Final Project

Master of Engineering in Mechanical and Aerospace Engineering

Design of a Gearbox for an electric FSAE vehicle

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Abstract

The main objective of this project is to design a new gearbox for an electric Formula SAE car. The last car had one motor for each wheel, and for the new car, only one motor will be used to give traction to the rear wheels.

The gear selection has been done by using the Optimum Lap software. Which after giving the main parameters of the car as well as the geometry of the circuit, returns the data needed to design the powertrain system, among others.

A comparison haas been done between the different transmission systems available. After deciding that the use of a gearbox is the best option, another comparison has been done to choose the type of gears. To do so three different basic designs have been done with CATIAV5. The selected option is to use three stage spur gears.

To design the tooth form, materials and shaft-hub conection selection, the KISSSOFT software has been used. To optimize the different parts after designing them with CATIAV5, the parts have been meshed with HyperMesh and processed in Abaqus.

The final design weights less than 6kg and contains all the parts needed inside of the gearbox. The connections to the motor or the wheels have not been designed.

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1. Introduction

In this project, a Gearbox for an electric Formula SAE Vehicle has been designed. This one will be used in the IIT Motorsports for the 2016 competitions. I have already been a member of a Formula SAE team for two years. One of the years in charge of the transmission, which was a belt-pulley design.

The motor selection as well as all the electric part of the powertrain is already defined. The speed, torque and power are given as information. The dimensions of the wheels and tires were also selected before the design.

The aim is to compare different designs and find the one that gives the vehicle the better performance. As in all the motorsports competitions, the main objectives are to reduce weight, improve the reliability and control the costs.

2. Transmission Systems

2.1. Gearbox

One of the common designs is the use of a gearbox to provide the speed ratio desired. This is probably one of the most reliable transmission systems although, depending on the materials used and the design, it can also be the heaviest. It is usually more expensive than other solutions. Another disadvantage could be that it needs lubrication. On the other hand, it does not require the axis to be parallel and it is probably the most compact transmission system.

2.2. Pulleys and belt

The small pulley (driver) is connected to the motor axle whereas the big pulley (driven) is fixed to the tractive wheels' axle. The belt can be synchronous or asynchronous but the first are usually preferred because they allow a better traction control.

A very interesting advantage of this system is that it does not require lubrication, although the belt needs to be tensioned and is a little less reliable than a gearbox. It usually needs more space to achieve the same speed ratio.

2.3. Chain

This system is very similar to the belt system but it uses a chain instead of a belt. This implies that the rotational transmission will always be synchronous. It usually needs less space than the classic pulleys-belt system because it does not need to be tensioned so much (although it does need to be tensioned). It also needs some lubrication. It can be considered quite reliable, providing that a chain is usually less likely to break than a belt.

2.4. Transmission system election

One of the most important benefits of an electric vehicle is to recover energy during the breaking. The Formula in which this gearbox will be used is going to have regenerative breaking. In order to recover the energy while breaking, the powertrain must be prepared. The gearbox and the pulleys with belts, both allow its use. In the other hand, the use of a chain will not allow to recover the energy, and all the energy would be converted to thermal energy and loosed. Since the weight of the batteries is really important in the motorsports car, and the efficiency is also valued in the competition. The use of a chain as transmission system is discarded.

The use of belts would weight a little bit less than a gearbox, around 2 kilograms. But it has some disadvantages. The precision needed in order to work properly is much more difficult to achieve, just a few degrees will reduce the effective contact area of the belt and produce problems. The use of a gearbox can make the car weight a few more kilograms, but it will be more reliable and compact. Its volume it is also important, since it will be held in a reduced space.

For this reasons, the transmission system that has been designed is a gearbox.

3. Gearbox Ratio

Before choosing the transmission system that will be used, knowing the gearbox ratio is fundamental.

To choose it, a maximum speed calculation is done, since the maximum motor speed is known, as well as the wheel diameter. To get a reference for the maximum velocity wanted, a research on previous competitions is done. The range is between 100-110 km/h.

The data needed is the following:

Motor Maximum Velocity (w)	3200 r/min
Wheel Radius (r)	0.223 m
Differential Ratio	1
Mass (with driver)	350 kg
Driven Type	2WD
SCL	-1.5
SCD	2.78
Air density	1.225 kg/m^3
Tire radius	0.223 m
Final Drive Ratio	2.3

Table 1, Vehicle Parameters

Now the desired ratio can be found by isolating some formulas.

$$w = \frac{3200r}{min} \times \frac{1min}{60s} \times \frac{2\pi}{1r} = 335.10\, rad/s$$

$$v = 120\frac{km}{h} \times \frac{1000m}{1km} \times \frac{1h}{3600s} = 33.33m/s$$

$$w \times \tau \times r = v$$

$$\tau = \frac{v}{w \times r} = 0.43$$

$$i = \frac{1}{\tau} = 2.3$$

To prove that the chosen ratio is convenient, another study is done. This time it will not just limit the maximum speed but simulate a lap on a track. Using the Optimum Lap simulator, introducing some basic parameters of the car and a track geometry, some crucial information would be obtained.

More data of the tires is also given as well as the engine speed vs engine torque data.

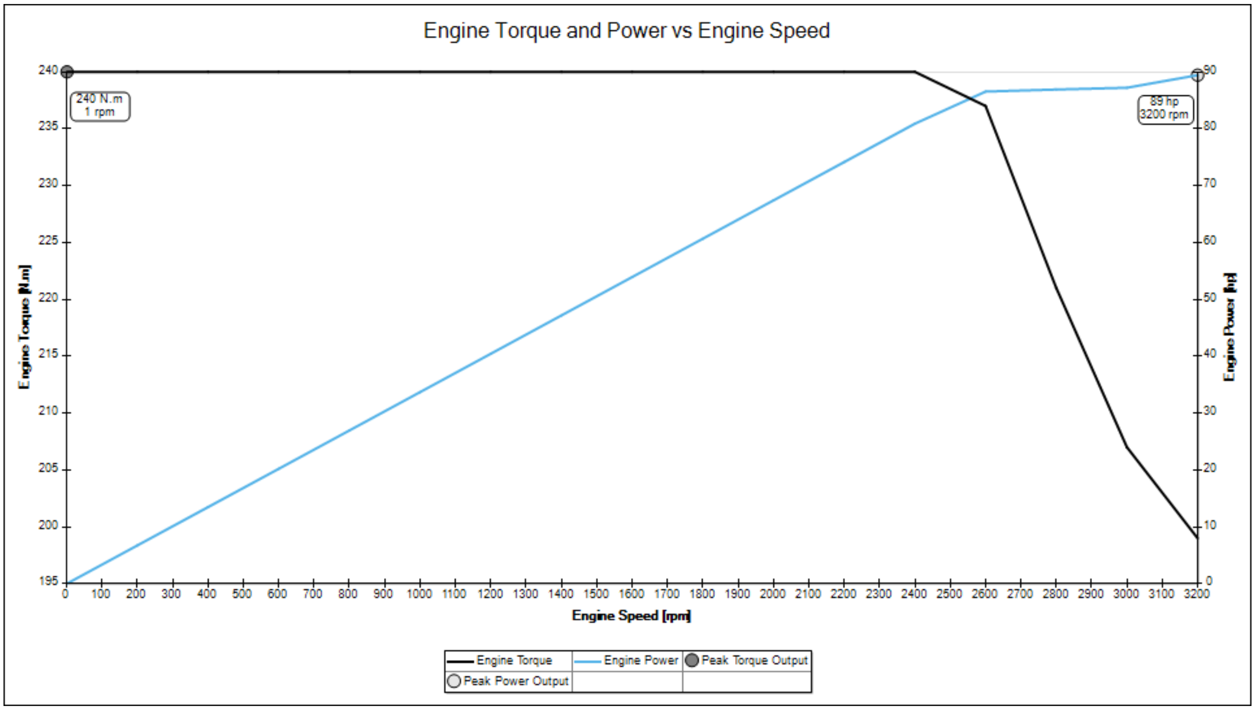

Figure 1, Engine data

The track used in this simulation is one of the Formula SAE, the Formula SAE Michigan Endurance Track.

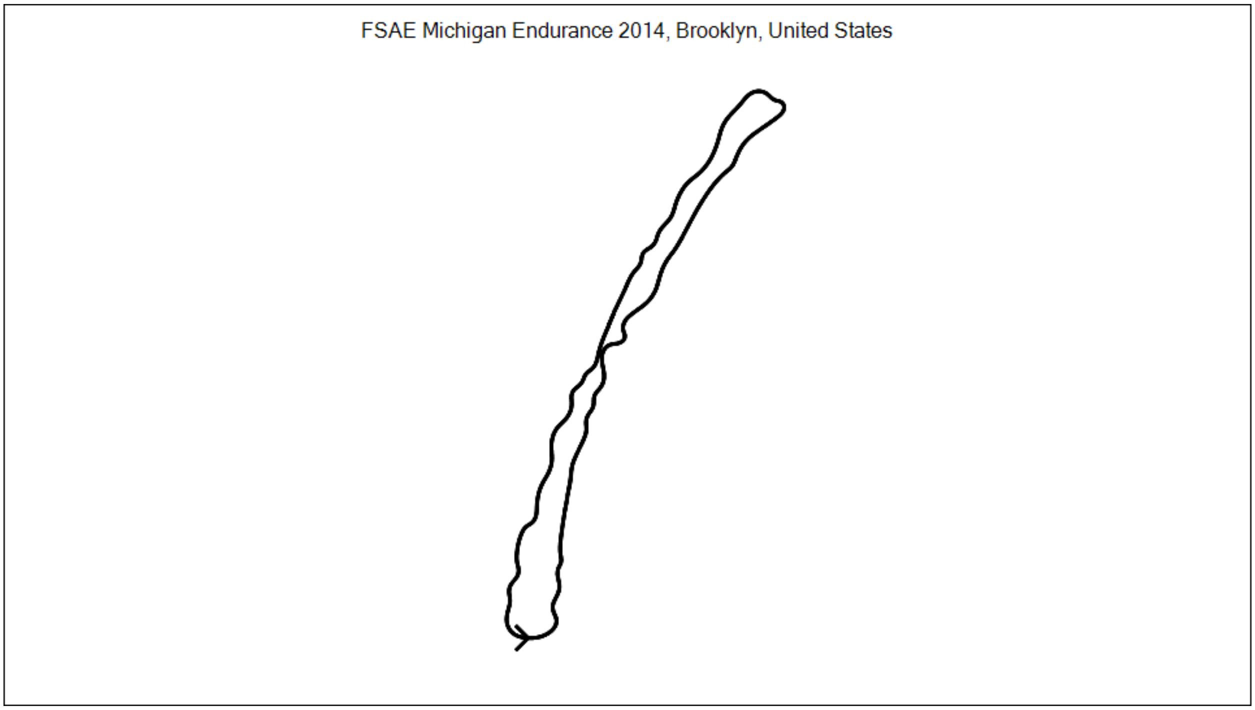

Figure 2, FSAE Michigan Track

From the simulation a group of different graphs are given.

