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Major Project Final Presentation
Group No. 24

Development of All-Wheel Driveline for All-Terrain Vehicles and Gradability and Acceleration Simulations using IPG CarMaker

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Project work at RVCE

Introduction-MOTIVATION

- **Market Motivation:** The global all-wheel drive (AWD) vehicle market size is projected to be worth around US$ 680 billion by 2030 expanding at a CAGR of 5.1% from 2021 to 2030.
- **Industrial Motivation:** Titanium Alloys and Aluminium alloys have emerged in recent years due to their high strength to weight ratio.
- **Research Motivation:** Titanium Alloys have high strength-to-weight ratio and high torsional rigidity, hence can be used as shaft material. Aluminum Alloys have lesser density and has good machinability, hence can be used as casing material.
- **Societal motivation:** The need of better traction on the roads which increases fuel efficiency.
- **Knowledge Motivation**: During execution of the project, students will learn AGMA standards, CATIA, SOLIDWORKS, ANSYS and IPG CarMaker.

Open Differential

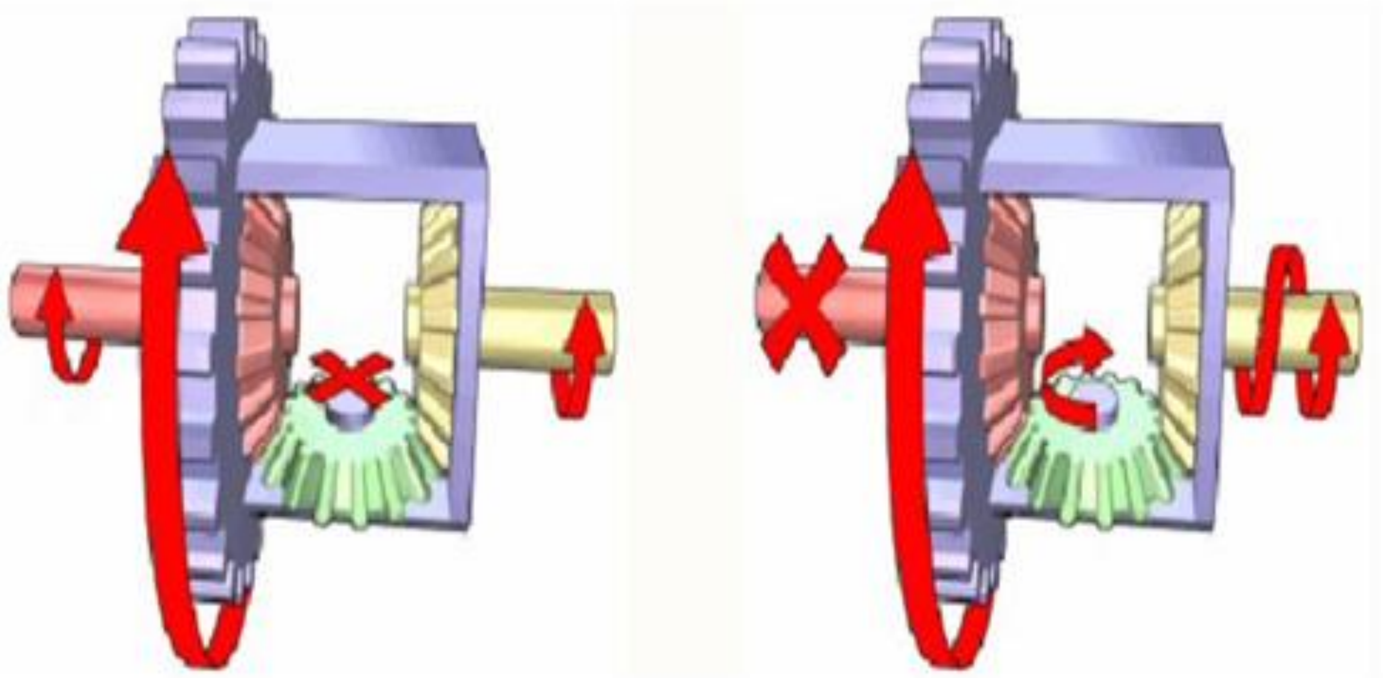


Fig 1. Basic model of open differential

- ➢A differential in its most basic form comprises two halves of an axle with a gear on each end, connected together by a third gear making up three sides of a square
- ➢This basic unit is then further augmented by a ring gear being added to the differential case that holds the basic core gears – and this ring gear allows the wheels to be powered by connecting to the drive shaft via a pinion
- ➢The left image shows the differential with both wheels turning at the same speed, while the right image illustrates how the inner gears engage when one

Continuously Variable Transmission

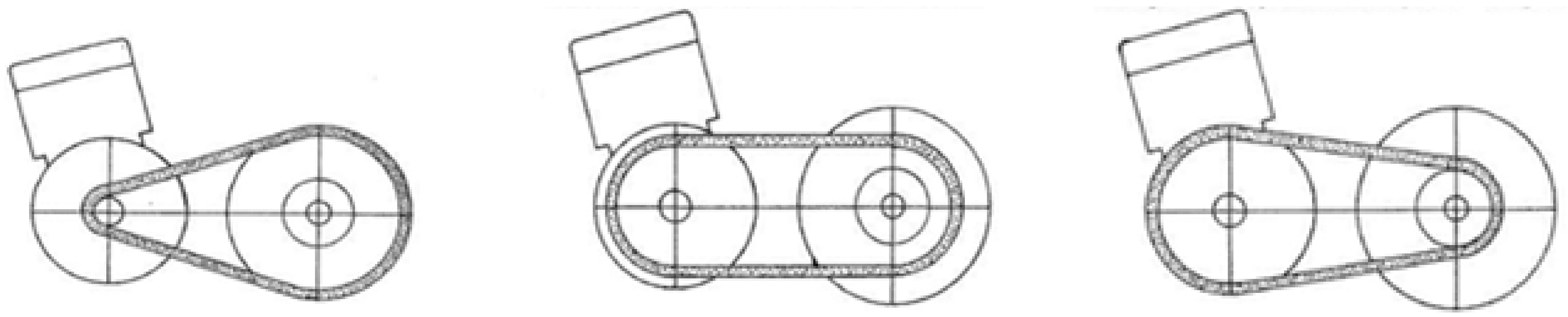


Fig 2. Belt position at low ratio (left), direct drive (center) and high ratio (right)

- A Continuously Variable Transmission (CVT), also known as a shiftless transmission, single-speed transmission, or step less transmission, is an automatic transmission that can change seamlessly through a continuous range of effective gear ratios
- The flexibility of a CVT with suitable control may allow the engine to operate at a constant RPM while the vehicle moves at varying speeds. CVTs are proven to have a better fuel economy which is achieved due to the enablement of the engine running at its most efficient RPMs

Research Objectives

Identification of Constraints: Engine Characteristics, Space, Weight, Cost, center of gravity, wheel base, track width ,suspension points and other parameters required for the calculation.

Calculation of overall reduction and dividing this reduction into different stages.

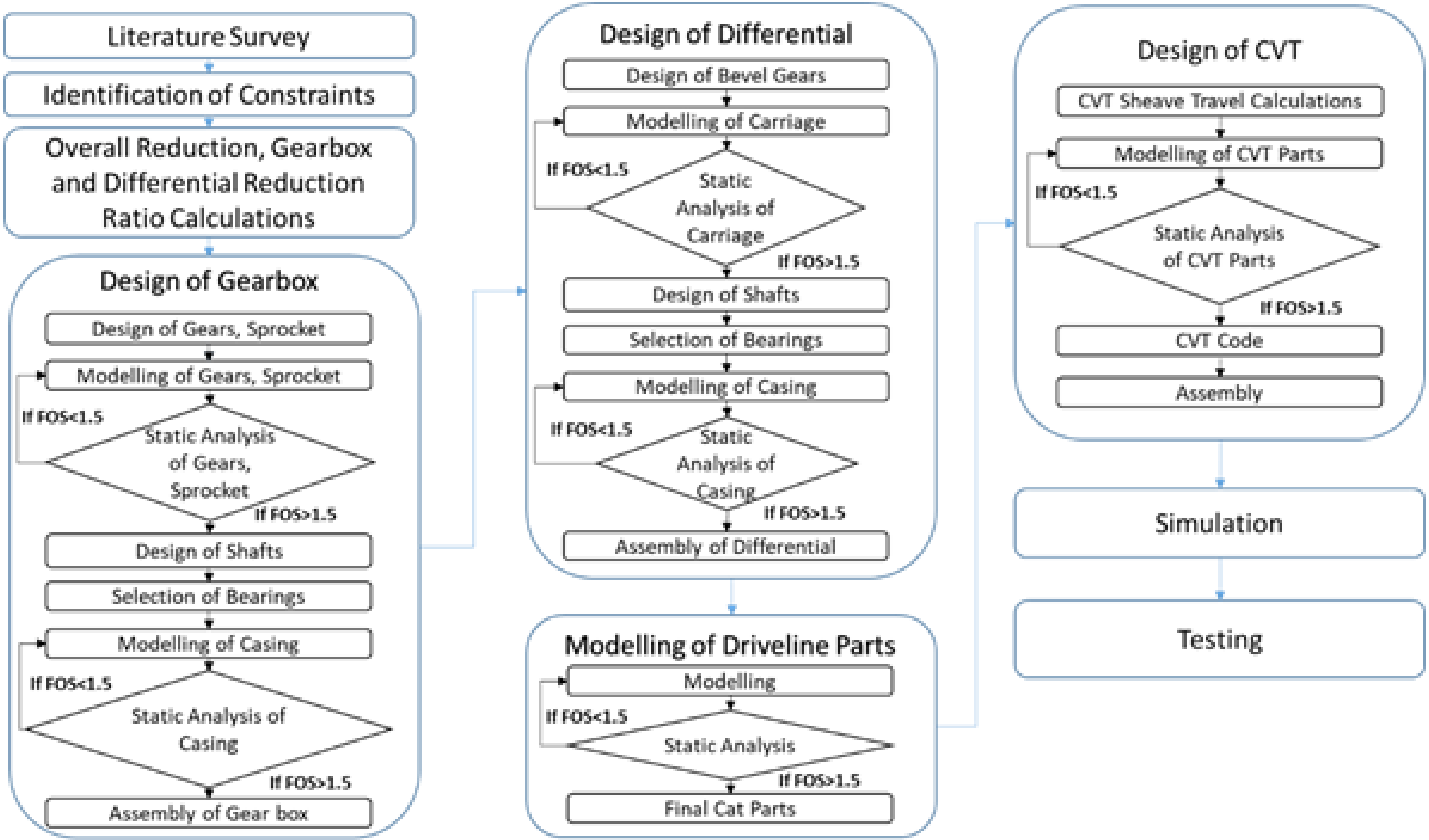
Design of gear and shaft.

Modelling and analysis of all driveline components.

Design of CVT.

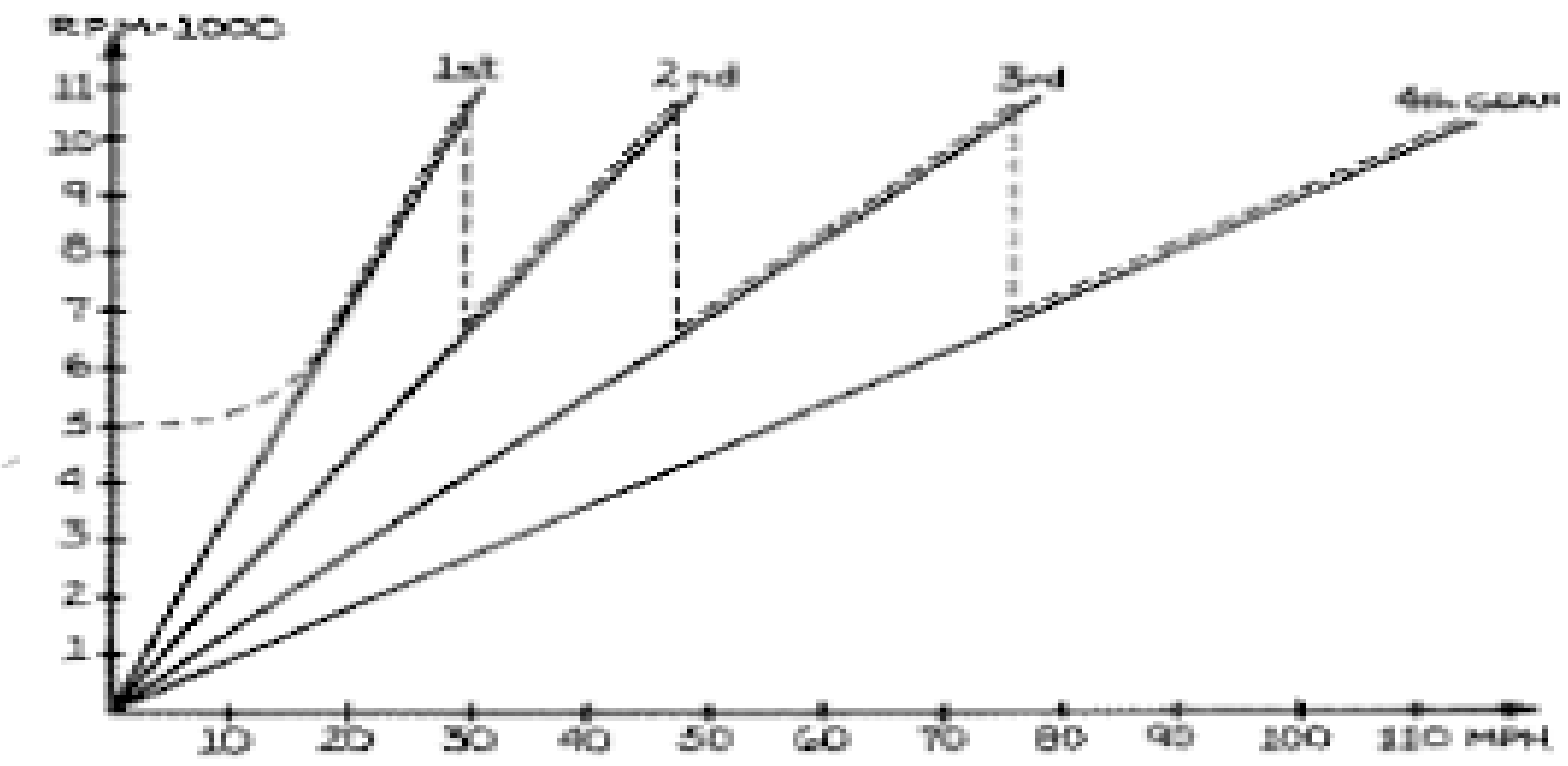
Acceleration and gradability Simulations using IPG Carmaker

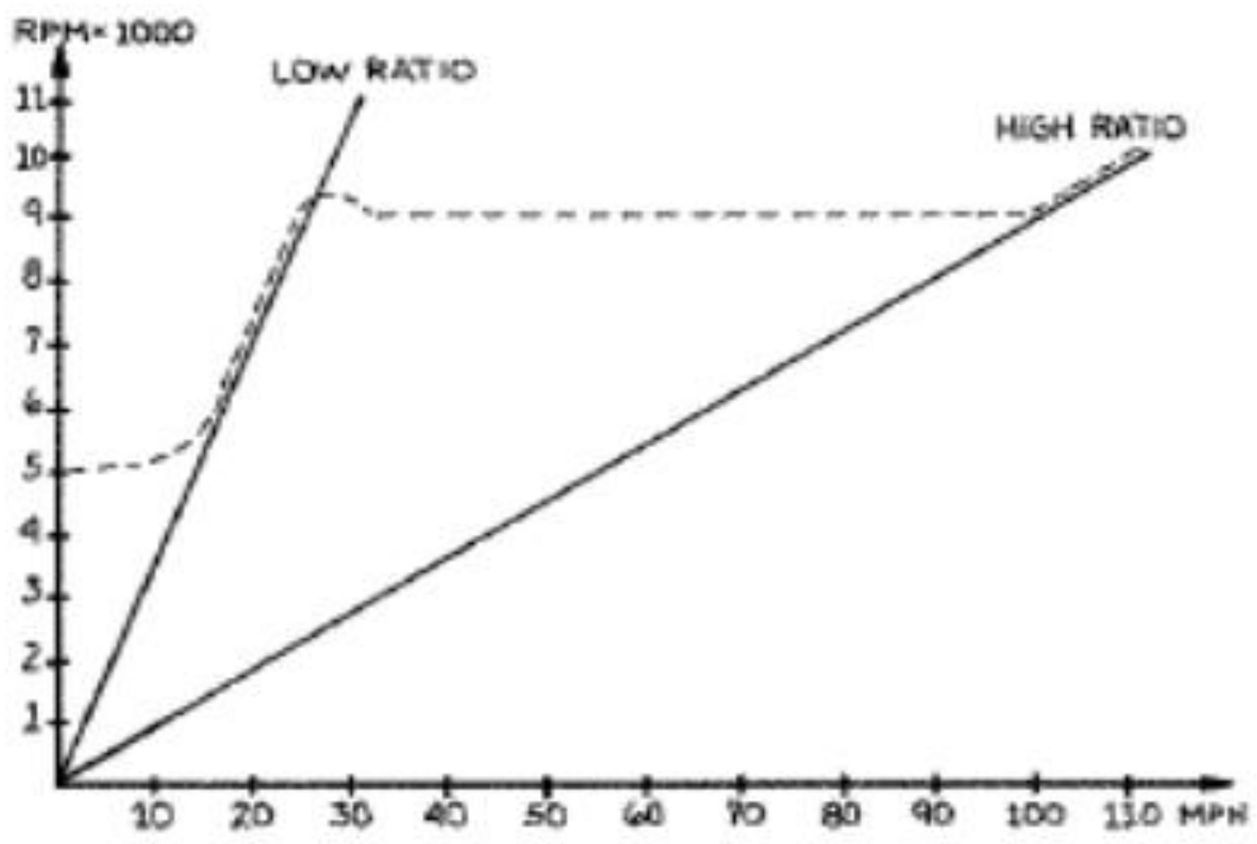
Methodology



Problem Definition

- ✓ Our project main focus is to develop all wheel driveline for All Terrain Vehicles. This novel design helps all terrain vehicle to provide power to front wheels. This assists for improved Gradeability and drawbar pull.
- ✓ Comparing performance of Manual gearbox to automatic gearbox(CVT), we found CVT will perform without losing its peak power and provide optimum torque at all condition.





Problem Definition

- ✓ Considering the commercially available Open Differential, the specification available for Cars are with turning radius ranging from 6-10m with high power rating, but for All Terrain Baja Vehicles turning radius needs to be lesser so that the vehicle maneuvers sharp corners without losing traction to the driving wheels. In our Baja Prototype the turning radius is 3m.
- ✓ Without open differential, while taking a turn the inner wheel has to slip ($v < \omega r$) and the outer wheel will drag out ($v > \omega r$). This leads to loss of power.

While taking a right turn the left wheel has to travel more distance; this means more speed to left wheel

Problem Definition

- ✓ Power to front wheel drive is usually split using transfer case which weighs around 30 kg which is more than weight of engine (25kg). So provision of power split is been provided at intermediate shaft and chain drive has been used in order to lower propeller shaft position, effectively center of gravity of vehicle is decreased.
- ✓ Custom gearbox is designed for required reduction ratios and spool setup is used to eliminate differential in rear wheel drive. Spool setup acts as locked differential since this is preferable for all terrains.
- ✓ Since there were no commercially available all wheel drive line oem components meeting specification of required power rating and required torque, we design for our required specification.

● Engine characteristics

All Baja teams across the world have to use a factory spec. Briggs and Stratton Model 19 with a specified set of rules as mentioned in section B.2.7.9 of the BAJA SAE rulebook.

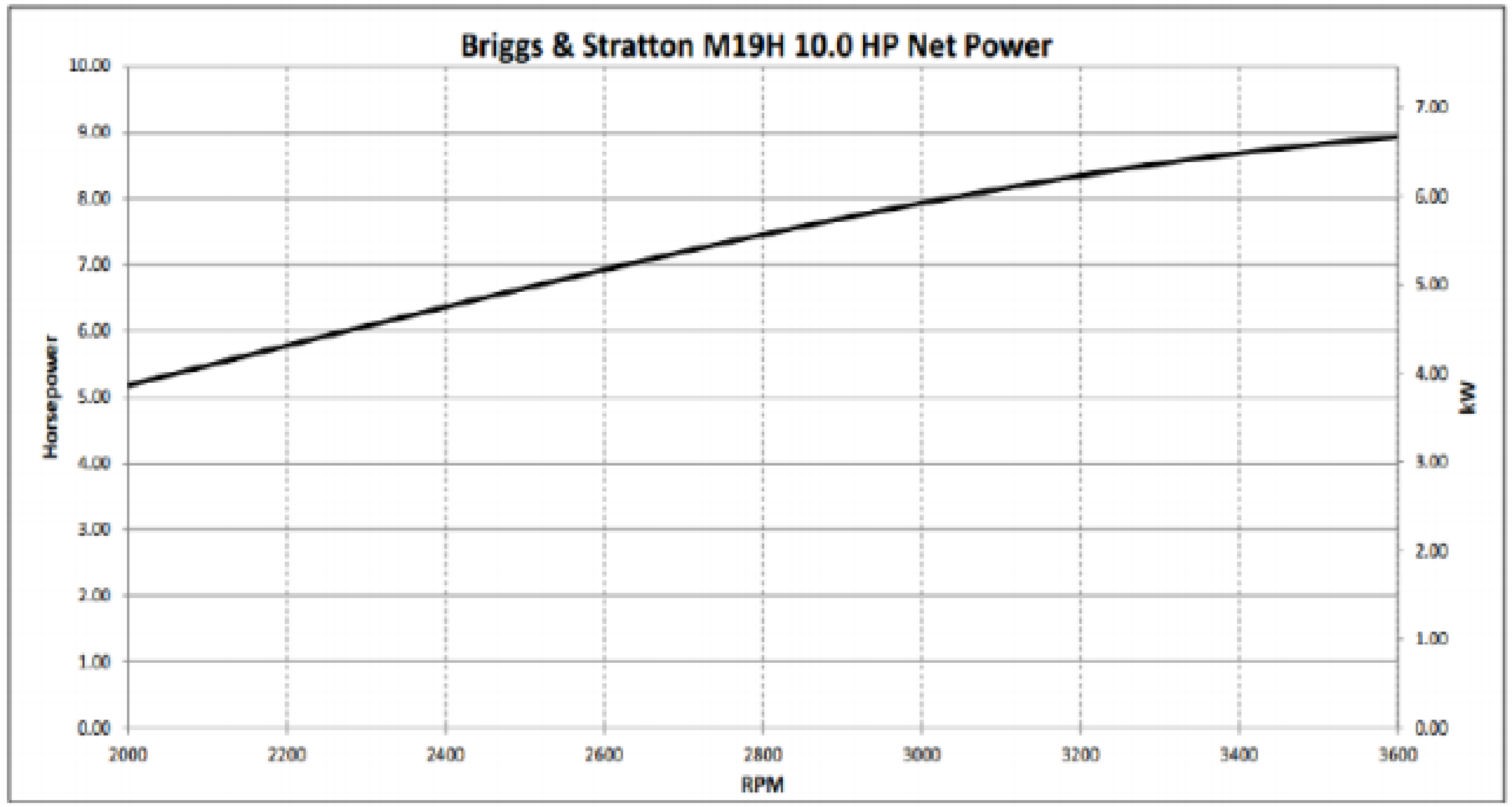


Figure 1: Model 19 Power Curve

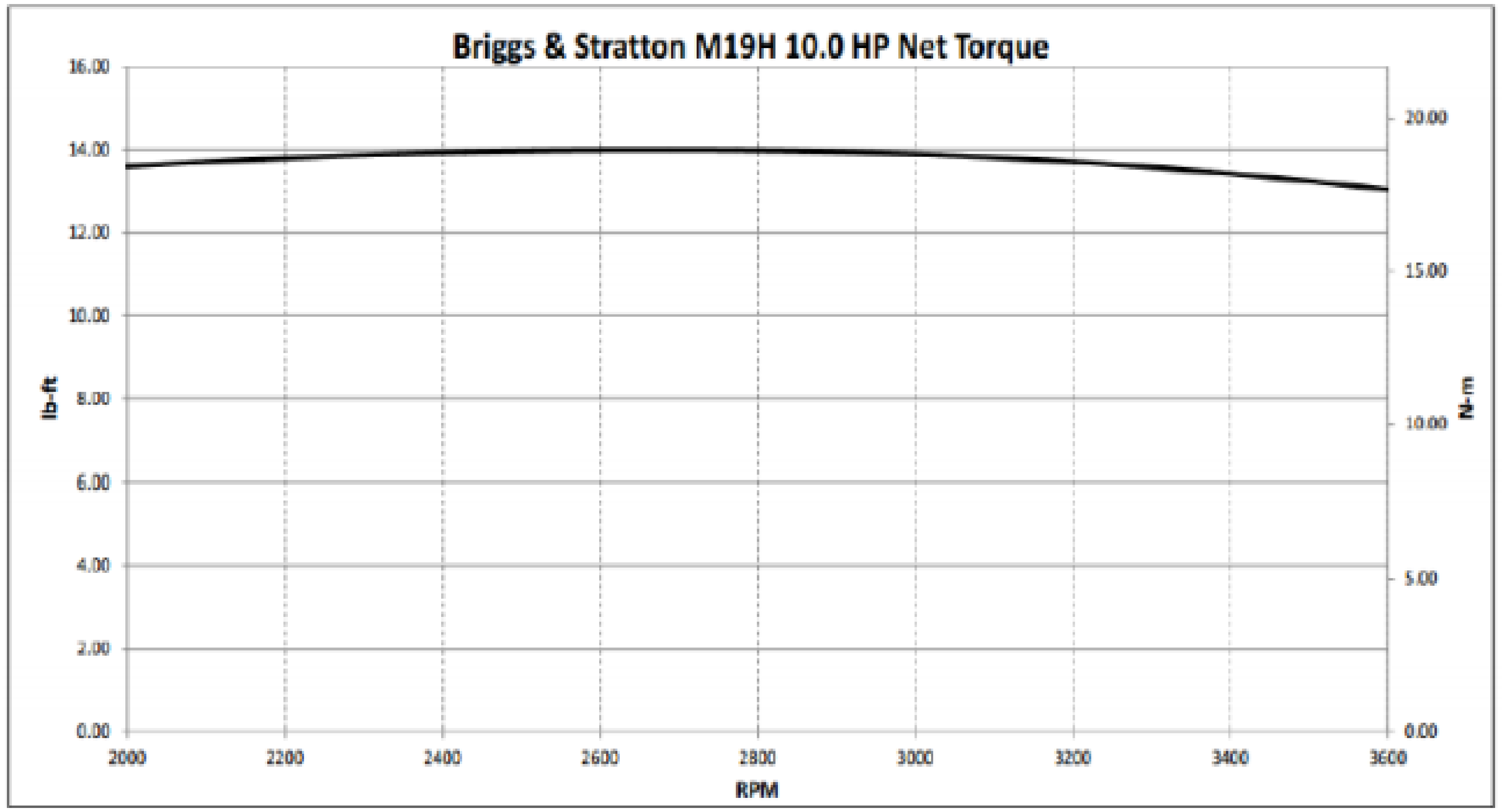


Figure 2: Model 19 Torque Curve

Parameter	Value	RPM
Maximum Torque	19.5 N-m	2600
Maximum Power	9 BHP	3800

Other reference values required for our calculation are as follows

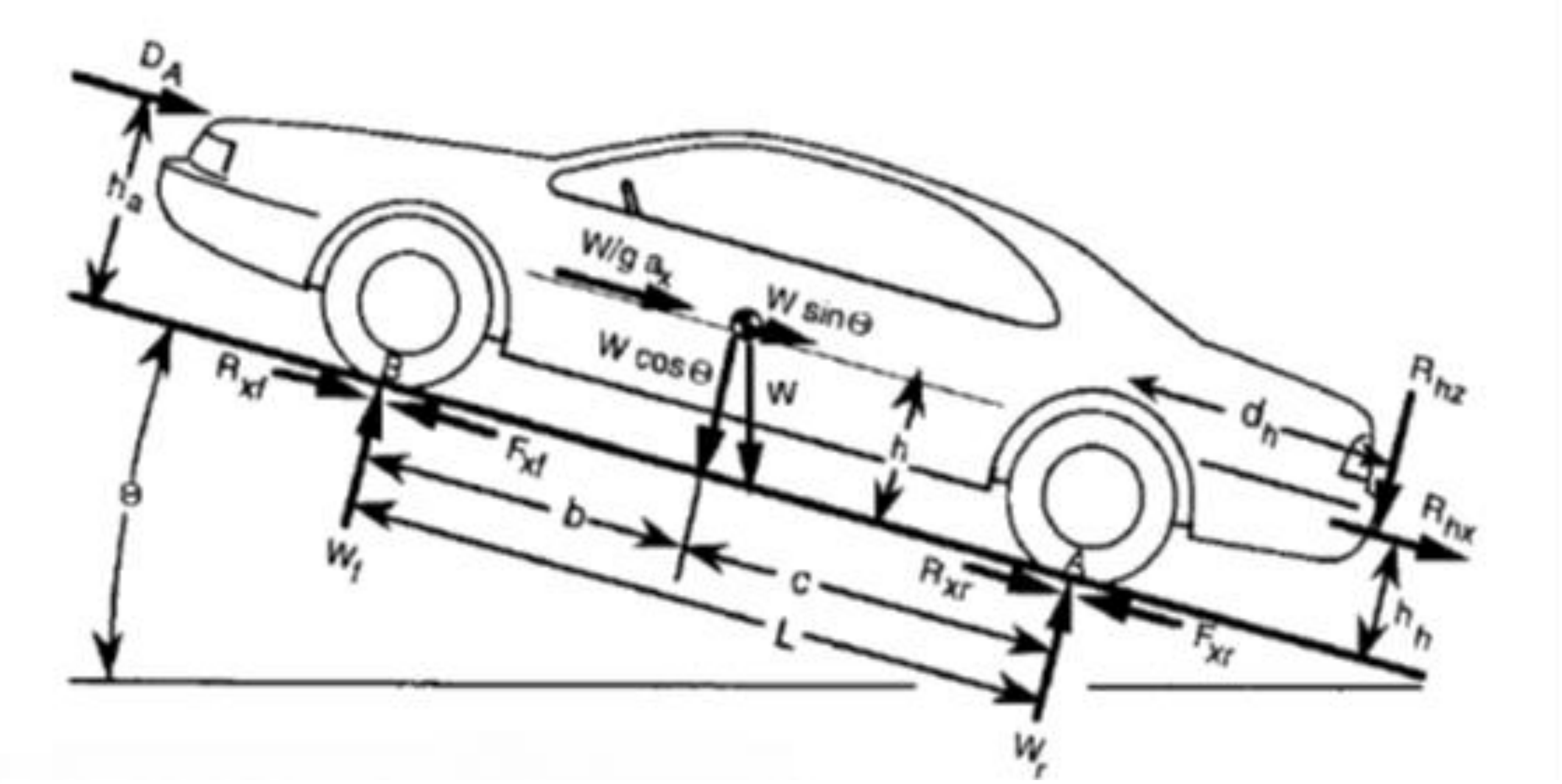
- W = Weight of Car (with driver) = 220 kg = 2158.2 N
- b = Distance of C.G from Front Axle = 0.45L = 0.594 m
- h = Distance of C.G from Ground = 0.54 m
- L = Wheelbase = 1.320 m
- Co-efficient of friction between tire and concrete = 1.0
- Co-efficient of rolling resistance between tire and concrete = 0.02
- Co-efficient of Rolling Resistance between tire and sand = 0.06
- Radius of Wheel = 0.2794 m

1. **Calculating overall reduction using gradeability.**

To calculate low ratio, we needed to find out the maximum torque needed by the car in the most demanding scenario, like the hill climb event. To overcome the hill climb, the tractive force at the wheels must be greater than all the resisting forces.

Tractive Forces-

According to Fundamentals of Vehicle Dynamics – Thomas D. Gillespie;



$$W_r = (W_b\cos\Theta + R_{hx}h_h + R_{hz}(d_h+L) + (W/g)a_x h + D_a h_a + Wh\sin\Theta)/L$$

$$W_f = (W_b\cos\Theta - R_{hx}h_h - R_{hz}(d_h+L) - (W/g)a_x h + D_a h_a - Wh\sin\Theta)/L$$

As there are no hitch forces and drag forces can be neglected at low speeds,

$F_x = \mu_{t-c} * (W_f + W_r)$

From (i) & (ii),

$(W_f + W_r) = (W*c*\cos\Theta + W*b*\cos\Theta)/L$

$\therefore F_x = W\cos\Theta = 2158.2\cos\Theta$ **-(iii)**

Resistance Forces-

$F_r = W\sin\Theta + C_{r-rt-c} * W\cos\Theta$

$\therefore F_r = 2158.2\sin\Theta + 43.164\cos\Theta$ **-(iv)**

Since, $F_x = F_r$

Solving (iii) & (iv),

$\Theta = 44.42°$

$\therefore F_x = 1541.44$ N

Now since,

$F_x * R_w = T_e * N_{low} * \eta$ **-(v)**

@N = 1900rpm;

$T_e = 1.0627*10^{-9}*N^3 – 1.0008* 10^{-5}*N^2 + 0.0302*N – 10.231$

$\therefore T_e = 18.3092$ Nm

$\eta = 0.816$

$R_w = 0.2794$m (Radius of wheel)

N_{low} =28.826

2. Calculating overall reduction using Topple angle.

For car to topple, the W_f force at front will be zero,

Taking moment about A,

$W_f*l - W*\cos(\Theta)*c + W*\sin(\Theta)*h=0$

Since, $W_f=0$

Therefore, $\tan(\Theta)=c/h$

$\Theta=53.35$

$Fx=w*\cos(\Theta)=1288.28$ N

$F_x * R_w = T_e * N_{low} * \eta$

N_{low} =24.092

3. Calculating overall reduction using maximum torque condition

Maximum torque that can be given to the wheels,

$F_x = \mu_{t-c}*W = 0.85*2158.2 = 1876.16$ N

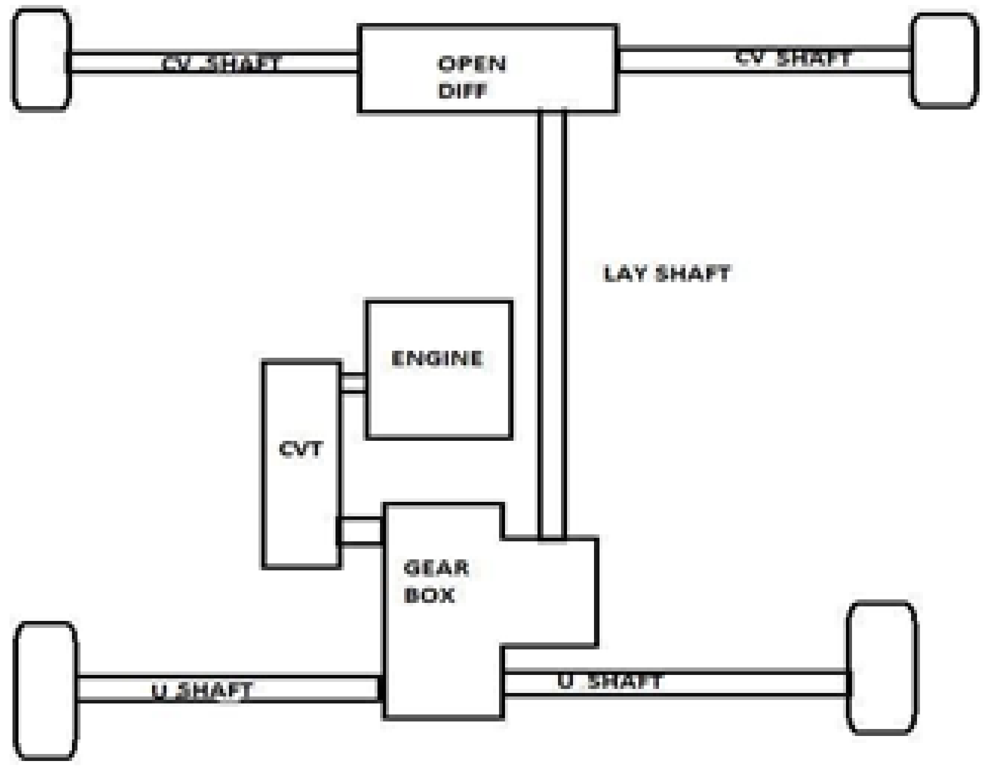
(For maximum torque conditions, $\mu_{t-c}= 0.85$)

Using (v),

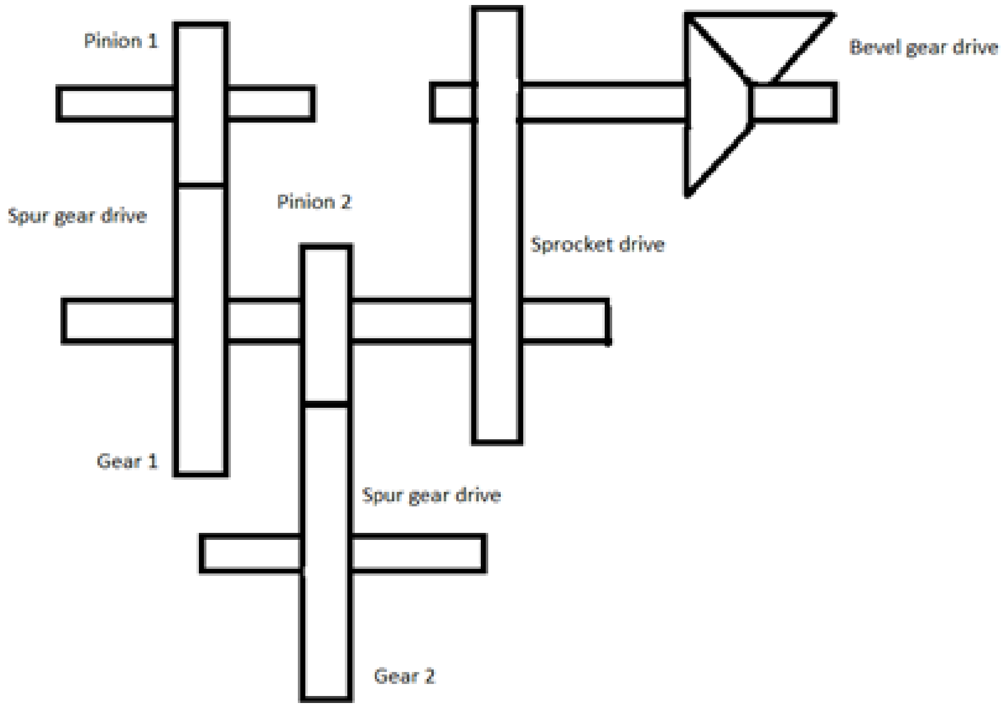
N_{low} = 34.3066

Therefore, overall reduction, **N_{low}=34.3066** (maximum value of above method)

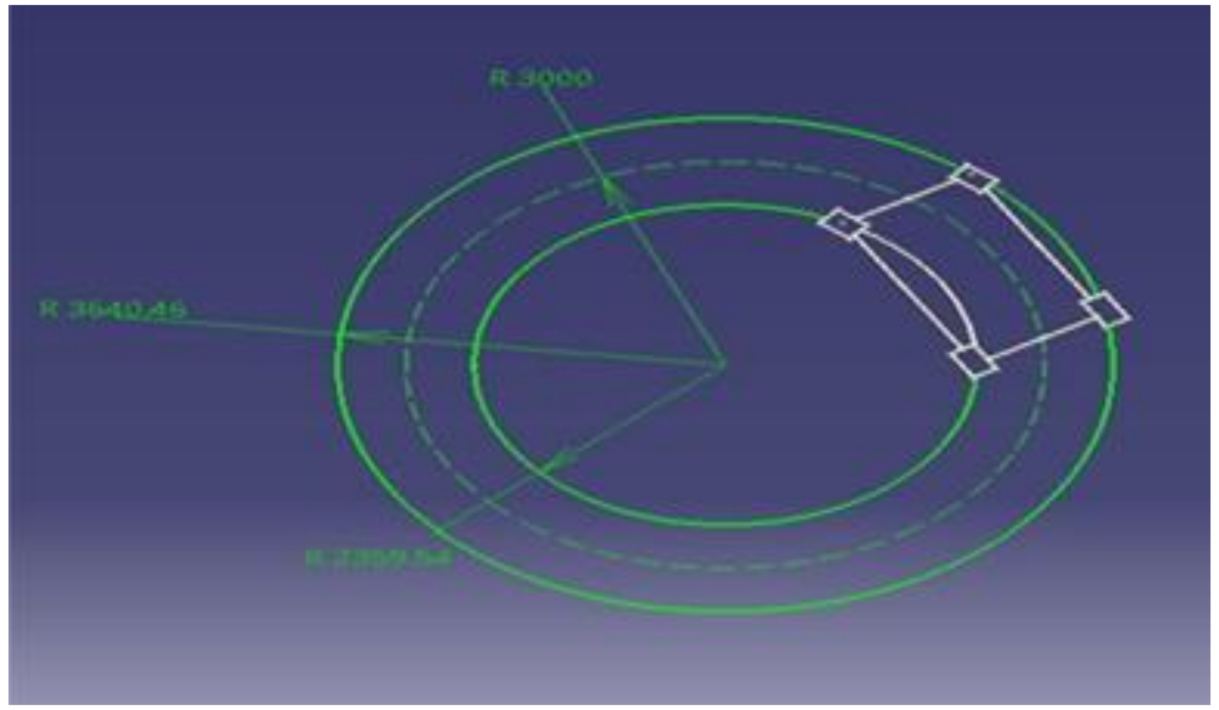
Block diagram of driveline of the car



Block diagram of the Gearbox



Calculation of Side and Spider Gear Ratio



Turning radius=3000mm Front track width=1280.92mm Wheel radius=275.4mm

Circumference of wheel=2*π*275.4 …….(i)

Circumference of inner wheel trajectory=2*π*2359.54 ……(ii)

Circumference of outer wheel trajectory=2*π*3640.46 …….(iii)

Ratio (ii /i) =8.5676 ……..(iv)

This ratio tells us that the inner wheel should rotate 8.5676 times to complete one rotation.

