Design and Analysis of Electric Vehicle Gearbox and Differential

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Abstract - Electric vehicles (EVs) use simpler drivetrain because of the characteristics of the torque speed curve of an electric motor. The Electric Vehicle market is rapidly growing and the transmission unit of an EV is not the same as a Combustion Vehicle. Every EV's torque requirement to obtain the desirable performance is different. This paper focuses on the part development and analysis of a single speed two stage reduction compound gearbox for electric vehicle to obtain all the desirable characteristics. Furthermore, the design factors including material selection for gears and shafts, gear ratio selection on the basis of torque requirement, number of teeth in a helical gear and bearing selection is considered. All the hand calculations were carried out in MS Excel to make it simpler and less time consuming. The gearbox was designed as per the parameters obtained by the Excel calculations on Autodesk Inventor. Then the design was optimized for weight reduction by carrying out the structural analysis of gears and shaft.

***Key Words*:** Electric Vehicle, Gearbox, Two-stage reduction, Helical Gear, MS Excel, Autodesk Inventor.

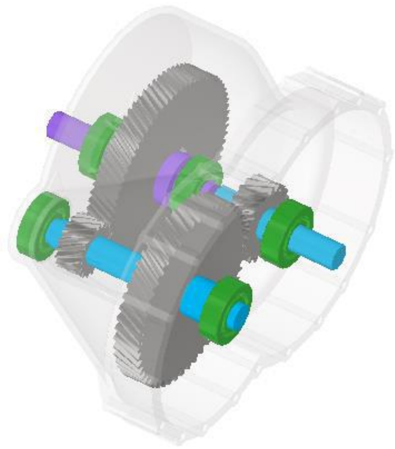
1.INTRODUCTION

Gearbox is a mechanical device to increase or decrease the torque via speed reduction or increase. The gearbox is thus, that important piece of automotive puzzle that gives you optimum power when you need it and conserves power when there is enough momentum already occurring in the wheels. Electric vehicle uses a reducer. However, it does not matter when the load is small, such as flat running, but it is stable to use the transmission to control the motor output when the load is large, such as when going up a hill. There are many different terrains in India, as well as in most cities around the world; a lot of roads require high torque. There is a need for a transmission that is adaptable to various driving environments and that optimizes torque and speed according to the driving speed. The gearbox is dependent on the power output of the electric motor. Most of the gearboxes uses helical gears due to several advantages like they are more efficient, reduce energy costs, run cooler, are quieter, and require overall less maintenance. In short, helical gearboxes are excellent at saving money, time, and energy over the long-term. Helical gears use angular teeth, which are hardened to achieve high efficiency. At all the time 2-3 teeth are in constant mesh with other gear which not only reduces the load on one tooth but also smoothens the overall operation, reduces noise and create less vibration. A differential is a very important part of a gear train. The differential is the part of the gear train which allows both the driving wheels to rotate at different speeds. This is very much required during cornering.

During cornering, the inner wheel has to cover less distance as compared to the outer wheel and in order to do so they both have to rotate at different speeds. This is where the function of differential comes in. Over the years, a lot of research has been carried out on differential design and optimization in order to make it light weight and more efficient by reducing losses. In today's market, a lot of differentials are available for different purposes. Some of the main types of differential are: Open Differential, Limited Slip Differential (LSD), Torsen Differential, Spur Differential, Locked Differential etc. The basic working principle of all the differentials are the same but they vary in packaging size, performance and some added features. In this paper, according to the requirement of the power unit and the vehicle, an Open Differential was designed. At high RPMs, in any mechanical machine there will be thermal and frictional losses. To prevent these two losses cooling and lubrication plays a vital role in deciding the machine efficiency and performance. In gearbox, gear oil is used to lubricate the mating teeth, scuffing, wear, and more severe damage to the gear tooth surface can be prevented. Oil viscosity increases with pressure, making the lubrication thicker, which separates the mating gear tooth despite all of the stress on the gears. Lubrication oil can be used to cool down the surface as well, preventing high temperatures and overheating.

2. Methodology

In almost every electric vehicle there is a gearbox connected to a motor for smooth operations and to achieve high torque values at low rpm. It consists of four major components as follows:



1. Gears: Helical gears are the most common and widely used type of gear used in gearbox. The first step in designing a helical gear is to select a material according to the application. The most common material used in a gearbox are alloy steels (40Ni2Cr1Mo28 for driver gears and 15Ni2Cr1Mo15 for driven gears) having a surface hardness of 55RC and core hardness is greater than 350 BHN.

