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Design and Failure Analysis of Two-Stage Reduction Gearbox for an All-Terrain Vehicle

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Abstract: The automobile sector demands an all-0Qterrain vehicle. Wherein, single seater and moderated weight are desirable by considering the service environment. In this context, high torque is being employed while designing this kind of automobile systems. Hence, two stage reduction gearbox is a vital part of transmission in order to obtain the necessary speed and torque. Henceforth, sustainable gear design is must for such application. Therefore, the present study is exposing the failure modes of the gear box considering the dynamic condition of all-terrain vehicles. Furthermore, failure has been eliminated through simulations carried out by Ansys 18.2 and KissSYS software. wherein, the obtained results of the simulation are compared with those of theoretical calculations. On the other hand, the designed gear box has been assessed on the scale of reliability while in the dynamic condition in conjunction with assembly components. Moreover, FoS is increased from 0.77 of the previous level to 2.52 existing level due to the alteration of gear material, face width, module and number of gear teeth. This, in turn, eliminates the gear hunting phenomenon resulting in premature failure, one of the reasons for the gearbox.
Keywords: KissSYS, CAD Modeller, FEA Solver, Two Stage Reduction Gearbox, Gear Failure analysis.

I. INTRODUCTION

In this paper, we will be discussing the failure analysis of previous year gearbox and design methodology of two stage reduction gearbox which is majorly used in SAE Baja all-terrain vehicle (BAJA SAE is an intercollegiate engineering design competition for undergraduate and graduate engineering students. The object of the competition is to simulate real-world engineering design projects and their related challenges) [1], its FEA and gearbox reliability through KissSys.

A correctly designed gearbox can win you races, and other dynamic events and power loss can be minimum and smooth operation can be carried out without compensating on efficiency. The main goal when developing a vehicle transmission is to convert the power from the engine into vehicle traction as efficiently as possible, over an adjustable and desired velocity. Acceleration and velocity should be considered hand by hand, as races are decided in seconds and on the contrary, we want extended service life also for the same to avoid pre-mature failure and extend its service life.

Gears are one of the most critical and efficient mechanical components that have been used over the years for transmitting power or speed reduction. Geartrain design includes material selection, designing gears according to the beam and wear strength, validation through FEA.

Gear geometry selection plays a vital role in its efficiency, power transmission capacity, noise vibration, and harshness, etc. After examining all the factors and narrowing down our approach we finalized to use spur gear as they have excellent transmission capacity with minimum losses, manufacturing time is reduced and simplicity in design increases, as there is no moving or sliding contact between the gears, so it eliminates helical gear.

Another aspect which one needs to consider is its compactness of gearbox as it is connected to secondary pulley of CVT (continuously variable transmission)[2], In refereed study the author has discussed CVT’s working parameters with different set of variations and performance analysis and on another end drive shafts are connected to tires, so its overall design should not hamper the space as it would be near about engine and engine being one of the heaviest parts of entire buggy, slight change in its position would create unnecessary problems with tire assembly, roll cage spacing, wheelbase and wheel track measurements, Centre of gravity location which will further effect on vehicle dynamic calculation’s and performance of Buggy.

Herewith, for the modelling of the gear train, shafts, and another component we have used solidworks 2017 (sp x5) and followed by the assembly in the same. For other data calculation such as reliability, all the forces on gear, bearing life and serviceability and other engineering data useful in consideration of gear train were taken from KISSsys (2017) and Finite Element Analysis was taken into consideration with help of Ansys (18.2).

II. DESIGN REQUIREMENT'S

Gearbox drive train should not only be efficient enough to transmit power but compact and robust in design with enough FoS (factor of Safety). Previous year gearbox specification is shown below, and its failure analysis was carried out and propelled to the forefront in the investigation as following failures: -

1) Gear Material used didn't have quite good Mechanical Properties which are desired by Gear
2) Unequal Number of teeth on Gear and pinion were selected which could have been one of the reasons
3) Even Number of teeth were prevailing on Gear and Pinion which caused Gear Hunting and one of the Primary reasons for failure
4) Compactness and light weight parameters were given more importance then FoS which caused bending failure
5) Reliability of entire gearbox was around 500-1000 Hours with 0.77 as FoS, which couldn't sustain longer cyclic loads

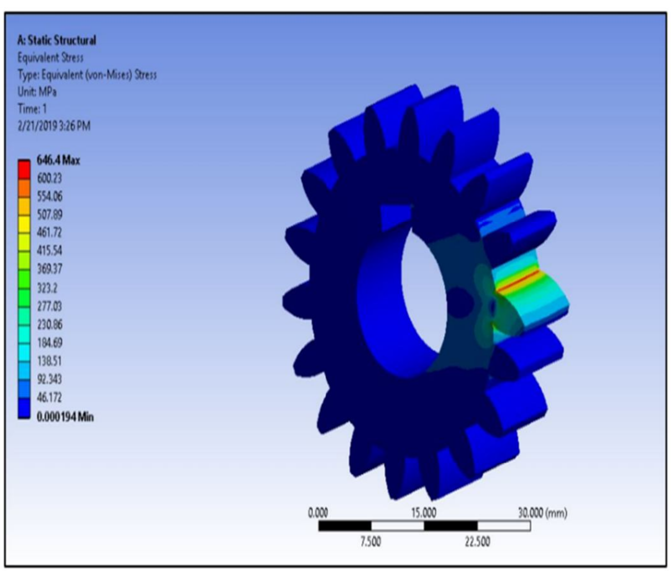
A. 2017 gearbox Component's Listing

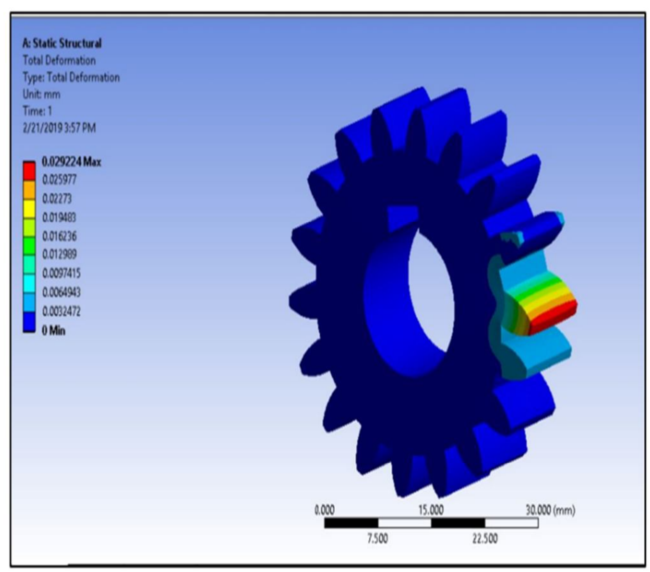
Sr No.	Parameter	Value	Remarks
1	Pinion 1	18 teeth (Module = 2)	36 mm PCD
2	Gear 1	56 teeth (Module = 2)	112 mm PCD
3	Pinion 2	18teeth (Module = 2)	36 mm PCD
4	Gear 2	70 Teeth (Module = 2)	140 mm PCD
5	Face width (Same for all)	12 mm	
6	Gear material (Same for all)	EN 8	
7	Hardness (Same for all)	310 BHN	
8	Roller ball Bearing	SKF 6204	
9	Gear and Shaft Material	EN 8	Post Heat Treatment

Table 1 Specification of 2017 Gearbox

B. PCD = Pitch Circle Diameter of the Gear

As shown in Table 1, all gears were of Spur gear category with 20degree full depth involute teeth and nominal shaft dia. was kept same, D = 15mm, further proceeding with Finite Element analysis for the same was carried out by applying equal force on the face of gear and the following results were derived