Ratio (iii) / (i) =13.2188 ……..(v)

This ratio tells us that the outer wheel should rotate 13.2188 times to complete one rotation.

Ratio (v/ iv) =**1.542**

Calculation of CVT, gearbox, and differential ratios.

Parameter	Reduction ratio
Overall ratio	34.3
CVT ratio	4
1st stage reduction	3
2nd stage reduction	2.83
Bevel reduction	1
Sprocket reduction	1
Differential reduction	2.83
Side and spider reduction	1.52

PUGH MATRIX

Grading Parameter	Weightage	EN24 AISI 4340	EN36 AISI 8620	Aluminum 7075	Titanium 6Al-4V	AISI 9310
Bending Strength	5	2	4	1	5	5
Contact Strength	5	4	5	1	2	5
Weight	4	3	3	5	4	3
Fabrication	3	3	2	5	4	1
Availability	3	5	3	4	2	1
Total	100	66	72	57	72	68

Titanium 6Al-4V was chosen as the shaft material.

PUGH MATRIX

Grading Parameter	Weightage	EN24 (AISI 4340)	EN36 (AISI 8620)	Aluminum 7075	Titanium 6Al-4V	AISI 9310
Bending Strength	5	2	4	1	3	5
Contact Strength	5	4	5	1	2	5
Weight	4	3	3	5	4	3
Fabrication	3	3	2	5	3	1
Availability	3	5	3	4	2	1
Total	100	66	72	57	56	68

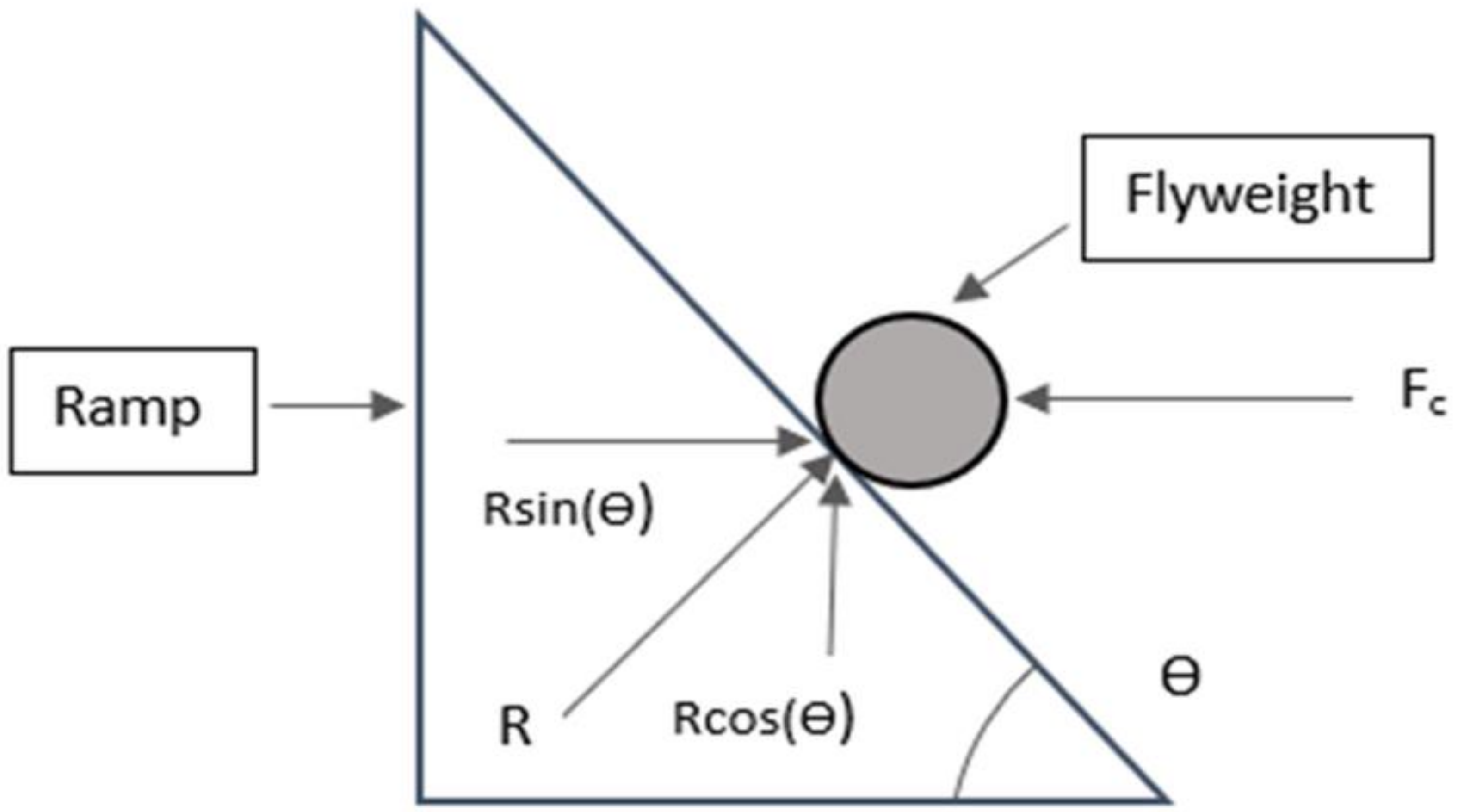
EN36 (AISI 8620) steel was chosen as the gear material.

PUGH MATRIX

Parameter	Weightage	EN24 (AISI 4340)	EN36 (AISI 8620)	Aluminum 7075	Titanium 6Al-4V
Torsional Strength	5	4	5	3	5
Fatigue Strength	5	4	5	1	3
Weight	4	3	3	5	4
Cost	3	5	3	1	1
Availability	3	5	2	4	4
Total	**100**	**82**	**74**	**61**	**71**

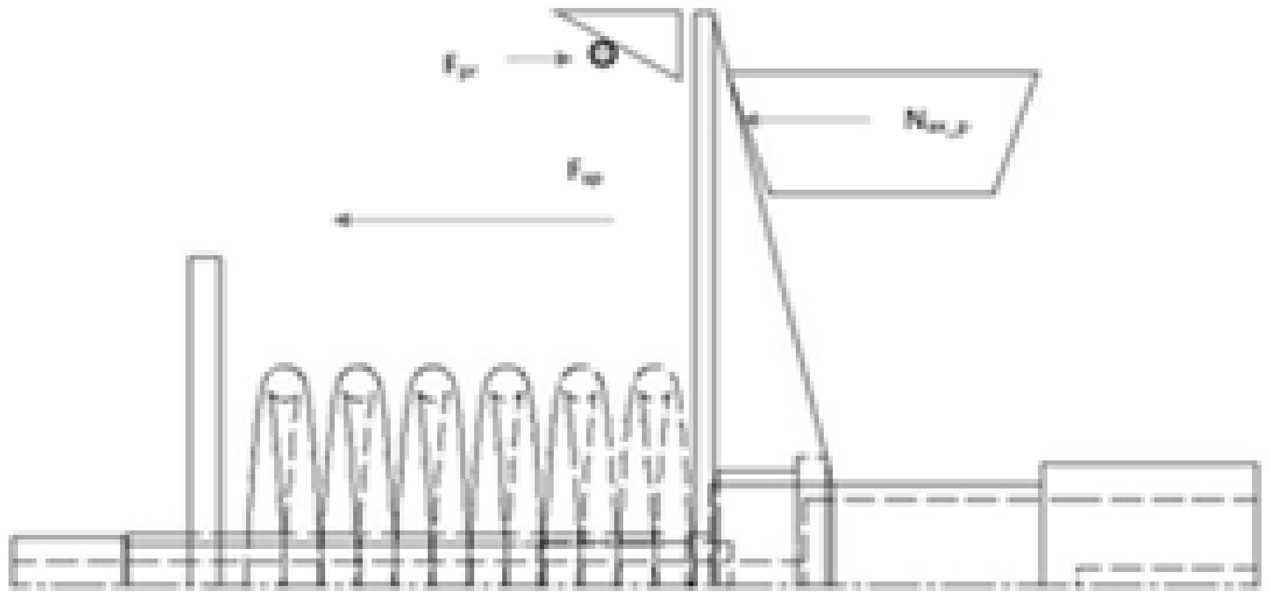
EN24 (AISI 4340) steel was chosen as the gear material.

Effect of Centrifugal Force on Ramp Profile



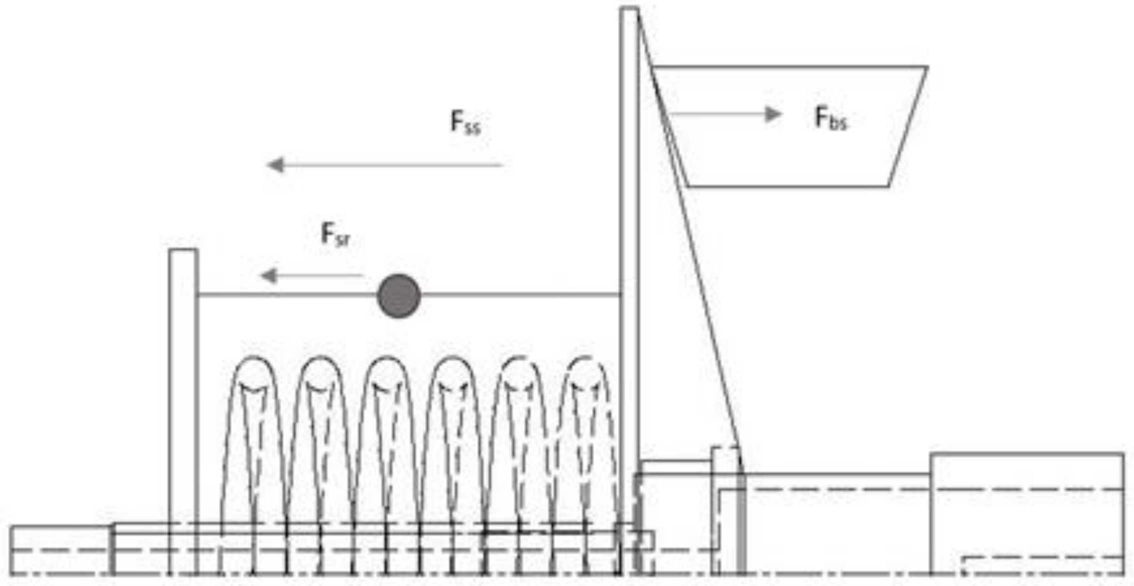
The figure depicts effect of centrifugal force on a ramp profile. The CVT model is synonymous with a tire and road model. If the tractive force is equal to the frictional force between the tire and the wheel, the wheel does not slip. Similarly, if the Side Force provided by the ramp assembly is equal to frictional force between belt and sheave, there is no belt slip. For this, we must analyse the different forces acting in the primary pulley.

FBD of Primary Pulley



$$F_{pr} - F_{sp} = N_{ax\_p}$$

FBD of Secondary Pulley



$$F_{sr} + F_{ss} = F_{bs}$$

CVT Design

- After selecting low and high ratio, we will now select standard belt available in market and also considering the load carried by it. The available belt and their parameters are show in below excel sheet.

Belt Parameters

Belt	Thickness	Included Angle	Radian	Top Width	Bottom Width	Pitch Length	Pitch Width	Pitch Thickness from top	Pitch Thickness from bottom
UA 459	16.1	28	0.488692191	29.9	21.9	822.3	27.5	4.81293712	11.28706288
UA 416	14.7	28	0.488692191	30	22.7	825	27.4	5.214015214	9.485984786
UA 400	13	28	0.488692191	29.8	23.3	806.7	27.4	4.81293712	8.18706288
UHQ 400 real	12.1	27.2	0.474729557	28.6	23.86	806.7	27.26	2.769448066	9.330551934
UHQ 400	12.1	28	0.488692191	29.8	23.7	806.7	27.4	4.81293712	7.28706288

- Now to select the best belt from the available belt, we fix primary post diameter and center to center distance according to our line diagram. Then we find the deviation in the belt length. The belt with minimum deviation is selected i.e., UHQ 400. Secondary sheave diameter is also determined in this excel sheet.

Selection of belt

Primary Post Dia	Primary Pitch Dia	Secondary Sheave Dia	Secondary Pitch Dia	C-C	Angle of Wrap Primary	Angle of Wrap Secondary	Pitch Length Needed	Deviation in belt length
30	52.57412576	219.9223773	210.296503	218.086	2.401613059	3.881572249	877.6048908	55.3048908
30	48.97196957	206.3159087	195.8878783	218.086	2.45449594	3.828689367	845.5398082	20.5398082
30	46.37412576	195.1223773	185.496503	218.086	2.492324088	3.790861219	822.5809212	15.88092122
30	48.66110387	200.1833116	194.6444155	218.086	2.459035773	3.824149534	842.7851448	36.08514478
30	44.57412576	187.9223773	178.296503	218.086	2.518393124	3.764792183	806.7547856	0.054785626

CVT Design

- When CVT is in high ratio, there are chances of contact between primary sheave, to avoid this contact we will calculate the maximum allowable primary sheave diameter and check with the calculated primary sheave diameter.
- When CVT is in low ratio, there are chances of contact between secondary sheave. Here we will calculate minimum secondary post diameter and check with the calculated secondary post diameter.
- Since we know primary post diameter and secondary sheave diameter, we can calculate primary sheave and secondary post diameter. Since both the diameter are under allowable condition, we can proceed to next step.

To avoid contact between primary sheaves		
Primary Sheave Angle	Max Primary Sheave Diameter	Corresp. Secondary Post Dia
26	159.5111286	73.65220757
26	159.9442762	77.01746023
26	159.0779811	79.57412681
26	153.88021	76.57401964
26	159.0779811	81.37412681

To avoid contact between secondary sheaves		
Secondary Sheave Angle	Min Secondary Post Dia	Corresp. Primary Sheave Dia
27	95.37991415	188.8600938
27	81.35691562	162.8803708
27	70.99644413	142.3877188
27	81.05573818	157.0612211
27	63.79644413	128.7119856

Primary Sheave and Secondary Post Diameter						
a	b	c	Primary Sheave Pitch Dia	f(AD)	Primary Sheave Dia	Secondary Post Dia
0.128	2248.030488	-336836.444	148.5792756	-6.40284E-10	158.2051499	72.81376919
0.128	2248.030488	-339191.7728	149.6095137	-1.53214E-06	160.0375441	77.07733819
0.128	2248.030488	-323227.8776	142.6244608	0.00085222	152.250335	75.19077806
0.128	2248.030488	-323227.8776	142.6244608	0.00085222	148.1633569	72.90379995
0.128	2248.030488	-323227.8776	142.6244608	0.00085222	152.250335	76.99077806

- The final dimension for CVT are listed below table.

Sheave and Post Dimensions					
Type	Primary Post Dia	Secondary Sheave Dia	Secondary Post Dia	Primary Sheave Dia	High Ratio Attained
For Calculated High Ratio	30	187.922	76.99	152.25	0.642
Final Dimensions	30	190	72	162	NA

CVT Dimensions

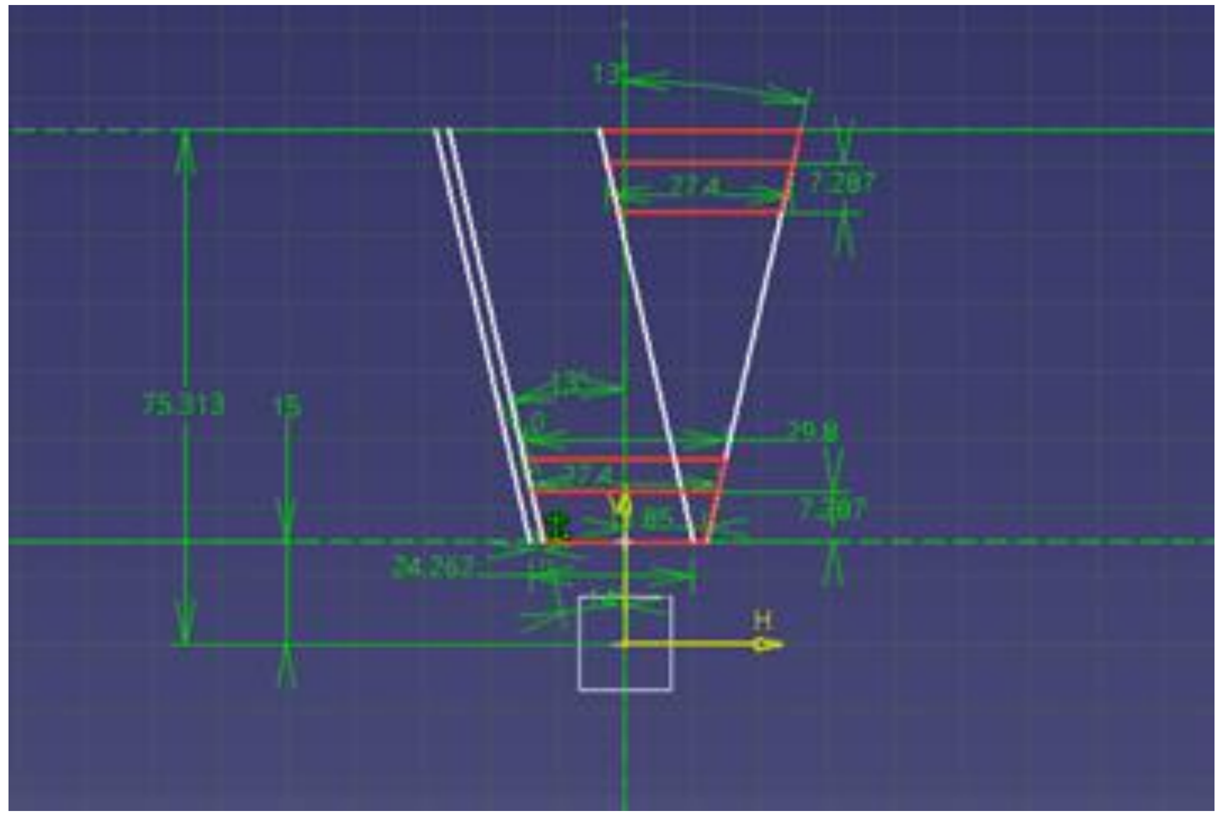
Sheave Angles and Belt Angle

Component	Angle (°)
Belt (α)	28
Secondary Pulley (β)	27
Primary Pulley (Ω)	26

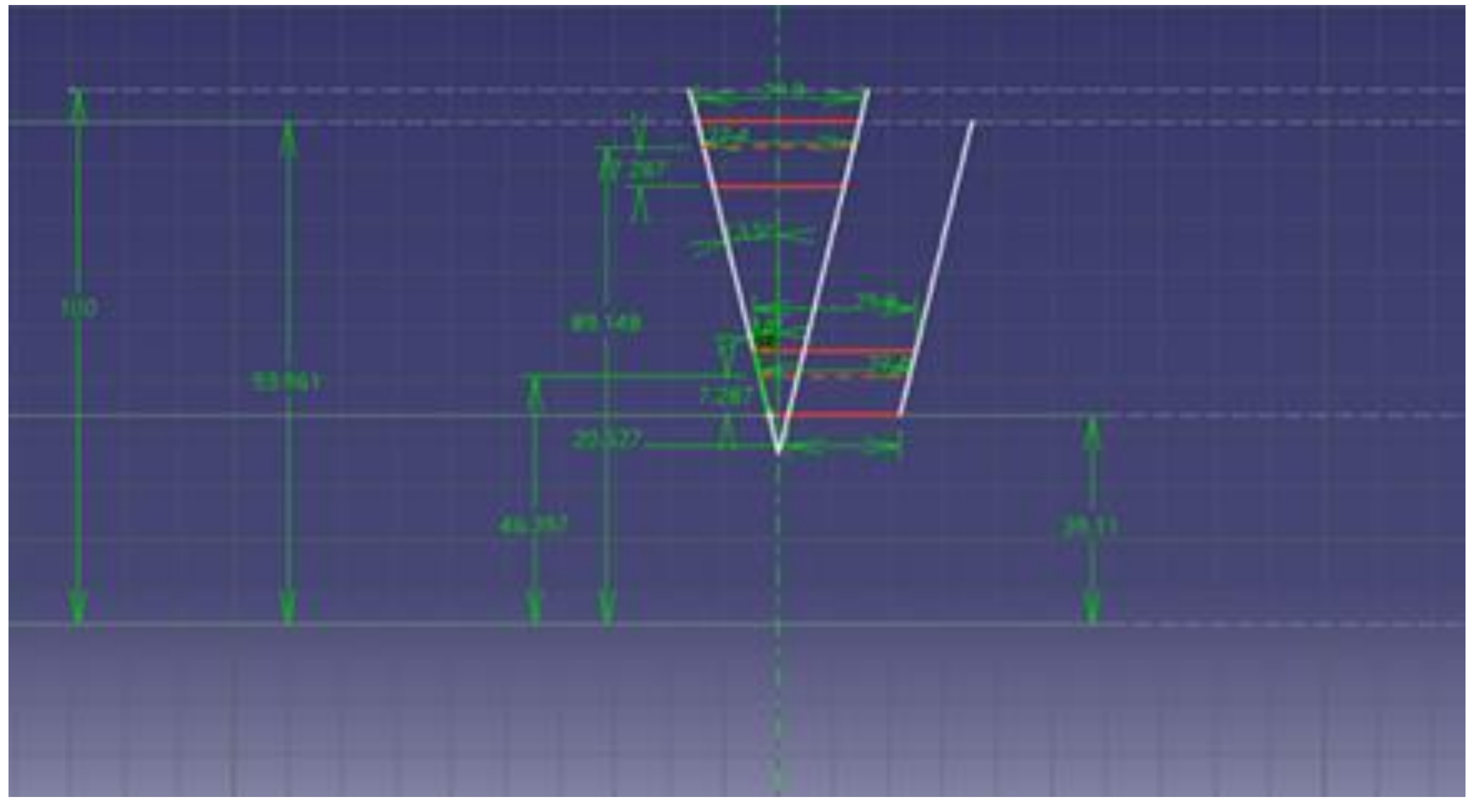
Dimensions of Primary and Secondary Pulley

Parameter	Primary Pulley	Secondary Pulley
Sheave Diameter	170 mm	200 mm
Post Diameter	30 mm	63.79 mm
Sheave Angle	26°	27°

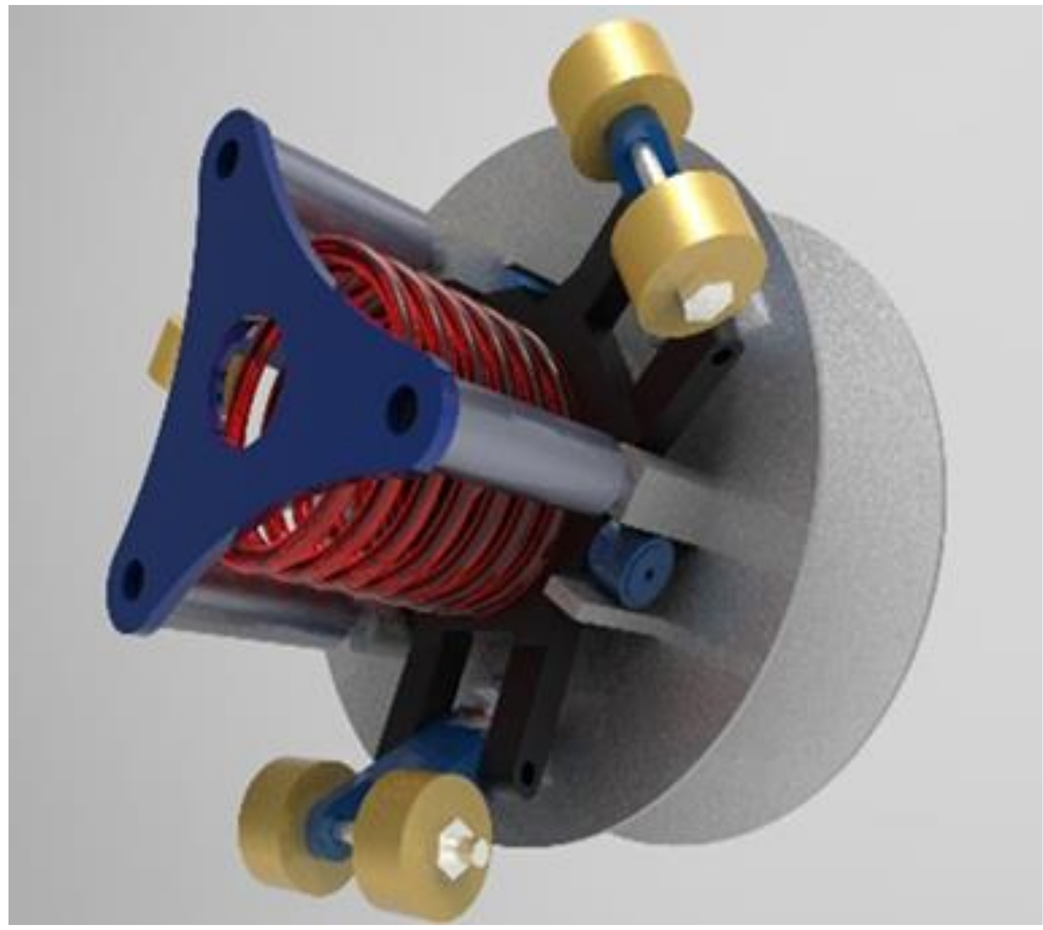
Primary Sheave Travel Line Diagram



Secondary Sheave Travel Line Diagram



Modelling of CVT



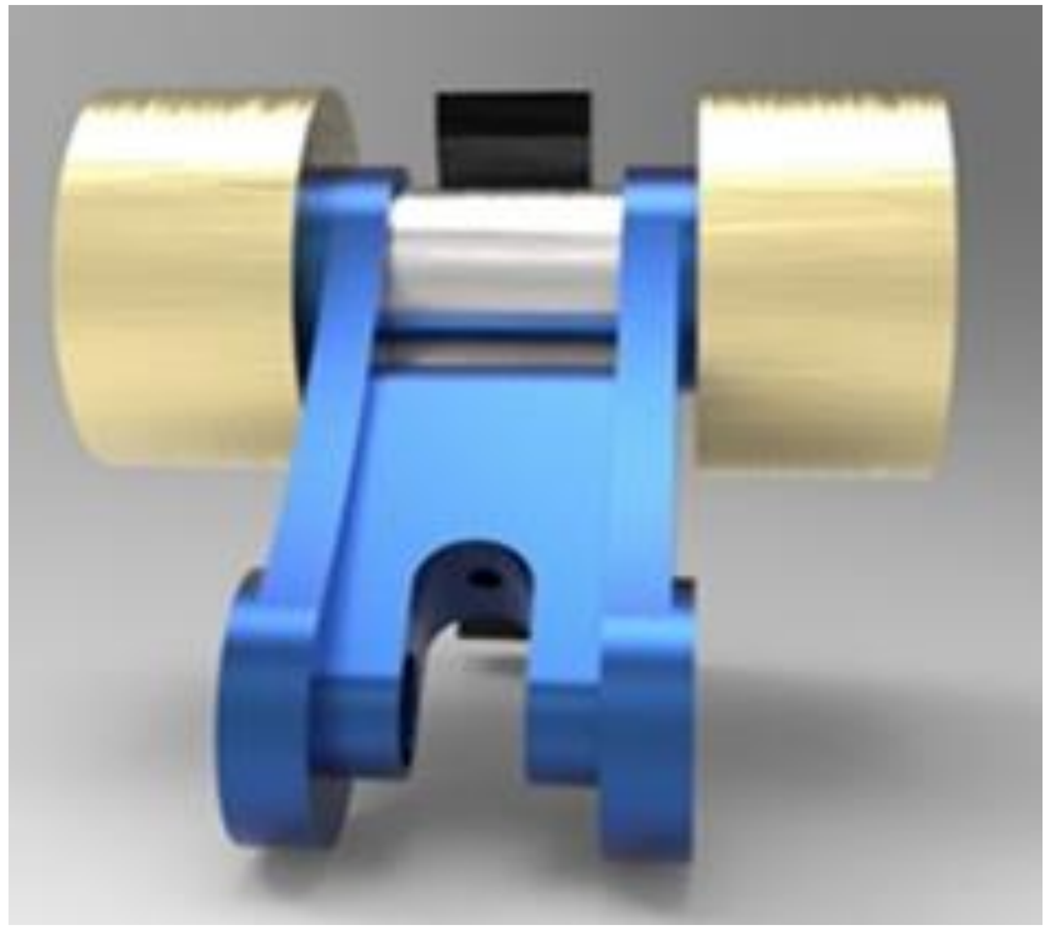
Primary Sheaves – There are two sheaves in CVT primary – Movable Sheave and Fixed Sheave. Both the sheaves have an angle slightly lower than that of the belt to offer the belt more grip. Their function is to change drive ratios by relative motion between the two.

Primary Ramp Holder – Serves as a housing for the ramps. The design criteria of the ramp holder are to reduce weight but at the same time, exhibit high rigidity. This means the CVT performs consistently through all load cases.

Primary Ramp – It has a specifically defined profile (using MATLAB) that allows the movable sheave to change drive ratio due to rolling of flyweights on it.

Flyweight Assembly –They aid the movement of the sheaves by tracing the ramp profile due to the action of centrifugal force.

Primary Post – It is connected to the crankshaft and holds the entire assembly of the primary in place. The movable sheave moves on the primary post.



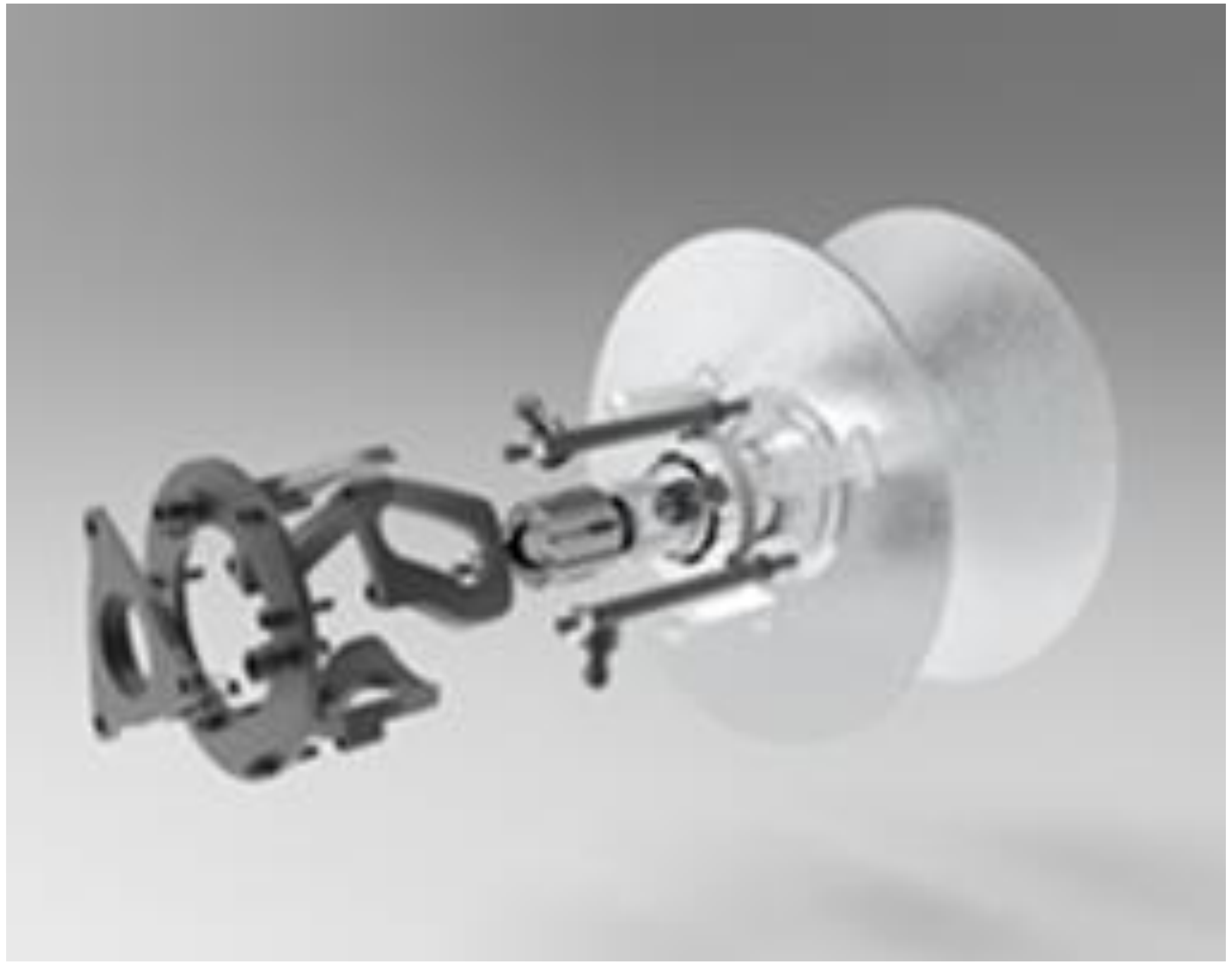
Secondary Pulley -

Secondary Sheaves –Just like the primary, this component also has two sheaves – Fixed Sheave and Movable Sheave. The movable sheave is integrated into the secondary post.

Secondary spring – They offer the opposing force to the movement of the sheaves due to the changing drive from the primary. They are mainly responsible for back shifting. The spring was changed from a torsional spring to a compression spring so as to obtain more accurate spring data and ease of assembly. To compensate for the relative motion between the fixed sheave and movable sheave, a thrust bearing was used.

Secondary Ramp – They have a specifically designed profile, on which the roller bearing is attached to the movable roll during shifting of ratios.

Belt – It is a V-Grooved belt that serves as a link from the primary to the secondary of a CVT. Change in diameter of the belt on the primary and secondary accounts for the changing drive ratios.



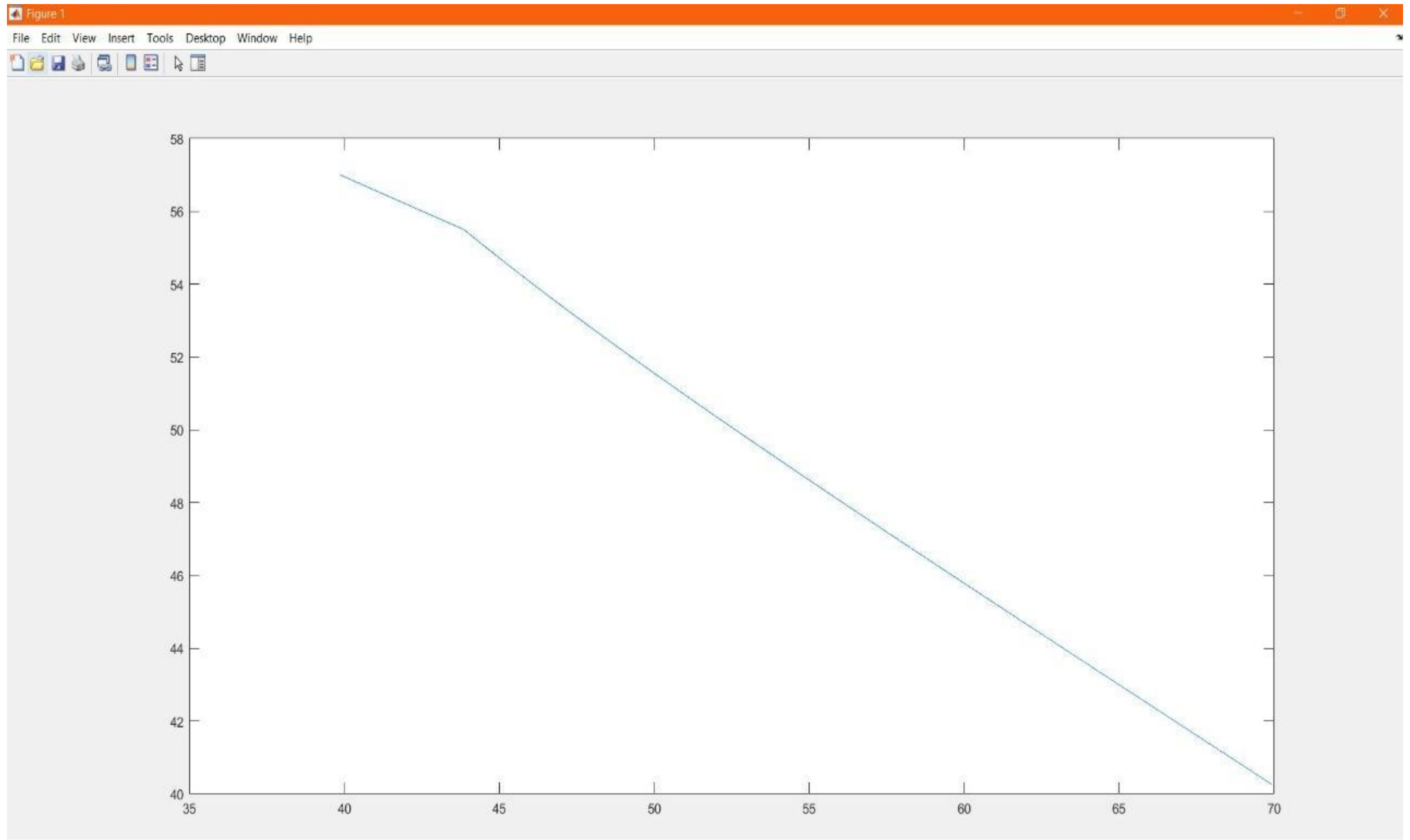
CVT Code

```
Ne = 2200; %Engagement RPM
R_ini = 22.28705; %Initial Pitch Radius at Engagement (m)
T_ini = -(1.4367*10^-6) * Ne^2 + 0.0076014*Ne + 8.9676; %Torque produced by Engine at N RPM (Nm)
Mu = 0.8; %Coefficient of Friction Between Belt and Aluminium
A = 13; %Primary Sheave Angle
B = A * pi /180;
K = 19.88; %Spring Stiffness (N/mm)
x_ini = 39.875; %Pretension Length (mm)
Fs = K*x_ini; %Pretension force (N)

m_w = 0.188; %Mass of Flyweight Side of flyarms 160gms weight, 35mm arm length(Kg)
m_r = 0.035; %Mass of Roller Side of flyarms (Kg)
L1 = 32.910; %Distance of COM of weight side of flyarms from Fulcrum (mm)
L2 = 35; %Distance of weights from fulcrum (mm)
L3 = 21.3378; %Distance of COM of roller side of flyarms from Fulcrum (mm)
L4 = 27.7; %Distance of rollers from Fulcrum (mm)
R = 65; %Radius of fulcrum from post centre (mm)
w = 2*pi*Ne / 60; %Angular Velocity at Engagement (rad/s)
y_ini =57; %Initial y position of Ramp (mm)
Fc = 3*w^2*((m_w*(L1/L4)*(R+((L1/L4)*(R-6.34-y_ini))))-(m_r*(L3/L4)*(R-((L3/L4)*(R-6.34-y_ini)))))/1000; %Centrifugal Force at Engagement (N)
theta = atan(Fs/Fc);
slope_p = (22.261361517981236 - 70.499576573252870)/(25.048 - 51.098);
i=1;
x(1) = x_ini; %Initialisation
y(1) = y_ini; %Initialisation
rp =[22.261361517981236,22.261361517981236,22.261361517981236,22.261361517981236];
while (x(i) < (x_ini + 4))
alpha(i) = atan(Fs/Fc);
k = tan(alpha(i));
SF(i) = K*x(i);
CF(i) = 3*w^2*((m_w*(L1/L4)*(R+((L1/L4)*(R-6.34-y(i)))))-(m_r*(L3/L4)*(R-((L3/L4)*(R-6.34-y(i))))))/1000;
RF(i) = CF(i)*k;
SF_P(i) = RF(i) - SF(i);
x(i+1) = x(i) + 1;
y(i+1) = y(i) - k;
i=i+1;
```

CVT Code

```
end
SF(i) = K*x(i);
CF(i) = 3*w^2*((m_w*(L1/L4)*(R+((L1/L4)*(R-6.34-y(i)))))-(m_r*(L3/L4)*(R-((L3/L4)*(R-6.34-y(i))))))/1000;
RF(i) = CF(i)*k;
SF_P(i) = RF(i) - SF(i);
rp(i) = (x(i) - x_ini - 4)*slope_p + 22.261361517981236;
alpha(i) = atan(Fs/Fc);
z= 0.05; %Increment Variable
Ns = 3400; %Shifting RPM
T_s = -(1.4367*10^-6) * Ns^2 + 0.0076014*Ns + 8.9676; % Torque at shifting RPM
Ws = 2*pi*Ns/60;
j=1;
while (x(i) < (x_ini + 30))%26.262 sheave travel + 3 mm extra
SF(i+1) = K*x(i);
CF(i+1) = 3*w^2*((m_w*(L1/L4)*(R+((L1/L4)*(R-6.34-y(i)))))-(m_r*(L3/L4)*(R-((L3/L4)*(R-6.34-y(i))))))/1000;
rp(i+1) = (x(i) - x_ini - 4)*slope_p + 22.261361517981236;
SF_B(i+1) = 1300*T_s * cos(B)/(2*Mu*rp(i+1));
alpha(i+1) = atan ((SF(i+1)+SF_B(i+1))/CF(i+1));
RF(i+1) = CF(i+1)*tan(alpha(i+1));
SF_P(i+1) = RF(i+1) - SF(i+1);
s =z * tan (alpha(i+1));
x(i+1) = x(i) + z;
y(i+1) = y(i) - s;
i = i+1;
j=j+1;
end
plot(x,y)
```



1. Design of gears

- After finding the overall reduction, this reduction is split into stages. This gear box consist of 4 stages, 2 spur gear reduction(which help to transfer power to rear wheel), 1 chain drive reduction and 1 bevel gear reduction(both chain and bevel gear help to transfer power to front). We have kept CVT low ratio as 4, so reduction of compound spur gear is 8.5. So each spur gear reduction is 3 and 2.8333. Chain drive reduction and bevel gear reduction are 1.
- After finding the reduction, we check for interference of gear teeth to determine minimum and maximum number of teeth on gear and pinion for a given module.

$$N_p = \frac{2k}{(1+2m)\sin 2\phi}\left(m + \sqrt{m^2 + (1+2m)\sin^2\phi}\right) \qquad N_G = \frac{N_P^2 \sin^2\phi - 4k^2}{4k - 2N_P \sin^2\phi}$$

Where, k=1 for full depth teeth

m=2.833(Gear ratio)

ϕ=20(Pressure angle)

- After finding the no. of teeth in pinion and gear, other parameter are found out using AGMA stress equation(From Shigley’s mechanical textbook (9th edition)) using excel sheet iterative method.(This also applies for bevel gear)
- After finding the parameters, the gear is modeled and is analysed for weight reduction.