2. Housing: As shown in Fig.1 Autodesk Inventor is used to design the transmission where transparent one is the transmission housing. Some Transmission housing design are as follows:

- Fatigue/Strength
- NVH
- Weight

3. Shafts: To design a shaft for transmission static and dynamics loads on each point is calculated and after that BMD & SFD should be plotted to calculate bending moment and shear force on each point of the shaft and finally deflection of beam on each point is checked.

4. Bearings: In transmission housing, generally cylindrical roller bearings are used to allow high loads on shaft due to tangential force of gears while in operating condition. Sometimes deep-groove ball bearings are also used where load is less in dynamic conditions.

The overall design of the Differential system was carried out in Autodesk Inventor. The main components of a Differential system are: spider gears, side gears, differential housing and spider gear pin. The basic calculations for the gear parameters were carried out in MS Excel. Iterations were performed to obtain the best gear design as per the load requirements of the gear train and vehicle. These iterations were carried out to increase the load bearing capacity as much as possible and keeping the weight and size as small as possible. MS Excel was used instead of MATLAB or any other software was the ease of use and increase the understandability of the calculations. The design approach was the basic approach as per given in the design data book. The material selection was done keeping in mind the strength to weight ratio and availability. A thorough market research was carried out by consulting the manufacturing companies and local machining shops in order to select the best material for our requirement.

Table -1: Material Selection Chart

Material Properties	20MnCr5	EN 8	EN 19
Tensile Yield Strength (MPa)	750	433	470
Ultimate Tensile Strength (MPa)	100	650	745
Density (kg/m^3)	7850	7800	7800

Assumption of the gear parameters and then calculate the design load and actual load on the gear and then changing the parameters in each iteration to get the design load greater than the actual load so the gears can bear the load from the power train. Then the final design is carried out in Autodesk Inventor of the gears and other components. After the design, the optimization of the components was carried out by doing analysis and removing the material wherever not required and hence reducing the overall weight of the system.

3. Design and Analysis

3.1. Analytical Calculations

3.1.1. Gearbox

The loads acting on the gear are bending load and contact load. According to our application following is done to calculate the center distance between the gears.

1. For moderate shock loading, at peak power of 80 kW, bending stress and contact stress on a gear is calculated as follows:

$\Sigma b=(1.4\times Kbl\times\sigma 1)/(nK\sigma)=(19.1\times10^6\times W)/(DP\ n)$

And, σ-1=$(0.35\times\sigma u)+120$

$=(0.35\times1550)+120=662.5$ N/mm2

$\sigma b=(1.4\times0.7\times662.5)/(2.5\times1.5)=173.1$ N/mm2

$\sigma c=CR\times HRC\times Kcl\ =26.5\times55\times0585$

$=852.6$ N/mm2

Where,

Kbl: Life Factor

Kσ: Fillet stress concentration factor

σ-1 : Endurance limit in revised bending

σc : Contact Stress (N/mm2)

σb : Bending Stress (N/mm2)

n: Factor of safety

2. Calculation of center distance:

After calculating stresses on the gear center distance is calculated as follows:

$a\geq(i+1)3\ \sqrt[3]{((0.7/\sigma c\ )^2\times(E[Mt])/i\varphi)}$

$[Mt]=K0\times KKd\times Mt$

And,
$Mt=(P\times60)/2\pi N=(80\times10^3\times60)/(2\times\pi\times13000)=58.795$ N-m

K0=1.25 , for moderate shocks

(Assume, KKd=1.3)

Therefore, $[Mt\ ]=1.25\times1.3\times58.795$

$[Mt\ ]=95.54$ N-m

Assume, $\varphi=0.5$ and $E=2.1\times10^5$ N/mm2

$a\geq(2.5+1)3\ \sqrt[3]{((0.7/852.6)^2\times(2.1\times10^5\times95.54)/(2.5\times0.5))}$

$a\geq94.82$ mm

Hence, a = 95 mm;

Where,

a: Center to center distance between driver and driven gear; Mt: Pinion Torque; i: Speed Reduction Ratio

