K-4 DRIVETRAIN DESIGN REPORT

CONTENTS

1. Design objectives
2. Final drive ratio
    - 2.1 Track layout
    - 2.2 LapSim parameters
    - 2.3 Results
        - 2.3.1 Laptime vs FDR
        - 2.3.2 Speed data
            - 2.3.2 (a) Highest speed vs FDR
            - 2.3.2 (b) Average speed vs FDR
        - 2.3.3 Acceleration data
            - 2.3.3 (a) Max. longitudinal acceleration vs FDR
            - 2.3.3 (b) Max. longitudinal deceleration vs FDR
            - 2.3.3 (c) Max. lateral acceleration vs FDR
        - 2.3.4 Fuel consumption vs FDR
        - 2.3.5 No. of gear shifts vs FDR
        - 2.3.6 % use of a particular gear
            - 2.3.6 (a) % use of gear 1 vs FDR
            - 2.3.6 (b) % use of gear 2 vs FDR
            - 2.3.6 (c) % use of gear 3 vs FDR
            - 2.3.6 (d) % use of gear 4 vs FDR
            - 2.3.6 (e) % use of gear 5 vs FDR
        - 2.3.7 Traction models vs FDR
            - 2.3.7 (a) Traction model for FDR = 7.82 (44 teeth)
            - 2.3.7 (b) Traction model for FDR = 8.01 (45 teeth)
            - 2.3.7 (c) Traction model for FDR = 8.20 (46 teeth)
    - 2.4 Inferences drawn from results
    - 2.5 Rear sprocket selection
3. Differential
    - 3.1 Choice of differential
    - 3.2 Ramp angle configuration selection
    - 3.3 Lubrication
        - 3.3.1 Advantages
        - 3.3.2 Characteristics
4. Chain tensioning mechanism
    - 4.1 Transition to a new mechanism
    - 4.2 Design parameters and Limiting conditions
    - 4.3 Geometry
    - 4.4 Offset Calculation
5. Differential mounts and bearings
    - 5.1 Design decisions
    - 5.2 Calculations

- 5.3 Eccentric disk design consideration
- 5.4 ANSYS static structural simulation
- 5.5 Bearing selection and installation
    - 5.5.1 Selection parameters
    - 5.5.2 Type of bearings
    - 5.5.3 Desired load and speed capabilities
    - 5.5.4 Selection
    - 5.5.5 Tolerances for interference fitting

6. Sprocket adapter
    - 6.1 Need of an adapter
    - 6.2 Calculations
    - 6.3 Types of loading on the adapter
        - 6.3.1 Normal loading
        - 6.3.2 Shock loading
        - 6.3.3 Max. longitudinal acceleration
        - 6.3.4 Max. longitudinal deceleration
        - 6.3.5 Max. lateral acceleration
        - 6.3.6 Chain breaking force
    - 6.4 ANSYS static structural simulation
7. Half shafts

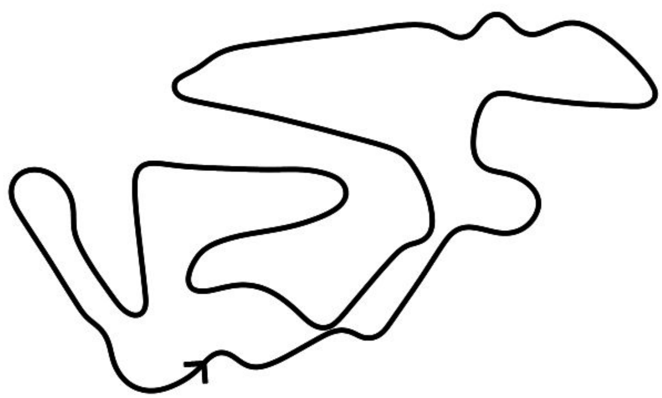
1. Design objectives

The goal of this project was to improve upon the drivetrain design of the previous car K3 by reducing the complexity, weight, and maintenance time. One of the most significant steps taken to accomplish this was to transition from a turnbuckle to an eccentric chain tensioning mechanism. This was chosen because it reduced the time to adjust the chain, a vital factor during the actual competition environment and reduced the overall part count of the system, making the packaging more efficient. Also, the turnbuckle mechanism supported a major part of the load on the threaded tie rods, wearing them out and making it less reliable. The lightweight differential mounts, machined out of AL 7075-T6 are about 1.2 kg lighter than those used in K3, while still being strong enough to support the loads on the differential assembly under different loading conditions. Furthermore, a Lapsim was performed on the Optimum Lap software to evaluate an optimal final drive ratio. Finally, new half-shafts were made of AISI 4340 steel and nitrided on the splined portion to prevent failure due to torsional stress.

2. Final drive ratio

2.1 Track layout

The final drive ratio (FDR) is the reduction between the gearbox and the differential. Because it's not possible to adapt the gear ratios, the torque at the wheels and the vehicle speed has to be regulated by the final drive. To determine an optimal FDR delivered by a sprocket-chain drive, we performed a Laptime simulation on the Optimum Lap software. Our approach was not to judge the vehicle performance by optimizing one or two parameters alone, say top speed or straight line or lateral acceleration or fuel economy but to have a bit of each, since trading off one parameter for another would ensure a good overall performance in all the dynamic events and not just one. To do so, a FSG endurance track layout was selected from the software repository to better analyze the vehicle performance.



FSG endurance track map

Total Track Length	1423.5 m
Percent Left Corners	47.75 %
Percent Right Corners	35.57 %
Percent Straights	16.68 %
Average Corner Radius	97.79 m
Minimum Corner Radius	4 m
Longest Straight	22.3 m

Track data

2.2 LapSim parameters

For the simulation, a range of FDRs was considered, corresponding to the number of teeth that the rear sprocket would have (the front stock sprocket has 15 teeth). This was varied from 34 teeth to 50 teeth. The following vital parameters were evaluated against the range of FDRs:

- Highest speed
- Average speed
- Max. longitudinal acceleration and deceleration
- Max. lateral acceleration
- Lap time
- Vehicle traction model (to evaluate traction limit)
- Fuel consumption
- No. of gear shifts in a lap
- % use of a particular gear

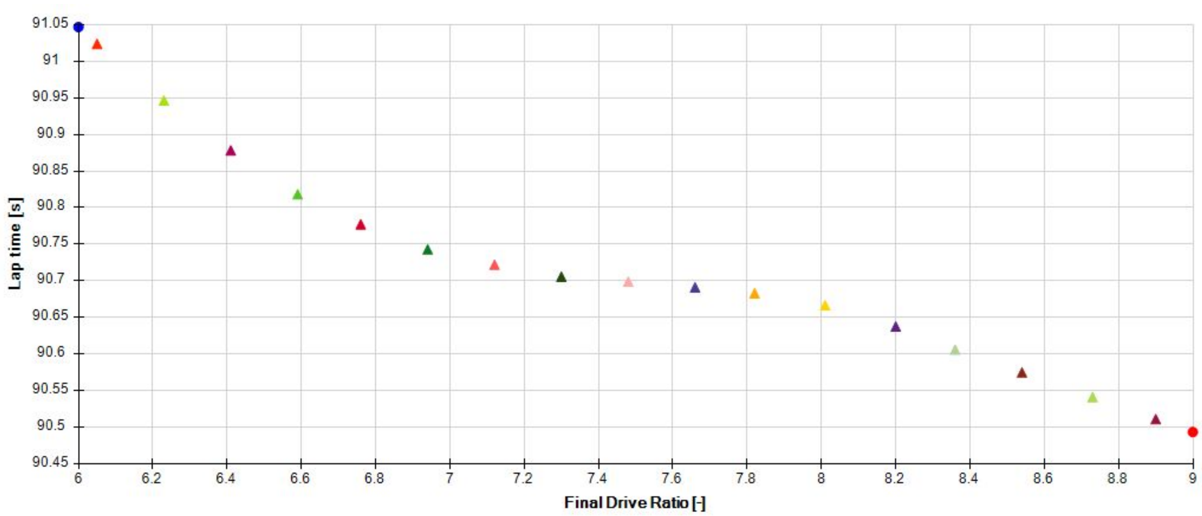
A separate check was also made of the time taken to complete the 75m long acceleration run in each FDR.

(Note: the FDRs shown in the graphs are formulated as:

Primary gear reduction (2.67) x $\frac{No.\ of\ teeth\ in\ rear\ sprocket}{No.\ of\ teeth\ in\ front\ sprocket\ (15)}$ *)*

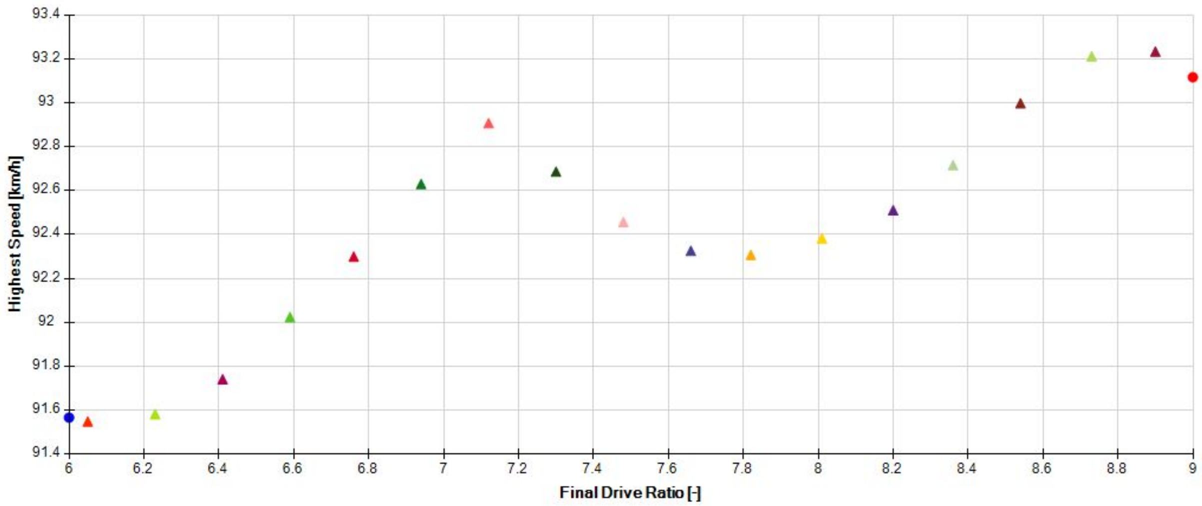
2.3 Results

2.3.1 Laptime vs FDR

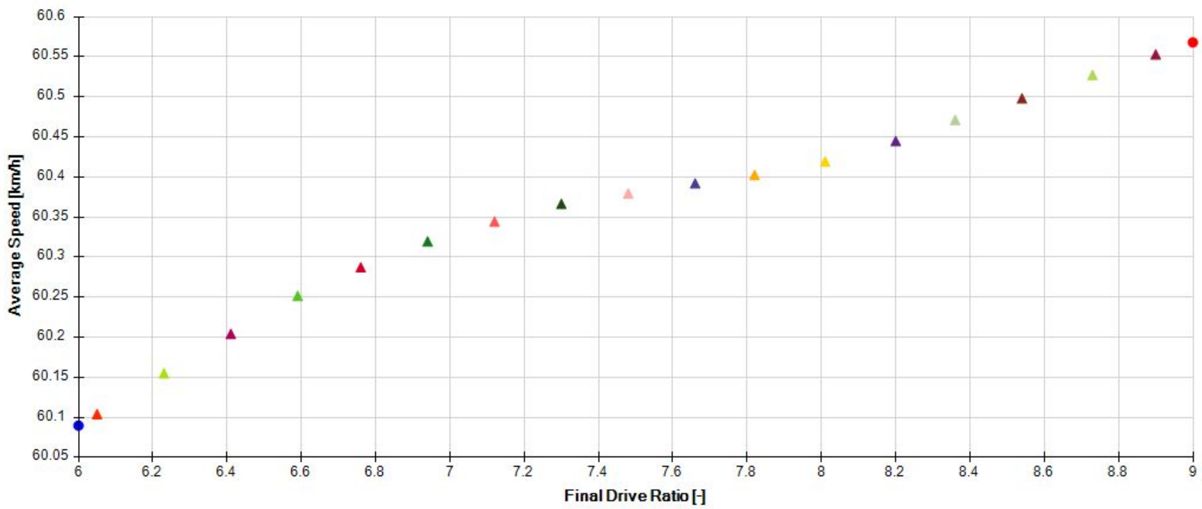


2.3.2 Speed data

2.3.2 (a) Highest speed vs FDR

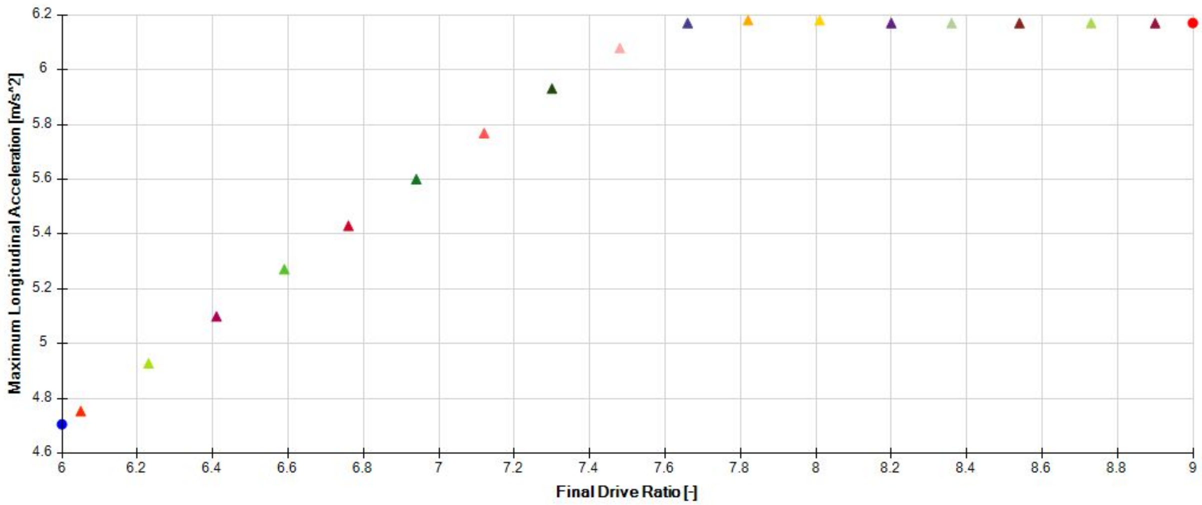


2.3.2 (b) Average speed vs FDR

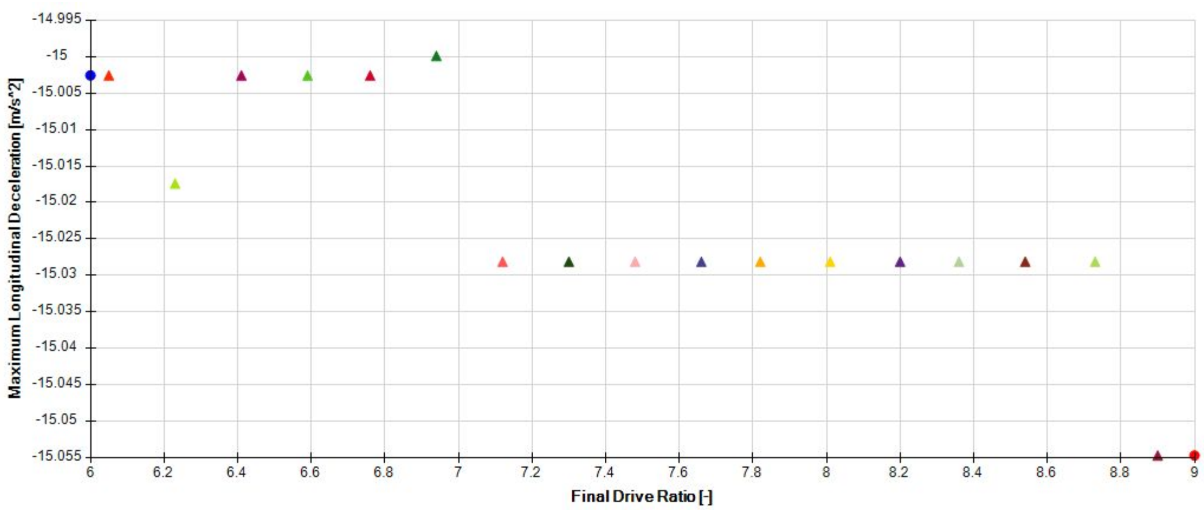


2.3.3 Acceleration data

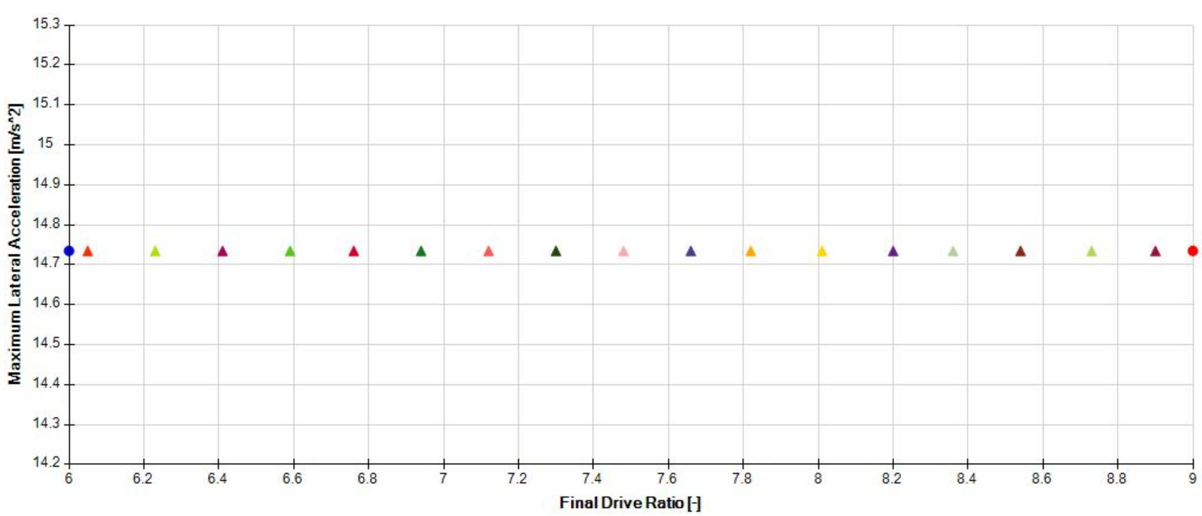
2.3.3 (a) Max. longitudinal acceleration vs FDR



