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Design and Analysis of 2-Stage Gearbox for ATV Applications

Prathmesh Surnis¹, Dr. Naresh Jaiswal²

¹Mechanical Engineering Department, Pune Vidyarthi Griha's College of Engineering and Technology and G.K. Pate (Wani) Institute of Management, Pune.

²Assistant Professor, Mechanical Engineering Department, Pune Vidyarthi Griha's College of Engineering and Technology and G.K. Pate (Wani) Institute of Management, Pune.

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.

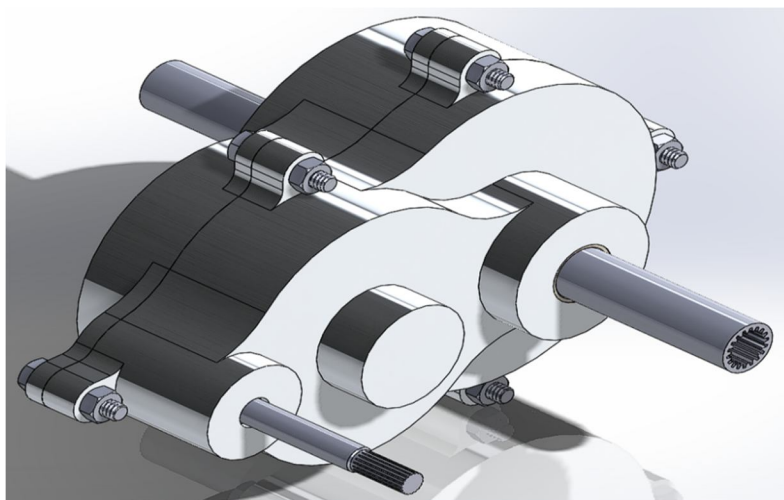


Figure 1. Reduction gearbox

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

- To obtain an output Top speed of 50 kmph.
- 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:

$$RR = GVW * f_r * \cos(\theta)$$

$$AR = 0.5 * \rho (air) * v^2 * A * Cd$$

$$GR = GVW * \sin(\theta)$$

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°.

$$RR = 240 * 9.81 * 0.14 * \cos(30.35) = 28.44 \text{ N}$$

Aerodynamic force was considered as 0 in this case.

$$GR = GVW * \sin(30.35) = 1189.42 \text{ N}$$

$$\text{Thus, } TTR = RR + AR + GR = 1217.871 \text{ N.}$$

The total amount of force which can be transferred by the wheels without slipping can be found out by $TTE_{max} = GVW * \mu * \sin(\theta) * 0.60$.

$$TTE_{max} = 1218.83 \text{ N.}$$

Gradeability is selected such that the $TTE_{max} \geq TTE$.

$$\text{Torque at wheels } (T_w) = TTE * 0.283337 = 345 \text{ Nm.}$$

C. Calculation of Gearbox Reduction

$$\text{Torque at wheels } (T_w) 345 = 19.7 * 3.9 * 0.75 * GR * 0.99 * 0.99 \Rightarrow GR = 6.1$$

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: $13.89(50\text{kmph}) = \frac{3800}{0.5} * 0.75 * \frac{1}{GR} * \frac{2\pi}{60} * 0.283337 \Rightarrow GR = 12.17$

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:

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

Torque at wheels considering 10.5 reduction ratio:

$$T_w = 19.7 * 3.9 * 0.75 * 10.5 * 0.99 * 0.99 = 592.996 \text{ Nm}$$

$$RR = 2354 * 0.014 = 32.956 \text{ Nm}$$

Engine rpm = 2800

$$\text{Wheel rpm} = \frac{2800}{3.9} * 0.75 * \frac{1}{10.5} * 0.99 * 0.99 = 50.26$$

$$\text{Wheel velocity} = 2\pi * \frac{50.26}{60} * 0.283337 = 1.49309$$

$$AR = 0.5 * 1.122 * 1.49309^2 * 0.78 * 1.08 = 1.05 \text{ Nm}$$

$$\text{Net Force on wheel (F)} = \frac{592.996}{0.283337} - (1.05 + 32.956) = 2058.893$$

$$\text{Acceleration} = F * \frac{9.81}{2354} = 8.58 \text{ m/s}^2$$

D. Gear Dimension Selection

The Material used for gear is case hardened alloy steel 15Ni4Cr1: Sut=1500N/mm², Syt=850 N/mm² 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

$$PCD_p = \frac{Z_p * M_n}{\cos \phi} = \frac{18 * 2}{\cos 15^\circ} = 37.27 \text{ mm} \quad \text{and} \quad PCD_g = \frac{Z_g * M_n}{\cos \phi} = \frac{63 * 2}{\cos 15^\circ} = 130.44 \text{ mm}$$

$$\text{Centre to Centre distance} = \frac{M_n * (Z_p + Z_g)}{2 * \cos(\phi)} = \frac{2 * (18 + 63)}{2 * \cos(15^\circ)} = 83.85 \Rightarrow \text{C-C} \sim 85 \text{ mm}$$

$$\text{Virtual No. of teeth for pinion (Z'p)} = \frac{Z_p}{\cos(\phi)^3} = \frac{18}{(\cos 15^\circ)^3} = 19.97$$

$$\text{Virtual No. of teeth for gear (Z'g)} = \frac{Z_g}{\cos(\phi)^3} = \frac{63}{(\cos 15^\circ)^3} = 69.90$$

$$\text{Face width (b)} \geq \frac{\pi * M_n}{\sin(\phi)} \Rightarrow b = \frac{\pi * 2}{\sin(15^\circ)} = 24.27 \sim 25 \text{ mm}$$

$$\text{Input torque to Stage 1 (Mt)} = 19.7 * 3.9 * 0.75 = 57.6225 \text{ Nm} = 57622.5 \text{ Nmm.}$$

$$P_t = \frac{2 * M_t}{PCD_p} = \frac{2 * 57622.5}{37.27} = 3092.17 \text{ N}$$

$$P_r = P_t * \frac{\tan(\alpha_n)}{\cos(\phi)} = 3092.17 * \frac{\tan(20^\circ)}{\cos(15^\circ)} = 1165.16 \text{ N}$$

$$P_a = P_t * \tan(\phi) = 3092.17 * \tan(15^\circ) = 828.54 \text{ N}$$

The beam Strength is given by the following equation:

$$S_b = M_n * b * \sigma_b * Y \quad [Y \text{ is Lewis form factor for } Z' \text{ and } \sigma_b \text{ is permissible bending stress}]$$

$$\Rightarrow Y = \pi * \left(0.154 - \frac{0.912}{Z_p'} \right) = \pi * \left(0.154 - \frac{0.912}{19.97} \right) = 0.340 \quad \text{and } \sigma_b = S_{ut}/3 = 500 \text{ Nmm}^2.$$

Thus, S_b is calculated as:

$$S_b = 2 * 25 * 500 * 0.340 = 8500 \text{ N}$$

The Wear Strength is given by the following equation:

$$S_w = \frac{b * Q * P * C * D * p * K}{\cos(\phi)^2}$$

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

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

$$S_w = \frac{b * Q * P * C * D * p * K}{\cos(\phi)^2} = \frac{25 * 1.556 * 37.27 * 6.76}{\cos(15)^2} = 10504.32 \text{ N}$$

