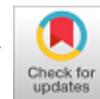


# Design and Analysis of an Epicyclic Gearbox for an Electric Drivetrain



Timir Patel, Ashutosh Dubey, Lokavarapu Bhaskara Rao

**Abstract**— This paper focusses on the designing and optimizing of a planetary gearbox for an FSAE formula electric car. The conventional arrangement of single or double stage reduction transmissions are bulky and uneconomical for an FSAE car because of their less power to weight ratios and high volume to weight ratios. A planetary gearbox gives a balance between weight and power transmitted. The procedure for the design was an iterative process to determine the optimum dimensions and materials for the gearbox. The method utilized the standard equations for gear calculations and computational simulations for designing and CAD modelling for various parts of the gearbox. The transmission shafts and bearings are designed using the standard force equations. The lubrication system is also selected according to established methods.

**Keywords:** Epicyclic Gearbox, Electric Drivetrain, Planetary Gearbox, FSAE formula electric car, Gear Design, Shaft Design, Gear Lubrication

## I. INTRODUCTION

In today's age, conventional fuels are depleting, and the automobile industry is adapting the concept of electrically powered vehicles. Planetary gearboxes are widely used in the efficient transmission of power in electric vehicles. These types of gearboxes are compact and have a higher torque to weight ratio than compound gear trains. With more efficiency than that of normal series gearing, their use is becoming more and more prevalent in the automotive field, especially in the case of formula student electric vehicles [1]. Low spatial requirement and a high power-to-weight ratio make these gearboxes superior, especially when compared to double-step single reduction gearboxes [1]. The planetary arrangement also keeps the motors concentric to the driveshaft, thus providing more efficient packaging of components [1]. High transmission ratios can be obtained from relatively compact arrangements of gears using a planetary gearbox instead of using compound gear trains.

## II. OBJECTIVE

The main objective for this system is to make a trade-off between speed and torque to provide a necessary push to accelerate the vehicle.

The torque provided by the motor is not enough to accelerate the vehicle at the required rate and thus the torque needs to be increased at the expense of speed. This can be achieved by placing an efficacious transmission system between the motor and the differential (which has a 1:1 transmission ratio). An arrangement of gears is an ideal system to achieve this goal. Gears are used in almost every Transmission system to transmit power from one shaft to another and to regulate the speed and torque. Gears have a high efficiency and can be manufactured using basic machining processes. The different gear arrangements that can be used are a single stage gear reduction, a multistage reduction and a planetary or an epicyclic gear train. In single and multi-stage gear train, the gear axes are fixed, and the gears rotate about their respective axes. However, in a planetary gear train, the axes of some of the gears are not fixed but rotate about the axes of other gears. Planetary gear trains can transmit at very high transmission ratios and require smaller gears in a compact space [2].

## III. METHODOLOGY

First, to start designing the transmission system, the power obtained and the output parameters from the motor is needed. The motor used in our car is the EMRAX 208, which is an AC 3-phase synchronous motor, delivering a maximum power of 75 kW. The maximum torque obtained is 120 N-m @ 5500 RPM [3]. The required speed and torque for achieving the desired acceleration and speed of the car are calculated and thus the transmission ratio for the system can be obtained. Now, to transmit power to the differential placed at the center of the drive shaft, a chain drive will be used whose input will be the output from the gearbox. Thus, the gearbox transmission ratio can be obtained. Now, the gearbox arrangement is made such that the annular or the ring gear will be stationary, and the sun gear will be used as the input with the carrier as the output as shown in figure 1. This arrangement can be used for gear ratios of 3:1 or more and is relatively simple to implement. The design procedure for the gears is carried out in accordance with the PSG Data Book and the formulae used are referred from the book as well. An important thing to note is that the planet gears need to be arranged with bearings and so the planet gears need to be designed while considering the bearing dimensions that can be used and the gear dimensions must be adjusted accordingly.

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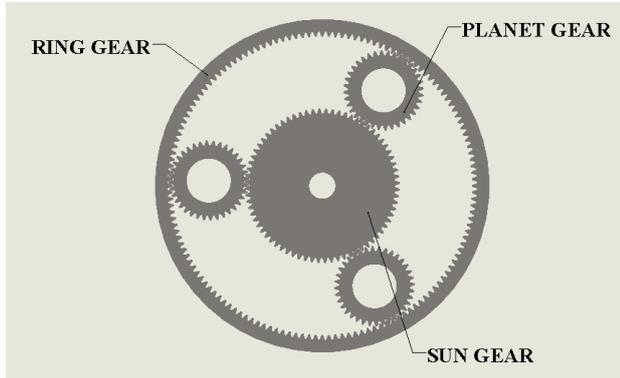
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According to DIN 3990, the material below the root of the gear teeth must be 3-6 times the module of the teeth [4]. Computational simulations are carried out to ensure the safety of the design. After the gear design, the transmission shafts and the bearings are selected. An appropriate lubrication system is selected, and the housing is designed accordingly.