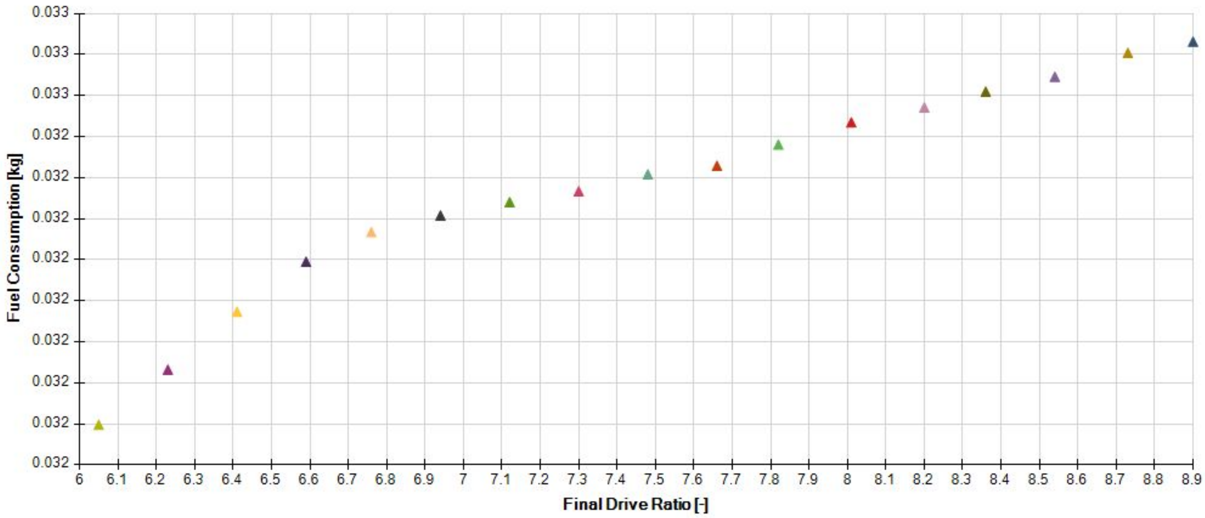
2.3.3 (b) Max. longitudinal deceleration vs FDR



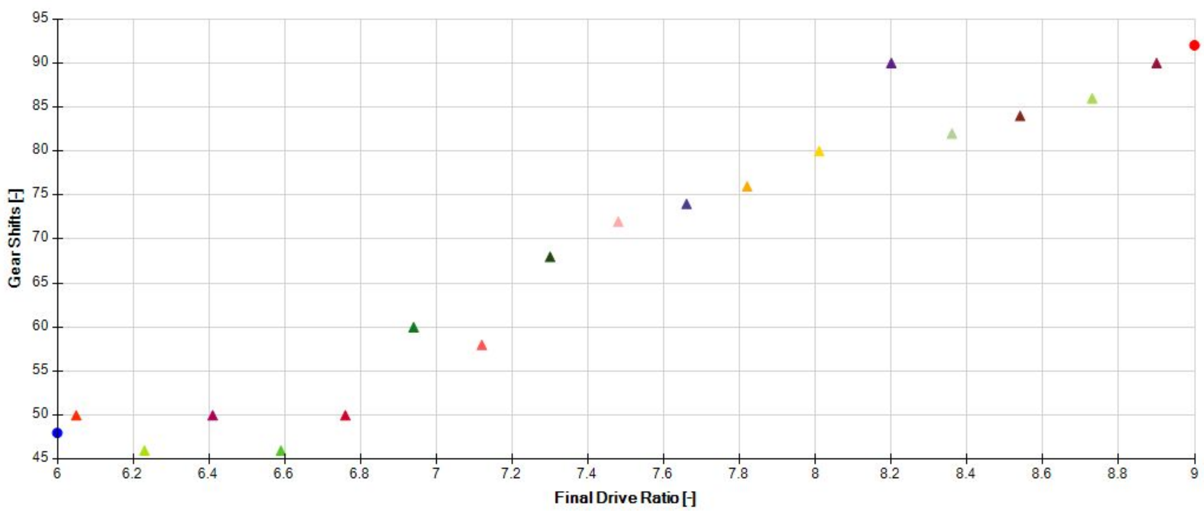
2.3.3 (c) Max. lateral acceleration vs FDR