Now, to calculate effective force on the tooth,

Max torque is 19.7 Nm @ 2800 rpm \Rightarrow the rpm at Stage 1 is $\frac{2800}{3.9} * 0.75 = 538.46$, the C_v is calculated as:

$$C_v = \frac{5.6}{5.6 + \sqrt{v}} \quad [v \text{ is pitch line velocity}] \quad \text{and } C_s = 1.25$$

$$v = \frac{\pi * P * C * D * p * r * p * m}{60000} = \frac{\pi * 37.27 * 538.46}{60000} = 1.05 \text{ m/s} \Rightarrow C_v = \frac{5.6}{5.6 + \sqrt{1.05}} = 0.845$$

$$C \text{ (deformation factor)} = 11400 \text{ N/mm}^2$$

$$e = e_p + e_g$$

For IS Grade 5, the error is $e = 5 + 0.4\phi$ and $\phi = M_n + 0.25\sqrt{P * C * D}$

$$e_p = 5 + 0.4(2 + 0.25\sqrt{37.27}) = 6.41 \mu\text{m}$$

$$e_g = 5 + 0.4(2 + 0.25\sqrt{130.44}) = 6.94 \mu\text{m}$$

$$e = 6.41 + 6.94 = 13.35 \mu\text{m} = 0.01335 \text{ mm}$$

Therefore, the dynamic load can be calculated by using the Buckingham's Load equation:

$$P_d = \frac{21 * v * (C * e * b * (\cos\phi)^2 + P_t) * \cos\phi}{21 * v + \sqrt{C * e * b * (\cos\phi)^2 + P_t}} = 1366.18 \text{ N}$$

Thus, P_{eff} can be calculated as:

$$P_{eff} = C_s * P_t + P_d = 1.25 * 3092.17 + 1366.18 = 5231.4 \text{ N}$$

Hence, the factor of safety for bending and wear can be calculated as:

$$f_{os}(\text{beam}) = S_b / P_{eff} = 8500 / 5231.4 = \mathbf{1.62}$$

$$f_{os}(\text{wear}) = S_w / P_{eff} = 10504.32 / 5231.4 = \mathbf{2.007}$$

- 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 as it has sufficient factor of safety.

Lead angle	15°
Normal Module (M_n)	3 mm
No. of teeth of Pinion (Z_p)	17
No. of teeth is Gear (Z_g)	51

$$PCD_p = \frac{Z_p * Mn}{\cos \phi} = \frac{17 * 3}{\cos 15^\circ} = 52.8 \text{ mm} \quad \text{and} \quad PCD_g = \frac{Z_g * Mn}{\cos \phi} = \frac{51 * 3}{\cos 15^\circ} = 158.4 \text{ mm}$$

$$\text{Centre to Centre distance} = \frac{Mn * (Z_p + Z_g)}{2 * \cos(\phi)} = \frac{2 * (17 + 51)}{2 * \cos(15^\circ)} = 105.6 \Rightarrow \text{C-C} \sim 106.5 \text{ mm}$$

$$\text{Virtual No. of teeth for pinion (Z'p)} = \frac{Z_p}{\cos(\phi)^3} = \frac{17}{(\cos 15^\circ)^3} = 18.86$$

$$\text{Virtual No. of teeth for gear (Z'g)} = \frac{Z_g}{\cos(\phi)^3} = \frac{51}{(\cos 15^\circ)^3} = 56.58$$

$$\text{Face width (b)} \geq \frac{\pi * Mn}{\sin(\phi)} \Rightarrow b = \frac{\pi * 3}{\sin(15^\circ)} = 36.41 \sim 37 \text{ mm}$$

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

$$P_t = \frac{2 * Mt}{PCD_p} = \frac{2 * 199662}{52.8} = 7563 \text{ N}$$

$$P_r = P_t * \frac{\tan(\alpha_n)}{\cos(\phi)} = 7563 * \frac{\tan(20^\circ)}{\cos(15^\circ)} = 2849.82 \text{ N}$$

$$P_a = P_t * \tan(\phi) = 7563 * \tan(15^\circ) = 2026.5 \text{ N}$$

The beam Strength is given by the following equation:

$$S_b = Mn * b * \sigma_b * Y \quad [Y \text{ is Lewis form factor for } Z' \text{ and } \sigma_b \text{ is permissible bending stress}]$$

$$\Rightarrow Y = \pi * \left(0.154 - \frac{0.912}{Z'p} \right) = \pi * \left(0.154 - \frac{0.912}{18.86} \right) = 0.33192 \quad \text{and} \quad \sigma_b = S_{ut} / 3 = 500 \text{ Nmm}^2.$$

Thus, S_b is calculated as:

$$S_b = 3 * 37 * 500 * 0.33192 = 18421.56 \text{ N}$$

The Wear Strength is given by the following equation:

$$S_w = \frac{b * Q * PCD_p * K}{\cos(\phi)^2}$$

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

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

$$S_w = \frac{b * Q * PCD_p * K}{\cos(\phi)^2} = \frac{37 * 1.5 * 52.8 * 6.76}{\cos(15^\circ)^2} = 21231.76 \text{ N}$$

Now, to calculate effective force on the tooth,

$$\text{Max torque is } 19.7 \text{ Nm @ } 2800 \text{ rpm} \Rightarrow \text{the rpm at Stage 2 is } \frac{2800}{3.9} * 0.75 * \frac{1}{3.5} * 0.99 = 152.3, \text{ the } C_v \text{ is calculated as:}$$

$$C_v = \frac{5.6}{5.6 + \sqrt{v}} \quad [v \text{ is pitch line velocity}] \quad \text{and} \quad C_s = 1.25$$

$$v = \frac{\pi * PCD_p * rpm}{60000} = \frac{\pi * 52.8 * 152.3}{60000} = 0.42104 \text{ m/s} \Rightarrow C_v = \frac{5.6}{5.6 + \sqrt{0.42104}} = 0.89616$$

C (deformation factor) = 11400 N/mm²

$e = e_p + e_g$

For IS Grade 5, the error is $e = 5 + 0.4\phi$ and $\phi = Mn + 0.25\sqrt{PCD}$

$e_p = 5 + 0.4(3 + 0.25\sqrt{52.8}) = 6.92\mu\text{m}$

$e_g = 5 + 0.4(3 + 0.25\sqrt{158.4}) = 7.46\mu\text{m}$

$e = 6.92 + 7.46 = 14.38\mu\text{m} = 0.01438\text{mm}$

Therefore, the dynamic load can be calculated by using the Buckingham's Load equation:

$$P_d = \frac{21 \cdot v \cdot (C \cdot e \cdot b \cdot (\cos\phi)^2 + Pt) \cdot \cos\phi}{21 \cdot v + \sqrt{C \cdot e \cdot b \cdot (\cos\phi)^2 + Pt}} = 911.94 \text{ N}$$

Thus, P_{eff} can be calculated as:

$P_{eff} = C_s \cdot Pt + P_d = 1.25 \cdot 7563 + 911.94 = 10365.7 \text{ N}$

Hence, the factor of safety for bending and wear can be calculated as:

$fos(\text{beam}) = S_b / P_{eff} = 18421.56 / 10365.7 = 1.77$

$fos(\text{wear}) = S_w / P_{eff} = 21231.76 / 10365.7 = 2.048$

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. ($S_{yt} = 1100\text{N/mm}^2$ and $S_{ut} = 1150\text{N/mm}^2$)

The shafts are designed based on ASME code.

