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Design and Analysis of 2-Stage Gearbox for ATV Applications

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Abstract: The main objective of this project is to design and develop a 2-Stage- Reduction gearbox for all terrain vehicles application. Gearbox is a mechanical unit consisting of series of gears within a housing (casing) or a simple gear train. It is the main component of the powertrain system as it provides speed and torque conversions from the Engine or CVT to the wheels. The gears inside of gearbox can be any one of a number of different types of the gear that are available in the market i.e., from spur gears to worm gears and planetary gears as well. In this report the gearbox is made from a helical gear train. This report focuses on selection of overall reduction ratio of the gearbox, material selection, selection of gear dimensions, selection of shaft dimension, design of key and bearing selection from SKF bearing catalogue. The bending and wear strengths are calculated to determine factor of safety and later, Finite element analysis is performed to validate the results.
Keywords: Geartrain, Shaft Design, SKF Bearings, Solidworks, FEA.

I. INTRODUCTION

An All-Terrain-Vehicle usually has to undergo different uneven and irregular road conditions. In order to perform well on these road surfaces the powertrain system of the vehicle must produce sufficient torque to conquer the steep slopes and droops put forward by it. Thus, the role of a gearbox is of prime importance. The main source of power in this powertrain subsystem is a Four stroke, Single cylinder, 305cc, 10Hp Briggs and Stratton Engine which can produce a maximum torque of 19.7Nm @2800 rpm. The torque provided by the engine is extremely low. Therefore, the engine is coupled with a CVT which is further coupled with a two-stage reduction gearbox. Such arrangement is capable of producing a torque sufficient enough to not only move the vehicle but also to overcome the rough terrain. The CVT used in this system is JITech CVT which has a low ratio of 3.9 and a high ratio of 0.5. The Design and development of the gearbox is based on various factors such as Gradeability, Grade Resistance, Rolling resistance. Taking into considerations all of these parameters, the required torque on the wheel is calculated and thus the overall reduction ratio of the gearbox is calculated. The maximum velocity is restricted to 50 kmph in this design and this plays a vital role in selecting the overall gearbox reduction.

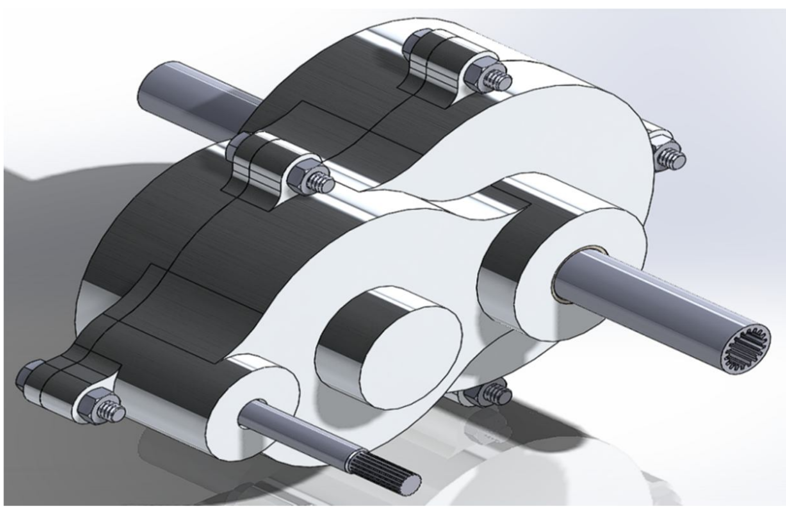

Figure 1. Reduction gearbox

II. DESIGN CALCULATIONS

A. Design Considerations

1) Vehicle Specifications

Weight Distribution	60:40 (Rear: Front)
Gross Vehicle Weight (GVW)	180+60=240 Kg
Tire rolling radius	0.283337 m
Drag Coefficient (Cd)	0.44
Frontal Area	$0.78m^2$
Coefficient of friction (μ)	1.0
Coefficient of rolling resistance (f_r)	0.014
CVT low ratio and high ratio	3.9:1 and 0.5:1

2) Transmission Specifications

Engine Specifications (Briggs and Stratton)		CVT Specifications (JITech)	
Maximum Torque	19.7 Nm @ 2800 rpm	Low Ratio	3.9:1
Maximum Power	10 HP @ 3600 rpm	High Ratio	0.5:1
		Centre to Centre distance	214 mm

3) Design Targets

a) To obtain an output Top speed of 50 kmph.
b) To obtain Maximum torque and acceleration

The Total Tractive effort (TTE) can be calculated as shown: -
TTE = RR+AR+GR
The vehicle must be able to produce enough torque in order to move the vehicle against the road loads. These loads mainly comprise of Rolling resistance, Aerodynamic resistance and Grade Resistance. They can be calculated as shown:
RR= GVW*f_r*cos(θ)
AR= $0.5 * \rho\ (air) * v^2 * A * Cd$
GR=GVW*sin(θ)

B. Torque at Wheels

Gradeability: After various iteration Gradeability of 58.55% was achieved. This implies that inclination was found to be 30.35°. Thus, all of the driving resistance forces were calculated in order to find maximum torque at wheel when vehicle accelerates on a surface with inclination of 30.35°.
RR=240*9.81*0.14*cos (30.35) =28.44 N
Aerodynamic force was considered as 0 in this case.
GR= GVW*sin (30.35) =1189.42 N
Thus, TTR=RR+AR+GR = 1217.871 N.
The total amount of force which can be transferred by the wheels without slipping can be found out by TTE_{max} = GVW*μ*sin(θ)*0.60.
TTE_{max} = 1218.83 N.
Gradeability is selected such that the TTE_{max} >=TTE.
Torque at wheels (T_w)= TTE*0.283337 = 345 Nm.

C. Calculation of Gearbox Reduction

Torque at wheels (T_w) 345 = $19.7 * 3.9 * 0.75 * GR * 0.99 * 0.99$ => GR=6.1
The top-speed of the vehicle is limited to 50 Kmph and the max engine rpm is 2800. Therefore, max velocity of vehicle can be calculated as: 13.89(50kmph) = $\frac{3800}{0.5} * 0.75 * \frac{1}{GR} * \frac{2\pi}{60} * 0.283337$ => GR=12.17

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In order to get sufficient torque at wheel optimum Gear Ratio of 10.5 is selected. Splitting of the final GR is shown below:

STAGE 1	3.5
STAGE 2	3

Maximum theoretical velocity:
$\frac{3800}{0.5} * 0.75 * \frac{1}{10.5} * \frac{2\pi}{60} * 0.283337$ = 16.10 m/s = 57.96 kmph

Torque at wheels considering 10.5 reduction ratio:
Tw= $19.7 * 3.9 * 0.75 * 10.5 * 0.99 * 0.99$ = 592.996 Nm
RR=2354*0.014 = 32.956 Nm
Engine rpm = 2800
Wheel rpm = $\frac{2800}{3.9} * 0.75 * \frac{1}{10.5} * 0.99 * 0.99$ = 50.26
Wheel velocity =$2\pi * \frac{50.26}{60} * 0.283337$ = 1.49309
AR= $0.5 * 1.122 * 1.49309^2 * 0.78 * 1.08$ = 1.05Nm
Net Force on wheel (F) = $\frac{592.996}{0.283337} - (1.05 + 32.956)$ = 2058.893
Acceleration = $F * \frac{9.81}{2354}$ = 8.58 m/s^2

D. Gear Dimension Selection

The Material used for gear is case hardened alloy steel 15Ni4Cr1: Sut=1500N/mm^2, Syt=850 N/mm^2 BHN: 650 and IS GRADE 5.

1) *Stage 1:* The reduction for stage 1 is 3.5. Helical Gearset is to be used. After performing sufficient iterations, the following gear dimension were finalized has it has sufficient factor of safety.

Lead angle	15°
Normal Module (Mn)	2 mm
No. of teeth of Pinion (Zp)	18
No. of teeth is Gear (Zg)	63

PCDp =$\frac{Zp*Mn}{cos\varphi}$ =$\frac{18*2}{cos15°}$= 37.27mm and PCDg= $\frac{Zp*Mn}{cos\varphi}$ = $\frac{63*2}{cos15°}$ = 130.44 mm

Centre to Centre distance = $\frac{Mn*(Zp+Zg)}{2*\cos(\varphi)}$ = $\frac{2*(18+63)}{2*\cos(15°)}$ = 83.85 => C-C~85mm

Virtual No. of teeth for pinion (Z'p) = $\frac{Zp}{\cos(\varphi)^3}$ = $\frac{18}{(\cos 15°)^3}$ = 19.97

Virtual No. of teeth for gear (Z'p) = $\frac{Zp}{\cos(\varphi)^3}$ = $\frac{63}{(\cos(15°)}$ = 69.90

Face width (b) ≥ $\frac{\pi*Mn}{\sin(\varphi)}$ => b= $\frac{\pi*2}{\sin(15°)}$=24.27 ~ $25mm$

Input torque to Stage 1 (Mt) = 19.7*3.9*0.75 = 57.6225 Nm = 57622.5 Nmm.