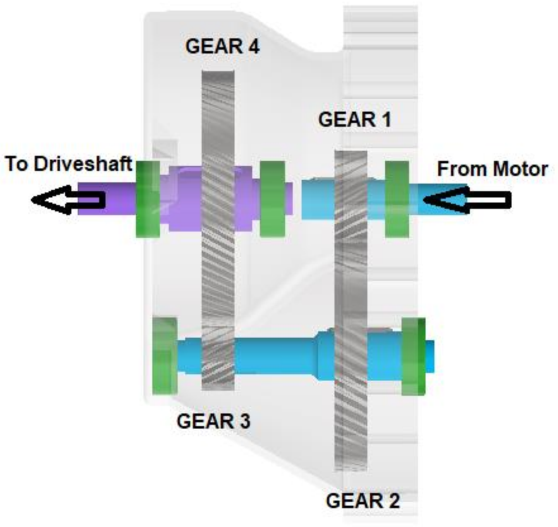


Fig 2. Preliminary Design

Center-to-center distance and speed reduction ratio is known now. Using software Autodesk Inventor 2021 we can directly calculate the module, number of teeth and pitch circle diameter of the gear by directly inserting the calculated values. Also, power, speed and torque values can be calculated at each reduction.

Table -2: First Gear Reduction

Parameter	Value		Unit
Desired Gear Ratio	2.250		–
Center Distance	95		mm
Module	2.250		mm
Pressure Angle	20		degree
Helix Angle	30		degree
	Gear 1	Gear 2	
No. of Teeth	21	52	_
Face Width	20	20	mm
Pitch Diameter	54.56	135.1	mm
Power	80	78.4	kW
Speed	3466	1399.73	rpm
Torque	220.411	534.863	N-m

In 2-speed gearbox, compound gear arrangement has been used in which output power of 80 kW from the motor is directly transmitted from Gear 1 to Gear 2, increasing the torque to 534.863 N-m at 3466 rpm.

Table -3: Second Gear Reduction

Parameter	Value		Unit
Desired Gear Ratio	4		–
Center Distance	95		mm
Module	1.750		mm
Pressure Angle	20		degree
Helix Angle	30		degree
	Gear 1	Gear 2	
No. of Teeth	19	75	_
Face Width	20	20	mm
Pitch Diameter	38.394	15.554	mm
Power	78.4	76.832	kW
Speed	1399.73	354.60	rpm
Torque	543.864	2069.078	N-m

Final drive speed at peak torque of motor will be 354.60 rpm and torque will be 2069.078 N-m.

3.1.2. Differential System

- Gears

Once the calculations are completed, the design of the system starts. The initial design consists of the gears designed according to the parameters obtained from the latest iteration of the Excel sheet. The calculations were carried out for the maximum required gear ratio between the 2 wheels under cornering. According to this, the number of teeth and module of the gears were assumed.

Table -4: Gear Ratio Calculations

S No.	**Parameter**	**Value**	**Unit**
1	Radius (R)	0.33	m
2	π	3.14	_
3	Front Track Width	1.54	m
4	Turning Radius	5.1	m
5	Outer Wheel Trajectory Radius	6.64	m
6	Inner Wheel Trajectory Radius	3.56	m
7	(A) Circumference of Wheel	2.09	m
8	(B) Circumference of Outer Wheel Trajectory	41.72	m
9	(C) Circumference of Inner Wheel Trajectory	22.37	m
10	(D) Ratio (C/A)	10.72	_
11	(E) Ratio (B/A)	19.99	_
12	(F) Ratio (E/D)	1.9	_
13	**Reduction Between Spider**	**1.9**	_

	and Side Gears		
14	Number of teeth on Bevel Pinion	16	_
15	Number of teeth on Bevel Gear	30	_

With the help of Design Data Book, the other parameters were decided and the first iteration of gear parameters is obtained.

Table -5: Initial assumptions for first iteration

S No.	Parameter	Symbol	Value	Unit
1	Gear Ratio	i	1.9	_
2	Minimum number of teeth on Pinion	$Z_{p\ min}$	15.13	_
3	Minimum number of teeth on Bevel	$Z_{g\ min}$	7.96	_
4	Number of teeth in Pinion	Z_p	16	_
5	Number of teeth in Gear	Z_g	30	_
6	Module	m	2	mm
7	Diameter of Pinion	D_p	32	mm
8	Diameter of Gear	D_g	60.8	mm
9	Addendum	h_a	2	mm
10	Dedendum	h_f	2.5	mm
11	RPM of Gear	N_g	504.1454	rpm
12	RPM of Pinion	N_p	957.87626	rpm
13	Power	P	80000	W

These parameters were put into the calculations of Design Stresses and Induced Stresses. The formulas were obtained from different sources for calculation of design stresses and induced stresses.