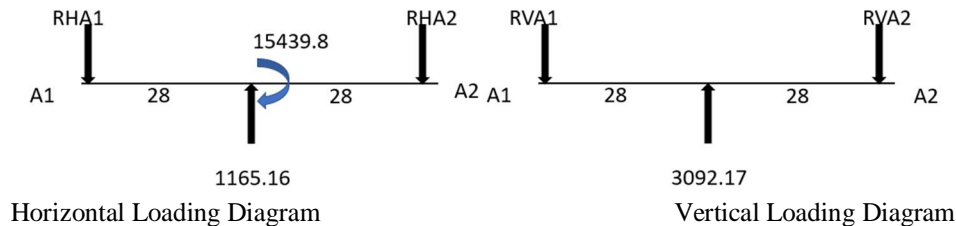
$\tau_{max} = 0.18 \cdot S_{ut}$, since keyways are present,

$\tau_{max} = 0.75 \cdot 0.18 \cdot 1150 = 155.25 \text{ N/mm}^2 \dots\dots(I)$

$K_b = 1.5$ and $K_t = 1$ [Since the load applied is gradual] and thus the τ_{max} is given by,

$\tau_{max} = \frac{16}{\pi \cdot d^3} \cdot \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \dots\dots\dots(II)$

1) Loading Condition of Input Shaft A1A2 ($M_t = 57622.5 \text{ Nmm}$)



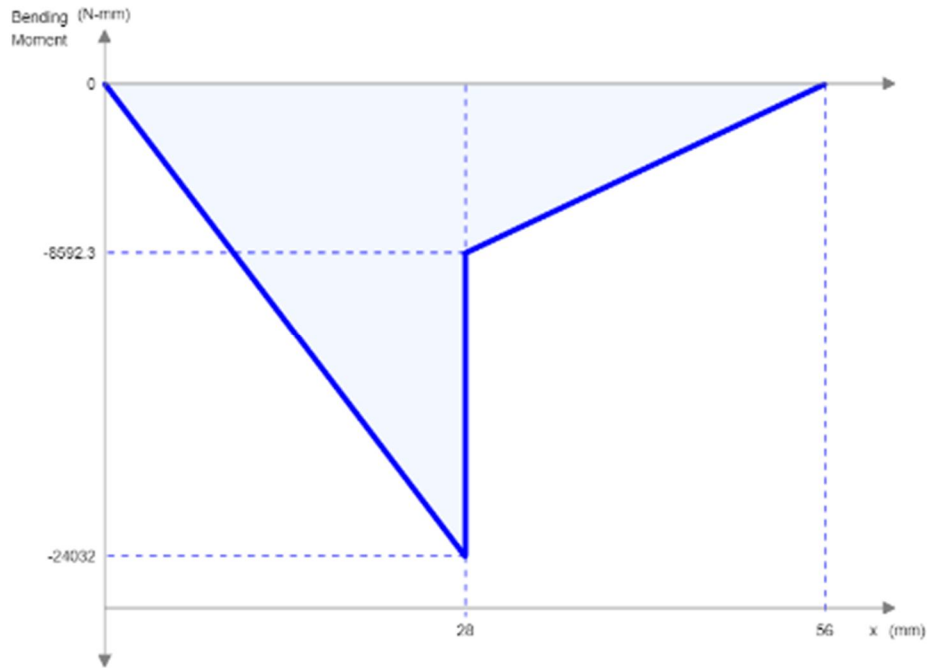
RHA1 and RHA2 can be solved by $\sum MA_1 = 0$ and equilibrium equation.

RVA1 and RVA2 can be solved by $\sum MA_1 = 0$ and equilibrium equation.

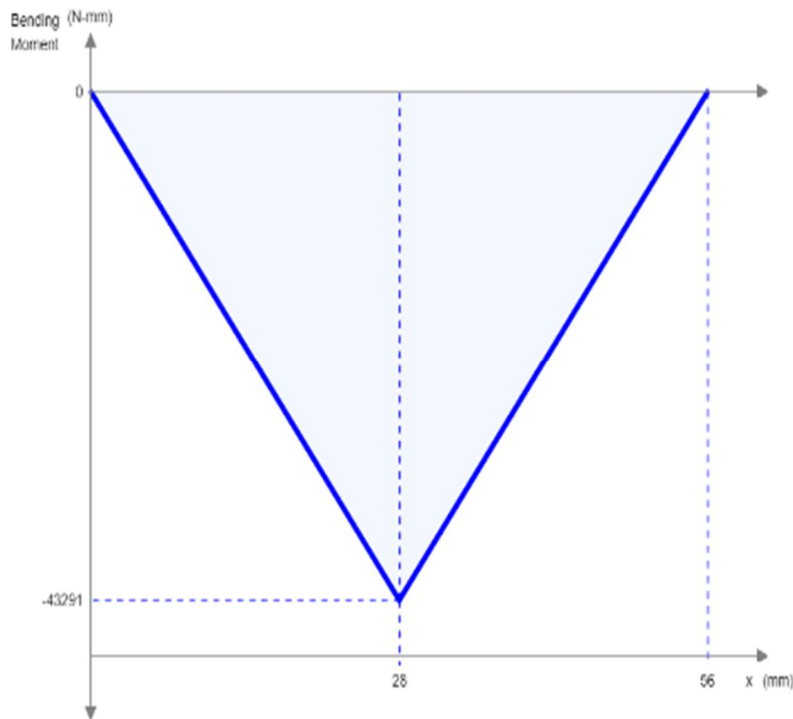
$RHA_1 = 858.29 \text{ N}$ and $RHA_2 = 306.87 \text{ N}$

$RVA_1 = 1546.1 \text{ N}$ and $RVA_2 = 1546.1 \text{ N}$

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



BMD for Horizontal Loading



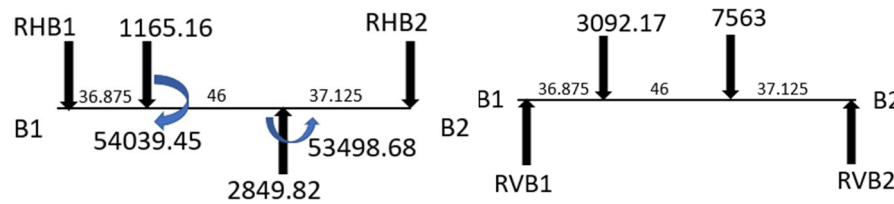
BMD for vertical Loading

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

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

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

2) Loading Condition of intermediate Shaft BIB2 ($M_t=199662 \text{ Nmm}$)



Horizontal Loading Diagram

RHB1 and RHB2 can be solved by $\sum MB_1=0$ and equilibrium equation.

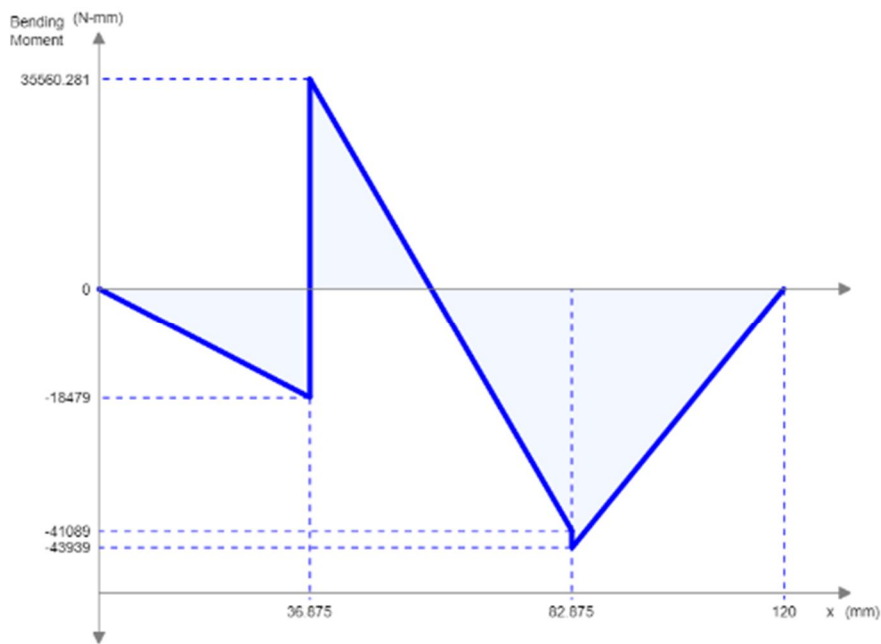
$RHB_1= 501.13 \text{ N}$ and $RHB_2=1183.5 \text{ N}$

Vertical Loading Diagram

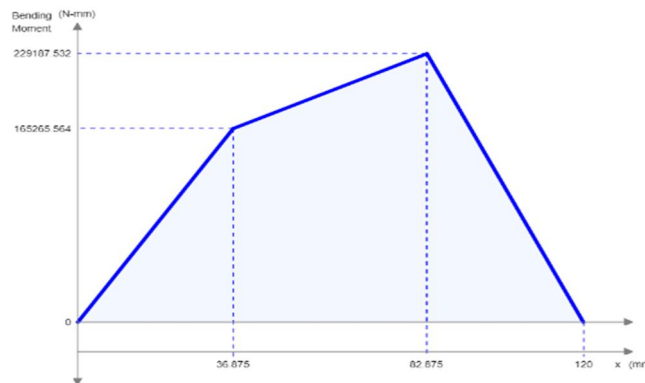
RVB1 and RVB2 can be solved by $\sum MB_1=0$ and equilibrium equation.

