Design of Efficient and Optimised Drivetrain Assembly

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Abstract - *In the automobiles industry, it is essential to design drive train with more fuel efficiency, fewer losses and reliability, and compact design. This article deals with the design of efficient and weight-optimised drivetrain design for Formula Student vehicle considering various parameters affecting it.*

Key Words: –

Drive Train, Efficiency, Optimisation, Final Drive Ratio, Sprocket, Differential, Differential, Brackets, shaft, Turn Buckle.

1. INTRODUCTION

What is Drivetrain?

The drivetrain system transmits power from the engine to wheels and all necessary components between them. The different types of power train are;

1. Chain Drive - Power transmitted via chain from driving sprocket to driven sprocket. The chain drive is the cheapest type of drive train. Eventually, it requires time to time maintenance. There is less power loss as compared to belt drive.

2. Belt Drive - Belt is used to transmit power from driving pulley to driven pulley. A belt drive is smooth with fewer jerks as compared to chain drive. The belt drive is expensive and has considerable power loss.

3. Shaft Drive - The shaft that's connected to the gearbox output via a universal joint, which is essentially a coupling that facilitates the transmission of rotary power at any selected angle. It is the most expensive type of drive train with negligible power loss.

Considering cost, power losses & ease to assemble chain-driven drivetrain was analyzed.

1.1The components of the chain-driven drivetrain are:

1. Driving Sprocket - Located at the output shaft of the engine. Used to transmit power from engine to driven sprocket.

2. Driven Sprocket - Connected with the driving sprocket via a chain. Used to transmit power to the shaft.

3. Differential - Used to transmit power, and it can rotate wheels at different speeds to prevent understeer, oversteer, and slipping at corners.

4. Differential Carrier Brackets - These are mounts for holding differential and shafts.

5. Half Shafts: - Half shafts are power transmission shafts used to transmit power from the differential to the wheels.

2. OBJECTIVE

1. To find the final drive ratio for maximum fuel efficiency
2. Reduce maximum possible mass from the drivetrain assembly to reduce inertia losses and enhance the performance.

3. SELECTION OF FINAL DRIVE RATIO

What is Final Drive Ratio?

In chain driven system the final drive ratio is the ratio of teeth on driven sprocket to the driving sprocket.

Vehicle dynamics simulation carried on **Optimum Lap** software gave the final drive ratio. It is a lap time estimation tool that utilizes a basic quasi-SteadyState vehicle model. (The vehicle is allowed to accelerate in a corner until it reaches its maximum allowed corner speed. Similarly, the vehicle is allowed to decelerate in a corner as well).

Optimum Lap rapidly analyze the characteristics of a vehicle on a given track. The input parameters are selected by the engine/transmission specification and the set of wheels used in the car.

The engine's behavior determines the input parameters for the powertrain under the different rotational speeds, the torque values of torque are selected by torque vs. rpm graph of KTM Duke 390 engine with stock ECU.

3.1 Simulation Result

For simulation result we selected 3D batch chart where we can find optimum lap time and fuel efficiency at different final drive ratio and longitudinal coefficient of friction as the coefficient of static friction depends on tread design and compound, tire construction, inflation pressure, road surface, tire load, and temperature.

As Fuel consumption is related to combined effect of transmission ratios and final drive ratio but we have 6-speed transmission of KTM we have to find most efficient final drive ratio over track by using optimum lap.

By graphs provided Figure 1 Lap time vs Final drive ratio& Figure 2 by the optimum lap of fuel consumption and lap speed and comparing results of other parameters like lap speed acceleration we selected **3.42** as final drive ratio.

Figure 1 Lap time vs Final drive ratio

4. 4. SELECTION OF POWER TRANSMITTING CHAIN

Gears, chains, or belts can transmit the power of the engine to the shafts. Yet, the chain is more suitable than the belt system by considering the variation in speed and load. The gear assembly is not as economical as the engine and shaft assembly are not too close to each other; therefore, we selected a roller chain drive system. Chain tension is continuously acting on the chain, which leads to the formation of fatigue on-chain; this causes the chain breakage. Considering factors in Table 1, we selected the

KTM 520X1R chain which has **15.875** mm pitch and breaking load nearly **36.3KN.**

Table 1- Dimensions and the breaking load of the roller chain

Chain	Roller Diameter	Pin Length	Plate Thickness		Weight per 100 links	Avg. Tensile Strength
			Inner	Outer		
	mm	mm	mm	mm	kg	kgf (kN)
520 Z3	10.32	21.40	2.2	2.4	1.87	4240 (41.6)
525 Z3	10.32	23.70	2.4	2.6	2.10	4690 (46.0)
530 Z3	10.32	25.30	2.4	2.6	2.19	4690 (46.0)
428 X1R	8.50	20.80	2.0	2.0	1.12	2800 (27.5)
520 X1R	10.22	20.30	2.0	2.0	1.55	3700 (36.3)
520 X1R₂	10.22	20.30	2.0	2.0	1.67	3900 (38.2)
520 X1R₃	10.22	20.30	2.2	2.0	1.73	4050 (39.7)
525 X1R	10.22	22.60	2.2	2.2	1.87	4050 (39.7)
525 X1R₃	10.22	22.60	2.4	2.2	2.03	4200 (41.2)
530 X1R	10.22	24.90	2.4	2.4	2.08	4200 (41.2)
428 HPO	8.50	20.00	1.7	1.7	1.11	2250 (22.1)
520 HDS	10.22	18.60	2.2	2.2	1.55	4020 (39.4)
520 HDR	10.22	17.80	2.0	2.0	1.47	3700 (36.3)
420 HDR	7.77	15.05	1.6	1.6	0.74	2000 (19.6)
428 HDR	8.50	18.30	2.0	2.0	0.97	2500 (24.5)

5. SPROCKET DIAMETER CALCULATION

The work of the sprocket is to transfer the torque of the engine to both half shafts connected to wheels. Sprocket is driven by pinion with the help of a chain wound on the sprocket.