Pt = $\frac{2*Mt}{PCDp}$ = $\frac{2*57622.5}{37.27}$ = 3092.17 N

Pr = Pt * $\frac{\tan(\alpha n)}{\cos(\varphi)}$ = 3092.17 * $\frac{\tan(20°)}{\cos(15°)}$= 1165.16 N

Pa = Pt*tan(φ) = 3092.17 * tan (15°) = 828.54 N

The beam Strength is given by the following equation:

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Sb=Mn*b*σb*Y [Y is Lewis form factor for Z' and σb is permissible bending stress]

=>Y=$\pi * \left(0.154 - \frac{0.912}{Z'p}\right) = \pi * \left(0.154 - \frac{0.912}{19.97}\right)$ = 0.340 and σb = Sut/3 = 500Nmm^2.

Thus, Sb is calculated as:
Sb=$2 * 25 * 500 * 0.340$= 8500 N

The Wear Strength is given by the following equation:
Sw=$\frac{b*Q*PCDp*K}{\cos(\varphi)^2}$

Q=$\frac{2*Z'g}{Z'g+Z'p} = \frac{2*69.90}{69.90+19.97}$ = 1.556 [Ratio factor]

K=$0.16 * \left(\frac{BHN}{100}\right)^2 = 0.16 * (6.5)^2$ = 6.76 [Load factor]

Sw=$\frac{b*Q*PCDp*K}{\cos(\varphi)^2} = \frac{25*1.556*37.27*6.76}{\cos(15)^2}$ = 10504.32 N

Now, to calculate effective force on the tooth,
Max torque is 19.7Nm @2800rpm => the rpm at Stage 1 is $\frac{2800}{3.9} * 0.75 = 538.46$, the Cv is calculated as:
Cv=$\frac{5.6}{5.6+\sqrt{v}}$ [v is pitch line velocity] and Cs=1.25

v=$\frac{\pi*PCDp*rpm}{60000} = \frac{\pi*37.27*538.46}{60000}$ = 1.05m/s => Cv = $\frac{5.6}{5.6+\sqrt{1.05}} = 0.845$
C (deformation factor) =11400 N/mm^2

e=ep+eg
For IS Grade 5, the error is e=5+0.4ϕ and ϕ=Mn+0.25$\sqrt{PCD}$
ep= 5+0.4(2+0.25$\sqrt{37.27}$) = 6.41μm
eg= 5+0.4(2+0.25$\sqrt{130.44}$) = 6.94μm
e=6.41+6.94=13.35μm = 0.01335mm

Therefore, the dynamic load can be calculated by using the Buckingham's Load equation:
Pd=$\frac{21*v*(C*e*b*(\cos\varphi)^2+Pt)*\cos\varphi}{21*v+\sqrt{C*e*b*(\cos\varphi)^2+Pt}}$ = 1366.18 N
Thus, Peff ca be calculated as:
Peff=$Cs * Pt + Pd$ = 1.25*3092.17+1366.18 = 5231.4 N
Hence, the factor of safety for bending and wear ca be calculated as:
fos(beam)=Sb/Peff = 8500/5231.4 = **1.62**
fos(wear)=Sw/Peff = 10504.32/5231.4 = **2.007**

2) *Stage 2:* The reduction for stage 1 is 3. Helical Gearset is to be used. After performing sufficient iterations, the following gear dimension were finalized has it has sufficient factor of safety.

Lead angle	15°
Normal Module (Mn)	3 mm
No. of teeth of Pinion (Zp)	17
No. of teeth is Gear (Zg)	51

PCDp $=\frac{Zp*Mn}{cos\varphi}=\frac{17*3}{cos15°}=$ 52.8mm and PCDg= $\frac{Zp*Mn}{cos\varphi}=\frac{51*3}{cos15°}=$ 158.4 mm

Centre to Centre distance = $\frac{Mn*(Zp+Zg)}{2*\cos(\varphi)}=\frac{2*(17+51)}{2*\cos(15°)}=$ 105.6 => C-C~106.5mm

Virtual No. of teeth for pinion (Z'p) = $\frac{Zp}{\cos(\varphi)^3}=\frac{17}{(\cos 15°)^3}=$ 18.86

Virtual No. of teeth for gear (Z'p) = $\frac{Zp}{\cos(\varphi)^3}=\frac{51}{(\cos 15°)^3}=$ 56.58

Face width (b) ≥ $\frac{\pi*Mn}{\sin(\varphi)}$ => b= $\frac{\pi*3}{\sin(15°)}$=36.41 ~ $37mm$

Input torque to Stage 2 (Mt) = 19.7*3.9*0.75*3.5*0.99 = 199.662 Nm = 199662 Nmm.

Pt = $\frac{2*Mt}{PCDp}=\frac{2*199662}{52.8}=7563$ N

Pr = Pt * $\frac{\tan(\alpha n)}{\cos(\varphi)}$ = 7563 * $\frac{\tan(20°)}{\cos(15°)}$= 2849.82 N

Pa = Pt*tan(φ) = 7563 * tan (15°) = 2026.5 N

The beam Strength is given by the following equation:
Sb=Mn*b*σb*Y [Y is Lewis form factor for Z' and σb is permissible bending stress]

=>Y= $\pi*\left(0.154-\frac{0.912}{Z'p}\right)=\pi*\left(0.154-\frac{0.912}{18.86}\right)=$ 0.33192 and σb = Sut/3 = 500Nmm^2.

Thus, Sb is calculated as:
Sb= $3*37*500*0.33192$= 18421.56 N
The Wear Strength is given by the following equation:
Sw= $\frac{b*Q*PCDp*K}{\cos(\varphi)^2}$

Q= $\frac{2*Z'g}{Z'g+Z'p}=\frac{2*56.58}{56.58+18.86}=$ 1.5 [Ratio factor]

K= $0.16*\left(\frac{BHN}{100}\right)^2=0.16*(6.5)^2=$ 6.76 [Load factor]

Sw= $\frac{b*Q*PCDp*K}{\cos(\varphi)^2}=\frac{37*1.5*52.8*6.76}{\cos(15)^2}=$ 21231.76 N

Now, to calculate effective force on the tooth,
Max torque is 19.7Nm @2800rpm => the rpm at Stage 2 is $\frac{2800}{3.9}*0.75*\frac{1}{3.5}*0.99=152.3$, the Cv is calculated as:

Cv= $\frac{5.6}{5.6+\sqrt{v}}$ [v is pitch line velocity] and Cs=1.25

v= $\frac{\pi*PCDp*rpm}{60000}=\frac{\pi*52.8*152.3}{60000}=$ 0.42104m/s => Cv = $\frac{5.6}{5.6+\sqrt{0.42104}}=0.89616$

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C (deformation factor) =11400 N/mm^2
e=ep+eg
For IS Grade 5, the error is e=5+0.4ϕ and ϕ=Mn+0.25$\sqrt{PCD}$
ep= 5+0.4(3+0.25$\sqrt{52.8}$) = 6.92μm
eg= 5+0.4(3+0.25$\sqrt{158.4}$) = 7.46μm
e=6.92+7.46=14.38μm = 0.01438mm

Therefore, the dynamic load can be calculated by using the Buckingham's Load equation:
Pd=$\frac{21*v*(C*e*b*(cos\varphi)^2+Pt)*cos\varphi}{21*v+\sqrt{C*e*b*(cos\varphi)^2+Pt}}$ = 911.94 N

Thus, Peff ca be calculated as:
Peff=$Cs*Pt+Pd$ = 1.25*7563+911.94 = 10365.7 N
Hence, the factor of safety for bending and wear can be calculated as:
fos(beam)=Sb/Peff = 18421.56/10365.7 = **1.77**
fos(wear)=Sw/Peff = 21231.76/10365.7 = **2.048**

Summary of Factor of safety calculated above:

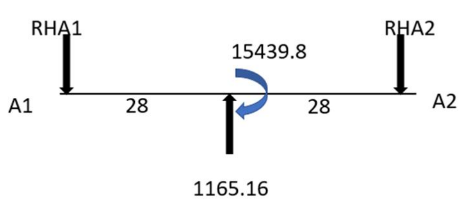
	Fos (beam)	Fos(wear)
STAGE 1	**1.62**	**2.007**
STAGE 2	**1.77**	**2.048**

E. Shaft Design

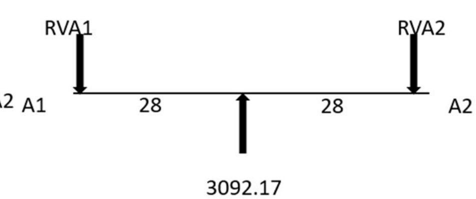
20MnCr5 is selected as the material for shaft. (Syt=1100N/mm^2 and Sut=1150N/mm^2)
The shafts are designed based on ASME code.
τ max=0.18*Sut, since keyways are present,
τ max=0.75*0.18*1150=155.25 N/mm2 ……(I)

Kb=1.5 and Kt=1 [Since the load applied is gradual] and thus the τ max is given by,
τ max=$\frac{16}{\pi*d^3}*\sqrt{(KbMb)^2+(KtMt)^2}$ ………. (II)

1) Loading Condition of Input Shaft A1A2 (Mt=57622.5 Nmm)



Horizontal Loading Diagram



Vertical Loading Diagram

RHA1 and RHA2 can be solved by $\sum MA1$=0 and equilibrium equation.