2.3.4 Fuel consumption vs FDR

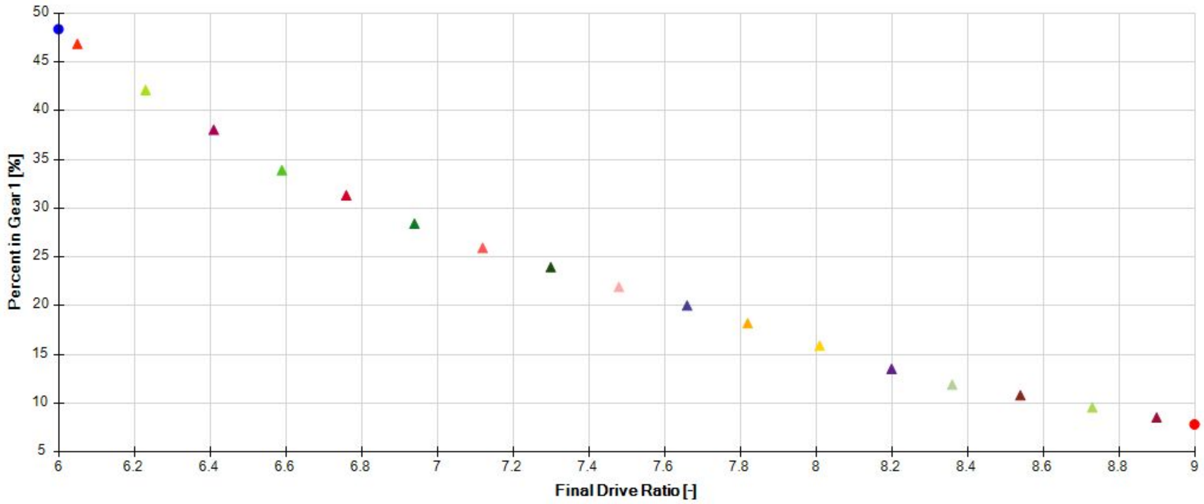


2.3.5 No. of gear shifts vs FDR

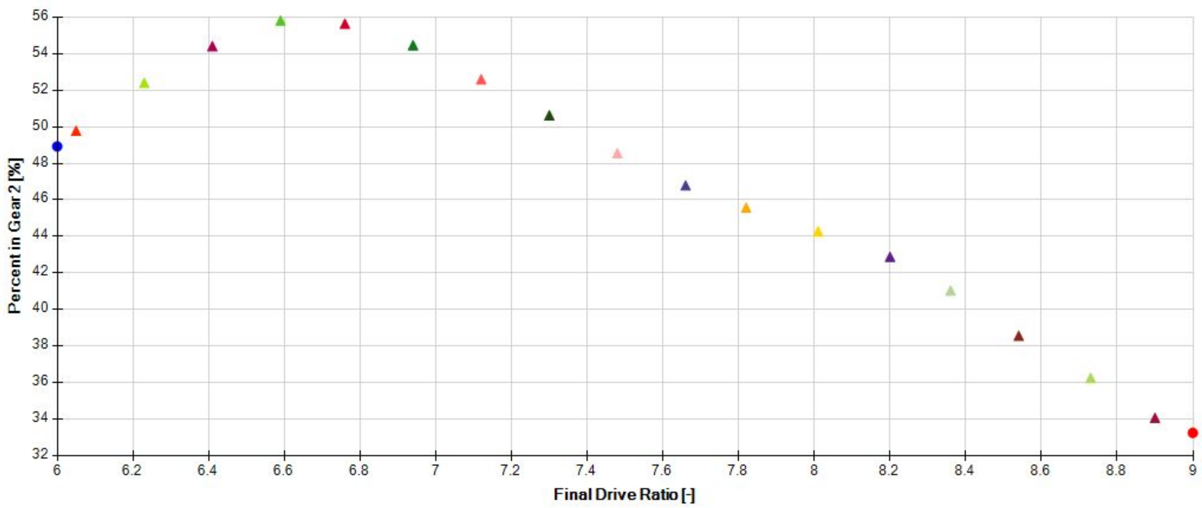


2.3.6 % use of a particular gear

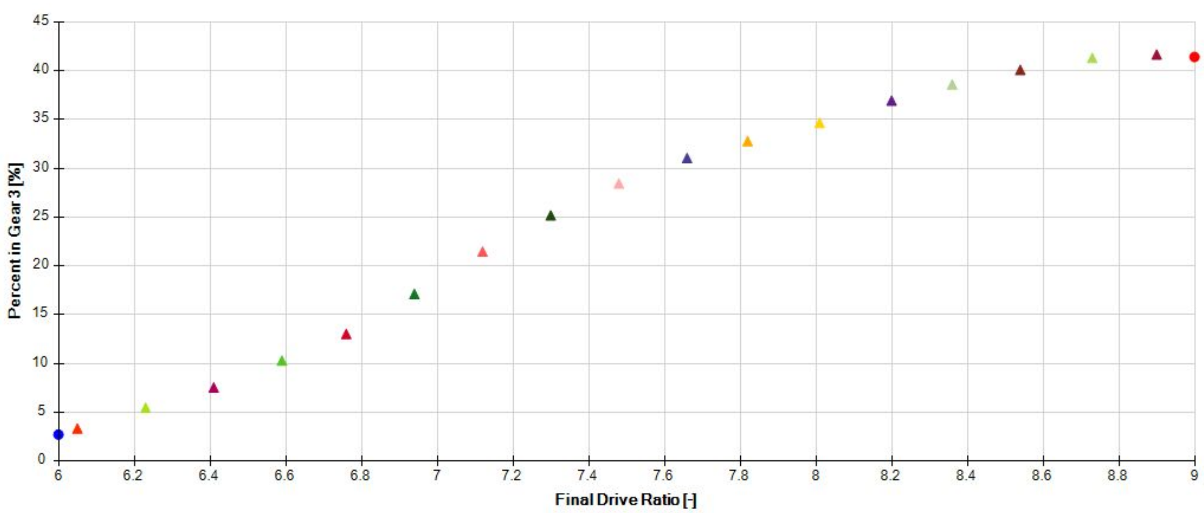
2.3.6 (a) % use of gear 1 vs FDR



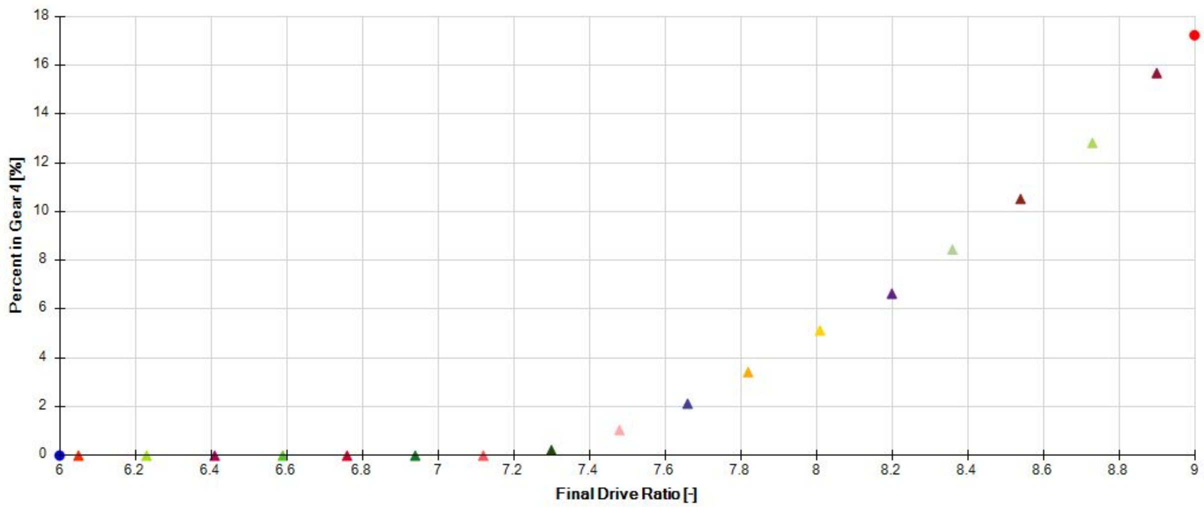
2.3.6 (b) % use of gear 2 vs FDR



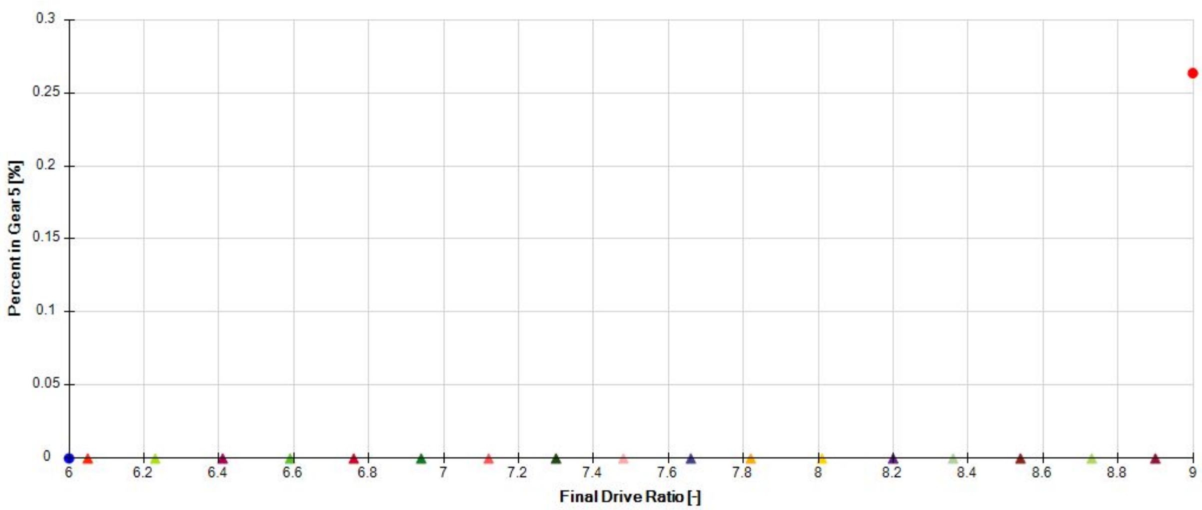
2.3.6 (c) % use of gear 3 vs FDR



2.3.6 (d) % use of gear 4 vs FDR

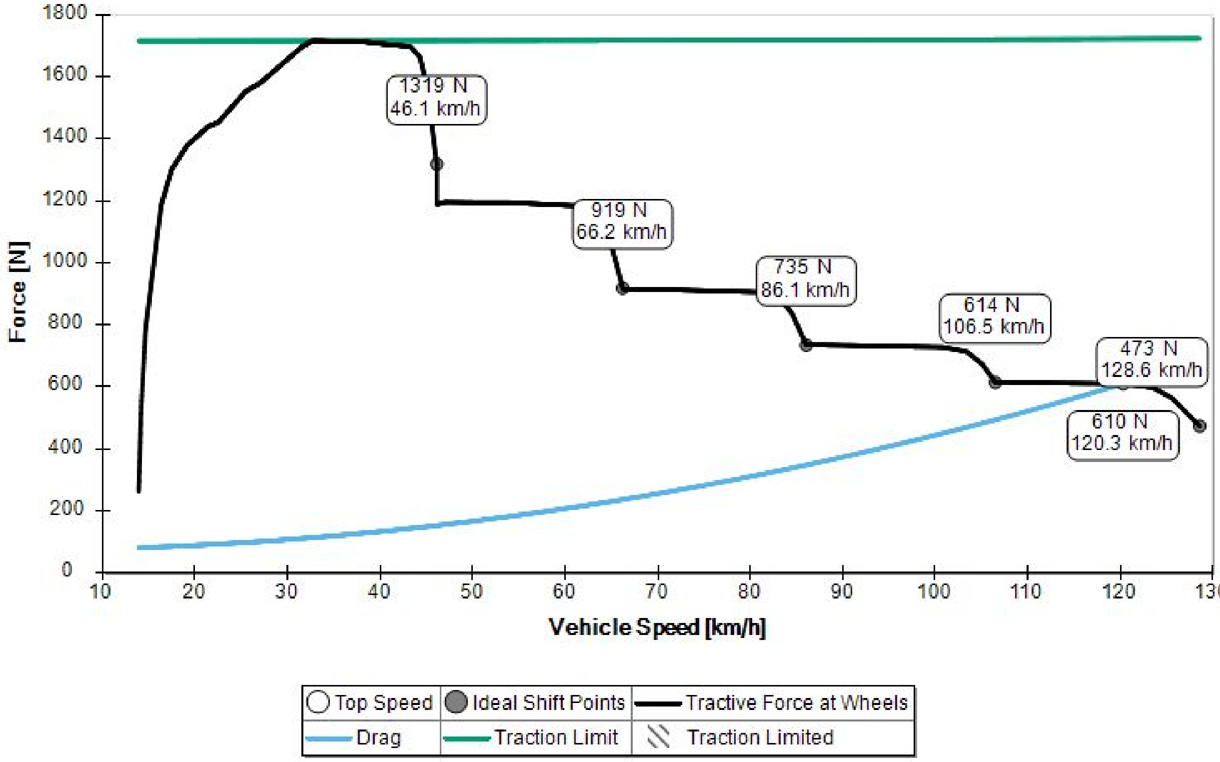


2.3.6 (e) % use of gear 5 vs FDR

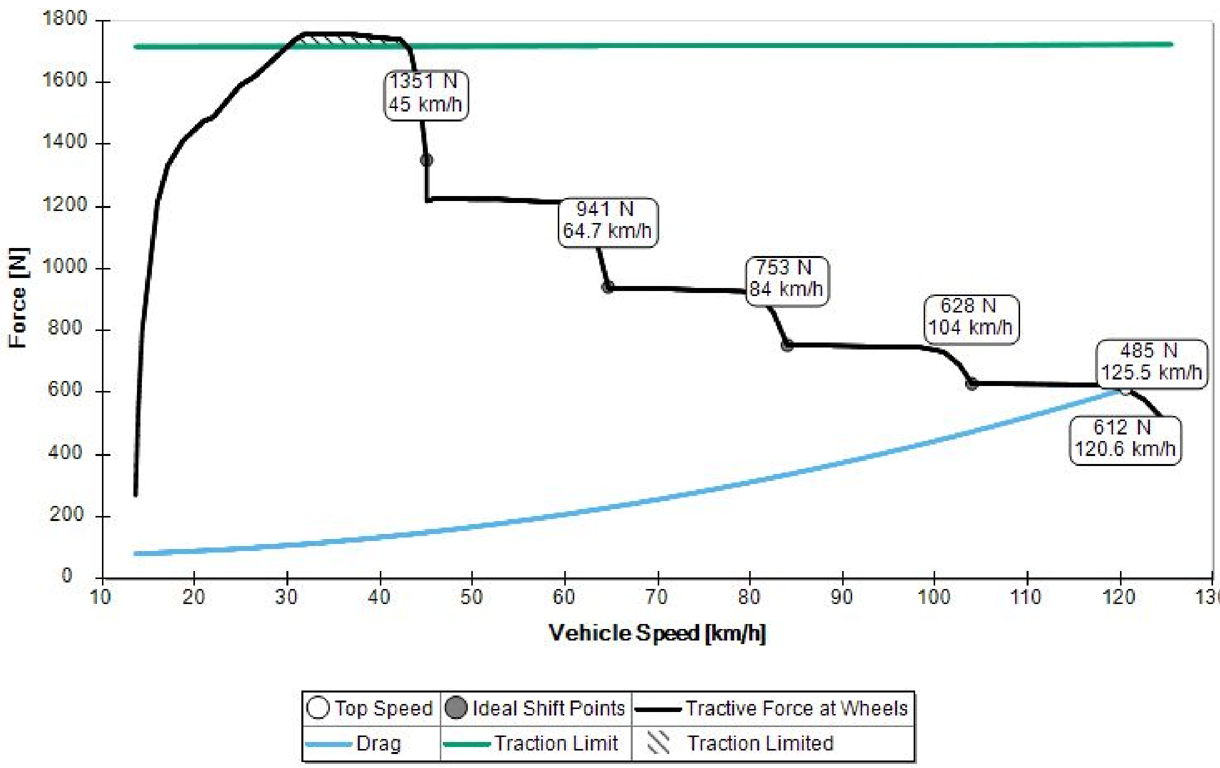


2.3.7 Traction models vs FDR

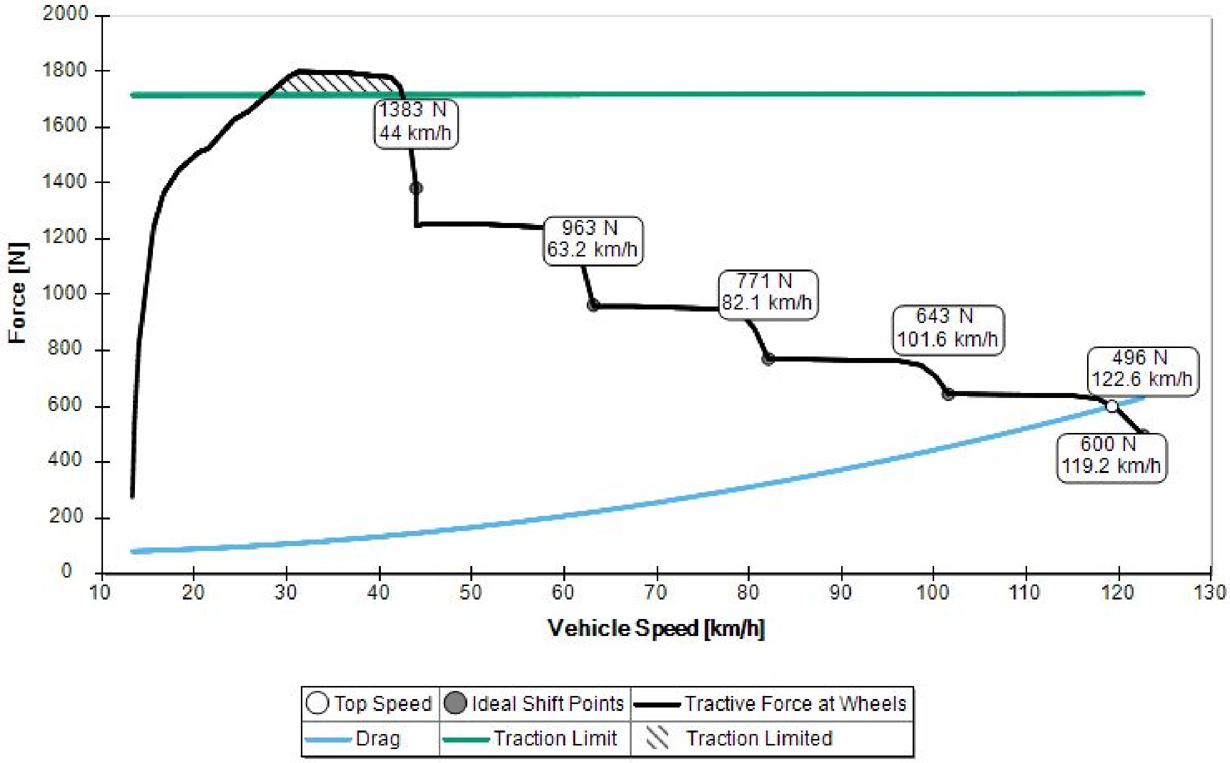
2.3.7 (a) Traction model for FDR = 7.82 (44 teeth)



2.3.7 (b) Traction model for FDR = 8.01 (45 teeth)



2.3.7 (c) Traction model for FDR = 8.20 (46 teeth)



2.4 Inferences drawn from results

1. One of the most interesting observations to make is from the track map data is that the % of corners (83.32%) far outweigh the % of straights (16.68%). As aforementioned, our approach was not just to optimize just a few parameters at the cost of others, i.e., achieving the least possible lap time or the highest possible speed wasn't the sole measure of vehicle performance. Rather, we also took into consideration some non-racing parameters like fuel efficiency as well which contribute significantly to the overall points (100 points) and is a parameter that teams rarely look after. We could aim to achieve the max. speed possible but this would heavily cost us the fuel efficiency points. Rather, we could aim for a decent top speed and this would earn us good fuel efficiency points as well. With so many corners, optimizing solely for speed doesn't make sense.

2. Around the FDR of 8, the lap times are low enough with excellent top speed (about 120 kmph!) and average speed of 60.5 kmph.

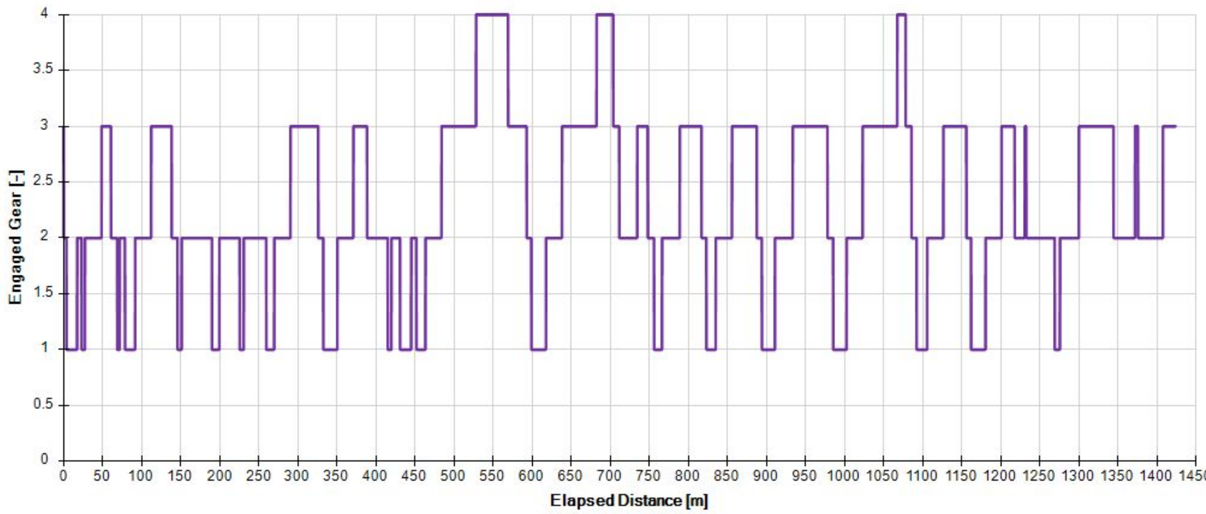
3. Around the FDR of 8, the max. lateral acceleration graph slopes upward and achieves a constant max. value of about 6.18 m/s^2.

4. Max. longitudinal deceleration maintains a consistent value of near about 15 m/s^2 throughout the FDR range.

5. Speed starts increasing and lap times start decreasing from FDR of 8.2 to 8.8, but as obvious from the fuel efficiency graph, this region consumes the majority of fuel! Also, this is the region where the no. of gear shifts made are maximum which is not desirable since, in the case of manual gear shifting, time is lost doing so many gear shifts. Such FDRs are not feasible.

6. While FDRs slightly below 7.8 perform well on fuel efficiency and the no. of gear shifts, the lap times are high, the average speed is lower than the FDRs around 8 and the max. longitudinal acceleration is low as well. This region, therefore, is also not feasible.

7. In context to the % use of a particular gear, the graphs reveal that gears 1,2 and 3 are the most used, with the % of gear 4 used remaining less than 20% in the higher FDRs and negligible use of gear 5. Also evident from the graph shown below, this indicates that choosing a FDR that uses the first three gears in a good combination so as to have a good starting acceleration (as in after exiting a corner) at lower gears as well as high speeds at higher gears should be an obvious decision. FDRs around 8 satisfy this requirement.



Engaged gear vs elapsed distance

8. As obvious, at FDRs of 8.01 and higher, the vehicle becomes more traction limited, meaning that the torque supplied by the engine to the wheels exceeds the traction limit offered by the ground.

2.5 Rear sprocket selection

Eventually, the optimal choice of implementing a FDR of 8.01 was decided which performed fairly well on all the listed parameters, promising a better performance and drivetrain efficiency in the dynamics events. Luckily, the stock rear sprocket that we already possessed, had 45 teeth and hence, it was decided to implement it in the car. Moreover, this saved us the time and money of designing and manufacturing a new sprocket, had the optimal FDR been too different.

3. Differential

3.1 Choice of differential

One of the most important issue with differentials is its weight. Also, a differential must be lubricated, hence must be located in a sealed housing. In normal vehicles, a differential is mounted in a stationary housing. Because of weight and packaging concerns, a FS car cannot afford to implement heavy stationary housing. To contain the lubricating oil, a housing which rotates with the differential must be incorporated into the total design.

A Drexler limited slip differential was chosen to be used in the car. The previous differential, the Quaife differential, was replaced due to the following disadvantages:
1. Being a torque-biased differential, it fails to split the torque between the wheels in case one of the rear wheels loses traction.
2. The Torque Bias Ratio (TBR) being fixed, cannot be adapted according to acceleration and deceleration conditions.
3. Weighs about 3.3 kg without housing.

To overcome these limitations, a Drexler LSD was chosen, since it is specially designed for formula student competitions and is run by several teams.

Drexler Limited Slip Differential

The advantages of Drexler LSD are:

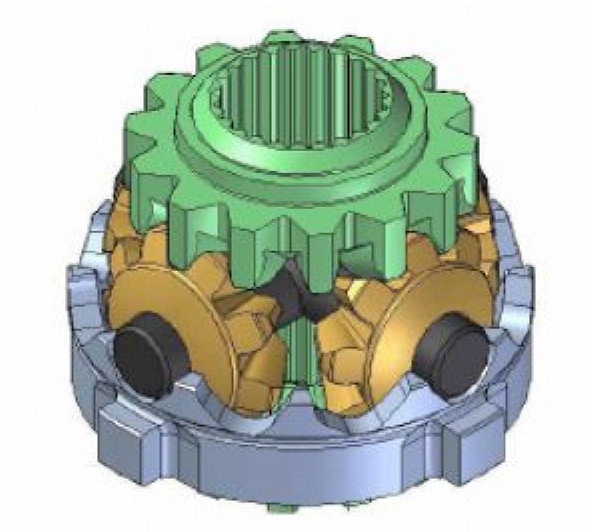
1) Weighs 2.4 kg including housing (very light compared to other differentials)
2) It has a 1.5-way setup, hence behave differently while accelerating and decelerating
3) 3 different combinations of ramp angles are provided to alter the torque bias ratio as required.
4) Max Applicable Torque:1200 Nm is far more than the operating region of a FS Car
5) Oil sealed

6) Long life

3.2 Ramp angle configuration selection

The Drexler Motorsport FSAE LSD ramp angle setup options are as follows:

Angle= 30° ➜ Approx. 88% lock-up
Angle= 40° ➜ Approx. 60% lock-up
Angle= 45° ➜ Approx. 51% lock-up
Angle= 50° ➜ Approx. 42% lock-up
Angle= 60° ➜ Approx. 29% lock-up
The available ramp angle pairs are (30º,45º), (40º,50º) and (45º,60º).



Spider gears and ramp angles

The hole in the ground ramp for the cam has a chamfer. When the ground ramp is driven the chamfer pushes the ground ramp to the outside. With this movement, clutch plates are pressed together which creates friction. How hard the clutch plates are pressed together depends on the shape of the chamfer. Half of the clutch plates are connected to the side gear and the other half to the ground ramp. The friction creates a connection between the side gear and the housing, and so between the engine and the wheels. When there is deceleration the cam pushes to the other side of the hole in the ground ramp and a different amount of pressing force is applied on the clutch plates.