The formula for finding the diameter of the sprocket is

D = P x cosec (180/t2)

Where P is chain pitch which is wound on sprocket and t is the number of teeth on the sprocket.

The formula for finding the number of teeth of the sprocket is

N1 / N2 = t2 / t1

Where N1 / N2 is the final drive ratio and t1 is the number of teeth on pinion and t2 is the number of t1 is the number of teeth on the sprocket.

We know the value of the final drive ratio (**3.42**) and the number of teeth on the pinion i.e. The pinion sprocket

(driving sprocket) which has the number of teeth is **14.**

Therefore, we get the number of teeth on driven sprocket **t2 = 48**

We also know the value of chain pitch which is **P= 15.875** mm hence we can obtain the value of the diameter of the sprocket is **242 mm**.
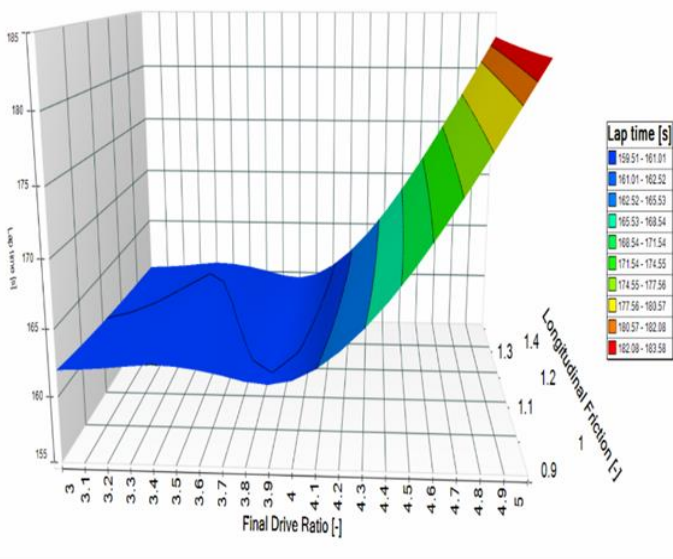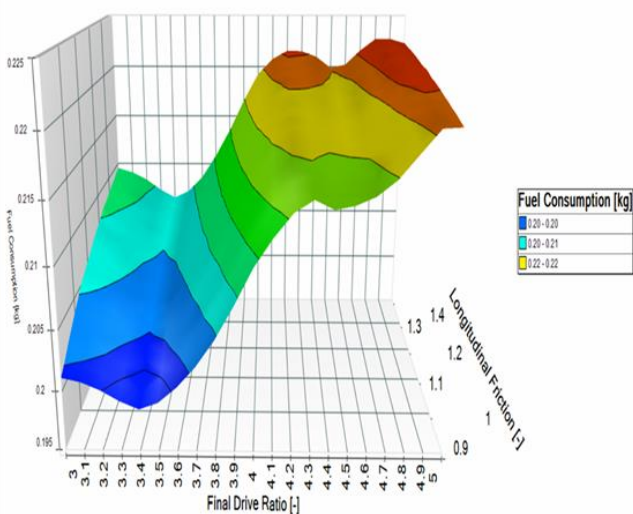
Figure 2-final drive ratio vs Fuel consumption

6. Maximum chain tension=maximum torque on sprocket / radius of sprocket

Maximum torque on sprocket=Maximum Engine Torque *Combined Gear ratio*Final drive ratio

=34*7.07*3.42

=822 Nm

Therefor maximum chain tension=

822/0.121

=6.7 KN

8. Sprocket Teeth Back Tension

The back tension generated due to wrapping the chain around the sprocket is responsible for the deformation of the sprocket.

Tk = T0 x (sin (θ)/sin (θ+2 β)) ^k-1

Where,

Tk = Tooth back tension at teeth k

T0 = Chain tension = 6.7 x 10 ^3 N

N = Number of teeth = 48

2 β = 360/N = 7.5

Tooth Number (k)	Back Tension (Tk)	Force on tooth = (Tk -T(k+1))	X– com (N)	Z– com (N)
1	6700	2200	1970	- 960
2	4500	1410	1327	- 444
3	3090	990	980	- 190
4	2100	670	668	40
5	1430	456	454	31
6	974	307	300	61
7	667	197	186	64

Table 2 Forces on respective tooth

K = Number of engaged teeth = ((wrap angle x N)/360) We calculate the value of Back Tension up to 7 teeth, as tension becomes very less after 7TH teeth.

F=Tk-T(k+1) N

The chain is making **-19.86⁰** angle from the negative X-axis due to this back tension is getting split into X and Z components and a number of the tooth increases the chain angle is also increased by angle spacing between each tooth around the chain which is 2 β.