**Fig. 1. Planetary Gear Arrangement**

### IV. DESIGN PARAMETERS

The motor used is the EMRAX 208 which is an AC 3-phase synchronous motor which delivers a maximum power of 75kW. The maximum torque obtained is 120 N-m@5500 RPM. Now, assuming the weight (m) of the car with the driver will be 320 kg and the radius (r) of the wheel is 9 inches, to achieve a speed of 0-100 km/h or 0-27.77 m/s in less than 4 seconds (which is our target), the acceleration should be,

$$a = \frac{27.77 - 0}{4} = 6.94 \text{ m/s}^2$$

Now, to find the torque required at the differential will be,

$$T = m \times a \times r = 320 \times 6.94 \times 9 \times 0.0254 = \mathbf{507.67 \text{ N-m}}$$

Thus, if we need 507.67 N-m at the differential, then the Final Drive gear ratio needed will be

$$\text{Final Drive ratio} = \frac{507.69}{120} = 4.23 = \text{approx. } \mathbf{4.5}$$

The top speed in RPM when the gear ratio is 4.5 is  $\frac{5500}{4.5} = 1223 \text{ RPM}$ .

Therefore,

$$\text{velocity} = \frac{\pi d N}{60} = \frac{\pi \times 18 \times 0.0254 \times 1223}{60} = 29.26 \text{ m/s} = \mathbf{105.34 \text{ km/h}}$$

Thus, the design parameters for the transmission system are shown in table-I,

**Table-I: Design Parameters for the transmission system**

Torque at the differential	507.67 N-m
RPM at the differential	1223 RPM
Final Drive Ratio	4.5

Now, the transmission ratio from the chain drive, which is connected to the differential, will be **1.5**.

Therefore, the transmission ratio required from the planetary gear set would be,

Gear box ratio,

$$i = \frac{\text{Final Drive Ratio}}{\text{Chain Drive Ratio}} = \frac{4.5}{1.5} = \mathbf{3}$$

Thus, the design gear box ratio will be **3:1**.

### V. GEAR DESIGN

#### Gear Material

18CrNiMo7-6 Chrome-Nickel-Moly Carburing Steel generally supplied annealed to HB 229(max). Carbured and heat treated, it develops a hard wear resistant case of HRC 60-63 and a tough strong core with a typical tensile strength range of 900-1300 MPa, in small to fairly large sections [5]. The material properties are shown in table-II.

#### Applications

Heavy-duty arbors, bushings, wear pins, bearings, sprockets, gears and shafts etc. Or can be used for high tensile applications un carbured but through hardened and tempered [5].

**Table-II: Material Properties (EN10084 - 18CrNiMo7) [5]**

Typical Chemical Analysis					
Carbon	0.18%				
Silicon	0.20%				
Manganese	0.70%				
Chromium	1.65%				
Nickel	1.55%				
Molybdenum	0.30%				
Typical Mechanical Properties – Quenched and Tempered at 200°C					
Section	Yield Strength	Tensile Strength	Elongation	Impact Izod	Hardness
mm	MPa	MPa	%	J	HB
25	1050	1295	14	45	380
50	950	1160	15	51	340
100	815	1010	16	53	300

#### Gear Calculation

We initially assumed the module(m) as 2 mm and number of teeth on planet gears ( $Z_p$ ) are 18 as suggested by AGMA. As the planet and the sun gear are made of the same material so by default, the planet is going to be treated as the weaker element and all the designs are based on the planet.