We analyzed the worst-case scenario wherein the car maneuvers around the corner with max. lateral acceleration in order to get the maximum lock-up torque possible. When the car accelerates through the corner, the outer wheels turn faster than the inner ones and also, there is lateral load distribution. Weight transfers to the tires on the outside of the turn, causing them to have more traction than the inner tires. Assuming that the car takes a right turn around a corner, the following calculations were performed to obtain the lock-up torque.

Load on the outer drive wheel(left)   =135g

Load on the inner drive wheel(right) =25g

Defining Torque Bias Ratio(TBR) = max.(M_L,M_R) / min.(M_L,M_R) (M_L= torque on left wheel M_R= torque on right wheel)

= 135g/25g (Torque is proportional to the load on the wheels)

= 5.4

This means that the wheel torque on the left is about 5.4 times the wheel torque on the right. Hence, the clutch pack on the left wheel side will be pressed more than that on the right wheel side.

Defining Lock-Up Torque S(%) = {|M_L-M_R| / (M_L+M_R)} * 100
= {|135g-25g| / (135g+25g)} * 100
= (11/16)*100
= 68.75%

This means that the pressure applied on the clutch pack on the side of the left wheel (84.375%) will be about 68.75% more than that on the side of the right wheel (15.625%). Now, according to the specifications of the Drexler LSD, the nearest lock-up torque available is 60% and the corresponding ramp angle is 40°. This 40° ramp angle is paired with 50° ramp angle. Hence, the most suitable ramp angle set chosen is (40°,50°). During acceleration, the 40° ramp angle is active giving a 60% lock-up torque (TBR=4) while during deceleration, the 50° ramp angle is active giving a 42% lock-up torque (TBR=2.45).

3.3 Lubrication

The lubrication of the differential is done using Castrol Syntrax Limited Slip 75W-140. Castrol Syntrax Limited Slip 75W-140 is fully synthetic hypoid gear oil formulated for use in both conventional and limited slip differentials. Its flash point (228^0 C) is well above and pour point (-54^0 C) is well below the operating temperatures ($\sim 25^0$ C).

3.3.1 Advantages

1. Reduces noise and vibrations in the axle.
2. Excellent shear stability maintaining performance over the life of the lubricant.
3. Exceptional stability at high temperatures (due to high Viscosity Index) extending the life of lubricant and axle.
4. Effective wear resistance ensures protection under high loads and prevents damage to components.
5. Very good low-temperature fluidity properties increase protection at startup.

3.3.2 Characteristics

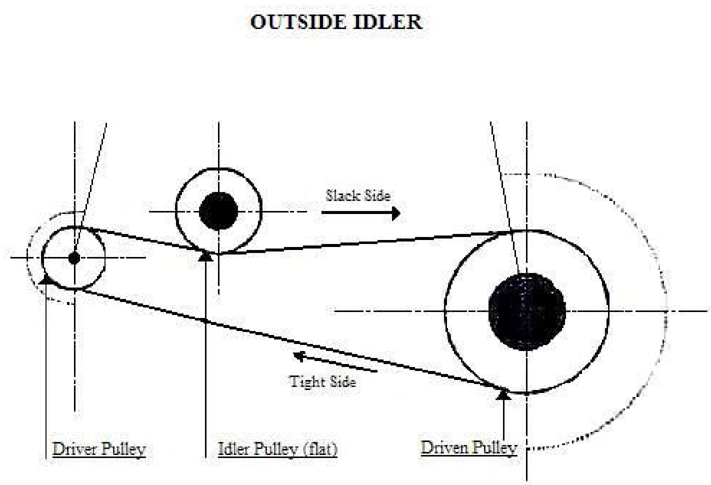
Name	Method	Units	Syntrax Limited Slip 75W-140
Viscosity, Kinematic 100C	ASTM D445	mm^2/s	24.7
Viscosity, Brookfield @ -40C	ASTM D2983	mPa.s (cP)	120000
Appearance	Visual	-	Bright & Clear
Flash Point, COC	ISO 2592	°C	228
Pour Point	ISO 3016	°C	-54
Density @ 15C	DIN EN ISO 12185	g/ml	0.857
Viscosity, Kinematic 40C	ASTM D445	mm^2/s	175
Viscosity Index	ISO 2909	None	174

Castrol Syntrax Limited Slip 75W-140 properties

4. Chain tensioning mechanism

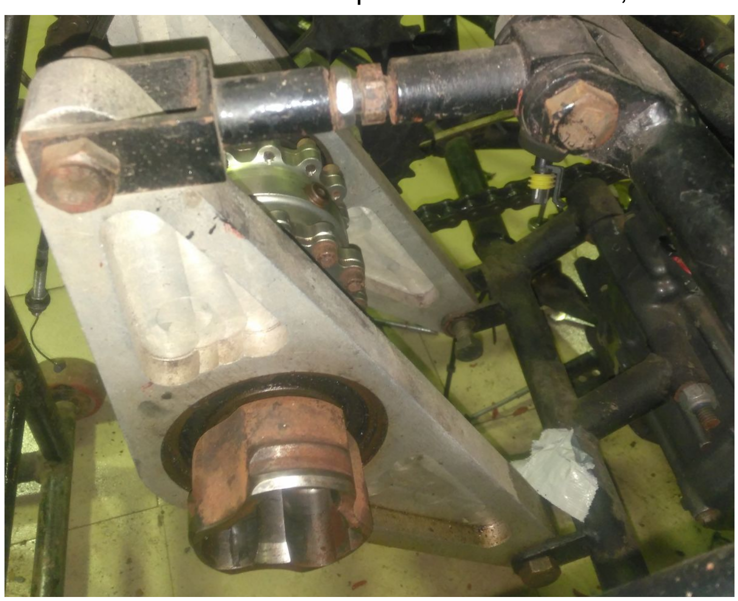
4.1 Transition to a new mechanism

The chain used to drive the car is 520. An efficient chain tensioning method is designed to keep the chain in correct tension to prevent wear of the chain and prevent the possibility of hopping off the sprocket. The previous two cars, K2 and K3 used idler pulley tensioner and turnbuckle tensioner respectively.



Idler pulley chain tensioner

The pulley causes mechanical loses as it uses power to rotate itself, hence it was discarded.



Turnbuckle chain tensioner

The turnbuckle mechanism was discarded because the bolt couldn't bear the stresses and the threads were wearing out. Also, this design increased the part count leading to complexity.

Hence in the current car, it was decided to employ an eccentric chain tensioning mechanism. The design consists of an eccentric disk which houses the differential bearings. The eccentric disk is mounted to the differential mounts. The centreline of the bearing housing is offset from the center of the eccentric disk by 10.5 mm.

4.2 Design parameters and Limiting conditions

The various parameters that are crucial in the design of the eccentric disk are as follows:
1. The initial distance between the front and rear sprocket at absolute chain length.
2. The angular rotation of the disk.
3. The offset of center.

The design had to have a few boundary conditions and they were as follows:
1. The smallest change in length of the chain achievable had to be smaller than approx.0.5cm to make the mechanism feasible at subtle changes in tension of the chain(slack).
2. The largest achievable change in length had to be less than that of single link of the chain, as otherwise the chain would have lost its value! The limit was set to approximately 2.0cm.

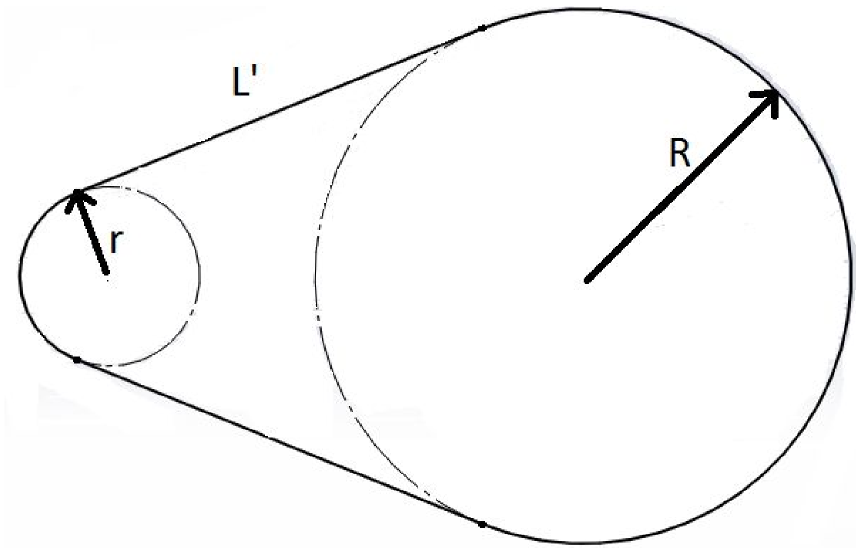
4.3 Geometry

Deciding the geometry was the most crucial part in the design of the eccentric disk. The sketch was created and bound to the following conditions:
1. Tangency of the chain to the rear sprocket.
2. Tangency of the chain to the front sprocket.
3. Center to center distance of the front and rear sprocket.

The sketch of the chain wrapping around both the sprockets were made taking the line joining the centers as the reference(x-axis) to avoid ambiguity. In the sketch the front and rear sprockets were represented by their pitch circles. The sketch includes the circle enclosing the eccentric disk. The eccentric disk has a hole with a variable offset and the circle representing the hole remains concentric to the pitch circle the rear sprocket at all times. The point of tangency also had to decided for the case of zero chain tensioning. This was done by counting the number of teeth enclosed within the chain and thereby calculating the angle of wrap for both the sprockets. The geometry includes a few variables that had to be decided for the design and they are as follows:
1. Offset.
2. Angle of rotation of the eccentric disk.
3. Change in length of chain.

The objective was to decide the optimum change in length of the chain bound to the above mentioned boundary conditions. As a matter of concern, this value also depended on the center to center distance between the two sprockets and hence, was fixed to a value of 201mm.

4.4 Offset Calculation



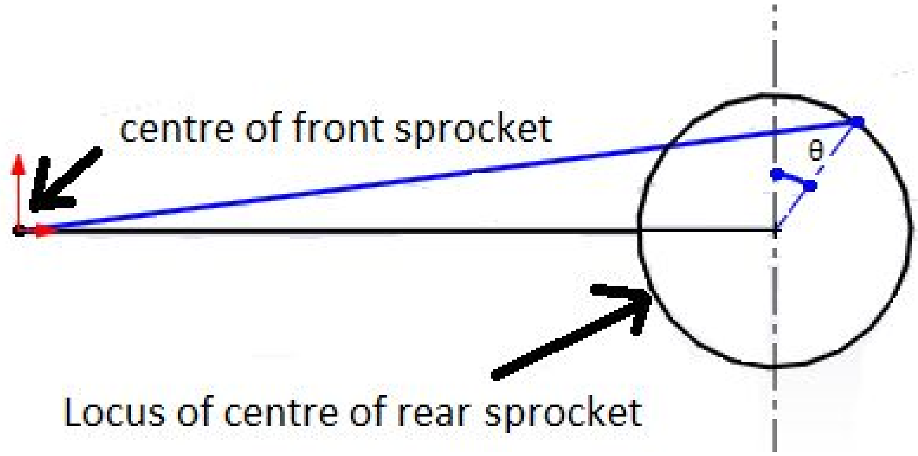
From the diagram shown above, we calculate the total length of the chain wrapping both the sprockets.

The following denotations will be used:
L' = half of the length of the chain not in contact with any of the sprockets.
r = radius of drive sprocket.
R = radius of driven sprocket.
P = total length of the chain wrapping both the sprockets.

$P = 2L' + r[\,\pi - ((2\phi\pi)/180)\,] + R[\,\pi + ((2\phi\pi)/180)\,]$
$\quad = 2L\cos(\phi) + r[\,\pi - ((2\phi\pi)/180)\,] + R[\,\pi + ((2\phi\pi)/180)\,]$ $\quad (L' = L\cos(\phi))$
$\quad = 2[\,L^2 - (R - r)^2\,]^{1/2} + \pi(R + r) + (\pi/90)(\phi)(R - r)$ $\quad (L\cos(\phi) = [L^2 - (R-r)^2]^{1/2})$
$\quad = 2[\,L^2 - (R - r)^2\,]^{1/2} + \pi(R + r) + (\pi/90)(R - r)[\,\sin^{-1}((R-r)/L)\,]$ $\quad (\sin(\phi) = (R-r)/L)$

Since 'ϕ' and 'L' are two variables here, we have expressed 'ϕ' in terms of 'L' to simplify the calculation.

Therefore, $P = 2[\,L^2 - (R - r)^2\,]^{1/2} + \pi(R + r) + (\pi/90)(R - r)[\,\sin^{-1}((R - r)/L)\,]$ ------→ (i)



According to the diagram shown above, we analyze the change in the total length of the chain wrapping both the sprockets when the eccentric disk is rotated through an angle '$\boldsymbol{\Theta}$' ranging from 0 to 90°. We need to find out an optimum value of the offset 'd' of the disk with respect to the differential mount.

From the diagram,
$-\sin(\boldsymbol{\Theta}) = (D^2 + d^2 - L^2)/(2Dd)$
$\Rightarrow L = (D^2 + d^2 + 2Dd \sin(\boldsymbol{\Theta}))^{1/2}$ ------→ (ii)
Here, 'L' = center-to-center distance between the two sprockets (set 201 mm @ $\boldsymbol{\Theta}$=0) and 'D' = 190.422mm (obtained from (ii) @ $\boldsymbol{\Theta}$=0).

Thus, at $\boldsymbol{\Theta}$=0 and R = 113.79mm, r = 38.18mm, we get 'P' = 889.18mm from (i).

Now, we will evaluate the results for the case when '$\boldsymbol{\Theta}$' = 0, 'P' = 889.18mm and offset 'd' = 10.5mm.
Substituting rest of the values in (i), we get L = 190.79mm as the center-to-center distance between the two sprockets.

We now perform various iterations by ranging the value of $\boldsymbol{\Theta}$ from 0 to 90° in (ii) and get corresponding values of L from (ii) which are then substituted in (i) to get different values of P. From these values of P we find out the change in the total length of the chain wrapping both the sprockets after each iteration.
We take offset 'd' =12mm and calculate the change in total length of chain for change in θ by 20°.

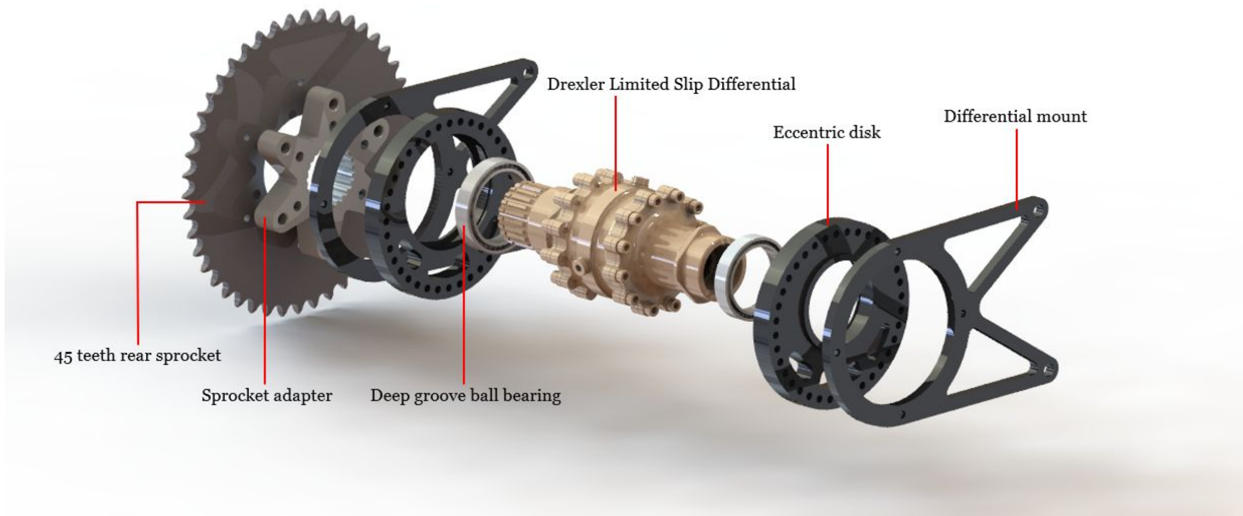
θ	P(mm)	Δ P(mm)
20	896.872291	7.452868853
40	903.3235476	6.451256619
60	908.053375	4.729827454
80	910.5526545	2.499279438

The change in length was seen to be very high so we repeat this procedure by taking offset 'd' = 10.5mm and get the change in the total length of the chain with a step of 10°.