$RVB_1=4481.778 \text{ N}$ and $RVB_2=6173.402 \text{ N}$

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



BMD for Horizontal Loading



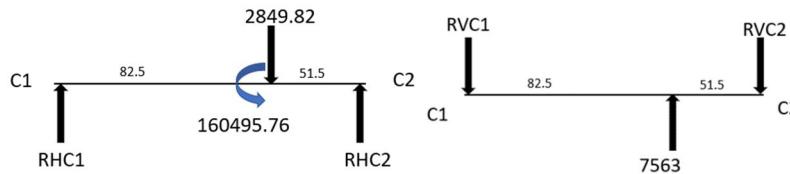
BMD for vertical Loading

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

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

From (II) $155.25 = \frac{16}{\pi \cdot d^3} * \sqrt{(1.5 * 233361.4)^2 + (199662)^2} \Rightarrow d = 23.65 \text{ mm} \sim 25 \text{ mm}$

3) Loading Condition of intermediate Shaft C1C2 ($M_t=598980 \text{ Nmm}$)



Horizontal Loading Diagram

RHC1 and RHC2 can be solved by $\sum MC_1=0$ and equilibrium equation.

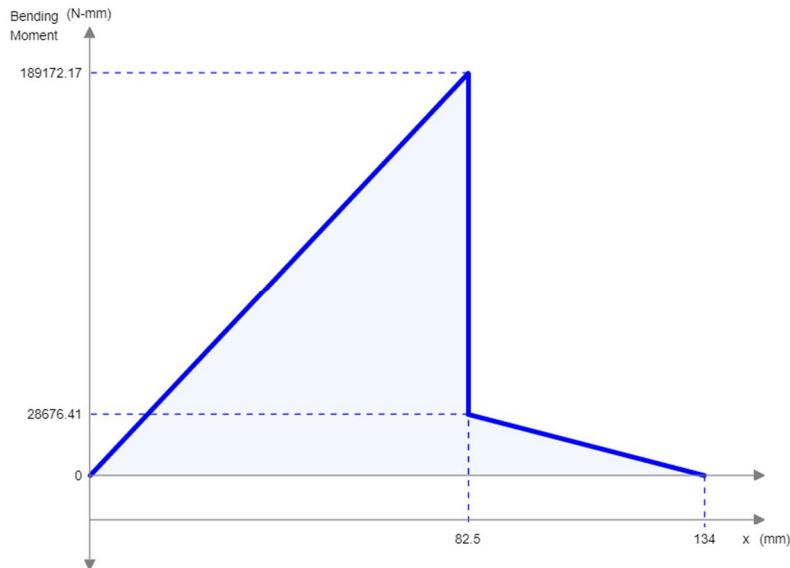
RHC1= 2292.996 N and RHC2=556.824 N

Vertical Loading Diagram

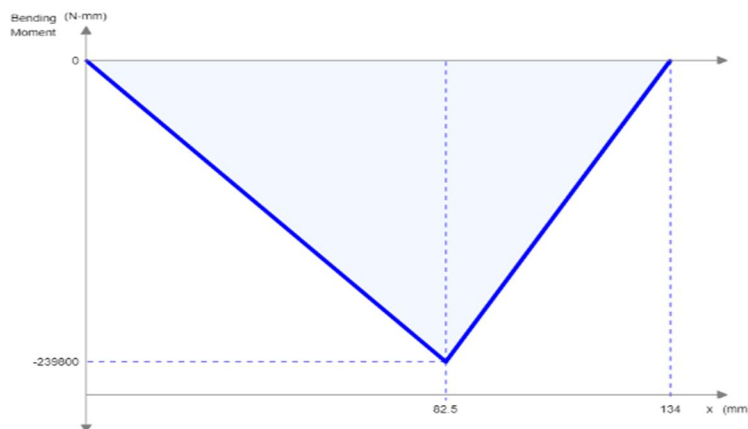
RVC1 and RVC2 can be solved by $\sum MC_1=0$ and equilibrium equation.

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, $M_b = \sqrt{189172.17^2 + 239800^2} = 305434.36 \text{ N}$

From (II) $155.25 = \frac{16}{\pi \cdot d^3} * \sqrt{(1.5 * 305434.36)^2 + (598980)^2} \Rightarrow d = 29.14 \text{ MM} \sim 30 \text{ mm}$

F. Design of Key

Material used is 50C4 ($S_{yt}=S_{yc}=460 \text{ N/mm}^2$).

According to Maximum Shear Stress theory;

$\Rightarrow S_{sy}=0.5S_{yt}$ (Yield strength in Shear is 0.5 times Yield Strength in tension)

Considering factor of safety as 3: $\tau = \frac{S_{sy}}{f_{os}} \Rightarrow \tau = 0.5 * \frac{460}{3} = 76.67 \text{ N/mm}^2$.

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

$$d=15\text{mm}, b=h=d/4=3.75.$$

$$l = \frac{2 * 57622.5}{76.67 * 15 * 3.75} = 26.75 \sim 25\text{mm}$$

2) For Intermediate Shaft

$$d=25\text{mm}, b=h=d/4=6.25$$

$$l = \frac{2 * 199662}{76.67 * 25 * 6.25} = 33.34 \sim 34\text{mm}$$

3) For Output Shaft

$$d=30\text{mm}, b=h=d/4=7.5.$$

$$l = \frac{2 * 598980}{76.67 * 30 * 7.5} = 69.44 \sim 70\text{mm}$$

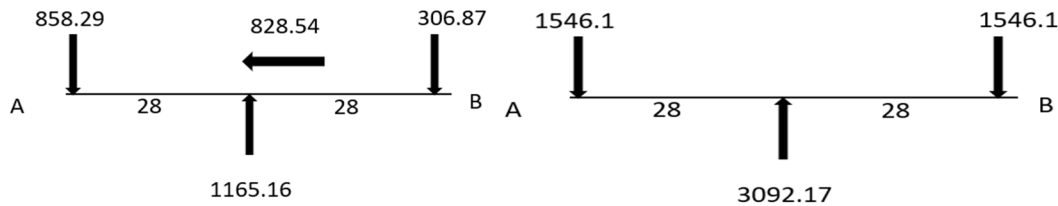
G. Selection of Single Row Tapered Roller Bearing

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 ($d=15 \text{ mm}$)



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:

$$FrA = \sqrt{(858.29^2 + 1546.1^2)} = 1768.35 \text{ N}$$

$$FrB = \sqrt{(306.87^2 + 1546.1^2)} = 1576.26 \text{ N}$$

$$K_a = 828.54 \text{ N}$$

Let us consider bearing 30302 (Skf catalogue) for the further calculations, the respective bearing details are given below:

$d \text{ (ID)} = 15 \text{ mm}$

$D \text{ (OD)} = 42 \text{ mm}$

$b \text{ (width)} = 14.25 \text{ mm}$

$C \text{ (dynamic load constant)} = 27.7\text{kN}$

$C_o \text{ (static load constant)} = 20\text{kN}$

$e = 0.28$

$Y = 2.1$

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

$$FaA = FaB + Ka$$

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

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

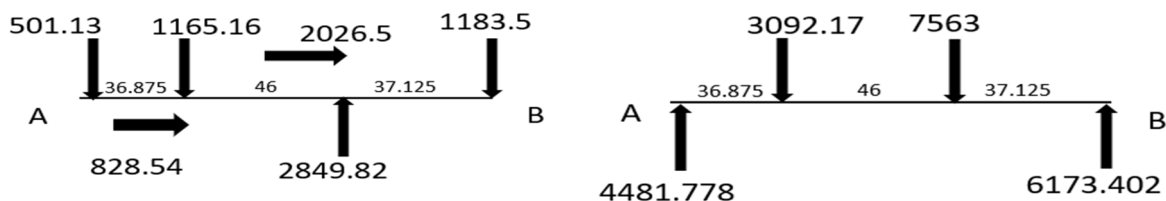
$$\text{And } FaA = 375.3 + 828.54 = 1203.84 \text{ N}$$

Now, to check if the bearing is suitable:

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

Hence SKF bearing 30302 is Selected.