Diameter of Planet ( $D_p$ ) =  $m \times Z_p$

(1)

$$\text{Ratio, } i = \frac{(\text{Teeth of sun}(Z_s) + \text{Teeth of ring}(Z_r))}{(\text{Teeth of sun}(Z_s))} \quad [6]$$

(2)

From the above two equations we get the values shown in table-III,

**Table-III: Dimensional Values of gear**

Sr. No	Type	No of teeth	Diameter	RPM
1.	Planet	36	72	1834
2.	Sun	72	144	5500
3.	Ring	144	288	0

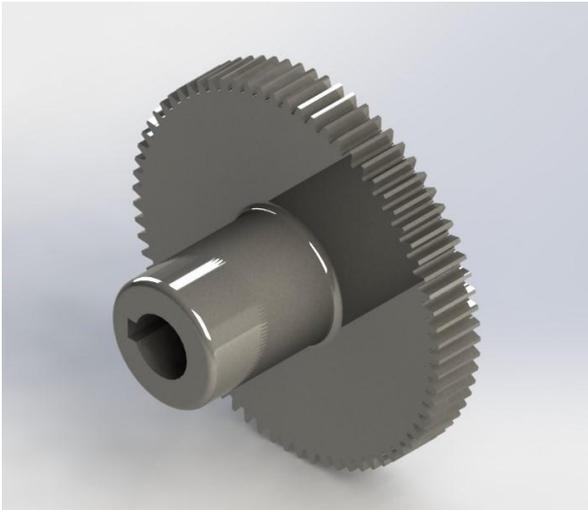


Fig. 2. Sun Gear

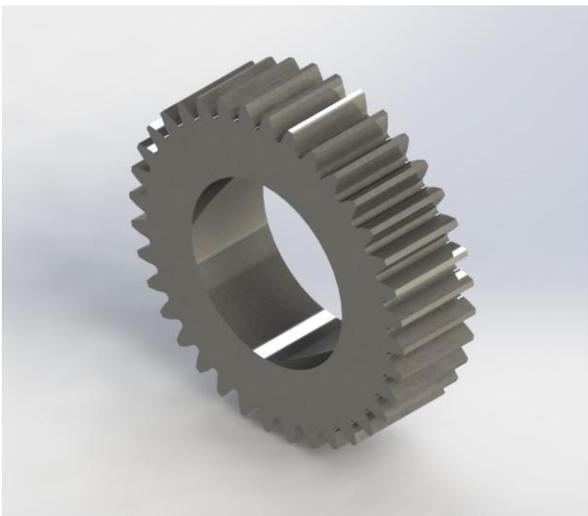


Fig. 3. Planet Gear



Fig. 4. Ring Gear

**Procedure for the Gear calculations**

Ultimate strength = 1200MPa  
Yield strength = 850MPa  
Surface Endurance strength = 1500Mpa

$$Power(P) = Tangential Load(F_t) \times Pitch Line Velocity(v) \tag{3}$$

$$Pitch Line Velocity = \frac{\pi D_p N_p}{60} \tag{4}$$

$$RPM of planet(N_p) = Velocity ratio of Sun and Planet \times RPM of Sun(N_p) \tag{5}$$

From the above three equations, Pitch line velocity and Tangential Load is calculated.

$$F_t = 555.555 N$$

$$v = 41.469 m/s$$

$$T = 120 N - m$$

**Lewis Equation**

$$F_t = \sigma_w P_c y_p b \tag{6}$$

Where: -

- $\sigma_w$  is the allowable stress
- $P_c$  is the circular pitch which is equal to  $\pi$  times of module
- $y_p$  is the form factor which is based on tooth profile ( $20^\circ$  involute) and number of teeth
- $b$  is the face width of the planet

$$\sigma_w = \sigma_d \times C_v, \text{ where } C_v = \text{coefficient of velocity} \tag{7}$$

$$\sigma_d = \frac{\sigma_u}{f_{os}}, \text{ where } \sigma_d = \text{ultimate strength} \tag{8}$$

$$C_v = \frac{0.75}{0.75 + \sqrt{v}}, \text{ for precision gears with } v > 20 m/s \tag{9}$$

$$y_p = 0.154 - \frac{0.912}{Z_p}, \text{ for } 20^\circ \text{ full involute teeth} \tag{10}$$

By using the Lewis equation, the face width can be calculated for the sun, planet and the ring gears.

**Beam Strength (F<sub>s</sub>)**

Considering a gear tooth as a cantilever beam its strength can be calculated under the tangential load.

$$F_s = \sigma_d P_c y_p b \tag{11}$$

**Dynamic Load (F<sub>d</sub>)**

It is sum of tangential load (F<sub>t</sub>) and dynamic induced load (F<sub>i</sub>)

$$F_d = F_t + F_i \tag{12}$$

$$F_i = \frac{21v(bc + F_t)}{21v + \sqrt{bc + F_t}}, \text{ where } c \text{ is the deformation factor} \tag{13}$$

**Wear Load (F<sub>w</sub>)**

$$F_w = D_p b Q k \tag{14}$$

$$Velocity ratio factor (Q) = \frac{2i}{i+1} \tag{15}$$

$$Combination Factor (k) = \frac{(\sigma_{es})^2 \sin \phi}{1.4} \left( \frac{1}{E_p} + \frac{1}{E_g} \right) \tag{16}$$

where,

$\sigma_{es}$  is Surface endurance limit  
 $\phi$  is Pressure angle

$E_p$  and  $E_g$  are the Young's modulus of the planet and the sun gears respectively.