θ	P(mm)	Δ P(mm)
10	892.7961631	3.328336325
20	895.9992393	3.20307623
30	898.9835632	2.984323875
40	901.663827	2.68026387
50	903.9648503	2.301023242
60	905.8231442	1.85829395
70	907.1881302	1.364985979
80	908.0230441	0.834913908
90	908.3055583	0.2825141749

The change in length was satisfying all the design constraints for an offset of 10.5mm. So an offset of 10.5mm was chosen.

5. <u>Differential mounts and bearings</u>



Exploded view of drivetrain assembly

5.1 Design decisions

1. All components must be as light as possible without compromising the structural integrity.
2. Since the rear chassis is very tightly dimensioned, the mounts must be packaged as precisely as possible.
3. To mount the differential, two options were considered:

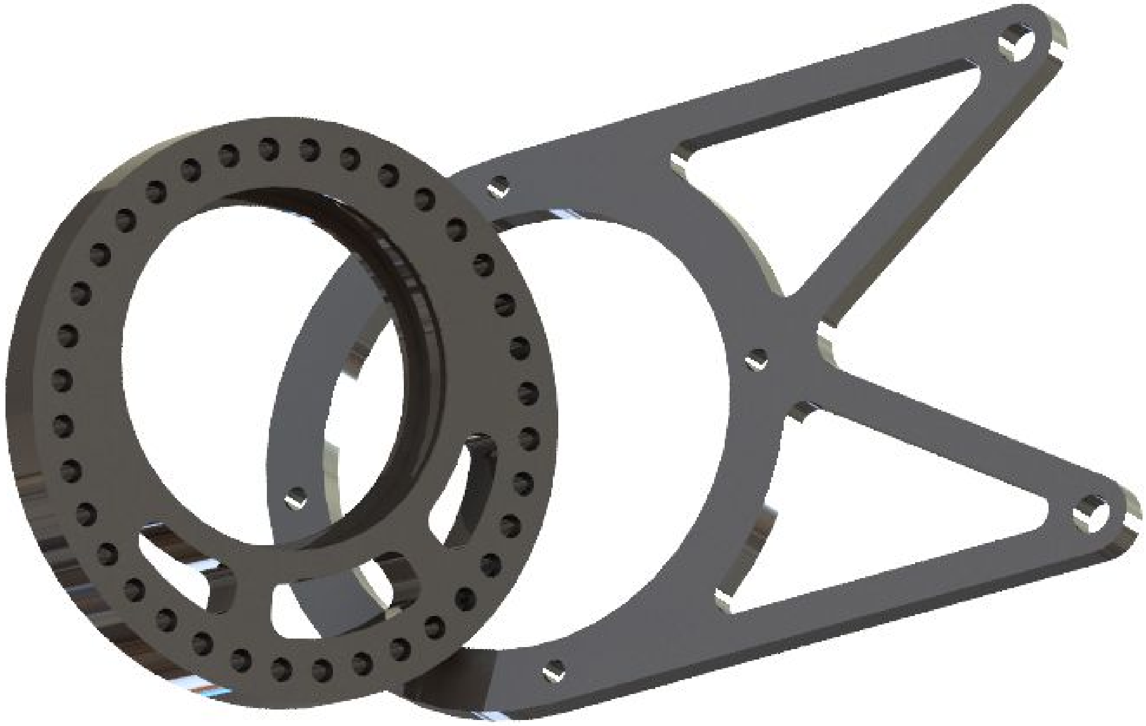
Option 1: Mounting the differential to the rear chassis.

This option, though implementable, has some limitations. Firstly, this would possibly interfere with the suspension setup which might lead to a mount size that would be overweight or large in size. Even if this issue would be ironed out in the design phase, mounting issues might arise during the assembling of the components. Secondly, this design might increase the load on the rear chassis, which would shift the vehicle CG further rear, leading to the possibility of oversteering during corners. Not affecting the suspension setup seemed the better solution, leading to an independent mount design.

Option 2: Mounting the differential to the engine mounts

This solution gave a freedom to optimize the mount design without having any effect on the suspension setup. Moreover, this would allow us to move the differential towards the engine, thus shifting the vehicle CG forward and achieving a static front-rear weight distribution balance. An added benefit would be the reduction in the polar moment of inertia of the vehicle, leading to less resistance on vehicle during cornering.

Based on these options, it was decided to design the mounts to be mounted to the engine mounts.



Differential mount and eccentric disk

The mount has four 6 mm holes to attach the eccentric disk and two 10 mm holes to attach to the engine mounts.

5.2 Calculations

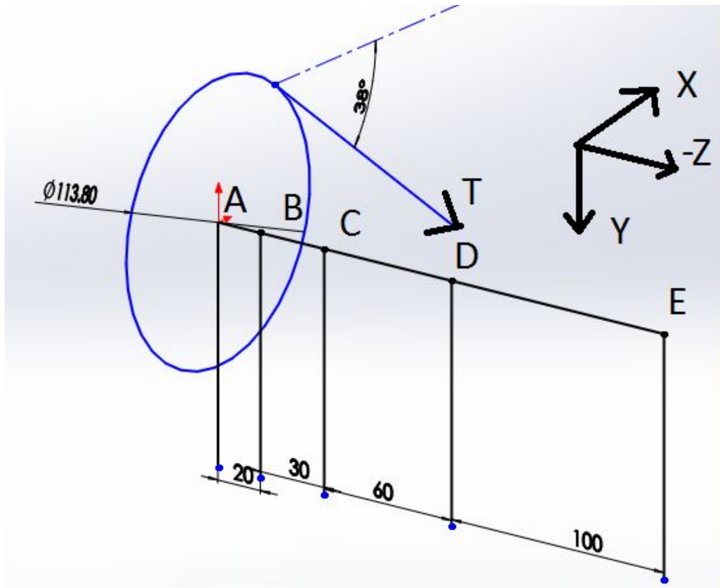
Engine Data:
Maximum Engine Torque(T)=33Nm
Final Drive Ratio(FDR)=3
Primary Gear Ratio(P)=2.667
Ist Gear Ratio(G_1)=2.667

Rear Sprocket Data:
Pitch(P)=15.875 mm
Angle per teeth(2β)=180/45=4^0
Pitch Radius of Rear Sprocket(r)=15.875/(2×sin(4^0))=0.1138 m
Pressure Angle(ø)=17.55^0
Wrap angle(A)= 256^0

Mass of components:
Rear sprocket=1.5 kg
Adapter=.3kg

Bearings=.5kg
Differentials=2.4kg

The figure shows the arrangement of the CG of components:
A: Rear Sprocket B: Adapter C: Left Differential Mount D: Differential E: Right Differential Mount



(All dimensions in mm)

Maximum torque at rear sprocket "M" = T× FDR× P× G_1=33× 3× 2.667× 2.667=704 Nm
Tension at the k^{th} roller in the chain = $T_k = T_0 \times \{\sin ø \div \sin (ø + 2β)\}^{k-1}$
No. of teeth in contact = A/360*45=32
For k=32,
$T_k = T_0 \times \{\sin 17.55 \div \sin (17.55 + 4)\}^{31} \approx 0$

$M=(T_0-T_k)\times r=T_0\times r$
$\Rightarrow T_0$=M/r=704/.1138=6186.3N
Tension in the chain=6186.3N
$T_x = T_0 \cos\Theta$ = 6186.3*cos 38 = 4876N
$T_y = T_0 \sin\Theta$ = 6186.3*sin 38=3852N

Balancing the forces along x and y-direction:

Along X: $R_{x1}+R_{x2}$=4876 N
Along Y: $R_{y1}+R_{y2}$=3810+15+3+5+24+5=3862 N

Balancing the moments along x and y direction:

Along X: 0.02× 3 + 0.05× 5 + 0.11× 24 + 0.21× 5 - $0.05R_{y1}$ - $0.21R_{y2}$=0
Along Y: 0.05× R_{x1}+0.21× R_{x2} =0

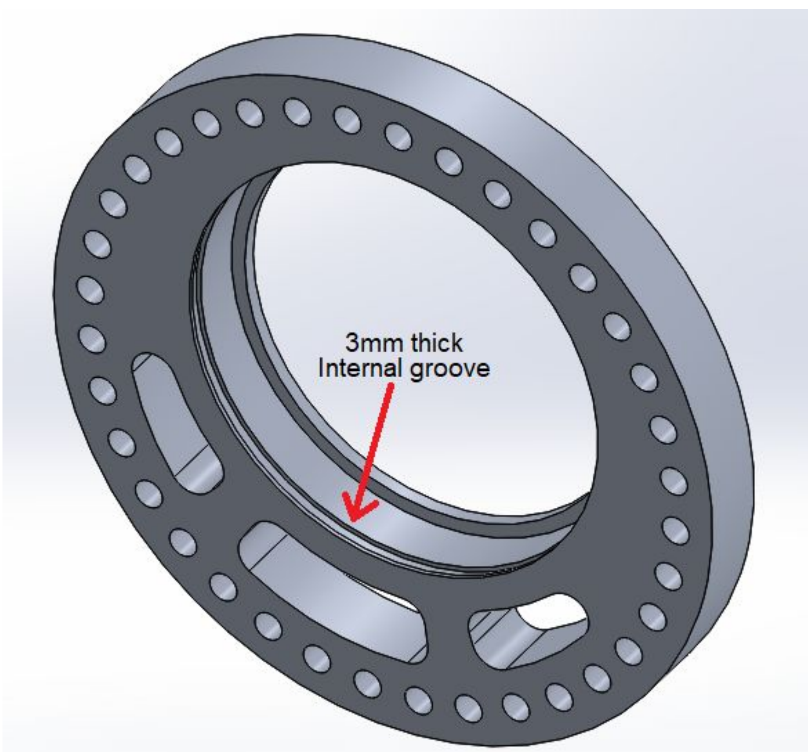
Solving these equations we get:
R_{x1} = 6400N

R_{x2} = -1524N
R_{y1} = 5050N
R_{y2} = -1189N

5.3 Eccentric disk design consideration

In any design process, following a “worst-case scenario” approach is always recommended to ensure the reliability of a component in any situation. Sometimes, this might lead to over-designing the part but where special attention needs to be given, such trade-offs are always feasible. Following this principle, a design feature was incorporated into the eccentric disks.

In the eccentric chain tensioning assembly, the eccentric disk is bolted to the differential mount and is seated on the differential itself via a deep groove ball bearing interface. The bearing’s outer race is press-fitted (or interference-fitted) into the disk and its inner race is press-fitted onto the differential’s circular tubular surface. In order to prevent the bearing from accidentally coming out of the disk due to the transfer of axial force via half shafts and slipping of the disk-bearing interface, an internal groove was made inside the disk as shown.



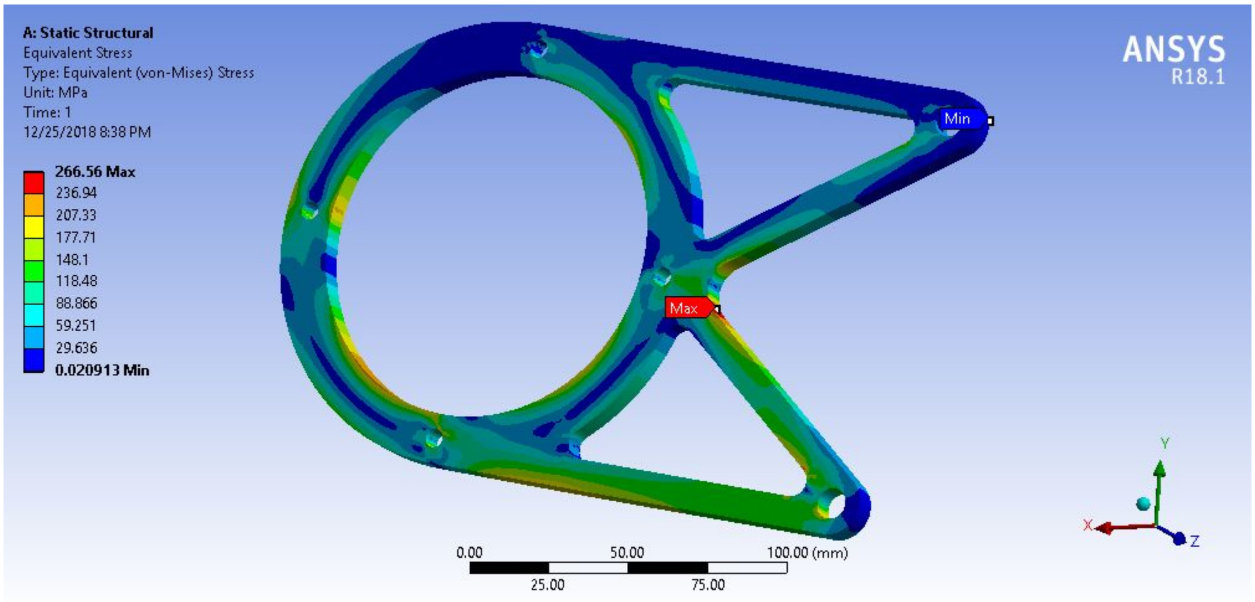
Internal groove for circlip

After press-fitting the bearing into the disk, a circlip is mounted inside the groove which blocks the circular hole thereby preventing the bearing from slipping out of the disk.

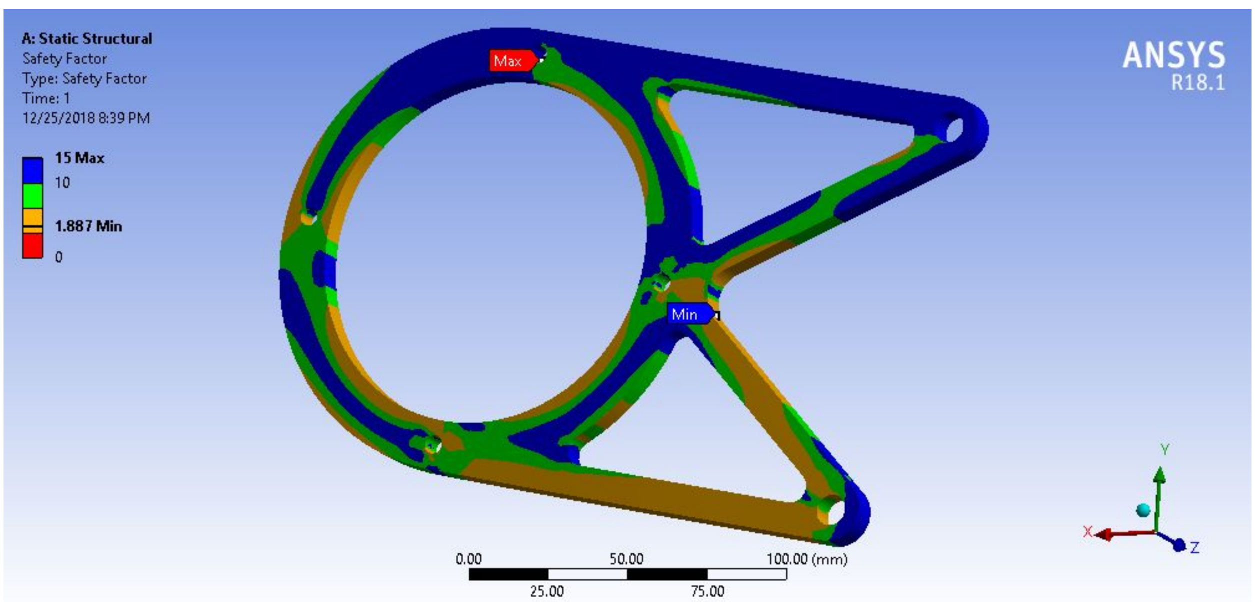
5.4 ANSYS static structural simulation

Five different loading conditions were simulated:
1. Normal loading with just the forces in longitudinal and downward direction
2. Normal loading along with a max. longitudinal acceleration of 6.18 m/s^2
3. Normal loading along with a max. longitudinal deceleration of 15.03 m/s^2
4. Normal loading along with a max. lateral acceleration of 14.73 m/s^2
5. Chain breaking load (14kN) along with max. longitudinal acceleration of 6.18 m/s^2

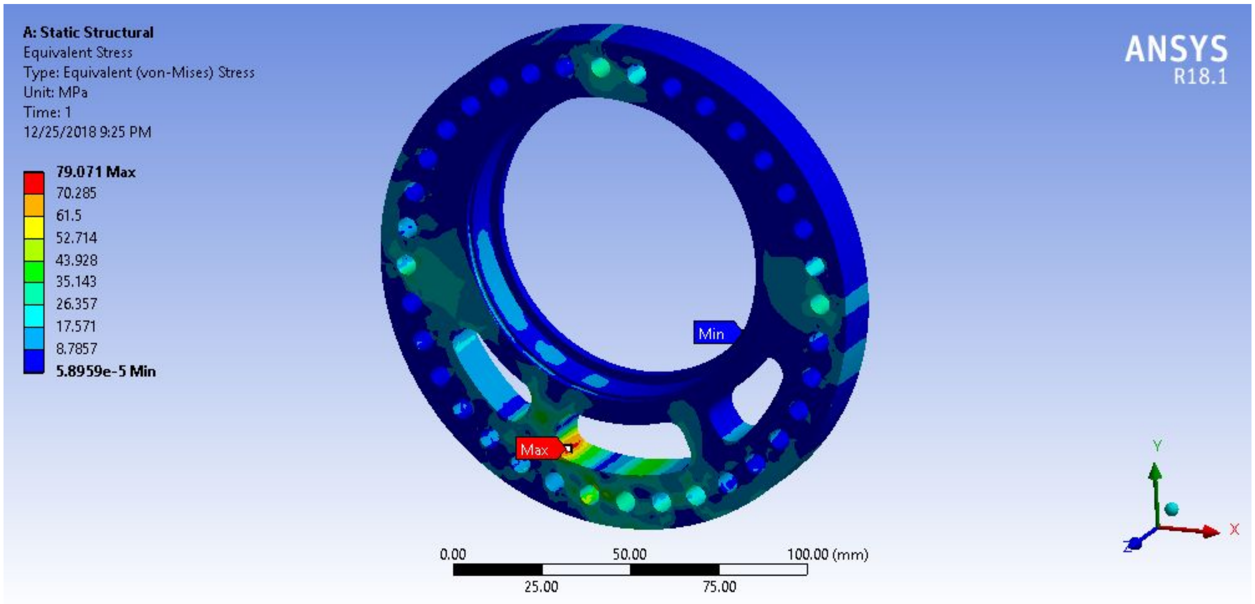
As obvious, the 5th case yields the min. FoS. Aluminium 6061-T6, Aluminium 6082-T6 and Aluminium 7075-T6 were considered as the appropriate materials. Simulation results showed that under the 5th loading case, Aluminium 7075-T6 would perform the best, leading to its selection and implementation.