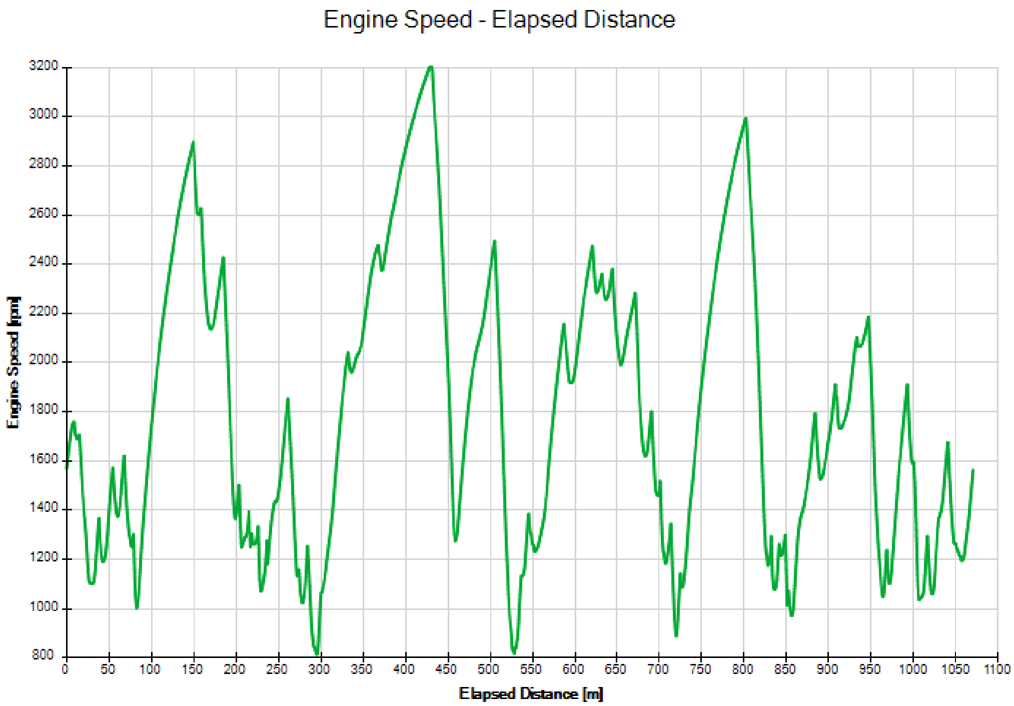

Figure 3, Engine Speed vs Elapsed Distance

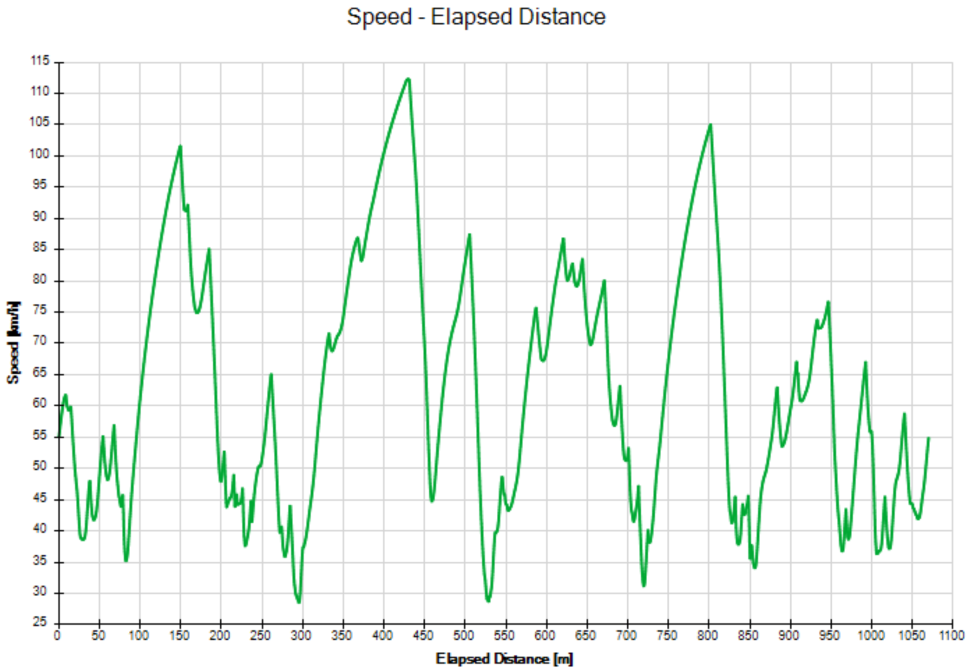

Figure 4, Speed vs Elapsed Distance

From this two graphs we can see that the maximum velocity of the motor is not exceeded. Also the maximum longitudinal speed of the vehicle is in the range speculated.

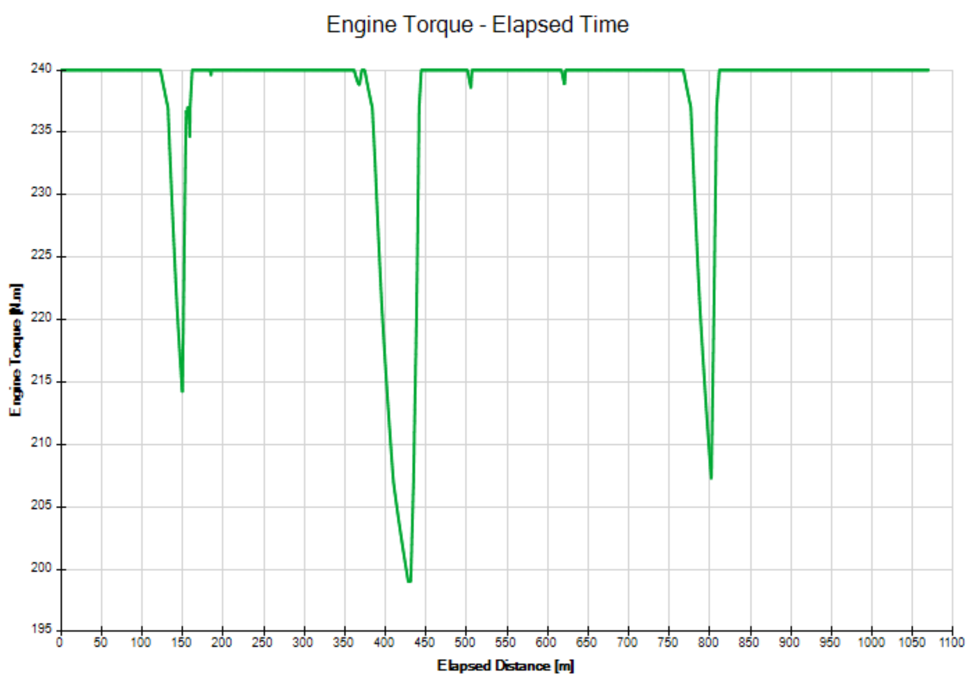

Figure 5, Engine Torque vs Elapsed Time

In this last graph we can see that the maximum torque given is also take in to account. We can also see that the torque is at its maximum for most of the lap. This is not true, since the tires cannot achieve such longitudinal force and would slip, and the traction control will enter in action. But since it is just a simulation to see the behavior of the gear ratio, no further analysis are done.

4. Stage Selection

Once decided that the transmission system that best fits the requirements is a gearbox. Now the number of stages will be discussed, as well as the kind of gears used. Three different models are considered: one stage with spur gears, two stages with spur gears and one stage with bevel gears. One simplified design for each model is done. The designs will just have the following parts: gears, shafts, bearings and housing. We are looking for reduced weight and reduced dimensions. The efficiency of all them is considered approximately the same. No bolts, oil, tripod housing, or complex designs to reduce weight for the gears are considered in this analysis.

The data required to do the designs is the gear ratio that was calculated before. The distance between the axles is also needed. In order to know the distance, the dimensions of the motor and the differential are considered. The distance is going to be the sum of their radius plus a safety factor. The radius of the cylindrical motor is 114 mm and the differential radius is 50 mm. The safety factor is going to be a 10%. The distance there will be $a = 1.1 \times (114 + 50) = 180\ mm$

4.1. One Stage Spur Gears

For this design two different steel gears, two equal steel shafts, four equal bearings and an aluminum housing are considered. The first gear has a diameter of 120 mm and the second one a diameter of 280 mm. The total weight of the assembly is of 14.7 kg.