- **AGMA Stress Equations**

Two fundamental stress equations are used in the AGMA (American Gear Manufacturing Association) methodology, one for bending stress and another for pitting resistance (contact stress). In AGMA terminology, these are called stress numbers.

Gear Bending Stress Equation for Bevel Gear (σ)

$$S_t = \frac{W^T}{F} P_d K_o K_v \frac{K_s K_m}{K_x J}$$

Gear Bending Stress Equation for Spur Gear (σ)

$$S_t = \frac{W^T}{F} P_d K_o K_v K_s \frac{K_h K_b}{J}$$

Where,

WT = Tangential transmitted load, lbf

Ko = Overload factor

Kv = Dynamic factor

Ks = Size factor

Kh = Load-distribution factor

Kb =Rim-thickness factor

Pd = Transverse diametral pitch

F = Face width of the narrower member, inch

Km = Load-distribution factor

Kx = lengthwise curvature factor

YJ = Geometry factor for bending strength

Wt - Tangential Tooth Load

Wt= (33000×H×ŋ)/V

H = Power output of the engine

ŋ = Efficiency

V = Pitch-line Velocity

Gear Contact Stress Equation for bevel gear

$$S_C = \sigma_C = C_P * \left(\frac{W^t}{F * d_P * I} * K_O K_V K_M C_S C_{SC}\right)^{0.5}$$

Gear Contact Stress Equation for spur gear

$$S_C = \sigma_C = C_P * \left(\frac{W^t}{F * d_P * I} * K_O K_V K_s K_m C_f\right)^{0.5}$$

Where,

W_t = Tangential load, lbf

K_o = Overload factor

K_v = Dynamic factor

C_s= Size factor for pitting resistance.

F = Face width (mm)

K_m = Load-distribution factor

K_s = Size factor

I= Geometry factor for Pitting resistance

d_p =Outer pitch diameter of pinion (in)

C_{xc}=Crowning factor for pitting resistance

C_p= Elastic Coefficient for pitting resistance

C_f = Surface Condition Factor

- We use excel sheet method to determine the gear parameter. The excel sheet for pinion 1 and gear 1 is shown in the next slide. In the same way we can calculate gear parameter for remaining gear.

- Gear parameters

Name	No of teeth	Module	Face Width(mm)
Pinion 1	15	2.5	9.3
Gear 1	45	2.5	9.3
Pinion 2	18	2.5	13.2
Gear 2	51	2.5	13.2
Bevel pinion	26	2.5	20.6
Bevel gear	26	2.5	20.6

Excel Sheet Iterative Method to determine Gear Parameters

Name	Teeth (Z)	Lewis Form Factor (Y)	Teeth of Mating Gear	Gear Ratio (Mg)	F.O.S Bending	F.O.S Contact	Module (m) mm	Pitch Circle Dia (Dp) in	Facewidth (F) mm	Face Width (F) in	Diametrical Pitch (P) tooth / in	RPM of Engine	Drive Ratio	Pinion RPM	Pitch Line Velocity (V) ft / min	Power (H) HP	Efficiency	Tangential Tooth Load (Wt) lbf	Ko	Kv	Ks	Cpf	Cpm
	15	0.29	45	3	1.262012538	1.290991637	2.5	1.476377953	11.7	0.460629921	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	0.97727306	0.0062	1
	15	0.29	45	3	1.251784865	1.285749732	2.5	1.476377953	11.7	0.460629921	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.0062	1
	15	0.29	45	3	1.074252273	1.191090429	2.5	1.476377953	10	0.393700787	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00166667	1
	15	0.29	45	3	1.178857563	1.247734723	2.5	1.476377953	11	0.433070866	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00433333	1
	15	0.29	45	3	1.157976445	1.23663478	2.5	1.476377953	10.8	0.42519685	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.0038	1
	15	0.29	45	3	1.15275305	1.233842525	2.5	1.476377953	10.75	0.423228346	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00366667	1
	15	0.29	45	3	1.14752841	1.231043267	2.5	1.476377953	10.7	0.421259843	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00353333	1
Pinion 1	15	0.29	45	3	1.137075386	1.225423553	2.5	1.476377953	10.6	0.417322835	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00326667	1
	15	0.29	45	3	0.99020718	1.143548608	2.5	1.476377953	9.2	0.362204724	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	-0.00046667	1
	15	0.29	45	3	1.000730431	1.149608979	2.5	1.476377953	9.3	0.36614173	10.16	2600	4	650	251.33577	6.88954	0.85	768.8981023	1.75	1.097	1	-0.0002	1
	15	0.29	45	3	1.011248642	1.15563469	2.5	1.476377953	9.4	0.37007874	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	6.6667E-05	1
	15	0.29	45	3	1.021761817	1.16162628	2.5	1.476377953	9.5	0.374015748	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00033333	1
	15	0.29	45	3	1.053271159	1.17940154	2.5	1.476377953	9.8	0.385826772	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00113333	1
	15	0.29	45	3	1.074252273	1.191090429	2.5	1.476377953	10	0.393700787	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00166667	1
	15	0.29	45	3	1.168419495	1.242198474	2.5	1.476377953	10.9	0.429133858	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.00406667	1
	15	0.29	45	3	1.189290654	1.253243894	2.5	1.476377953	11.1	0.437007874	10.16	2600	4	650	251.335771	6.88954	0.85	768.8981023	1.75	1.0969	1	0.0046	1

Cpf	Cpm	Cma	Kh	Kb	Yj	Bending Stress PSI	Allowable Bending Stress (St)	Reliability R	Yz	Number of rotations of wheel	Gear 2	Pinion 2	Second Stage Reduction	Reduction	Number of rotations of gear	Yn	F.O.S Bending	Cp	Zi	Zn	Contact Stress	Allowable Contact Stress	F.O.S Contact
0.0062	1	0.07337642	1.07957642	1.16	0.25	159372.653	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.262012538	2279.122	0.12047183	1.30600732	314215.1298	275000	1.290991637
0.0062	1	0.07337642	1.06366113	1.16	0.25	160674.802	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.251784865	2279.122	0.12047183	1.30600732	315496.1614	275000	1.285749732
0.00166667	1	0.07252502	1.05935335	1.16	0.25	187228.169	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.074252273	2279.122	0.12047183	1.30600732	340569.5277	275000	1.191090429
0.00433333	1	0.07302594	1.06188742	1.16	0.25	170614.578	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.178857563	2279.122	0.12047183	1.30600732	325108.4523	275000	1.247734723
0.0038	1	0.07292578	1.06138062	1.16	0.25	173691.172	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.157976445	2279.122	0.12047183	1.30600732	328026.602	275000	1.23663478
0.00366667	1	0.07290074	1.06125392	1.16	0.25	174478.207	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.15275305	2279.122	0.12047183	1.30600732	328768.9447	275000	1.233842525
0.00353333	1	0.07287569	1.06112722	1.16	0.25	175272.598	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.14752841	2279.122	0.12047183	1.30600732	329516.5292	275000	1.231043267
0.00326667	1	0.07282561	1.06087382	1.16	0.25	176883.862	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.137075386	2279.122	0.12047183	1.30600732	331027.671	275000	1.225423553
-0.00046667	1	0.07212407	1.05732592	1.16	0.25	203119.398	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	0.99020718	2279.122	0.12047183	1.30600732	354728.3447	275000	1.143548608
-0.0002	1	0.072174	1.057579	1.16	0.25	200983.5	112000	0.95	0.885376	10000	51	18	2.833333	8.5	85000	1.589964	1.00073043	2279.122	0.120472	1.306007	352858.33	275000	1.149609
6.6667E-05	1	0.07222433	1.05783279	1.16	0.25	198893.01	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.011248642	2279.122	0.12047183	1.30600732	351018.4563	275000	1.15563469
0.00033333	1	0.07227445	1.05808623	1.16	0.25	196846.547	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.021761817	2279.122	0.12047183	1.30600732	349207.9266	275000	1.16162628
0.00113333	1	0.0724248	1.05884651	1.16	0.25	190957.745	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.053271159	2279.122	0.12047183	1.30600732	343944.8662	275000	1.17940154
0.00166667	1	0.07252502	1.05935335	1.16	0.25	187228.169	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.074252273	2279.122	0.12047183	1.30600732	340569.5277	275000	1.191090429
0.00406667	1	0.07297586	1.06163402	1.16	0.25	172138.762	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.168419495	2279.122	0.12047183	1.30600732	326557.4007	275000	1.242198474
0.0046	1	0.07307602	1.06214081	1.16	0.25	169117.856	112000	0.95	0.88537608	10000	51	18	2.83333333	8.5	85000	1.58996378	1.189290654	2279.122	0.12047183	1.30600732	323679.2989	275000	1.253243894

2. Design of sprocket

- Sprocket and Chain Drive is used to lower Engine position and thus reducing center of gravity of car. By this overall performance of car is improved.
- Center to Center Distance is assumed based on available space in rear packaging of driveline.
- Minimum number of teeth is chosen from Shigley’s mechanical textbook (9th edition).
- Proper chain type ANSI 50-1 is selected and pitch chosen is 15.88mm (From Shigley’s mechanical textbook (9th edition)).
- Both tangential load and centrifugal load is taken for design calculation.

Total Tooth Load: P= Pt +Pc +Ps

Pt = Torque/Radius

= 228/ ((30*9.525)/2*Pi)

= 5302.32N

Pc = mv^2

= $10.16*(0.975)^2$

= 9.67N

Ps = k*w*a

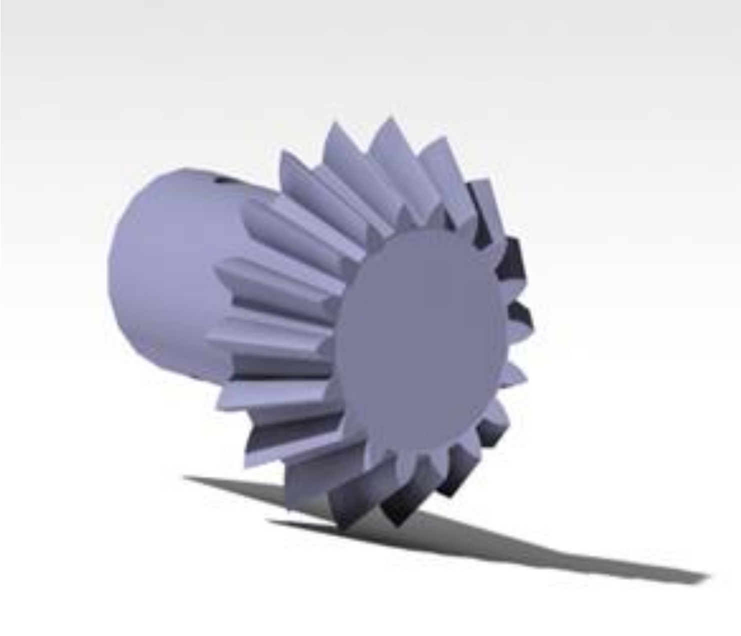
= 2*(20*9.81) *(0.096)

= 37.67N

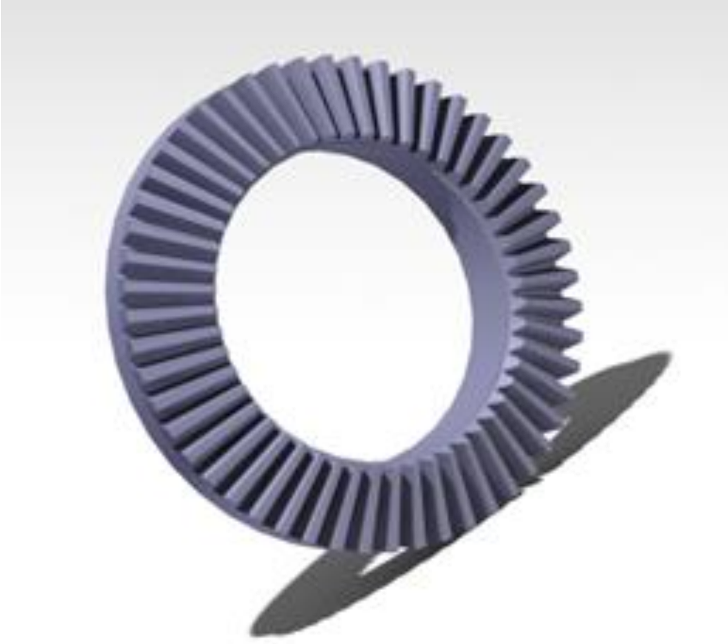
P=5349.65N

Where,
P= total tooth load
Pt= Tangential load
Pc= centrifugal load
Ps= Service Factor Load
m= mass per unit length of sprocket
v= velocity of chain drive
k= service factor
w= weight per unit length
a= center-center distance

- Proper lubrication is selected and corrected c-c distance is calculated for positioning of sprockets.
- After finding sprocket parameters, the sprocket is modelled and analysed for weight reduction.

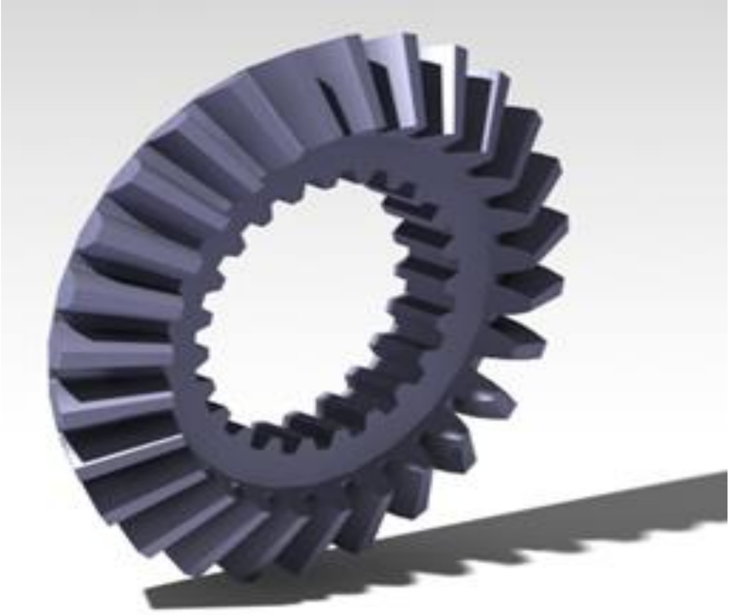


Bevel Pinion



Crown Gear

Spider Gear



Side Gear

Differential Gears

Pinion 1

Gear 1

Pinion 2

Gear 2

Sprocket

Bevel Gear

Gearbox Gears

3. Design of shaft

- Shaft design involves determination of the correct shaft diameter to ensure satisfactory strength & rigidity when the shaft is transmitting power under various loading conditions. The diameters of the shaft are chosen based on the different loads that are acting on the shaft and their positions with respect to the two bearings on which each shaft rests.

$$d_0 = \sqrt[3]{\frac{16\left(\sqrt[2]{(C_m M)^2 + (C_t T)^2}\right)}{\pi\tau(1-K^4)}}$$

An excel sheet iteration method was used to determine the value of d_i by altering the values of K, C_m, C_t.

Shaft 1 Diameter Calculation								
Material	Shear Strength (Mpa)	k	M (N-mm)	T (N-mm)	Cm	Ct	Do(mm)	Di(mm)
Titanium	550	0	55133.25478	76000	2.5	2.25	13.7224385	0
	550	0	55133.25478	76000	2.5	2.25	15.7082708	0
	550	0	55133.25478	76000	2.25	2	15.1427887	0
	550	0	55133.25478	76000	2.5	2.25	15.7082708	0
	550	0.3	55133.25478	76000	2.5	2.25	15.7509136	4.7252741
	550	0.4	55133.25478	76000	2.5	2.25	15.8446489	6.3378596
	550	0.8	55133.25478	76000	2.5	2.25	18.7246164	14.979693
	550	0.82	55133.25478	76000	2.5	2.25	19.1970146	15.741552
	550	0.84	55133.25478	76000	2.5	2.25	19.7631749	16.601067
	550	0.848	55133.25478	76000	2.5	2.25	20.0222314	16.978852
	550	0.85	55133.25478	76000	2.5	2.25	20.0903563	17.076803

- In the same way all shaft calculation are done in excel sheet.

Final Shafts Dimensions

Final Shaft Dimensions	d_o (mm)	d_i (mm)
Shaft1	19	9
Shaft2	22	12
Shaft3	35	26
Shaft4	22	15
Shaft5	28	24
Differential Bevel Input Shaft	25	15
Differential Output Shaft	29	25

Gearbox shaft 1

Gearbox shaft 2

Gearbox shaft 3

Gearbox shaft 4

Gearbox shaft 5

Differential output shaft

Differential bevel input shaft

Selection of Bearing

$$L_{10mh} = a_{xyz} \Big| * \frac{1000000}{60*p} * \left(\frac{c}{p}\right)^{p}$$

L_{10mh} = Rating life in operating hours

a_{xyz} = Life modification factor

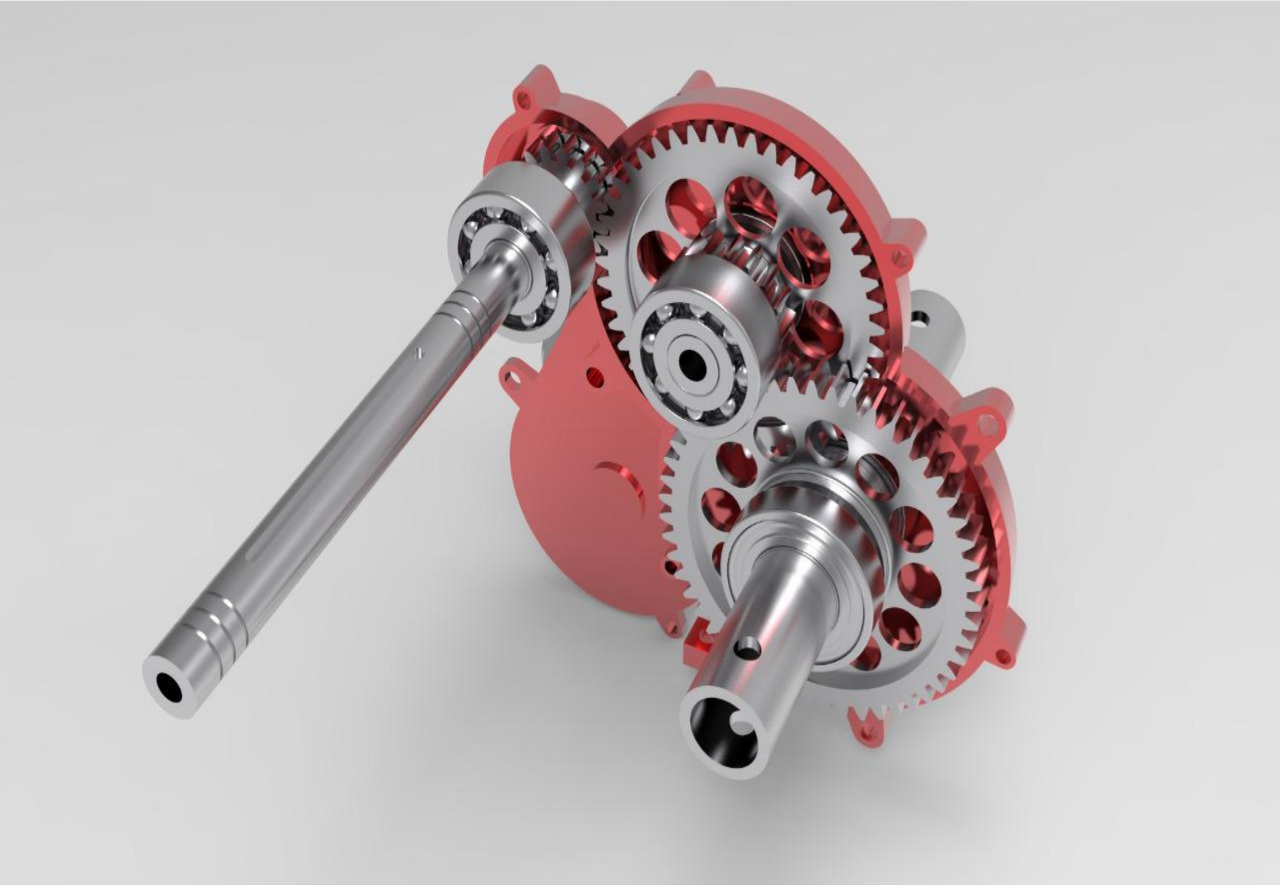
C = Basic load rating (kN)

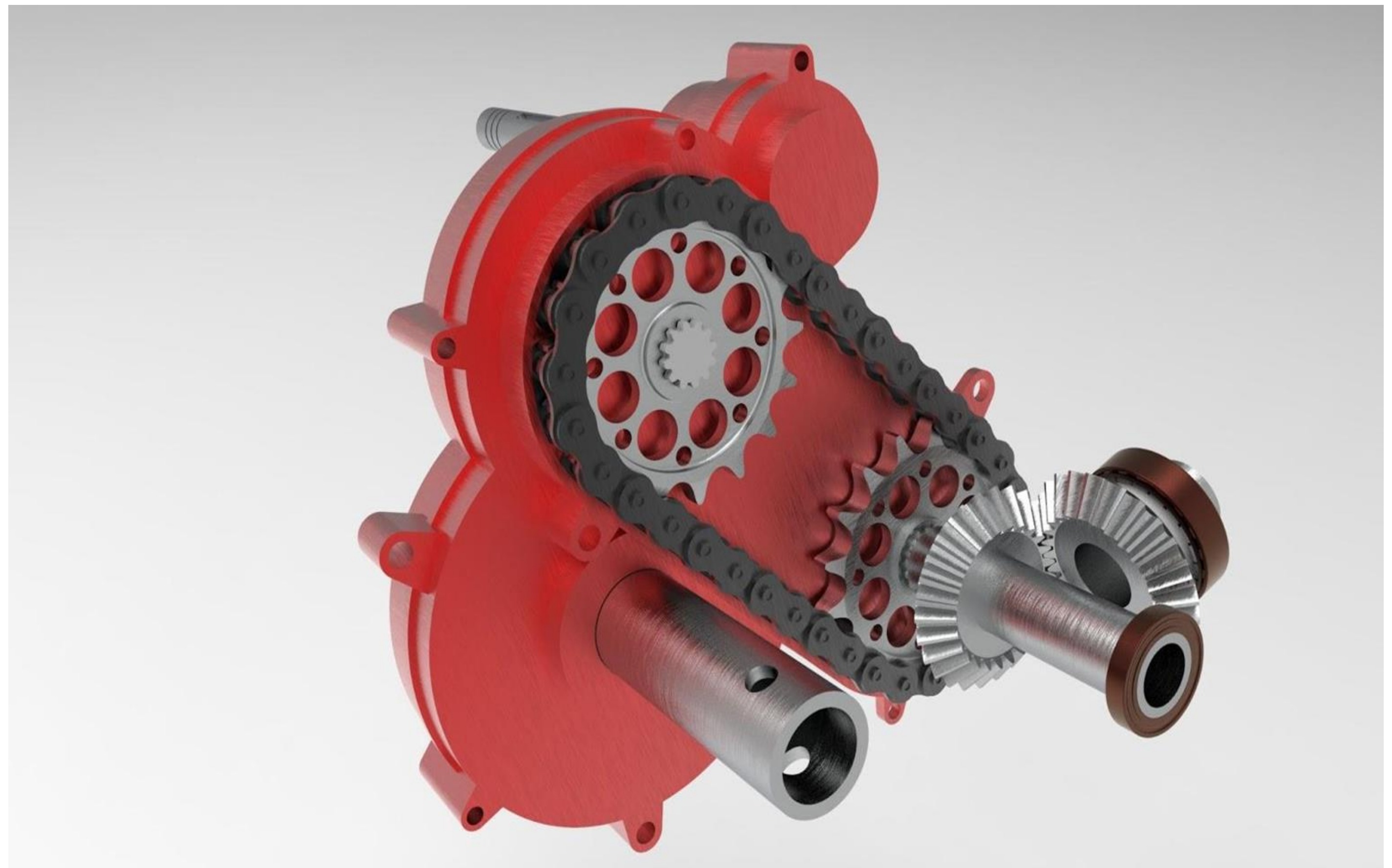
P = Equivalent dynamic bearing load (kN)

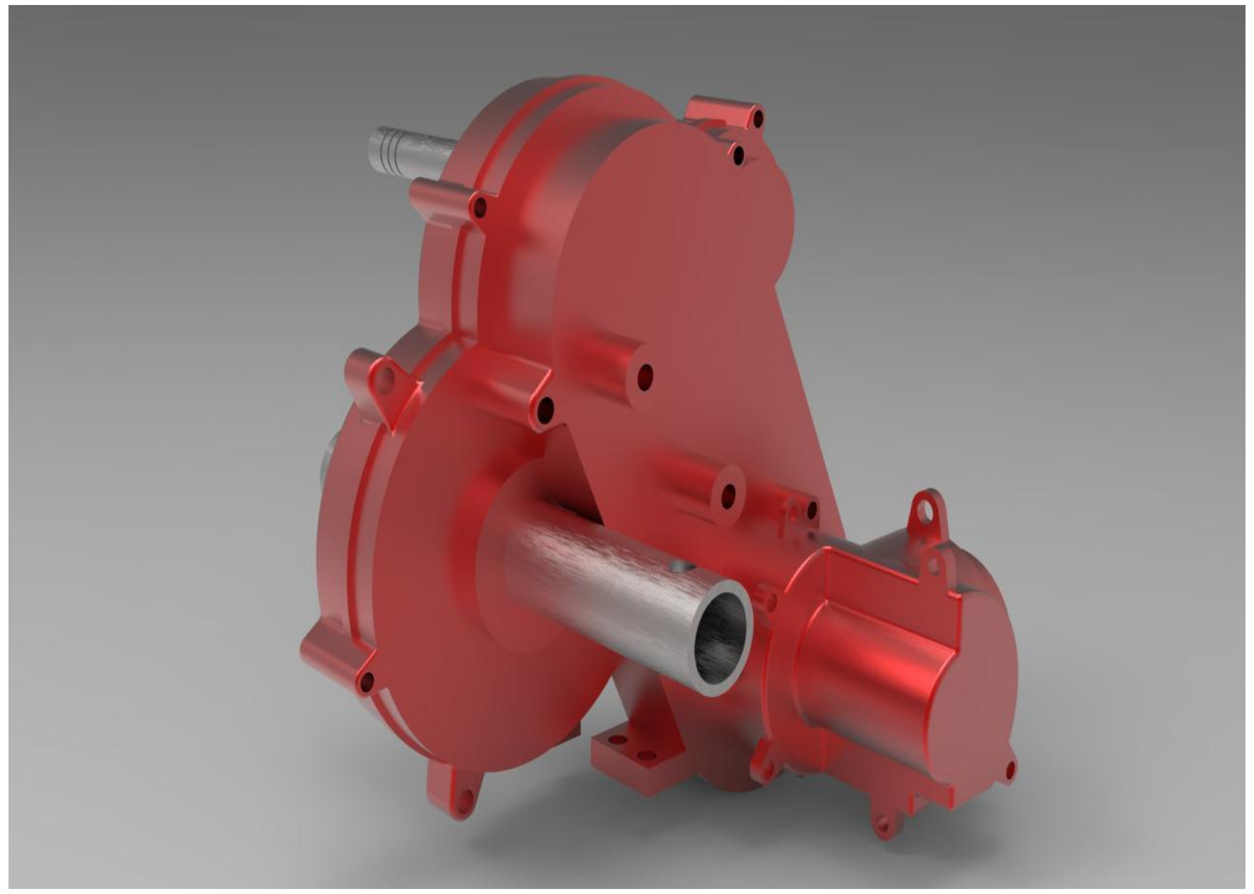
n = Rotational speed (r/min)

p = Exponent of life equation

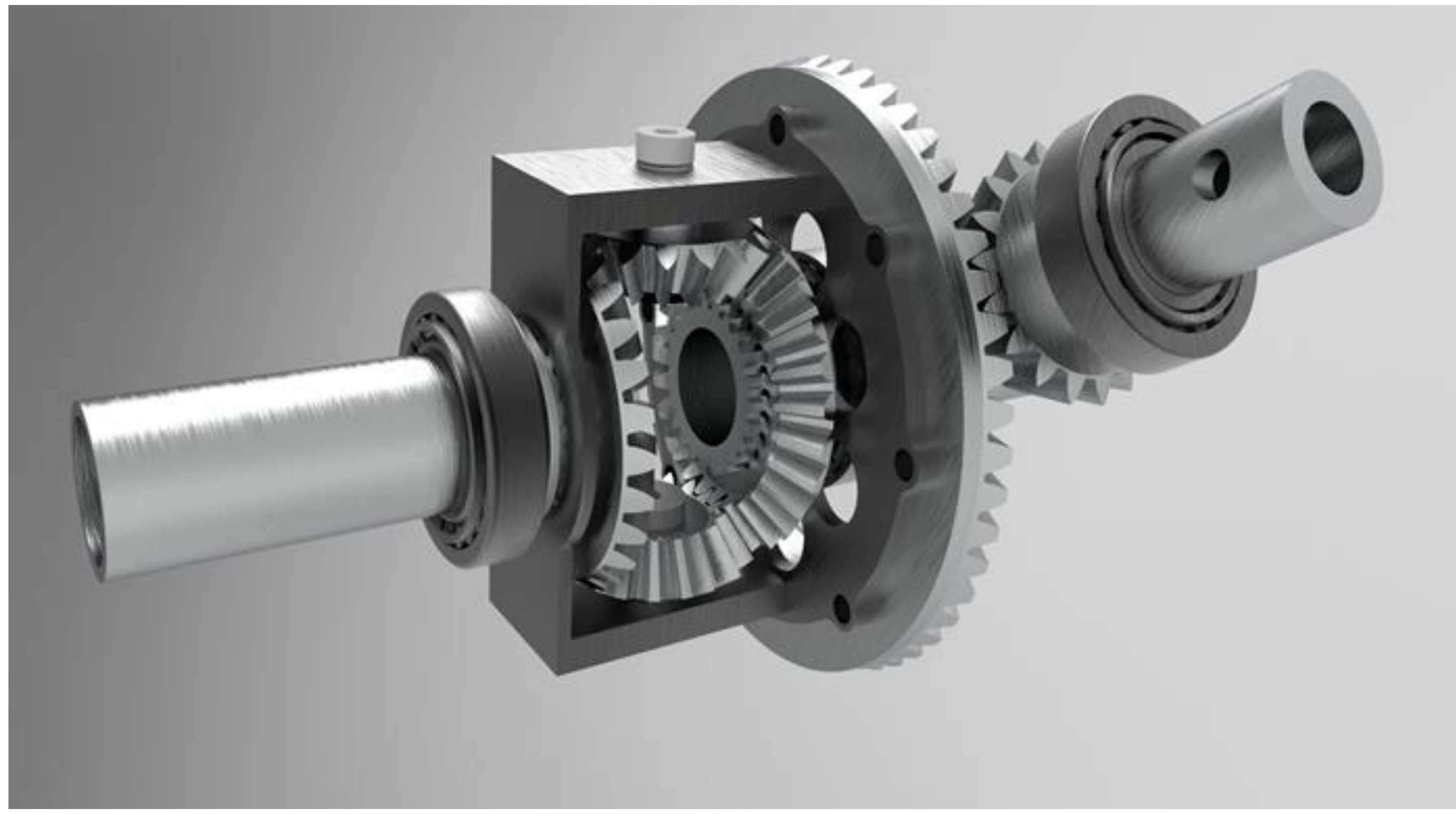
Finalised Gearbox Bearings								
Shaft	Position	Bearing	Di(mm)	Do(mm)	b(mm)	W (Kg)	C (KN)	Life
Shaft 1	Left	4304 ATN9	20	52	21	0.21	23.4	285.74
	Right	4203 ATN9	15	35	11	0.045	8.06	170.23
Shaft 2	Left	4304 ATN9	20	52	21	0.21	23.4	89.22
	Right	6204 - 2RSH	20	47	14	0.11	13.5	284.45
Shaft 3	Left	6007 2RS1	35	62	14	0.16	16.8	474.06
	Right	61907-2RS1	35	55	10	0.08	10.8	355.84
Shaft 4	Left	32205 B	25	52	19.25	0.19	44.5	260.56
	Right	61805 2RZ	25	37	7	0.02	4.36	286.26
Shaft 5		320/28 X	28	52	16	0.14	39	320.56
Diff Pinion		3200X	25	47	15	0.11	33.2	273.12
Diff Output		L45400	29	50.292	14.224	0.11	31.8	603.37







Reduction Gearbox



Open Differential

For analyzing driveline components, we will consider full torque conditions. Our gearbox acts as a lock differential, so when the car is on only one wheel on ground and rest in air, then all the torque is transferred to this wheel. During this condition we are doing static analysis. When it comes to fatigue analysis, we will consider average torque condition as there are rare conditions for a car to be on one wheel.

Inboard Yoke without adapter

Inboard Yoke with adapter

Outboard Yoke

Half Shaft

CV shaft

Long propeller shaft

Propeller yoke

Propeller Half-shaft

Rear Hub

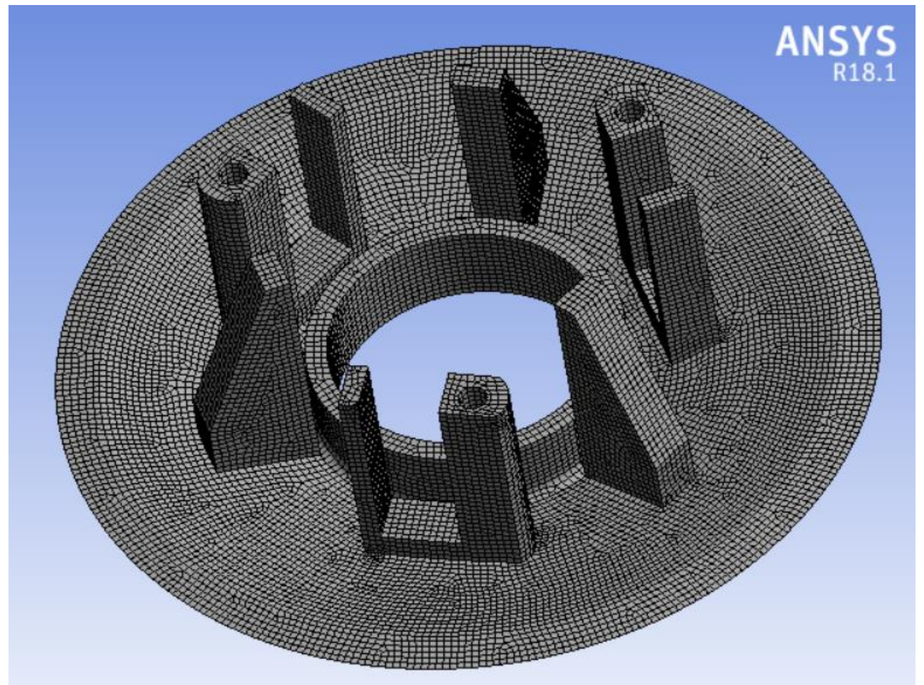
Top view of All wheel driveline

Isometric view of All wheel driveline

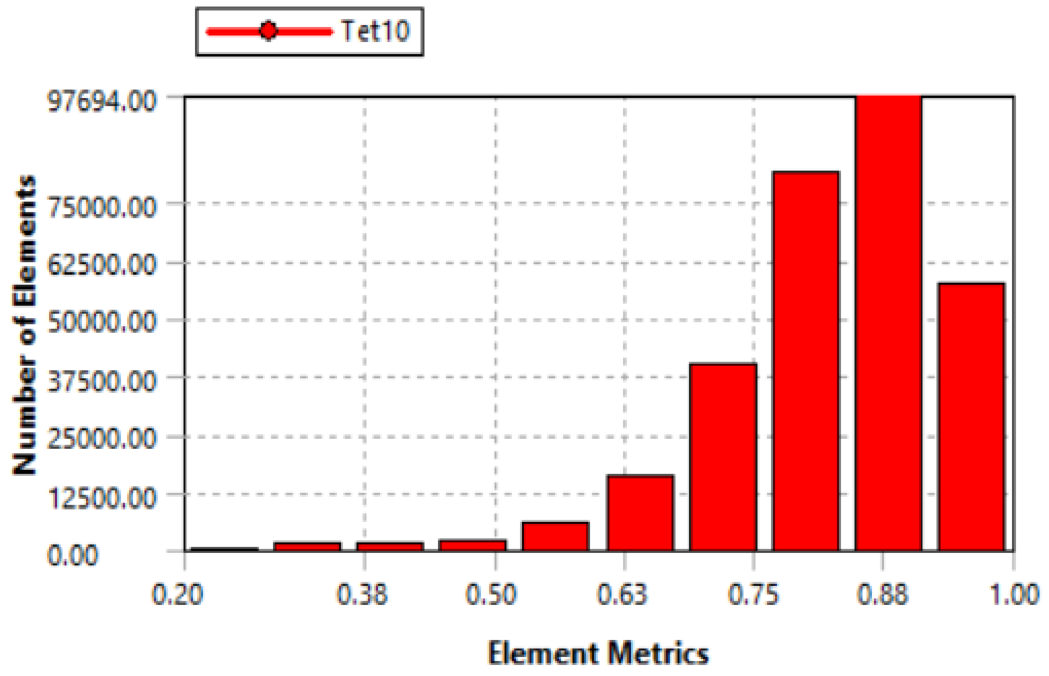
Result and Discussion

FEM Modelling

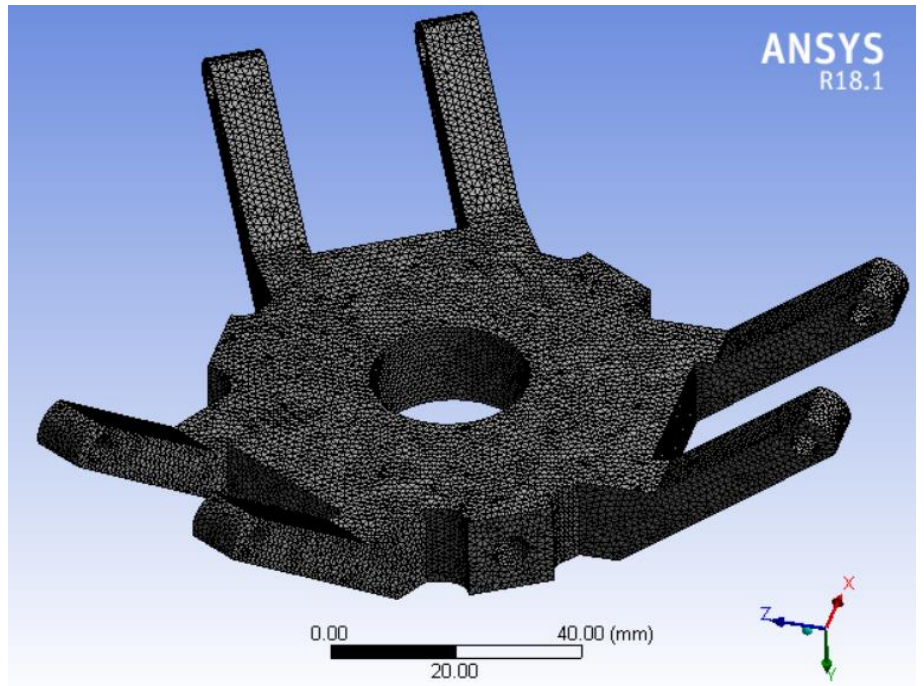
Meshing parameters for the components of the primary pulley has been chosen based on the complications of the geometry. Mesh quality parameters such as Jacobian, Aspect Ratio and Element quality have been considered.