RVA1 and RVA2 can be solved by $\sum MA1$=0 and equilibrium equation.

RHA1= 858.29 N and RHA2=306.87 N

RVA1=1546.1 N and RVA2=1546.1 N

Now, Bending Moment Diagram is Draw for both horizontal and vertical as shown below:

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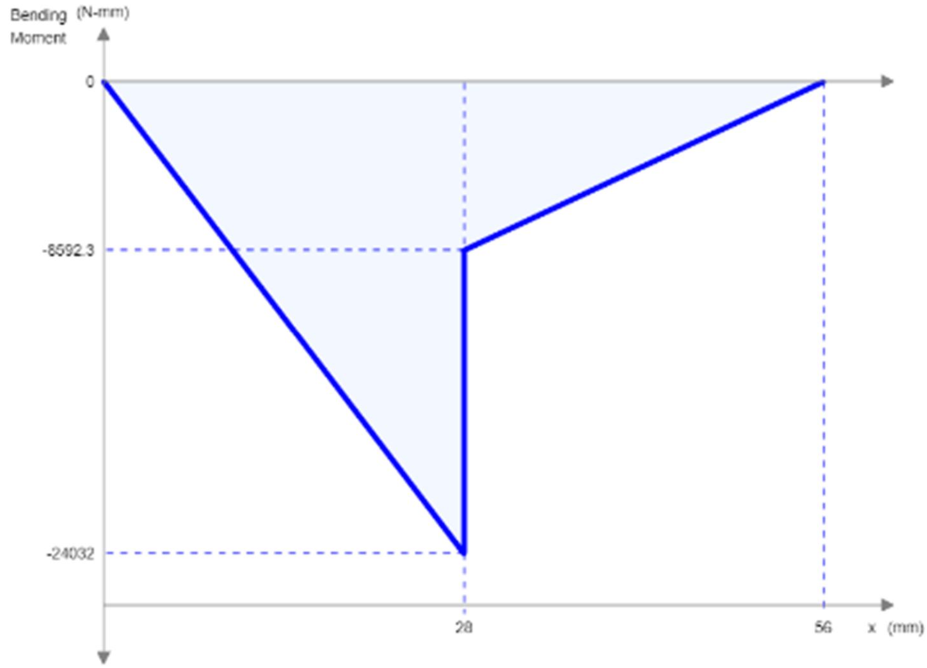
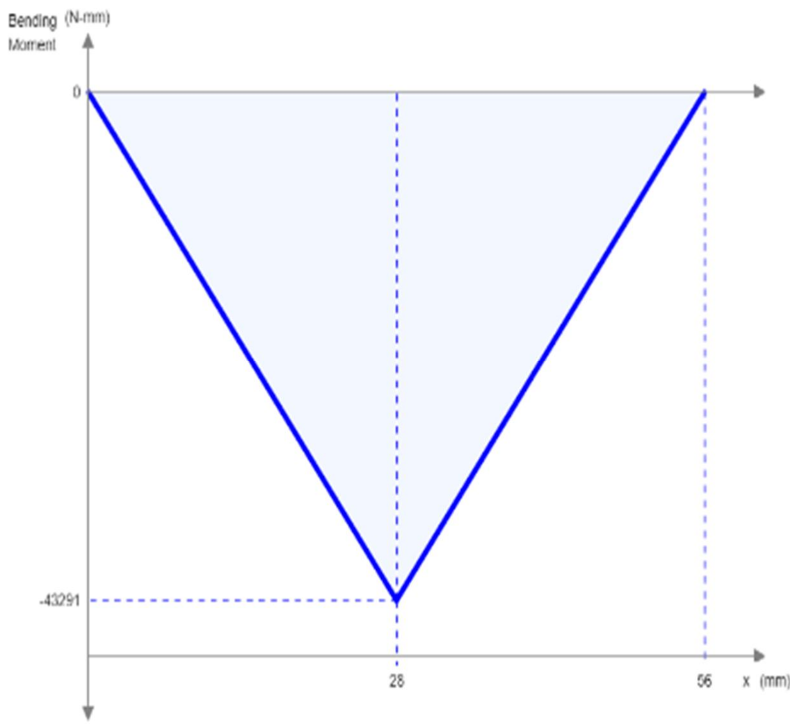


BMD for Horizontal Loading



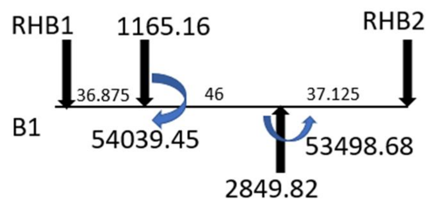
BMD for vertical Loading

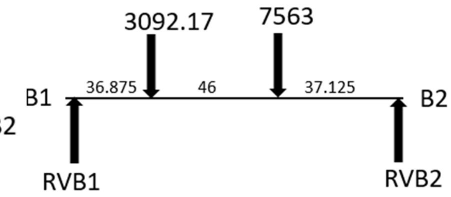
Maximum Bending moment in Horizontal and Vertical diagram is -24032 Nmm and -43291Nmm respectively.

Therefore, Mb=$\sqrt{24032^2 + 43291^2}$= 49514.12N

From (II) 155.25=$\frac{16}{\pi * d^3} * \sqrt{(1.5 * 49514.12)^2 + (57622.5)^2}$ => d=14.55mm~ $\mathbf{15}\boldsymbol{mm}$

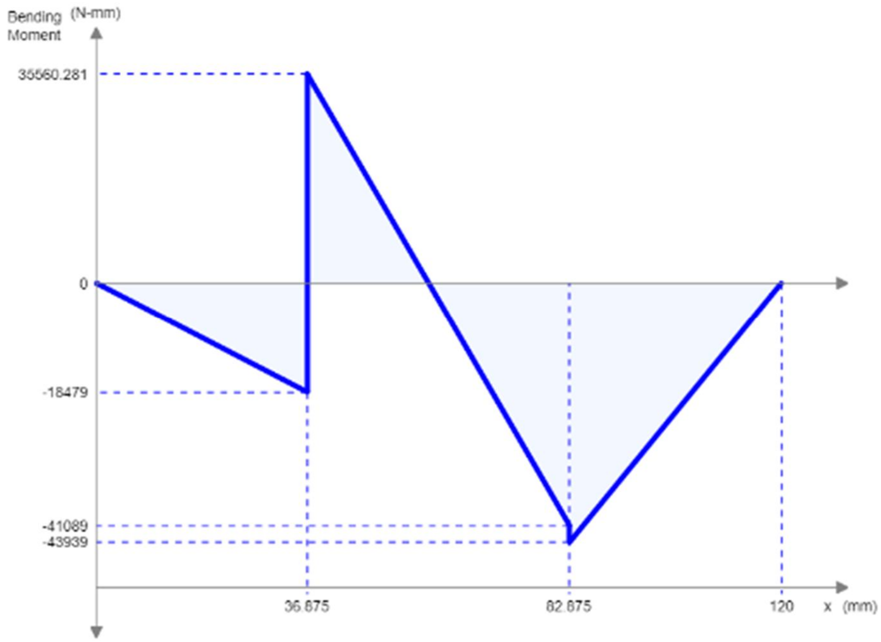
2) *Loading Condition of intermediate Shaft B1B2 (Mt=199662 Nmm)*



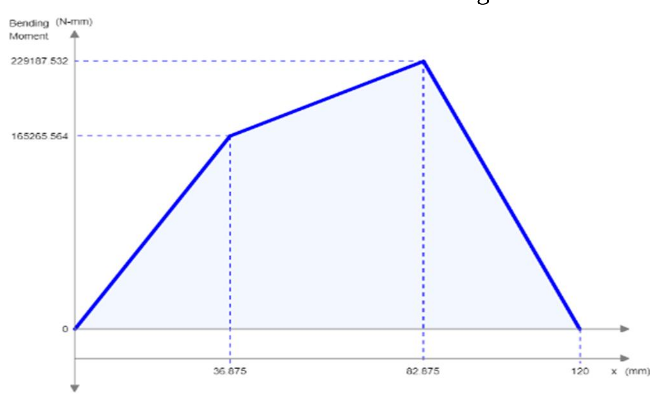


Horizontal Loading Diagram	Vertical Loading Diagram
RHB1 and RHB2 can be solved by $\sum MB1$=0 and equilibrium equation.	RVB1 and RVB2 can be solved by $\sum MB1$=0 and equilibrium equation.
RHB1= 501.13 N and RHB2=1183.5 N	RVB1=4481.778 N and RVB2=6173.402 N

Now, Bending Moment Diagram is Draw for both horizontal and vertical as shown below:



BMD for Horizontal Loading



BMD for vertical Loading

Maximum Bending moment in Horizontal and Vertical diagram is -43939 Nmm and 229187.532 Nmm respectively.