7. DESIGN OF SPROCKET

According to the selected final drive ratio (i.e., 3.4), the number of teeth of the driving sprocket is 14, and similarly, the driven sprocket has 48 teeth. The thickness of the sprocket is 6.35mm, about the 520 chains used. The sprocket has been designed on CATIA, considering the holes on the differential for mounting. After designing, the sprocket analyzed on Ansys concerning the back tensions calculated for each tooth gave maximum stress concentration regions, where removal of a considerable amount of material for weight reduction was possible. EN8 (AISI 1040) medium carbon steel material was finalized for the sprocket considering its mechanical properties.

Table 3-Mechanical properties of EN8 (unalloyed medium carbon steel)

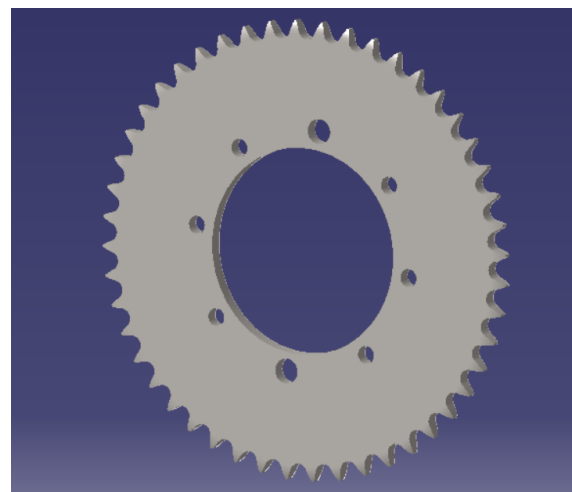| Max Stress	700-775 N/mm²
Yield stress	385 N/mm²
Max Elongation	16 % min
Hardness	201-255 Brinell

Figure 3-Design of sprocket without weight reduction

The initial weight of the sprocket was 2370grams. The material is removed from the low-stress concentration areas after sprocket weight is 1106 grams; the FEA of sprocket (without weight reduction) *Figure 5.* We successfully reduced 1264grams of weight.

Sprocket is flame hardened(quenched) for better tensile strength

7. ANALYSIS OF SPROCKET -

Analysis type - Static Structural

Material - EN8 (unalloyed medium carbon steel)

Meshing - *Multizone – quad/tri* Meshing

1. Fine relevance Centre
2. High Smoothing
3. Fine Span center
4. Element size - 4mm

Boundary Conditions -

The sprocket has fixed support at the mountings of the differential. And the components (X &Z component) of force (back tension) were applied on the first 7 teeth.
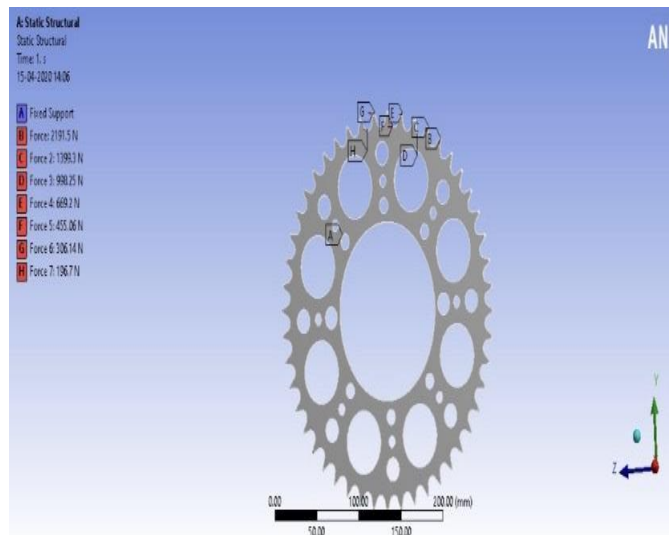
Figure 4- Boundary Conditions

Result: -

1.Max Von mises stress =129.02 Mpa (with weight reduction)

2.Factor of safety=3.68

Since the yield strength of unalloyed medium carbon steel (EN8) (385MPa) is greater than the maximum von misses stress (129.02 Mpa) the design is confirmed as safe.
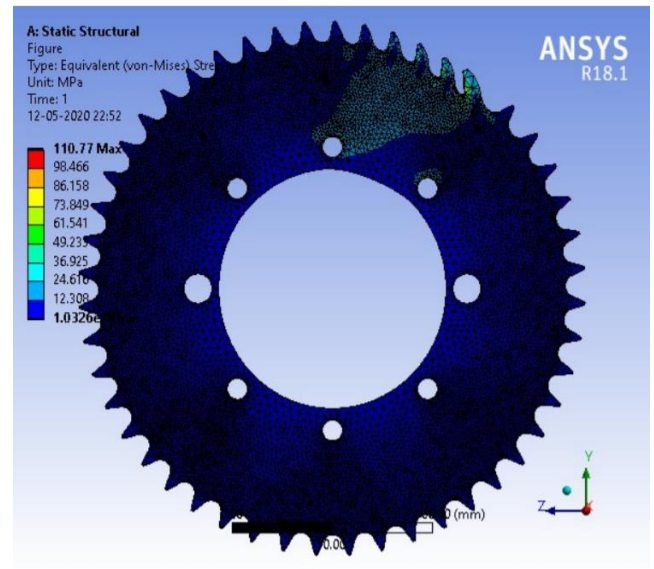