Table -6: Design Stress Calculations

S No.	Parameter	Symbol	Value	Unit
14	Ultimate Tensile Strength	σu	1550	N/mm2
15	Rockwell Hardness	HRC	58	HRC
16	Pinion Pitch cone angle of Pinion	γp	27.8	degree
17	Pinion Pitch cone angle of Gear	γg	62.2	degree
18	Beam Strength	σb	516.67	N/mm2
19	Virtual Number of teeth on Pinion	Z'p	18.08	_
20	Virtual Number of teeth on Gear	Z'g	65.27	_
21	Pitch cone distance	A0	68.71	mm
22	Lewis Form Factor for Pinion	Y'p	0.404	_
23	Lewis Form Factor for Gear	Y'g	0.510	_
	Design according to	Pinion		
24	Face Width from Cone Distance		22.90	mm
25	Face Width from Module		20.00	mm
26	Final Face Width	b	20.00	mm
27	Bending Stress	F_b	6373.78	N
28	Ratio Factor	Q'	1.6	_
29	Brinell Hardness	BHN	601	BHN
30	Load Stress Factor	K	5.78	_
31	Wear Strength	F_w	5622.02132	N

In the above table, for safe design, the Bending stress should be greater than Wear Strength.

Table -7: Induced Stress Calculations

S No.	Parameter	Symbol	Value	Unit
32	Pitch Line Velocity	V	1.60	m/s
33	Tangential Force	F_t	49846.19	N
34	Velocity Factor	K_v	0.789	_
35	Application/Service Factor	K_a	1.25	_
36	Load Distribution Factor	K_m	1	_
37	Effective Load	F_{eff}	38822.54	N
38	Factor of Safety	FOS	2	_
39	Wear Load	F_W	77645.0859	N
40	Mean Radius of Pinion	r_{mp}	10.7	mm
41	Mean Radius of Gear	r_{mg}	20.3	mm
42	Pinion Pitch Error	e_p	0.00977439	mm
43	Gear Pitch Error	e_g	0.00996904	mm
44	Pitch Error	e	0.01974343	mm
45	**Maximum Tangential Force**	F_{tmax}	**62307.7386**	**N**
46	Gear Deformation Factor	C	217.177779	N/mm
47	**Buckingham's**	F_d	**7736.9838**	**N**

	Dynamic Load		**7**	
48	**Final Effective Load**	F_{Eff}	**70044.7225**	**N**
49	Design FOS	N_f	1.1085073	_
	Wear Load should be greater than Bending Stress		**Pass**	

- Shafts

There are 2 main shafts to be designed in a differential system: the driving shaft and the spider gear shaft. The driving shaft is the one which transfers the torque from the differential system to the wheels and the pinion shaft is the one which transfers the torque from the differential housing to the side gears through spider gears. The spider gear shaft only takes the bending load so the calculations were carried out accordingly. But the side gear shaft takes all the forces: bending, axial and torsional. The calculations for the side gear shaft are carried out accordingly.

Table -8: Shaft Calculations

SPIDER GEAR PIN				
S No.	**Parameter**	**Symbol**	**Value**	**Unit**
1	Torque	T	1515.3242	Nm
2	Pin Length	L	98.22	mm
3	Force	F	4984.98627	N
4	Distance between Bevel Gear	L_b	77.82	mm
5	Distance between Bevel Gear and Casing	L_{bc}	10.2	mm
6	Reaction Force	R	6291.76755	Nmm
S No.	**Bending Moment at**		**Value**	**Unit**
1	A		0	Nmm
2	B		50846.8599	Nmm
3	C		50846.8599	Nmm
4	D		0	Nmm
7	Diameter		14.64	mm
SIDE GEAR SHAFT				
S No.	**Parameter**		**Value**	**Unit**
1	Ratio of OD and ID	C	0.6	_
2	Load Factor	K_t	1	_
3	Shear Force	τ	82.53	N/mm^2
4	Outer Diameter	d_o	47.54	mm
5	Inner Diameter	d_i	28.52	mm

- Bolts

The differential system is connected to the gear train by bolt and nuts. There are particular calculations for these bolts. The calculations decide the number and the size of bolts to be used.