2) Intermediate Shaft ($d=25 \text{ mm}$)



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:

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

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

$$Ka = 828.54 + 2026.5 = 2855.04 \text{ N}$$

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) = 38.1 kN

C_o (static load constant) = 33.5 kN

$e = 0.37$

$Y = 1.6$

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

$$FaB = FaA + Ka$$

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

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

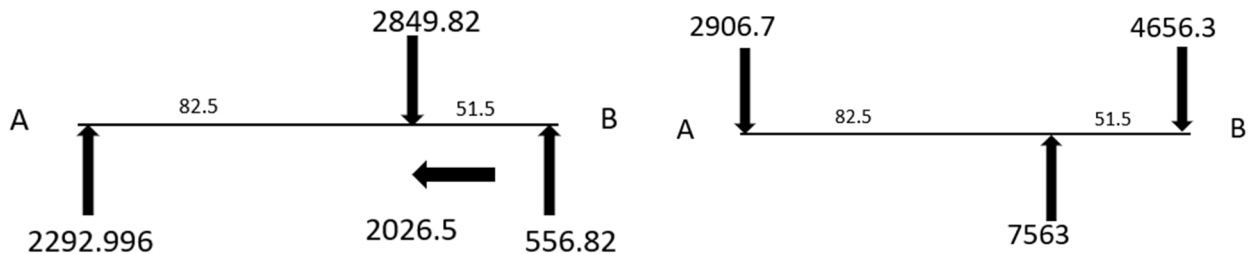
$$\text{And } FaB = 1409.28 + 2855.04 = 4264.32 \text{ N}$$

Now, to check if the bearing is suitable:

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

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:

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

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

$$Ka = 2026.5 \text{ N}$$

Let us consider bearing 32006X (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) = 43.9 kN

Co (static load constant) = 44 kN

e = 0.43

Y = 1.4

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

$$F_{aA} = F_{aB} + K_a$$

$$F_{aB} = 0.5 * \frac{F_{rB}}{Y}$$

$$F_{aB} = 0.5 * \frac{4689.5}{1.4} = 1674.82 \text{ N}$$

$$\text{And } F_{aA} = 1674.82 + 2026.5 = 3701.32 \text{ N}$$

Now, to check if the bearing is suitable:

$\frac{F_{aA}}{F_{rA}} = \frac{3701.32}{3702.26} = 0.99 > e$ $P = 0.4 * F_{rA} + Y * F_{aA}$ $P = 0.4(3702.26) + 1.4(3701.32)$ $P = 1480.904 + 5181.848$ $P = 6662.752 \text{ N}$ Consider $L_{10} = 50$ million rev and Load factor = 1.2 $C_1 = P * 50^{0.3} * 1.2$ $C_1 = 25853.89 \text{ N}$ Which is less than C_0 Hence the Bearing is Suitable	$\frac{F_{aB}}{F_{rB}} = \frac{1674.82}{4689.5} = 0.357 < e$ $P = F_{rB}$ $P = 4689.5 \text{ N}$ $C_1 = P * 50^{0.3} * 1.2$ $C_1 = 18196.95 \text{ N}$ Which is less than C_0 Hence the Bearing is Suitable
---	--

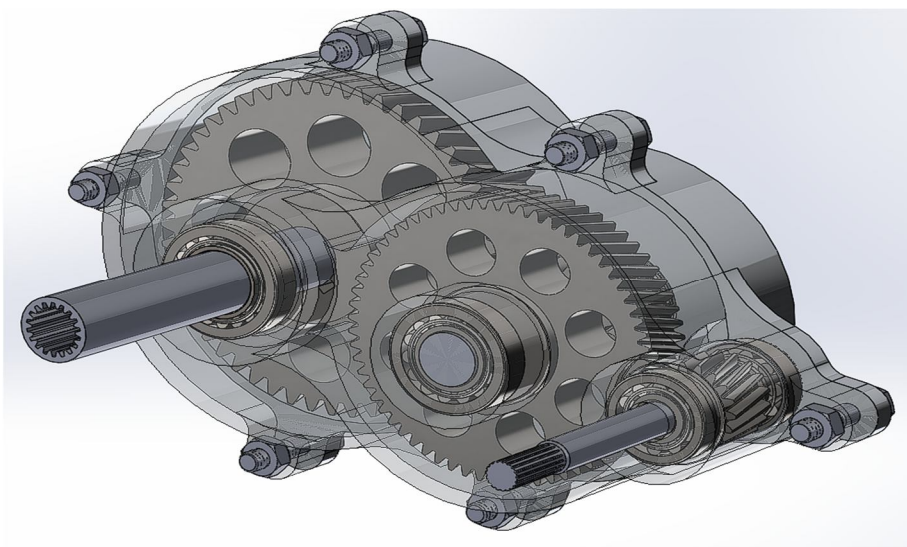
Hence SKF bearing 32006X is Selected.

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.



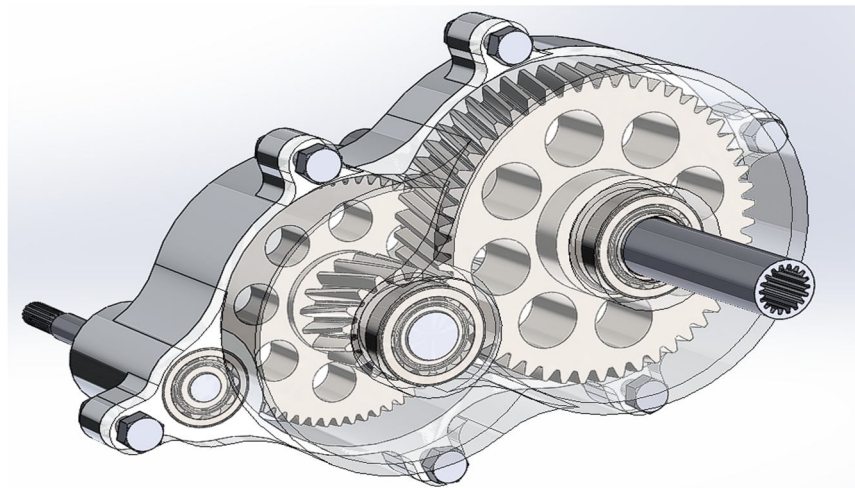


Figure 2. Overview of Gearbox

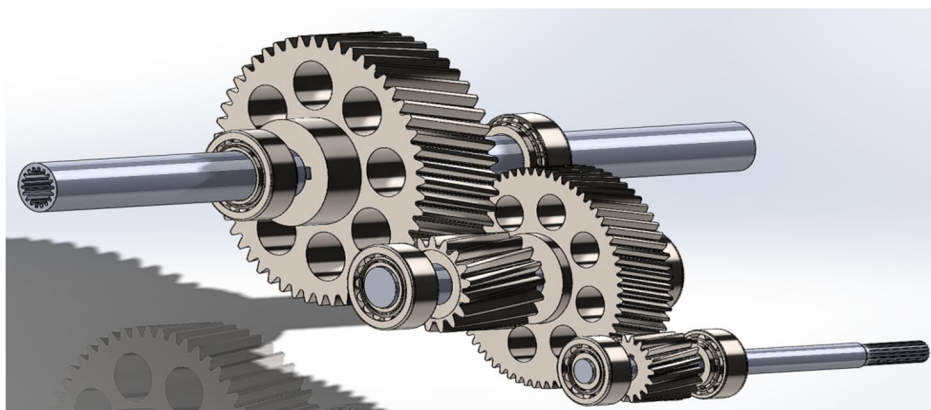


Figure 3. Helical Geartrain.