Equivalent Von Moises stress = 645 Mpa
FoS= 500/645 = 0.77

Total deformation = 0.025 mm

Figure 1 Analysis of 2017 Gearbox

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			Gear 1	Gear 2
Normal angle	2 mm	Number of teeth (z)	18	56
Pressure angle at normal section	20	Face width (b) mm	12	12
Gear 1	Spur	Profile shift Co-efficient (X')	0.4090	-0.4090
Helix angle at reference circle	0	Quality (ISO 1328:1995) (Q)	6	6
Centre distance	74 mm			
Result with load spectrum (own input)				
Contact ratio	1.561		Gear 1	Gear 2
		Actual tip circle (mm)	41.636	114.364
		Root safety	0.796	0.745
		Flank Safety	0.719	0.719
		Safety against scuffing (integral temp)	3.103	
		Safety against scuffing (flash temp)	4.146	

Table 2 Values in KissSYS

			Gear 1	Gear 2
Normal angle	2 mm	Number of teeth (z)	18	70
Pressure angle at normal section	20	Face width (b) mm	12	12
Gear 1	Spur	Profile shift Co-efficient (X')	0.3703	-0.3703
Helix angle at reference circle	0	Quality (ISO 1328:1995) (Q)	6	6
Centre distance	88 mm			
Result with load spectrum (own input)				
Contact ratio	1.581		Gear 1	Gear 2
		Actual tip circle (mm)	41.481	142.519
		Root safety	0.972	0.946
		Flank Safety	0.643	0.702
		Safety against scuffing (integral temp)	3.859	
		Safety against scuffing (flash temp)	6.055	

Table 3 Values in KissSYS

As shown and results were obtained in Fig.1 and Fig.2, 1st and 2nd stage reduction gear pair analysis was carried out in FEA Solver and KissSYS. Correspondingly Pinion FoS was obtained 0.796 and 1.13 respectively. So, we have examined that Gear pair was not at all safe for the usage in race and service life was not at all desired with so less in the scale of Hours. We plotted the System Failure graph and Gears reliability graph for 10,000 Hours as shown in Fig.4 and Fig.5 and it can be clearly witnessed that Gears are not reliable more than 500-1000 Hours. Hence decreasing overall gearbox life and causing failure in the pre-mature stage.