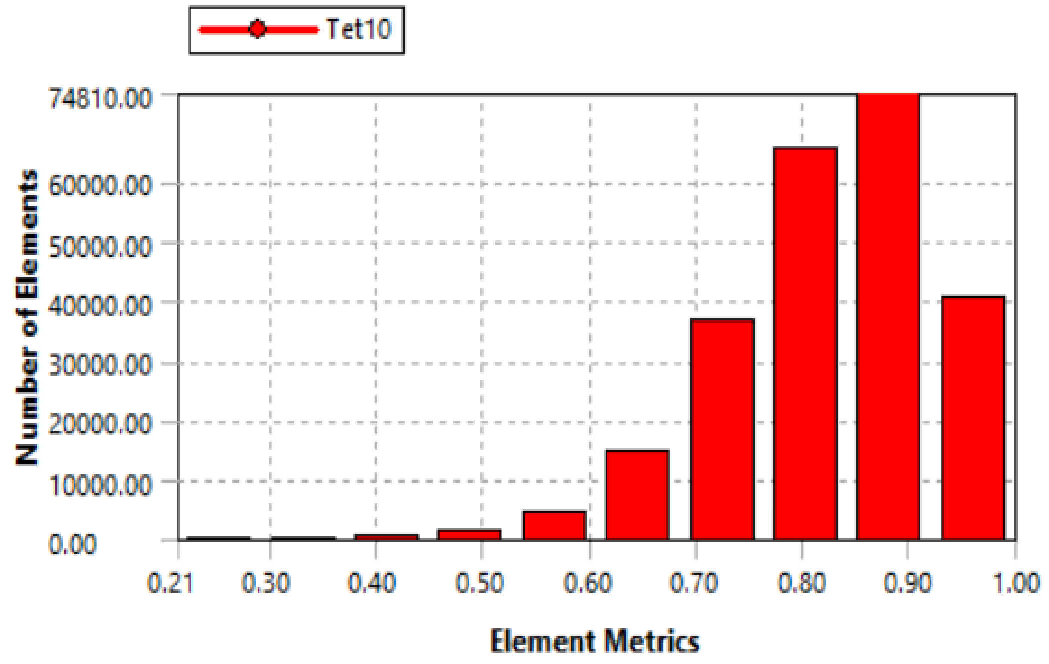
Meshed Primary Plate



Element quality of Primary Ramp Plate



Meshed Primary Fly-Arms Holder



Element quality of Primary Fly-Arms Holder

Mesh Parameters	
Method	Adaptive
Mesh Elements	Tetragonal
Element Sizing	1.5 mm

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.82	> 0.6
Jacobian	0.99	> 0.6
Aspect Ratio	1.89	< 10

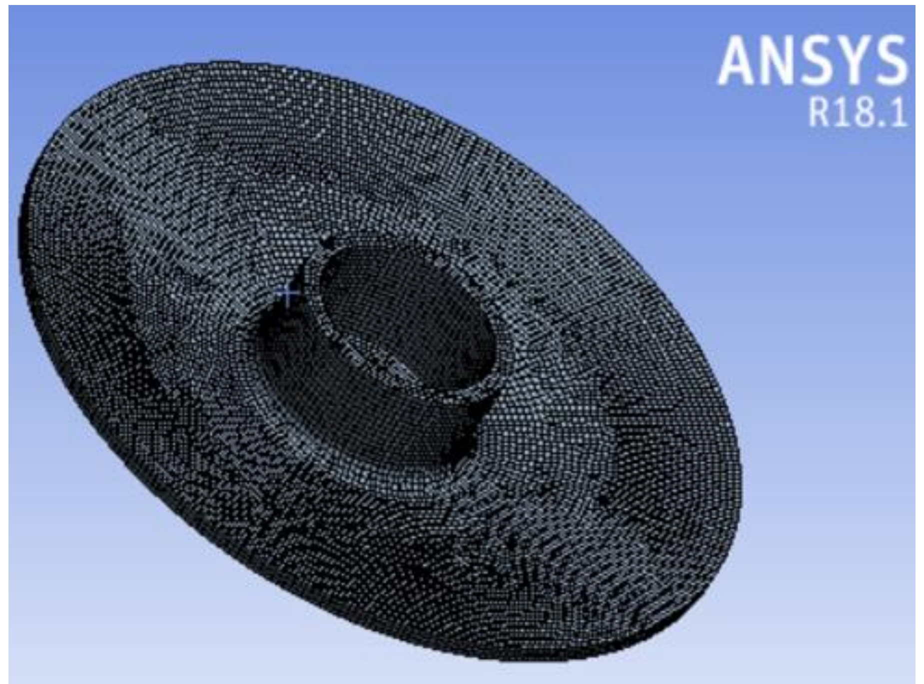
Mesh Parameters	
Method	Adaptive
Mesh Elements	Tetrahedron
Element Sizing	1 mm

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.83	> 0.6
Jacobian	0.99	> 0.6
Aspect Ratio	1.87	< 10

Result and Discussion

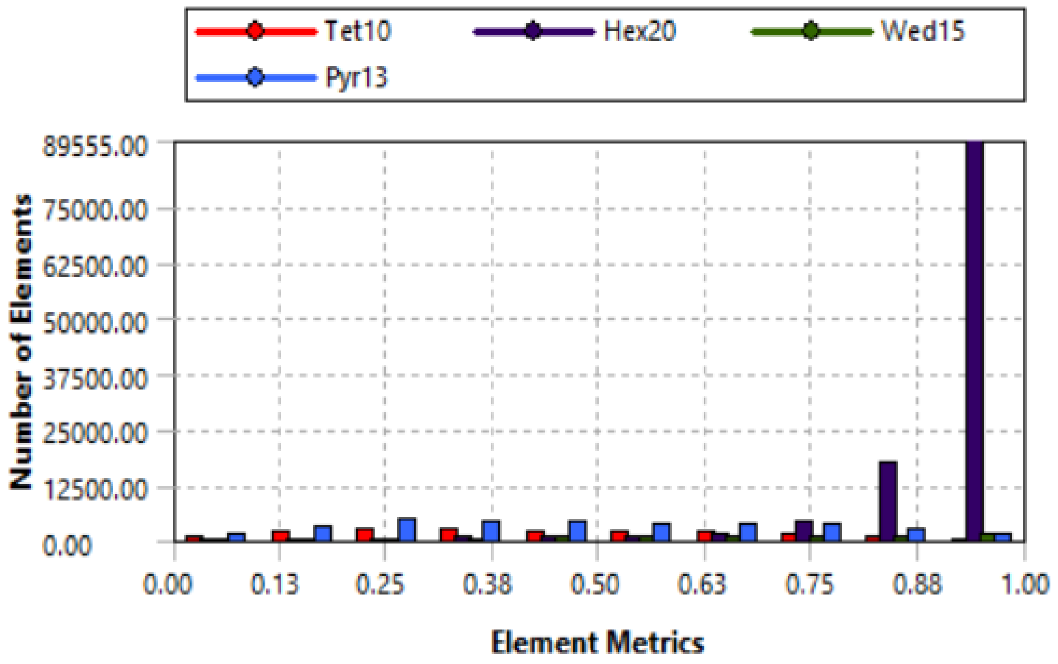
FEM Modelling

Components of the Primary Pulley Contd.



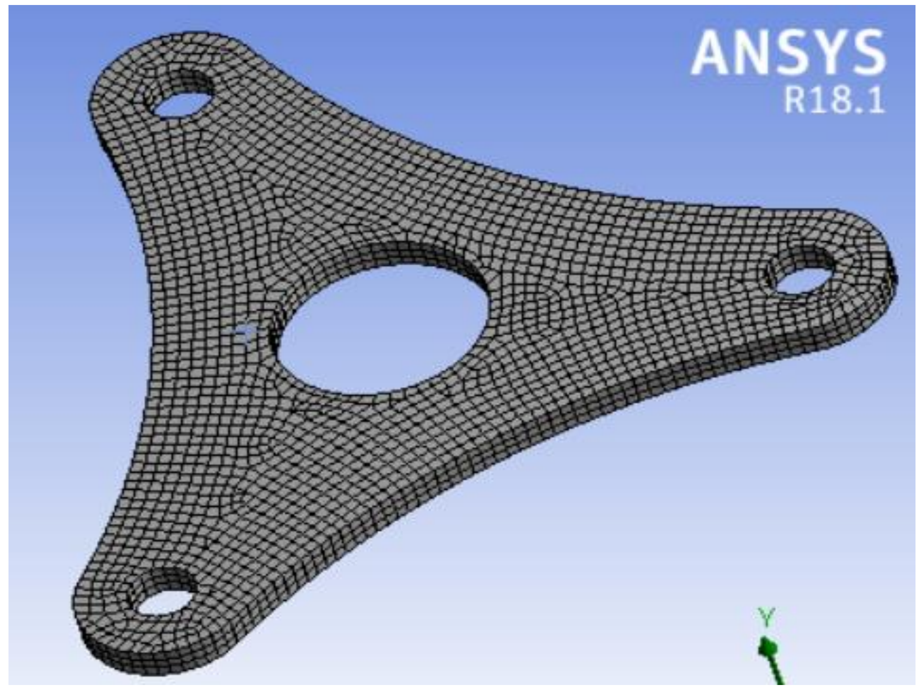
Meshed Primary Movable Sheave

Mesh Parameters	
Method	Hex Dominant
Mesh Elements	All Quad
Element Sizing	1 mm



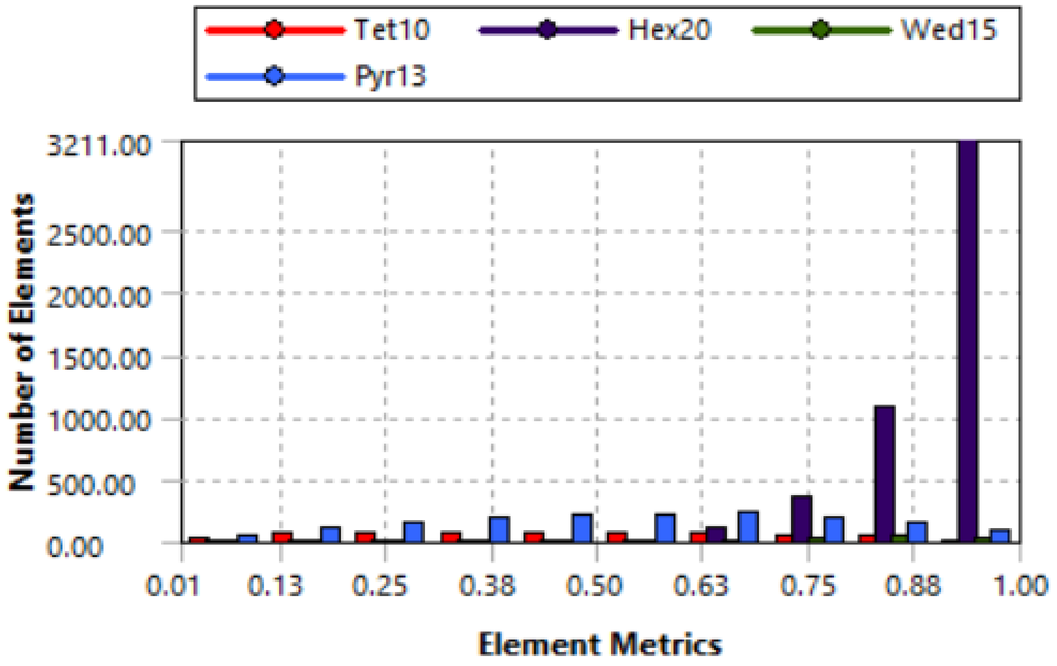
Element quality of Primary Movable Sheave

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.80	> 0.6
Jacobian	0.75	> 0.6
Aspect Ratio	3.38	< 10



Meshed Primary Spring Cap

Mesh Parameters	
Method	Hex Dominant
Mesh Elements	All Quad
Element Sizing	1.5 mm



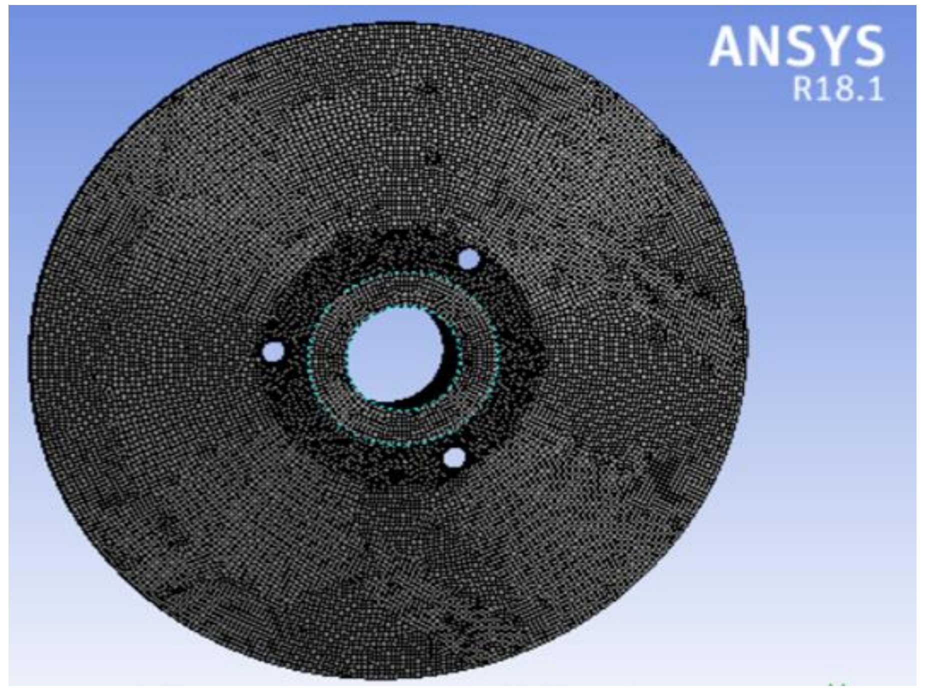
Element quality of Primary Spring Cap

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.80	> 0.6
Jacobian	0.73	> 0.6
Aspect Ratio	3.34	< 10

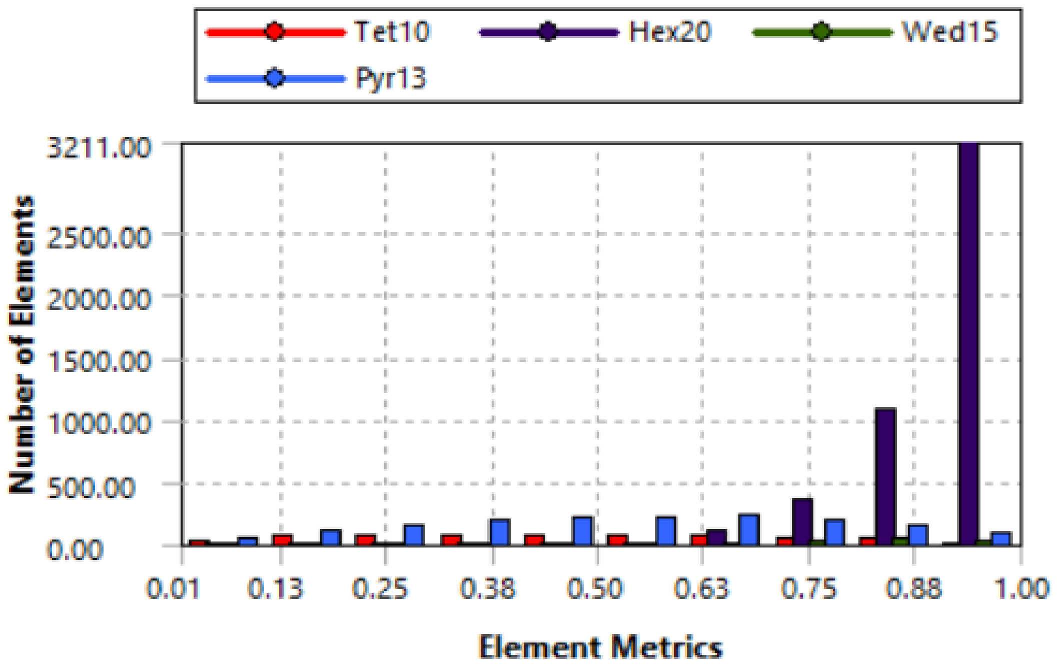
Result and Discussion

FEM Modelling

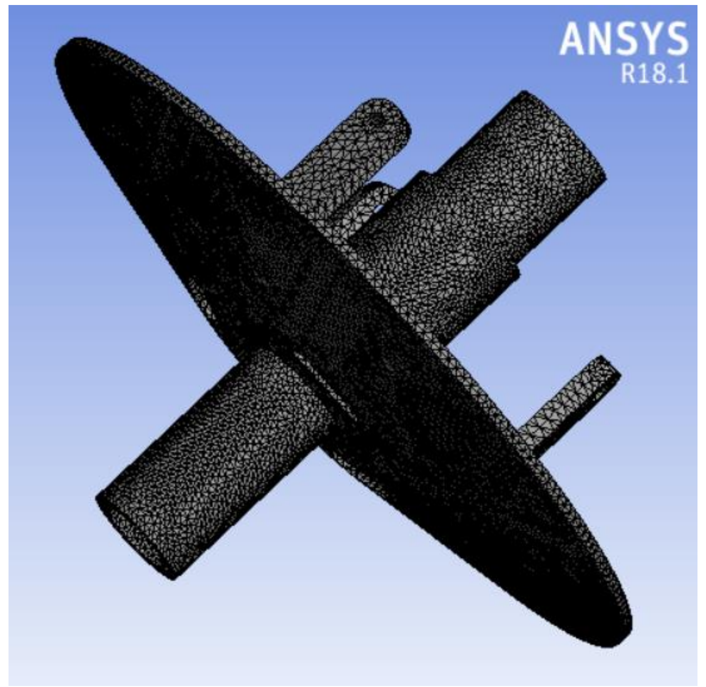
Meshing parameters for the components of the secondary pulley has been chosen based on the complications of the geometry



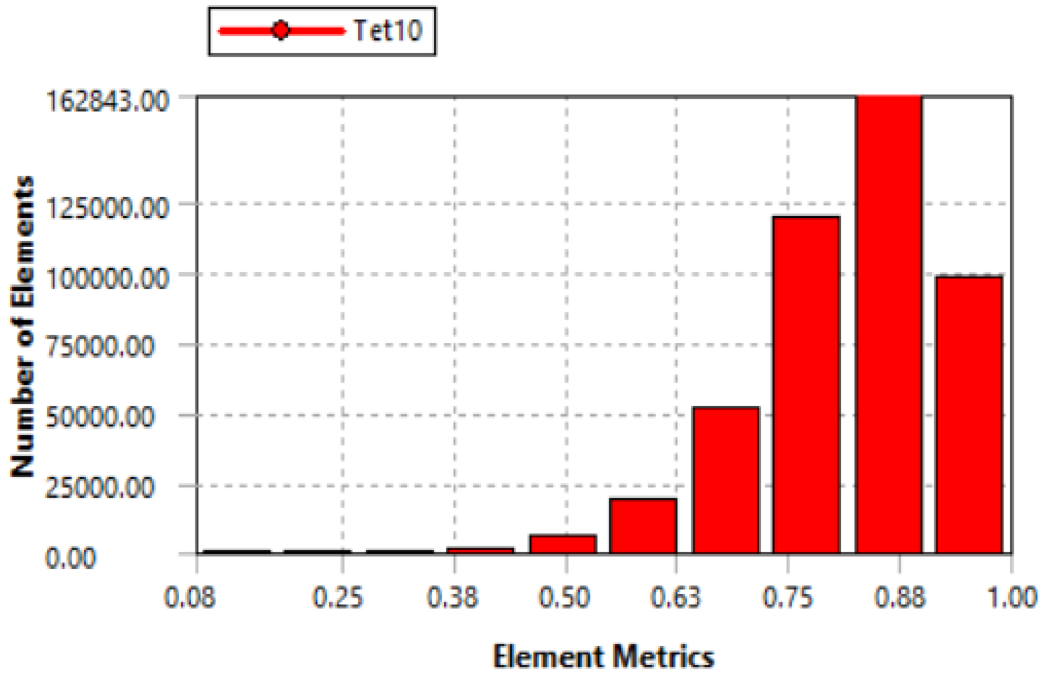
Meshed Secondary Movable Sheave



Element quality of Secondary Movable Sheave



Meshed Secondary Fixed Sheave



Element quality of Secondary Fixed Sheave

Mesh Parameters	
Method	Hex Dominant
Mesh Elements	All Quad
Element Sizing	1 mm

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.76	> 0.6
Jacobian	0.66	> 0.6
Aspect Ratio	3.37	< 10

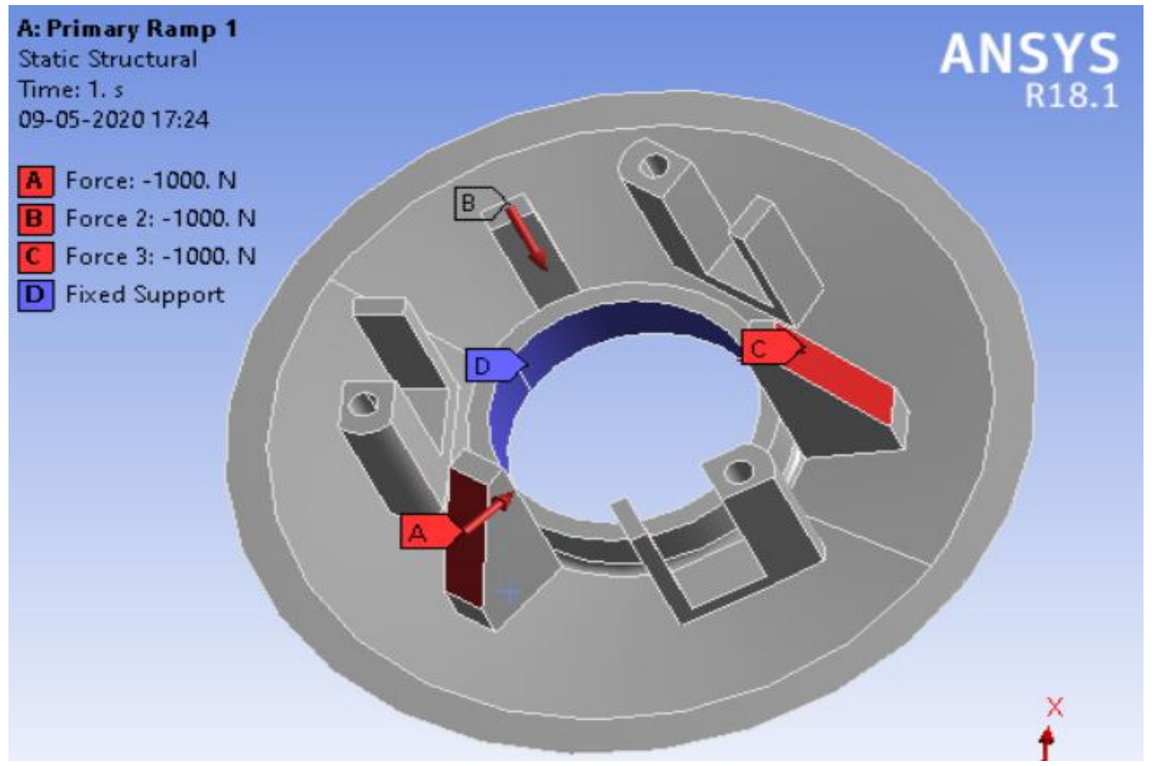
Mesh Parameters	
Method	Adaptive
Mesh Elements	Tetrahedron
Element Sizing	1.5 mm

Mesh Quality Parameter	Average Values	Desirable Values
Element Quality	0.83	> 0.6
Jacobian	0.99	> 0.6
Aspect Ratio	1.91	< 10

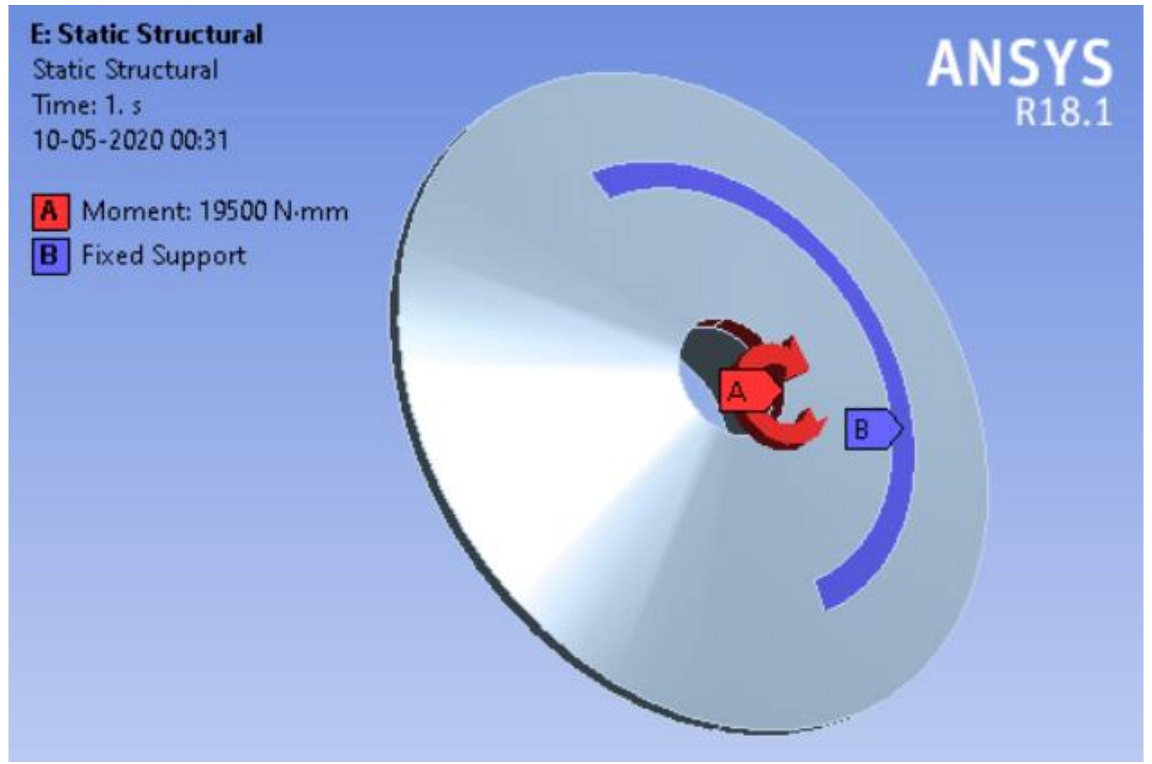
Result and Discussion

FEM Modelling

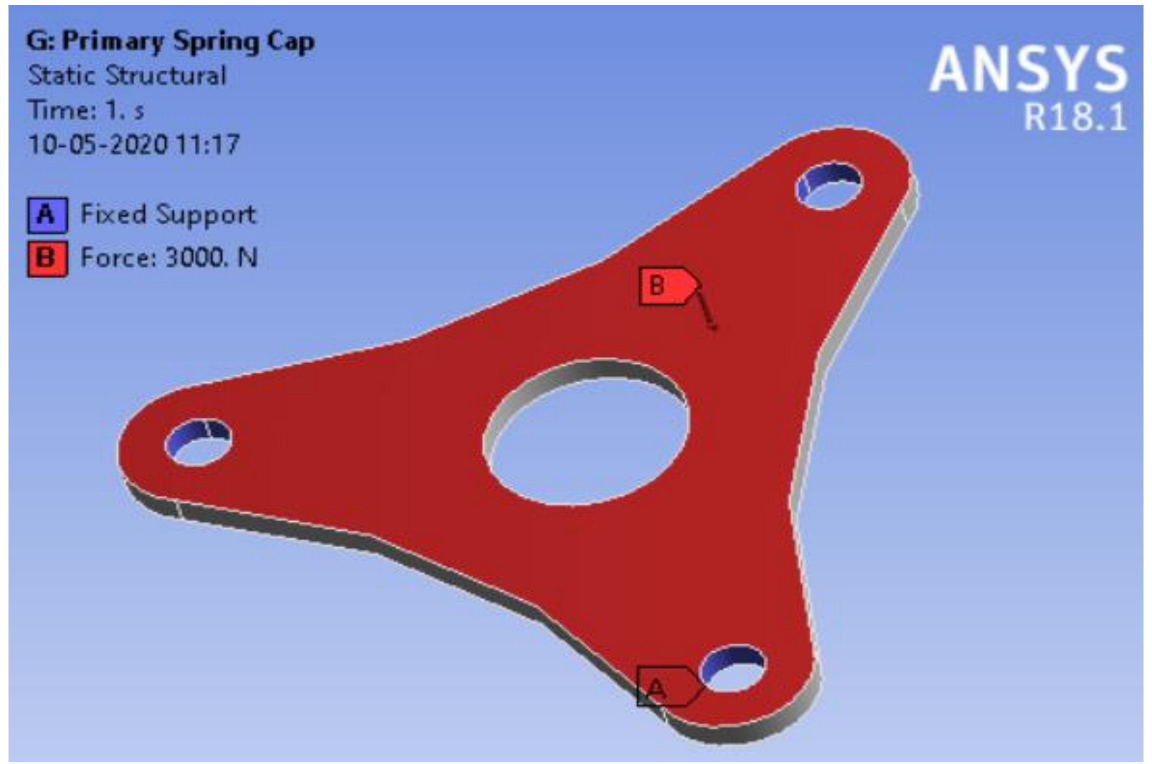
The boundary conditions for the components of the primary and secondary pulley