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.

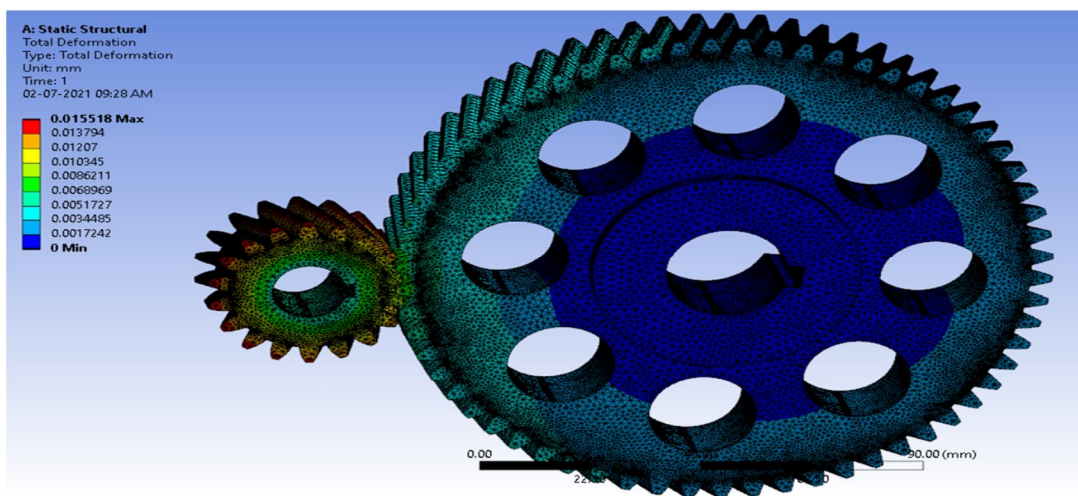


Figure 4. Stage 1 Gearset Deformation

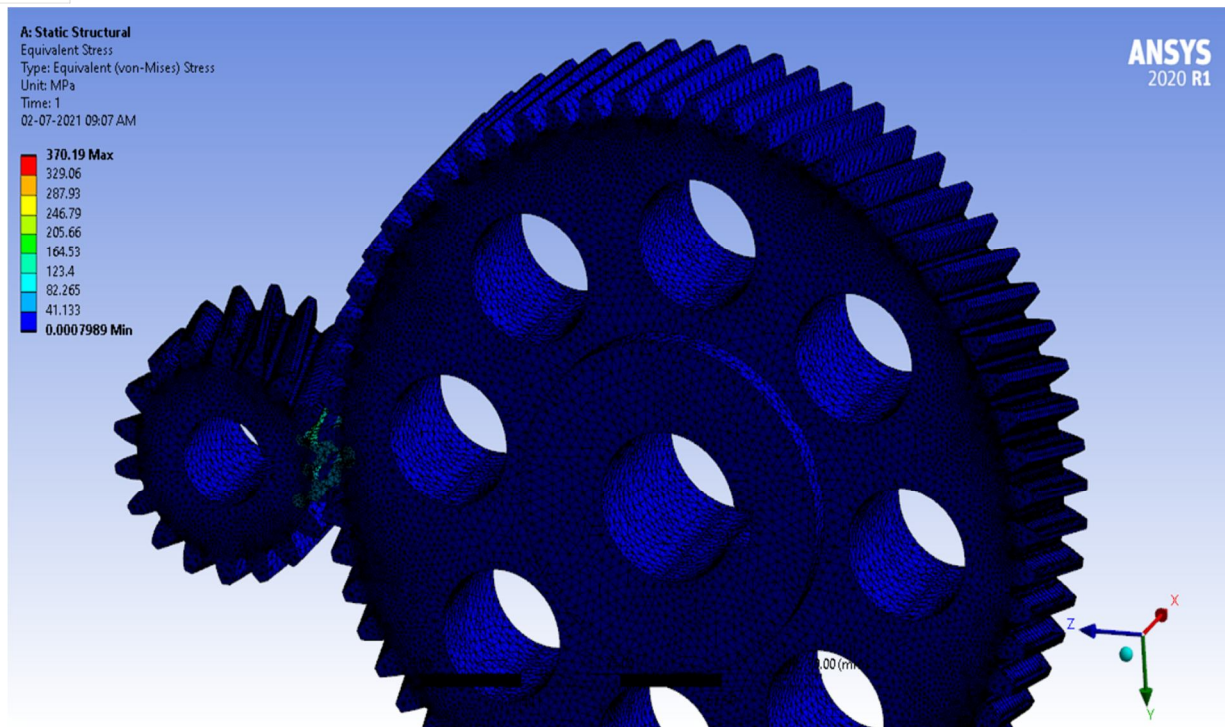


Figure 5. Stage 1 Gearset Equivalent Stress

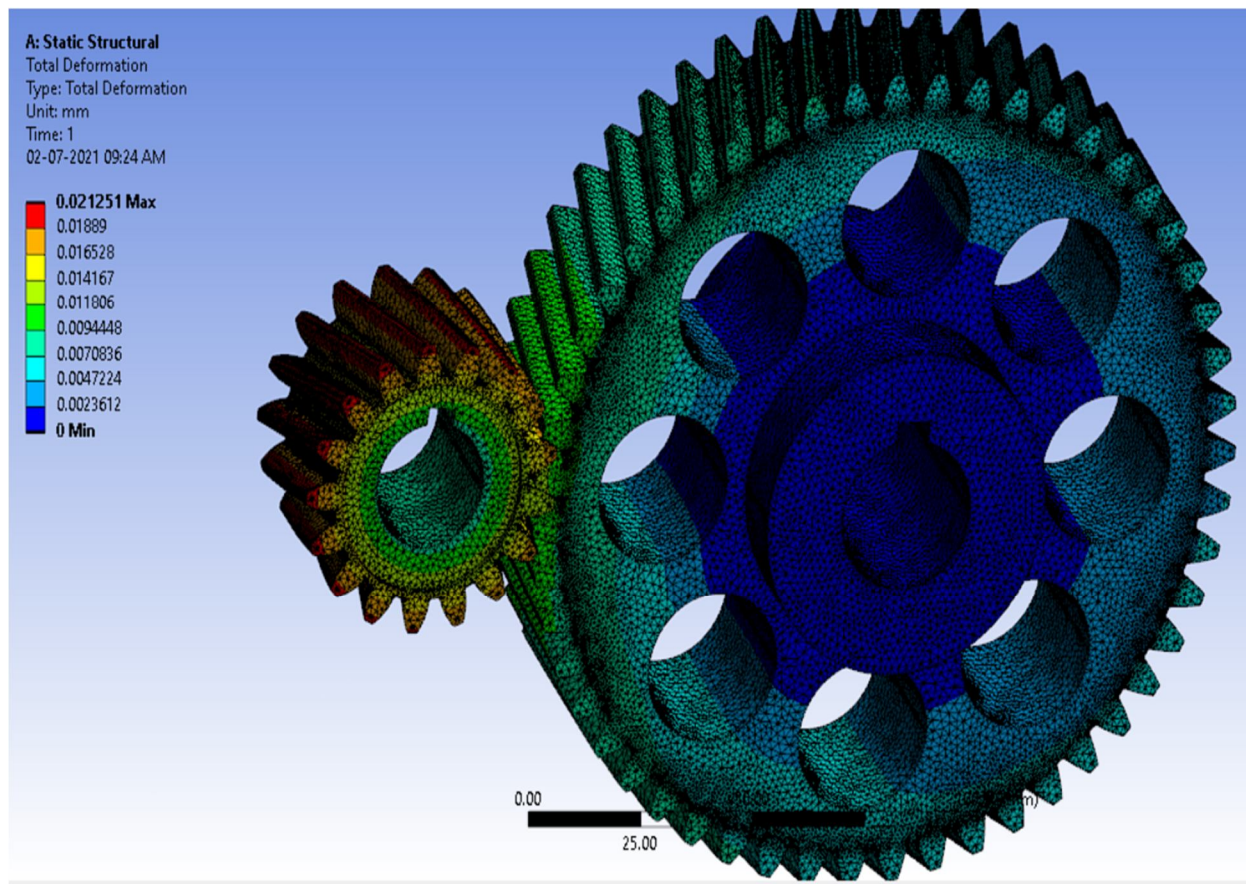


Figure 6. Stage 2 Gearset Deformation