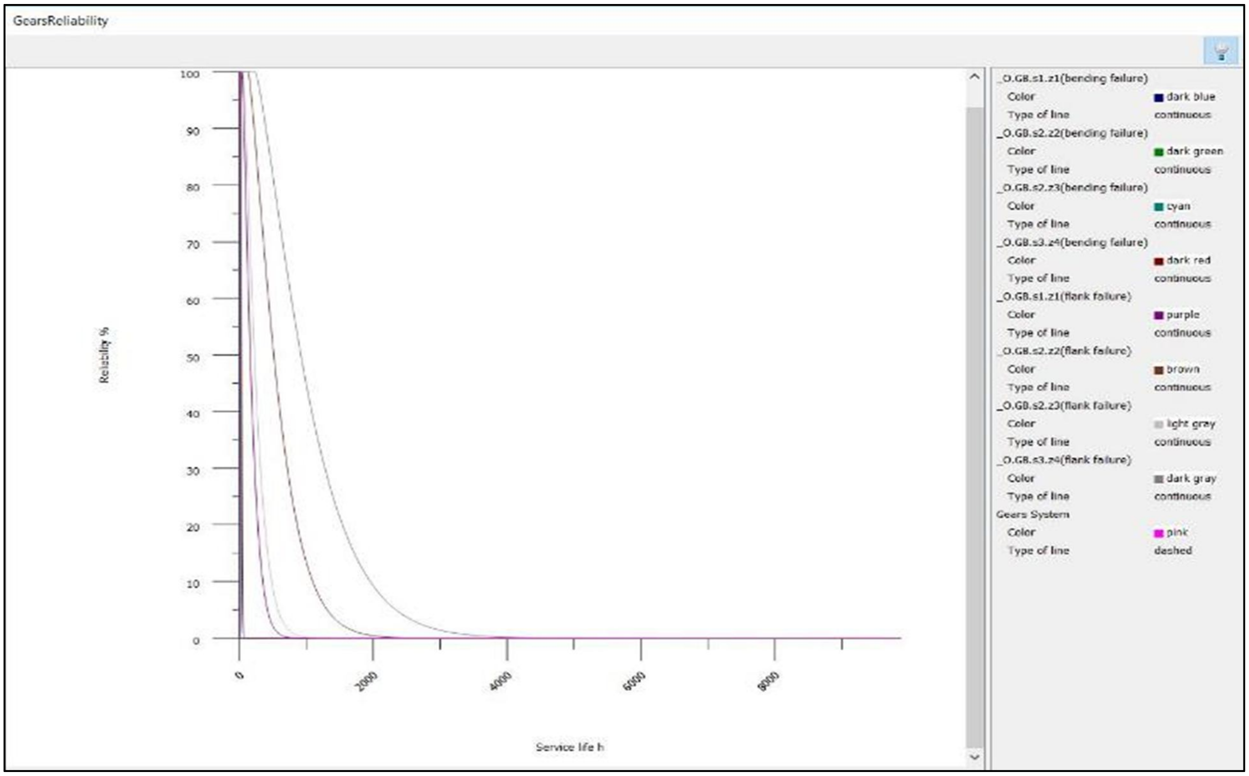

Figure 2 Gear reliability vs Service hours

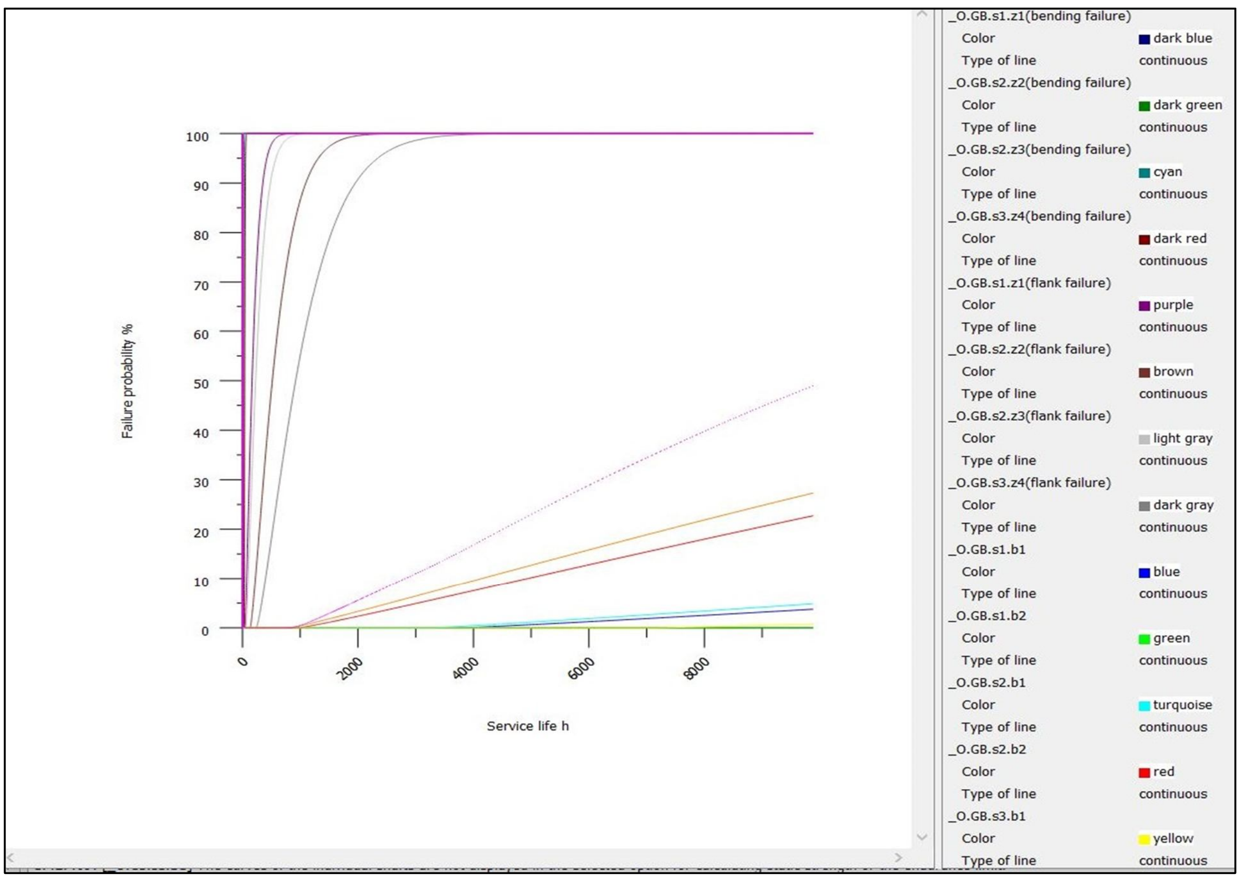

Figure 3 System failure probability vs Service Life (Hours)

The Old gear system had very little probability to cross 1500 Hours mark as shown in Fig.4 and Fig.5, all the dynamic components like Gears and Bearings were failed even before the designated life or the required life. So, compromising FoS (Factor of Safety) over size and compactness caused the failure which was a setback and another factor could be Gear Hunting [3] because for first pair pinion has 18teeth and gear has 56 teeth, so by calculating GCD (Greatest Common divisor) we can say that 2^{nd} tooth would have come under contact after every rotation and on that particular tooth stresses have been marginally higher. The second pair also had the same problem having 18 teeth on pinon 70 teeth on the gear, so every time 2^{nd} tooth would have been under contact after rotation. We will eliminate this problem in our new proposed design and another failure which were caused by bending and wear will also be eradicated from current operational design. So, to overcome these failures we have proposed a robust, Light Weight, compact, performance assured gearbox design with 10,000+ hours of service life reliability.

III.DETAILED DESIGN OF ATV (ALL TERRAIN VEHICLE) GEARBOX

A. In case of All-Terrain vehicle, as per SAE BAJA 2019 rulebook, engine model is fixed M19 Vanguard commercial engine (19L232-0054 G1) and their specifications are given below:

SAE Baja in 1976 the Society of Automotive Engineers (S.A.E.) launched the 'Baja' collegiate competition. This competition series challenges engineering students to design, engineer and build from the ground up an off-road vehicle, each powered by a Briggs & Stratton engine. Beginning in 2017, this program will now be powered by our M19 Vanguard commercial engine (19L232-0054 G1).[1]

SPECIFICATIONS		
MODEL/TYPE(S) 19L232-0054 G1	DISPLACEMENT 305cc	BORE/STROKE 3.12” / 2.44”
COMPRESSION RATIO 8.1 to 1	FACTORY TIMING 23 degrees BTDC	HP (GROSS)* 10.0 hp
OIL CAPACITY (DRY) 24 ounces	FACTORY SET RPM 3,800 RPM	FUEL TYPE 87 Octane

Table 4 Specifications of Engine

Our Desired output based on Traction requirement considering all resistances Such as Rolling, Driving, Air, Acceleration, Gradient, and Traction available we have following data Pre-requisite

Engine Idealising RPM: - 1750$\pm$100 RPM
Engaging RPM: - 1800 RPM
Maximum RPM available from Engine: - 3800 RPM
Max. Torque available - 20 N.m
Desired output RPM – 407 RPM
FoS (Factor of safety) $\geq$ 2 for gears

The desired gearbox for any all-terrain competition should be designed for the maximum output of Velocity, Torque, and acceleration and on the contrary gears FoS should not at all be compromised for several reasons such as Weight reduction, Size or manufacturing capability or any other reason because in actual dynamic condition load may differ from the calculated one so it’s necessary to have enough FoS without compromising on performance and analysis should be done accordingly.

B. Computation Of Gear Ratio For Gearbox Minimum Ratio According To Traction Requirement As

Traction is the total force that the tires of an automobile can provide/support. (In most cases it is used for acceleration/deceleration). It can also be referred to as the amount of longitudinal and/or lateral force the tires can withstanding before slipping/sliding. Traction is dependent on the friction between the tire and the ground surface (higher the friction, higher is the traction). Traction is also dependent on the normal load on the tires (higher the load, higher is the traction) It can be calculated as T=ǔ*N where ǔ is the coefficient and N is the normal force on a tire. The maximum torque the wheel can withstand without slipping is T*R where R is the radius of the tire[4]. The optimum amount of Traction is required under different dynamic conditions such as Sled Pull or overcoming a hill from less co-efficient of friction terrains like muddy swamps or water, [5] So here the author has discussed the importance of vehicle performances with respect to a different set of powertrains and validated through measured data

(Note that value of 'mǔ' drops once the tire starts rolling, hence traction is maximum at rest. It drops from the static friction coefficient value to the rolling friction coefficient value).

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C. Pre-Requisites Data:

Tire Dia = 22”
Value of static friction coefficient: - 0.75 [4]
Co-efficient Rolling Friction: - 0.03 [6]
Total Vehicle weight = 200 kg (With Driver)
Weight Distribution: - 65:35 (All the heavier parts such as Engine, Gearbox, Axle, CVT are mounted on rear part)
As shown in Fig.6, assuming maximum gradeability as per the dynamic event scenario = 30 degree
Weight in the rear wheel for traction = 65%
M=0.65(130)
= 130 Kg
Total Tractive force = Force required to overcome Static Friction + Force required to overcome rolling friction
F(t) = F1 + F2
F1 = (ŭ). (mg). cos30 + (mg). Sin30
= 1465.98 N or we can say that 1466 N
F2 = (ŭr). (u.mg. cos30) + (mg) Sin 30
= 52.26 N or 53 N
F(t) = 1466+53
= 1519 N
Traction force gives the torque required to be developed on the tire
The radius of Tire = 11”
Torque (T) = Ft. r
= 425 N.m

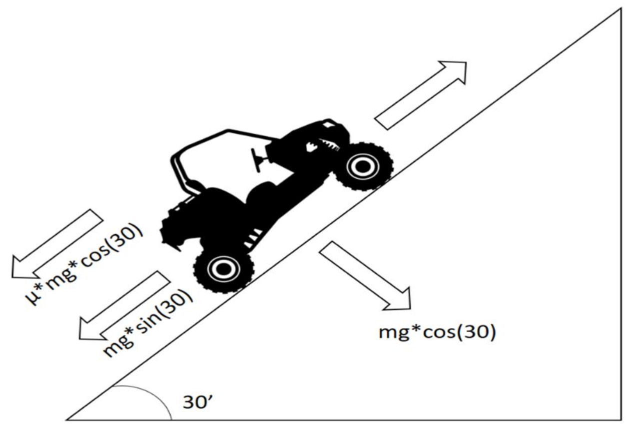

Figure 4 ATV illustrated with Forces

Max torque supplied by the engine is 19.85 N.m
So to overcome this situation we have introduced CVT [2],Author has prescribed the use of CVT its advantages over other driveline system and its performance characteristics, CVT has been installed in between Engine and Gearbox to supply required Torque on the wheels and get required reduction to get tires in rolling and attain desired performances.
CVT Ratio = 3: 0.43 [7]
Gear reduction Ratio (G) = 425/19.8*3
= 7.1582 or 8
On further computations we will witness the debunks of FoS is not achieved with Gear ratio 8, so we will take it with 9 to 9.33 and according to preferred numbers Gear tooth will be selected and for the best design and manufacturing point of view
Final RPM at Wheel = (3800/9) to (3800/9.33) = 423 RPM to 408 RPM

1) Material Used for Pinion & Gear:

Material	Density	Yield Strength	Ultimate Strength	Modulus of Elasticity	Poisson’s ratio
AISI 1038	7.845 g/cm^3	485 Mpa	570 Mpa	190-210 Gpa	0.27-0.30
AISI 5120	7.845 g/ cm^3	848 Mpa	1020 Mpa	190-210 Gpa	0.27-0.30
AISI 4340	7.845 g/ cm^3	470 Mpa	745 Mpa	190-210 Gpa	0.27-0.30

Table 5 Materials for Pinion & Gear

After assessing the materials in Table 3 and their properties post heat treatment, we finalized AISI (5120) for manufacturing of Gears and Shaft which had excellent Yield strength and High hardness (post heat treatment) and manufacturability has also been considered for the same.