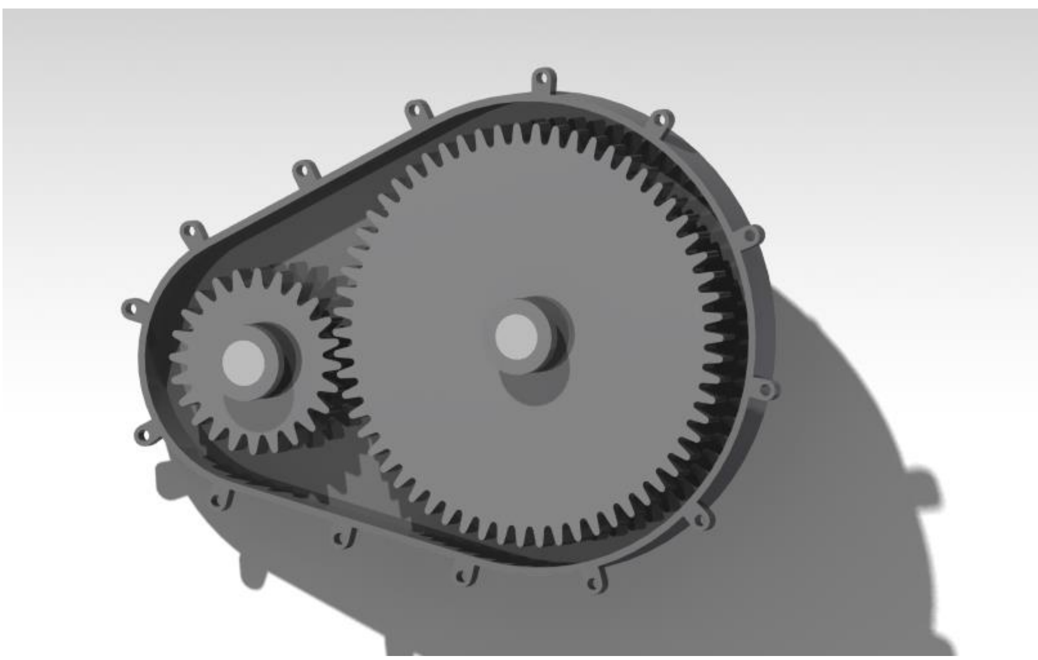

Figure 6, One Stage Gearbox render

4.2. Two Stages Spur Gears

For this design four different steel gears, three equal steel shafts, six equal bearings and an aluminum housing are considered. To determine the sizes of the gears, a solver in Excel was done. The variables used were the distance between the first and third axle and the total gear ratio. Some restrictions involving the minimum sizes of the gears and to avoid collisions were applied. Finally the solver proceeded to minimize the weight of the sum of the gears, or what is the same, their frontal area. The first gear has a diameter of 60 mm, the second a diameter of 87 mm, the third a diameter of 82mm and the fourth a diameter of 131mm. The total weight of the assembly is of 7.4 kg.

Figure 7, Two Stages Gearbox render

4.3. Straight bevel gears

For this design two different steel gears, two different steel shafts, three equal bearings and an aluminum housing are considered. The first gear has an average diameter of 70 mm and the second one an average diameter of 133 mm. The total weight of the assembly is of 6.75 kg and the housing dimensions 200x200x145mm.

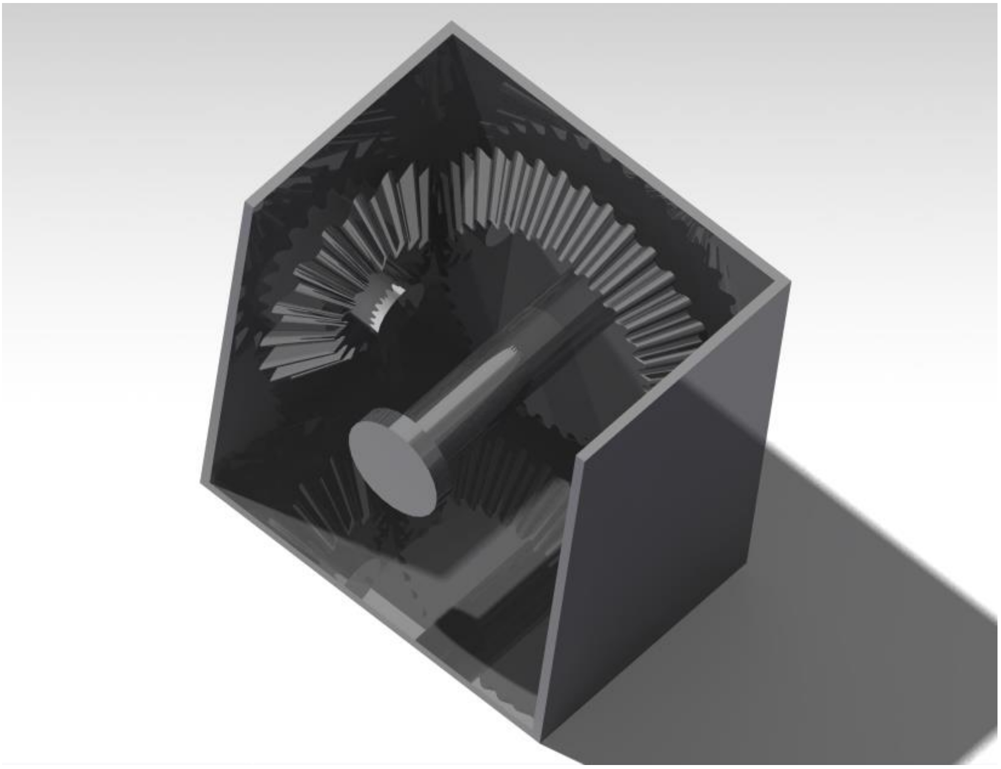

Figure 8, Bevel Gearbox render

4.4. Design election

As we can see, the one stage gearbox with spur gears is way far from the other ones if we compare the weight and dimensions. At first sight, there is no clear difference between the two stages spur gears and the straight bevel gears. They both weight approximately the same, but due to the complexity of the bevel gears assembly, design and tolerances, the chosen one is the two stage spur gears. Since the two gears of the second shaft are almost the same, for the final design there will be just one gear in the middle shaft. This way the gearbox will be thinner and lighter.

5. Gears selection

The gears are selected using the same solver as for the two stages gearbox before, with the difference that the two in the middle shaft are considered the same size. Once the dimensions are known, the next step is to find the specs of the gears. In order to do it, the software KISSSOFT is used. Introducing the following data, this software gives different valid options.

Power (kW)	28.3
Torque (Nm)	150
Force (N)	5000
Average speed (rev/min)	1800
Life hours (h)	100
Gear 1 diameter (mm)	60
Gear 2 diameter (mm)	81
Gear 3 diameter (mm)	138

Table 2, Gear Requirements

The selected option using the software is the following:

Gear	1	2	3
m	3	3	3
Z	21	24	48
Width (mm)	30	30	30

Table 3, Gears Geometry

The material selected for the gears is case hardening steel, 18CRNiMo7-6. The material properties are shown a continuation:

Rm, UTS (MPa)	1200
Rp (MPa)	850
E (MPa-1)	206000
V	0.3
P (kg/m3)	8930
σf lim/Sat (MPa)	460
σh lim/ Sac (MPa)	1500

Table 4, Gears Material

6. Shaft selection

Using the same software as before and introducing the data needed, the shaft material and dimensions are selected. The material is the C60 a heat treatable steel, not alloyed, through hardened. The external diameter of 30mm and the inner diameter of 15mm.

The material properties are the following:

Rm, UTS (MPa)	850
Rp (MPa)	580
E (MPa^{-1})	206000
V	0.3
P (kg/m^3)	8930

Table 5, Shaft Material

7. Gear Design

The units used in the simulation are the following:

Force	Distance	Stress	Torque
N	mm	MPa	Nmm

Table 6, Units

Knowing the dimensions and specs of the gears, we can proceed to optimize the designs, in order to reduce weight. The mesh is done in HyperMesh for all the models, and the processing with Abaqus. For the first design a key is used to connect the shaft and the gear.

7.1. First Gear

The element size is of 2mm. To apply the boundary conditions, one extra node has been created using a rigid body connection between the node in the center and all the inner nodes. This way all the restrictions applied on the center node will be applied at the inner nodes. One more node has been created on the top of one teeth in order to apply the load on the area.

The center node is fixed in the horizontal and axial, and the contact area with the key in the vertical direction. The force is applied as a compression on the area selected of the teeth.