**Criteria for safer design**

1. Static Load (Beam Strength) should be greater than Dynamic Load

$$F_s > 1.5F_d$$

2. Wear Load should be greater than dynamic Load

$$F_w > F_d$$

Note: - When both the criteria are satisfied only than the gear design should be consider as a SAFE design.

By using all the input values and the above equations are gear design calculation is c complete and the output values are in table-IV.

**Table-IV: Calculated Load Values**

Sr. No	Parameter	Value
1	Tangential Load, $F_t$	555.555 N
2	Face width, $b$	20 mm
3	Beam Strength, $F_s$	6467.1569 N
4	Dynamic Load, $F_d$	3851.541 N
5	Wear Load, $F_w$	10246.0158 N

**VI. SHAFT DESIGN**

**Material for the shafts**

Among the high number of steels, AISI 4340 steel is widely used due to a combination of high mechanical strength, ductility and hardness. This steel is applied in tractor and airplane crankshafts, shafts with high mechanical demands and vehicles in general [7].

Material = AISI 4340

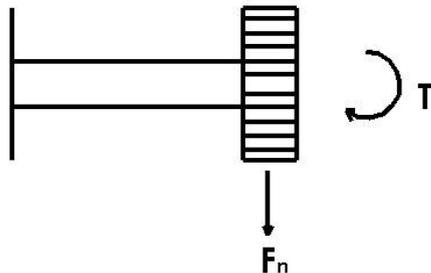
Yield Strength,  $\sigma_y = 470\text{MPa}$

Allowable Strength,  $\sigma_{allow} = 235\text{MPa}$

Shear Strength,  $\tau = 0.57\sigma_y = 267.9\text{MPa}$

Allowable Shear Strength,  $\tau_w = \tau/f.o.s = 133.95\text{MPa}$

**Input Shaft**



**Fig. 5. Normal Load and Torque on the Input shaft**

The length of the shaft (L) needs to be predetermined. The loads are shown on figure 5. The equations used to calculate the diameter of the shaft are given below.

$$\text{Tangential Load, } F_t = \frac{\text{Torque}(T)}{\text{Radius of the gear}} \tag{17}$$

$$\text{Normal Load, } F_n = \frac{F_t}{\cos\phi} \tag{18}$$

$$\text{Bending Moment on the Shaft, } M = \text{Length } (L) \times F_n \tag{19}$$

$$\text{Equivalent Torque, } T_{eq} = \sqrt{(k_b M)^2 + (k_t T)^2} \tag{20}$$

$$\text{Equivalent Moment, } M_{eq} = \frac{1}{2} [(k_b M) + \sqrt{(k_b M)^2 + (k_t T)^2}] \tag{21}$$

Here,  $k_b$  and  $k_t$  are the combined shock and fatigue factors for bending moment and twisting moment respectively. Using the values for the equivalent torque and the equivalent moment, we can find the diameter using the equations given below.

$$\tau = \frac{T y}{J}, \text{ where } J \tag{22}$$

$J$  = Polar moment of inertia of the shaft and  $y$  = radius of the shaft

$$T = \frac{\pi \tau d_1^3}{16} \tag{23}$$

$$\sigma_y = \frac{M y}{I}, \text{ where } M$$

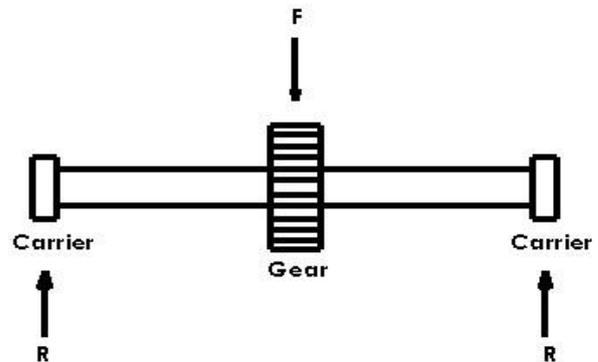
$I$  = moment of inertia of the cross section of the shaft and  $y$  = radius of shaft

$$\tag{24}$$

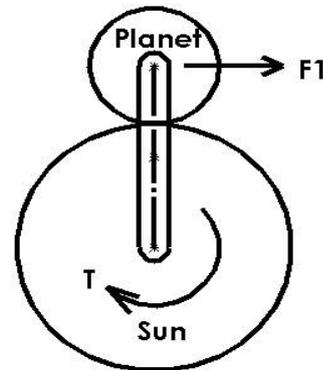
$$M = \frac{\pi \sigma_y d_2^3}{32} \tag{25}$$

Now, after calculating  $d_1$  and  $d_2$ , the higher of the two is selected as the shaft diameter.