Therefore, Mb=$\sqrt{43939^{2} + 229187.532^{2}}$= 233361.4 N

From (II) 155.25=$\frac{16}{\pi * d^3} * \sqrt{(1.5 * 233361.4)^2 + (199662)^2}$ => d=23.65mm~ $25\boldsymbol{mm}$

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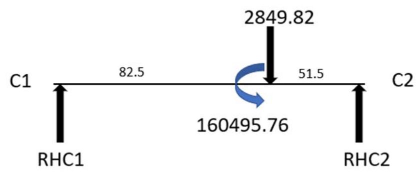
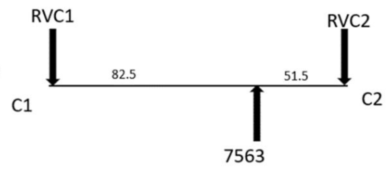
3) Loading Condition of intermediate Shaft C1C2 (Mt=598980 Nmm

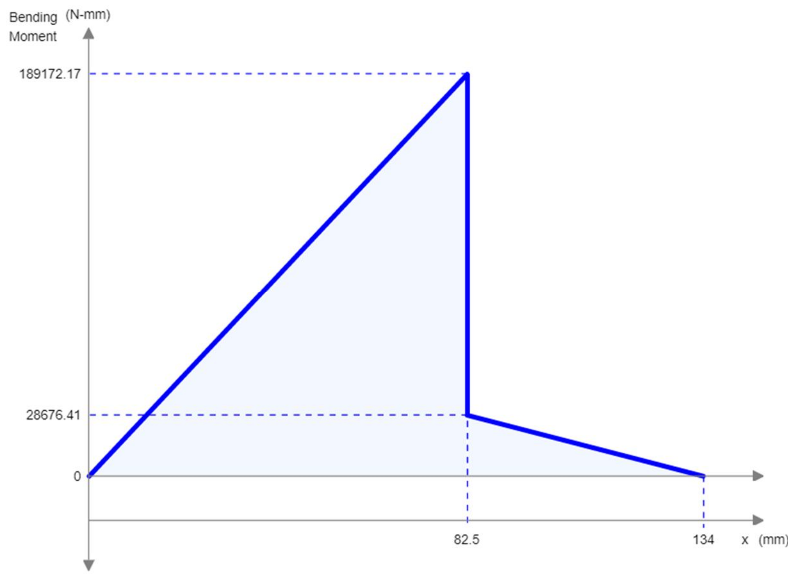


Horizontal Loading Diagram
RHC1 and RHC2 can be solved by $\sum MC1$=0
and equilibrium equation.
RHC1= 2292.996 N and RHC2=556.824 N

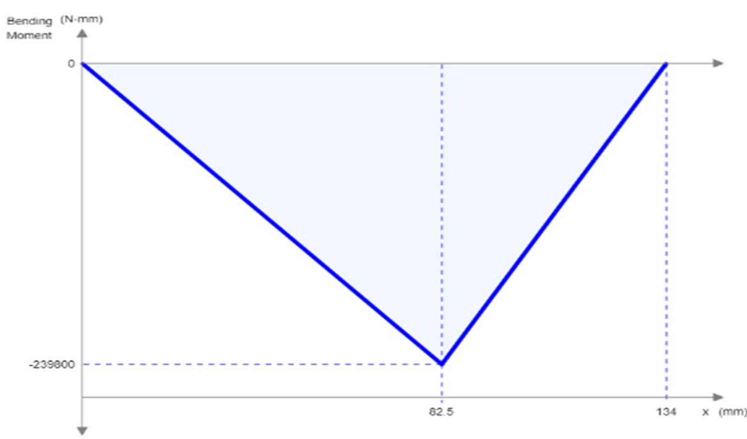


Vertical Loading Diagram
RVC1 and RVC2 can be solved by $\sum MC1$=0
and equilibrium equation.
RVC1=2906.7 N and RVC2=4656.3 N

Now, Bending Moment Diagram is Draw for both horizontal and vertical as shown below:



BMD for Horizontal Loading



BMD for vertical Loading

Maximum Bending moment in Horizontal and Vertical diagram is 189172.17 Nmm and -239800 Nmm respectively.

Therefore, Mb=$\sqrt{189172.17^2 + 239800^2}$= 305434.36 N

From (II) 155.25=$\frac{16}{\pi * d^3} * \sqrt{(1.5 * 305434.36)^2 + (598980)^2}$ => d=29.14MM~ 30$\boldsymbol{mm}$

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F. Design of Key

Material used is 50C4 (Syt=Syc=460 N/mm^2).
According to Maximum Shear Stress theory;
=>Ssy=0.5Syt (Yield strength in Shear is 0.5 times Yield Strength in tension)
Considering factor of safety as 3: $\tau=\frac{Ssy}{fos}$ => $\tau=0.5*\frac{460}{3}$= 76.67 N/mm^2.
The usual practise for square key is b=h=d/4. And the length of the key us given by:
$L=\frac{2Mt}{\tau db}$.

1) *For Input Shaft*
   d=15mm, b=h=d/4=3.75.
   $l=\frac{2*57622.5}{76.67*15*3.75}$ = 26.75 ~ $25mm$

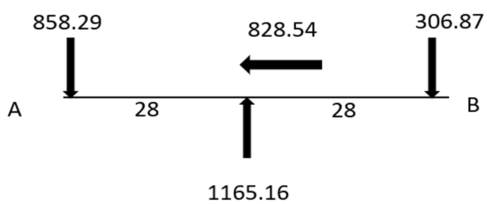
2) *For Intermediate Shaft*
   d=25mm, b=h=d/4=6.25
   $l=\frac{2*199662}{76.67*25*6.25}$ = 33.34 ~ $34mm$

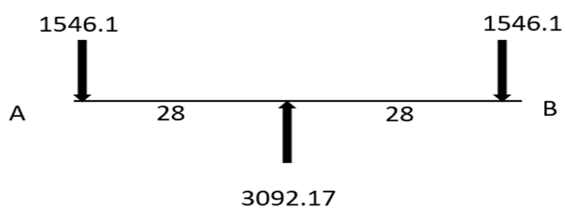
3) *For Output Shaft*
   d=30mm, b=h=d/4=7.5.
   $l=\frac{2*598980}{76.67*30*7.5}$ = 69.44 ~ $70mm$

G. Selection of Single Row Tapered Roller Bearing

Taper roller bearing can carry both axial and radial load and has more structural rigidity.
There are two types of arrangement possible for this particular bearing, they are: Face to Face arrangement and Back to Back arrangement. For this paper Face to Face arrangement is selected.
In order to calculate the total radial forces on the shafts following loading diagrams are drawn.

1) *Input Shaft (d=15 mm)*





The shaft is named AB such that the bearing will be placed at A and B with Face-to-Face configuration.
Now, to calculate the resultant radial force:
FrA = $\sqrt{(858.29^2 + 1546.1^2)}$ = 1768.35 N
FrB = $\sqrt{(306.87^2 + 1546.1^2)}$ = 1576.26 N
Ka= 828.54 N

Let us consider bearing 30302 (Skf catalogue) for the further calculations, the respective bearing details are given below:
d (ID)=15 mm
D (OD)= 42 mm
b(width)=14.25 mm
C (dynamic load constant) = 27.7kN
Co (static load constant) = 20kN
e=0.28
Y=2.1

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In this case for Face to Face configuration the following equations are used to calculated FaA and FaB : -

FaA=FaB+Ka

FaB=$0.5 * \frac{FrB}{Y}$

FaB= $0.5 * \frac{1576.26}{2.1}$ = 375.3 N

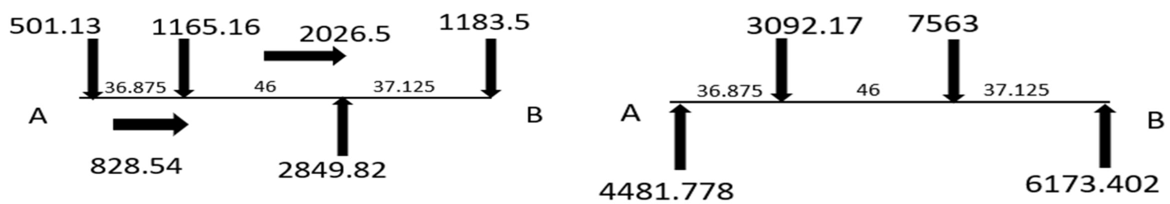
And FaA= 375.3+828.54 = 1203.84 N

Now, to check if the bearing is suitable:

$\frac{FaA}{FrA} = \frac{1203.84}{1768.35}$ = 0.68 > e P=0.4*FrA + Y* FaA P=0.4(1768.35) + 2.1(1203.84) P=707.34+2528.064 P=3235.404 N Consider L_{10} = 50 million rev and Load factor= 1.2 C1=$P * 50^{0.3} * 1.2$ C1=12554.54 N Which is less than Co **Hence the Bearing is Suitable**	$\frac{FaB}{FrB} = \frac{375.3}{1576.26}$ = 0.23 < e P=FrB P=1576.26 N C1=$P * 50^{0.3} * 1.2$ C1=6116.46 N Which is less than Co **Hence the Bearing is Suitable**

Hence SKF bearing 30302 is Selected.