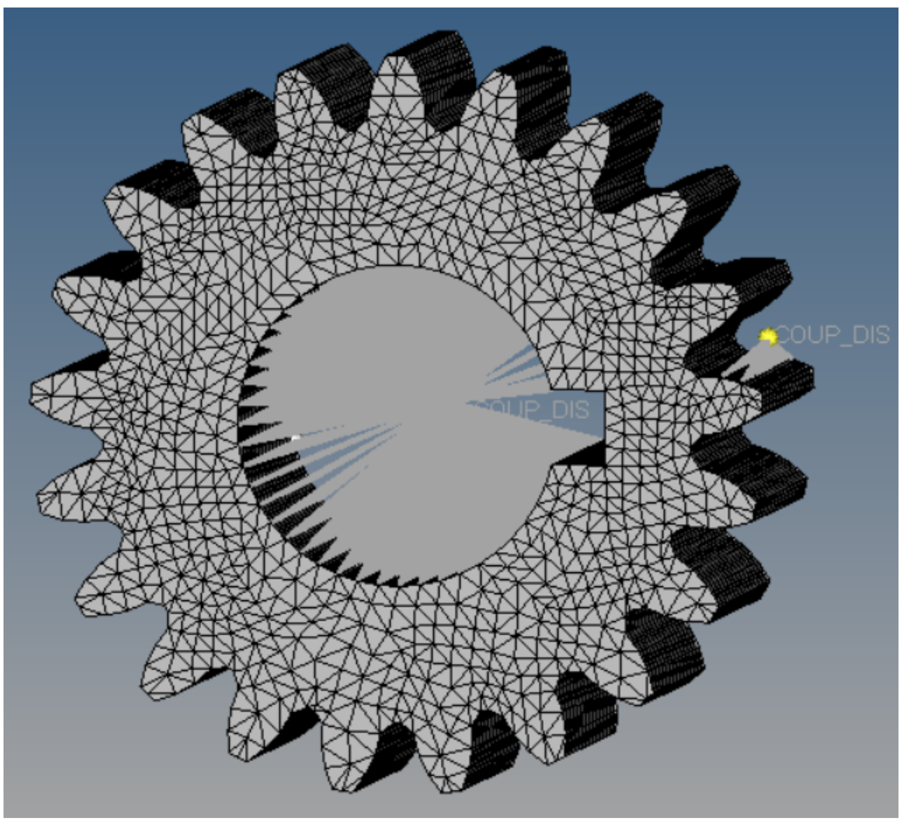

Figure 9, First gear preprocessing

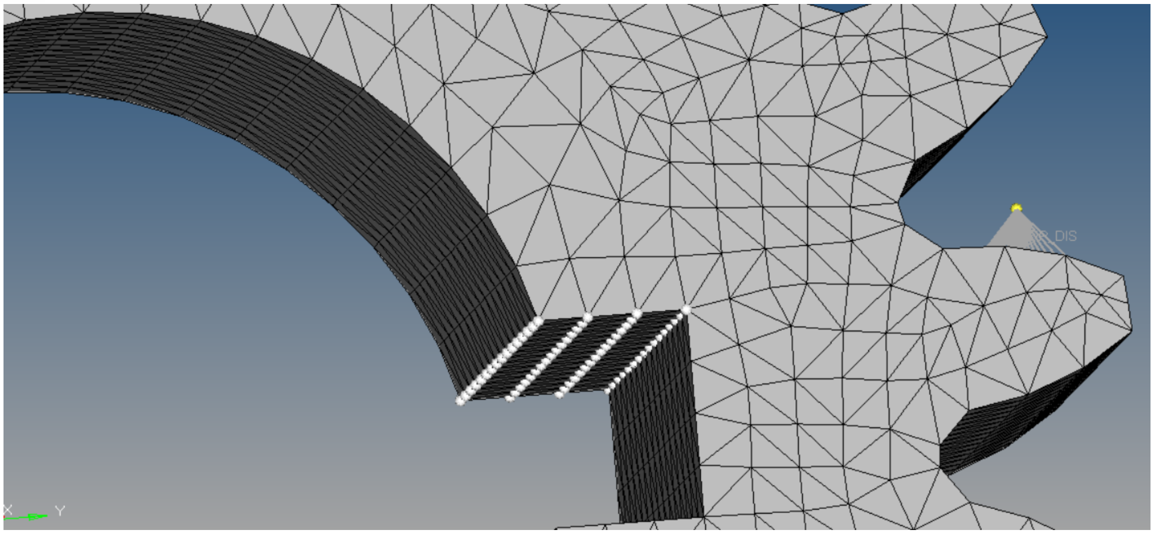

Figure 10, First Gear Boundary Conditions

The following picture shows the Stresses on the gear when applying 5000N of force. The peak stress is of 258 MPa and the weight of the gear is of 0.549 kg.

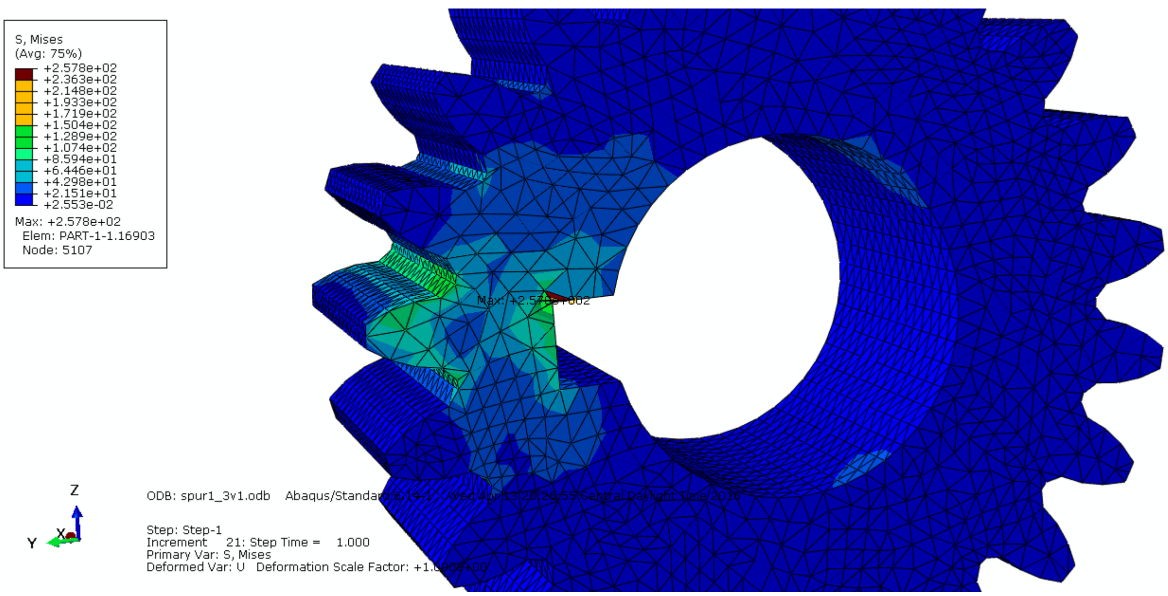

Figure 11, Stresses using a Key

Due to the stress concentration using just one key another solution has been considered. To use a shaft hub spline connection, this way all the torque will be shared among all the exterior surface of the shaft and interior surface of the gear.

Using the same software KISSSOFT, the splines have been calculated. All the shafts are the same for the different axles, in order to reduce costs and simplify the production of the assembly.

m	1.25
Z	26
b	30mm

Table 7, Spline Connection

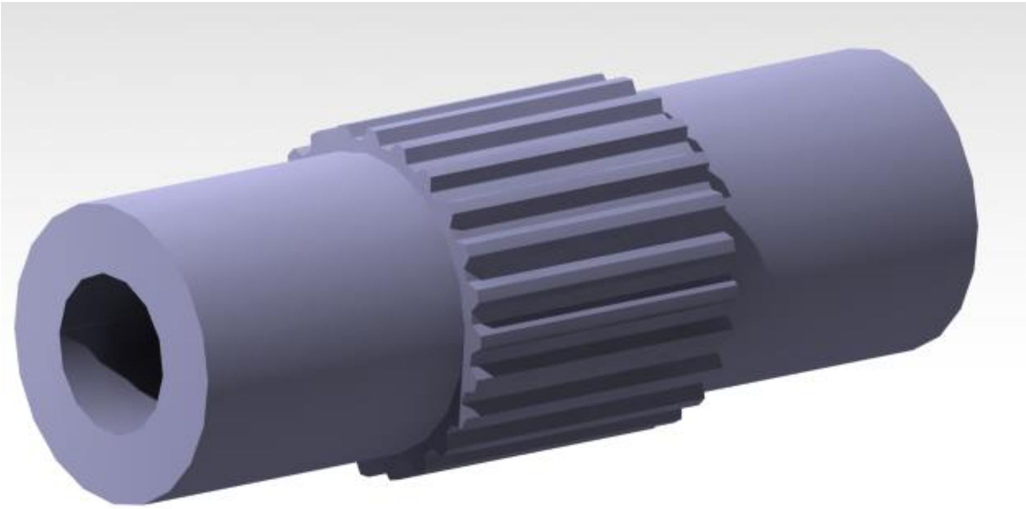

Figure 12, Spline Connection

To optimize the weight on the gears, the main factor that influences the geometry is the fatigue. Since the Finite Element Analysis of the fatigue is far more complex and need a lot more of computer resources, another method has been used. The typical values of the limit (S_e) for steels are half of the ultimate tensile strength. If the stresses do not reach this values, there is no fatigue in the component. Then, the following simulations would optimize the weight without reaching this stress.

The ultimate tensile strength is 1200 MPa, adding a security factor of 1.3, the maximum stress that can be achieved is 460 MPa

For the second version of the first gear, a pocket of a length of 7.5mm has been done at each side. There is no width reduction on the connecting area with the shaft, to assure the function of the spline. In this new shaft-spur connection, all the inner nodes has been fixed in all the directions. The load is applied to a node connected to the nodes of one side of the teeth.