**Carrier shaft**



**Fig. 6. Loads on the Carrier Shaft**



**Fig. 7. Torque on the carrier shaft**

The loads on the carrier shaft are shown in figures 6 and 7. The load  $F_1$  shown in figure 7 acting on the carrier shaft can be calculated using the equation.

$$F_1 = \frac{\text{Torque on the carrier}}{\text{Length of the arm}} \tag{26}$$

The normal force acting on the gear  $F_n$  is calculated in equation (18).

$$F = \sqrt{F_1^2 + F_n^2 + 2 \times F_1 \times F_n \times \cos \phi}$$

(27)

$$\text{Moment on Carrier Shaft}(M) = \frac{\text{Reaction force}(R) \times \text{Length of the Shaft}}{2}$$

(28)

$$\text{Torque on Planet}(T_p) = \frac{\text{Torque on sun gear}}{2}$$

(29)

Now, the diameter of the shaft can be calculated using the equations (20), (21), (22), (23), (24) and (25) and following the same procedure as the input shaft. For the output shaft that is connected to the carrier, the same equations are used. The final diameters for the shafts are given in table-V. The carrier shaft and the carrier are shown in figure 8 and figure 9 respectively.



Fig. 8. Carrier Shaft

Table-V: Shaft Diameters

Sr. No	Shaft	Minimum Safe Diameter
1	Input, $d_1$	25 mm
2	Carrier, $d_2$	20 mm
3	Output, $d_3$	25 mm

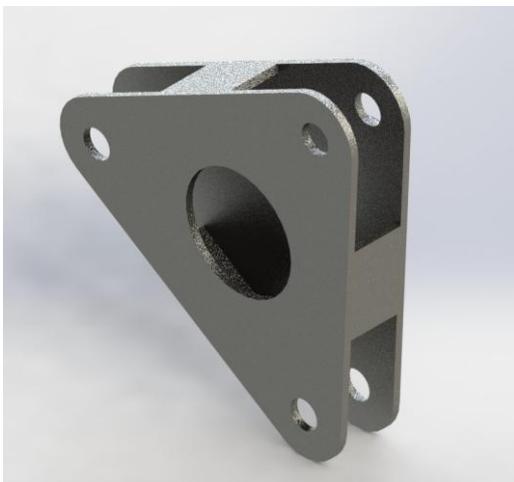


Fig. 9. Planet Carrier

### VII. BEARINGS

The bearing used between the planet gear and the carrier shaft is selected using the below equations which are referred from the PSG Data Book. The bearing type used is the needle roller bearing. Needle roller bearings come with numerous advantages compared to the other members of the

bearing family. They can handle a rigid and larger shaft in a given application. They possess a high load capacity compared to roller bearings of comparable OD or single-row ball. They cost less than the other types of bearings. Needle roller bearings offer great rolling characteristics in a small cross-section [8].

$$\text{Bearing Load} = \text{Equivalent load on the Carrier Shaft}$$

(30)

$$N =$$

$$\text{Speed(RPM) of the planet gear about its own axis}$$

(31)

The life of the bearing ( $L_h$ ) is assumed according to the required time. For our case, the life was 4000 hrs. This life is converted to life in revolutions ( $L$ ) used the below equation.

$$L = 60 \times N \times L_h$$

(32)

The dynamic load rating ( $C$ ) is calculated using the below formula.

$$C = W \times \left(\frac{L}{10^6}\right)^{10/3}$$

(33)

The bearing selected based on the values obtained is the SKF NKIS20 which is shown in figure 10 [9]. The characteristics of the needle roller bearing are as follows: - Inner race diameter = 20 mm, Outer race diameter = 42 mm, Dynamic Load factor = 32500 N, Race width = 20 mm.