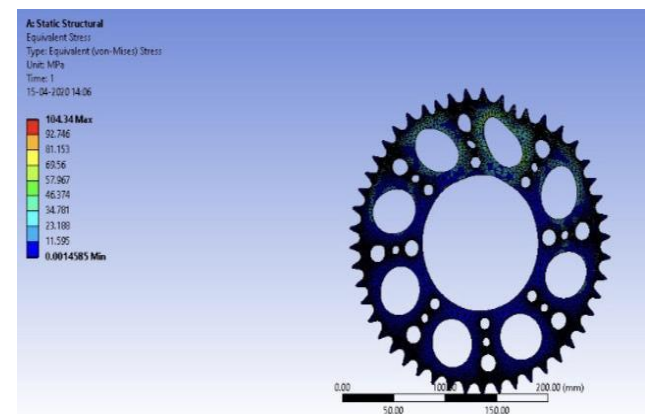*Figure 5- Max Von mises stress (without weight reduction)*

Figure 6- Max Von mises stress (with weight reduction)

8. THE DIAMETER OF HALF SHAFT CALCULATIONS

The shaft transmits the rotational motion from driven sprocket to the wheels of the vehicle. For transmitting the torque from the engine shaft. It must sustain under peak stress concentration and maximum bending moment due to weight; therefore, the diameter of half shaft is calculated by distortion energy theory with combined bending and twisting moment. FS (Factor of safety) is taken as 2 considering various loading factors.

We know,

$$d_o^3 = \frac{FS}{\sigma_y} \frac{32}{\pi} B \left\{ \left[M + \frac{F d_o}{8} (1 + \alpha^2) \right]^2 + \frac{3}{4} T^2 \right\}^{1/2}$$

Where **M** is maximum bending moment and **T** is maximum twisting moment. For axle material we selected the EN24 which has

M=352 Nm,

T=411 Nm

σ**y** (yield strength) ₌940 MPa

We get diameter of shaft d = 22mm

9. ANALYSIS OF SHAFT

Table 4 Material properties of EN24(AISI 4340)

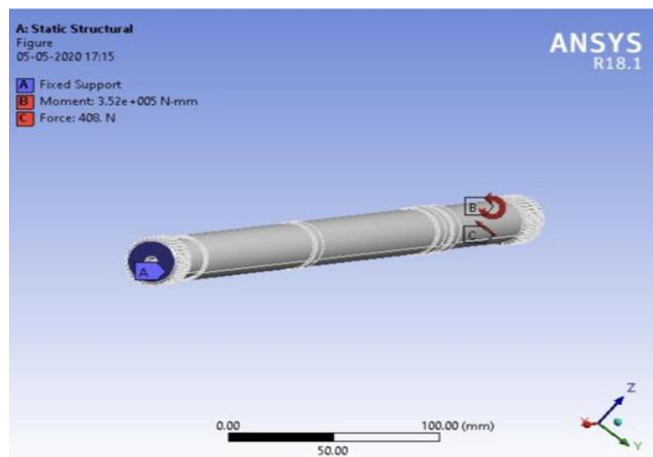| Max Stress	1225-1375 N/mm2
Yield Stress	1095 N/mm2
Max Elongation	10%
Hardness	363-429 Brinell
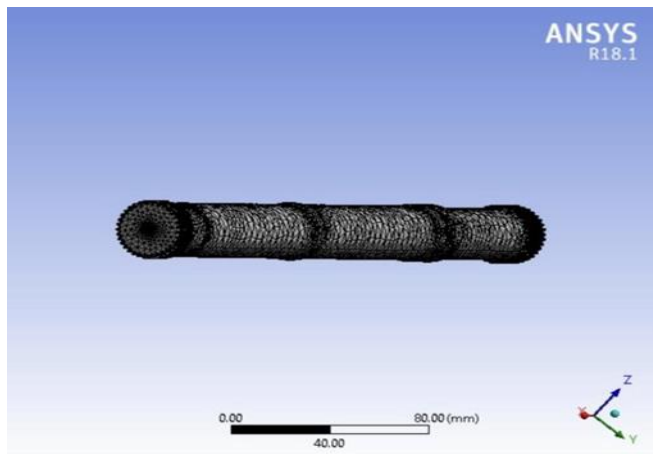
Figure 7 - Meshing

Figure 8 Boundary Condition

Meshing - Multi zone Quad/Tri Meshing type with

1. Fine Relevance Centre
2. High Smoothing
3. Fine Span Centre
4. Element Size - 4mm

Boundary Condition -

One end of the shaft assigned with fixed support and other end assigned with torque of 411Nm and bending moment of 352Nm.