S. No.	**Parameter**	**Symbol**	**Values**	**Unit**
1	Power	P	80	kW
2	RPM	n	504.1454	
3	No. of Bolts	N	8	
4	PCD	D	140	mm
5	Tensile strength of Bolt	S_{yt}	800	N/mm^2
6	FOS	fs	3	
7	Shear Stress on Bolts	S_{sy}	461.6	N/mm^2
8	Permissible Shear stress	τ	153.8667	N/mm^2
9	Torque transmitted by the shaft	M_t	1516093	N-mm
10	Diameter of the bolt	d	4.734365	mm

3.2. CAD Modelling

3.2.1. Differential System

The parameters obtained from above were used to create a 3D model of the gears in Autodesk Inventor 2021. The gears were automatically generated with the help of Autodesk Inventor Design Accelerator. The material was specified in the CAD model itself. The gears were edited for the shaft holes and hubs.

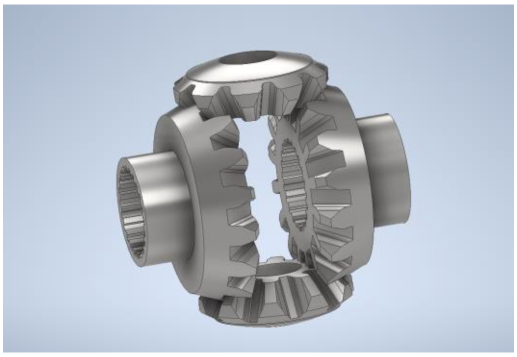


Fig 3. Gear Design

The hubs and splining of the side gears were designed according to the calculations. The casing was designed keeping in mind the torque that has to be transferred through the casing to the side gears and the packaging of the differential system. The differential pin was designed to take the load from the casing while the torque transfer happens.

Fig 4. Casing Design

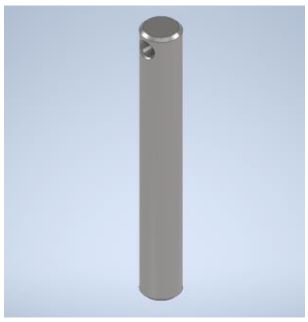


Fig 5. Pin Design

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3.2.2. Gearbox

The Gears are designed using the Autodesk Inventor Design Accelerator. The splines and weight reduction holes were made according to the calculations.

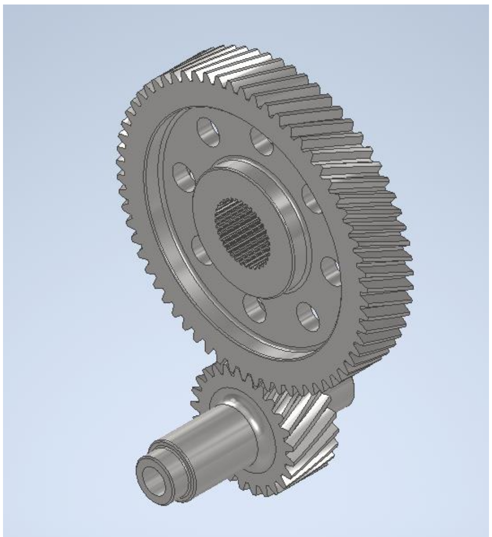

Fig 6. Reduction Gear Pair

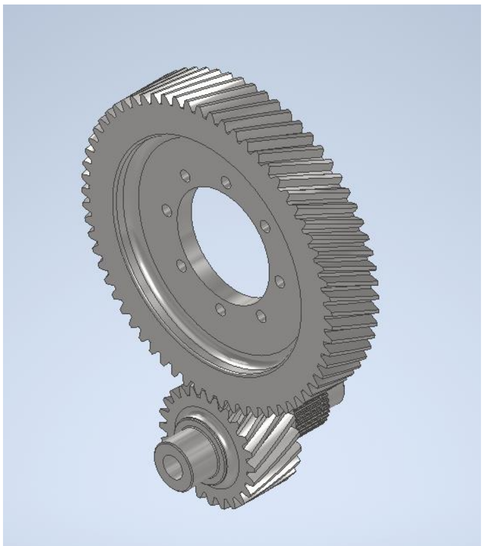

Fig 7. Second Reduction Gear Pair

The gearbox casing was designed so keeping in mind the packaging as well as the manufacturability of the casing. The casing will have to manufactured using casting technique so the design has to be compatible with the casting mold and avoid casting defects.