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2) *Designing Gears by Buckingham's Method:* Force analysis on Spur gear is considered and first, we will find out the module by estimation method and validate it with required FoS and same we will do for the wear failure analysis also[8]. In a previous study author has done by different AGMA methods as well as Buckingham's method and we are going to follow conventional Buckingham's method for designing gears under wear and Bending

$i' = \frac{3800}{422} = 9$ [9]

$i = \sqrt{i'} = 3$

For 20° pressure angle full depth involute teeth

Zp = 18, Zg = (18). (3) = 54

But we will take Zg = 55 to avoid Gear Hunting [10], which was one of the reasons in previous gearbox failure to make calculation simpler, manufacturability easy and have enough FoS we will keep Z1=Z3 and Z2=Z4 and we know when we have same material for gear as well as pinion, pinion will be the weaker element in the assembly, So we will make pinion safe against different failures

Now calculating speed for each shaft

$\frac{N1}{N2} = \frac{T2}{T1} = \frac{3800}{N2} = \frac{55}{18} = N_2 = 1244\ RPM$

$\frac{N3}{N2} = \frac{T3}{T4} = \frac{18}{55} * 1244 = N_3 = 408$ RPM

$Sb = m.b.\sigma_b.Y = 1155m^2$

$V_p = \frac{\pi.dp.np}{60.1000} = 3.58m$

$Ft = {}^{P}/_{V} = 2234.63/m$

$Kv = {}^{5.6}/_{5.6 + \sqrt{vp}}$

$S_b = FoS*Feff$

$1155m^2 = 2 * \frac{2234.36}{m} * (5.6 + \sqrt{3.53m})$

Now by solving Trial and error method, we will narrow down our answer to the nearest standard module [11]

m = 3

d3' = m*Z3 = 54 mm

d4' = m*Z4 = 165 mm

b (face width) = 20 mm

Finding out the Beam Strength of Tooth

Sb=m*b*Ý*Y = 11088 N

Tangential Force due to Rated Torque

Mt = (60x10^6) x (Kw)/2*3.141*Nb
= 61428.22365 N/mm

Pt = 2*Mt/d3' = 2275.1 N

For Grade of 6 machining,

e = 8 + 0.63 ø

$ø = m + 0.25\sqrt{d3'}$

Ep = 0.004837117 mm

ø = m+0.25 Sqrt(d4')

Eg = 0.006211308 mm

E=Ep+Eg = 0.011048425 mm

C (Deformation factor) = 11,400 N/mm^2

$V = \frac{(\pi*d3'*n3)}{60*1000} = 1.15$ m/s

Dynamic Load (Pd)

$Pd = \frac{(21*v*(C*e*b+Pt))}{(21*v + \sqrt{(C*e*b+Pt)})} = 1240.194804$ N

Effective Load (Peff)

Peff = (Cs.*Pt+Pd)
= 4652.873895 N

Wear strength for the pinion

$Q = \frac{2zg}{Zg+Zp} = 1.506849315$

$Sw = b * Q * d'p * K$
= 18812.71 N

The factor of Safety Validation[11]

Against Bending

FoS=Sb/Peff. = 11088/4652.87 = 2.380 or 2.40

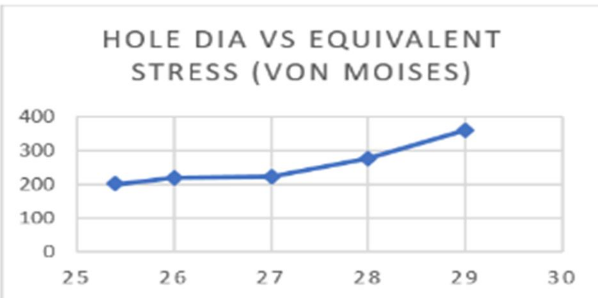
Against Wear

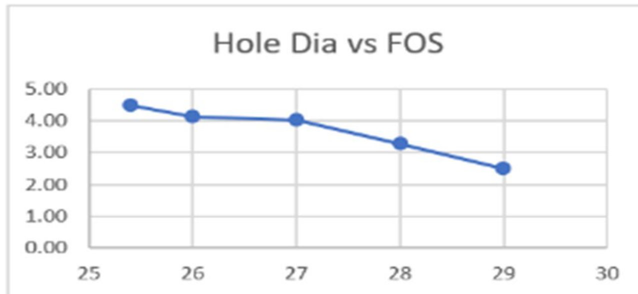
FoS= Sw/Peff. = 18812.71/4652.7 = 4.0423

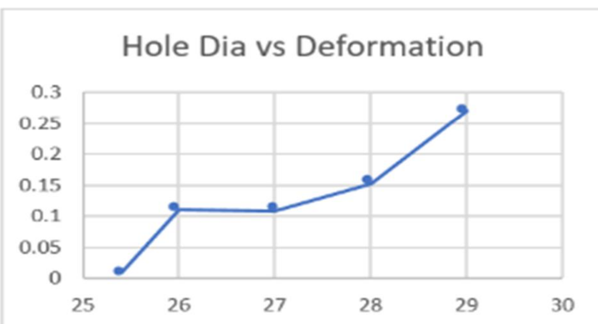
Above results obtained and analysed further we can say that investigating our gears against Bending and wear have been successfully been passed, as our required FoS was 2 and we are getting slightly more than required in bending as 2.40 and same for the wear as 4.04 and it is beneficial as sometimes in dynamic conditions we are uncertain in our case what road has to offer and forces may increase up to certain extent. Both the cases discussed are eliminating our previous failures and required performances are achieved within it.

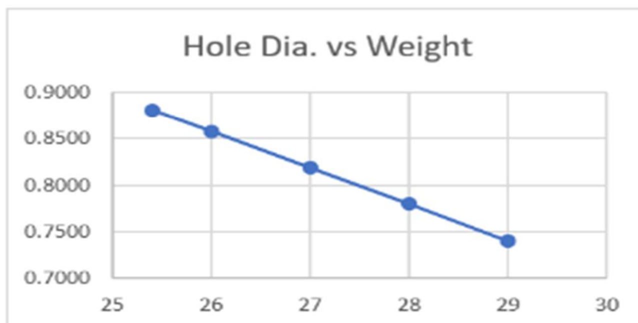
IV. 3D CAD MODELLING & FEA SOLVING

CAD Modelling for Spur gear was done in Solidworks 2017 sp x5, SolidWorks is a solid modelling computer-aided design (CAD) software and as per standards the pinion has been made and to reduce the weight (without compromising on FoS and performance), To reduce weight without compromising performance parameter we have drilled holes in the main body of Pinion and checked Equivalent stress, Deformation and FoS for the same.









As we can see from the graph's plotted above of various values, we could interpret some of the data like, As the Hole diameter increases, stresses on gear also increases and on the contrary FoS decreases, Weight also decreases, and deformation increases. Now as per our desired FoS we can say that Hole dia with 29 mm is optimum as we are getting enough FoS (2.52) and stresses are also under the limit (357.76 Mpa). The overall weight of the pinion has also been reduced from 880 grams to 740 grams, without compromising on FoS which an excellent thing is. So, we have successfully designed pinion with light weight and enough performance driving FoS.