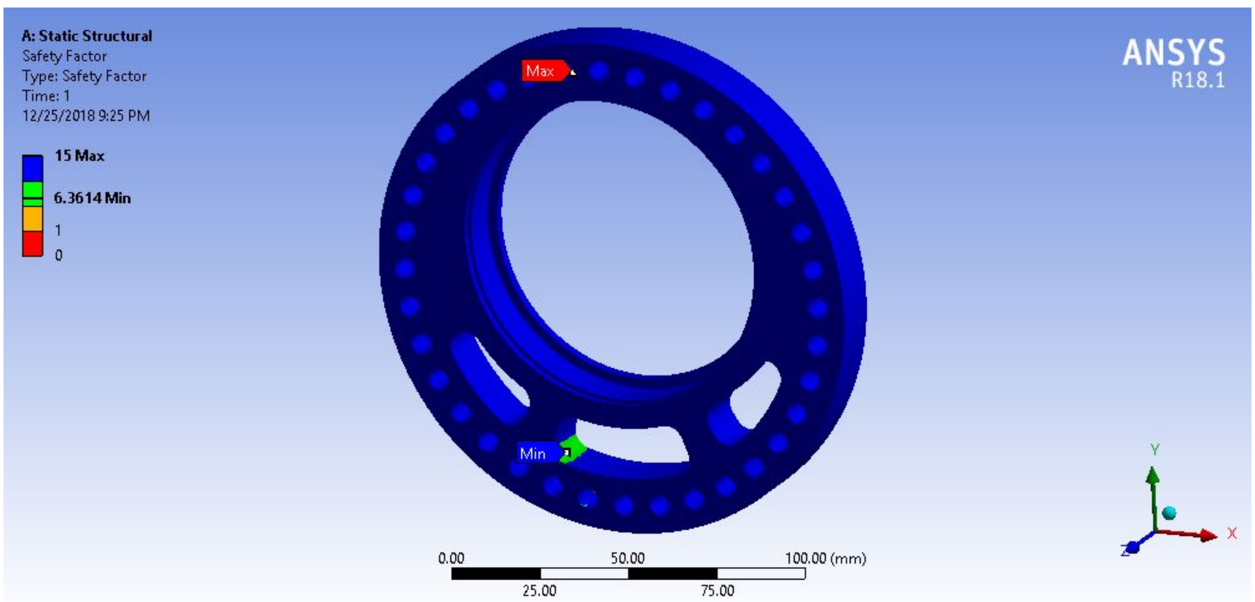
Chain breaking load von Mises stress



Chain breaking load (FoS = 1.89)



Chain breaking load von Mises stress



Chain breaking load von Mises stress

5.5 Bearing selection and installation

5.5.1 Selection parameters

Bearing selection was a part of the differential mount designing phase. In order to select the bearings, the main parameters taken into account were its nominal diameters (ID and OD), load rating, RPM rating and basic rating life. Temperature and lubrication were not critical parameters since the bearings would be operating in a normal open environment not subjected to extreme temperatures and proper timely lubrication would ensure their smooth operation. Sealed bearings seemed a better option since the open environment might lead to contamination due to dirt ingress, reducing the bearing life. The selection was partly eased due to the fact that since the bearings were to be press-fitted onto the differential, their nominal IDs were already known. We just needed to determine the optimum ODs and bearing width.

Since holes in the eccentric disks serve as housings for the bearings, it was important to:
1. Decide what hole diameter to be designed
2. Whether or not a ball bearing of that nominal diameter would be available to fit in the designd hole
3. Whether or not the disk would be strong enough by designing a hole of appropriate diameter so as to house a bearing

This analysis was important since it might happen that after designing a hole of a particular diameter and carrying out the structural simulations, it was found in the bearing selection phase that no bearing was available to fit in that particular hole, i.e., the hole was wrongly designed. This would lead to the changes in the disk design altogether, consuming time and delaying the manufacturing process.

5.5.2 Type of bearings

Based on the loads (on the differential mounts) already calculated, it was obvious that radial loads were most common, with a slight possibility of axial loads during cornering. Thus, it was decided to use deep groove ball bearings suitable for this purpose. In the setup, the outer race of the bearings will be fixed while only the inner race will rotate.

5.5.3 Desired load and speed capabilities

Based on the loads (on the differential mounts) already calculated and assuming that the possibility of a chain breaking force acting on the sprocket (and hence, the mount) would be very low, we considered the loading cases 2 and 3, which resulted in the maximum radial load of about 8kN acting on the bearings in a fixed direction (i.e., the bearing's inner race will rotate but the load will not rotate with it).

For the desired speed rating, the engine speed was assumed to be around 12000 RPM in the 4th gear, resulting in the rear sprocket speed of about 1300 RPM. Knowing that the vehicle would operate quite less at such high engine RPM, this bearing speed rating would suffice our need.

5.5.4 Selection

Based on the above requirements, appropriate bearings for the left and right sides were sorted out and the following were selected:

16011	[d:55 mm B:11 mm D:90 mm C:20.3 kN C_0:14 kN]
6011	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011 M	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011 N	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011 NR	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011-2RS1	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011-2Z	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011-RS1	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]
6011-Z	[d:55 mm D:90 mm B:18 mm C:29.6 kN C_0:21.2 kN]

Left bearing

For the left side: Bearing no. *16011* (open-bearing)

Nominal ID = 55
Nominal OD = 90
Width = 11
Dynamic load rating C = 20.3 kN
Mechanical limiting speed = 10000 RPM

To calculate the bearing life, following calculations were done:
Equivalent dynamic bearing load P = 8 kN
Basic rating life (at 90% reliability) = $\left(\frac{C}{P}\right)^3$ (in millions of revolutions) = 16 million revolutions (or 205 hours continuous operation @ 1300 RPM)

61910	[d:50 mm B:12 mm D:72 mm C:14.6 kN C_0:11.8 kN]
61910-2RS1	[d:50 mm D:72 mm B:12 mm C:14.6 kN C_0:11.8 kN]
61910-2RZ	[d:50 mm D:72 mm B:12 mm C:14.6 kN C_0:11.8 kN]
W 61910	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]
W 61910 R	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]
W 61910 R-2Z	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]
W 61910-2RS1	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]
W 61910-2RZ	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]
W 61910-2Z	[d:50 mm D:72 mm B:12 mm C:12.5 kN C_0:11.6 kN]

Right bearing

For the right side: Bearing no. *61910-2RS1* (rubber-sealed on both sides with normal internal clearance)

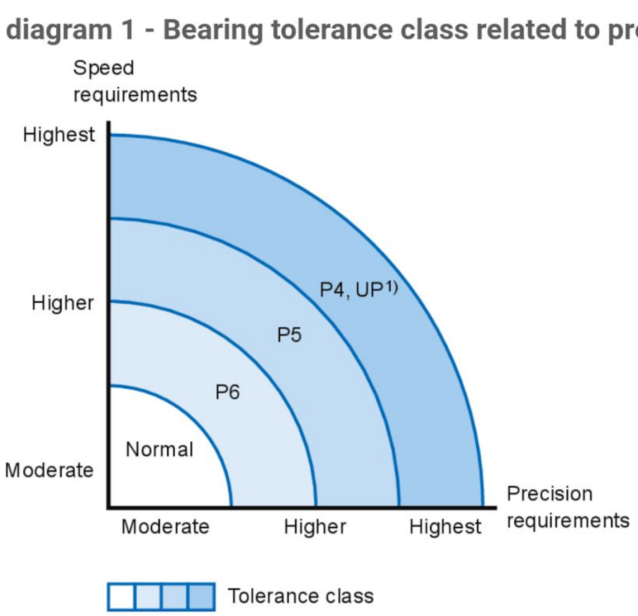
Nominal ID = 50
Nominal OD = 72
Width = 12
Dynamic load rating C = 14.6 kN
Mechanical limiting speed = 5600 RPM

To calculate the bearing life, following calculations were done:
Equivalent dynamic bearing load P = 2.5 kN
Basic rating life (at 90% reliability) = $\left(\frac{C}{P}\right)^3$ (in millions of revolutions) = 200 million revolutions (or 2564 hours continuous operation @ 1300 RPM)

5.5.5 Tolerances for interference fitting

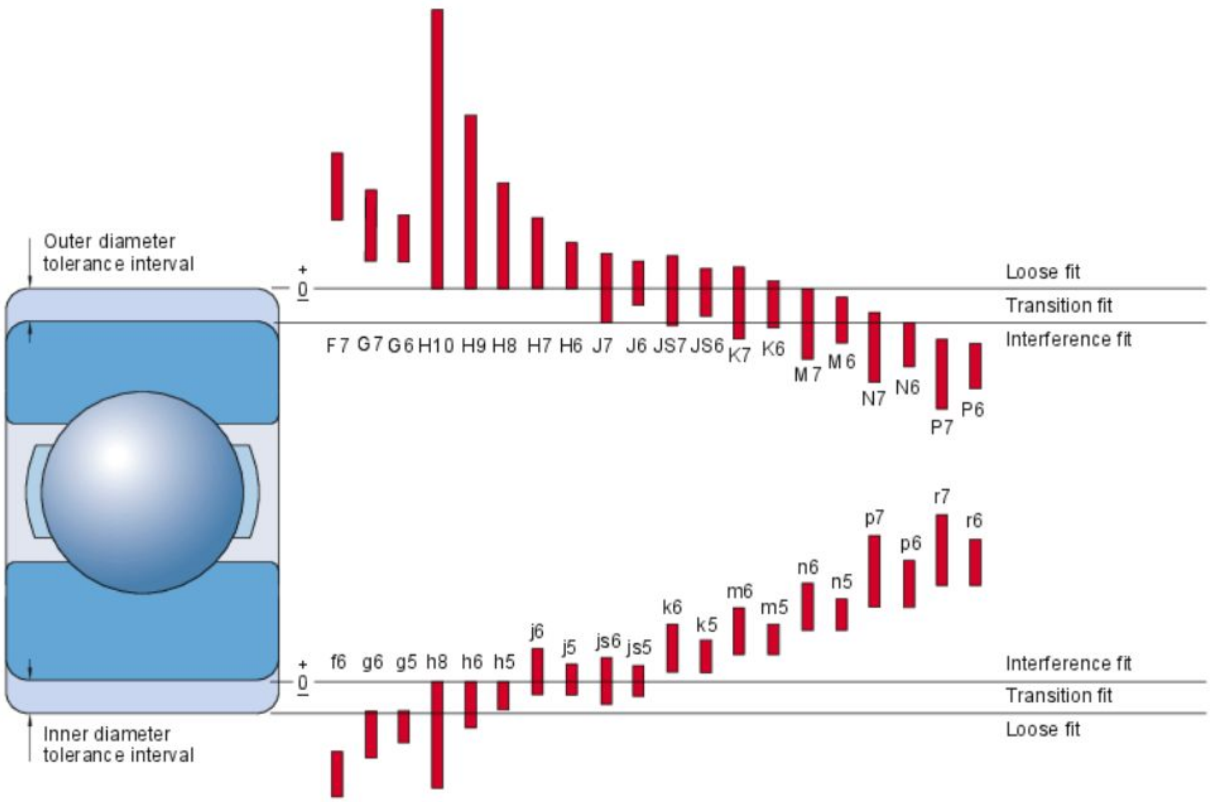
As aforementioned, the bearings were to be press-fitted onto the differential and into the eccentric disk mechanically. To do so, tolerances had to be determined for the bearing diameters, i.e., by what margin the bearing ID would be lesser than the differential's surface diameter and by what margin the bearing OD would be more than the eccentric disk's housing diameter. This was important since:
1. If loosely fit, the disk-bearing surface could come apart under the application of a slight axial load during operation
2. Too high a tolerance would result in the reduction of the clearance between the bearing balls and the races. This would stress the bearings during operation resulting in spalling failure.

Appropriate press-fit tolerance ranges can be found out on the SKF website based on the ISO 286-2 standard which takes into account the housing and shaft nominal diameter range and the tolerance class.



Bearing tolerance class

Fig. 1 - Position and width of shaft and housing tolerance classes



Tolerance classes and types of fit

For our analysis, we considered a P6/n6 tolerance class combination to ensure a reliable press-fitting, though negligible variation in results was observed with other tolerance class combinations. The following results were obtained using an excel spreadsheet for handy calculations:

	left side	right side
diff outer dia. (mm)	55	50
disk bearing housing dia. (as per CAD model) (mm)	89.95	71.94

bearing ID

		left side	right side
nominal dia.(mm)		55	50
tolerance (mm)	min	0	0
	max	0.054	0.054
TRUE ID (mm)	min	54.946	49.946
	max	55	50
theoretical interference (µm)	min	0	0
	max	54	54
probable interference (µm)	min	4	4
	max	50	50

bearing OD

		left side	right side
nominal dia. (mm)		90	72
tolerance (mm)	min	0	0
	max	0.015	0.013
TRUE OD (mm)	min	90	72
	max	90.015	72.013
theoretical interference (μm)	min	50	60
	max	65	73
probable interference (μm)	min	54	64
	max	61	69

6. Sprocket adapter

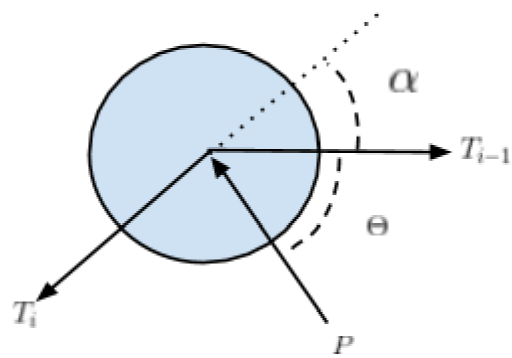
6.1 Need of an adapter

Since the design of first car KX-1, Team KART is using Quaife Limited Slip Differential which didn't require an adapter, instead, the sprocket was directly bolted to the differential. With the introduction of Drexler LSD, we required an adapter with a spline similar to the one on the differential. One of the first proposed designs was to embed the spline in the sprocket itself but considering the difficulty of manufacturing this design was switched to a splined adapter which would be cheaper and easier to manufacture.

6.2 Calculations

To get started with analysis first the forces on the adapter had to be calculated. These forces mainly came through the bolting points, weight of the adapter “W” and pseudo force during acceleration, retardation and cornering. The forces coming through bolting points are mainly due to torque given by the engine “M”, weight of the sprocket “W_{sp}” and due to tension in chain “T”.

In this chain sprocket drive the front sprocket which is driven by the engine drives the rear sprocket. In any chain sprocket drive consisting of two sprockets there is a tight side of chain with tension “T_t” and slack side of chain with tension “T_s”. For an “i^{th}” link of a chain in contact with sprocket.