2) *Intermediate Shaft (d=25 mm)*



The shaft is named AB such that the bearing will be placed at A and B with Face-to-Face configuration.

Now, to calculate the resultant radial force:

FrA = $\sqrt{(501.13^2 + 4481.778^2)}$ = 4509.7 N

FrB = $\sqrt{(1183.5^2 + 6173.402^2)}$ = 6285.82 N

Ka= 828.54 + 2026.5 = 2855.04 N

Let us consider bearing 30305 (Skf catalogue) for the further calculations, the respective bearing details are given below:

d (ID)=25 mm

D (OD)= 52 mm

b(width)=16.25 mm

C (dynamic load constant) = 38.1kN

Co (static load constant) = 33.5kN

e=0.37

Y=1.6

In this case for Face to Face configuration the following equations are used to calculated FaA and FaB : -

FaB=FaA+Ka

FaA=$0.5 * \frac{FrA}{Y}$

FaA= $0.5 * \frac{4509.7}{1.6}$ = 1409.28 N

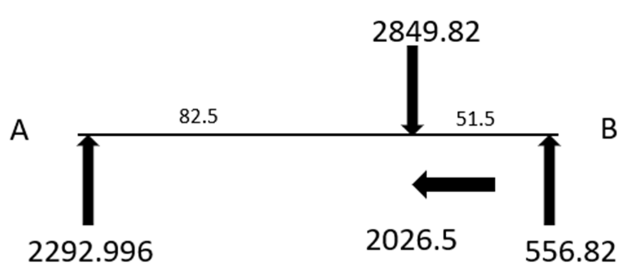
And FaB= 1409.28+2855.04 = 4264.32 N

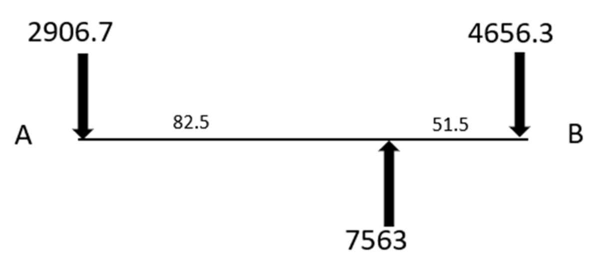
Now, to check if the bearing is suitable:

$\frac{FaB}{FrB} = \frac{4264.32}{6285.82}$ = 0.678 > e P=0.4*FrB + Y* FaB P=0.4(6285.82) + 1.6(4264.32) P=2514.328+6822.912 P=9337.24 N Consider L_{10} = 50 million rev and Load factor= 1.2 C1=$P * 50^{0.3} * 1.2$ C1=36231.87 N Which is less than C Hence the Bearing is Suitable	$\frac{FaA}{FrA} = \frac{1409.28}{4509.7}$ = 0.31 < e P=FrA P=4509.7 N C1=$P * 50^{0.3} * 1.2$ C1=17499.26 N Which is less than C Hence the Bearing is Suitable

Hence SKF bearing 30305 is Selected.

3) Output Shaft (d=30 mm)





The shaft is named AB such that the bearing will be placed at A and B with Face-to-Face configuration.

Now, to calculate the resultant radial force:

FrA = $\sqrt{(2292.996^2 + 2906.7^2)}$ = 3702.26 N

FrB = $\sqrt{(556.82 + 4656.3^2)}$ = 4689.5 N

Ka= 2026.5 N

Let us consider bearing <u>32006X</u> (Skf catalogue) for the further calculations, the respective bearing details are given below:

d (ID)=30 mm

D (OD)= 55 mm

b(width)=17 mm

C (dynamic load constant) = 43.9kN

Co (static load constant) = 44kN

e=0.43

Y=1.4

In this case for Face to Face configuration the following equations are used to calculated FaA and FaB : -

FaA=FaB+Ka

FaB=$0.5 * \frac{FrB}{Y}$

FaB= $0.5 * \frac{4689.5}{1.4}$ = 1674.82 N

And FaA= 1674.82+2026.5 = 3701.32 N

Now, to check if the bearing is suitable:

$\frac{FaA}{FrA} = \frac{3701.32}{3702.26}$ = 0.99 > e P=0.4*FrA + Y* FaA P=0.4(3702.26) + 1.4(3701.32) P=1480.904+5181.848 P=6662.752 N Consider L_{10} = 50 million rev and Load factor= 1.2 C1=$P * 50^{0.3} * 1.2$ C1=25853.89 N Which is less than Co **Hence the Bearing is Suitable**	$\frac{FaB}{FrB} = \frac{1674.82}{4689.5}$ = 0.357 < e P=FrB P=4689.5 N C1=$P * 50^{0.3} * 1.2$ C1=18196.95 N Which is less than Co **Hence the Bearing is Suitable**

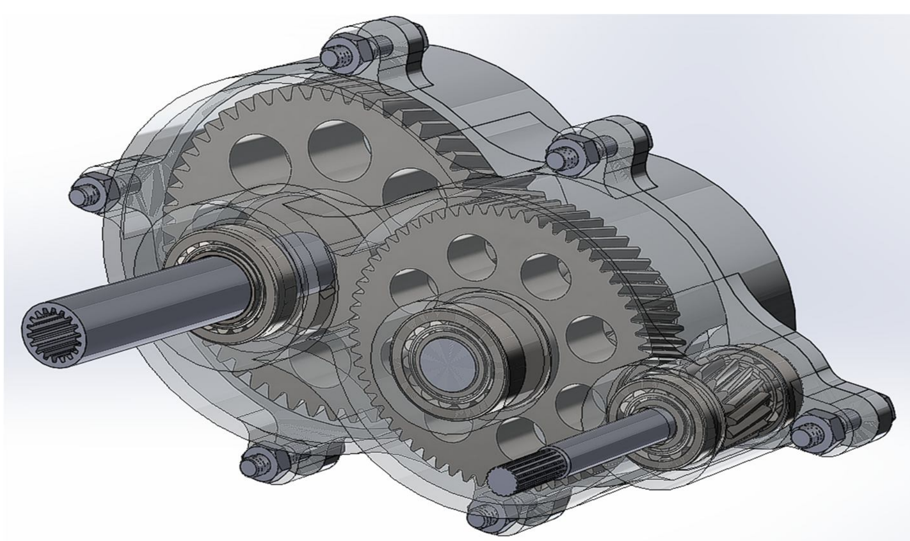
Hence SKF bearing 32006X is Selected.

4) Selected Bearings

SHAFT	SKF Bearings
INPUT SHAFT	30302
INTERMEDIATE SHAFT	30305
OUTPUT SHAFT	32006X

III. CAD IMAGES OF GEARBOX

The entire CAD model of the 2-Stage Helical gearbox was modelled in Solidworks 18.