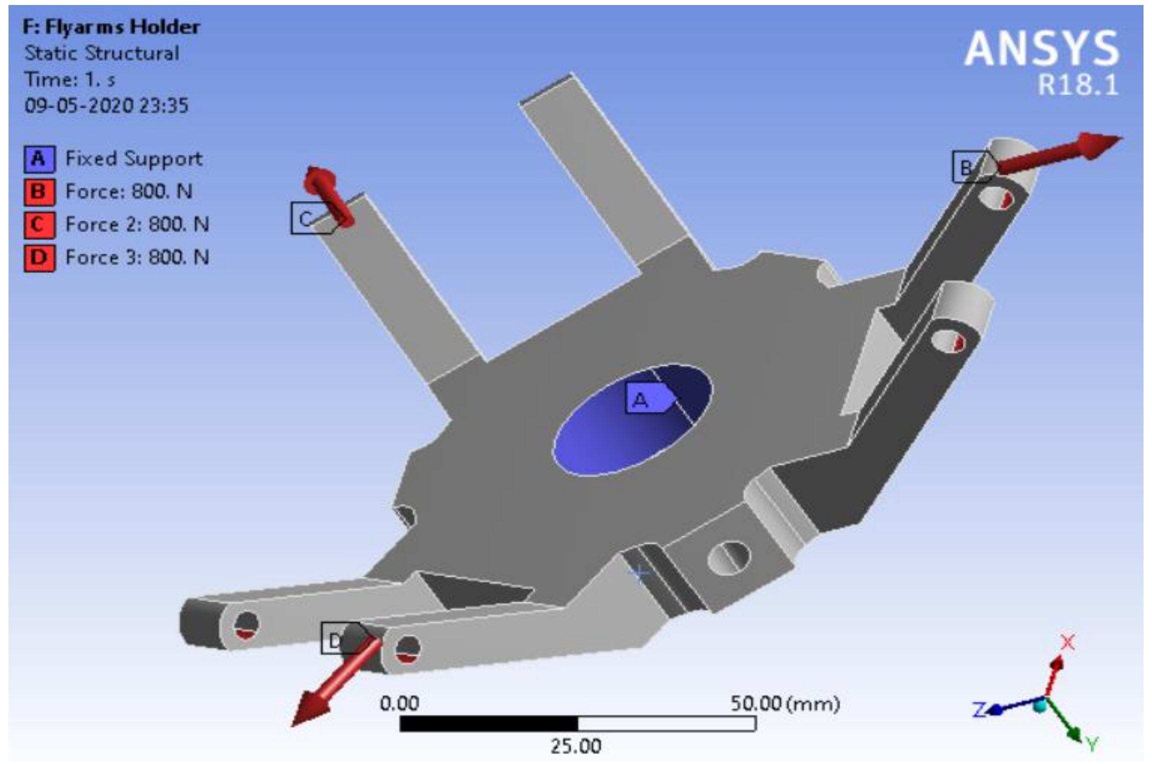
Primary Ramp Plate



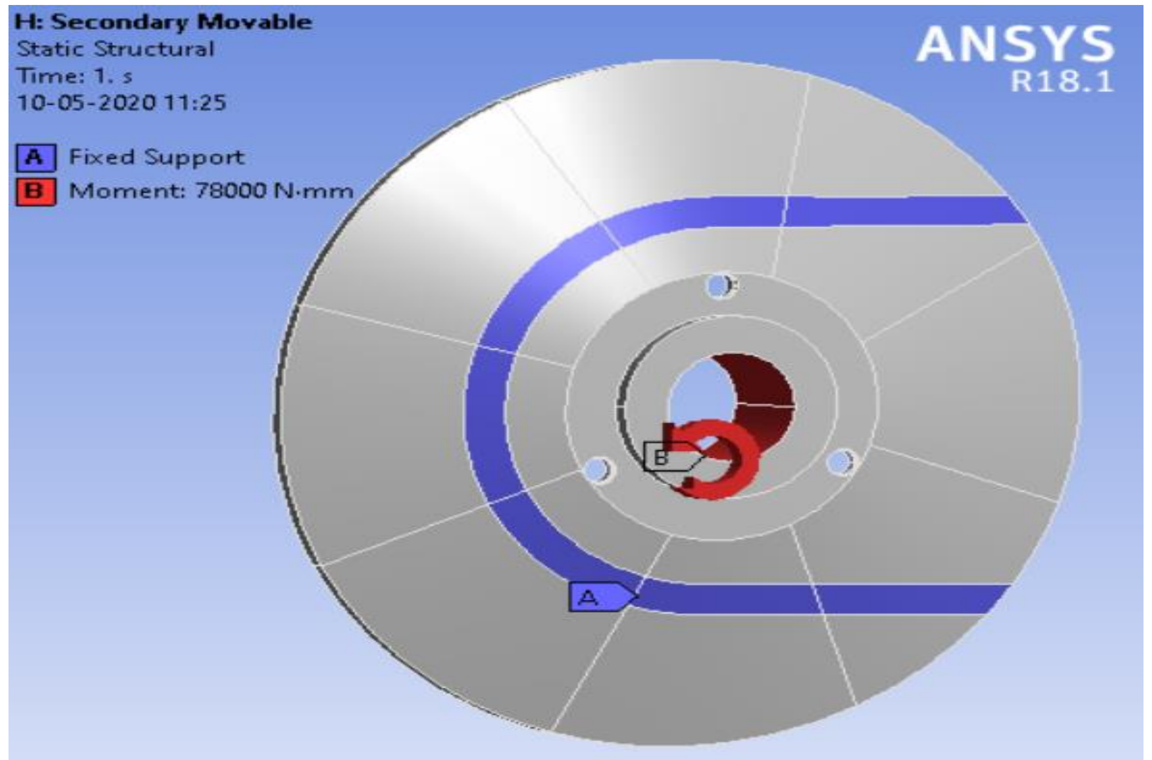
Primary Movable Sheave



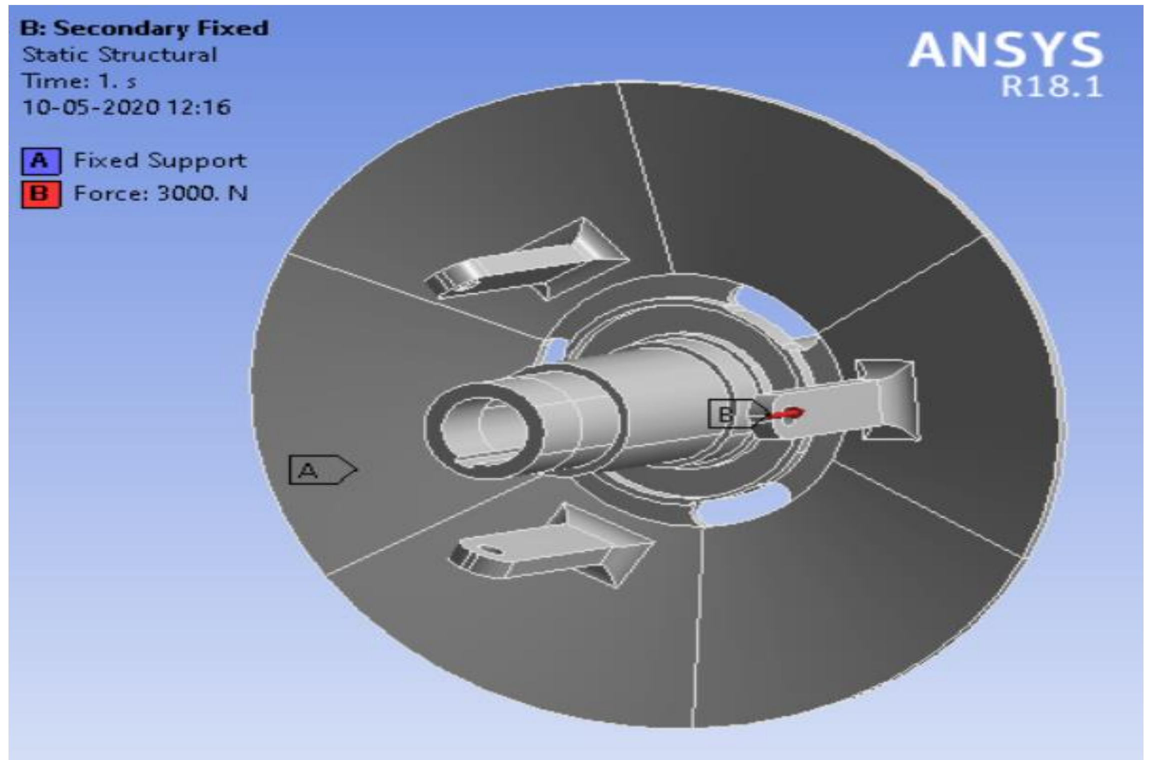
Primary Spring Cap



Primary Fly-Arms Holder



Secondary Movable Sheave

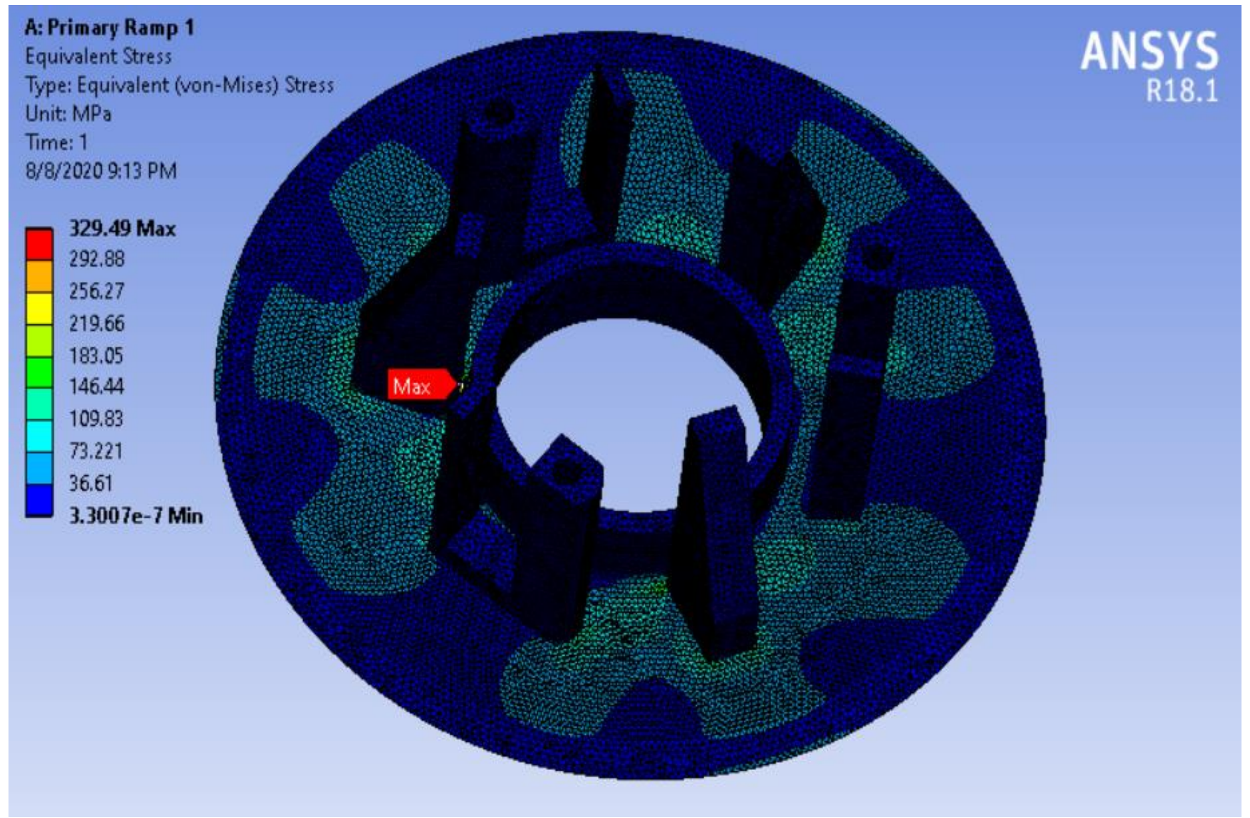


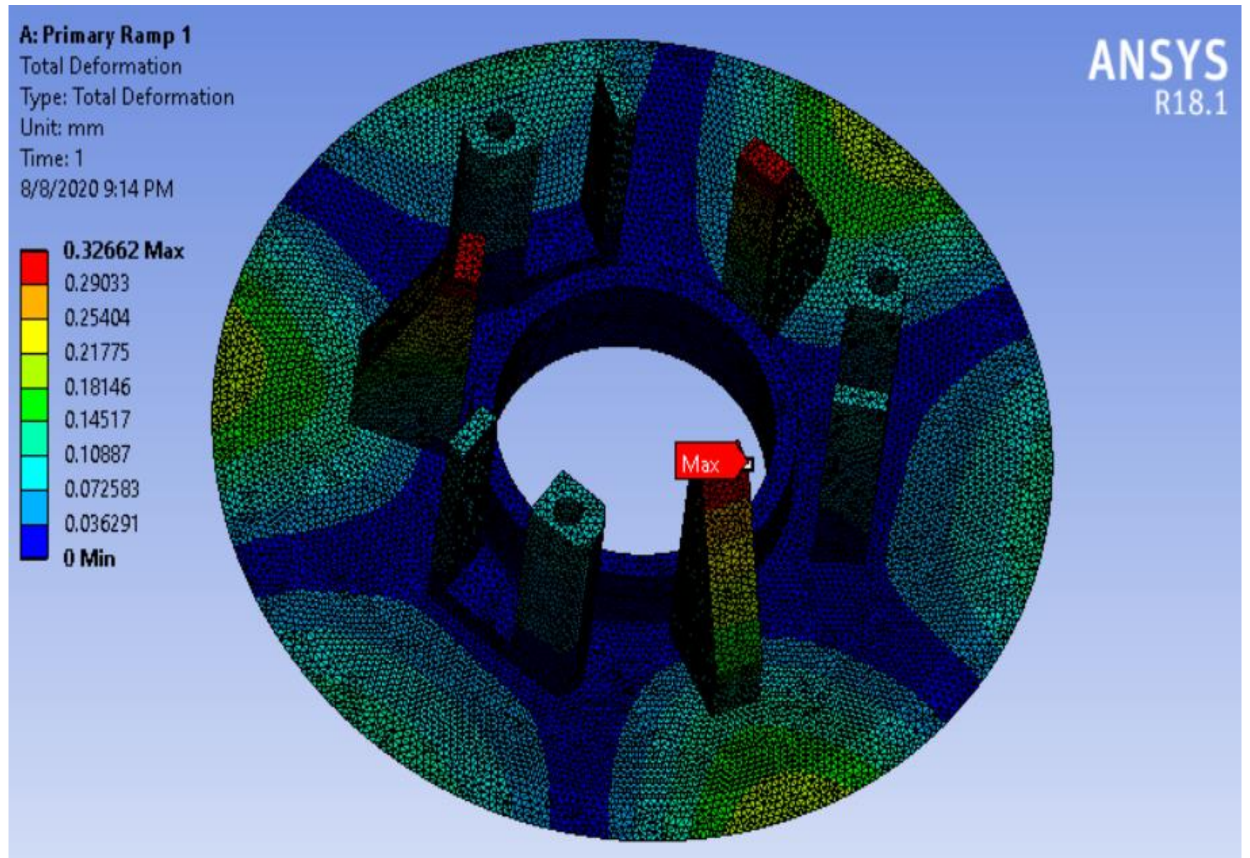
Secondary Fixed Sheave

Result and Discussion

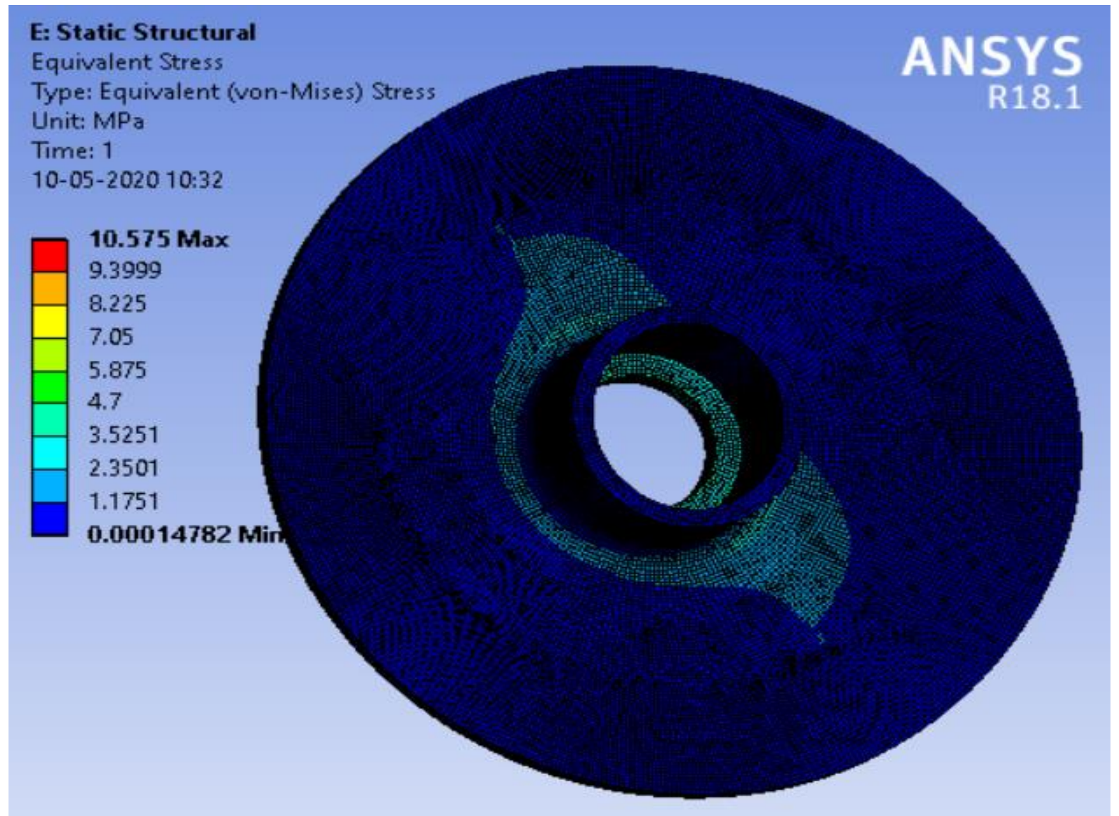
FEM Results

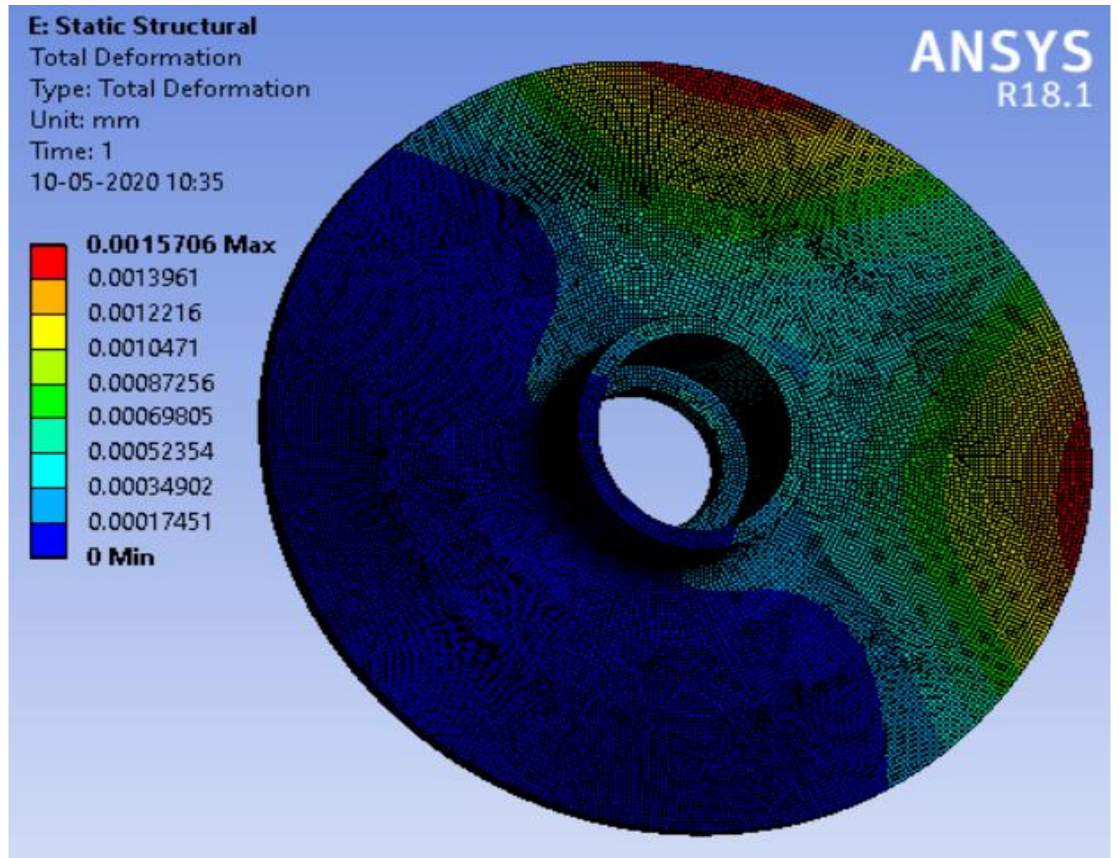
Primary Ramp Plate



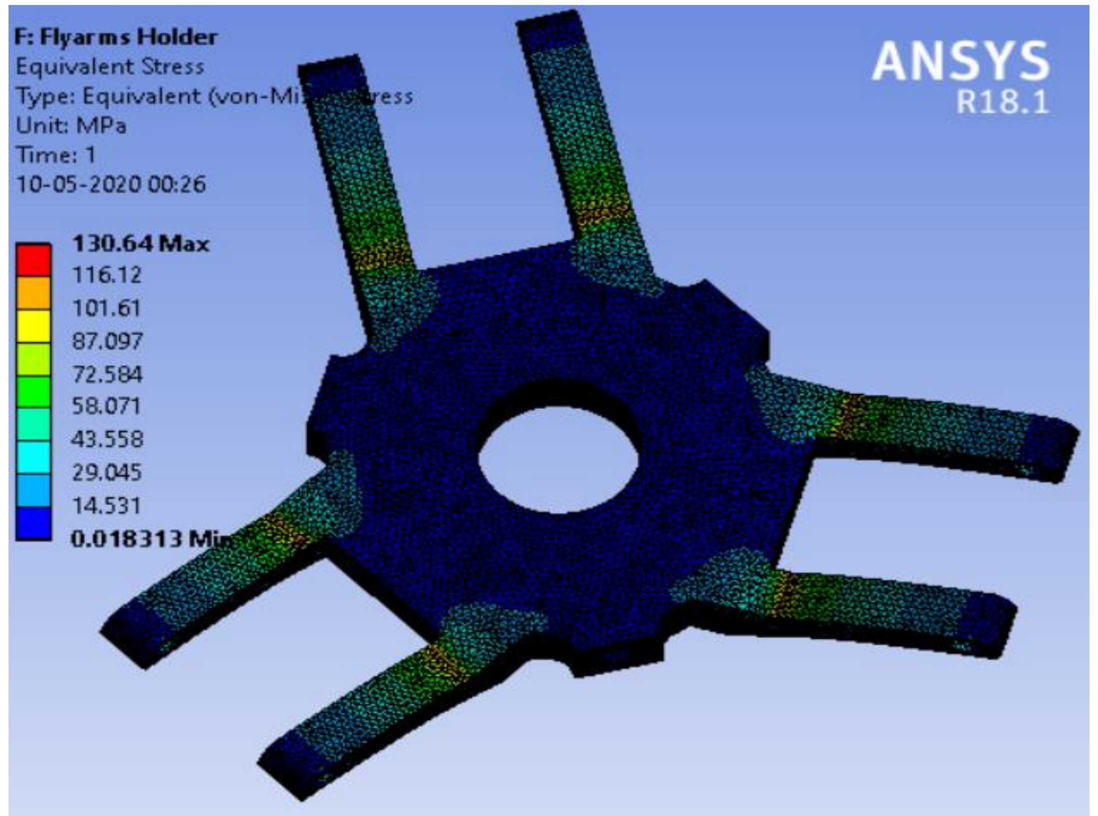


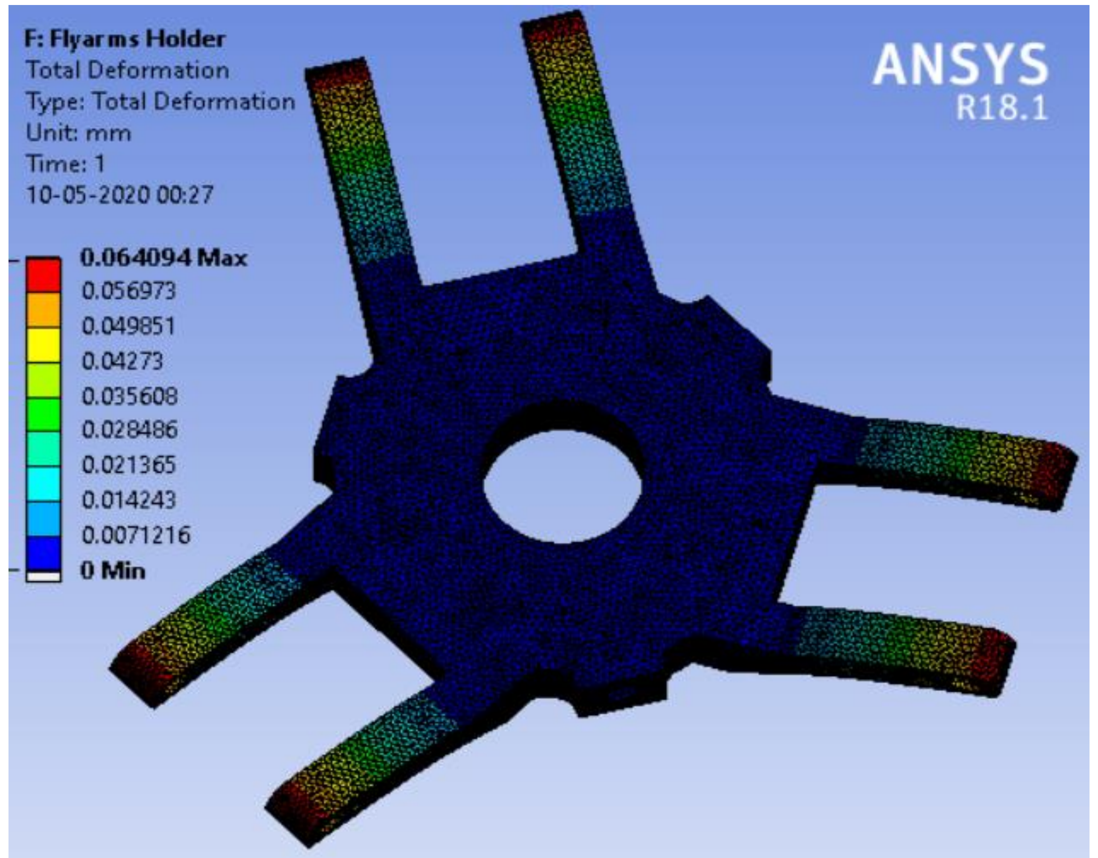
Primary Movable Sheave





Primary Fly-Arms Holder

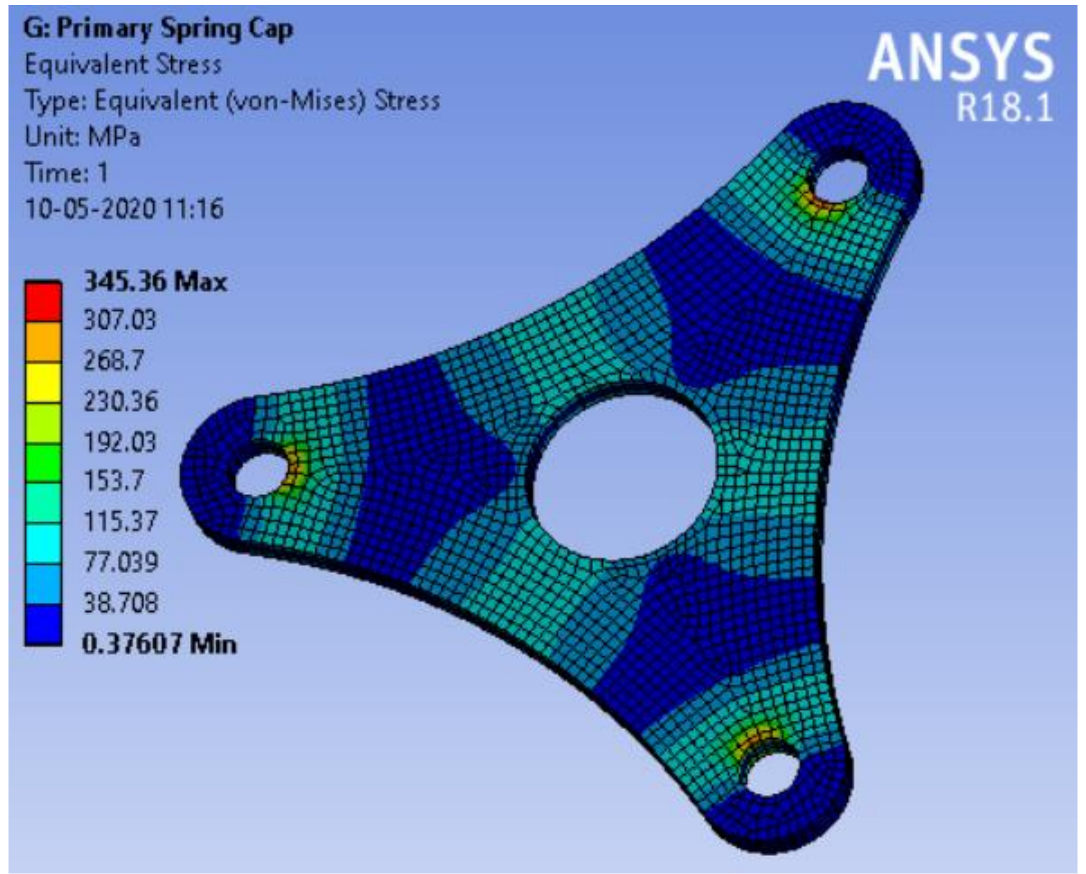


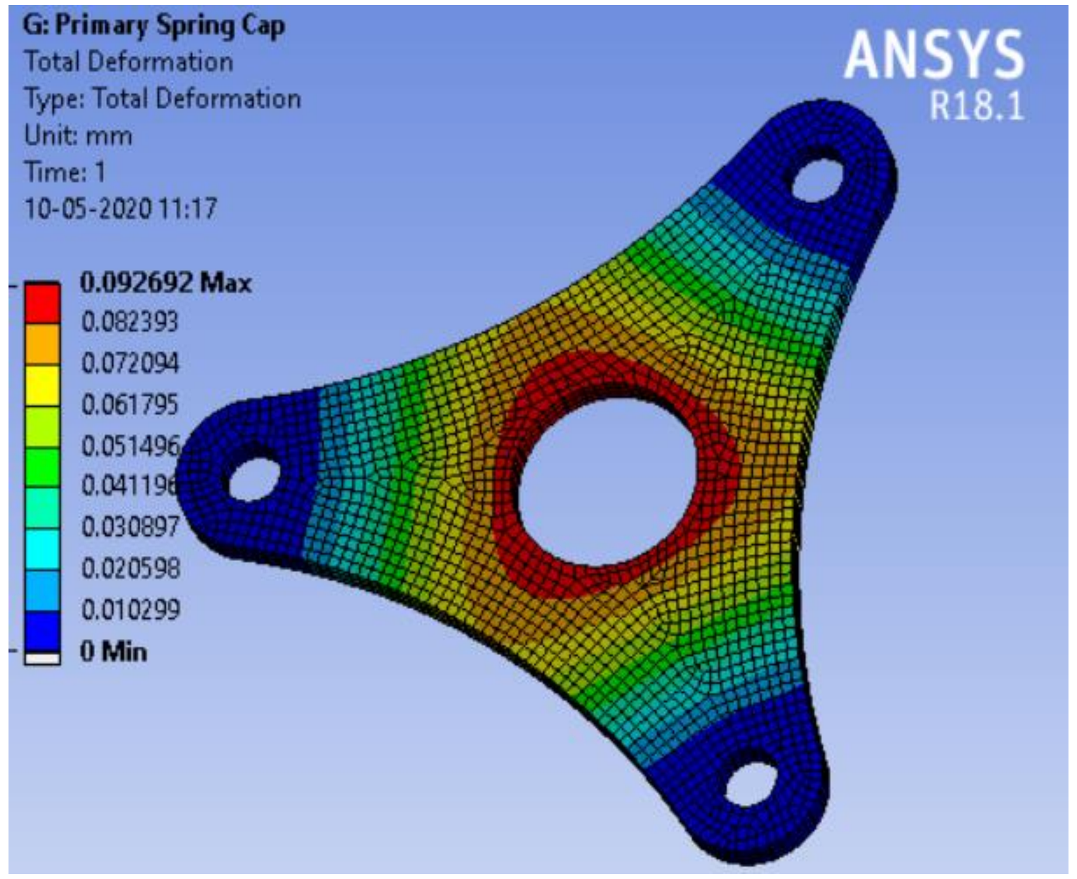


Result and Discussion

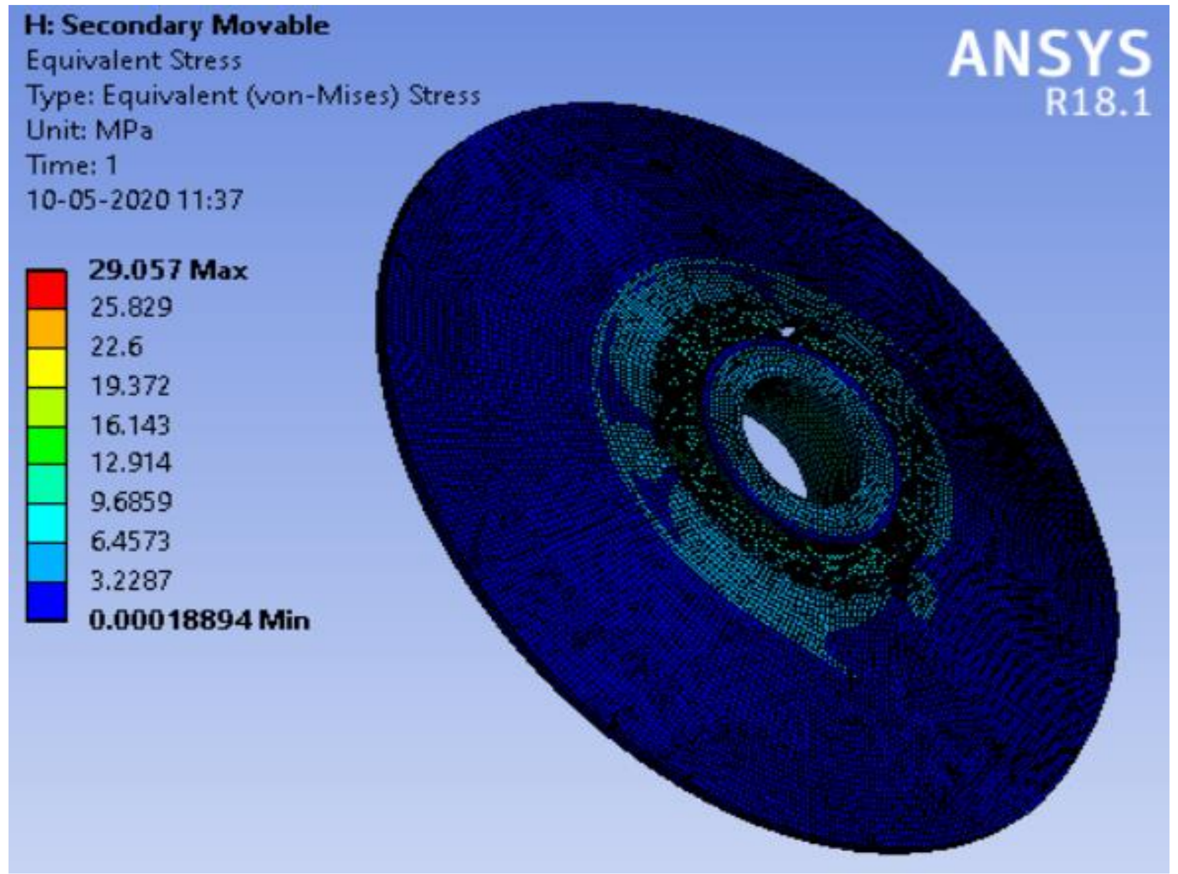
FEM Results

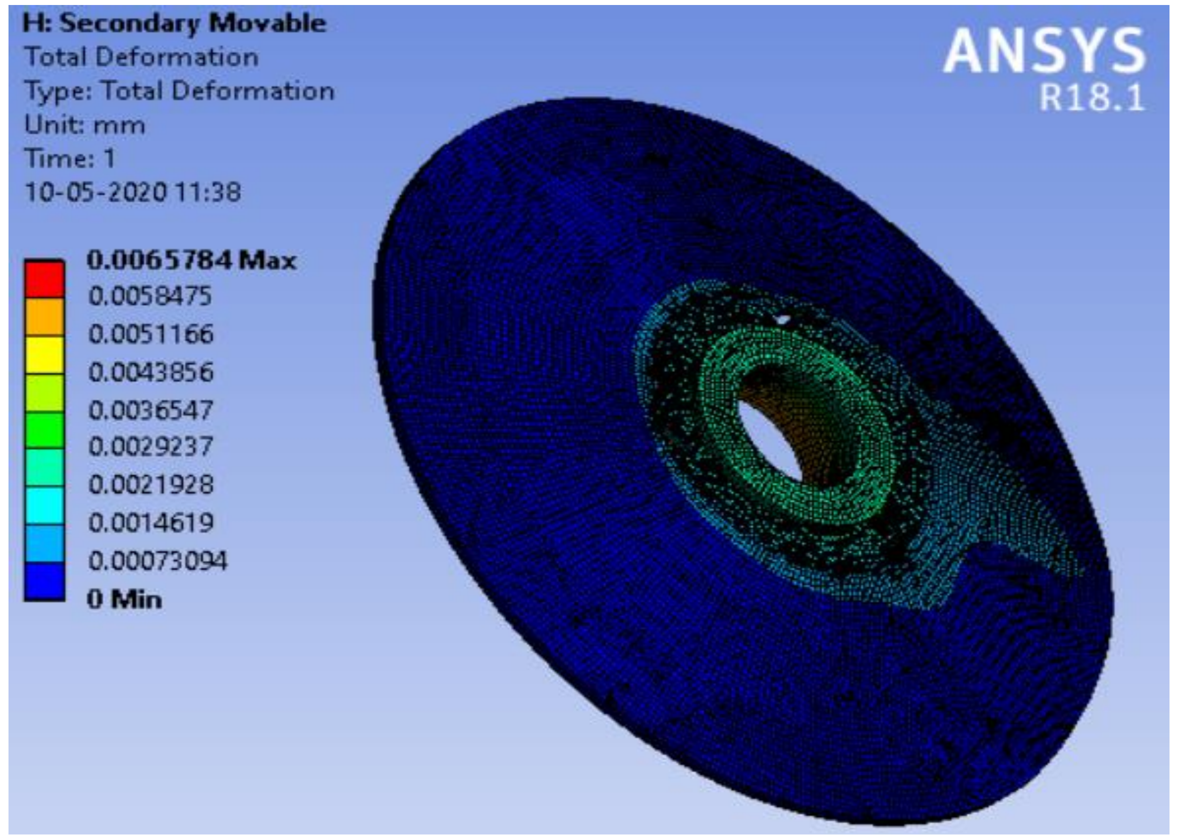
Primary Spring Cap



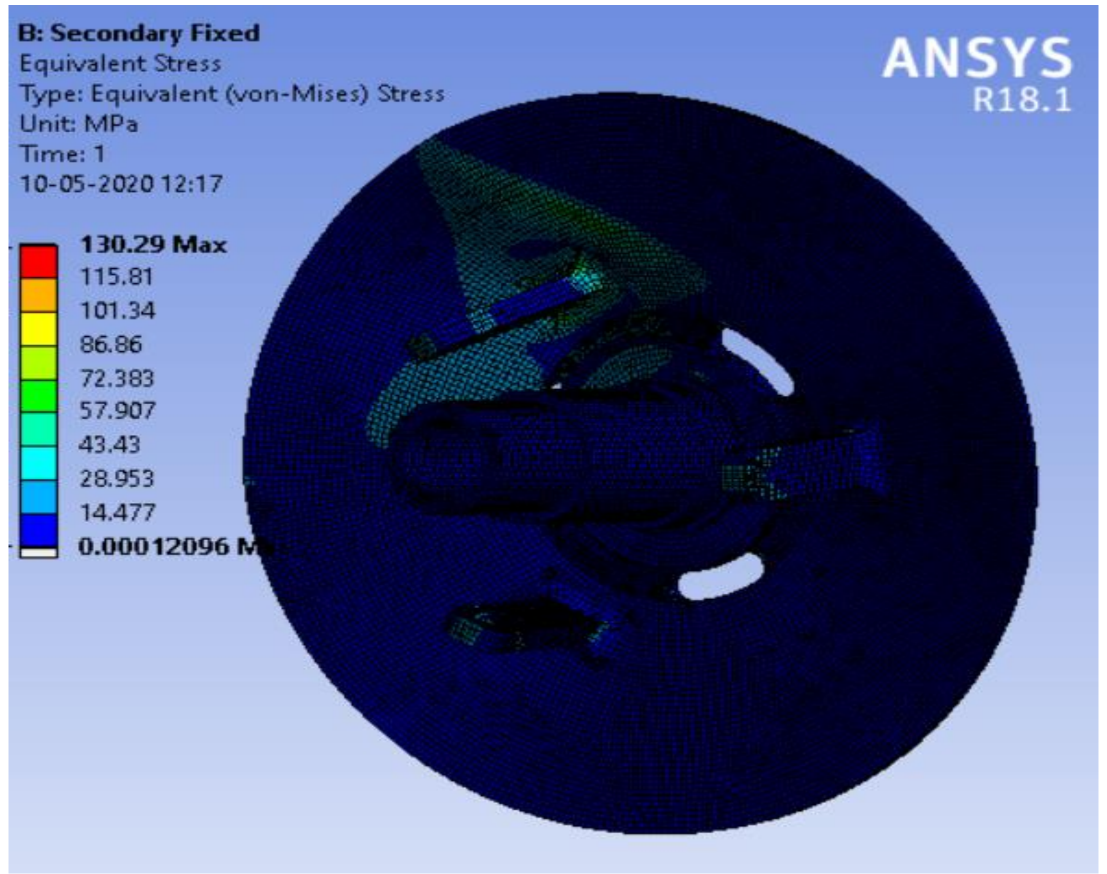


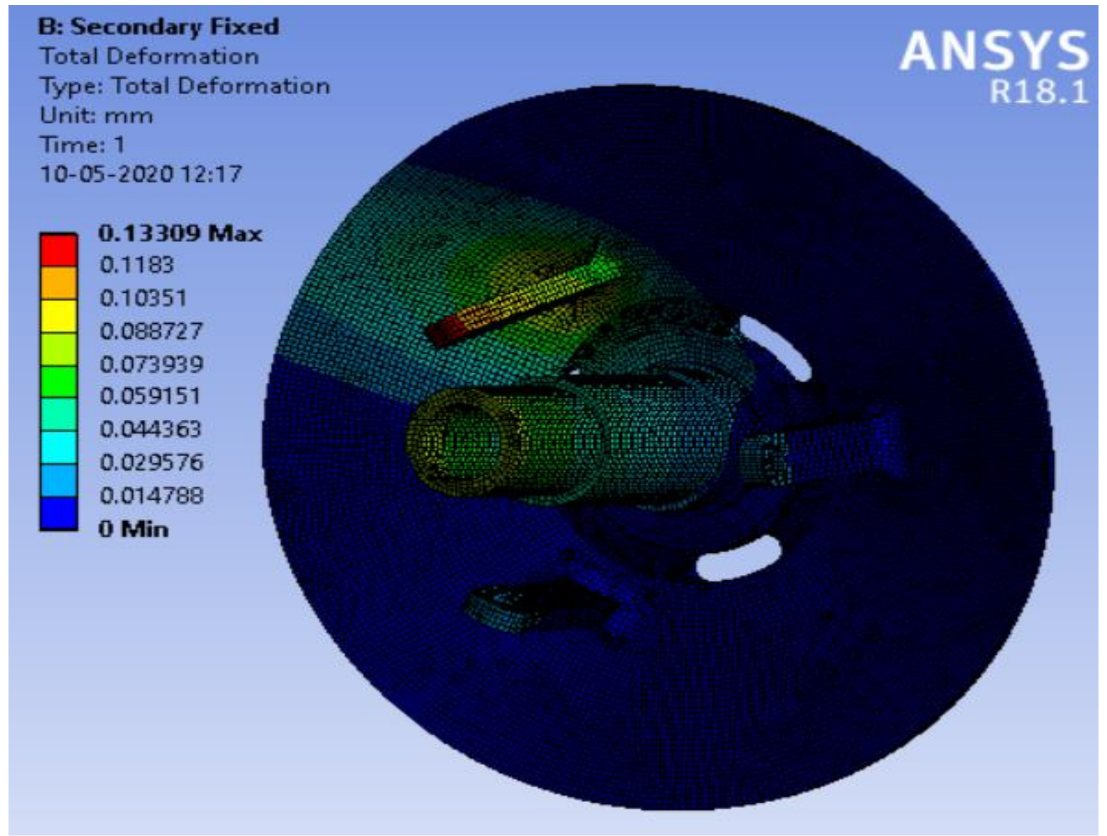
Secondary Movable Sheave





Secondary Fixed Sheave



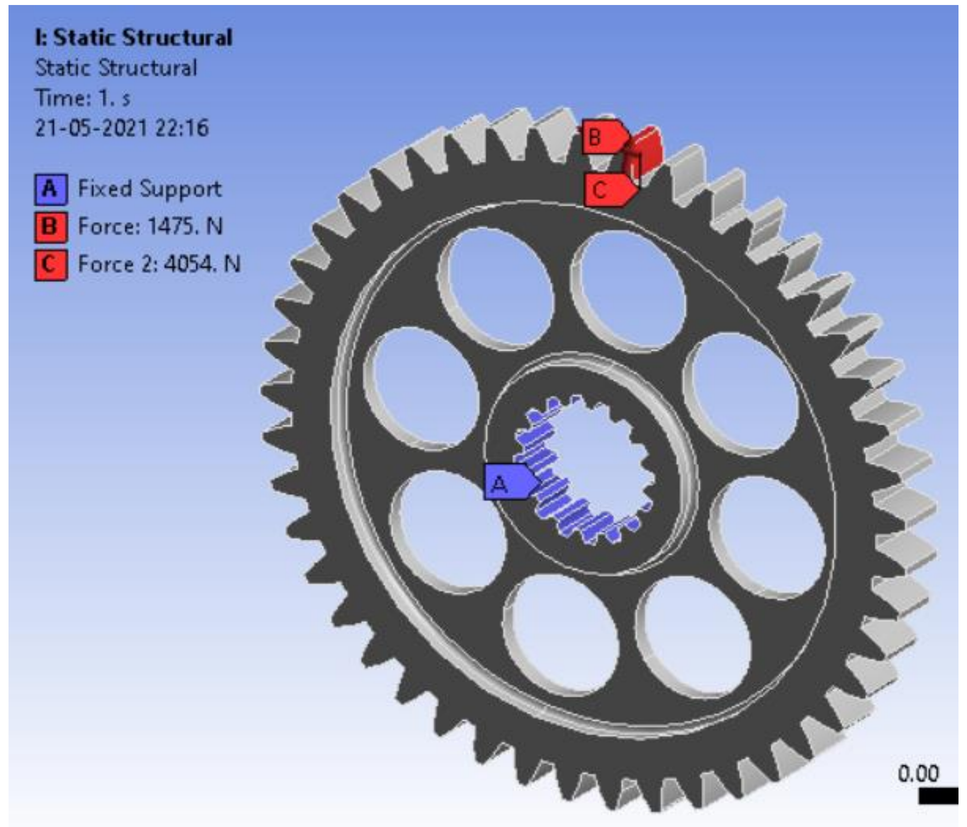


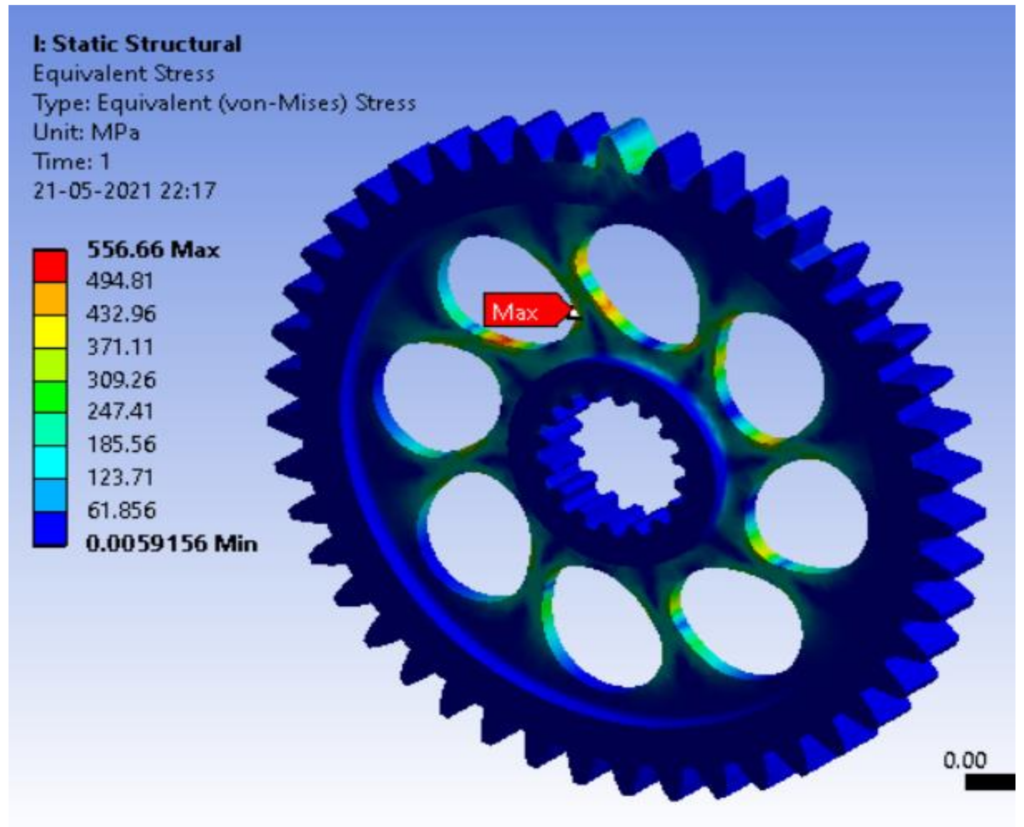
Result and Discussion

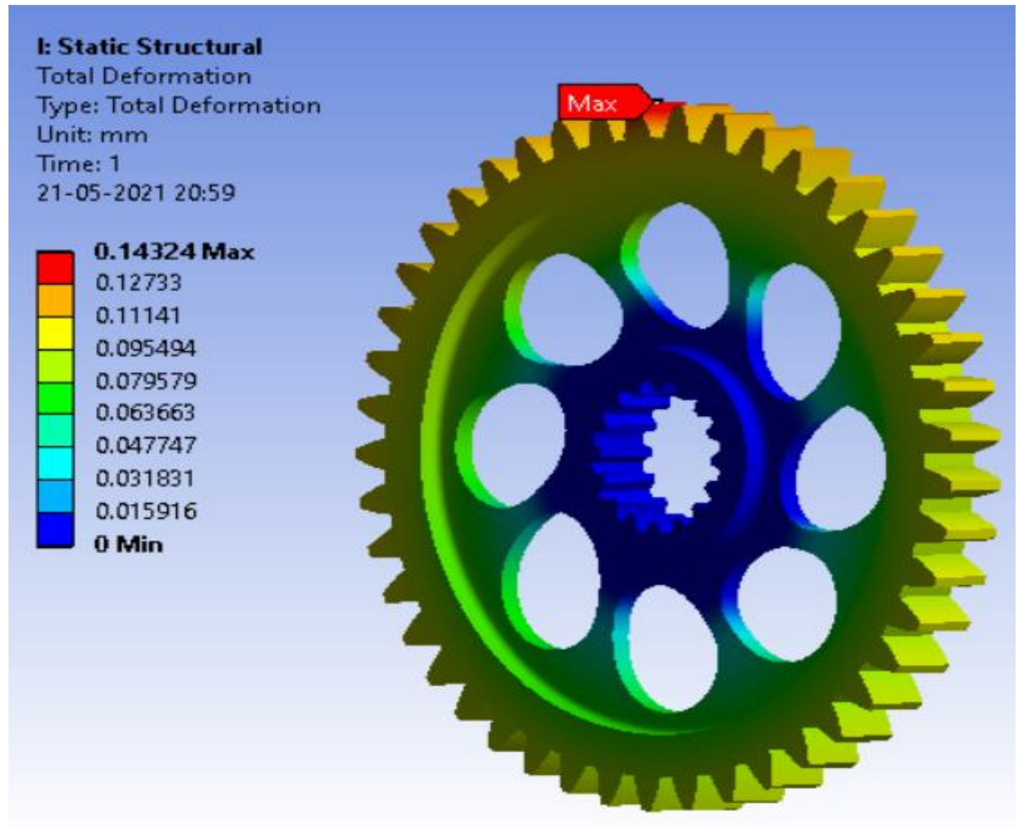
Components	Material	Max Stress (MPa)	Max Deformation (mm)	FOS
Primary ramp plate	Al 7075-T6	329.49	0.035	1.58
Primary movable sheave	Al 7075-T6	10.575	0.0159	49.17
Primary fly arm holder	Al 7075-T6	130.64	0.025	3.98
Primary spring cap	EN 24	345.36	0.0302	2.46
Secondary movable sheave	Al 7075-T6	130.29	0.133	3.99
Secondary fixed sheave	Al 7075-T6	29.05	0.0066	17.91