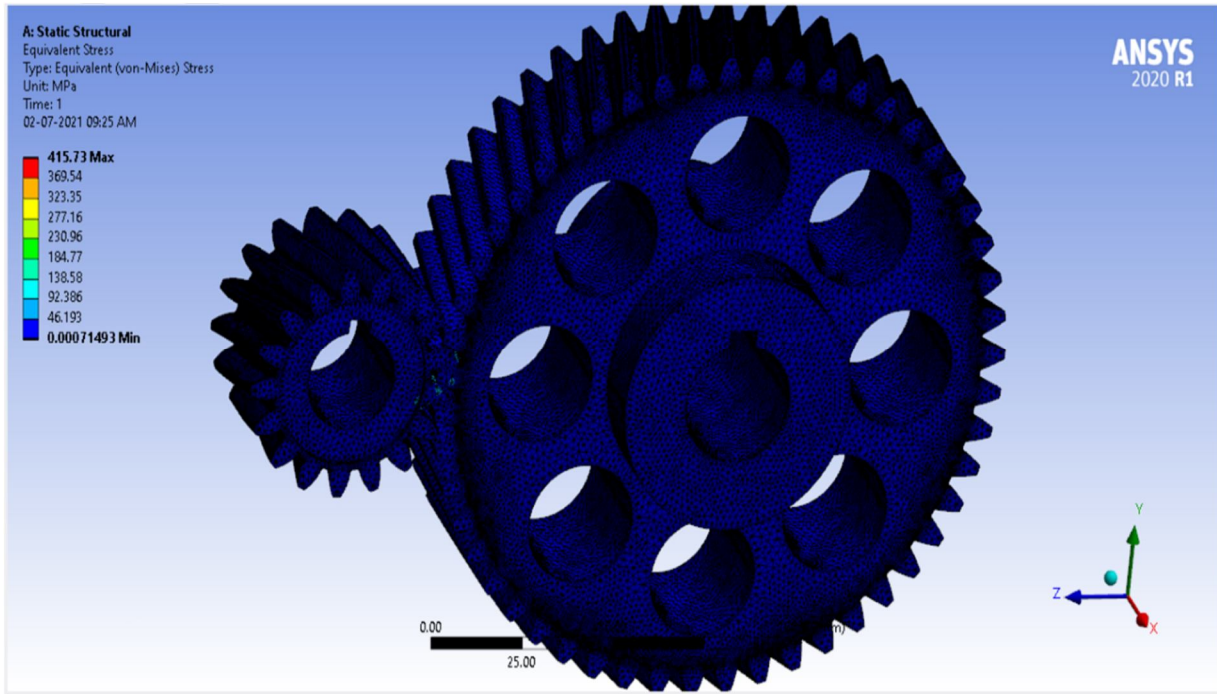


Figure 7. Stage 2 Gearset Equivalent Stress

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

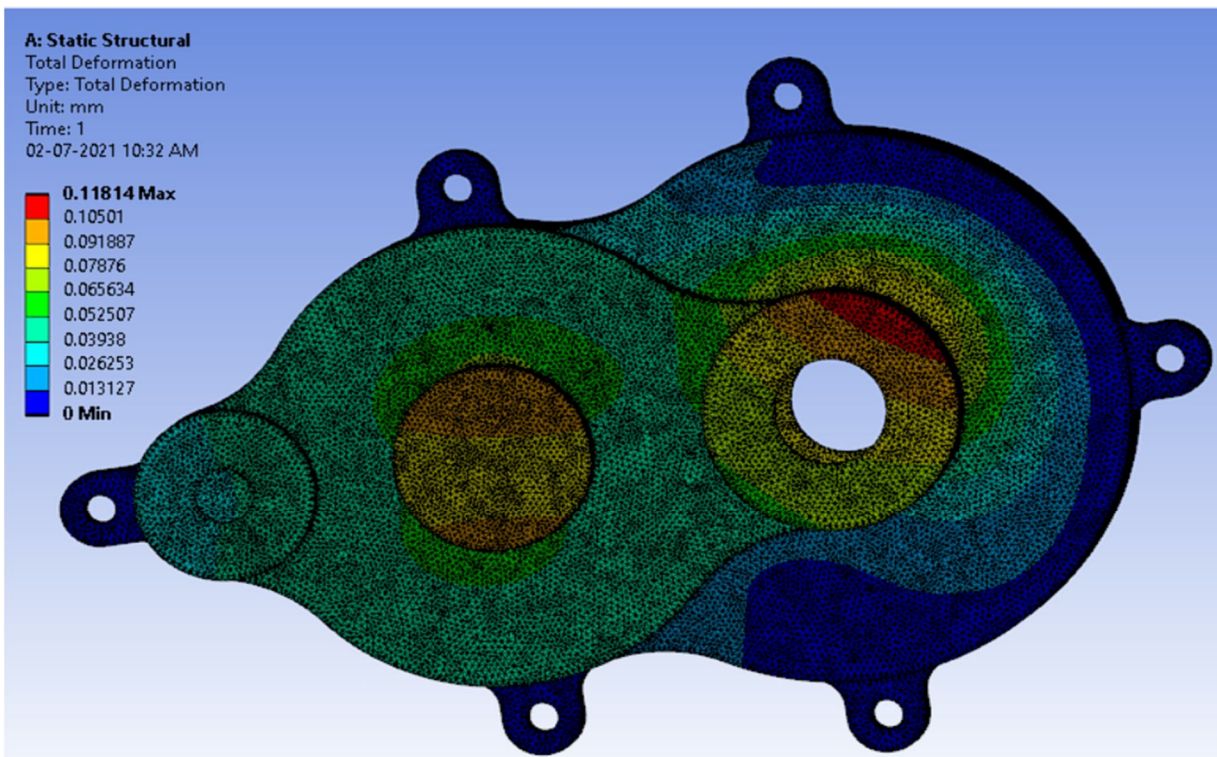


Figure 8. Casing Right Deformation

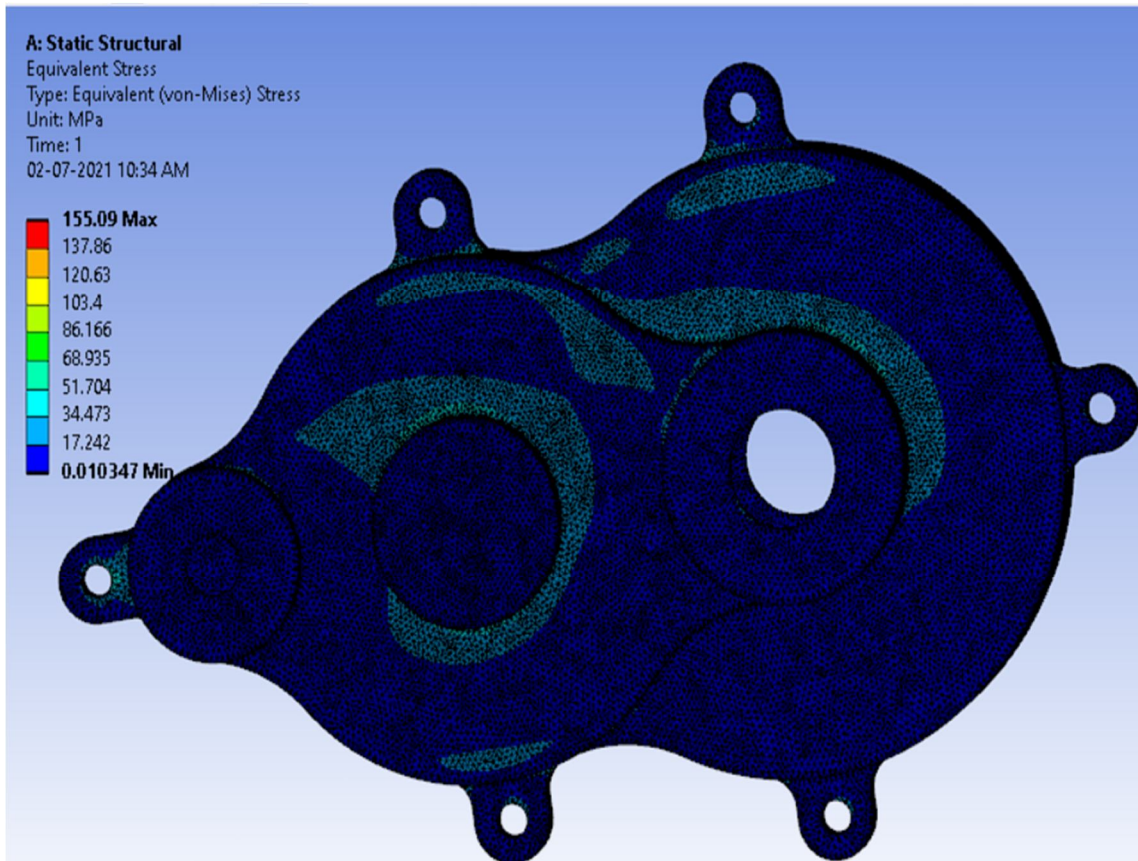


Figure 9. Casing right Equivalent Stress