Figure 5 Pinion CAD model

In the present study, FEA software ANSYS 18.2 Workbench has been used to determine Equivalent Stress (Von Moises), Total Deformation, Safety Margin, Stress Ratio and Factor of Safety. Pinion was designed in Solidworks 2017 (Spx5) and then imported through STEP file format and the necessary material properties are updated like Tensile yield strength, Ultimate Strength, Youngs Modulus, Poisson's Ratio, etc. The static structural analysis was done considering standstill position and sudden application of load as the worst condition scenario

The geometry has been meshed with fine sizing particles and 100% relevance centre. Giving the fine refinement for better results at the face of gear where the load is to apply [12]

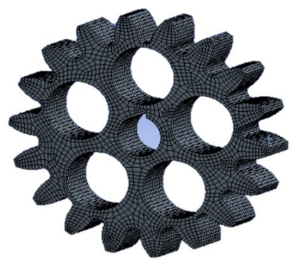


Figure 6 Pinion Mesh

Here a graph of FoS vs Equivalent Stress is obtained from exporting the data from ANSYS and plotting it against FoS. The maximum Equivalent Stress generated is 355 Mpa and corresponding 2.52 FoS is obtained which is nearly above from our desired FoS and on other hand weight reduction is also been done.

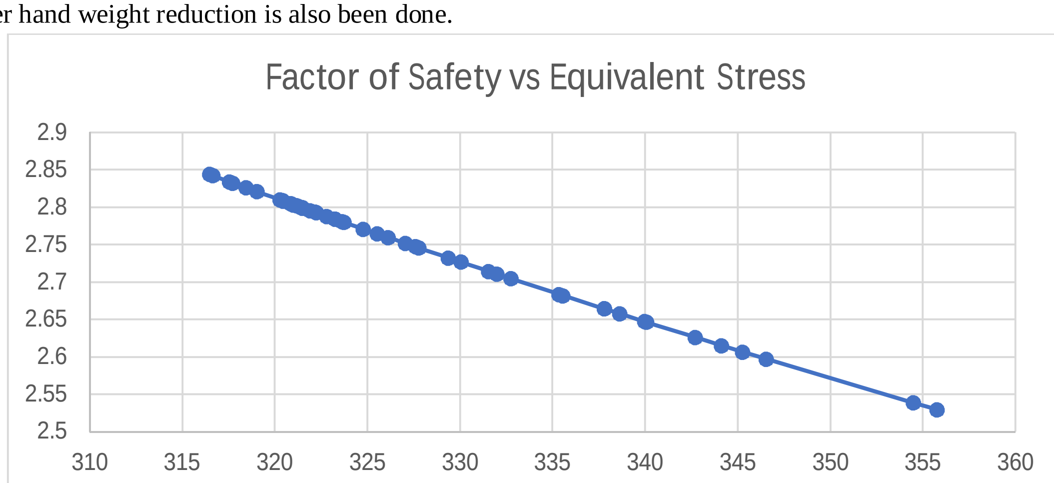
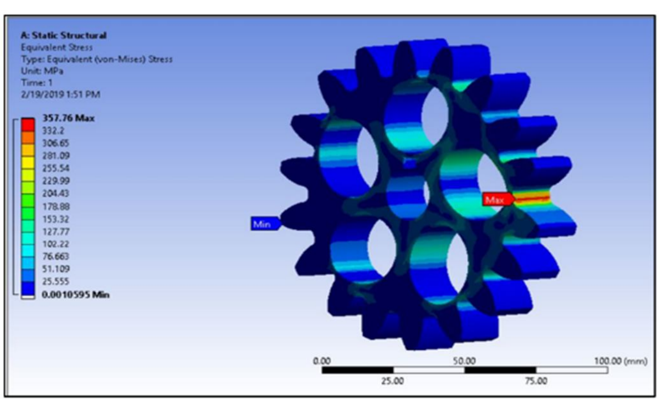
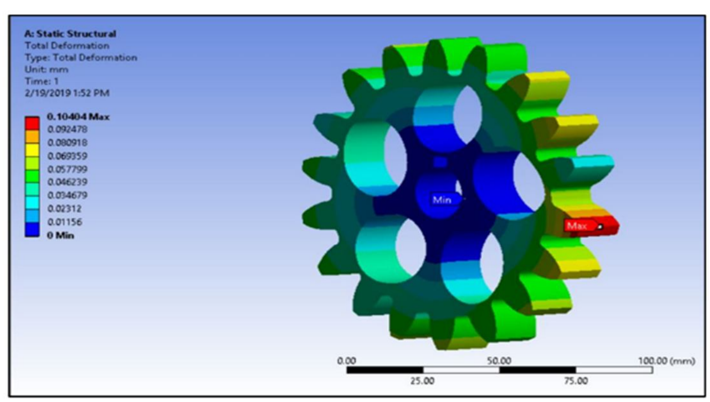


Figure 7 2018 Gear Analysis





Equivalent (Von-Moises) Stress:- 357.76 Mpa
FoS = 900/357.76 = 2.512

Total deformation = 0.104 mm

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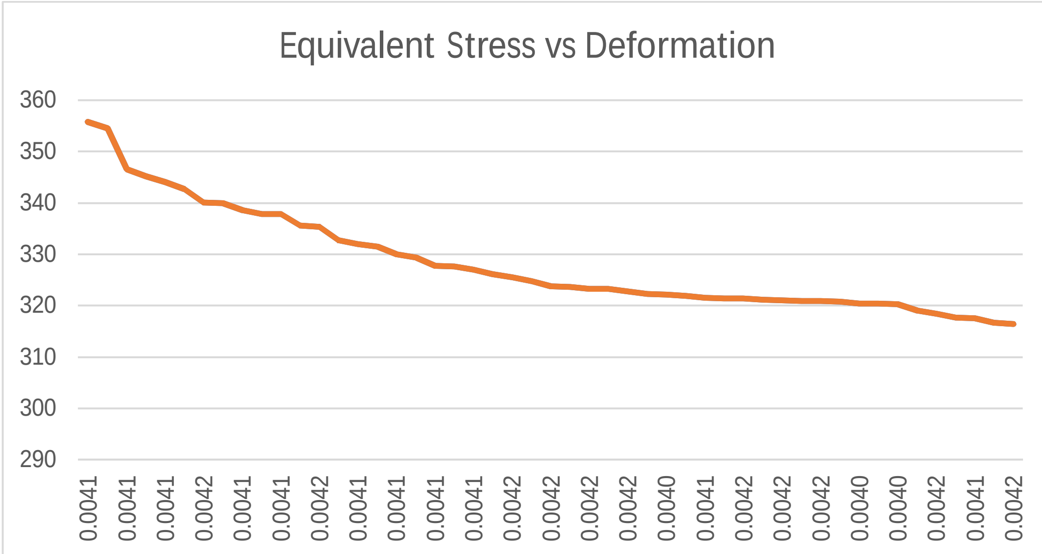

Graph of Equivalent Stress (Von-Moises vs Deformation) is obtained and is witnessed that graph decreases as the corresponding Deformation is decreased and examining at the values of deformation is very small so we can neglect or overshadow this parameter.

A. Validation through KissSys (2017)

KISSsoft is a Gear and Transmission system designing software used in many Automotive firms for Gearbox designing, manufacturing drawings and many more. Here to validate our data and other factors such as reliability, max equivalent stress on the shaft and corresponding deformation. Bearing selection and its reliability, service hours, all the loads and each and essential thing needed for the same.