The finite element analysis of the shaft done on Ansys workbench considering the bending moment and twisting moment for the shaft as boundary condition.
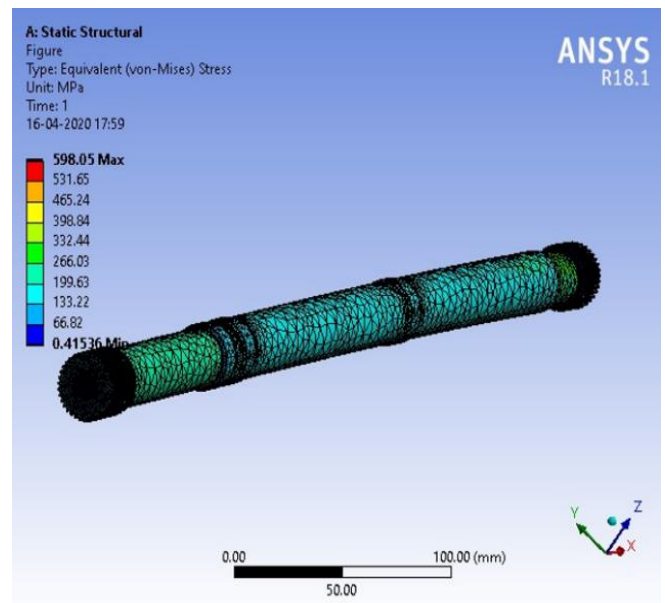

Figure 9- Analysis of shaft

Result of analysis-

The maximum von mises stress generated is 598 MPA and we got factor of safety nearly 1.8 by FEA analysis.

10.SELECTION OF DIFFERENTIAL – To select differential, we considered factors like reaction of differential under change in traction In the market there are 4 types of Differentials are available;

1. Open Differential

2. Limited Slip Differential (LSD)

3. Locking Differential(spool)

Open differential is cheap in price and easy to maintain but they tend to oversteer in rear wheel drive vehicle at high speed.

The Spool is solid axel it is locking differential it works as solid shaft which connect both wheels spools are very useful in straight acceleration as it divide torque equally bur it has disadvantages as tire wear due to locking effect and considering event like autocross, skid pad which requires vigorous turning it is not beneficial

The limited slip differential is made to overcome the issues of open differential and locking differential in a corner LSD would neither provide a full lock or full open situation instead it biases the power to wheel who has more traction.
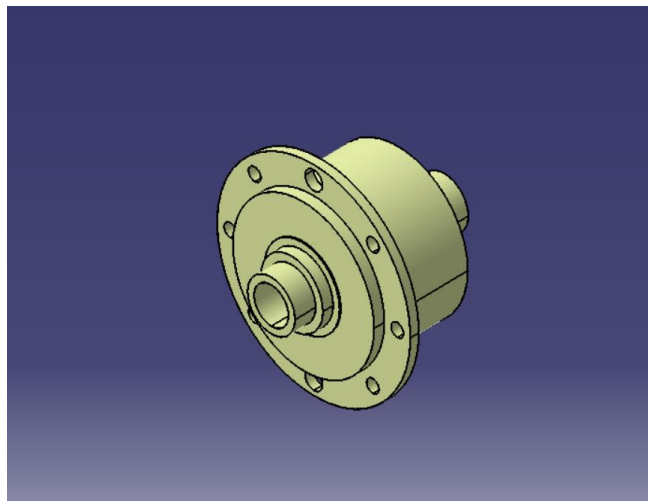

Figure 10-: Chain-Drive Sealed Quaife ATB Helical LSD Differential

By comparing the advantages & disadvantages of all types of Differentials we choose the Limited Slip Differential Torsion type provided by RT Quaife Engineering Ltd.

11. SELECTION OF BEARING –

Bearing is used to transfer the motion or torque, coming from differential. For the selection of bearing, we consider parameters like type of loading, Rotational Speed, and Rigidity. There are two types of the load's act on bearing Radial load and Thrust load. Thrust load acts parallel to the axis of rotation of bearing, and Radial load acts perpendicular to the axis of rotation of bearing. Rotational speed depends on engine speed at different rpm, so the rotational speed is also different at different rpm. The bearing has rigid enough to sustain the Thrust and Radial loads at different rotational speeds. Selection of bearing also done by choosing bearing which is suitable for differential which has dimensions as per the Differential dimensions.
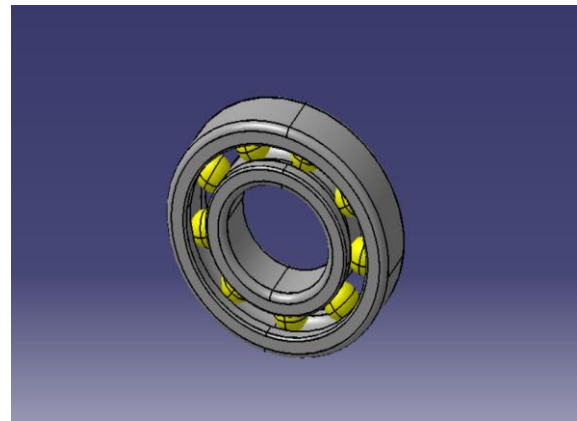

Figure 11- RLS 12 Deep groove ball bearing

On considering the parameters of size and load capacity we choose the RLS-12 Deep Groove Bearing which has high rigidity to sustain the Thrust and Radial loads at different RPMs and has the dimensions as per our Differential. We use the circlip on one side of the bearing and another end of the bearing closed this will restrict the bearing and keep it at one place.

12. SHEAR FORCE DIAGRAM FOR DIFFERENTIAL BRACKETS

Sprocket is fixed between two differential brackets and on sprocket chain wounded due to this chain tension is generated. This chain tension act between two brackets this chain tension produces share force on brackets, each bracket will be under different amount of force due to bending moment produced by chain tension.