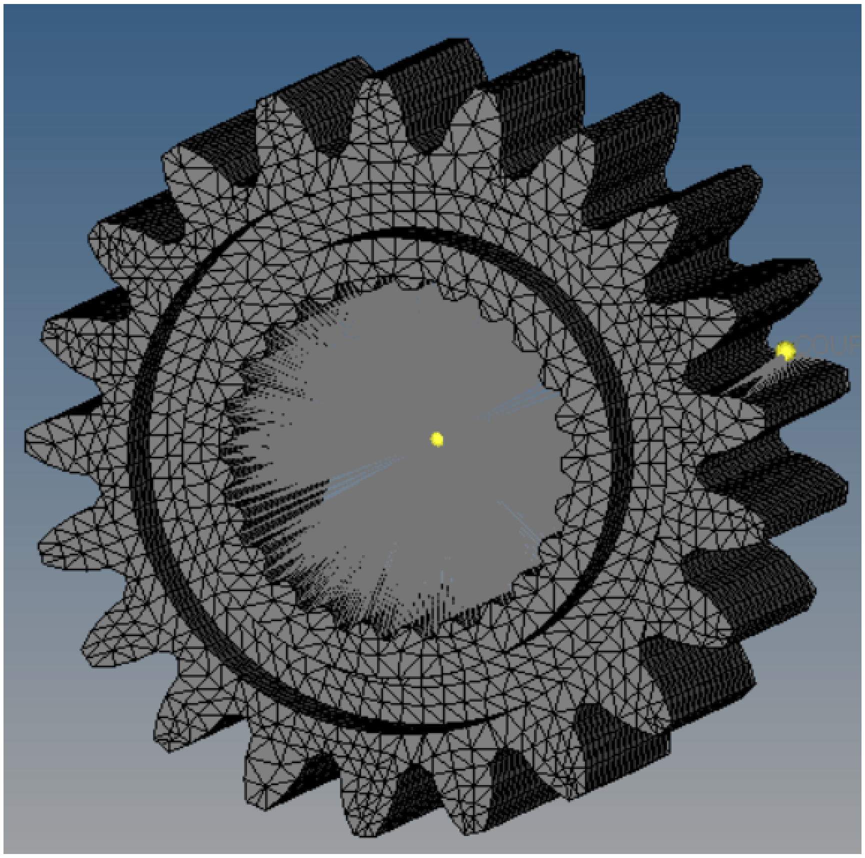

Figure 13, First Gear with a Spline Preprocessing

Repeating the same process as before for the new spur.

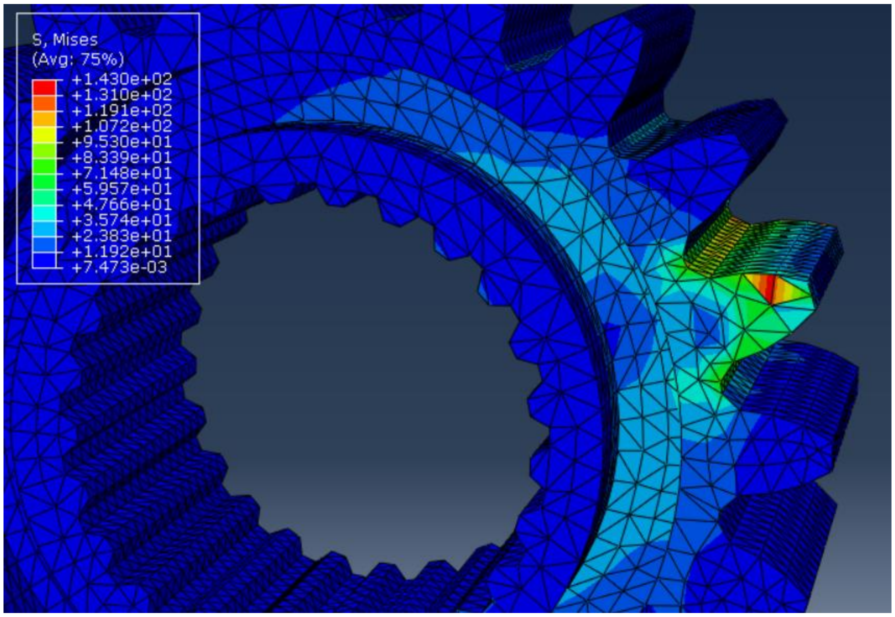

Figure 14, First Gear Stresses using a Spline and a 7.5mm pocket

The peak stress is 143 MPa, lower than in the first design even having a pocket. The reasons of it is because in this model, the area that has been fixed to avoid rotation are all the inner nodes, not just the key interaction as before. The element size used is still 2mm and the weight has been reduced to 0.453kg.

For the third version, the design will be the same than the version two but with a pocket length of 10mm per side.

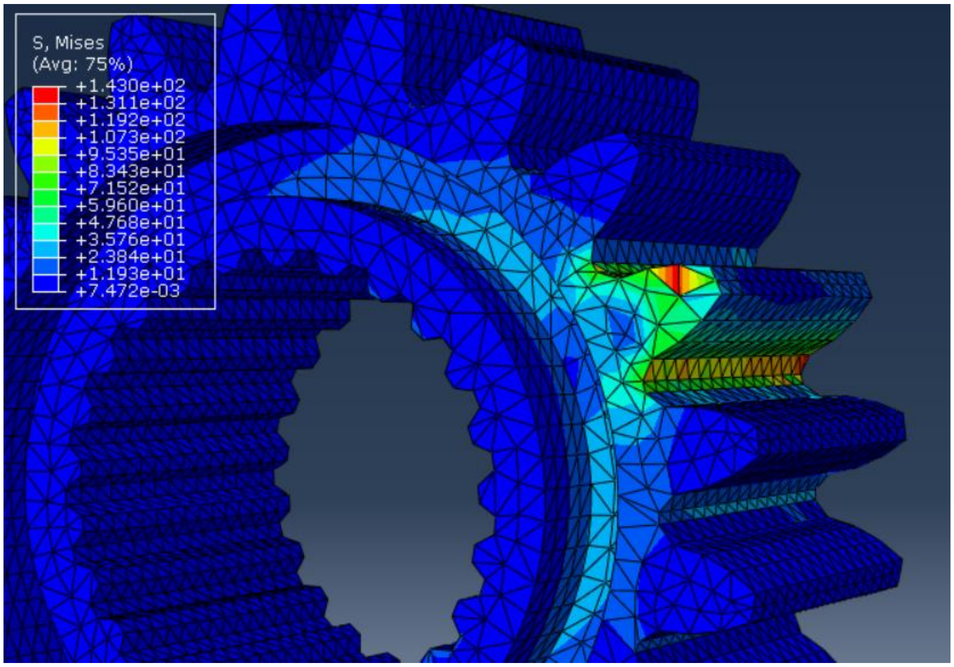

Figure 15, First Gear Stresses using a Spline and a 10mm pocket

The peak stress is still 143MPa at the same point, since the maximum stress is still in the area where the load is applied. The element size used is still 2mm and the weight has been reduced to 0.425kg.

No further reductions will be made, since the cost of reducing the material is not enough for the risks.

7.2. Second Gear

For the second spur gear the procedure is the same. Since the stress did not reach the limit with the pocket of 10mm per side in the first gear, the first simulation of the second gear would be directly with the pocket done. The mass without the hole is of 0.760kg, while with the hole is of 0.542kg. The element size is the same as in the last simulations, 2mm.

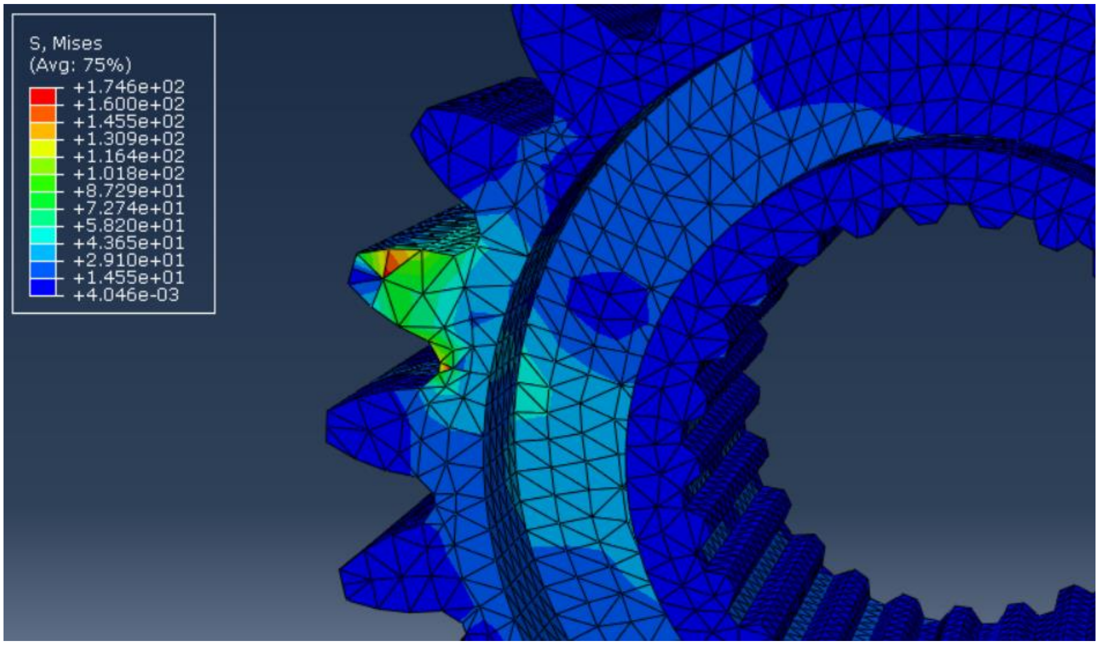

Figure 16, Second Gear Stresses using a Spline and a 10mm pocket

The peak stress is of 175 MPa, higher than in the first spur gear but still far from the limit of 460MPa. And the critical point is not in the optimized area where the material has been reduced. As in the first gear, no further reductions are done.

7.3. Third Gear

For the last gear and biggest one, the same simulation is done. As they were done in the second gear, the first simulation is already with a 10mm pocket per side. The mass without the hole is of 3.632kg. Once the pockets are done, the mass is reduced to 1.900kg. The element size of the mesh is still of 2mm.