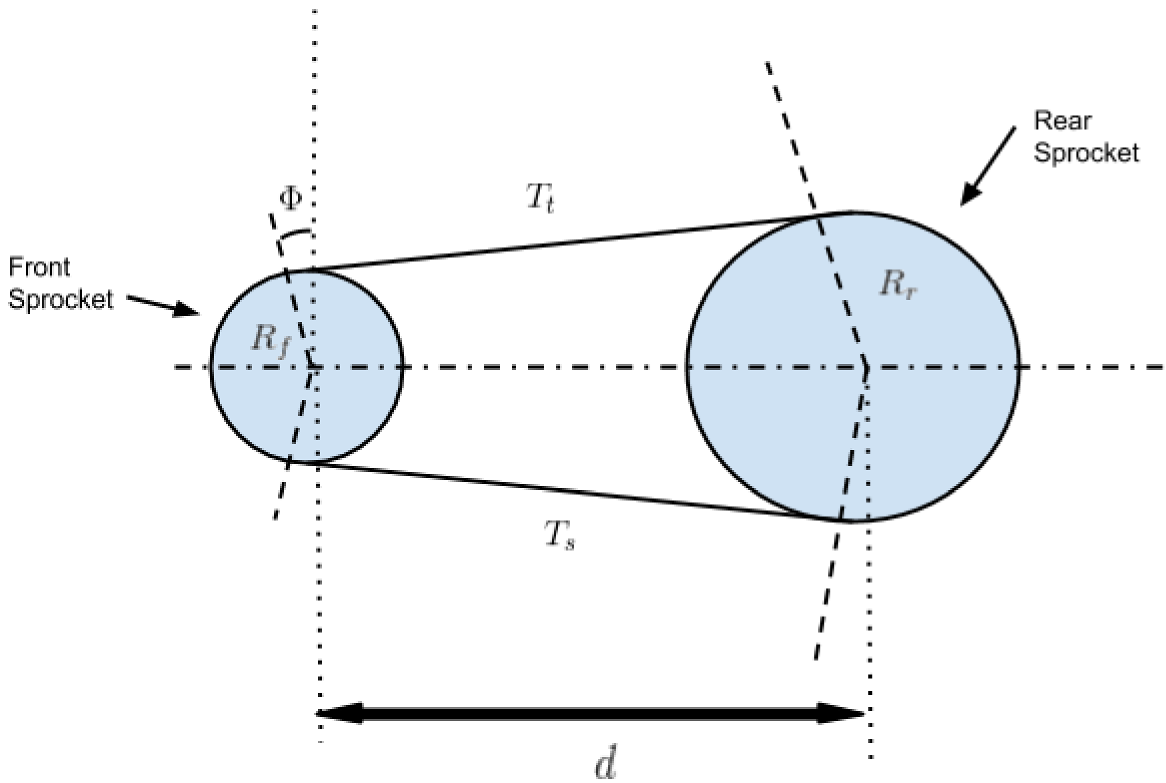


$$T_i = T_{i-1} sin\Theta / sin(\Theta + \alpha) \quad ......(1)$$

Similarly for “n^{th}” link,

$$T_s = T_t (sin\Theta / sin(\Theta + \alpha))^{n-1} \quad ......(2)$$

where Θ is articulation angle and α is the pressure angle.



Pitch radius of rear sprocket	R_r	**113.8mm**
Pitch radius of front sprocket	R_f	**35.6mm**
Distance between sprockets	d	**190.79mm**(approx.)
Front sprocket articulation angle	Θ_f	**24deg**
Rear sprocket articulation angle	Θ_r	**8deg**
Front sprocket pressure angle	α_f	**26.15deg**
Rear sprocket pressure angle	α_r	**17.55deg**
Front sprocket teeth	n_f	**15**
Rear sprocket teeth	n_r	**45**
	Φ	**38deg**

Considering forces on front sprocket -

$$M = R_f \times (T_t - T_s) = 35 \text{ Nm} \qquad ..(3)$$

Considering rear sprocket -

$$\tau = R_s \times (T_t' - T_s') \quad \text{....(4)}$$

where τ is torque on rear sprocket and hence on the adapter.

Considering the adapter

$$F_t = \tau / r$$

$$F_x = (T_s + T_t) \times cos\Phi \quad \text{..(5)}$$

$$F_y = ((T_t - T_s) \times sin\Phi + W_{sp}) \quad \text{...(6)}$$

r = distance of bolting points from the centre

F_t =tangential force due to torque on the bolting points

W_{sp} = weight of sprocket=15 N(approx.)

F_x and F_y are the forces in forward and vertically downward directions respectively

Among the above variables number of front sprocket teeth are fixed as the stock sprocket is being used with 15 teeth. The number of teeth of rear sprocket was simulated by using Optimum Lap as already detailed earlier, resulting in a 45 teeth rear sprocket.

Since, stock sprocket is being used so is the stock chain and thus the pitch diameter of the rear sprocket can be obtained using the formula

$$R_r = P/(2 \times sin(180/n_r))$$

where P= chain pitch and R_r = pitch radius of rear sprocket. Similar formula can be used for front sprocket

$$R_f = P/(2 \times sin(180/n_f))$$

Articulation angle for a sprocket can be obtained as

$$\Theta = 360/n$$

Pressure angle can be obtained through sprocket geometry and the distance between the two sprockets is obtained from the positions of engine and differential box in CAD of the car.

Considering all the variables obtained now we need to calculate forces on adapter for proceeding with the simulations. The forces are calculated as 6 different cases. The cases cover all the limiting force loading that the adapter will face while going round the track.

6.3 Types of loading on the adapter

6.3.1 Normal loading

This covers the phase when car is moving with a constant velocity on a straight i.e. it does not involve any acceleration components be it lateral or longitudinal. As discussed above we use following equations

$$T_t = T_t' \qquad \ldots(7)$$

From equation (2)

$$M = T_t \times (1 - (sin\Theta_f / sin(\Theta_f + \alpha_f))^{14}) \times R_f$$

$$\tau = T_t' \times (1 - (sin\Theta_f / sin(\Theta_f + \alpha_f))^{48}) \times R_r$$

but calculations show us that

$$T_s' \to 0$$

therefore assuming

$$T_s = T_s' = 0$$

we get

$$\tau = T_t \times R_r = M \times R_r / R_f$$

$$F_t = \tau / r = 1517 \text{ N}$$

$$F_x = 874 \text{ N}$$

$$F_y = 292 \text{ N}$$

6.3.2 Shock loading

Whenever the clutch is engaged or disengaged an impulse or shock is felt on the drivetrain. This shock travels through the chain to the sprocket and hence to the adapter. The force loading experienced by the adapter is covered in this case. Usually in this case the torque on the front sprocket is taken to be 5 times of max torque provided by the engine that is taken into consideration in normal loading. Rest of the calculations and equations being same as normal loading.

$$F_t = 7585N$$

$$F_x = 4368N$$

$$F_y = 1408N$$

6.3.3 Max. longitudinal acceleration

This case includes the situation when the car is accelerating with its maximum possible acceleration. We obtain the maximum value of acceleration through Optimum Lap simulation data which comes out to be $a_{max.}$= 6.18 m/s^2. Thus for simulation we apply acceleration a_{max} on the adapter in the forward direction, rest of the force values are kept same as in normal loading. It's important to note that we are still assuming.

$$T_t = T_t'$$

6.3.4 Max. longitudinal deceleration

Similar to previous case this case covers the part when car experiences maximum retardation as in case of braking at the end of a straight. This maximum retardation value is obtained from Optimum Lap simulation data r_{max} = 15.03 m/s^2. Similar simulations are performed as in case of max acceleration only here we apply the acceleration r_{max} in the opposite direction considering all the assumptions.

6.3.5 Max. lateral acceleration

Whenever car goes around a corner it experiences a centripetal acceleration. The maximum centripetal acceleration experienced by the car throughout the track was calculated through Optimum Lap simulation data, and it came out to be a_c = 14.73 m/s^2. Here we consider that the driver doesn't accelerate while going round the corner and car moves at constant speed. Thus for simulations the acceleration a_c is applied on the adapter along the Z-axis, keeping rest of the bolting point forces same as normal loading.

6.3.6 Chain breaking force

This case includes the forces experienced by the adapter just before the chain breaks. This is not so common phenomenon because FOS of chains are usually high as they are designed to bear high amount of loads. But for the safety purposes, we take this case into consideration. On the verge of breaking chain experiences a limiting tension in our case T= 36.6 kN. This tension acts on the rear sprocket and thus we take