Firstly, we will check our gear pair 1 & gear pair 2 with our manual calculations and FEA results. By all input parameters and other details, we will check the FoS and other data of gears. As our pinion is weak against Gear so we will check the design for same and setting up Calculation method with all the other factors such as Service factor, Dynamic factor, etc. [13]

			Gear 1	Gear 2
Normal angle	3 mm	Number of teeth (z)	18	55
Pressure angle at normal section	20	Face width (b) mm	20	20
Gear 1	Spur	Profile shift Co-efficient (X')	0.4090	-0.4090
Helix angle at reference circle	0	Quality (ISO 1328:1995) (Q)	6	6
Centre distance	109.5 mm			
Result with load spectrum (own input)				
Contact ratio	1.560		Gear 1	Gear 2
		Actual tip circle (mm)	62.454	168.546
		Root safety	3.014	2.838
		Flank Safety	1.392	1.392
		Safety against scuffing (integral temp)	3.926	
		Safety against scuffing (flash temp)	9.408	

			Gear 1	Gear 2
Normal angle	3 mm	Number of teeth (z)	18	55
Pressure angle at normal section	20	Face width (b) mm	20	20
Gear 1	Spur	Profile shift Co-efficient (X')	0.4090	-0.4090
Helix angle at reference circle	0	Quality (ISO 1328:1995) (Q)	6	6
Centre distance	109.5 mm			
Result with load spectrum (own input)				
Contact ratio	1.560		Gear 1	Gear 2
		Actual tip circle (mm)	62.222	168.778
		Root safety	2.237	2.173
		Flank Safety	1.086	1.156
		Safety against scuffing (integral temp)	3.472	
		Safety against scuffing (flash temp)	9.926	

Table 6 kisssoft data

After the iterative process running in KissSYS, it showed Root safety and flank safety for our all the gears out of which for our purpose pinion safety was more important which has come as follows: -

Pinion 1: - 3.014

Pinion 2: - 2.504

Minimum FoS for Pinion is 2.504 from KISsys out of all pinions which were still above from our desired results and hence we can say that our gear train design is safe in worst dynamic conditions offered: -

B. Comparing FoS of both the Software

Sr No.	Software	FoS	% Difference
1	ANSYS 18.2	2.512	0.319 %
2	KissSys	2.504	

Table 7 FOS Comparison (Software)

C. Comparing FoS of Softwares and Manual Calculation

Sr No.	Software	FoS	Manual Calculation	% Difference
1	ANSYS 18.2	2.512	2.40	4.66 %
2	KissSys	2.504	2.40	4.333 %

Table 8 Comparison (Software vs written calculation)

So, all the FoS Comparison has been carried out, having very less difference and we can say that our design is safe under all conditions and we have eliminated bending and wear failure from our previous design too.

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D. Other Dynamic Components (Bearings and shaft)

Bearing Selection is a crucial task, as all the three shafts will be rotating on its inner race's and cylindrical support will be given through shaft and bearings have certain life in hours and failure probability. So here we are assuming our gearbox life to be around 5-6 years, running at 6 hours a day. So, the required service life lies around 10,000 Hours minimum. So, by imperial calculations and iterations we finalized our bearing numbers and desired properties, as in previous bearings the required service life was not achieved and investigation showed that it could have failed in pre-matured stage, So by inputting all the dynamic parameters and road conditions uncertainties in our geartrain solver, we found that required service life of bearings was attained and results have been shared below [13]

Sr No.	Bearing Position	Minimum Attained service life (Hours)
1	Shaft 1	62714 h
2	Shaft 2	13587h
3	Shaft 3	23022h

Table 9 Bearings Service life

So, from maximum bearing life to be achieved out of our desired bearings, we finalized the bearings to be used on each shaft and shaft dia. & Strength was also checked with other parameters as Equivalent Stress, Deformation, and Max. Deflection when the required torque acts on the shaft

Shaft.	Bearing Name & Number	OD (Outer Dia)	ID (Inner Dia)	Number of Bearings
1	SKF 6304	52 mm	20 mm	2
2	SKF 6304	52 mm	20 mm	2
3	SKF 6304	52 mm	20 mm	2

Table 10 Bearing used for each shaft

1) *Shaft-1:* As our Bearings are fixed due to extended service life and more reliability needed, so we will check the same for other criteria whether dia is sustainable or not. So as per bearings ID, we will check the design parameters such as Maximum Equivalent Stress, Deformation and Maximum Deflection for 20mm Dia shaft, herein material used in all the shafts is the same which is used for the Gears (AISI 5120) [4]

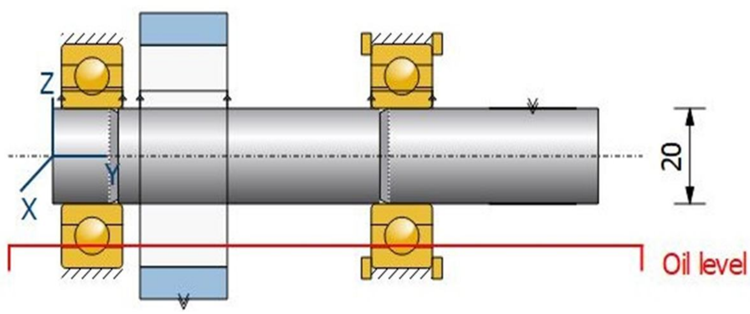

Figure 82 Shaft 1 with bearing in KissSYS

Sr No.	Parameters	Value
1	Maximum Equivalent Stress	26.58 Mpa
2	Maximum Deflection	12.94 ŭm (Micron)
3	Minimum Bearing Service Life	62714.68 H

Table 11 Shaft 1 Parameters

2) Shaft-2:

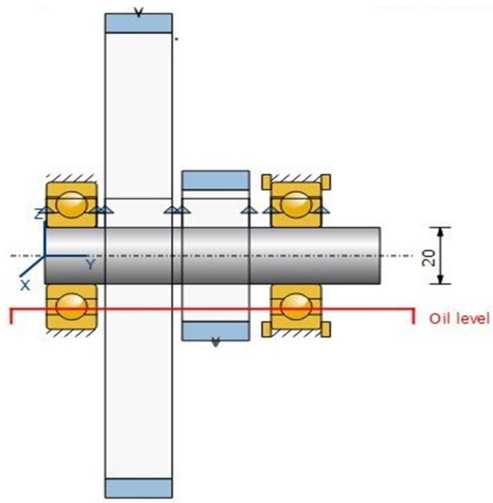

Figure 9 Shaft 2 with bearing in KissSYS

Sr No.	Parameters	Value
1	Maximum Equivalent Stress	81.58 Mpa
2	Maximum Deflection	10.35 ŭm (Micron)
3	Minimum Bearing Service Life	13587.68 H

Table 12 Shaft 2 Parameters

3) Shaft-3

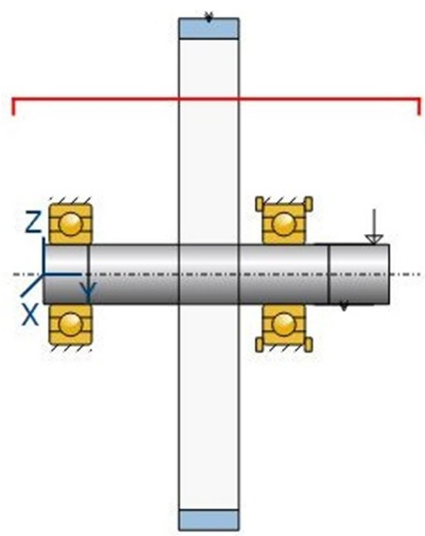

Figure 10 Shaft 2 with bearing in KissSYS

Sr No.	Parameters	Value
1	Maximum Equivalent Stress	210.77 Mpa
2	Maximum Deflection	30.52 ŭm (Micron)
3	Minimum Bearing Service Life	23022.06 H

Table 13 Shaft 3 Parameters

Analysing all the above results and examining parameters like equivalent stress and deformation we have decided to utilize shaft material same as the gear material (AISI 5120) for single piece manufacturing, less additional connections like a sleeve and other couplings due to dissimialr materails. Because gear material is having excellent mechanical properties Therefore FoS is prevailing quite high so there is now need of optimizing dia. because bearing ID needs to be maintained uniform as shafts OD.