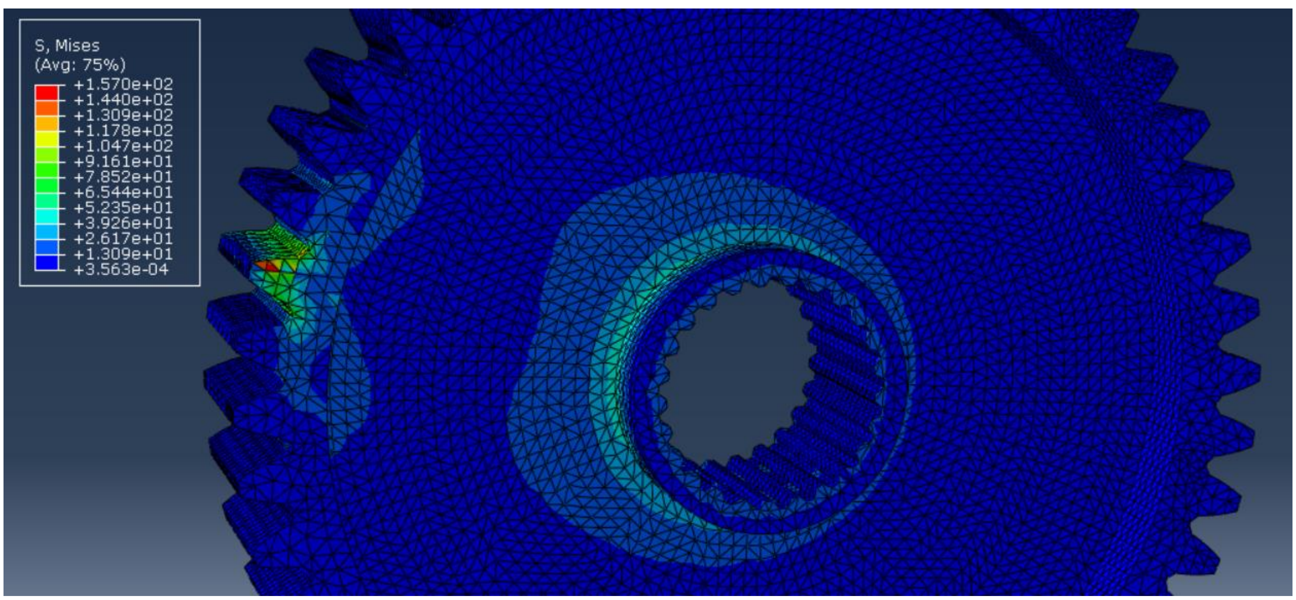

Figure 17, Third Gear Stresses using a Spline and a 10mm pocket

The peak stress achieved is of 157MPa on the teeth where the load is applied. Since the model has still a lot of material, a second operation is done in order to reduce weight. In this case 6 holes of 32mm of diameter are made. The weight is reduced to 1.521kg.

Figure 18, Third Gear final render

Repeating the simulation with the new model and still using the 2mm element size.

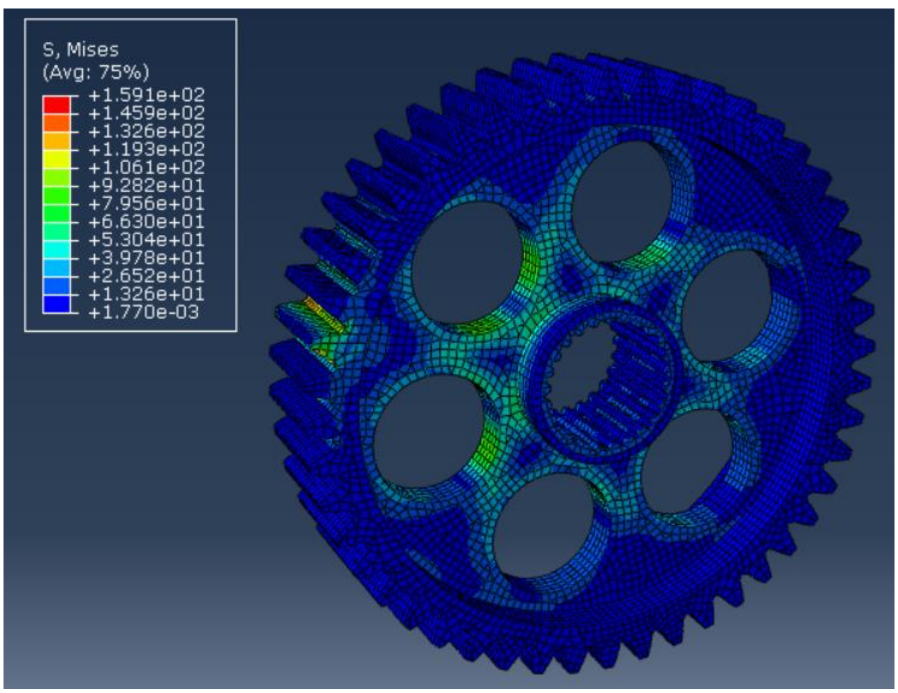

Figure 19, Third Gear Stresses

The peak stress achieved is of 159MPa. In this gear, from the first to the last design, the weight has been reduced to less than the half. This is the final design since there is no more material to take away.

In the following table the different phases of the optimization can be compared:

Gear	Version	Weight(kg)	Weight reduction(kg)
First	Normal	0.549	0.124
	Optimized	0.425	
Second	Normal	0.760	0.218
	Optimized	0.542	
Third	Normal	3.632	2.111
	Optimized	1.521	
All gears	Normal	4.941	2.453
	Optimized	2.488	

Table 8, Gear Weights

The total weight reduced is almost two and a half kilograms just for the gears. It may not look like a lot of weight, but knowing that the car will weight around 250kg, it is a 1%. Knowing this, the weight reduction is important in this case. The gears can still be further optimized a few grams, but without the chance of testing the simulation and the cost of a malfunction, those are the final designs.

8. Shaft Design

To corroborate that the shaft can hold the torque of the motor, a simulation is done. The element size is of 1mm. The whites nodes in the middle are fixed in the rotation plane, this ones are the ones that will be connected to the gear. To apply the torque one node is created on the outside, this has rigid connections to all the nodes on the external area of the shaft.

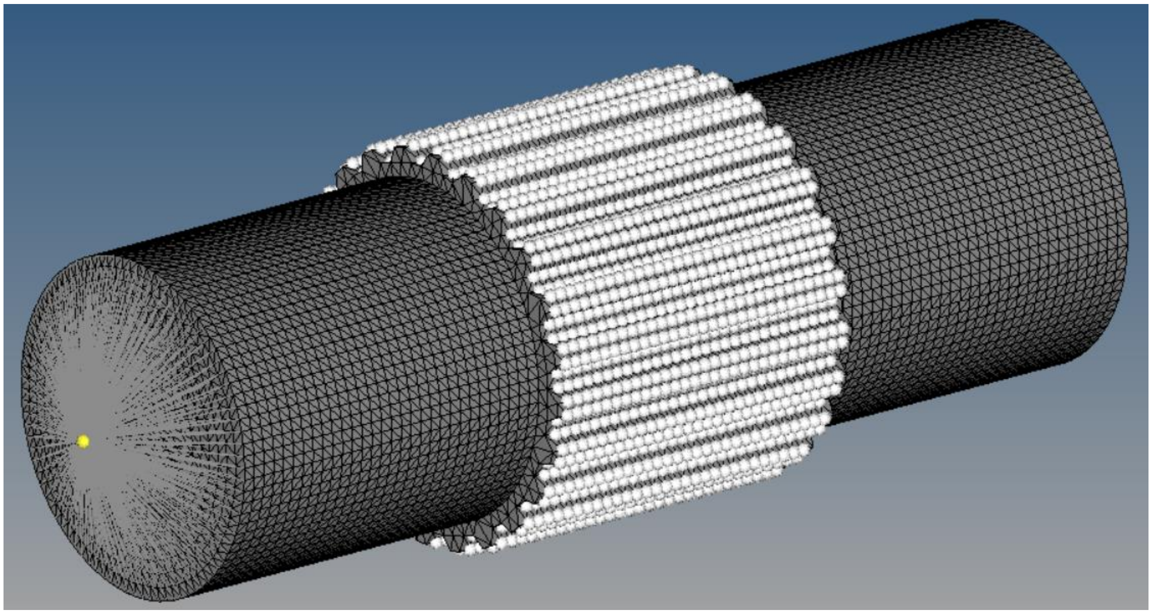

Figure 20, Shaft Preprocessing

Once the mesh and preprocessing is done, these are the results.

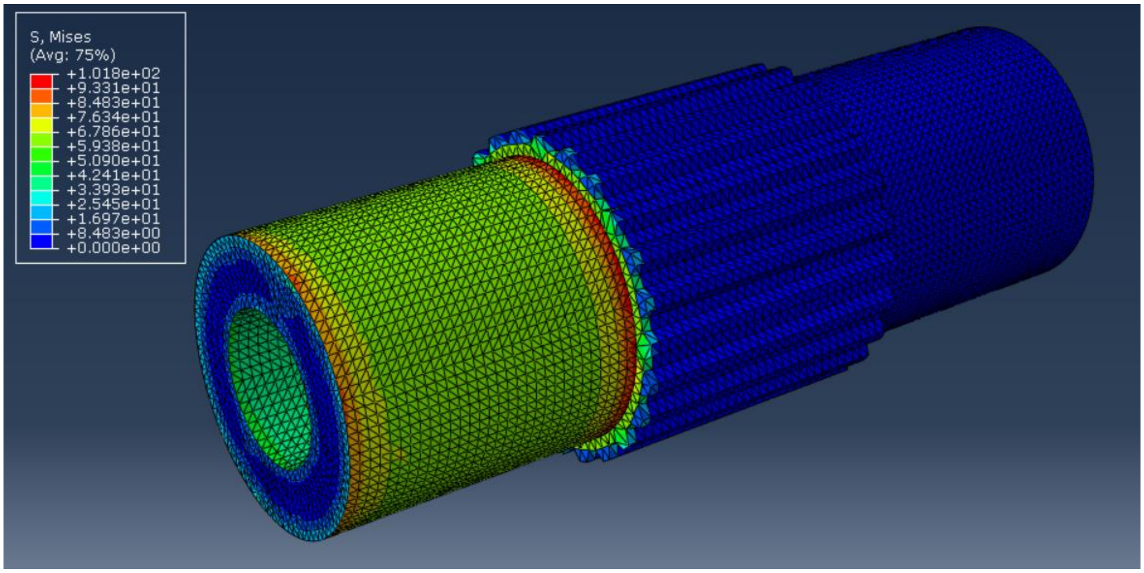

Figure 21, Shaft Stresses

The peak stress is of 102MPa, far from the maximum allowed. The maximum stresses are located where the spline starts.