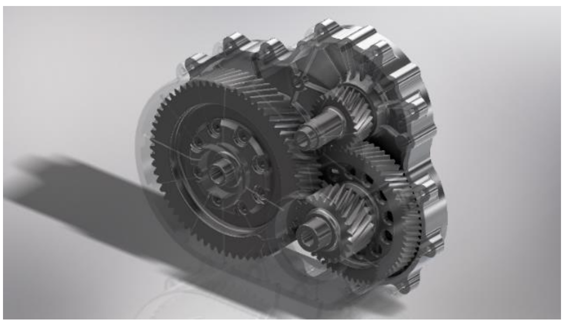

Fig 8. Gearbox Casing

3.3. Finite Element Analysis (FEA)

3.3.1. Gearbox

The structural analysis of the gear set was carried out on ANSYS Workbench 2020 R2. All the analysis were carried out in the Automotive Design Lab, Automobile Hanger, SRM IST, KTR Campus. The boundary conditions were set and the total deformation and equivalent stress was calculated. One of the gears was fixed and the load was applied on the tooth of the other gear.

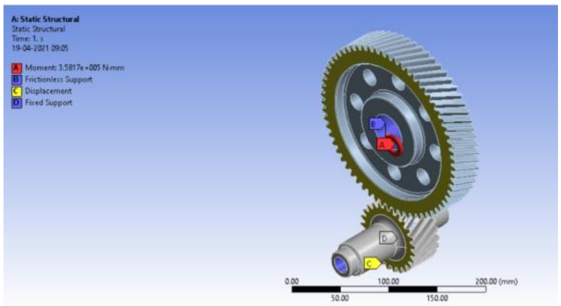

Fig 9. Boundary Conditions for First Reduction Gear Pair

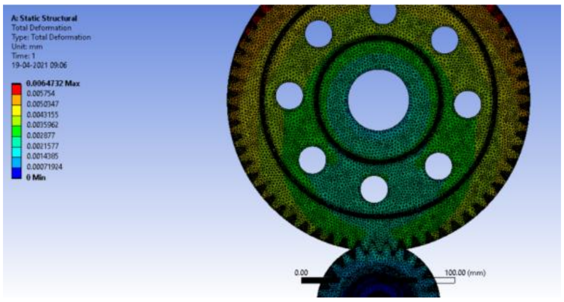

Fig 10. Total Deformation for First Reduction Gear Pair

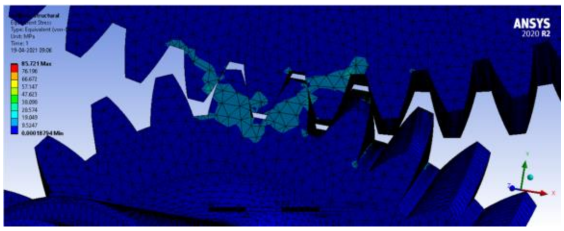

Fig 11. Equivalent Stress for First Reduction Gear Pair

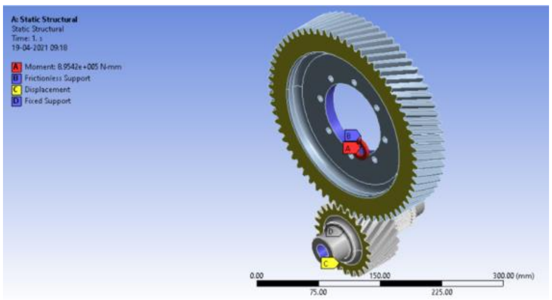

Fig 12. Boundary Conditions for Second Reduction Gear Pair

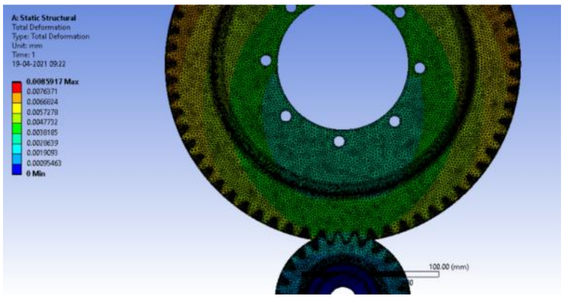

Fig 13. Total Deformation for Second Reduction Gear Pair

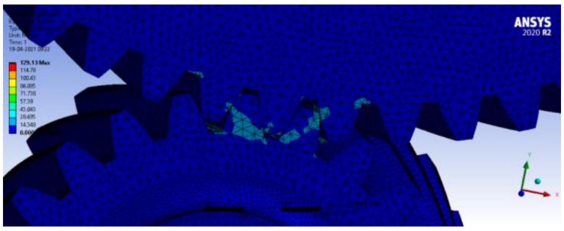