Result and Discussion

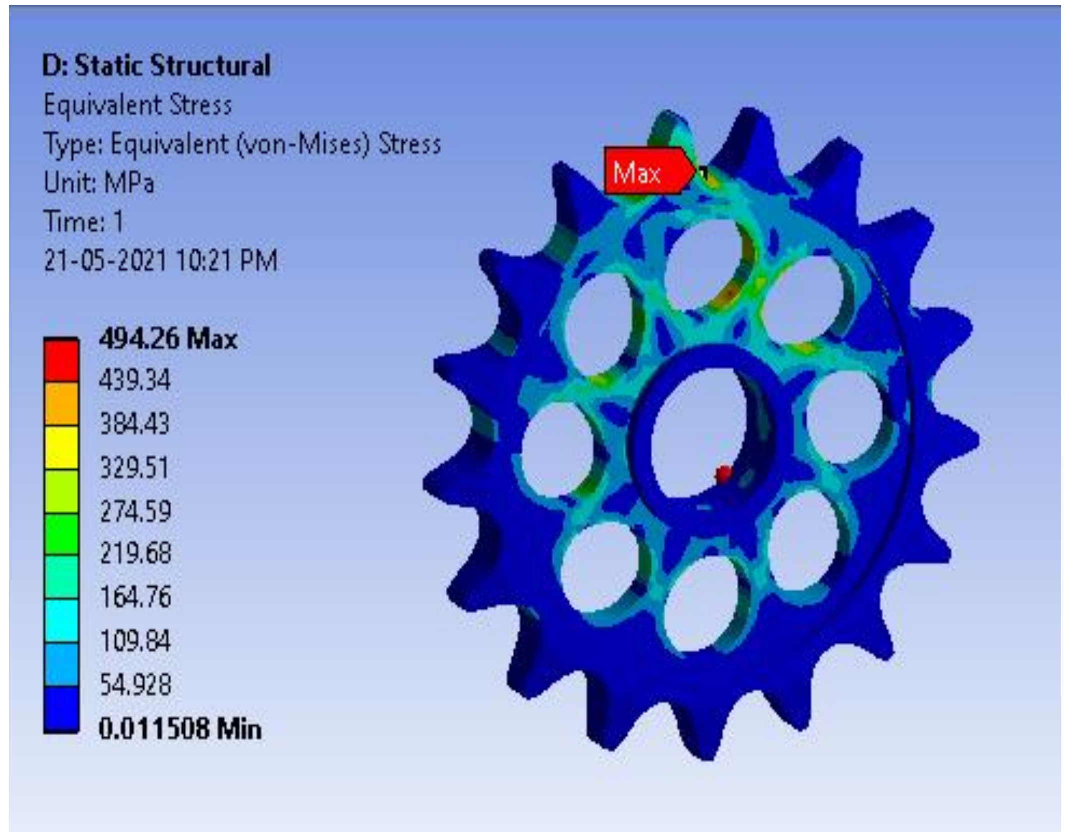
- Gear analysis(Gear 1)

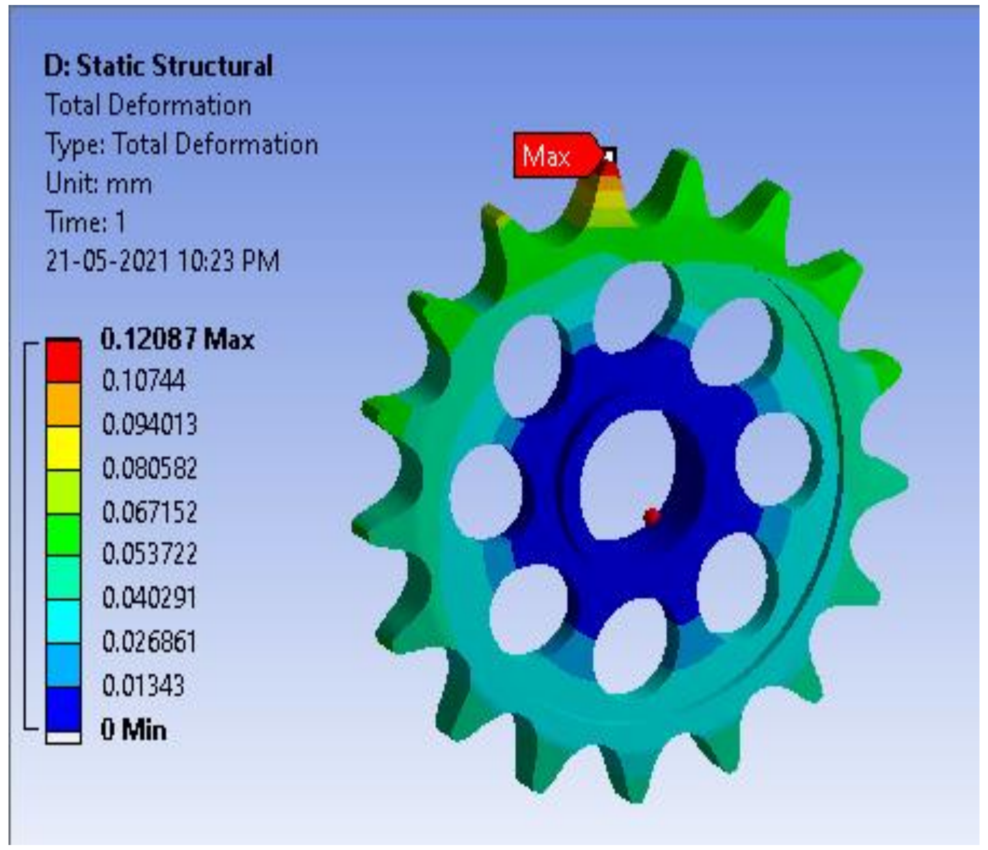






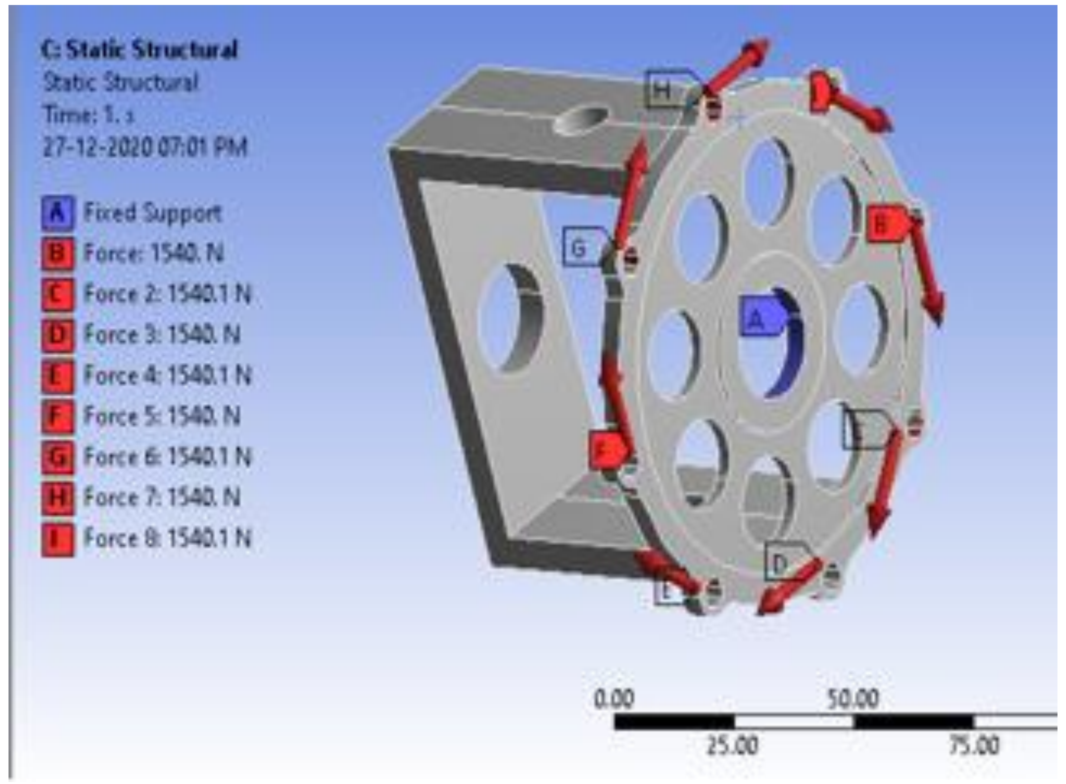
- Sprocket analysis

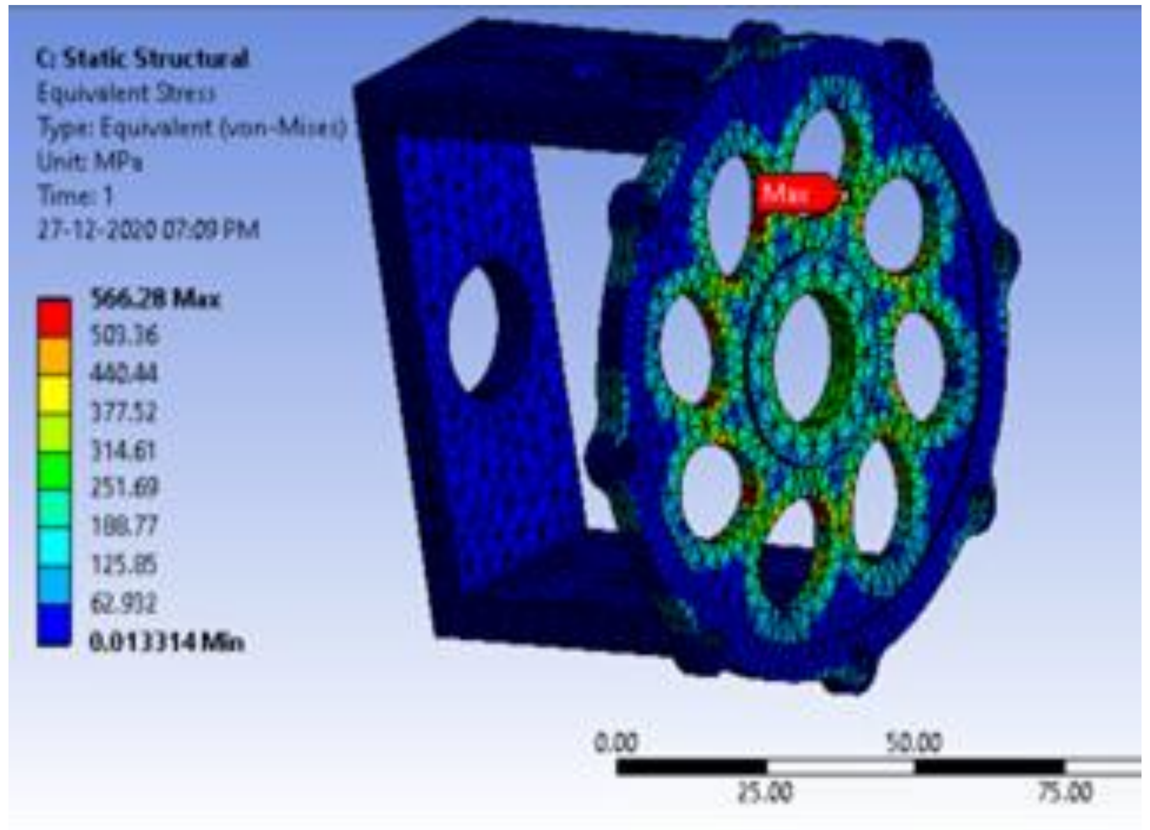


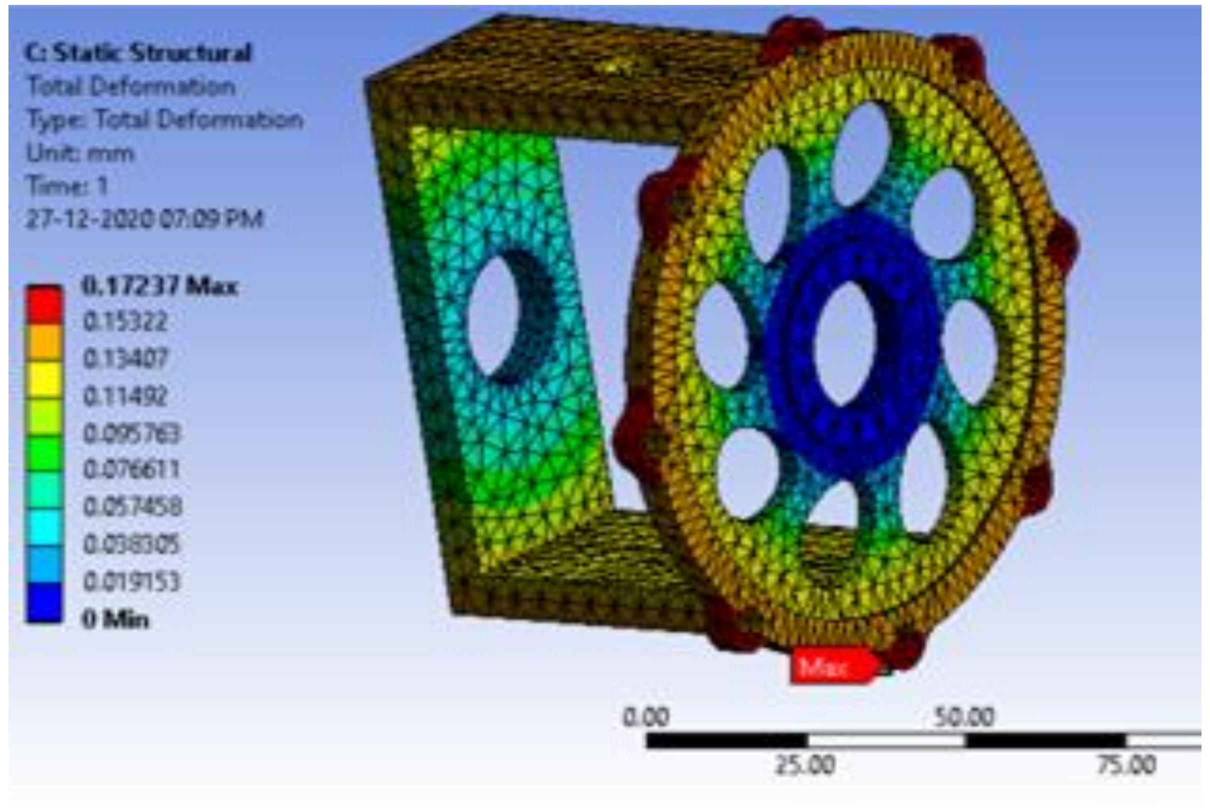


Result and Discussion

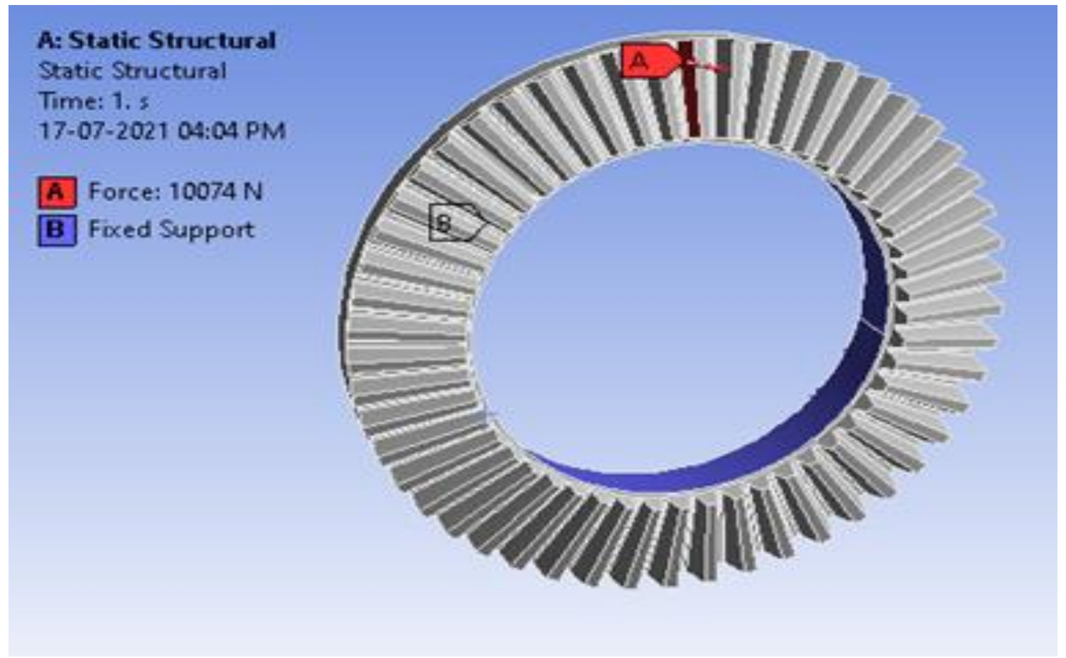
Analysis of carriage

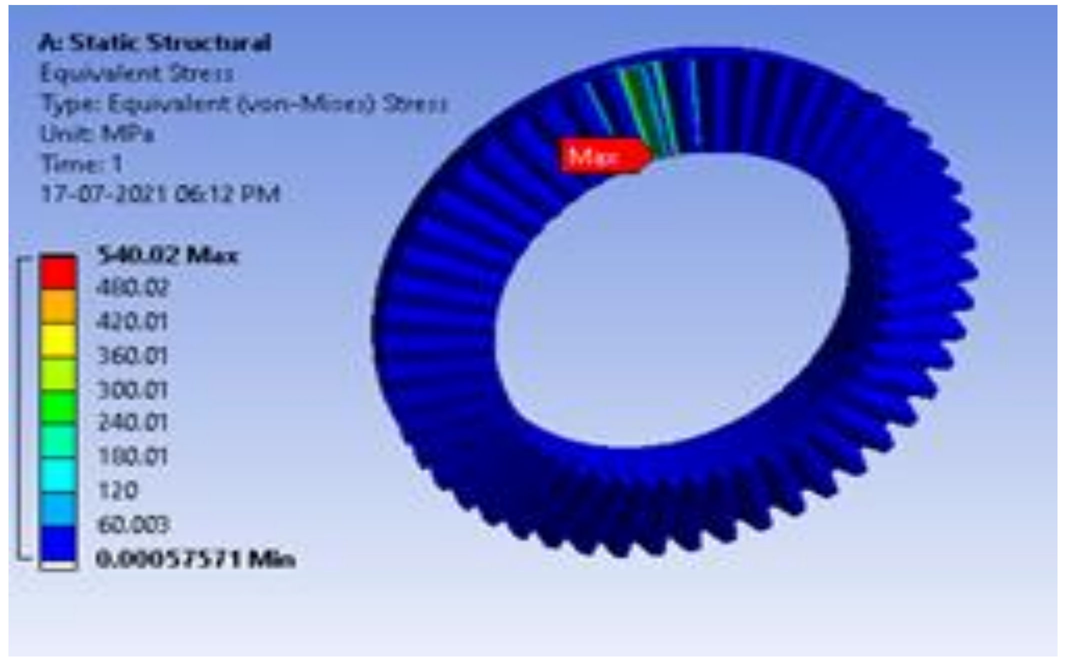


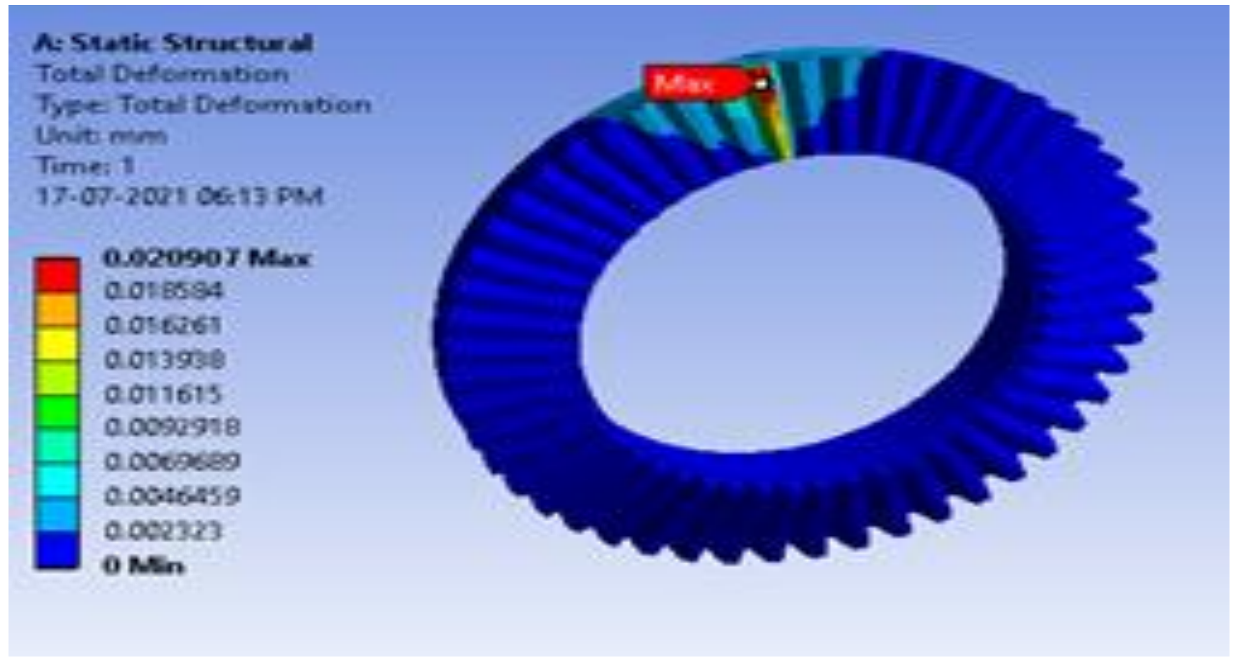




Analysis of bevel gear



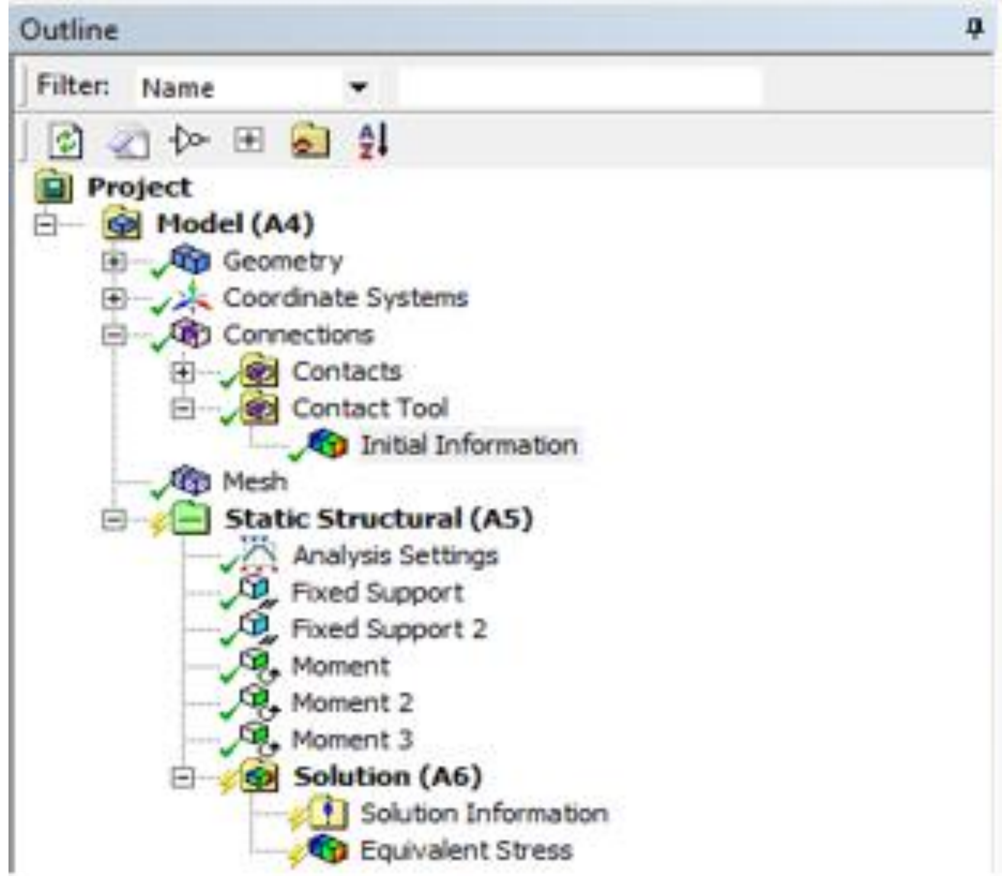


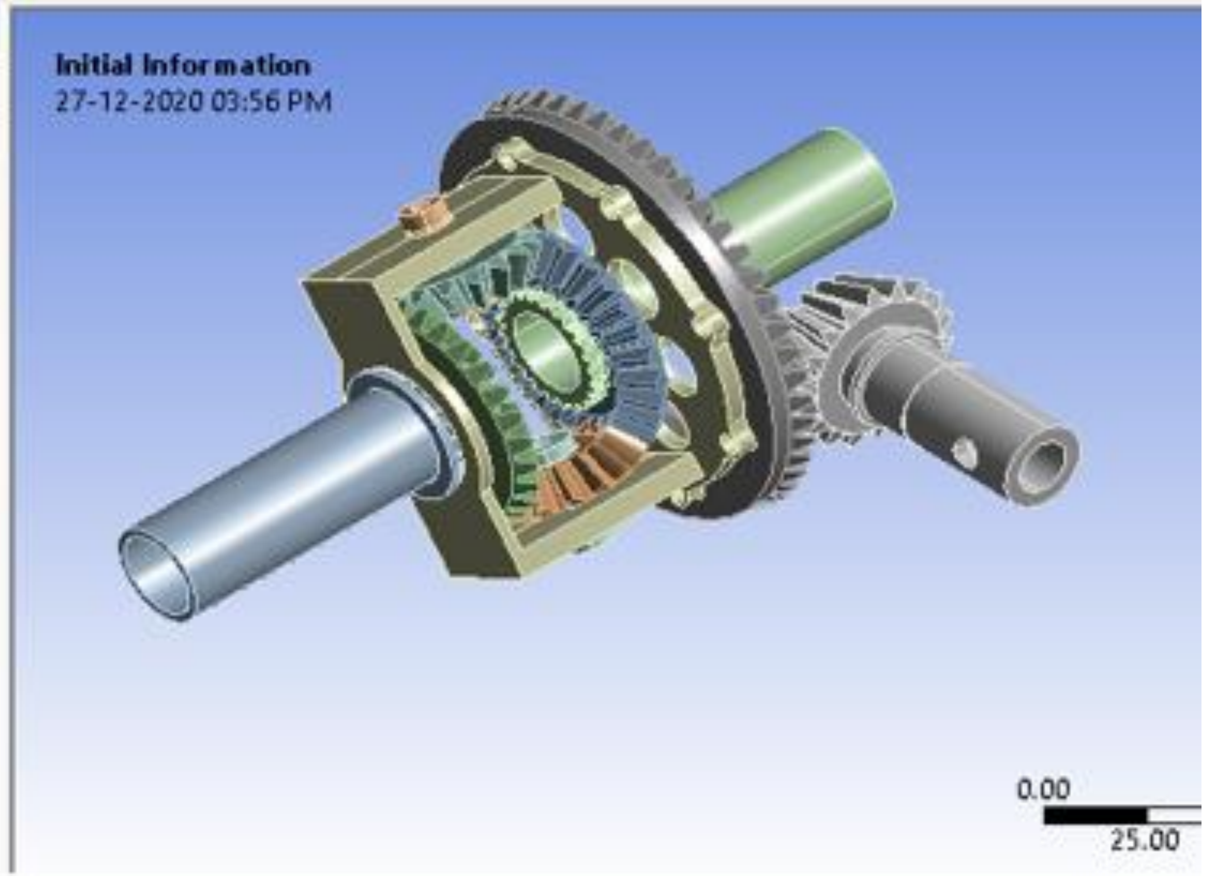


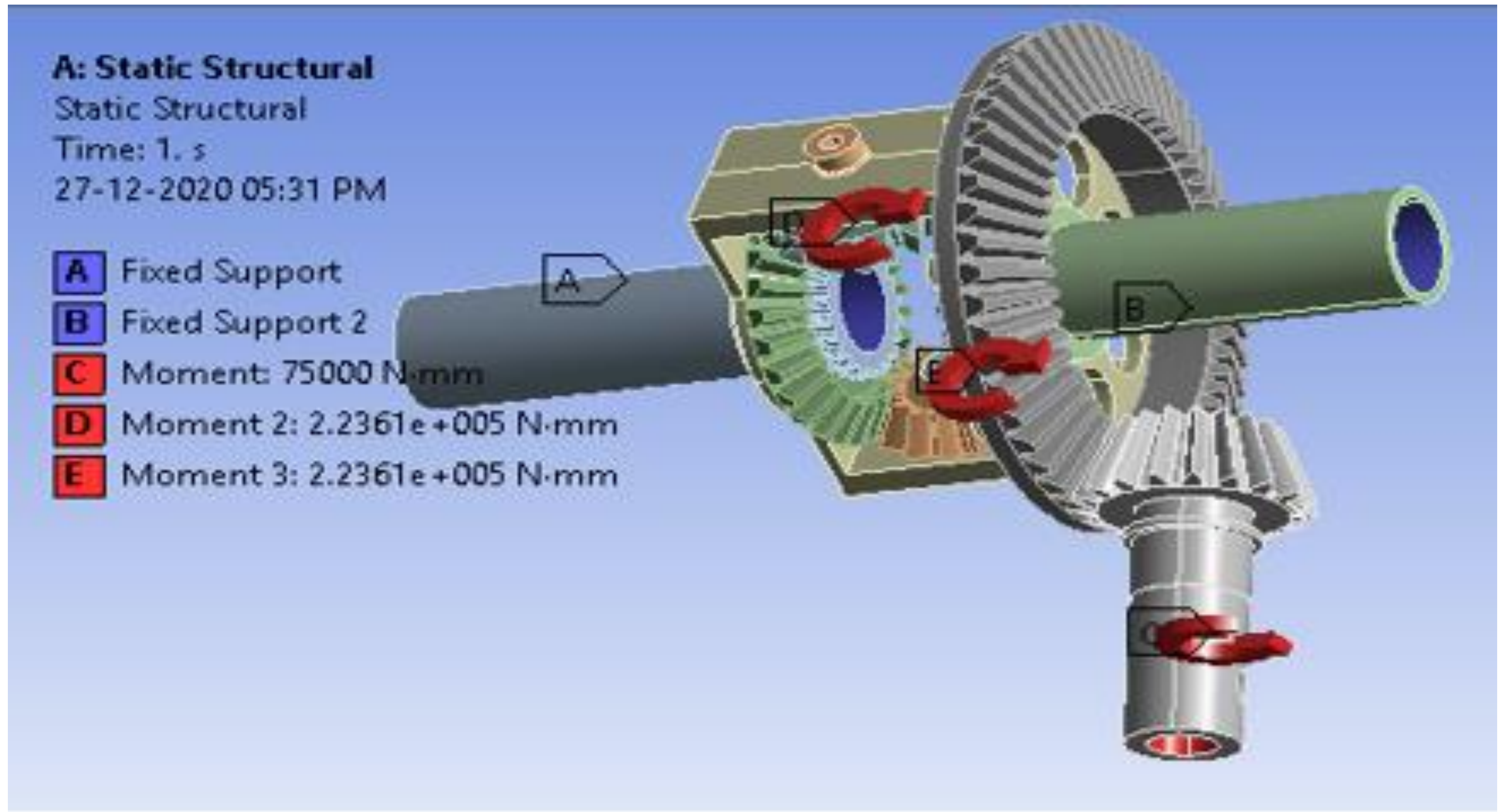
Result and Discussion

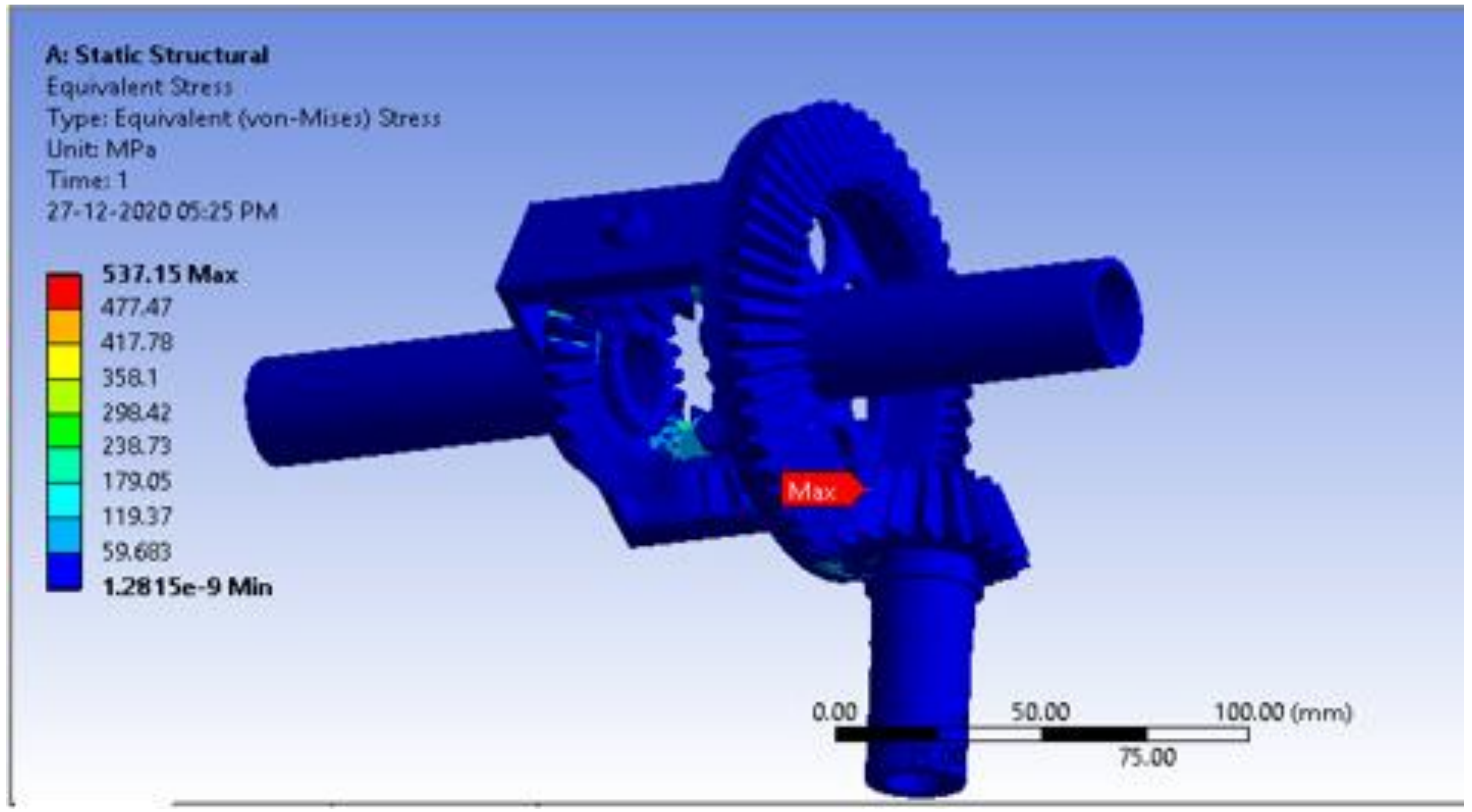
Components	Material	Max Stress (MPa)	Max Deformation (mm)	FOS
Gear 1	EN 36	556.66	0.143	1.61
Gear 2	EN 36	567.23	0.107	1.58
Sprocket	EN 36	494.26	0.121	1.82
Carriage	EN 24	566.28	0.18	1.51
Crown gear	EN 36	540.02	0.020	1.74

Assembly Analysis









- # Halfshaft

Calculation of force for half-shaft

Torque from engine=19.2 Nm.

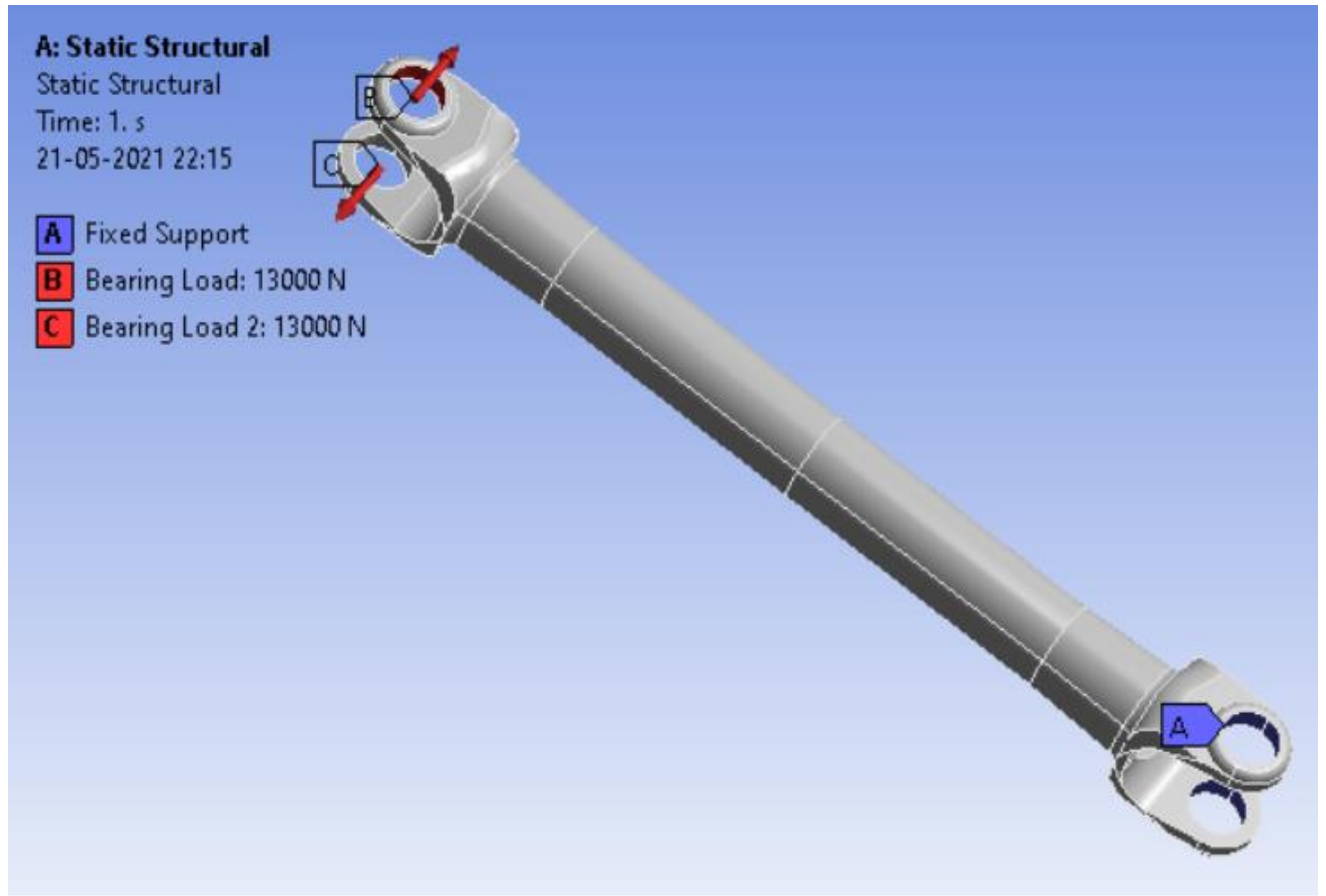
Overall Reduction=33.96.

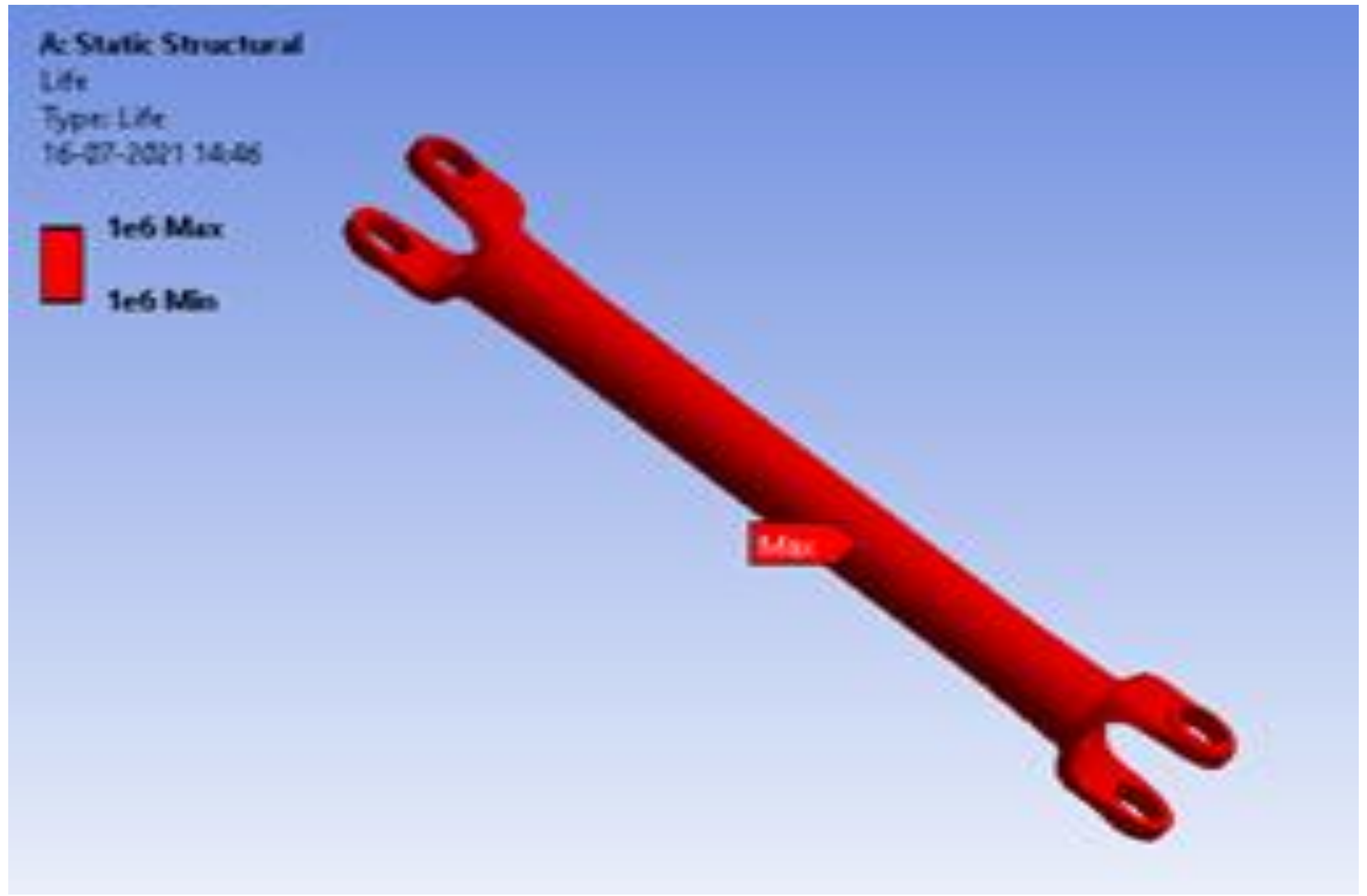
Centre to centre between cross bearing=49.85 mm.