$$T_t = 36.6kN$$

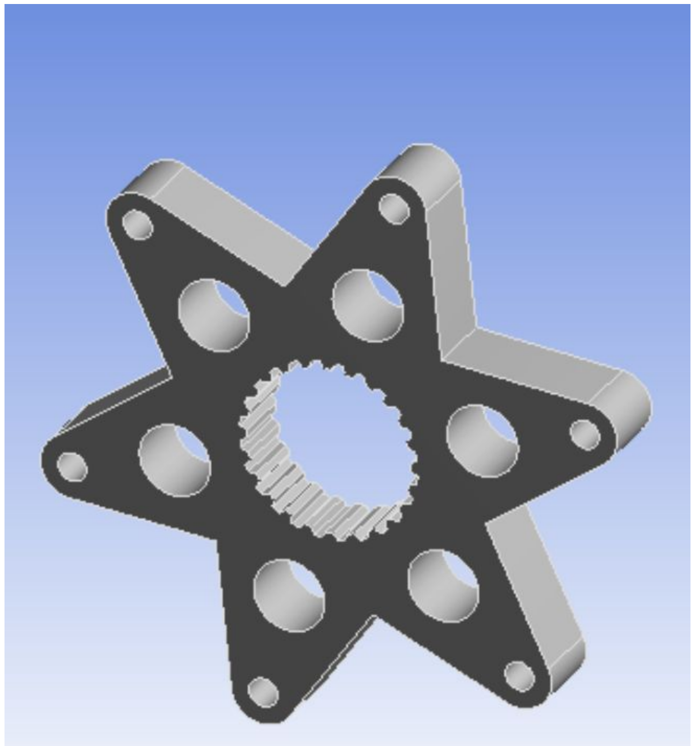
$$T_s \sim 0$$

$$F_t = 60027N$$

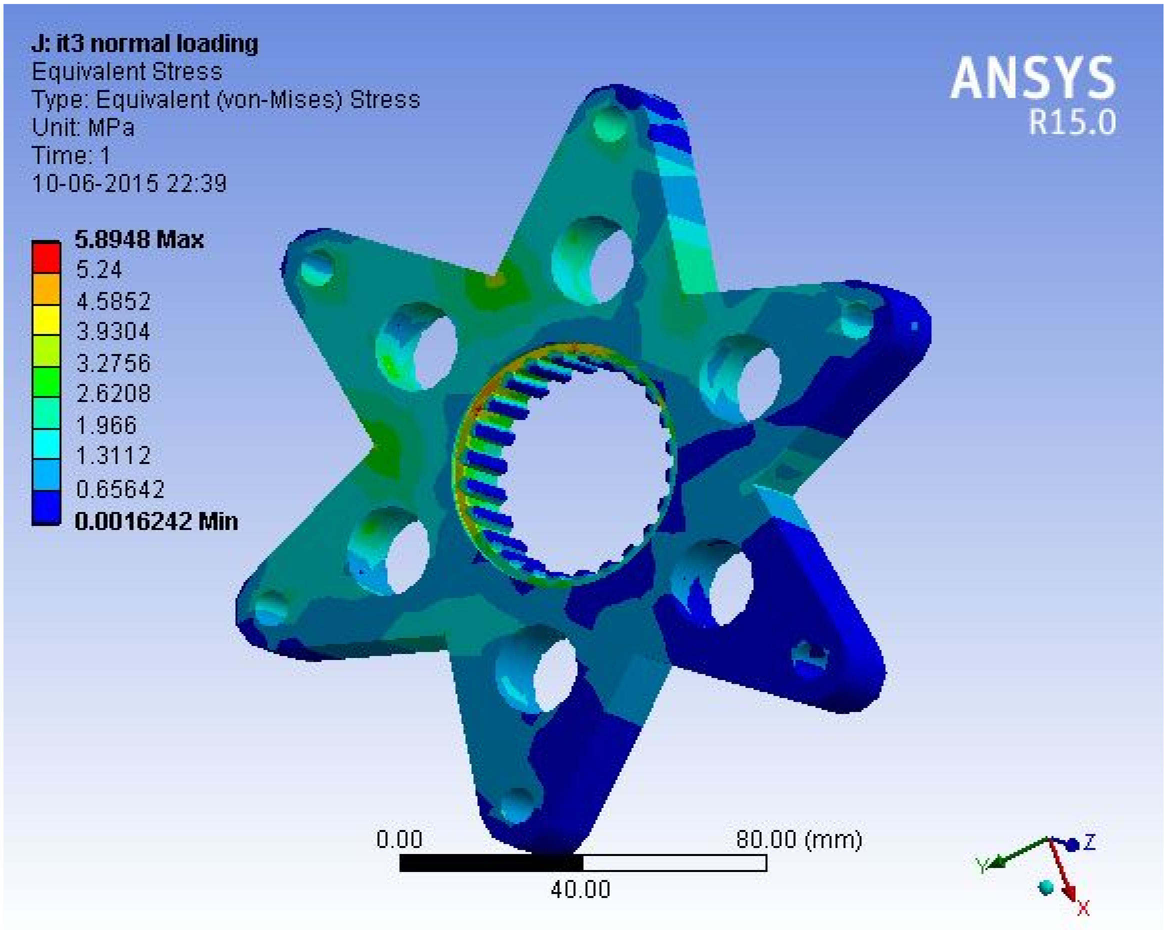
$$F_x = 34582N$$

$$F_y = 11049N$$

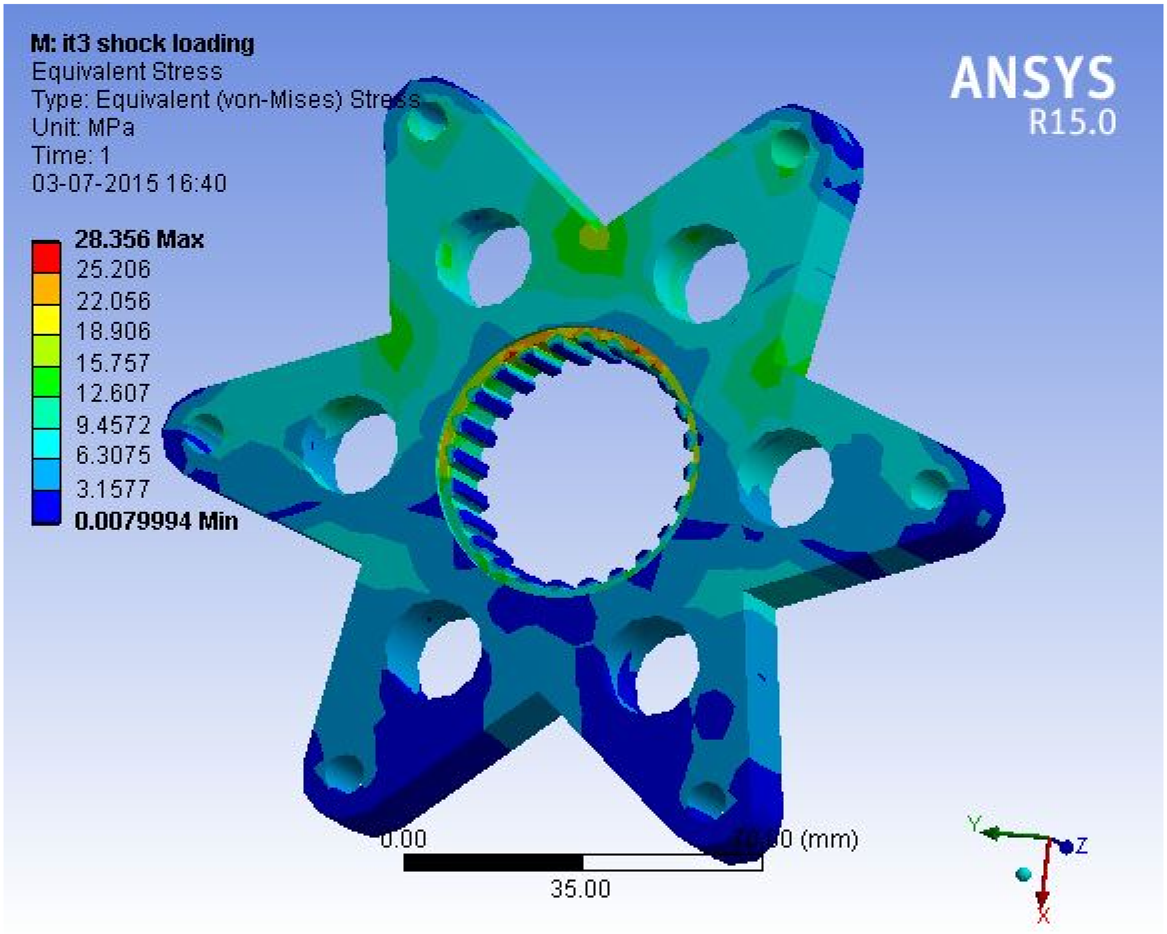
6.4 ANSYS static structural simulation



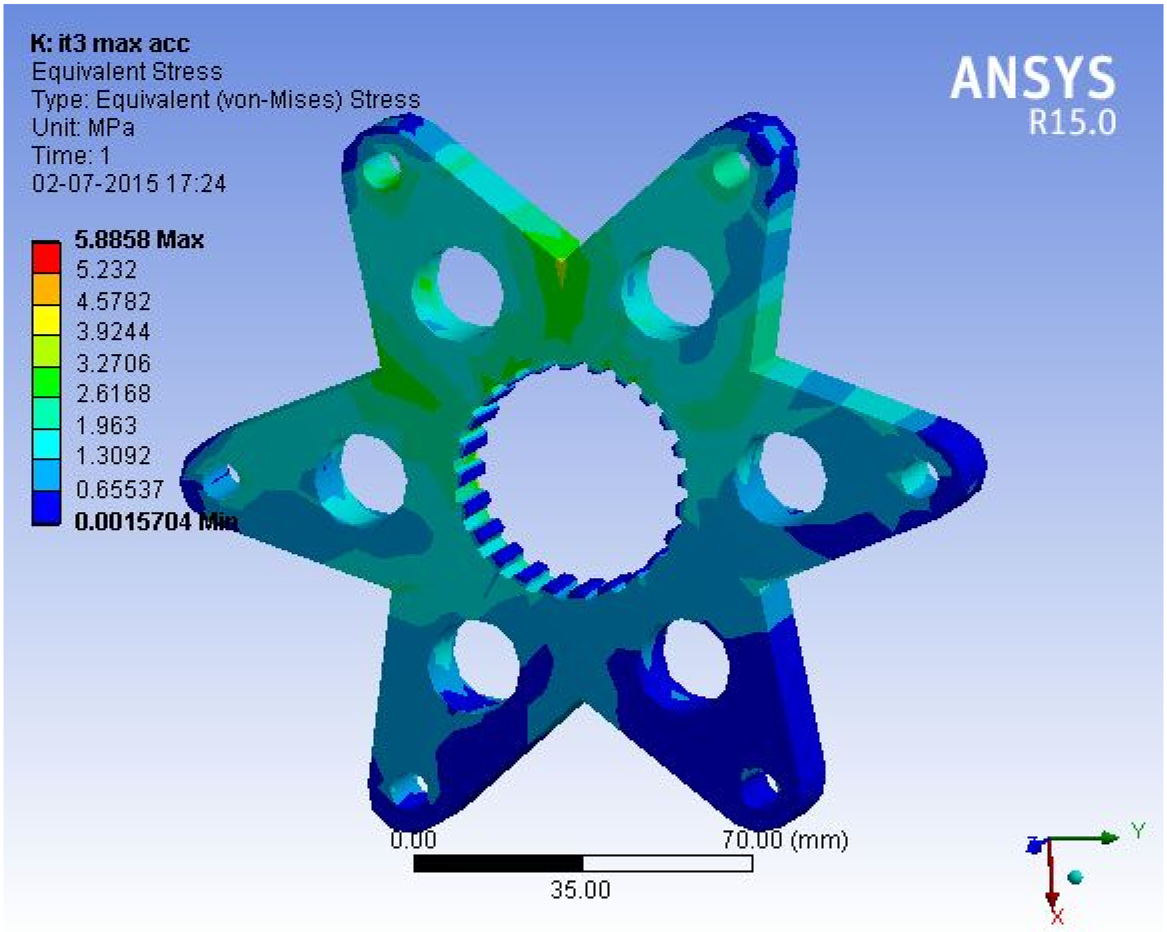
The loading for the above model was as follows:
1. The central spline was replaced by an equivalent and rigid body element with all the six degrees of freedom restrained. The constraint, being fixed support was applied on to this rigid body element.
2. The chain force was equivalently applied with equal distribution on these 6 rigid body elements representing the bolting holes. The equivalent torque was applied at the rigid body element representing the spline.



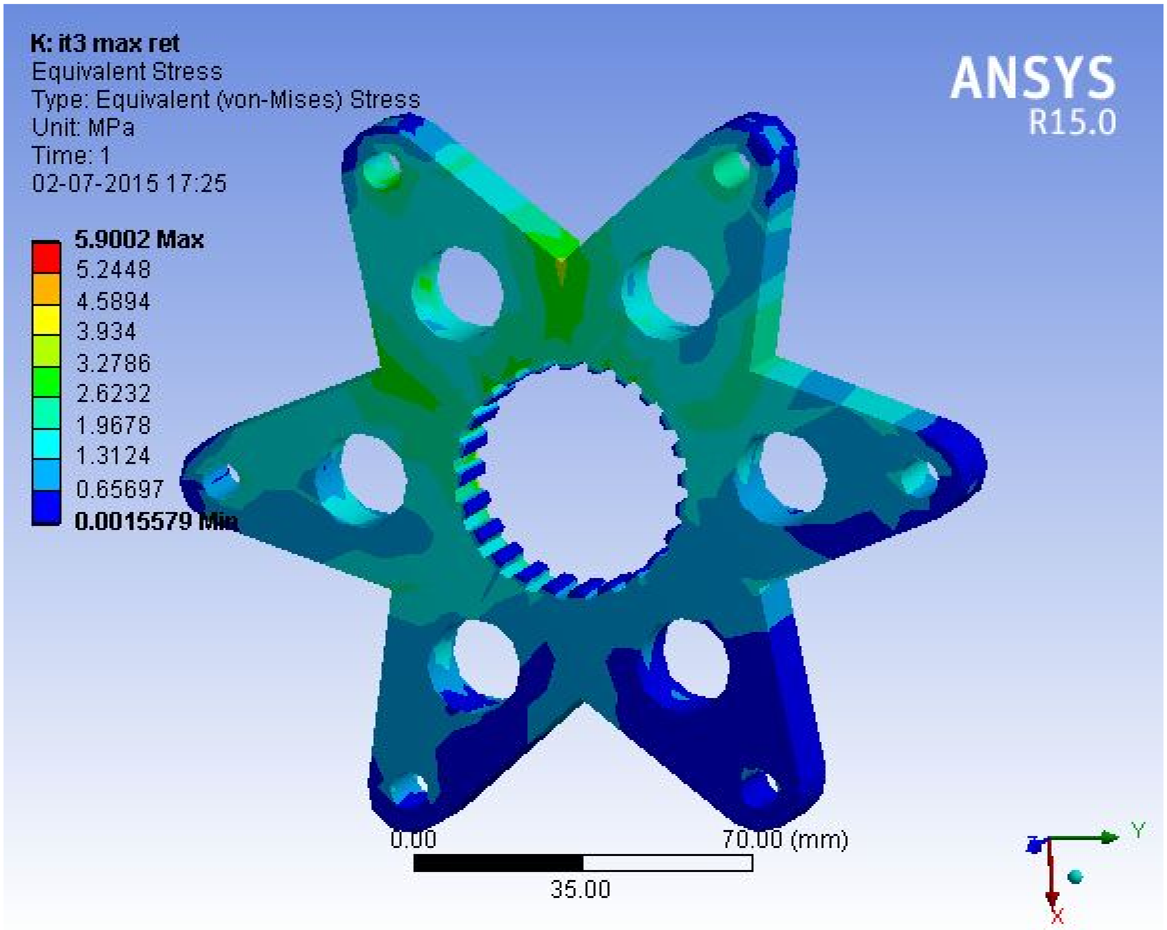
Normal Loading (FoS=85.33)



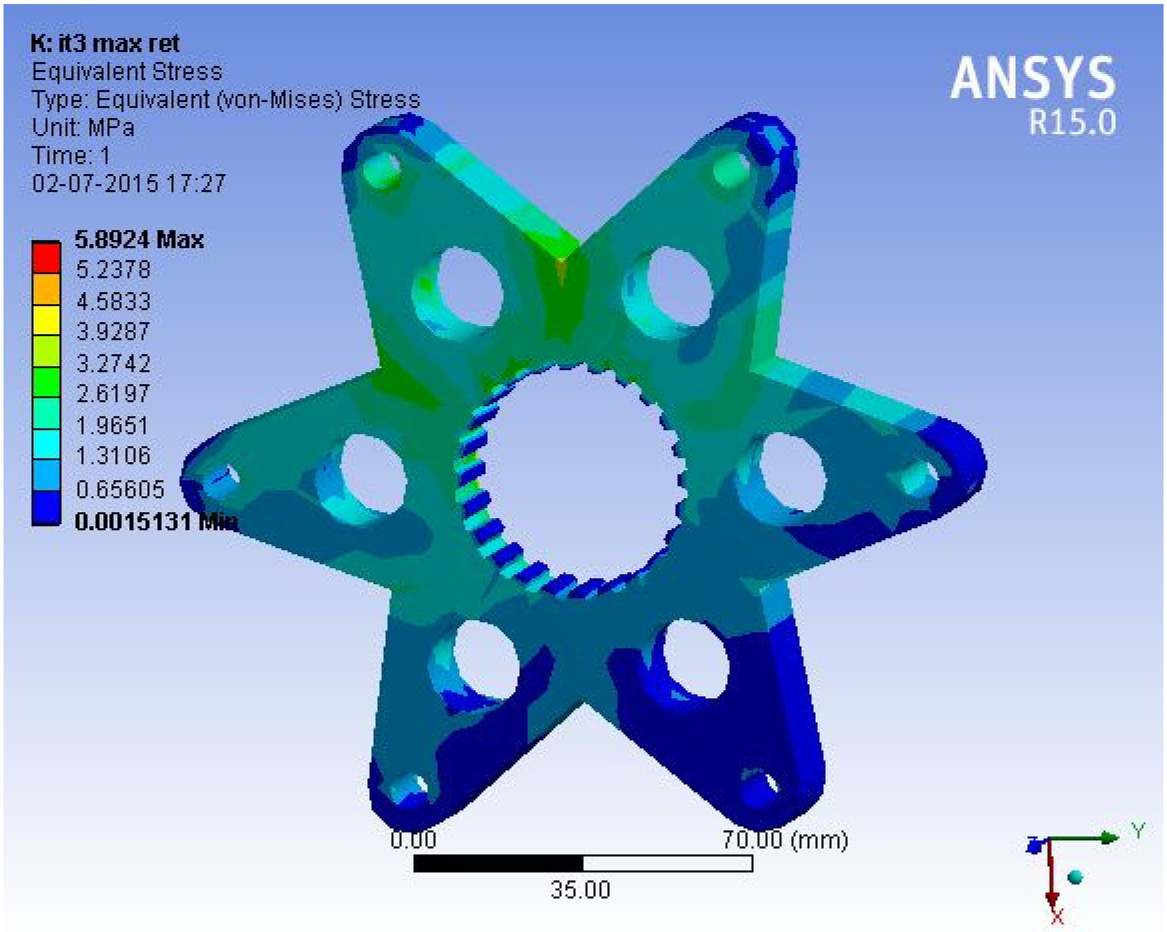
Shock Loading (FoS=17.74)



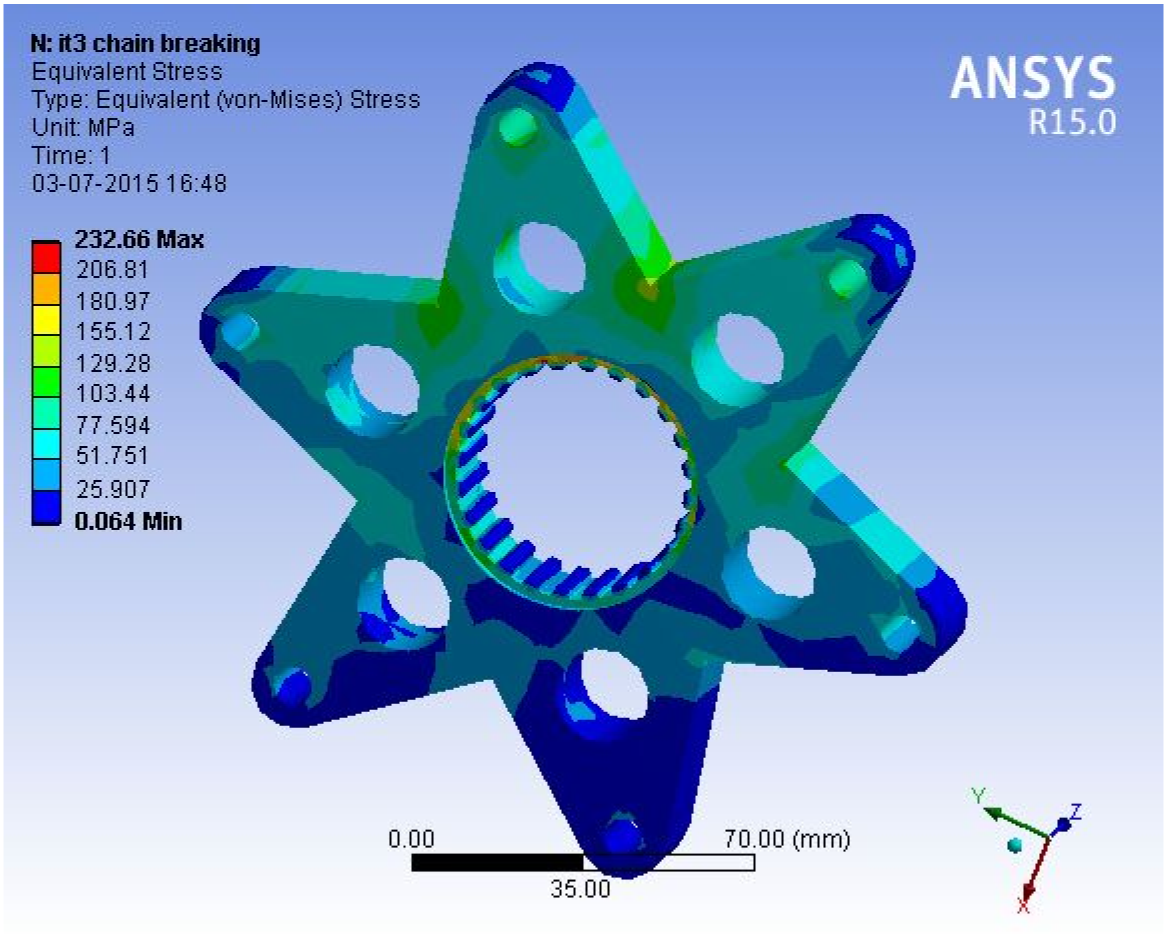
Maximum Acceleration (FoS=85.46)



Maximum deceleration (FoS=85.25)

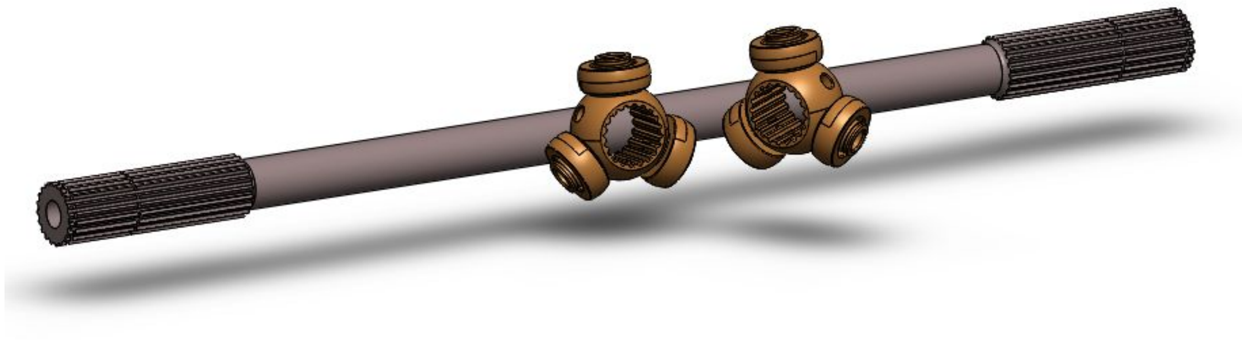


Maximum cornering force (FoS=85.36)



Chain breaking limit (FoS=2.16)

7. Half shafts



The half shafts were made from EN-24/AISI 4340. The reason for choosing AISI 4340 is that it has high shear strength and since ductile materials like steel are weaker in shear than in tension, it was important to choose a material having a high shear strength to prevent the failure (only shear stress due to engine torque was considered during the design and any axial load was neglected since it would be too small to have any effect). The splines were traced from the CV joint tripods and nitrided to 45 HRC (Rockwell Scale using a 120° sphero-conical diamond indenter) to prevent failure at splines. The half shaft was torsion tested and it could withstand 1.5 times the maximum torque until it failed at the splines. The CV joint used is tripod type.