L_{10} =(1200*60*600)/10^{\wedge 6}$
$L_{10}= 432$ million revolutions
Effective dynamic load is given by:-
Pe= V*Fa*Ka
Pe= 1*8542.026*1
Pe= 8542.026
$(L_{10})^{\wedge(1/3)}= C/P_e$
$432^{\wedge(1/3)}= C/9542.026$

C = 64.57 kN
Dynamic load rating = 64.57kN
Hence from bearing catalogue, according to dynamic load rating bearing number is selected.
Bearing No = 6017(Ball Bearing)

3. CAD MODELLING

3. 1 Gear and Pinion

After getting the output analytical data, initially gear pair was designed. An open differential generally consists of two or more gear pairs for sharing the load transmission. Considering the torque value it was decided to use two gear pairs. It was necessary to create a means to transfer the torque of rotating side gears to the shafts and ultimately to the driving wheels, hence a provision was made to accommodate shaft sleeves in the side gear itself. The shaft sleeve profile was first traced with the help of Co-ordinate Measuring Machine (CMM), and then the drawing was used in the CAD model. The gear length was decided according to the plunge length of the shaft in the gear profile, such that the shaft should not be thrown out of the gear profile in the maximum bump and droop condition of the wheel travel. The gears were then mounted on the bearings. The pinion was very small compared to gear and it was needed to revolve without support of bearings. Hence pinions were made with a hole to accommodate the centre pin with slide fit tolerance.

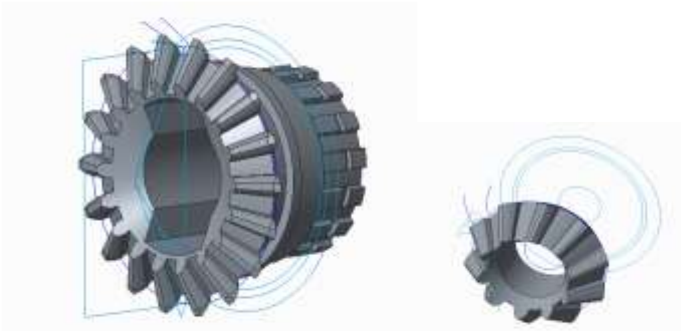

Fig-10: Gear and Pinion

3. 2 Centre Pin

The centre pin was designed considering the packaging of the gear pairs and ease of serviceability. The material from the central portion had been removed optimize to the weight. The material finalized for this part was 20MnCr5 steel. A small hole had been provided at one of the ends in order to couple the centre pin with the differential casing via a nut and a bolt such that centre pin won't slide of through the assembly.

Fig-11: Centre pin

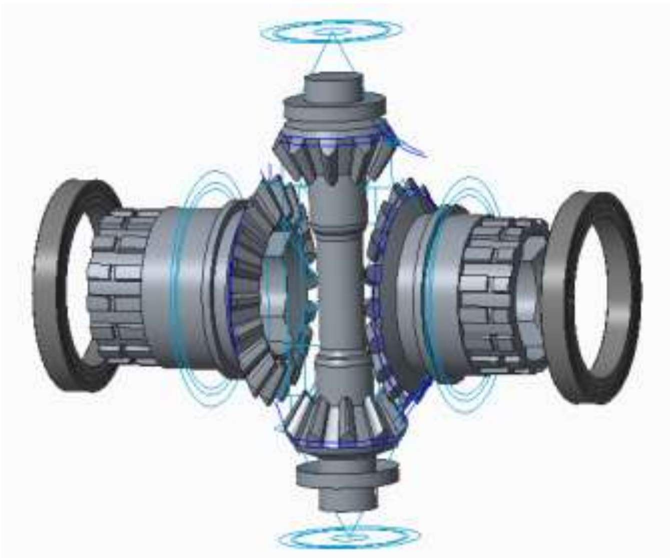

Fig-12: Final gear and pinion assembly

3. 3 Differential Casing

It was decided to design the casing in two parts for the convenience of assembly and ease of servicing. The overall designed was based on the packaging of the inner assembly. The side gears were designed to be mounted on the inner race of the inner bearings and the outer race of these bearings had been press fitted with tolerance of + 0.1mm in the inner side of the differential casing. The length of the casing was decided according to the length of side gears and the location of sprocket hub. Splines were provided on one of the casing to fix the sprocket hub on to it, ultimately mounting sprocket to the hub. Two casings were coupled to each other via 10 pairs of nut and bolts (M6 size). The casing was then mounted in a pair of bearings in differential mount plates.

Fig-13: Differential casing

3. 4 Differential Mount plate

Differential plate was designed considering the packaging constraints of the sprocket and the mounting points accessible on the chassis to mount the differential. The bearing's outer race was press fitted inside differential plates whereas the inner races were slide fitted on the differential casing. A small hole was provided on one of the differential plate's end in order to mount the bolt that would be passing through both the differential mounting plates holding them firmly in one plane.