Calculation of Shear Force

From Right Bearing to sprocket Distance is 13.57 mm and from Left Bearing to Sprocket Distance is 89.15 mm.
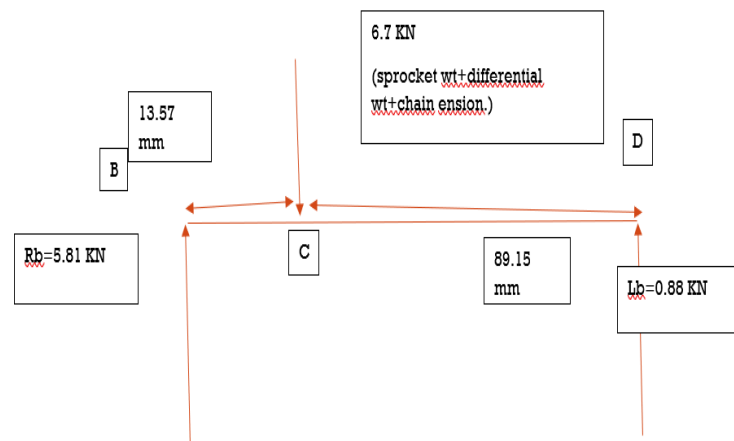

Figure 12- FBD of differential assembly

Shear Force at B = 5.81 KN

Shear Force at c = 5.81 KN – 6.7 KN = -0.88 KN
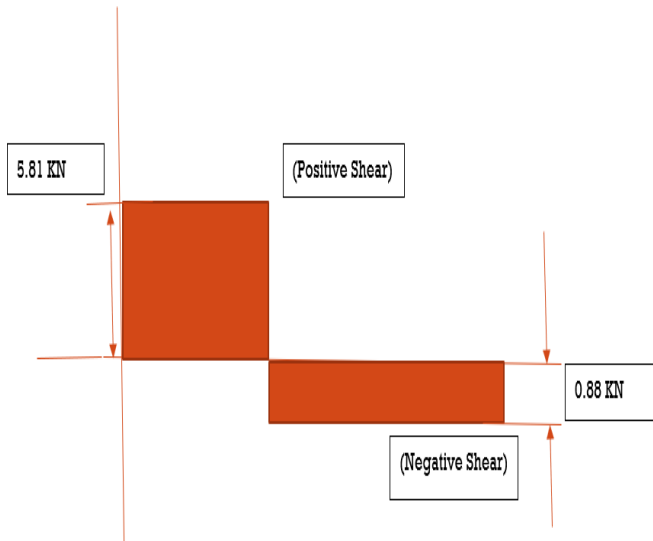
Shear Force at D = -0.88 KN

Figure 13- Forces acting on bearing

13. SELECTION OF CHAIN TENSIONER

 Chain Tensioners are of following types -

1. Rigid Differential Mount - In rigid differential mounts there are no moving components used to tension chain.

2. Spring Tensioner - This uses a spring-loaded idler gear to fix the slack in the chain.

3. Indexable Pivot Tensioner (Turnbuckle) - This mechanism works to tension the chain by pivoting the differential mounting above the axis using the turnbuckle.

4. Eccentric Chain Tensioner - The mechanism has symmetrical plates with an outer frame and eccentric plates.

From the above types of chain tensioners, we have selected an indexable pivot tensioner as it is easy to manufacture, low-priced and easy to assemble. The turnbuckle was designed with one end consisting of a ball joint and other end consisting of clevis joint for double shearing on differential side.
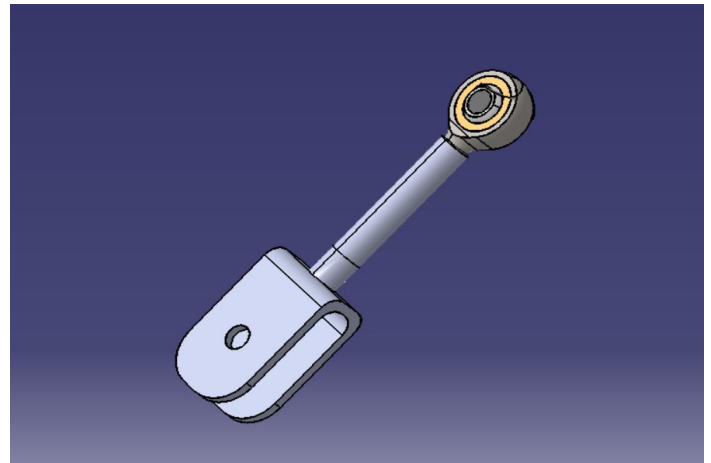

Figure 14- Turnbuckle

14. DESIGN OF CHAIN TENSIONER-

We have to design a chain tensioner which can sustain the maximum tension developed by drivetrain assembly which is 3.35 KN (chain tension).

Consideration of factor of safety 4 is done due to fluctuating tension due to engine assembly and dynamic braking on race track.

14.1. Design of rod for turnbuckle

The factor of safety of rod is find by principle of maximum stress theory and targeted FOS is 4 as rod is subjected under fluctuating tension, each rod in turnbuckle is design by considering tensile stress and twisting moment.