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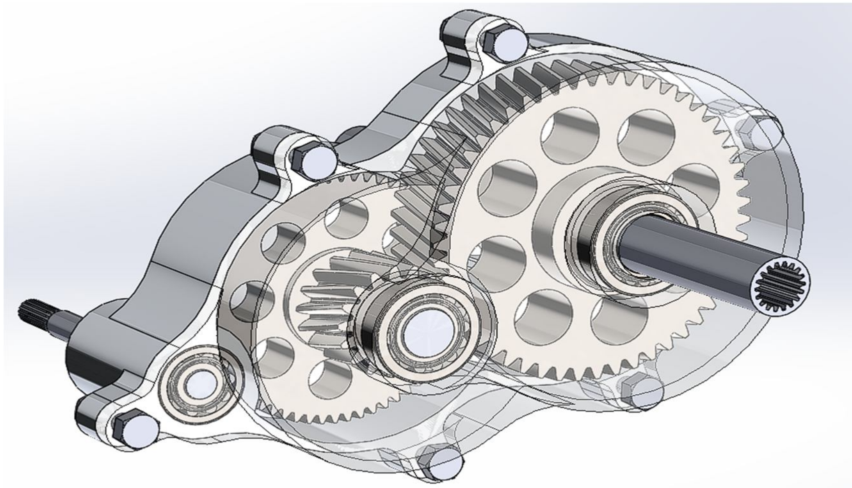

Figure 2. Overview of Gearbox

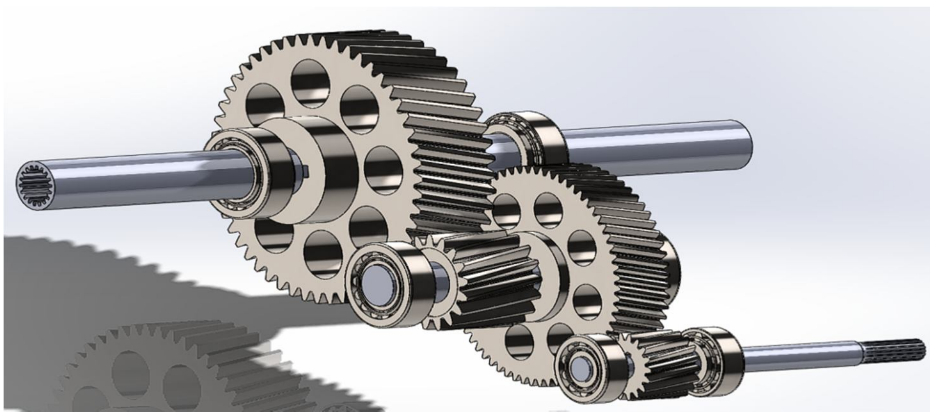

Figure 3. Helical Geartrain.

IV. ANALYSIS

The Analysis was done using Ansys Workbench software package for the material 15i4Cr1. In case of Helical Gearset, the Gear was fixed and the moment was applied on the pinion after giving it frictionless support. For Casing, the bolt holes were fixed and the radial and axial bearing load was given on the bearing support.