Fig. 10. SKF NKIS20 Needle Roller Bearing [9]

### VIII. LUBRICATION SYSTEM

Selecting the proper industrial gear lubricant is important for the long-term efficient operation of the gear drive. There are many factors to consider when selecting an industrial gear lubricant for any application. These factors are summarized in Table VI [10].

Table-VI:  
Factors Affecting Selection of Industrial Gear Lubricants [10]

Factor	Requirement
Gearing type <ul style="list-style-type: none"> <li>• Spur and bevel</li> <li>• Helical and spiral bevel</li> <li>• Hypoid</li> <li>• Worm</li> </ul>	Low slide, low speed Moderate Slide, moderate to high loading High slide, high loading Excessive sliding, moderate to high loading
Loading	Highly loaded industrial gear drives require the use of extreme pressure gear lubricants.
Surface finish	Rougher surfaces require high-viscosity oils, smoother surfaces can use lower viscosity oils

Transmitted Power	As load is increased, viscosity must be increased.
Gear speed	The higher the speed of the gear drive, the lighter the viscosity needs to be.
Materials compatibility	Some types of extreme pressure additives can attack yellow metals such as brass and bronze.
Temperature	The industrial gear lubricant's viscosity must be used must be selected based on the lowest and highest operating and/or ambient temperature experienced.

### Selection of Viscosity of lubricant

#### Method 1

We can calculate the Viscosity using the below equation [11].

$$V_{40} = \frac{7000}{\sqrt{V_1}} \quad (34)$$

where,  $V_{40}$  = Kinematic Viscosity @ 40°C, cSt

$V_1$  = Pitch line velocity of the lowest speed gear, feet per minute (fpm)

= 0.262 X speed (pinion rpm) X pinion diameter (inches)

If there is no oil cooler on the industrial gear drive, it's best to determine the maximum expected ambient temperature during operation and:

1. Increase one ISO Viscosity Grade if the ambient temperature exceeds 95°F (35°C).
2. Increase two ISO Viscosity Grades if the ambient temperature exceeds 122°F (50°C).

#### Method 2

A decent method for deciding the necessary viscosity for a gear drive is the AGMA 9005-E02 "Industrial Gear Lubrication" standard (in the past AGMA-D94). The AGMA 9005-E02 standard shows recommended viscosity grades for industrial gear drives working under ordinary loads over a scope of speeds and surrounding temperatures [11].

Calculated viscosity index (VI) is 76.988 by equation (34) at 40°C and from the plot given in figure 11 at temperature of 70°C VI is 68. So, finally a lubrication oil of viscosity index of 68 is selected.

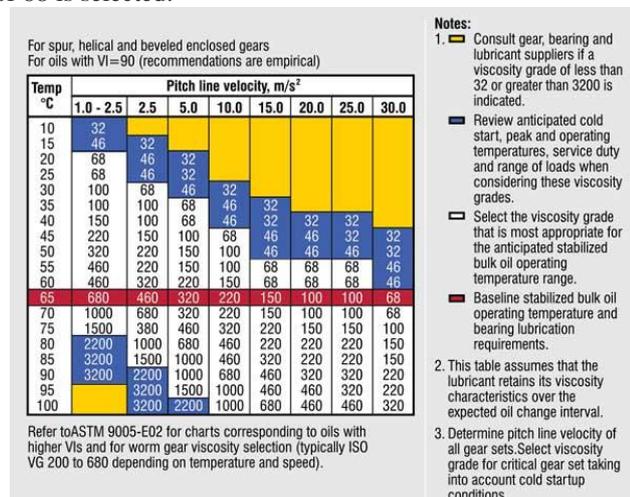


Fig. 11. Viscosity Index Chart [11]

ISO grade 68 oil od SAE20W lube oil selected for our gear box and sump lubrication mechanism is selected for the gear box according to reduction ratio, operating temperature and RPM of carrier.

## IX. GEARBOX HOUSING

The gearbox casing is an important transmission component. The strength of the gearbox casing is an important parameter to be considered while designing. In a power transmission system, the vibrations generated by the gear mesh are transmitted to the gearbox housing through shafts and bearings. The casing is an essential part of the gear box, it provides support to the shafts and bearings and hence the gear loading [12]. The primary requirements of a gearbox casing are rigidity and effective vibration damping properties. The material selected for the housing is Aluminum as it is light weight, has good machinability and higher strength to weight ratio. It is clamped together using six bolts and uses ring gear as a frame. The size of housing is equal to the size of the ring gear and has enough space inside the casing for a smooth lubrication mechanism. Silicon six-hole replaceable flange gasket and sealant is used to seal the lubricant oil however a small hole is provided at the top for releasing of the gases generated by the hot lubricant oil. The casing is shown in figure 12.