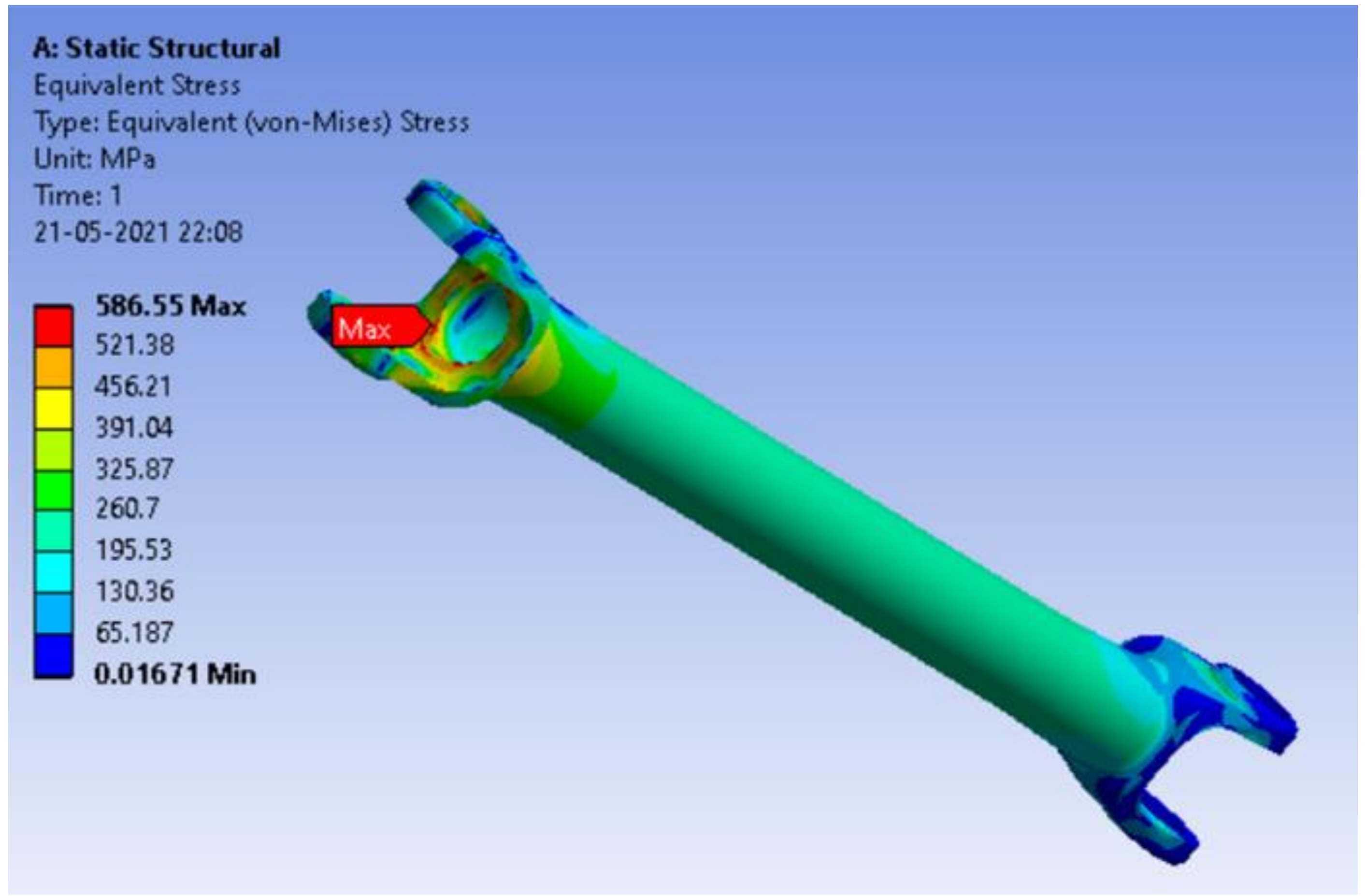
Force on the half-shaft=(19.2*33.96)/0.04985

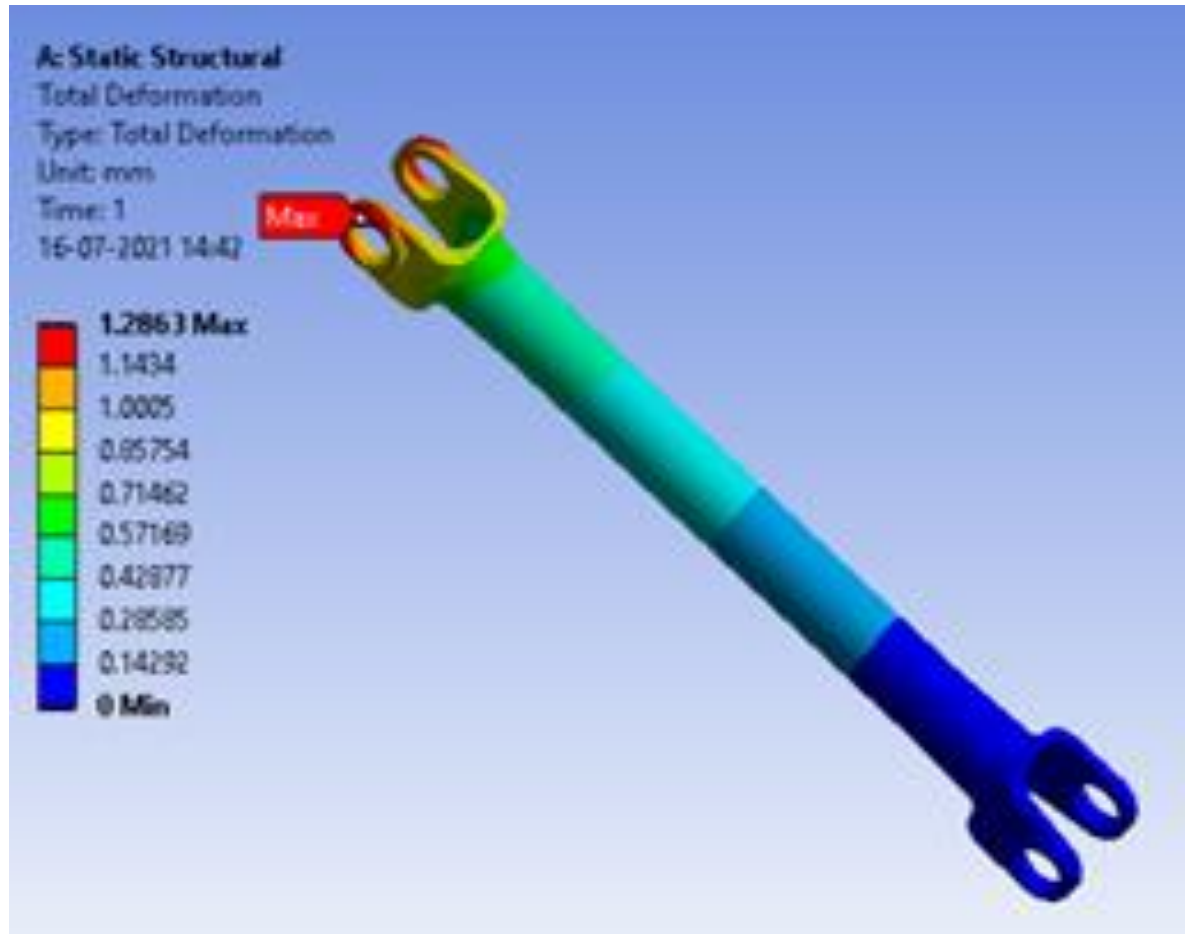
=13000 N

For fatigue analysis, we have considered average load.









Calculation of force for inboard yoke with and without adaptor

Torque from engine=19.2 Nm.

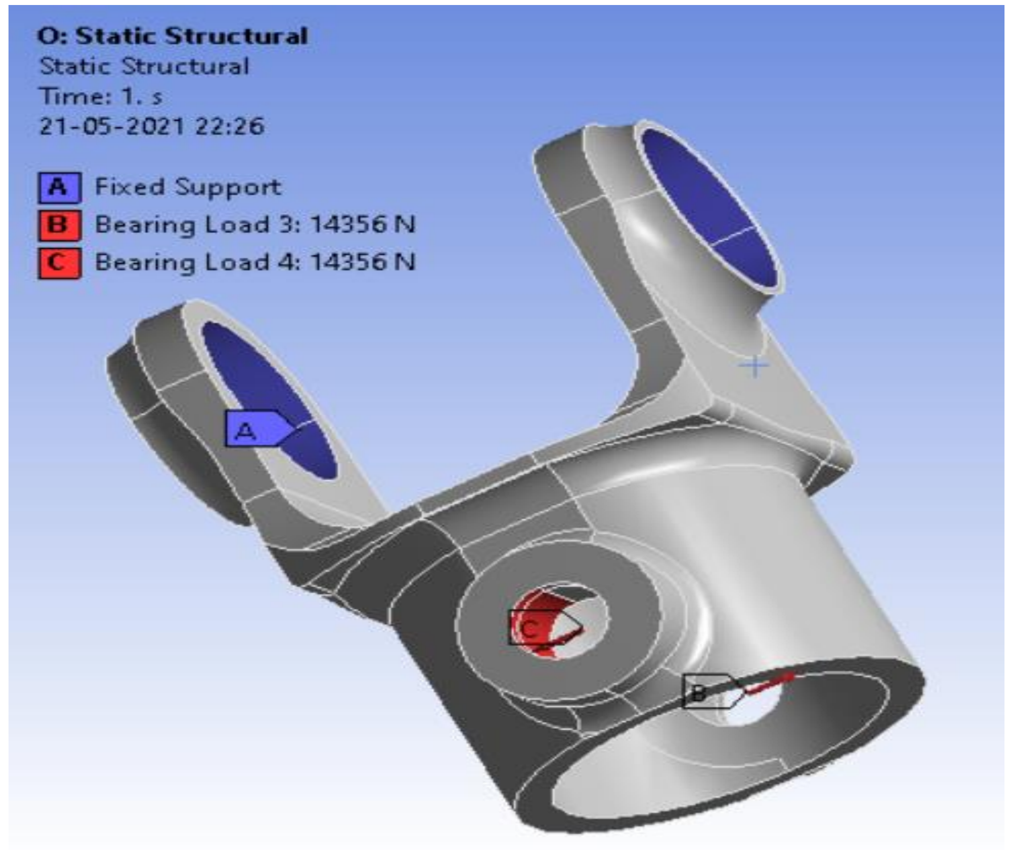
Overall Reduction=33.96.

Centre to centre between cross bearing=45.418 mm.

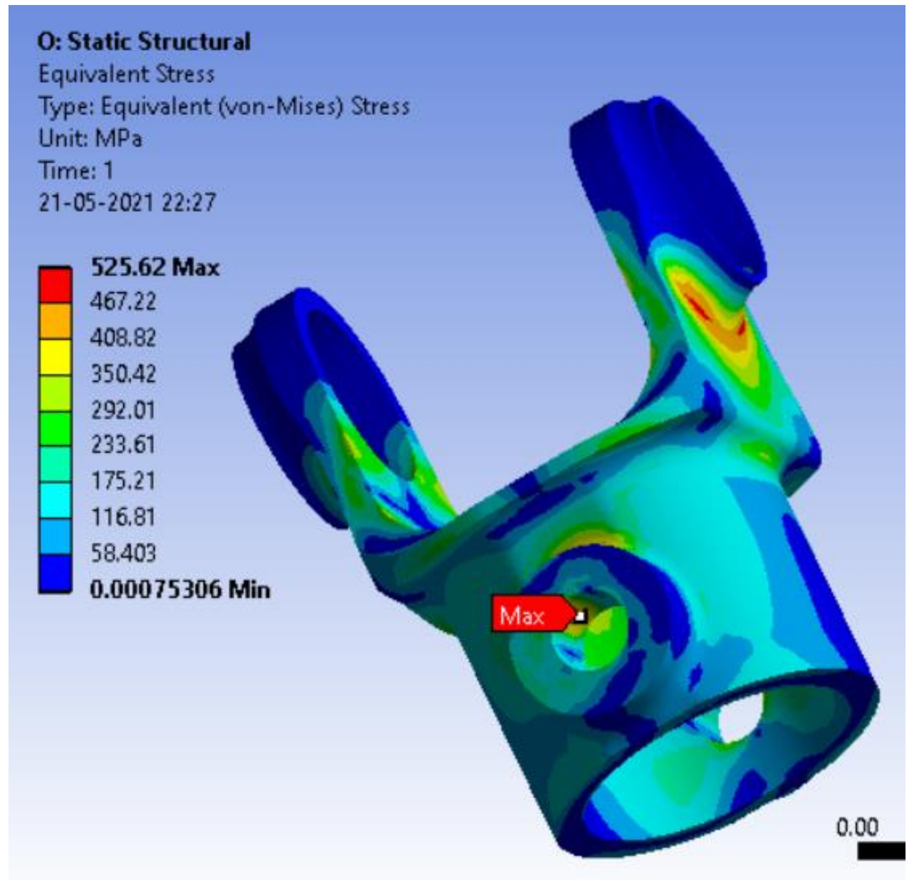
Force =(19.2*33.96)/0.045418

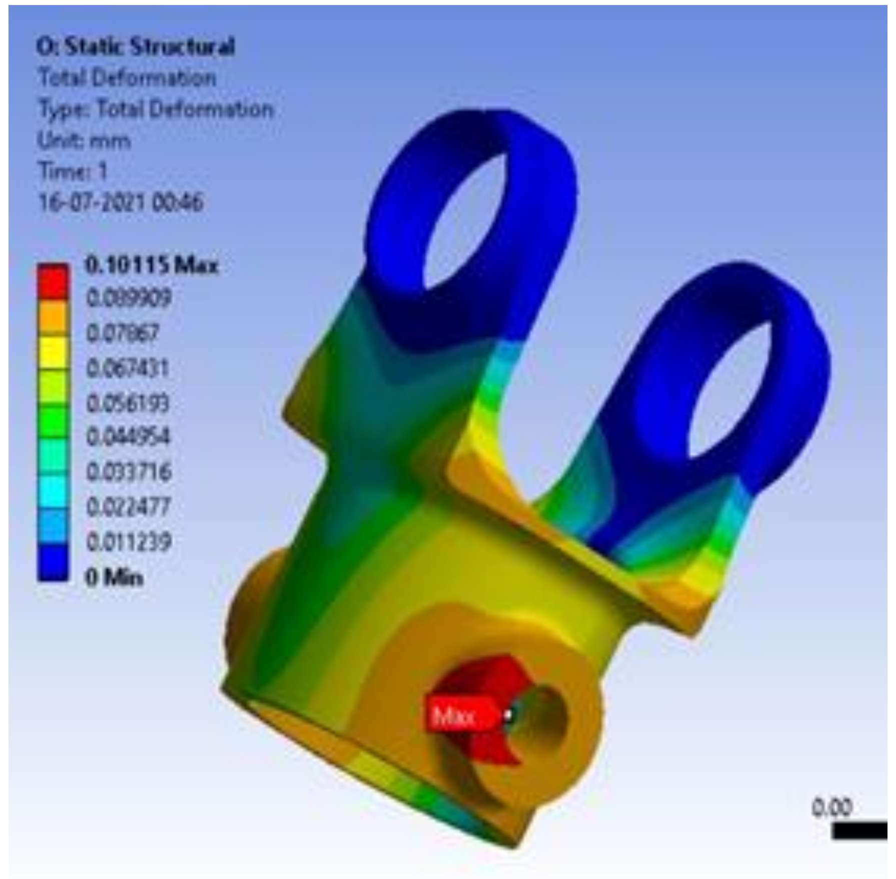
=14356 N

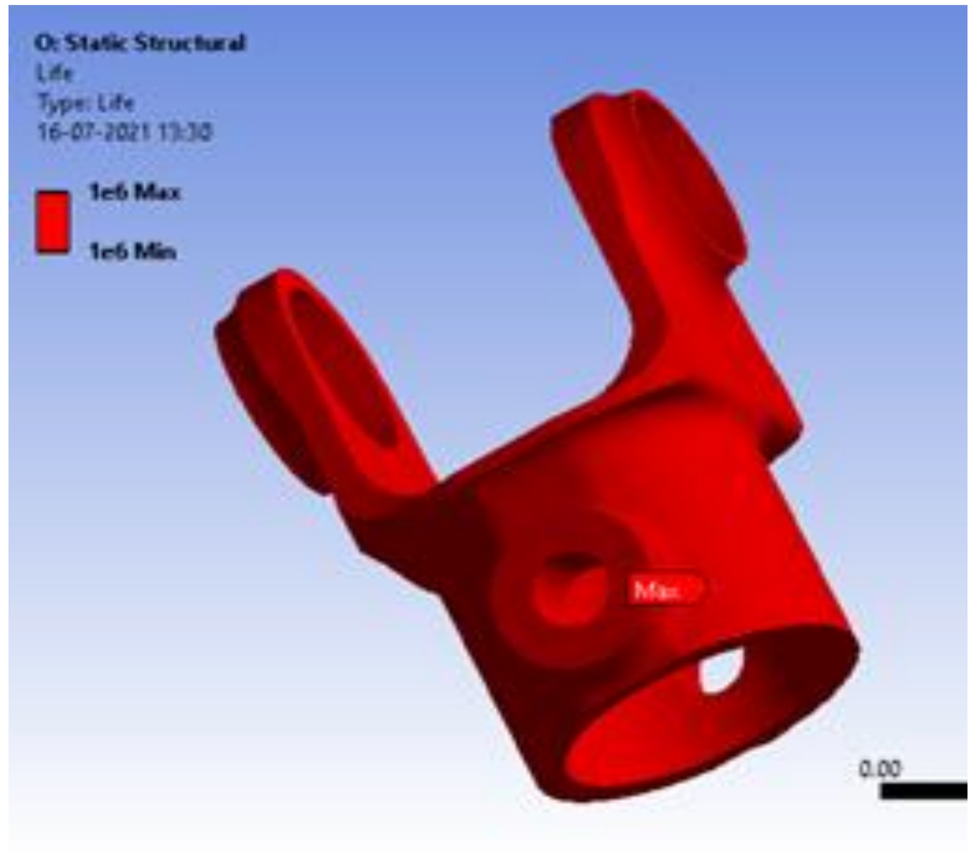
For fatigue analysis, we have considered average load.



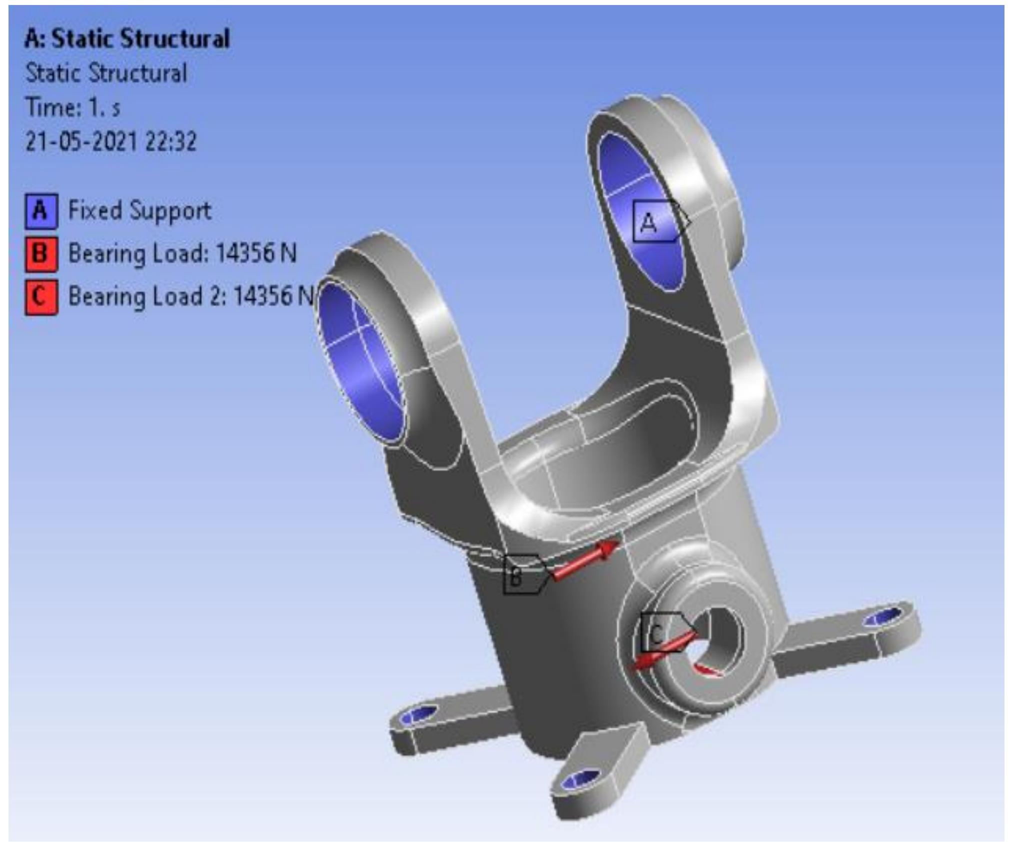
- Inboard yoke without adapter

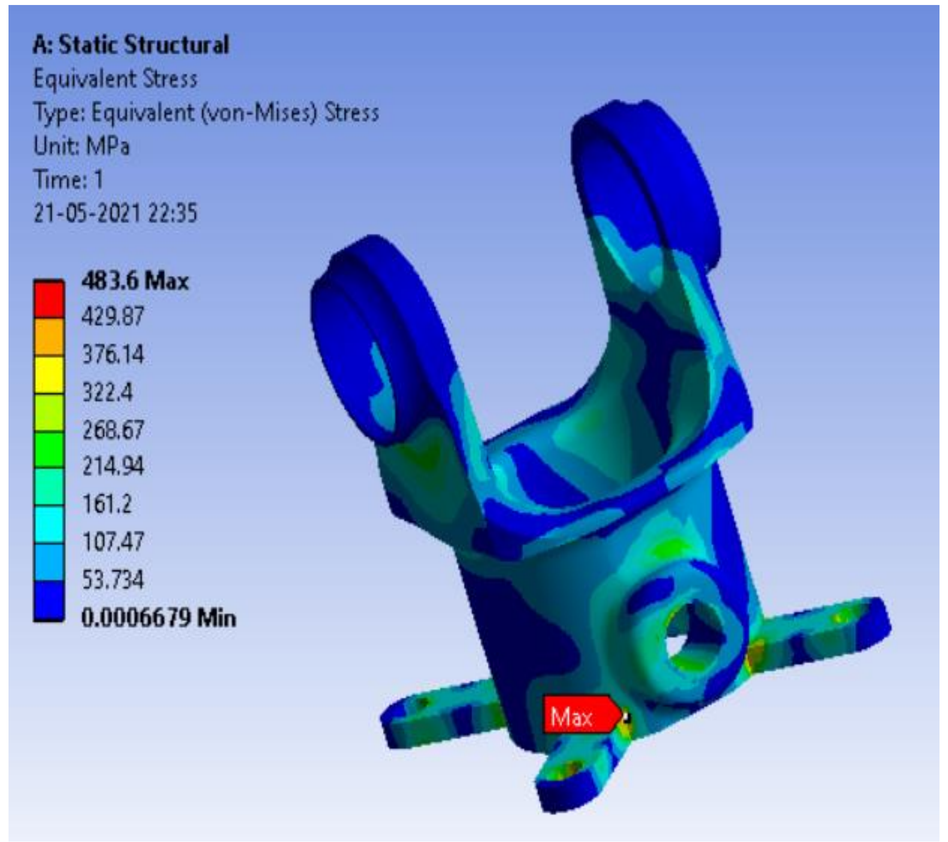


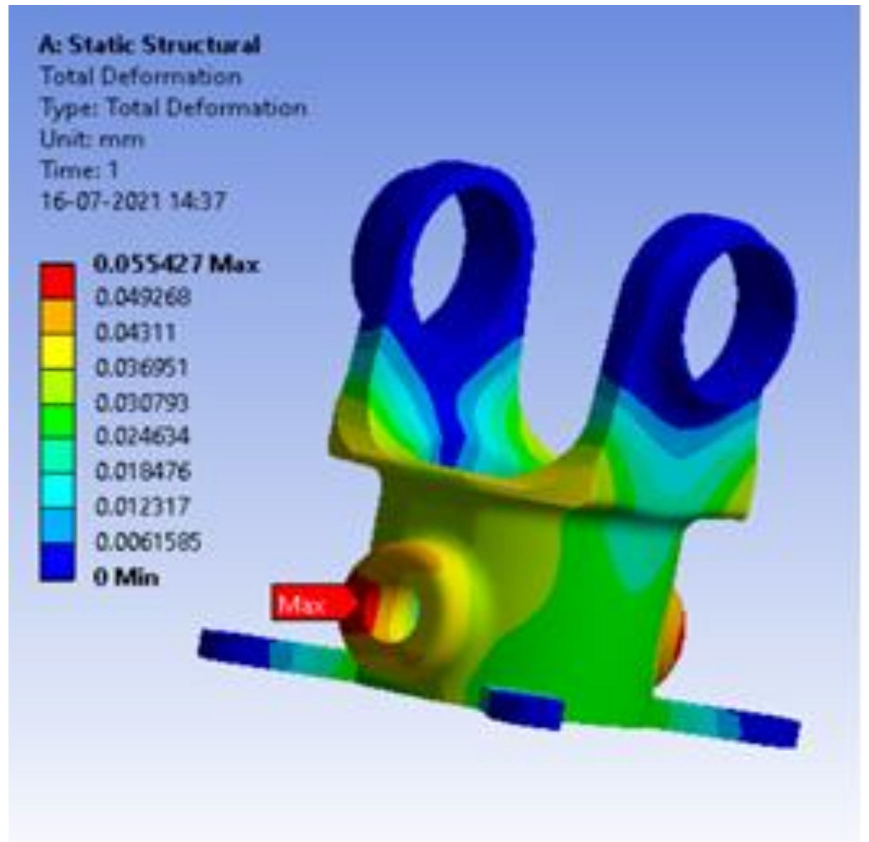


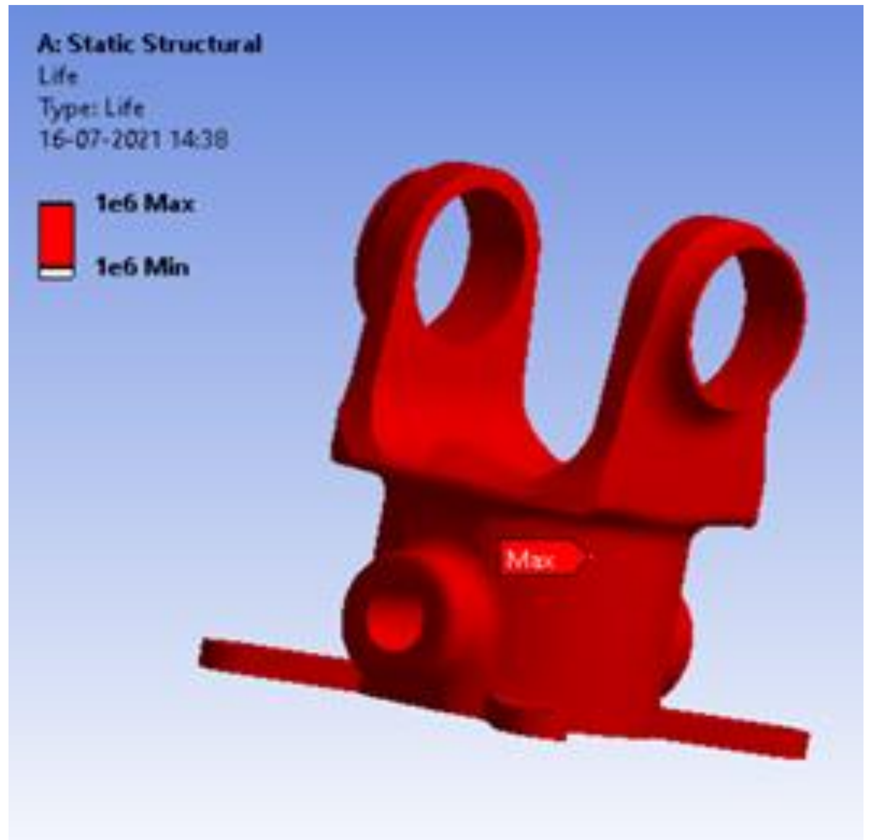


- Inboard yoke with adapter









Result and Discussion

Calculation of force for Outboard Yoke

Torque from engine=19.2 N m.

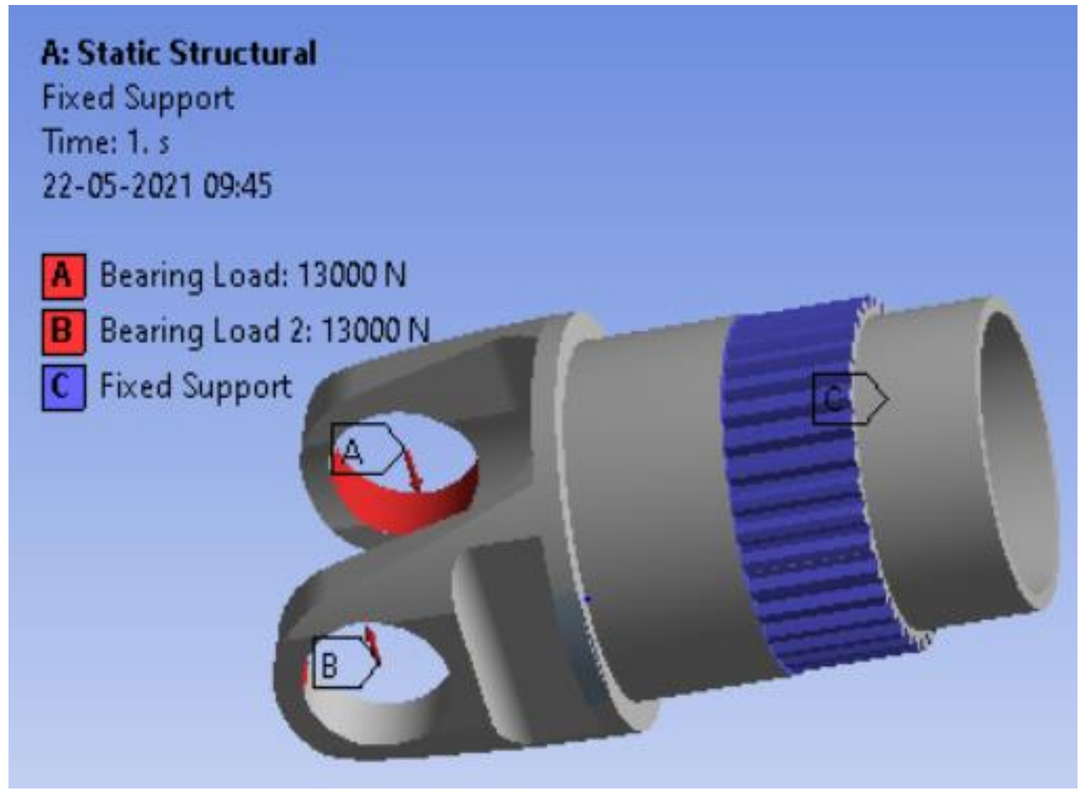
Overall Reduction=33.96.

Centre to centre between cross bearing=49.85 mm.

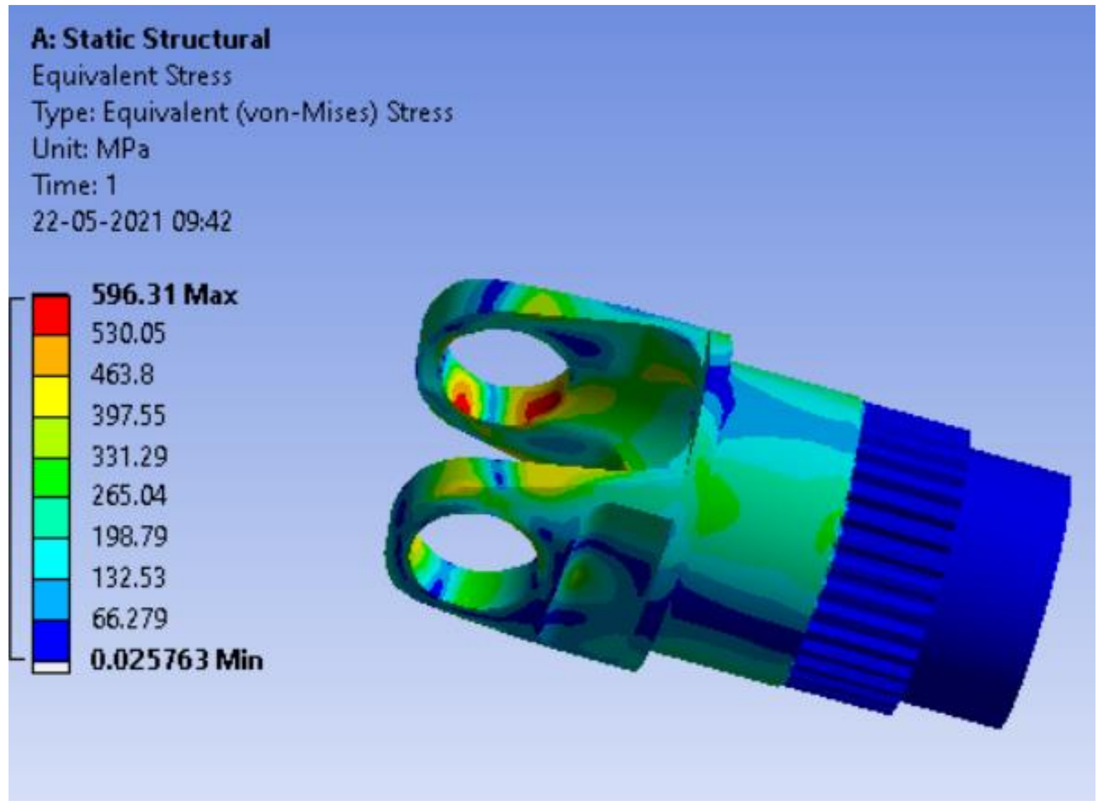
Force on the half-shaft=(19.2*33.96)/0.04985

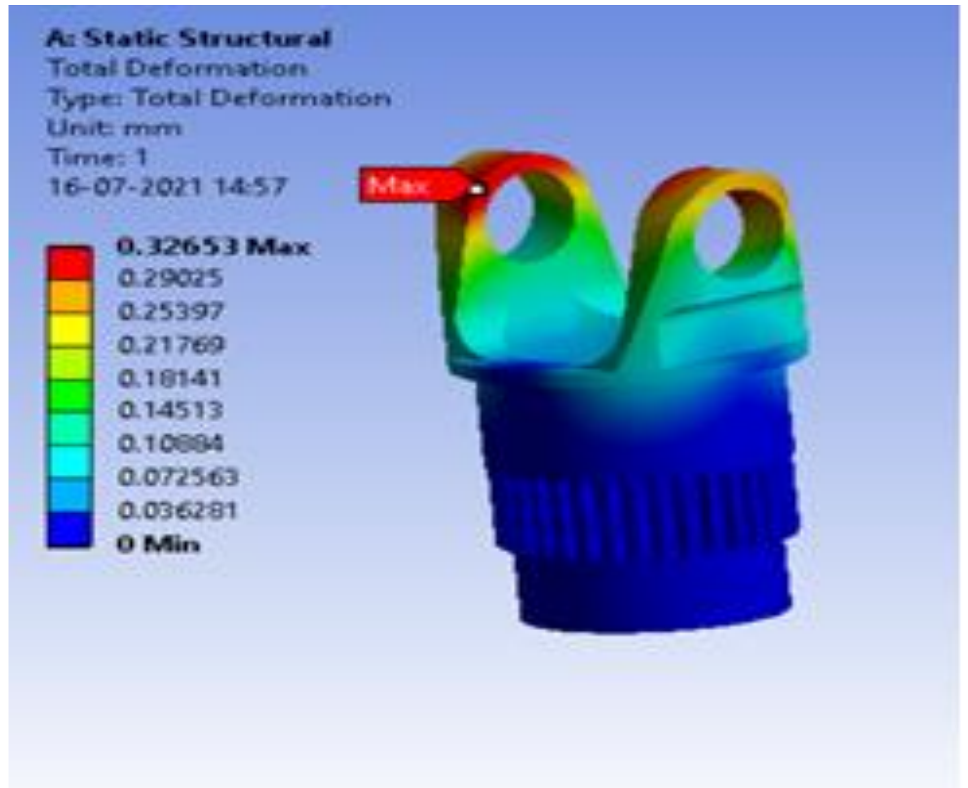
=13000 N

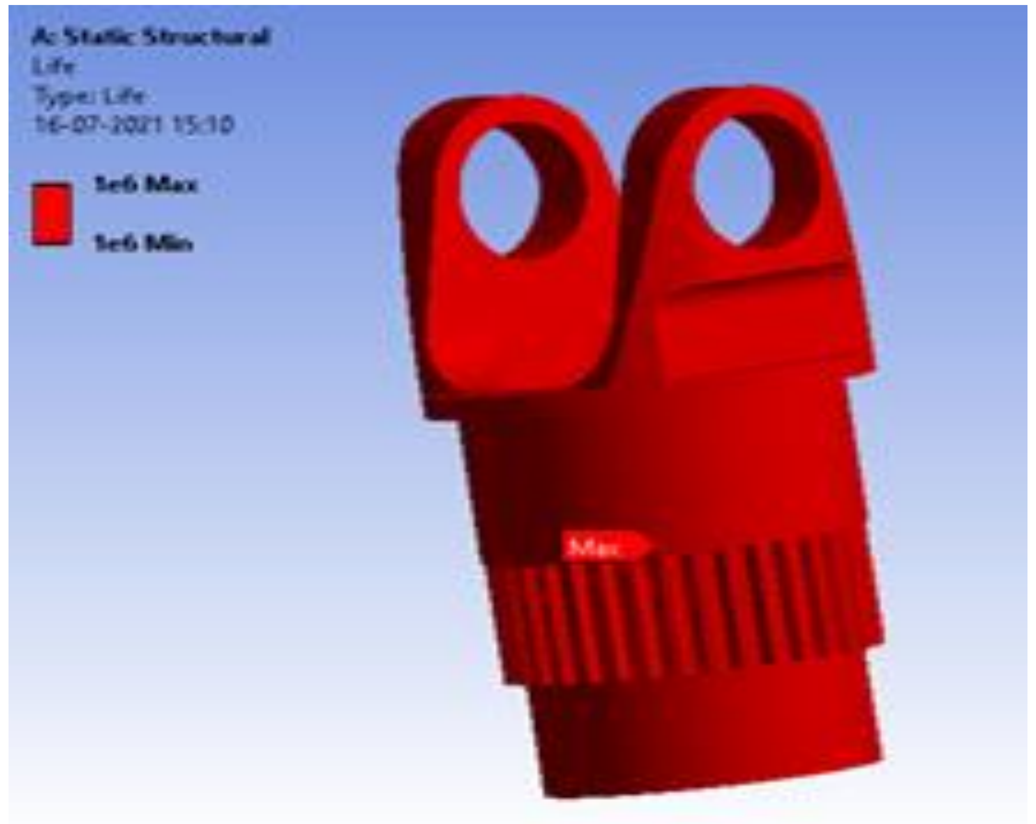
For fatigue analysis, we have considered average load.



- Outboard yoke







Calculation of force for Propeller shaft

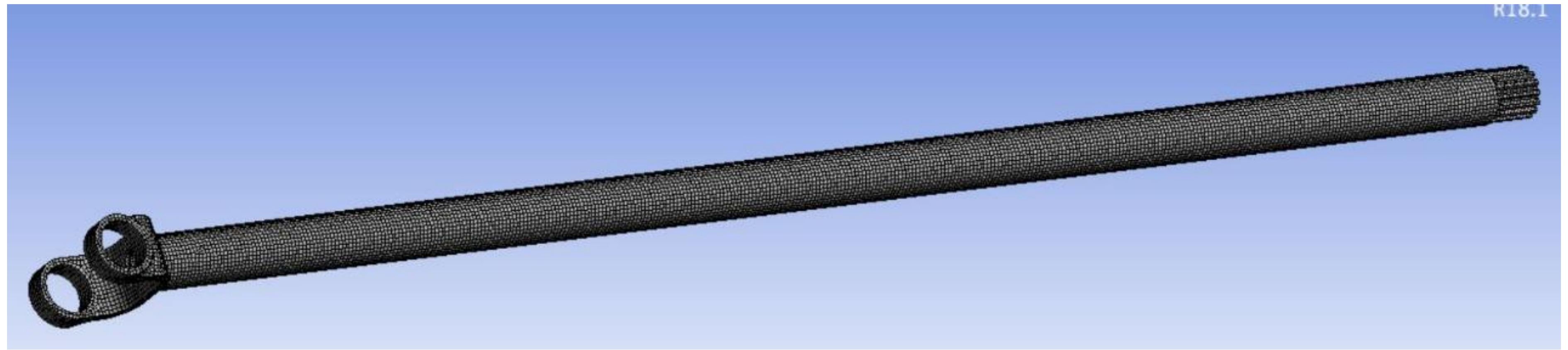
Torque from engine=19.2 Nm.