To help introduce the gears on the shaft as well to reduce the stresses on this critical area, a ramp has been made to avoid the 90 degree angle.

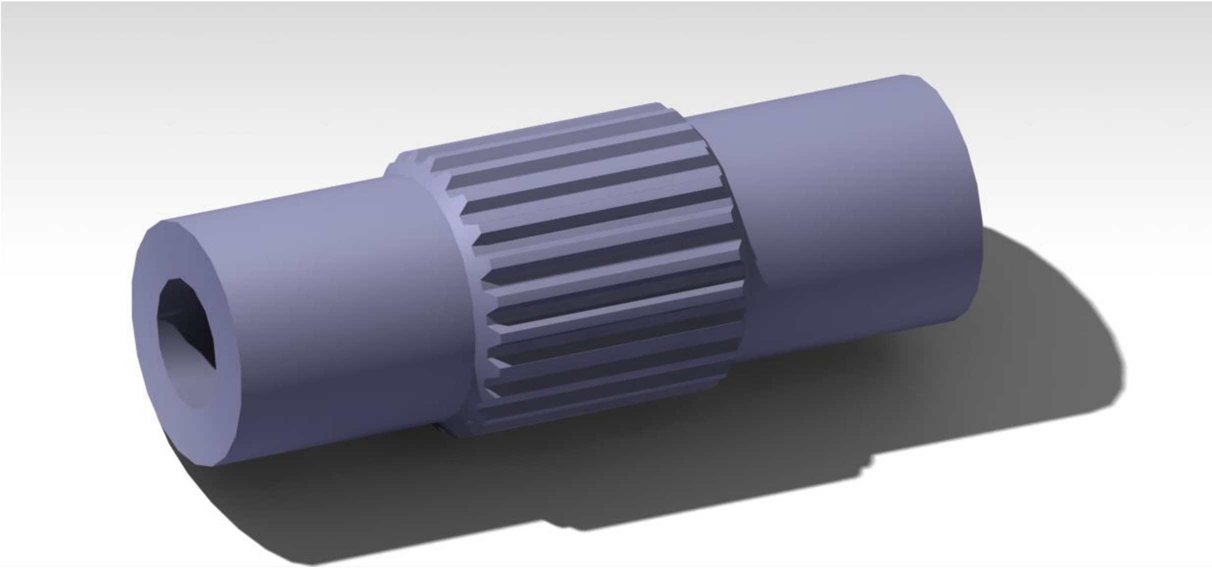

Figure 22, Shaft final design

9. Bearings selection

There are 6 equal bearings, 2 for each shaft. The force per each shaft is 5kN, so each bearing should be able to support 2.5kN of radial load. The inner dimension is determined for the shaft, which is 30mm of diameter. The external diameter cannot be bigger than the space inside the housing. The bearings should be able to arrive up to 3200 r/min.

There are different options when it comes to bearings. Some of them are designed to handle radial load, thrust load, or some combination of the two. The velocity and the dimensions also play an important role when it comes to choose.

- Ball Bearings: they can handle both radial and thrust loads, but can only handle a small amount of radial load. They are found in a wide array of applications and are very common. They deform easily when overloaded.

- Roller Bearings: designed to carry heavy loads. The cylindrical roller helps distribute the weight into a larger area, lowering the stress under the same load. However, means the bearing can handle primarily radial loads, but is not suited to thrust loads. Needle bearings have cylinders with smaller diameter, for when the space is an issue.

- Ball Thrust Bearings: designed to handle high thrust loads and low speeds.

- Roller Thrust Bearings: like ball thrust bearings, handle thrust loads. As well as in the regular bearings the difference is that the cylindrical rollers can handle bigger amounts of loads.

- Tapered Roller Bearings: designed to handle large radial and thrust loads.

Following a table of SKF is showing the performance of each type of bearing and comparing them:

matrix - Bearing types - design and characteristics

The matrix can only provide a rough guide so that in each individual case it is necessary to make a more qualified selection referring to the information given in the catalogue

Symbols
+++ excellent - poor
++ good -- unsuitable
+ fair ← single direction
←→ double direction

Design
1 Tapered bore
2 Shields or seals
3 Self-aligning
4 Non-separable
5 Separable

Characteristics
Suitability of bearings for
6 Purely radial load
7 Purely axial load
8 Combined load
9 Moment load
10 High speed
11 High running accuracy
12 High stiffness
13 Quiet running
14 Low friction
15 Compensation for misalignment in operation
16 Compensation for errors of alignment (initial)
17 Locating bearing arrangements
18 Non-locating bearing arrangement
19 Axial displacement possible in bearing

Bearing type	Variants	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
Deep groove ball bearings	a, b		a				+	+ ←→	+ ←→	- b+	+++ b+	+++ b+	+	+++	+++	-	-	++ ←→	+	--
Angular contact ball bearings							+	+ ←	++ ←	-	++	+++	+	++	++	-	-	++ ←	--	--
	a, b, c		b		a, b	c	++	+ ←→	++ ←→	+	+	++	++	+	+	--	--	++ ←→	+	--
							-	++ ←→	+ ←→	+	++	+	+	+	+	--	--	++ ←→	-	--
Self-aligning ball bearings							+	-	-	--	+++	++	-	++	+++	+++	+++	+ ←→	+	--
Cylindrical roller bearings							++	--	--	--	++	++	++	++	++	-	-	--	+++	+++
	a, a, b, b						++	+ a← b←→	+ a← b←→	--	++	++	++	+	++	-	-	++ a← b←→	+ a←	+ a←
full complement	a, b				a	b	+++	-	+ ←	--	-	+	+++	-	-	-	-	+ ←	+	+
	a, a, b, c		a				+++	-	+ a←→ b←	+	-	+	+++	-	-	--	--	+ a←→ b←	+ c	+ b← c←→
Needle roller bearings	a, b, c		a	c			++	--	--	--	+	+ a++	a++ b++	+	-	--	-- c++	--	+++	+++
	a, b, c		b, c				++	--	--	--	+	+	++	+	-	--	--	--	+++	+++
	a, b, c		b, c				+	+ c++ ←	+ ←	-	+	+	++	+	-	--	--	+ ←	--	--
Tapered roller bearings							++	++ ←	+++ ←	-	+	+	++	+	+	-	-	+++ ←	--	--
	a, b						+++	++ ←→	+++ ←→	+	+	+	+++	+	+	-	--	+++ ←→	-	--
Spherical roller bearings							+++	+ ←→	+++ ←→	--	+	+	++	+	+	+++	+++	++ ←→	+	--
CARB bearings							+++	--	--	--	+	+	++	+	+	+++	+++	--	+++	+++
full complement							+++	--	--	--	-	+	+++	+	+	+++	+++	--	+++	+++
Thrust ball bearings	a, b						--	+ a← b←→	--	--	-	++ a	+	-	+	-	--	++ a← b←→	--	--
	a, b						--	+ a← b←→	--	--	-	+	+	-	+	-	++	++ a← b←→	--	--
Needle Cylindrical roller thrust bearings	a, b						--	++ ←	--	--	-	a+ b++	++	-	-	--	--	++ ←	--	--
Spherical roller thrust bearings							--	+++ ←	+ ←	--	-	+	++	-	+	+++	+++	+++ ←	--	--

Table 9, Bearing Comparison

Since the loads will be basically radial, because the engine-gearbox would be attached to the same frame, and the driven gear would have a tripod taking off almost all of the thrust loads. The type that adapts more to the performance that is needed is the cylindrical roller bearings.

Among the cylindrical roller bearings, there are different types, the difference between them is basically the limitation of the displacement in either one, two or none axial direction.

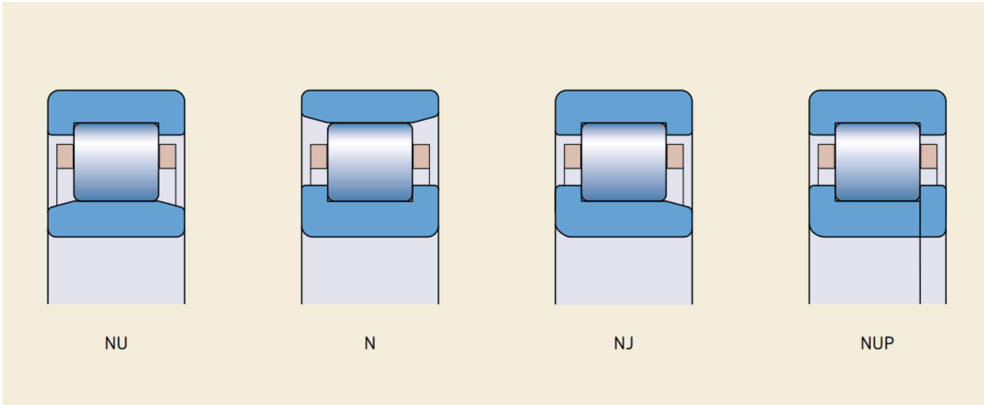

Figure 23, Roller Bearings types

To avoid redundancies and facilitate the assembly, two NJ are selected. One at each side to fix the shaft axially. Looking at the catalogue, the one that has all the requirements is the NJ 206 ECP, having a fatigue limit of 4.2kN, a limiting speed 14000r/min, and internal diameter of 30mm, an external diameter of 62mm, a width of 16mm and weighting 0.21kg.