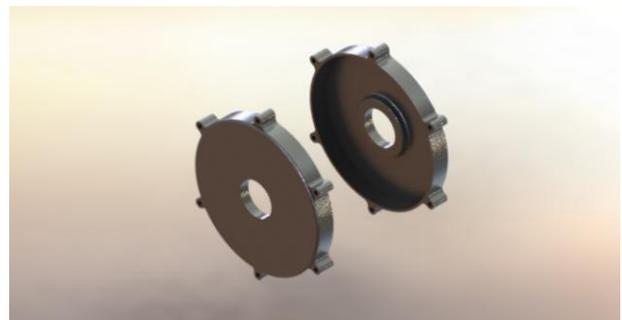


Fig. 12. Gearbox Housing

## X. SIMULATIONS

The CAD models are used to simulate the operating conditions and ANSYS Workbench is used to simulate the structural strength of the gears. The static structural simulation is selected in ANSYS Workbench. The CAD models are made using SOLIDWORKS. The mesh used is shown in figures 12 and 16. The results are shown in the below figures 13, 14, 15 and 17.



Fig. 12. Meshing between Sun and Planet Gear

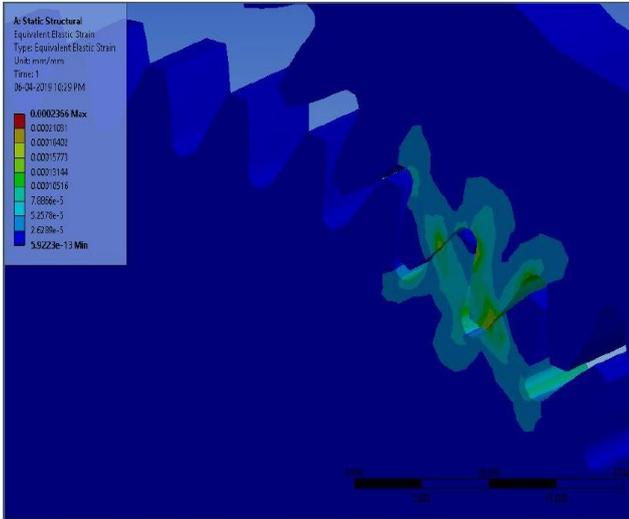


Fig. 13. Deformation in Sun and Planet Mesh

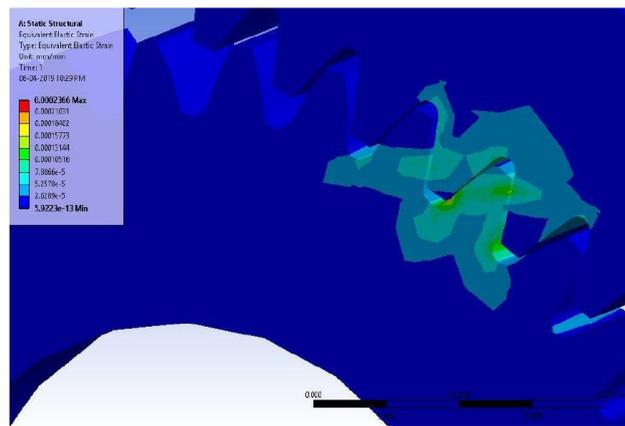


Fig. 14. Deformation in Planet and Ring Mesh

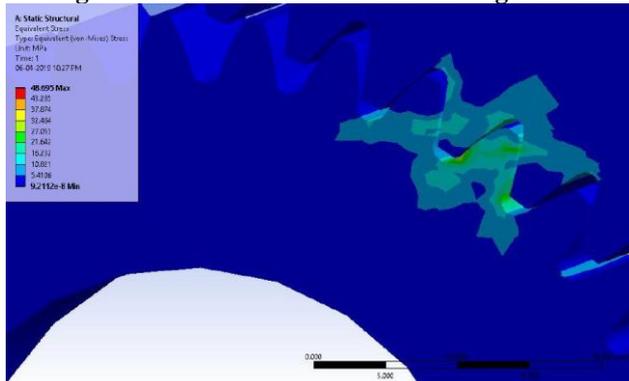


Fig. 15. Stress in Planet and Ring

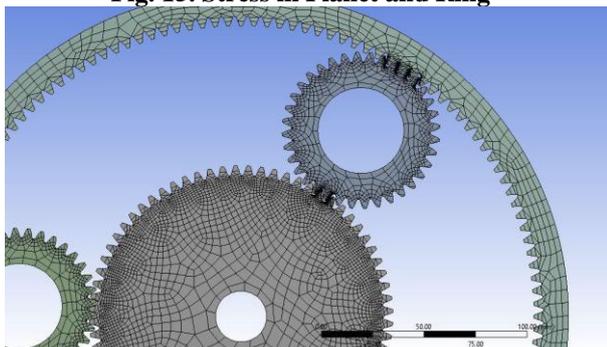


Fig. 16. Mesh

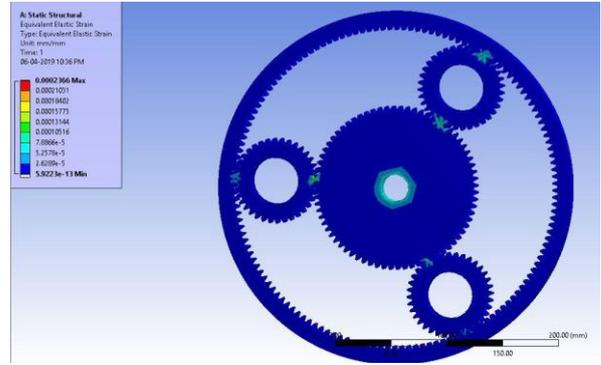


Fig. 17. Deformation in the planetary gear arrangement

## XI. RESULTS AND DISCUSSIONS

According to the above observations and simulations, we can imply that an epicyclic gearbox can be used as an ideal solution for transmitting power to the differential for an electric formula car. The gearbox can efficaciously transmit power and provide enough reduction to satisfy the static and dynamic requirements of the car in a compact design. The finite element analysis done on the gear geometry implied that the gears were safe and the maximum equivalent stress on the gear tooth did not exceed the allowable values under maximum load conditions. The maximum stress seen was around 48.695 MPa shown in figure 15 which is well below the allowable limit. The maximum equivalent strain reached a peak value of 0.0002366 shown in figure 14.

The material used is durable and can work against the wear load and compressive load on the gear teeth. The shaft material used is also suitable for the applied load and the shafts are designed keeping the operating conditions in mind. The bearings used are selected based on their dynamic load rating and are designed to handle a enough number of cycles according to the operation. The casing is made of 6061 alloy aluminium to reduce the overall weight of the gearbox. The carrier is designed keeping in mind machinability and reduced weight.