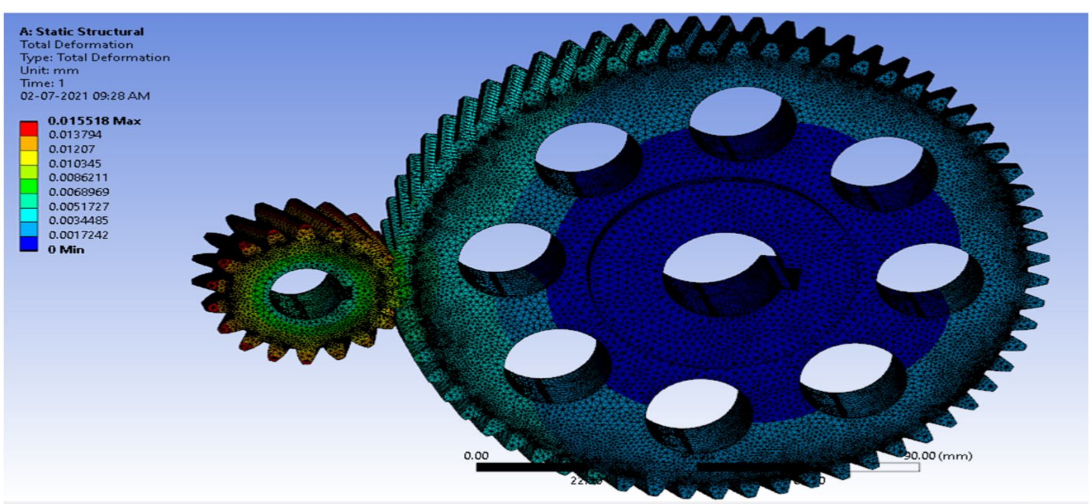

Figure 4. Stage 1 Gearset Deformation

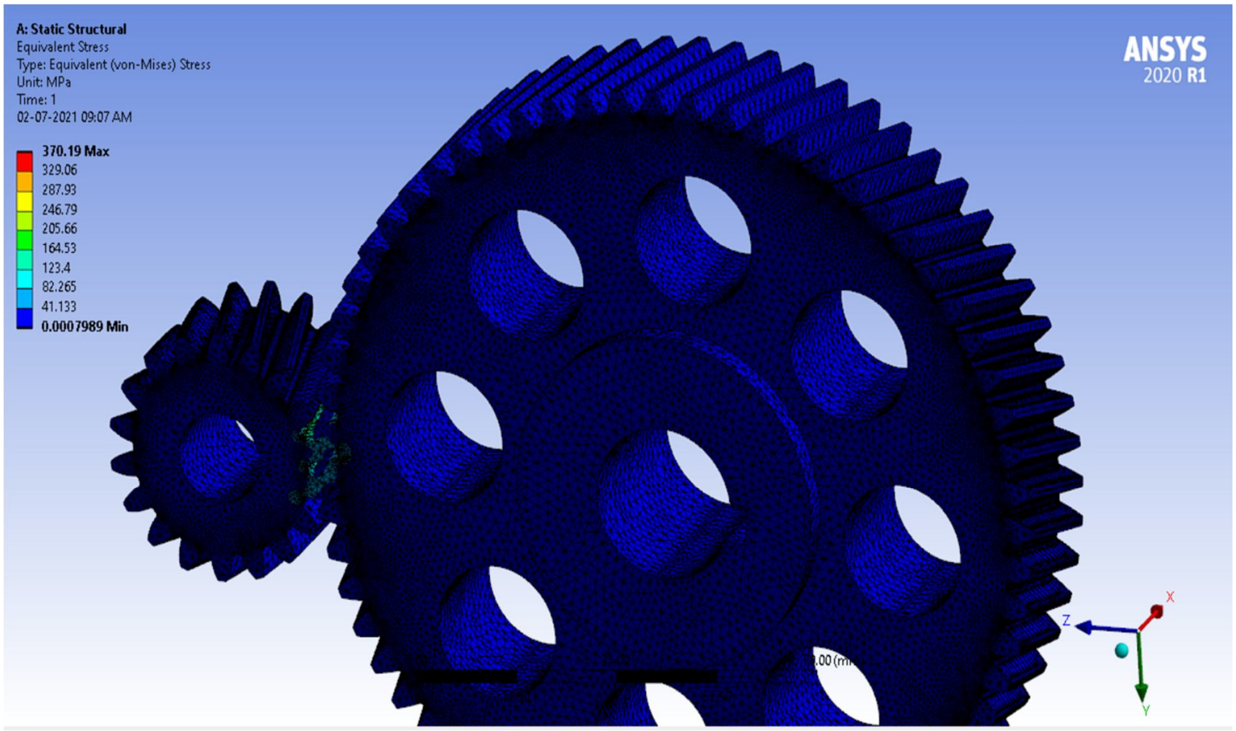

Figure 5. Stage 1 Gearset Equivalent Stress

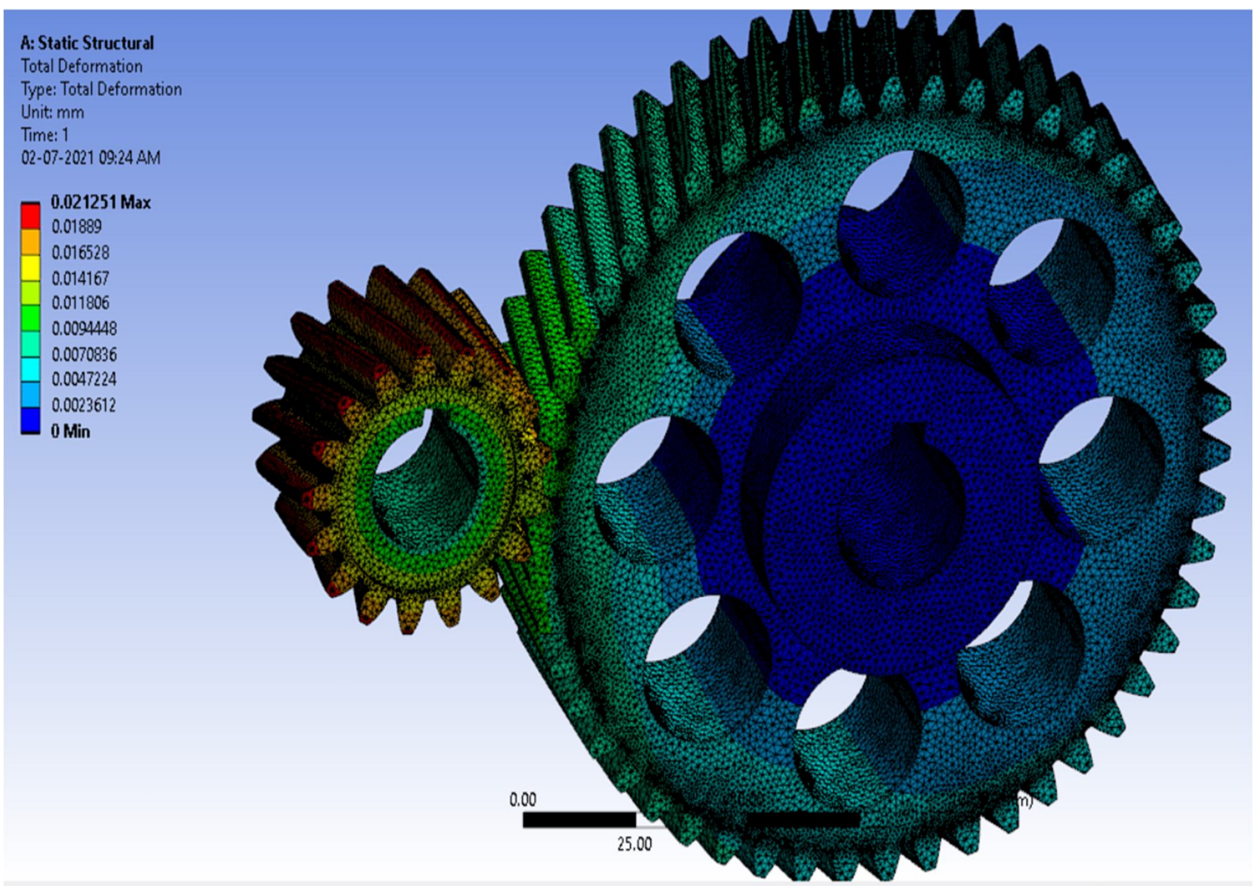

Figure 6. Stage 2 Gearset Deformation

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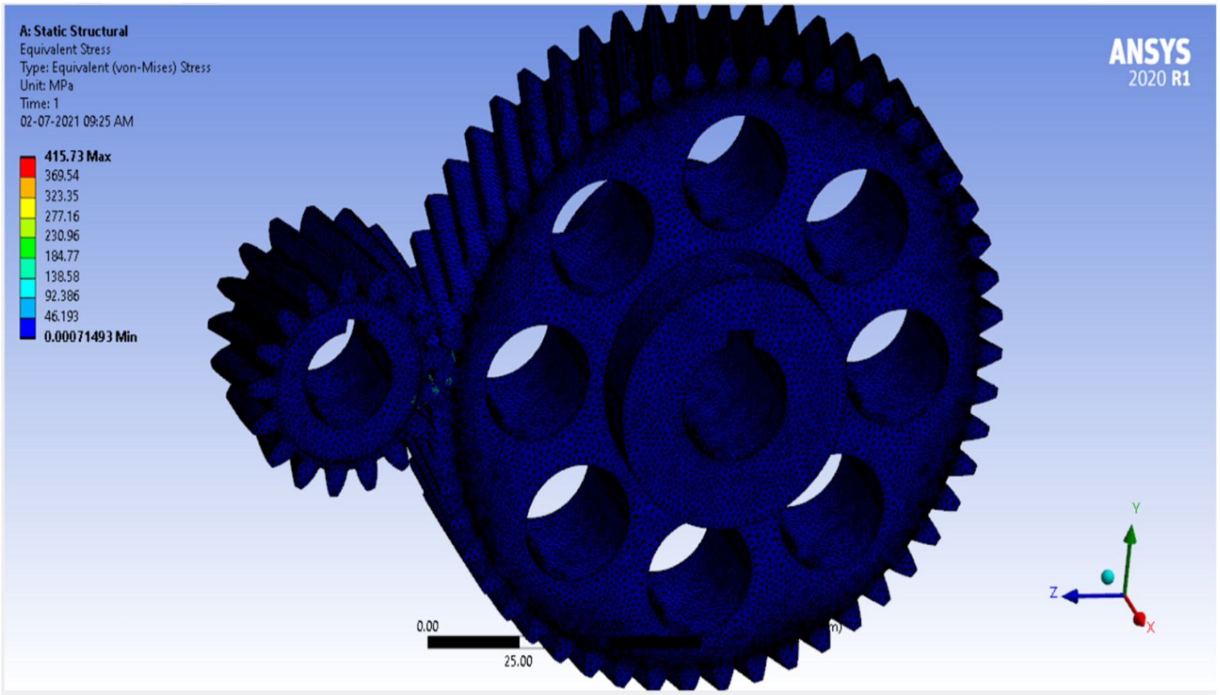