10. Housing design

The material selected for the housing is Aluminum. Its lightweight and mechanical properties are the best choice for this part. There are other alloys a little lighter but they are not worth the price, either the material or the machining of a more resistant material.

It is closed by 11 bolts, which are also used to attach it at the frame. The housing also has the space in order to accommodate the bearings. The size of the housing depends on the gears size, it has to be compact in order to reduce weight and occupy the less space possible inside the frame. It is also important that the gears get in contact with the oil so the connection between them gets lubricated.

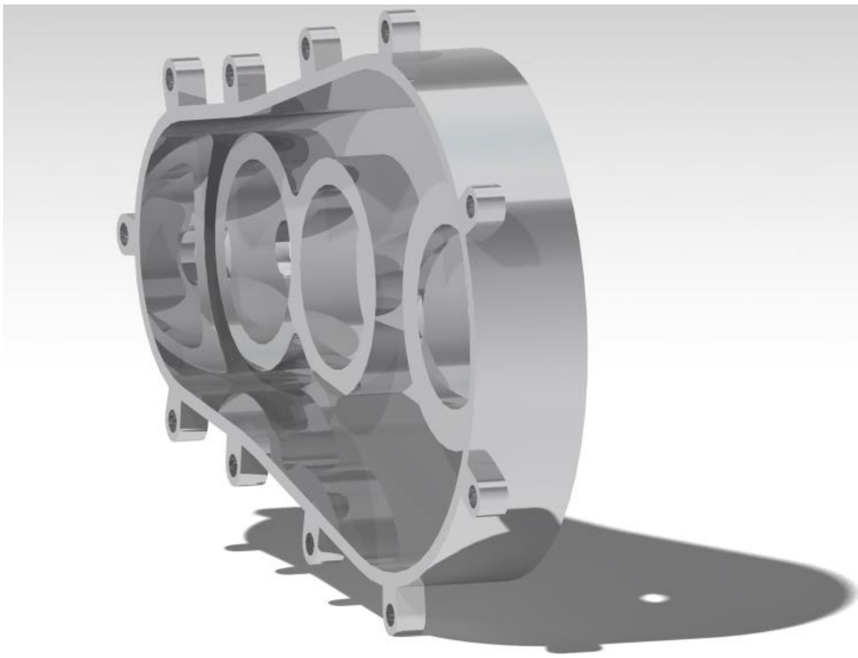

Figure 24, Housing

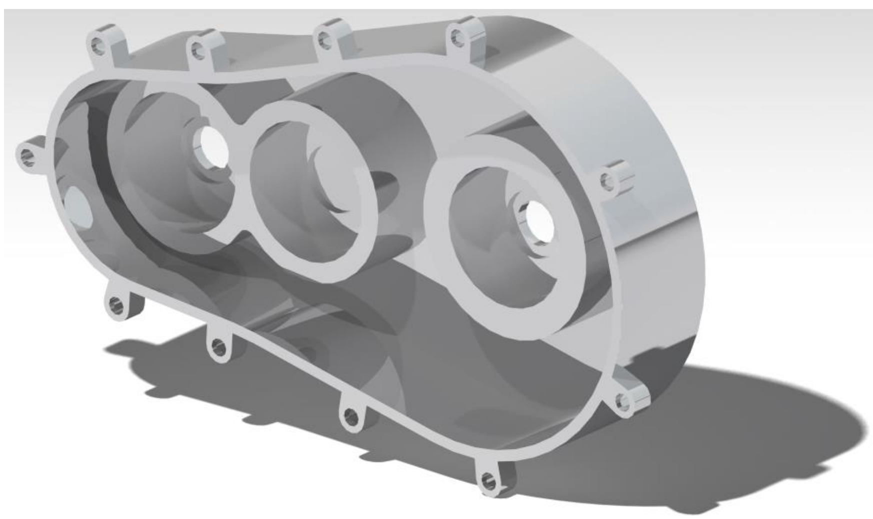

Figure 25, Housing

11. Final assembly

Here we can see the final design with all its components. The total weight is 5.96 kg.

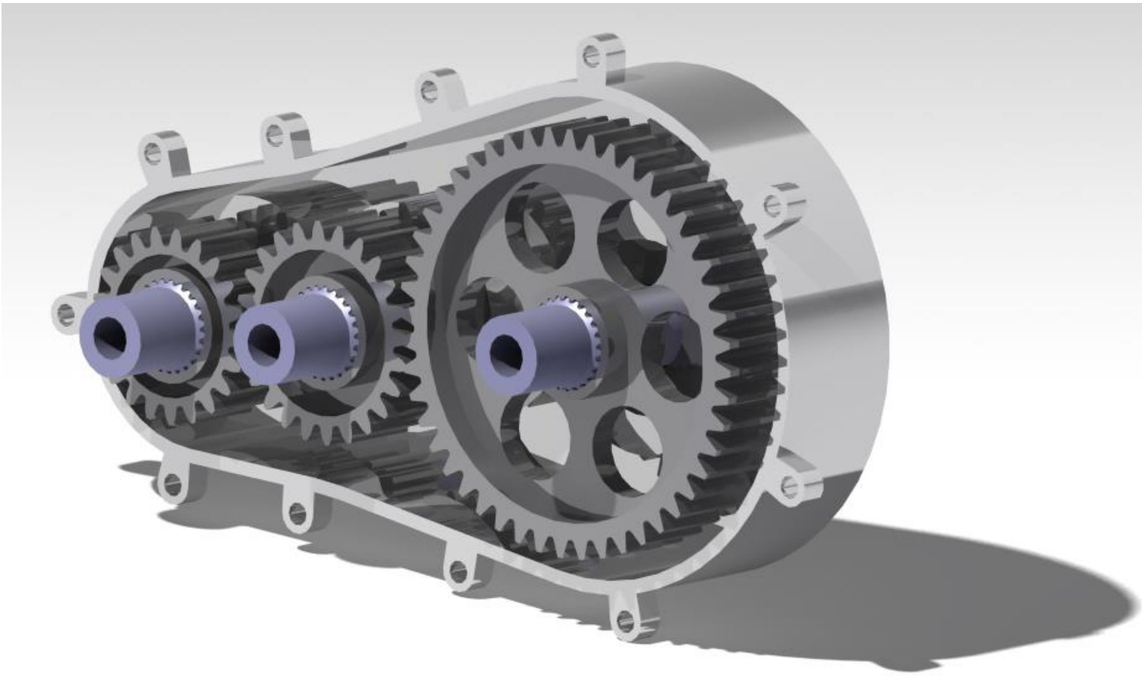

Figure 26, Final Assembly

In the next picture is the top view of the assembly without the housing.

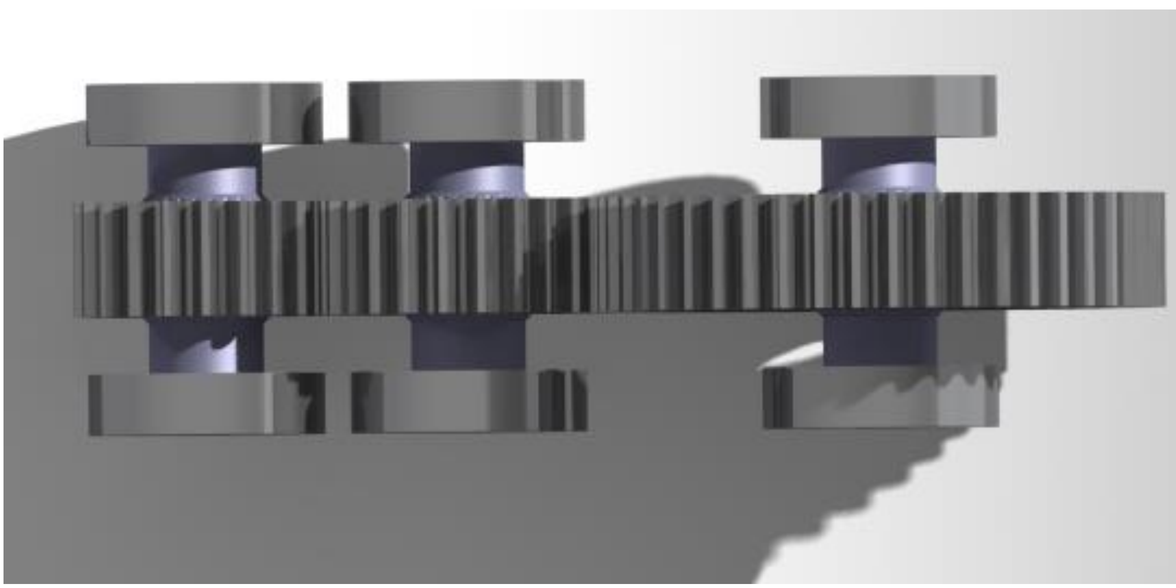

Figure 27, Final Assembly Gears

Results and Discussion

Using different software for each part of the project, a transmission system for the IIT Motorsports Formula SAE has been designed.

The reasons because the use of a gearbox is the best option as the transmission system are basically; is more compact than a pulley-belt system, and can be used to recover the kinetic energy from the brakes, not the case of a chain.

The gearbox ratio has been approximated by introducing some parameters of the car and studying its behavior in a Formula SAE track.

The specs that affected the number of stages, as well as the diameters of the different gears are the geometry of the motor. The big diameter of the motor, influenced in a way that one gear has been introduced in the middle just to make the transmission longer.

The design of the connection between the gearbox and the motor is still pending, as well as the implementation of the tripods.

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