The gears are designed based on the Lewis equation and checked from dynamic point of view and wear load point of view. The module is kept at an optimum value and the number of teeth on the gears are increased to accommodate the bearing in the planet gear. This is done to satisfy the condition that the bore diameter should be 60% - 70% of the root diameter of the gear.

ISO grade 68 oil od SAE20W lube oil selected for our gear box And Sump lubrication mechanism is selected for the gear box according to reduction ratio, operating temperature and RPM of carrier.

The final specifications of the gearbox are given in table-VII. The CAD models of the final gearbox arrangement is shown in figure 18. The interior of the gearbox is shown in figures 19 and 20.

# Design and Analysis of an Epicyclic Gearbox for an Electric Drivetrain

**Table-VII: Final Specifications**

Material	
Gears	18CrNiMo7- 6 Chrome-Nickel-Moly Carburing Steel
Shafts	AISI 4340 steel
Casing	Aluminium 6061 Alloy
Gear Specifications	
Module	2 mm
Face Width	20 mm
Teeth-Sun	72
Teeth-Planet	36
Teeth-Ring	144
Number of Planet gears	3
Shafts	
Input Shaft	25 mm
Carrier Shaft	20 mm
Output Shaft	25 mm
Bearing Specifications	
Input Shaft	SKF 6010-2Z (Deep Groove Ball Bearing)
Carrier Shaft	SKF NKIS20 (Needle Roller Bearing)
Output Shaft	SKF 6010-2Z (Deep Groove Ball Bearing)



**Fig. 18. Gearbox Assembly**



**Fig. 19. Gearbox interior without carrier**



**Fig. 20. Gearbox interior**

## XII. CONCLUSIONS

The result of this project is a very compact epicyclic gear reduction custom designed for the formula student electric car. We went through an iterative design process and all mechanical drivetrain components were scrutinized and researched to result in the lightest design possible. This report is significant research for the long-term vision of Formula student cars. In this project the gear box is designed and analyzed under static and wear load point of view keeping the optimum light weight design in the mind. A lubrication mechanism and suitable lube oil is also selected for the gear box along with suitable bearings.

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