Fig-14: Differential mount plate

Fig-15: Final Differential Assembly

3. 5 Material Selection for Differential Casing and Differential Mount Plates

Table-2: Material Selection Chart

Material Properties	EN19 Steel	Aluminium 6061 T6	Aluminium 7075 T6
Tensile yield Strength (MPa)	470	310	415
Tensile ultimate Strength (MPa)	745	400	460
Density (kg/m3)	7800	2700	2780
Cost / Kg	130	360	700

The casing and mount plates needed to be less in weight with adequate strength and optimum cost. The softer material like aaluminium made it possible to reduce machining cost and weight. Hence, Aluminium 6061 T6 was selected for these components.

4. Finite Element Analysis

Static structural simulation has been done for all the components. The material properties were not automatically selected by the software. Hence, the material properties were manually given to the software. After getting the properties of specific material the CAD model was imported to ANSYS 16.0 workbench. The mesh was created and analytical forces were applied on the components. The required solutions were selected such as von mises stresses, total deformation and the solutions are obtained.

4.1 Gear of Material 20MnCr5 Steel

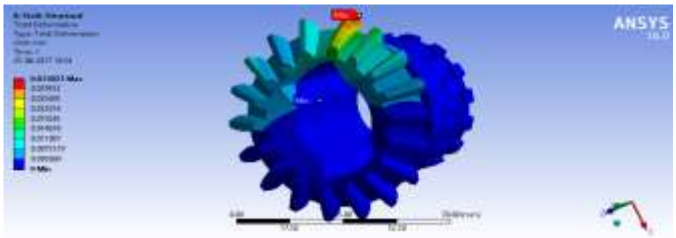

Fig-16: Total deformation in gear

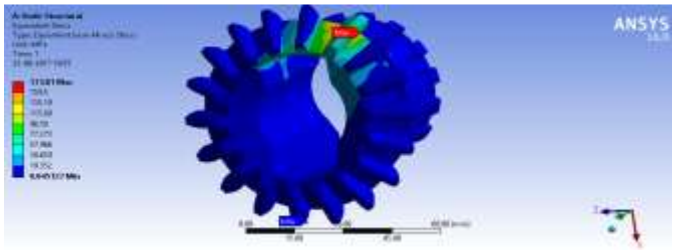

Fig-17: Von-mises stress in gear

4.2 Pinion of Material 20MnCr5 Steel

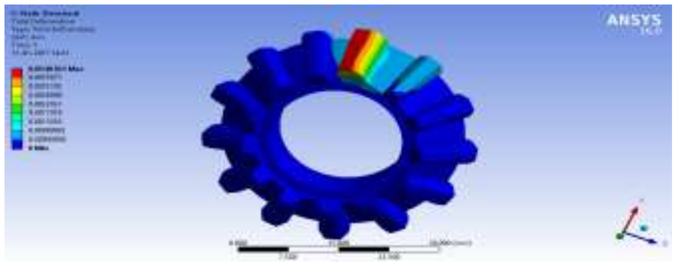

Fig-18: Total deformation in pinion

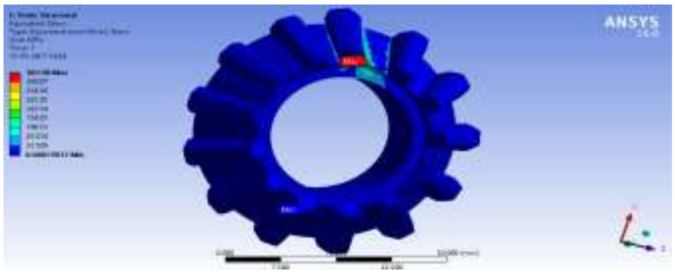

Fig-19: Von-mises stress in pinion

4.3 Differential Casing of Material 6061 T6

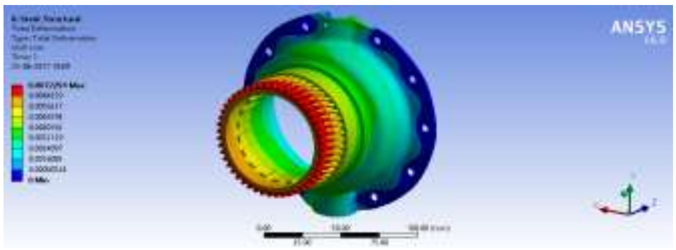

Fig-20: Total deformation in differential casing

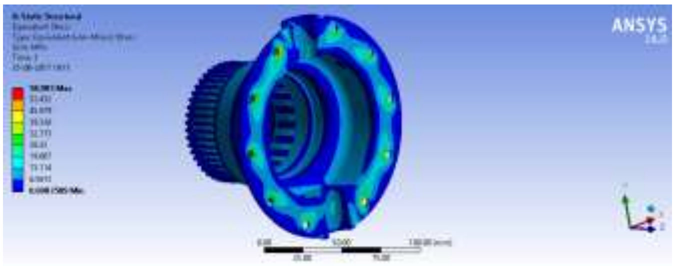

Fig-21: Von-mises in differential casing

4.4 Centre Pin of Material 20Mncr5 Steel

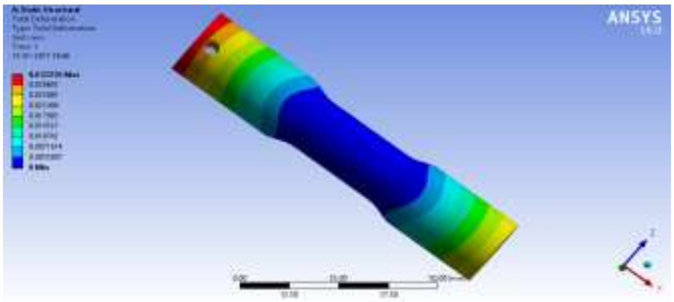

Fig-22: Total deformation in centre pin

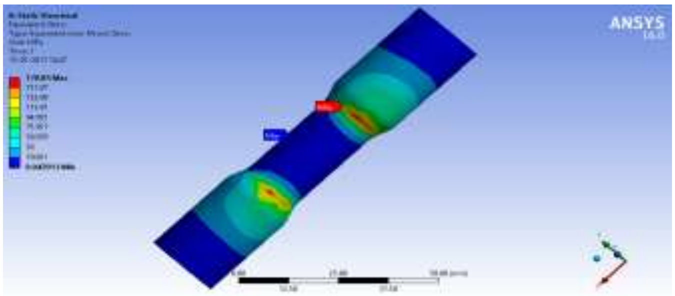

Fig-23: Von-mises stress in centre pin

4.5 Differential Mount Plates of Material 6061 T6

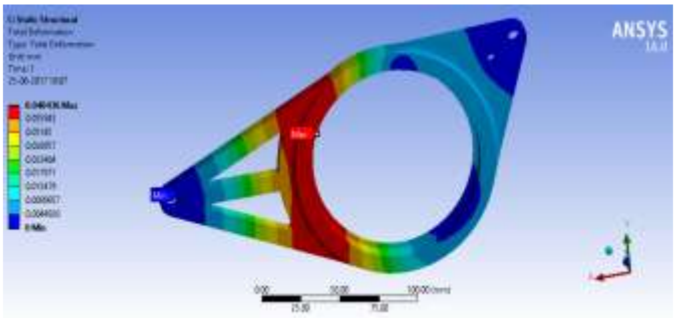

Fig-24: Total deformation in differential mount plate