V. RELIABILITY CURVES

Reliability of Bearings, Shaft, System and system failure data is very important whether to check the lifetime of the product and failure % chances of what we have decided or what is maximum life we want. As we will be considering the reliability of Bearings, Gear and whole system failure probability as we can say that all dynamic parts have more chances to fail rather than any static part in terms of hours and percentage, following graphs have been plotted with help of KISSsys as user input function to check life at designated hours and in terms of percentage.

A. Bearing Life

As interpretnded from the graph, Life of the bearings (cumulative) is obtained, a minimum of 92% at 10,000 Hrs. So we can consider is it safe and permissible within range and won't fail before our desired lifetime as minimum criteria followed by leading bearing manufacturers lies around 90% [14]

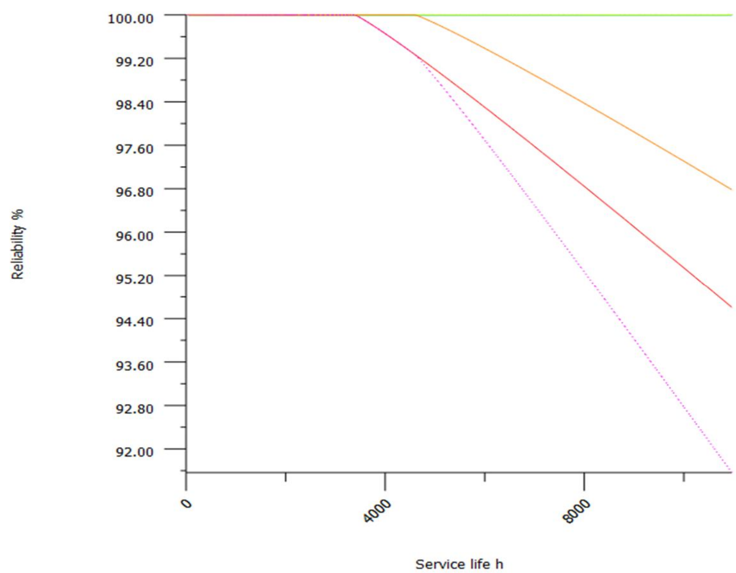

Figure 11 Reliability Vs Service life for Bearings

B. Gears Reliability

As we have designed our gears with enough FoS and material has been chosen wisely with high strength, gears have pretty much longer lifeline then entire dynamic parts. So here it is clearly seen from the graph that 100% reliability is obtained at 10,000 Hrs.

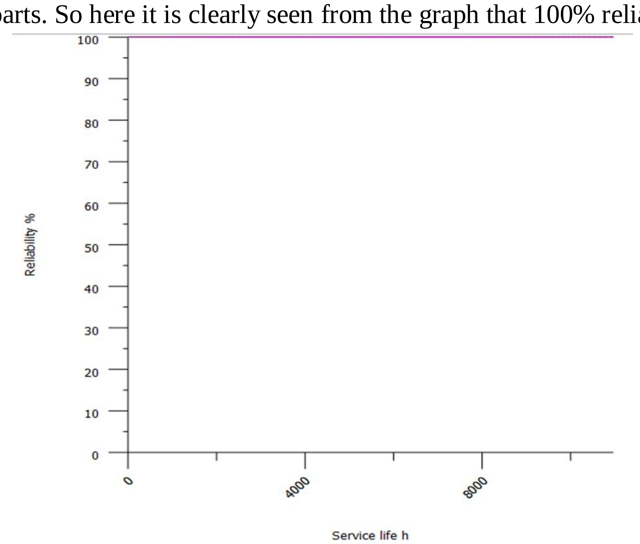

Figure 12 Reliability Vs Service life for Gear

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C. System Failure Graph

Here all the dynamic conditions and other scenarios are combined and the graph is plotted showing the entire system failure graph, so its failure probablity lies around 8% at maximum, which can be compensated and relied on.

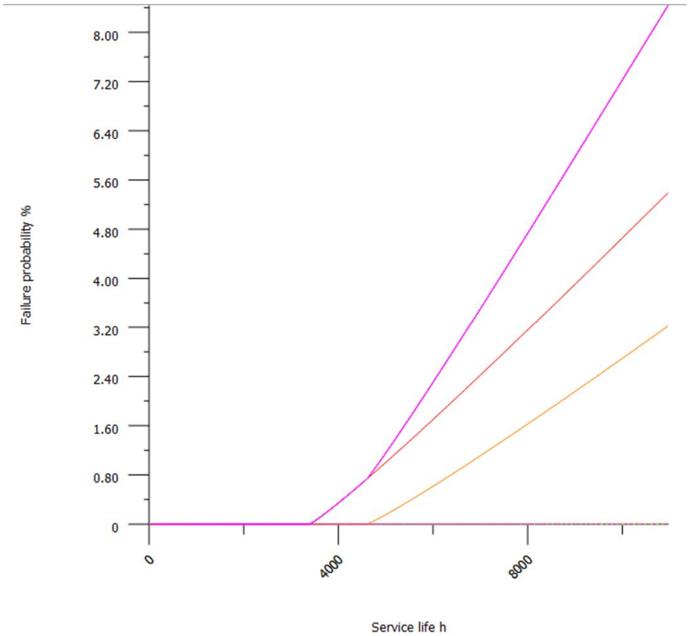

Figure 13 Reliability Vs Service life for System

D. System Reliability

At 10,000 Hrs we are obtaining 91% reliability of the entire system, which is falling under the required range and Hence we can say that our majority of the system is reliable enough to be trusted on under dynamic as well as stsic conditions

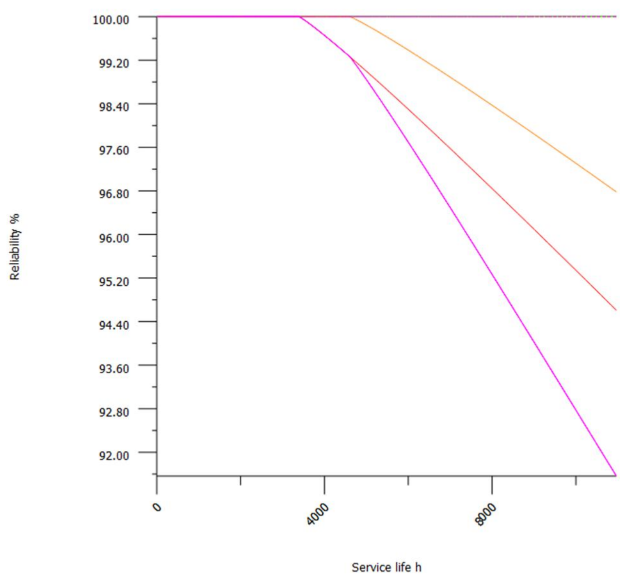

Figure 14 Reliability Vs Service life for System reliability

As from the above graphical data we can interpret that the most less reliability out of all components was achieved in bearings that are 92 %. That is more than required and hence we can say that our overall design is safe and overall reduction and RPM achieved was shown below considering all the oil losses and everything without compromising on FoS and compactness Final gearbox after designing as per the new parameters it would look like as shown in the figure below.