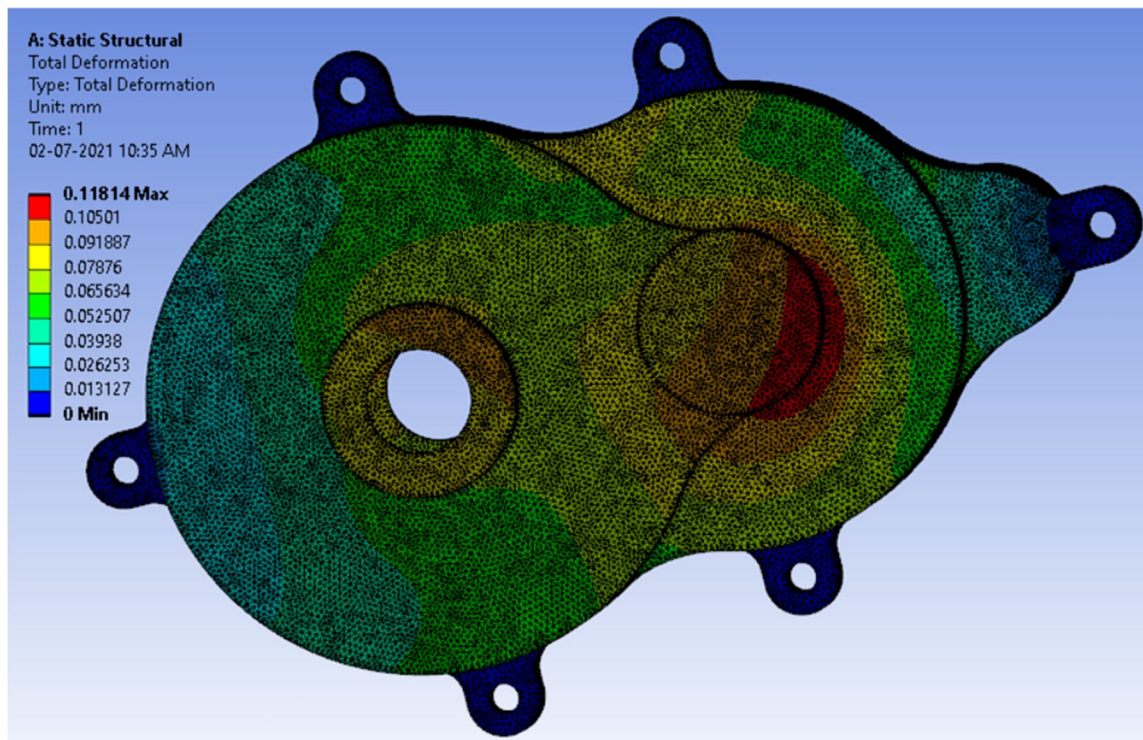


Figure 10. Casing left Deformation

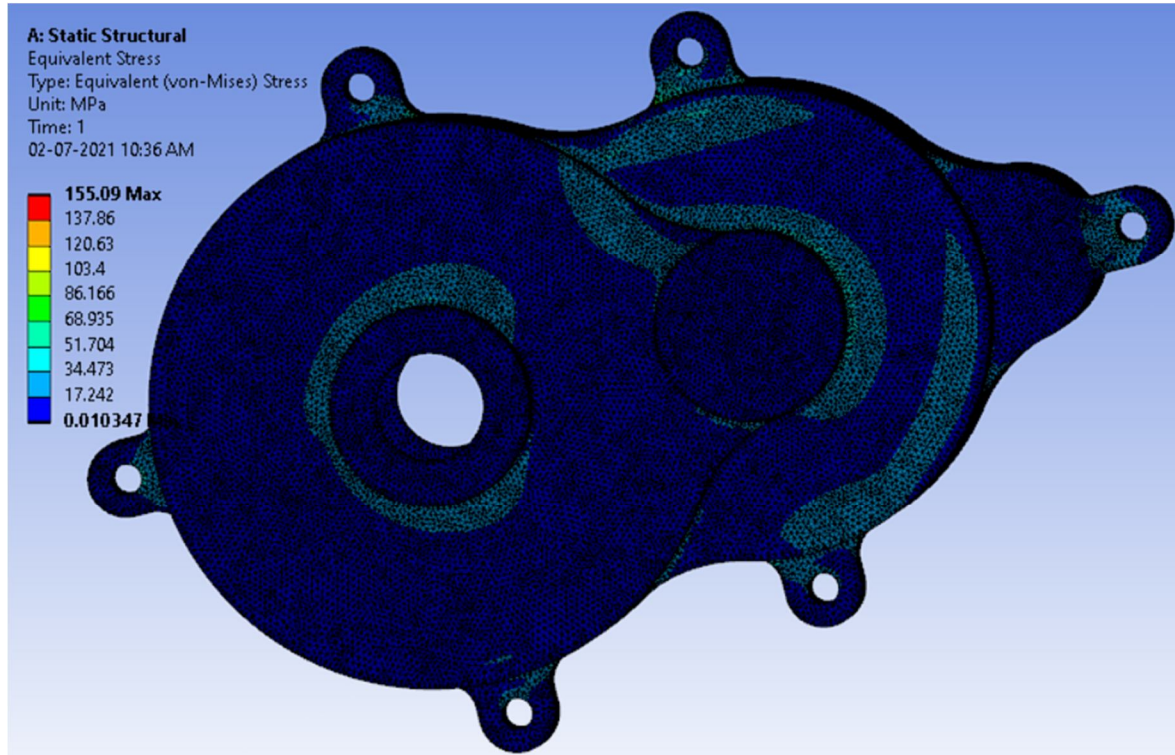


Figure 11. Casing left Equivalent Stress

COMPONENT	DEFORMATION (mm)	EQUIVALENT 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.

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