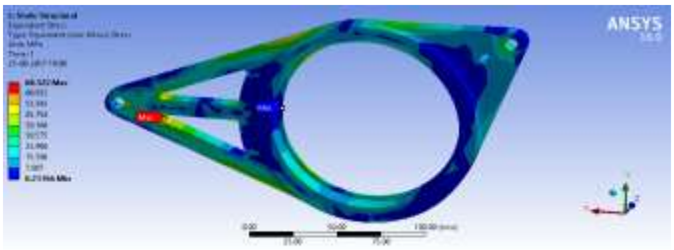

Fig-25: Von-mises stress in differential mount plate

5. Experimental Setup and Testing

The differential gears were machined by the process of gear hobbing. Pinion was case hardened(42HRc) as pinion is always weaker in bending than gear when same material is used. The gears were than wire cut according to the shaft profile in order to accommodate the shafts for transmitting the torque from differential to the wheels. The gears and pinions were than blackodized for corrosion resistance. The centre pin was turned on lathe and provided with a slide fit tolerance(negative 0.1mm) for the pinions to rotate about it.

The pinion material was harder than differential casing, hence shims were placed in between the pinions and the differential casing to avoid wear due to metal to metal contact. The differential casing and differential mount plates were machined using computer controlled Vertical Machining Centre(VMC). The overall weight of the complete assembly was found to be 3.6kg when measured practically on weighing machine. The complete assembly was than assembled on the quad vehicle.

The differential was tested for 30 days on offroad terrain including sharp corners, booby traps, sand gravel, minor and major bumps and depressions with 6 hours testing per day. The vehicle was let to take sharp corners and the tread pattern of tyre, wear of gear teeth were visually inspected.

Fig-26: Individual components of differential

Fig -27: Assembled differential on the quad bike.

Fig -28: Quad taking sharp corners during Endurance Race.

6. CONCLUSION

The differential assembly, after implementation in quad vehicle, satisfies the design requirements. It proves to be durable on rough terrain and critical operating conditions. No traction difference problem was observed and vehicle could maneuver sharp turns without losing traction. Thus, we can conclude that initial objective of differential for transmission system of quad vehicle is satisfied and serves as ground for further research on said system.

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