Figure 7. Stage 2 Gearset Equivalent Stress

COMPONENT	DEFORMATION (mm)	EQUIVALENT STRESS (Mpa)
STAGE 1	0.015518	370.19
STAGE 2	0.021251	415.73

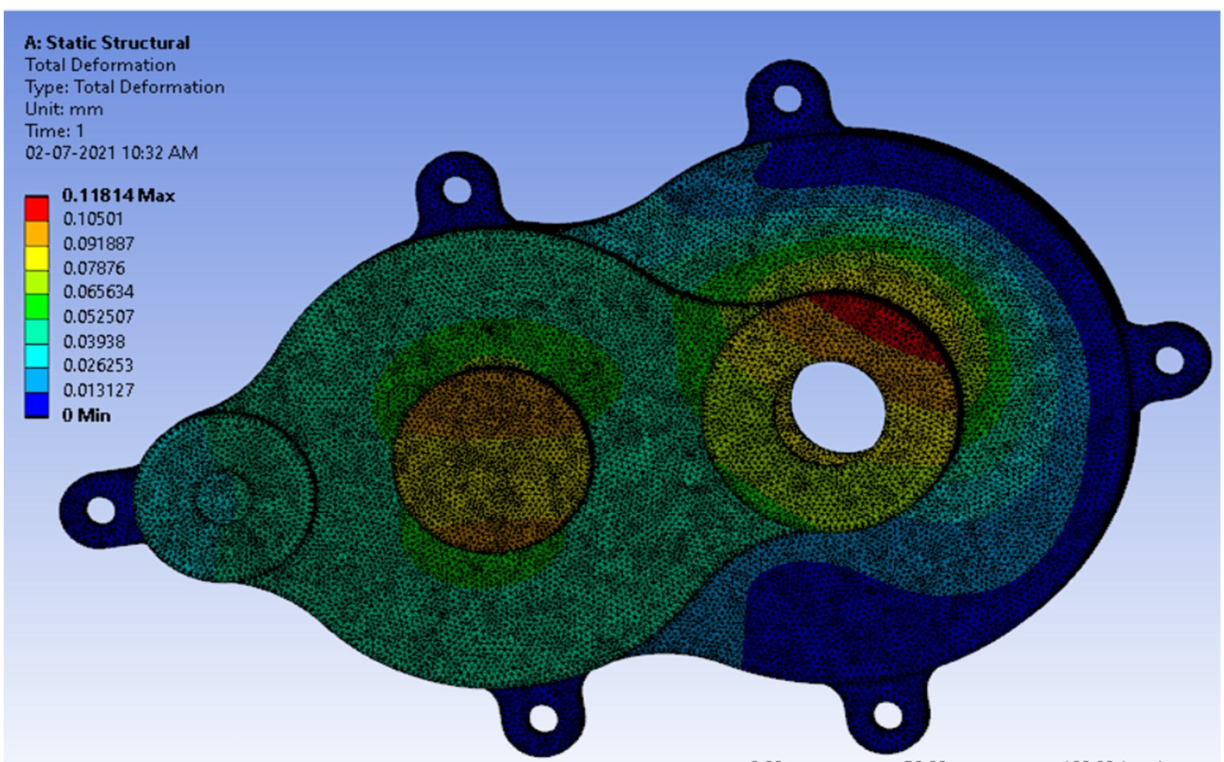

Figure 8. Casing Right Deformation

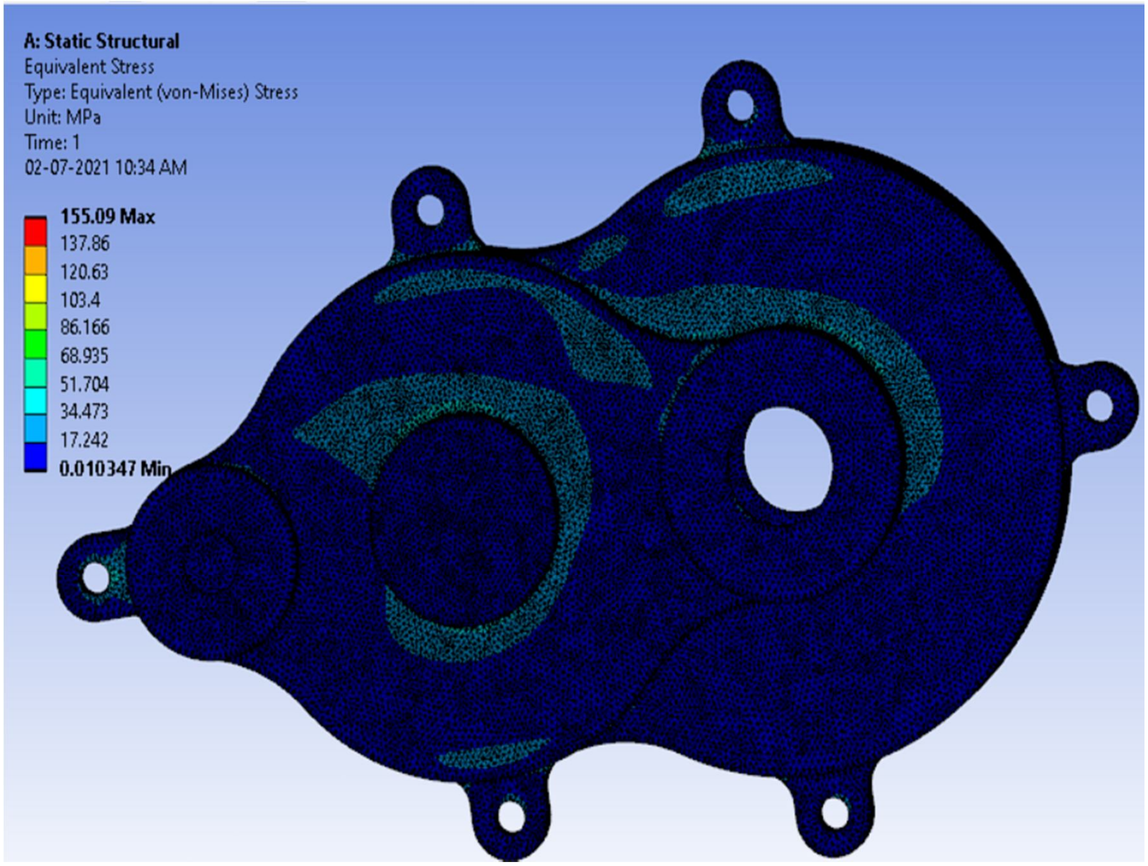

Figure 9. Casing right Equivalent Stress

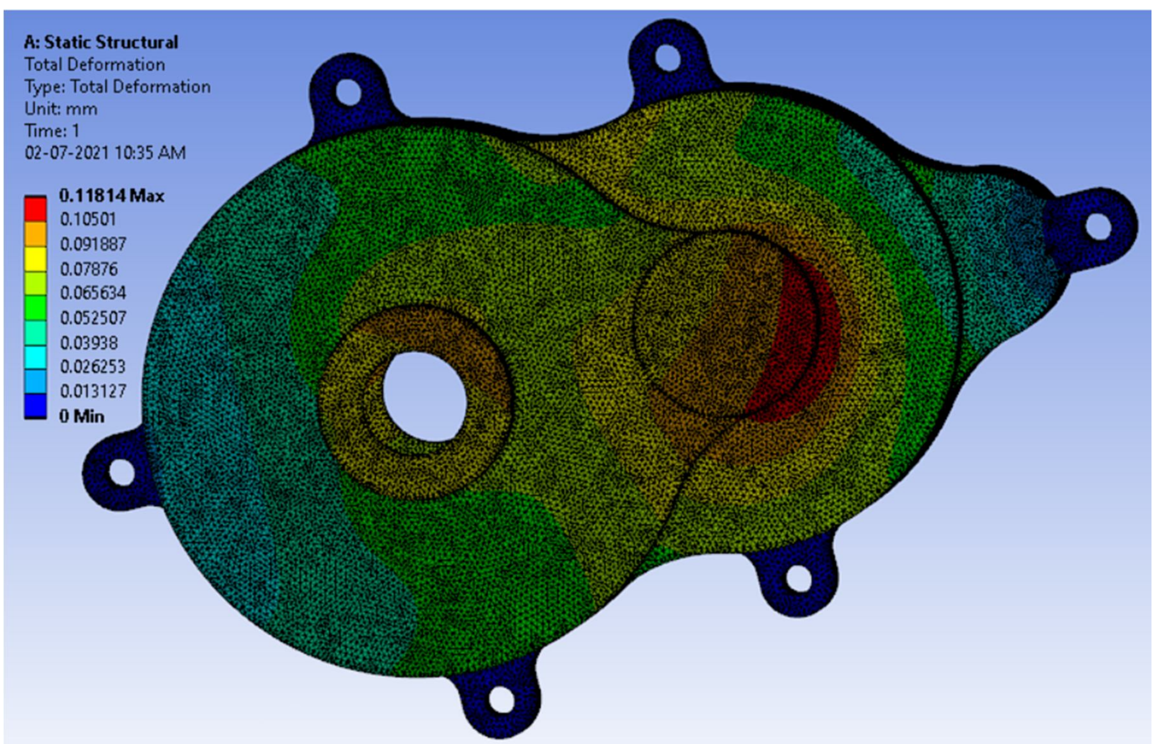

Figure 10. Casing left Deformation

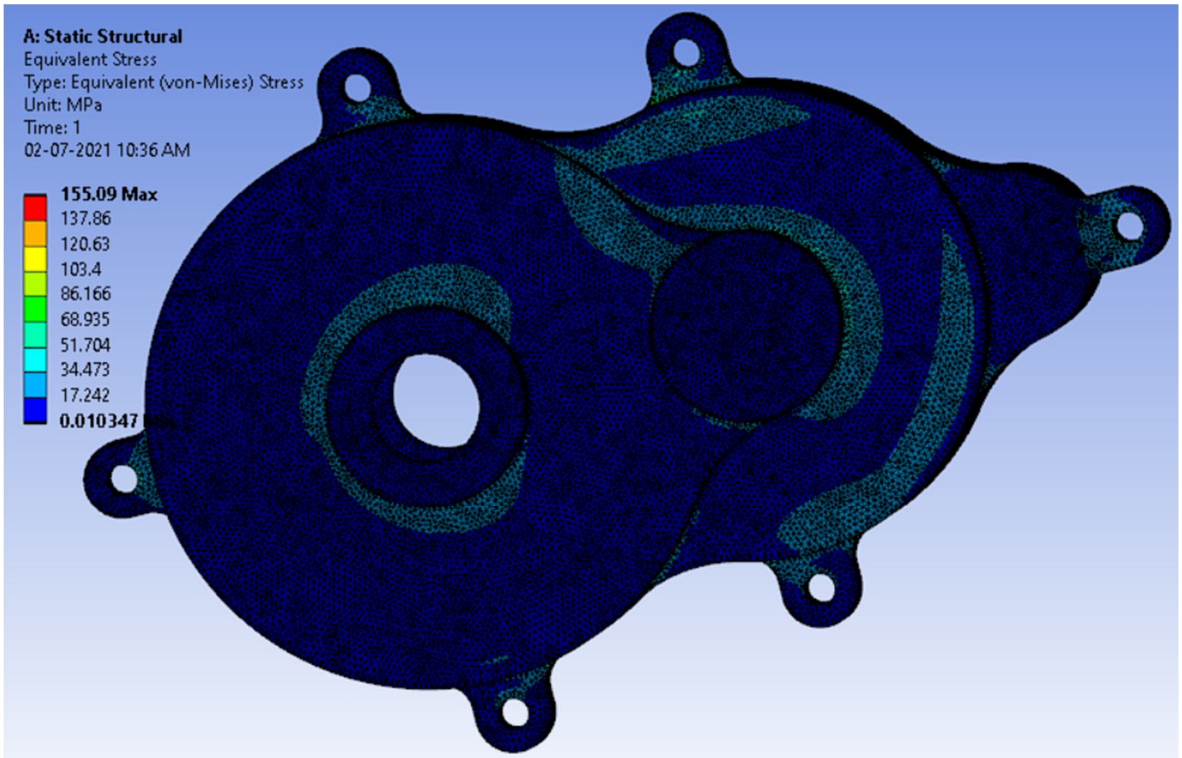

Figure 11. Casing left Equivalent Stress

COMPONENT	DEFORMATION (mm)	EQUIVAALENT STRESS (mm)
CASING LET	0.11814	155.09
CASING RIGHT	0.11814	155.09

V. FEA RESULTS

A. The factor of Safety of Stage 1 gearset is found to be 2.296.
B. The factor of Safety of Stage 2 gearset is found to be 2.0446
C. The factor of Safety of Casing gearset is found to be 3.243

VI. CONCLUSIONS

The major objective of this project was to Design, Develop and Analyze a 2-Stage Reduction Gearbox for ATV applications. The Overall gear ratio determined is found to be sufficient enough to propel the vehicle and achieve required speed and torque. This drivetrain was designed in such a way that it provides good acceleration, top speed and is reliable on different terrains. This powertrain system could attain a top speed of 50 kmph. 15Ni4Cr1 was selected as optimum material which would reduce the chances of failure.

REFERENCES

[1] Design of Machine Elements, 3rd Edition, V.B. Bhandari, McGraw Hill Publications
[2] Milliken and Milliken, Race Car Vehicle Dynamics.
[3] C. Smith, “Tune to Win”, AERO publishers Inc
[4] www.skf.com