Fig 14. Equivalent Stress for First Reduction Gear Pair

In the analysis of shafts, the inner part of the shaft is fixed and the torque is applied on the gear face.

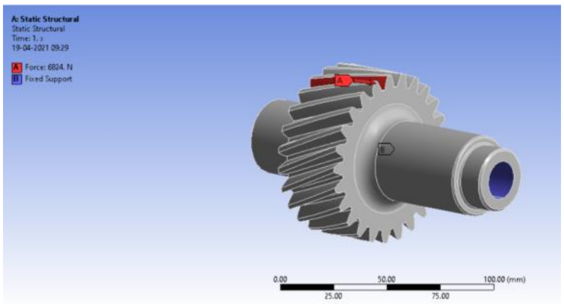

Fig 15. Boundary Conditions for First Reduction Shaft

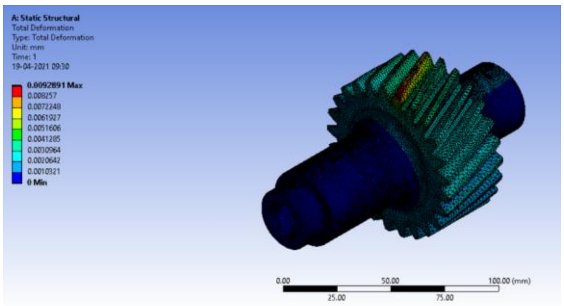

Fig 18. Total Deformation for First Reduction Shaft

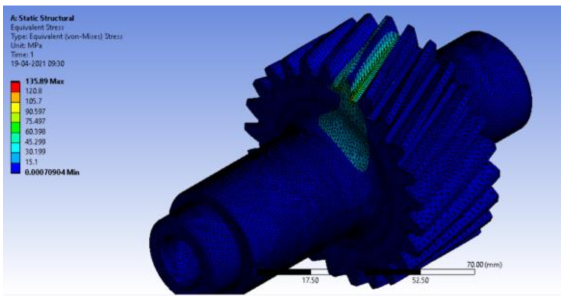

Fig 16. Equivalent Stress for First Reduction Shaft

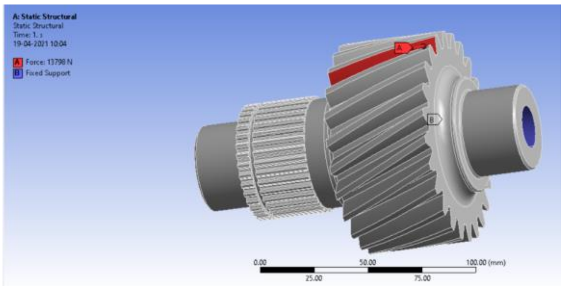

Fig 17. Boundary Conditions for Second Reduction Shaft

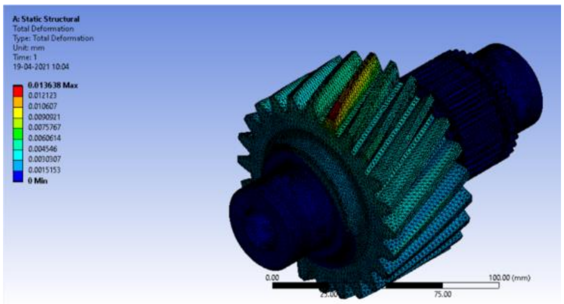

Fig 19. Total Deformation for Second Reduction Shaft

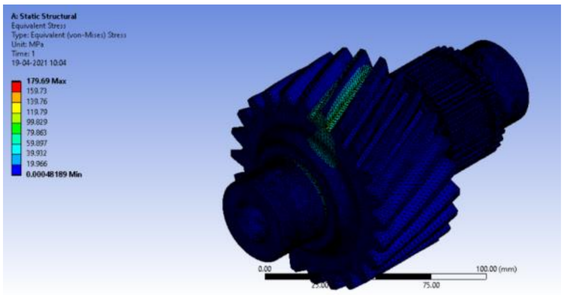

Fig 20. Equivalent Stress for Second Reduction Shaft

3.3.2. Differential System

The boundary conditions for the analysis of the differential bevel gears were to give the spider gears fixed support and the side gears frictionless support. Then torque is applied on the both the side gears in opposite direction.