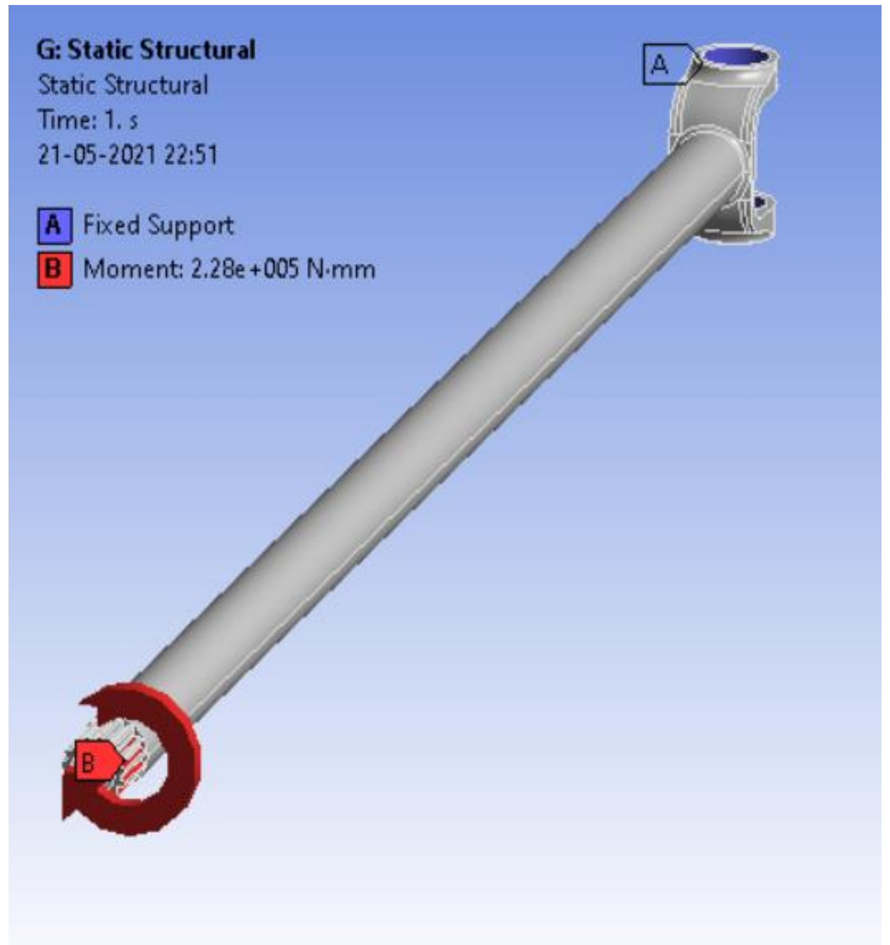
CVT Reduction=4, 1st stage gearbox reduction ratio=3,

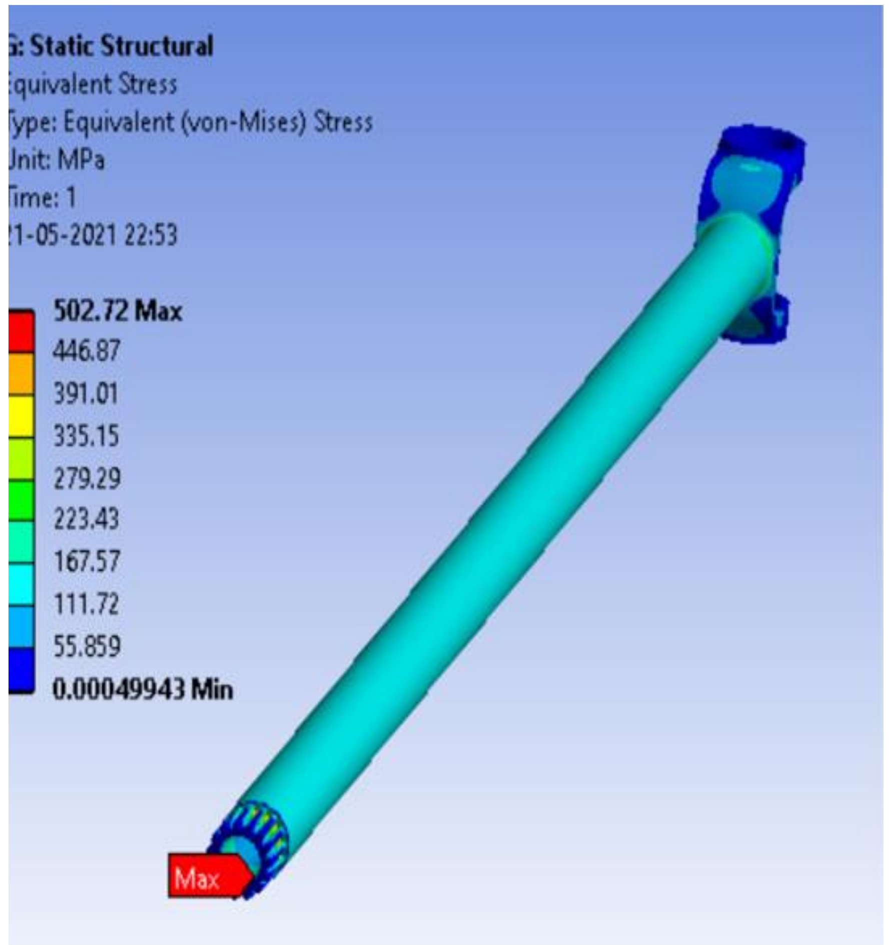
Torque on the Propeller shaft=(19.2*4*3)= 230.4 Nm

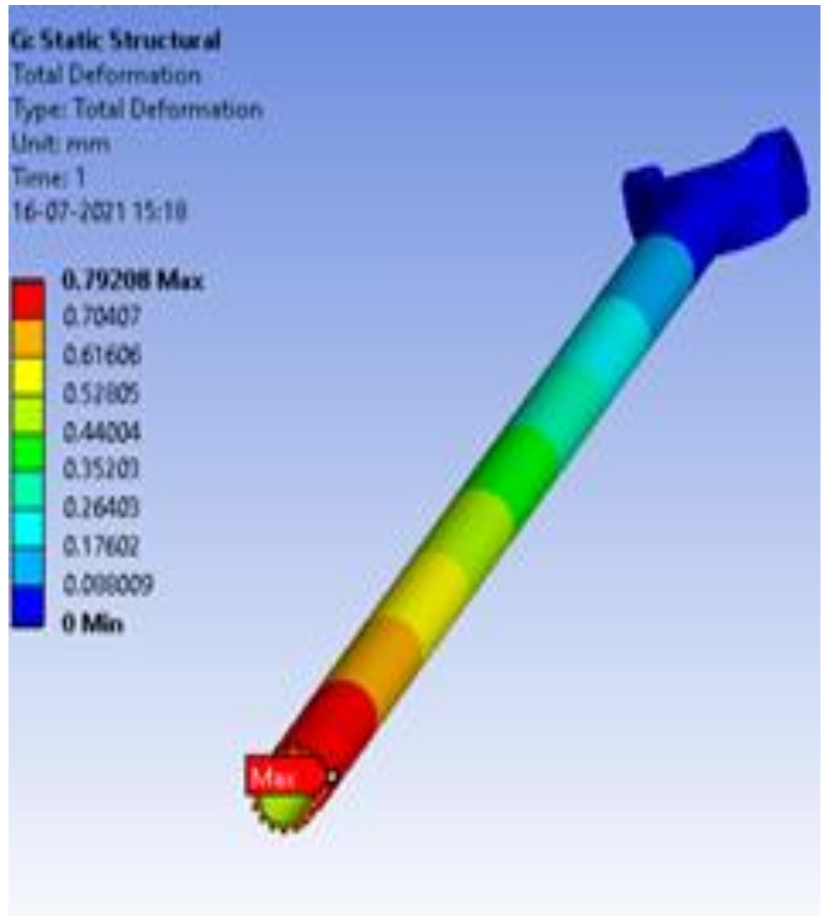
For fatigue analysis, we have considered average load.

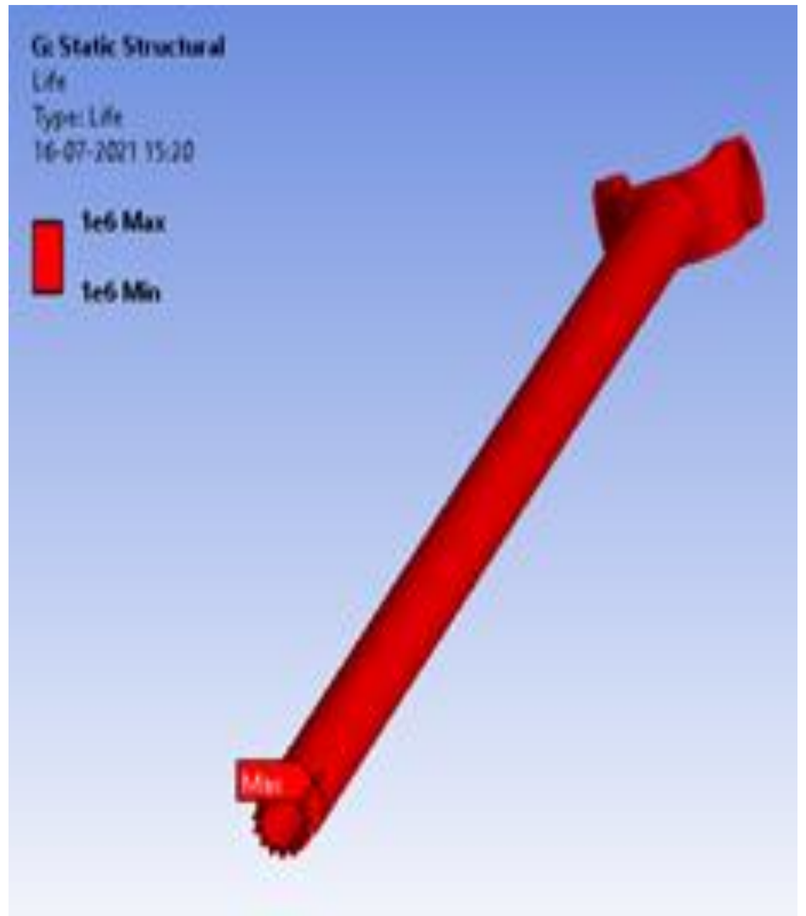
- Propeller Shaft











Calculation of force for hub

1. Bump load

Let us assume, the car falls from a height of 3m and lands on a single wheel and the shock compression time taken is 0.25s (Worst case scenario)

Therefore, let H = 3m; t = 0.2s

We know that, v^2 = 2gH => v = 7.668 m/s

And, v = u + a*t

7.668 = 0 + (0.2*a) a = 38.34 m/s^2

Now, F = ma

The assumed kerb weight (Including driver weight) of the car is 2000N;

m = 200 kg.

F = m*a

=> F = 200*38.34 = 7668 N ~ 8000 N

Therefore, Bump load = 8000 N

2. For corning load

Let us assume that the car is taking a curved path at a speed of 20 kmph and the curve is of radius 3m (As per Baja SAE track dimensions)

Therefore, r = 3m; v = 25 kmph = 7.94 m/s

Centrifugal force: F = m*(v^2)/r

= 190*(5.55^2)/3

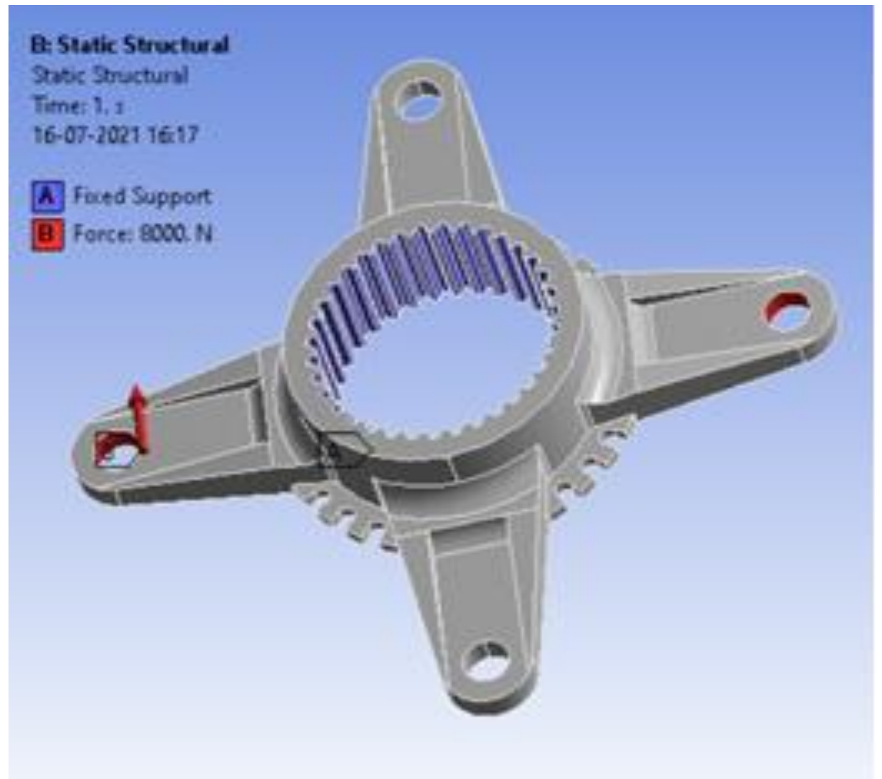
=> F = 3430.84 N ~ 3500 N

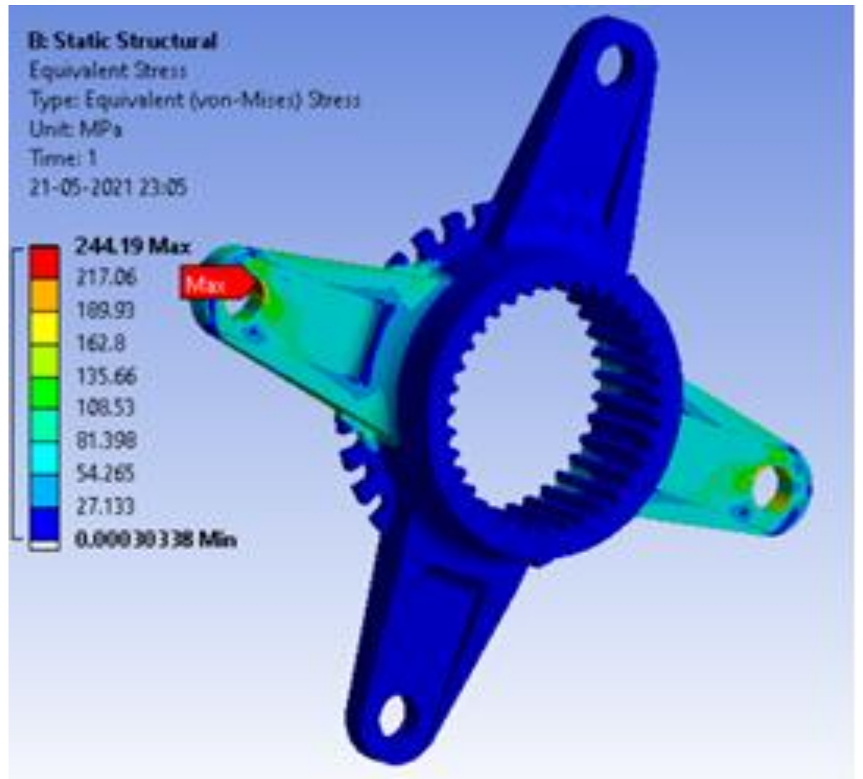
Therefore, Cornering load = 3500 N

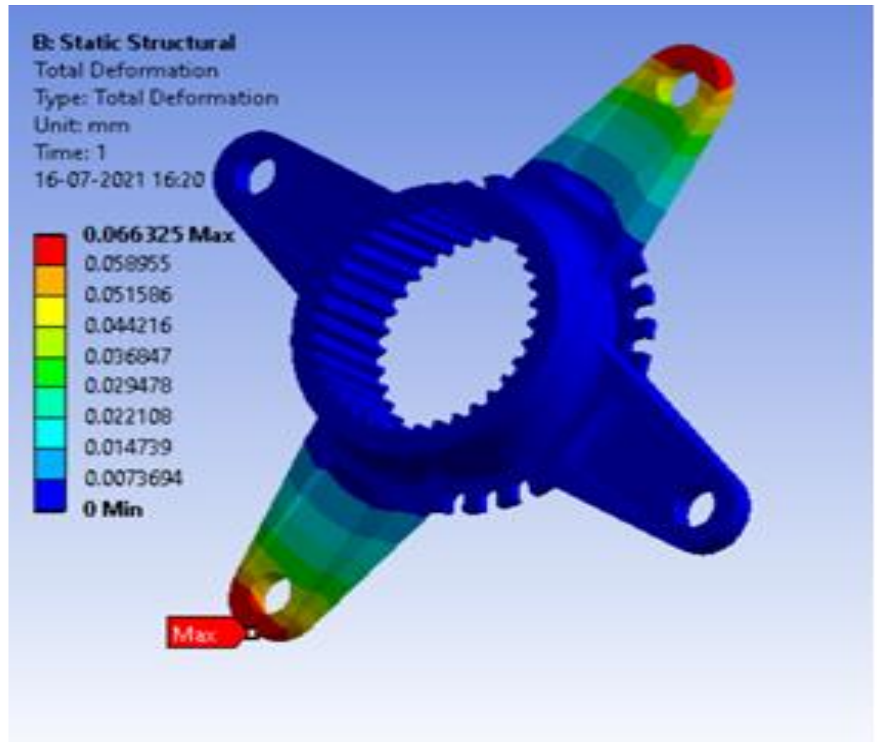
Result and Discussion

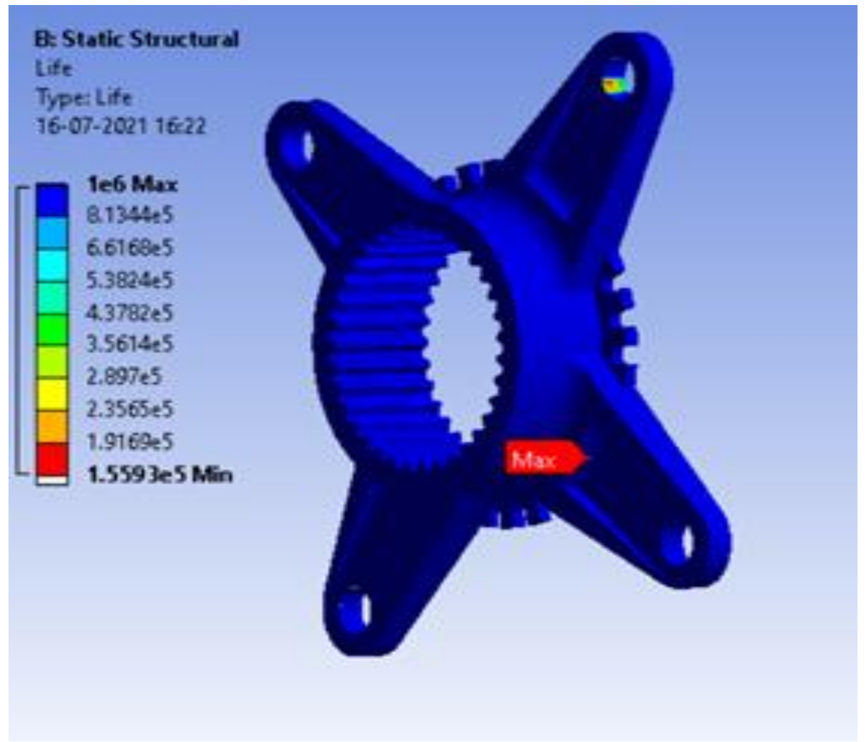
- Rear hub

1. Bump force

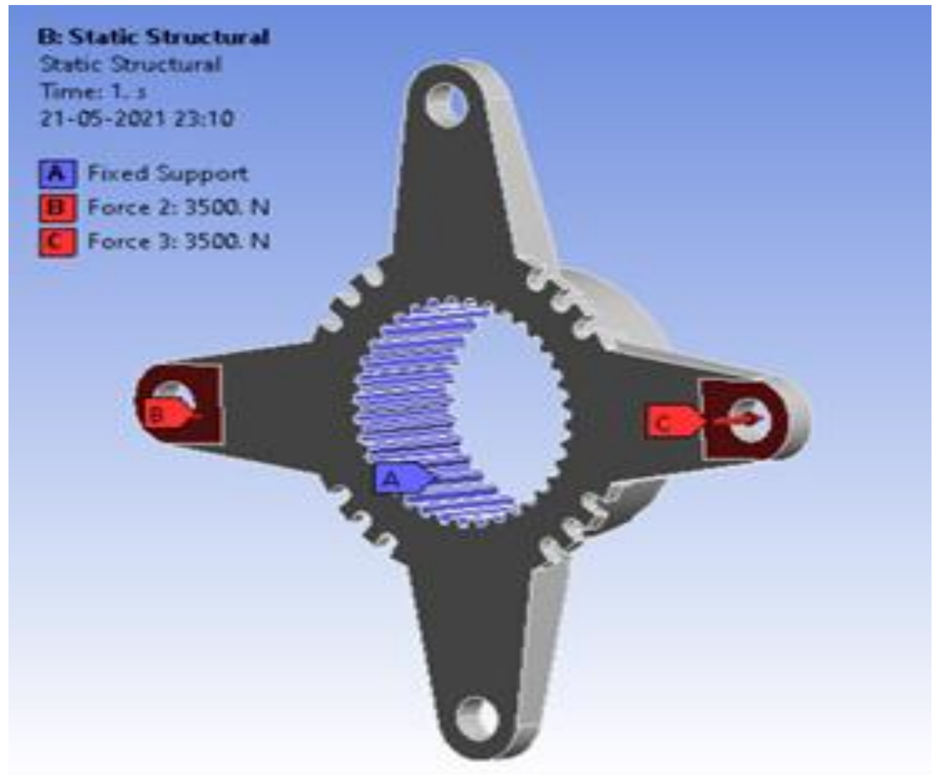


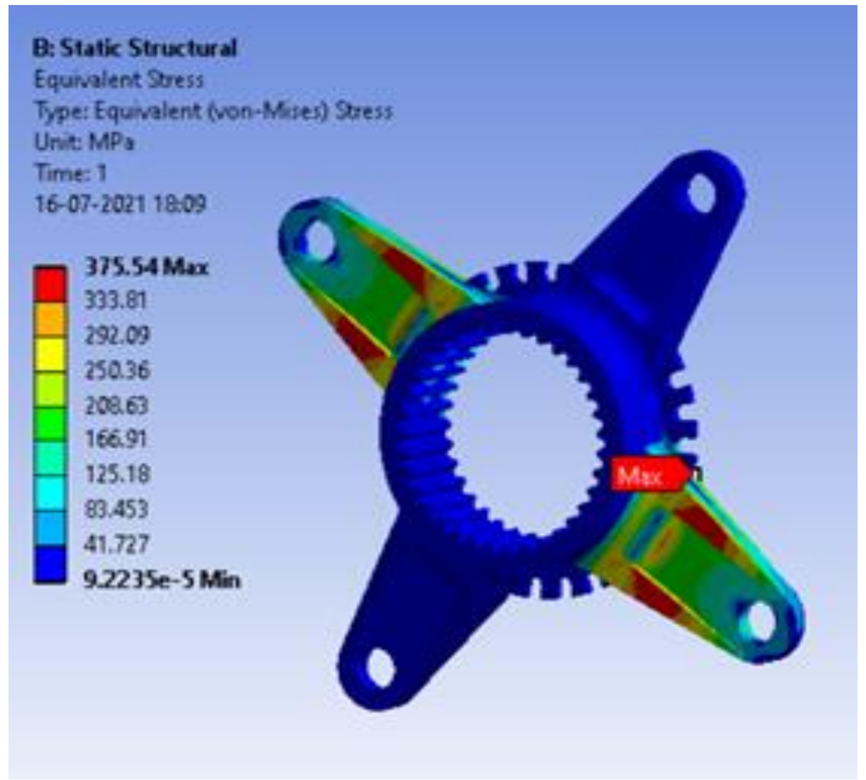


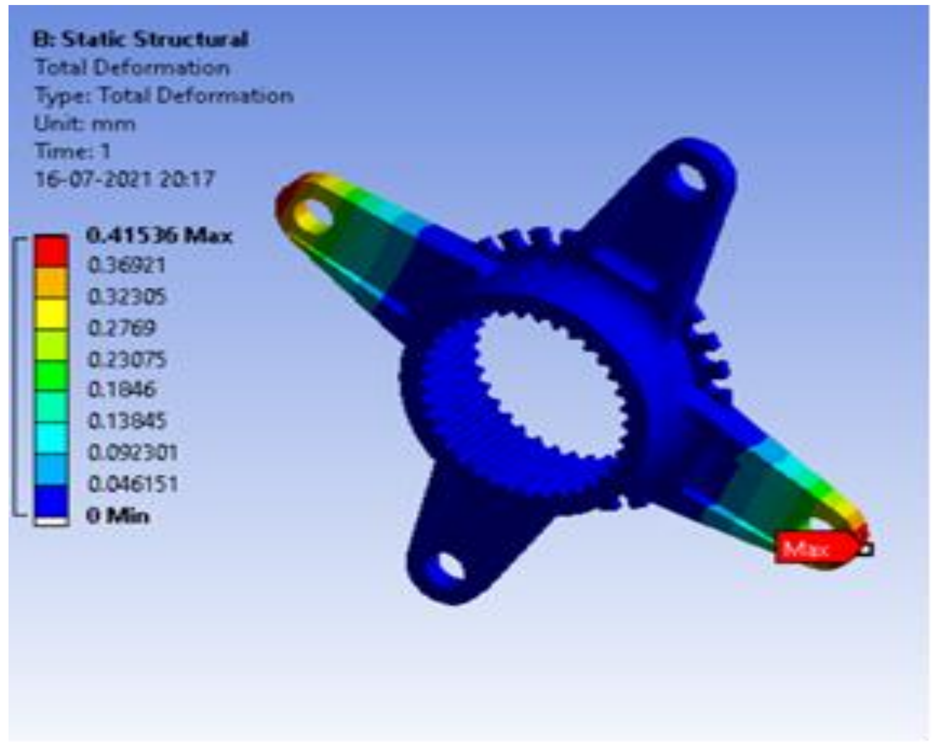


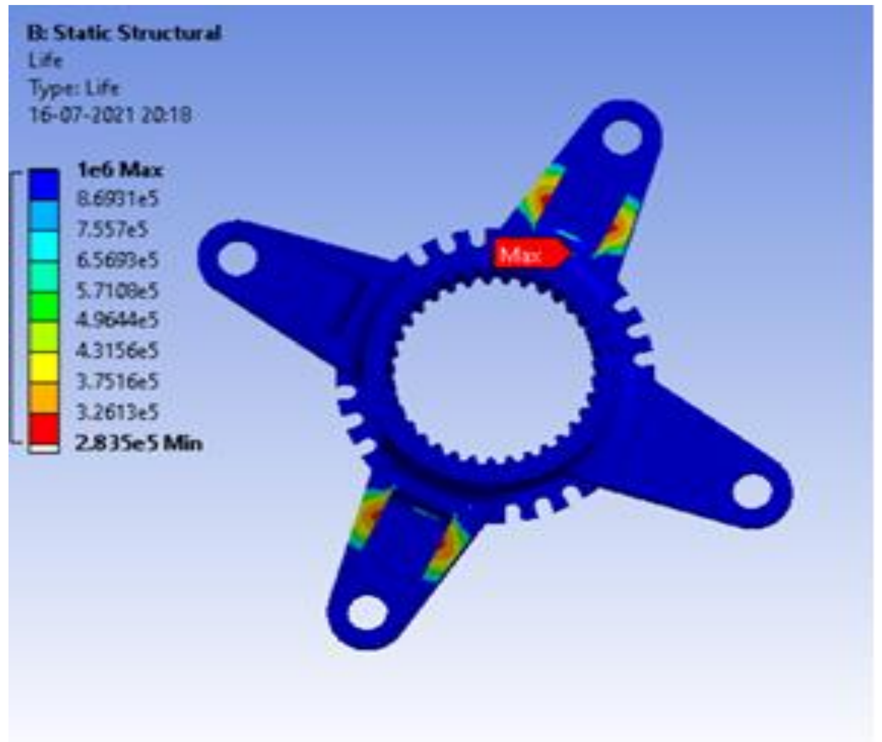


2. Cornering force





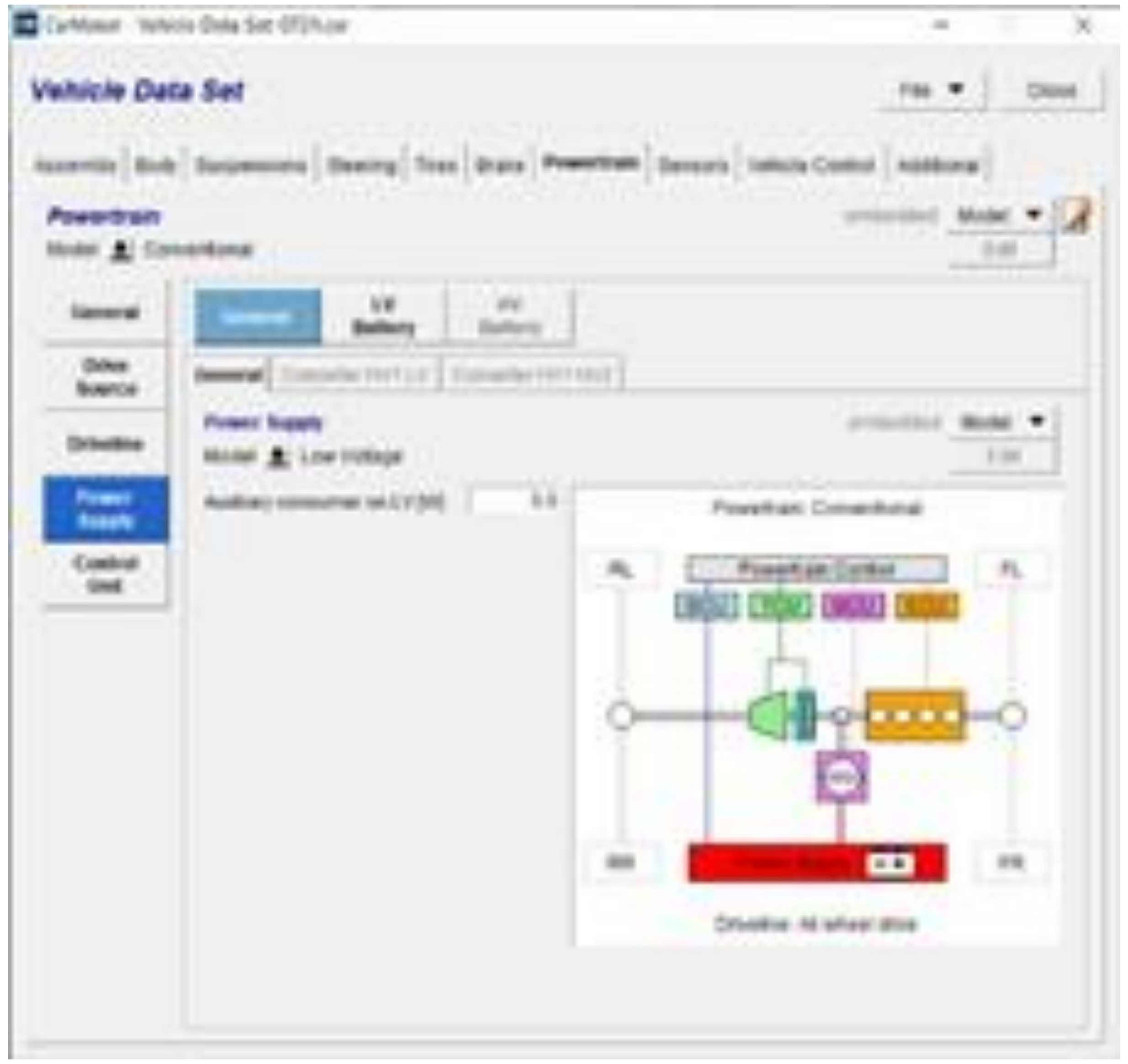




Result and Discussion

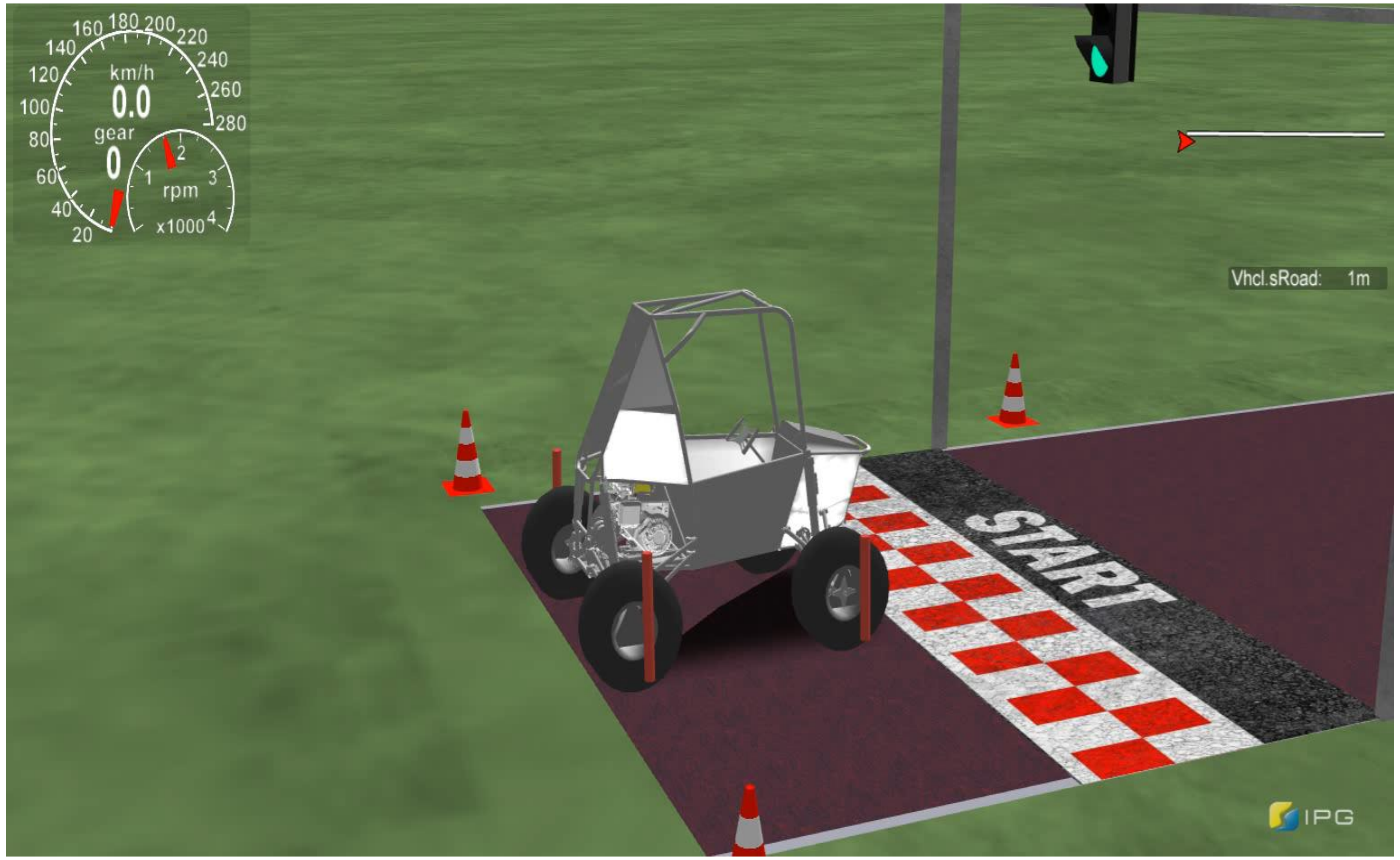
Components	Material	Max Stress (MPa)	Max Deformation (mm)	FOS
Inboard yoke without adapter	EN 24	525.62	0.101	1.61
Inboard yoke with adapter	EN 24	483.6	0.055	1.75
Half shaft	EN24	586.55	1.286	1.45
Outboard yoke	EN24	596.31	0.326	1.42
Long propeller shaft	EN 24	502.72	0.792	1.69
Propeller half shaft	EN 24	583.22	0.621	1.46
Propeller yoke	EN 24	578.28	0.381	1.47
Rear hub	Al 7075-T6	375.54	0.415	1.4

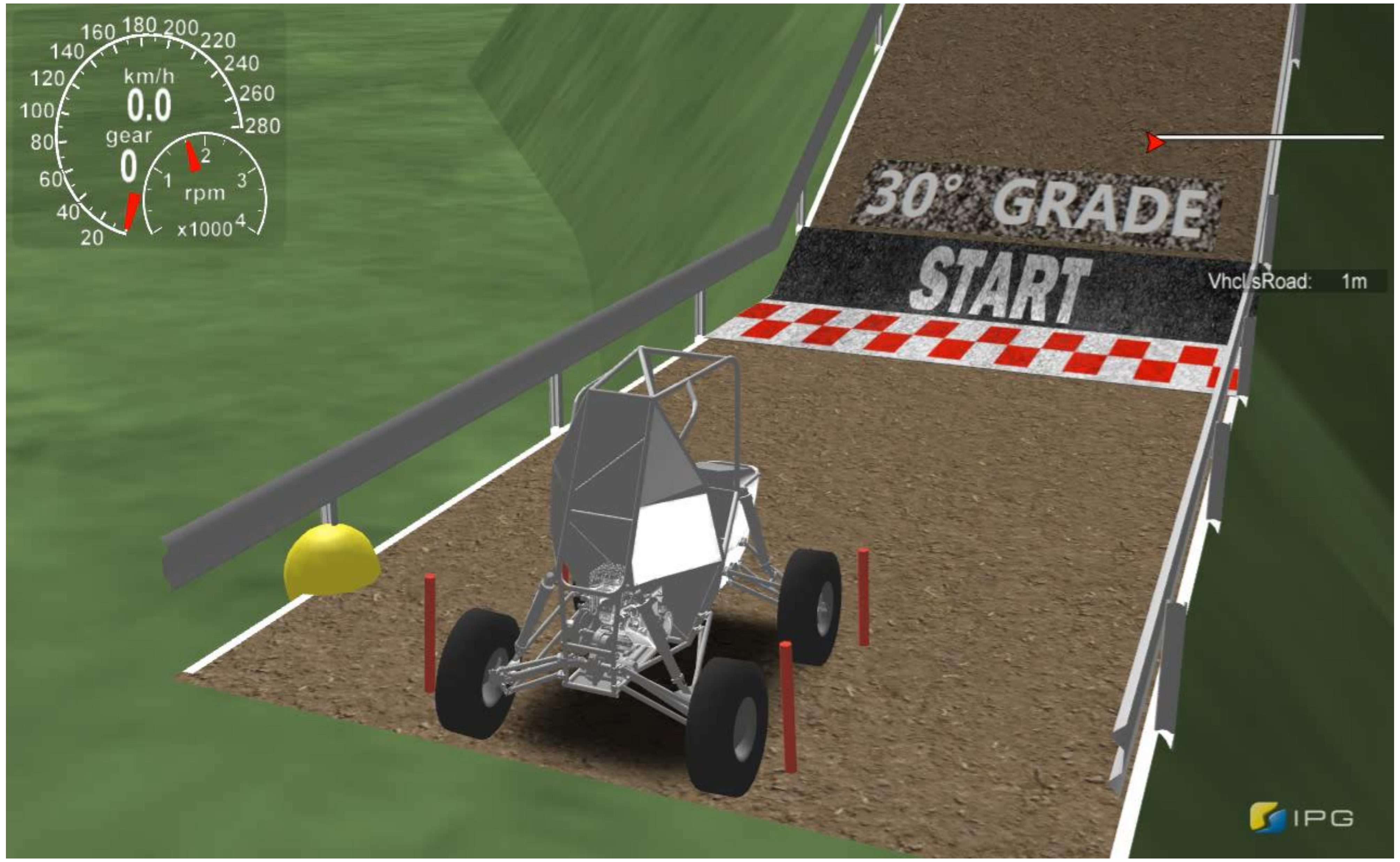
Parameters of various values of powertrain have been imported to Vehicle Data Set in IPG CarMaker software. This is show in Fig



Acceleration Run

Hill Climb Run





Conclusion

- The all-wheel driveline was designed for all-terrain vehicles which has an overall reduction ratio of 34.3 with the aim of improving gradability and performance.
- Tractive effort of 1981N, a drawbar effort of 1945N was achieved.
- By implementing chain drive in the transmission system , we were able to reduce the center of gravity of the car thus increasing the roll stability.
- By integrating input shaft and pinion the constraint on using splines or keys based on rim thickness factor was eliminated.
- Material for gears, shafts, casing and driveline components are EN36, Ti6Al4V, Aluminum 7075 and EN24 respectively and were selected based on design criteria using Pugh matrix.
- We have designed gear box, where gears are analysed for FOS of 1.56, Shaft is designed for FOS of 1.6 And sprocket is designed for FOS of 1.6.

Conclusion

- All driveline components are modelled and analysed, where FOS of half-shaft is 1.45, FOS of inboard and outboard yoke is 1.4, FOS of rear hub is 1.4.
- The gearbox reduction ratio of 8.5, differential reduction ratio of 1.8333 and differential side gear ratio of 1.65 were obtained for a turning radius of 3m.
- The All-wheel driveline, after implementation in IPG Carmaker, satisfies the design requirements. It proves to be durable on rough terrain and in critical operating conditions.
- All-Terrain vehicle achieved acceleration time of 4.82s for 150 feet and can go on 35 degree gradient terrains.
- Thus, we can conclude that the initial objective of an all-wheel driveline for all-terrain vehicles is satisfied and serves as ground for further research.