For rod we selected M8(d=8) bolt with 8.8 grade whose ultimate stress is 1000 MPa and yield strength nearly 640MPA and core diameter dc=7.1mm

By formula,

$$\tau_{max.} = \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2}$$

$$\sigma_t = \frac{P}{\frac{\pi}{4} d_c^2} \qquad \tau = \frac{16 M_t}{\pi d_c^3}$$

$$M_t = \frac{P d_m}{2} \times \frac{(\mu \sec \theta + \tan \alpha)}{(1 - \mu \sec \theta \tan \alpha)}$$

For ISO metric screw threads,
$$\theta = 30° \qquad \alpha \cong 2.5° \qquad d_m \cong 0.9\, d$$

μ=0.15

Mt=0.098xpxd

By calculation we got Factor of safety of 6.2 which satisfy our requirement.

14.2 Outside diameter of coupler nut (D)

The actual practice, the diameter of the coupler nut D is taken from 1.25d to 1.5d.

Therefore D=10mm

15. DESIGN OF DIFFERENTIAL CARRIER BRACKET

Design of Differential Carrier brackets involved four stages as follows:

1. Basic Design according to constraints and material selection: Initially a basic design was made on Catia according to design of chassis, half shaft mountings and bearing dimensions. Aluminum 6061 T6 was selected.

2. Topology Optimisation for Mass Reduction: A topology study was carried on hyper mesh for mass reduction. We used Volume tetra 3D meshing and following properties of mesh are recommended for better results

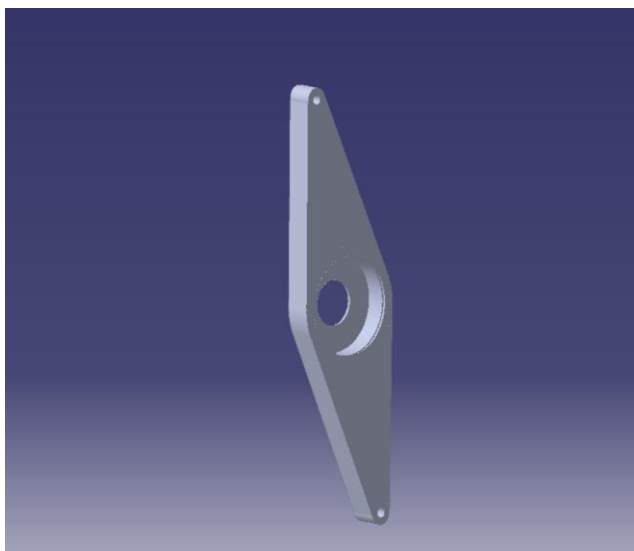a. Warpage < 5

b. Jacobian <0.4

c. Aspect Ratio<10

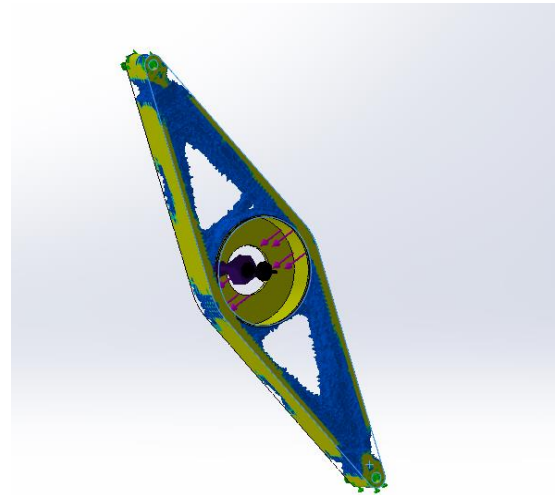*Figure 15-* differential carrier bracket

Figure 16- Topology Optimisation

3. Mass Optimised design -

After topology study we got regions where we can reduce mass. Taking that into consideration we designed optimised differential carrier brackets. Weight of unoptimised bracket was 1202.85 grams and the weight of the optimised design of bracket is 423.42grams   After topology study and optimisation in design we reduced 779.43grams of weight.

 4. Analysis of optimised design -

Analysis of the optimised design was carried out on Ansys to check whether it's safe or not.

Analysis Type - Static Structural

 Material - Aluminum 7075 (T6)

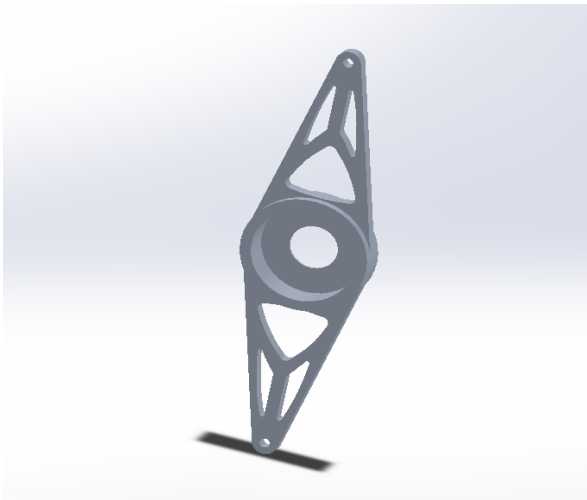*Figure 17-* Topology Optimisation result

Figure 18 optimised design

Meshing- Multi zone Quad/Tri Meshing type with:

1. Fine Relevance center
2. Fine Span Centre
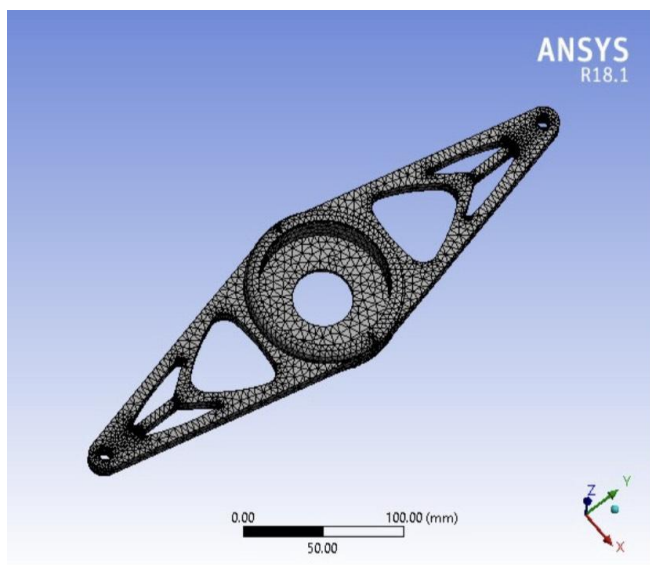3. High Smoothing
4. Element size - 4mm

Figure 19 – Meshing

Boundary Conditions:

The differential carrier bracket had fixed supports at the two mounting holes. A force of 6030N was applied radially outward from the bearing casing towards the engine side.
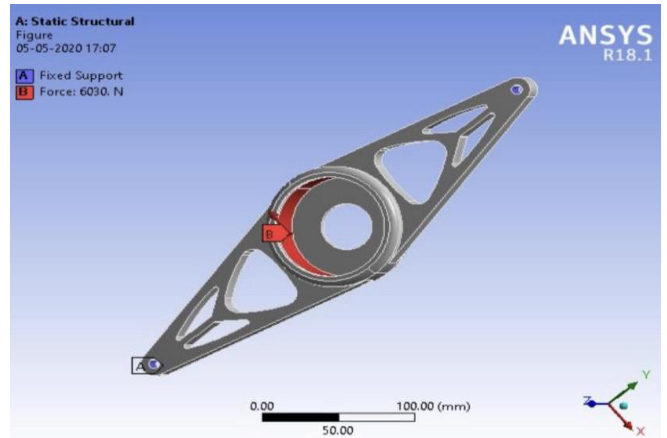

Figure 20-Boundary condition

Young's Modulus	71.7 GPa
Poisson's Ratio	0.33
Ultimate Tensile Strength	572MPa
Tensile Yield Strength	503Mpa

Table 5-Mechanical properties Aluminum 7075 (T6)

Result -

1. Max Von mises Stress = 293.77 MPa
2. Max Deformation = 0.17943mm
3. Factor of safety (FOS)= 1.71

Since the tensile strength of Aluminum 7075 T6 (503MPa) is more than max von mises stress (293.77MPa) the design is considered as safe.
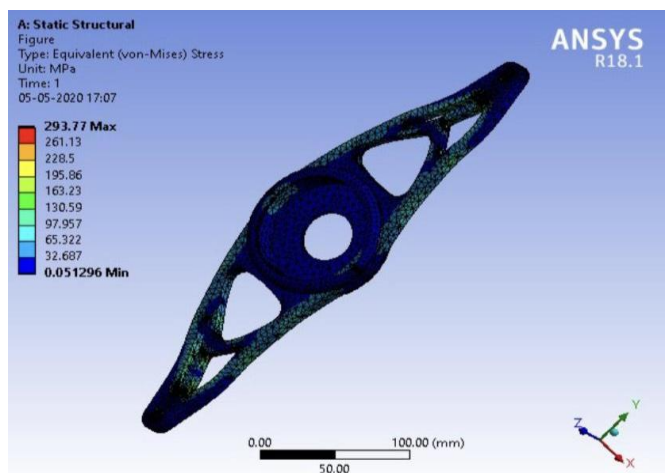

*Figure 21-A*nalysis result (von-mises) stress
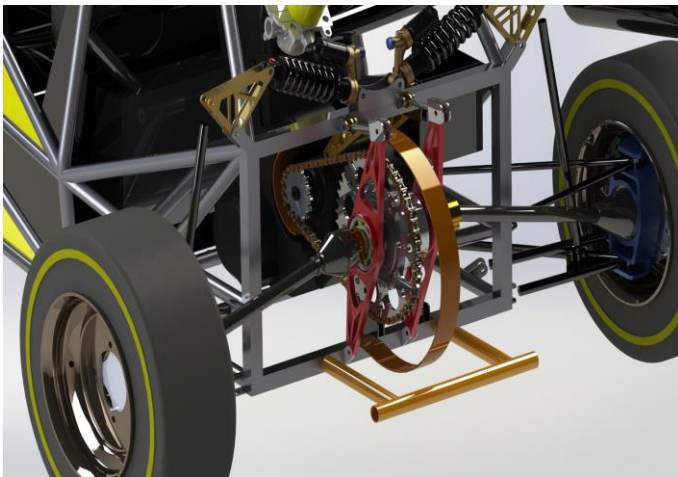
16. ASSEMBLY:

Figure 22-Drivetrain assembly

CONCLUSION –

The purpose of this article is not only to design and manufacture the drivetrain of the car but also provide an in-depth study of the processes and parameters considered for designing it. The FEA results indicates that the drivetrain system is safe to perform on track.

1. The final drive ratio for maximum efficiency was determined to be 3.4

2. The total possible weight reduction from sprocket and differential bracket collectively came out to be 2043.43 grams.

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