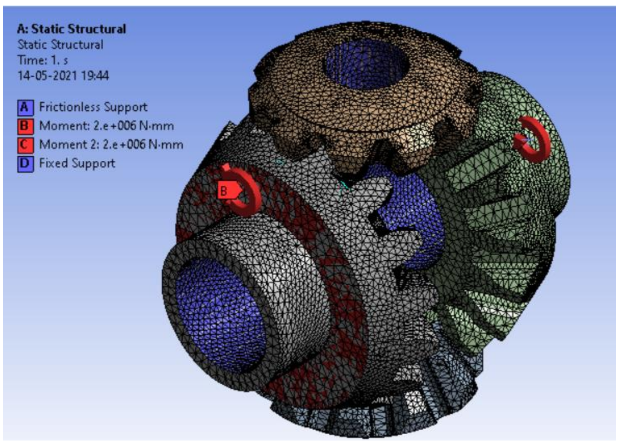

Fig 21. Boundary Conditions for Differential Bevel Gears

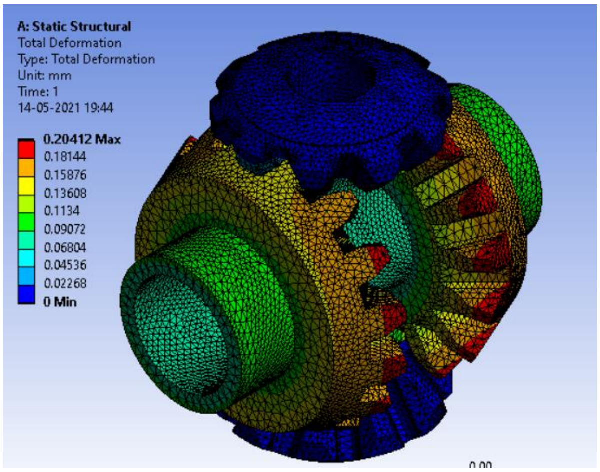

Fig 22. Total Deformation for Differential Bevel Gears

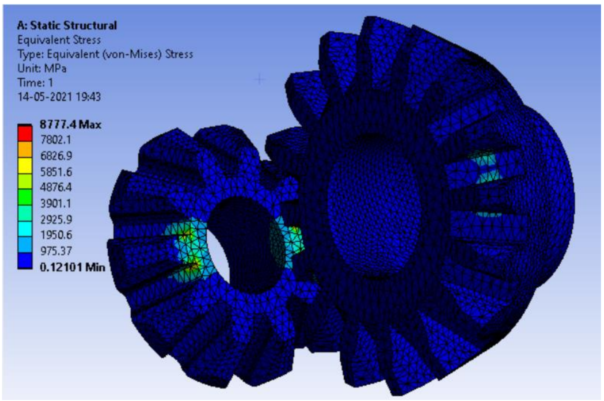

Fig 23. Equivalent Stress for Differential Bevel Gears

3. RESULTS AND CONCLUSIONS

The Gearbox Total weight was 38.625 kg in the first iteration. Then after the optimization of the design the weight came down to 34.970 kg including the weight of the bearings. The analysis results shows that the design is safe with the equivalent stress maximum at the gear meshing surface of 85.721 MPa and 129.13 MPa at first and second reduction respectively.

The results of the analysis show that the maximum deformation is on the teeth which are not meshed with the spider gears. The equivalent stress is maximum at the contact surface of the gears which is 8777.4 MPa.

The main purpose of this study is achieved by multiplying the torque using 2-speed reduction gearbox with a gear reduction ratio of 2.5 and 4 which will increase the peak torque coming from the motor to 10 times to the final drive. From the structural analysis, it can be concluded that the maximum stress and maximum deformation is on the side gears as the material of both gears are same. So, the design can withstand the loads and hence the design is safe.

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