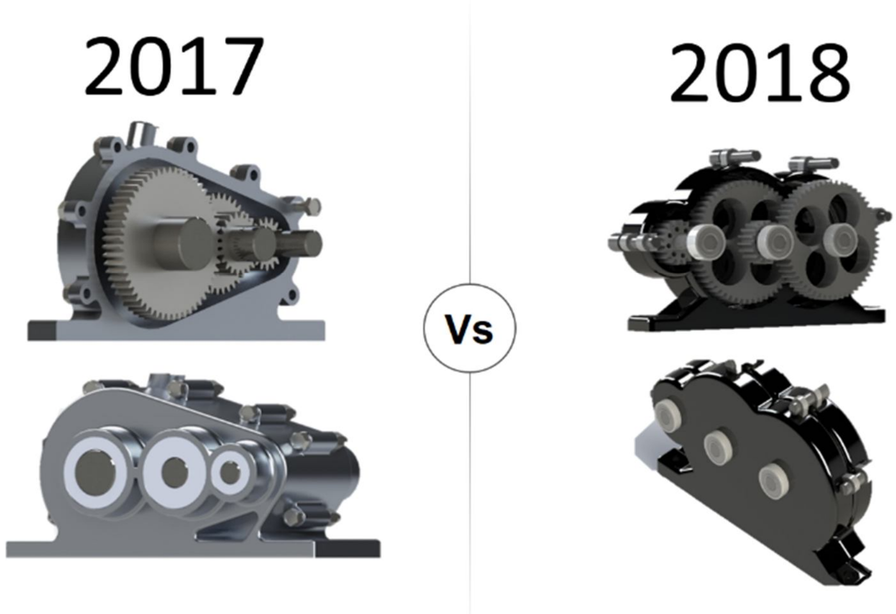

Figure 15 2017 Gearbox vs 2018 Gearbox

VI. RESULT ANALYSIS

A. *Previous year Gearbox*

Sr No.	Parameters	Qty	Value
1	Gears	(2 pinion + 2 gears)	2.85 kg
2	Shaft	(All 3)	0.650 kg
3	Bearing	(Gross 6)	0.5 kg
4	Casing	(Both sides)	3.68 kg
5	Bolting	4	0.560 kg
	Total		8.4 kg

Table 14 Previous year Gearbox Weight

B. *Present year Gearbox*

Sr No.	Part name	Qty	Gross weight
1	Casing	2 (Left and Right)	1.3 kg
2	Bolting	4	0.400 kg
3	Gears	4 (2pinion and 2 gear)	3.7 kg
4	Shaft	3	0.800 kg
5	Bearing	6 (SKF 6202)	0.6 kg
		Total weight	6.8 kg

Table 15 Present year Gearbox Weight

Sr No	2017	2018
1	8.4	6.8
Reduction of weight	19.04 %	

Table 16 Weight Reduction

Sr No.	Components	2017	2018
1	Gears	1st Pinion = 18teeth 1St gear = 56 teeth 2nd Pinion = 18 teeth 2nd Gear = 70 teeth	1st Pinion = 2nd Pinion = 18 Teeth 2nd Gear = 4th Gear = 55 Teeth
2	Face width	12 mm	20 mm
3	Gear Material	EN 8	AISI 5120
4	Hardness	300 BHN	340 BHN
5	Minimum FoS	0.796	2.501
6	Nominal Shaft Dia	15mm	20mm
7	Bearing	SKF 6202	SKF 6304
8	Casing material	AISI 6061-T6	AISI 7075-T6
9	Gearbox System Reliability (10,000 Hours)	15%	92%
10	Gears Reliability	100% (800 Hours)	100% (10,000 Hours)

Table 17 2017 vs 2018 Gearbox

VII. CONCLUSION

This paper unveils the more sophisticated methodology of designing the gearbox gear train, Bearings and shafts designing using the modern designing software's. By defining realistic driving conditions have been entered as an input to the software solver. And as a result, the designer can achieve more accurate results of strength, equivalent stress, deformation, safety factors, and other such parameters and validate the written calculations with Software acquired` results. Moreover, weight reduction and performance derived results can be achieved without contrary on Factor of Safety, following this methodology one can easily design and validate two stage reducers according to the required gear ratio and other important facts and failure can be prevented and required service life can be attained with good reliability.

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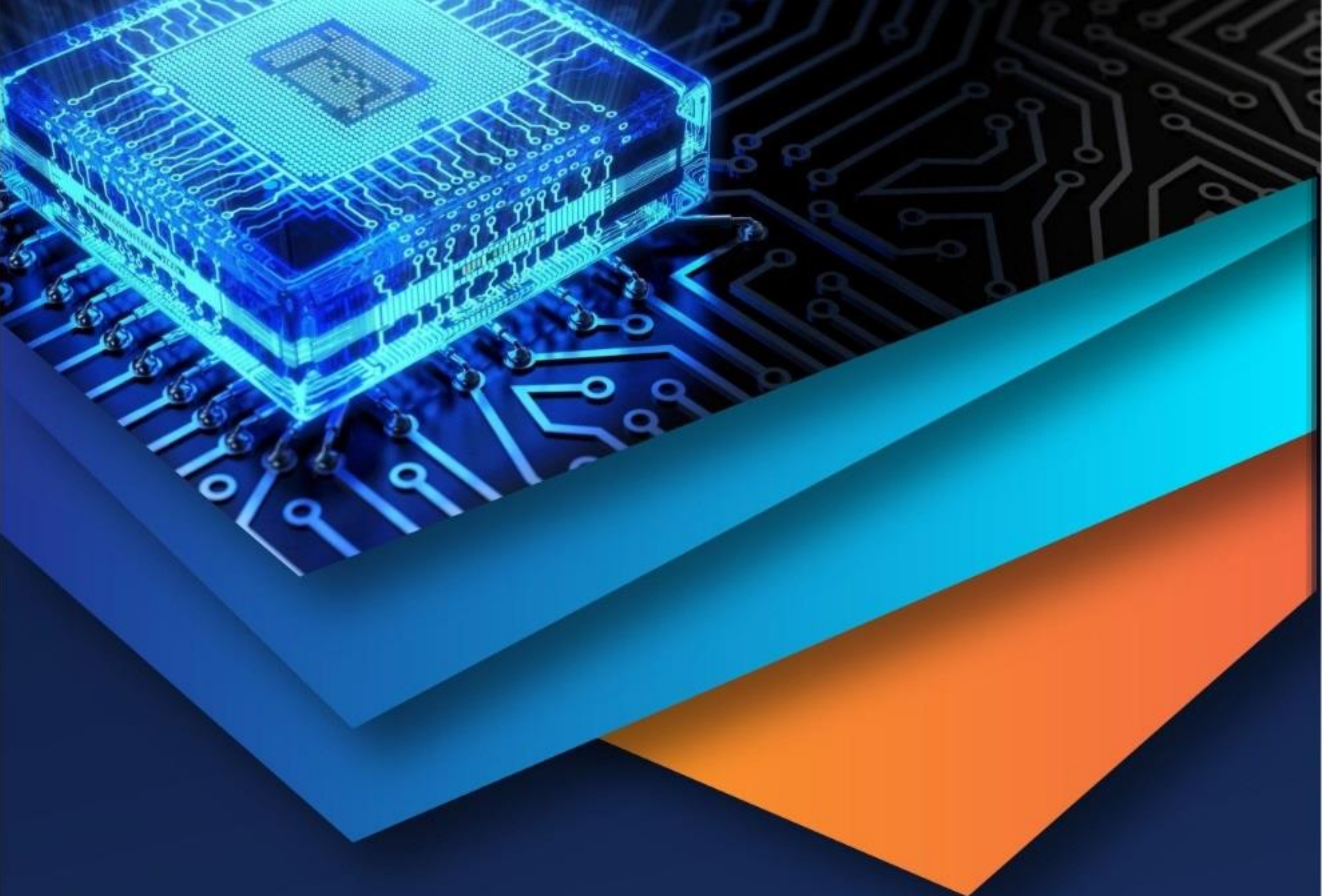

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