Tampereen ammattikorkeakoulu

Design of a Planetary Gear System for Formula Student Hybrid Drivetrain

Otto Makkonen

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ABSTRACT

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The purpose of this thesis was to design a planetary gearbox for Tampere Formula Students vehicle for the 2025 competition season. The need for a new hybrid drivetrain concept arose from the shortcomings of previous systems, and the possibility of a future electric car needing similar technologies.

The design started by setting the limitations for layout and preliminary design of encompassing elements. A layout concept for the gearing was chosen to fit the goals of achieving a light weight and high gear ratio. Gear life calculation was done using advanced simulation software and a complete 3D model of the system was created. All component designs were done with manufacturability in mind, while simulations were used to create lightweight and adequately structural parts.

A working gearbox design for TFS25 was achieved and a design documentation capable of guiding future design created. The goal of staying lightweight in a small space was achieved as drivetrain mass decreased by 40% and can be encompassed in a smaller wheel than previous designs. New tools for gear design were implemented successfully and can be used in the future. The time frame of this thesis did not allow for physical testing of the designed system, and this should be the focus of future system design.

Key words: transmission, formula student, planetary gearbox

CONTENTS

ABBREVIATIONS AND TERMS

TAMK	Tampere University of Applied Sciences
TFS	Tampere Formula Student
FSG	Formula Student Germany
FSAE	Formula SAE
Formula student	Design competition
I	Gear ratio
m	Module
d	Pitch diameter
z	Number of teeth
d_a	Tip diameter
d_f	Root diameter
α_b	Working pressure angle
α	Reference pressure angle
ε_α	Transverse contact ratio
a	Center distance
b	Face width
s_f	Safety factor
G	Gravitational acceleration
R_w	Wheel radius
D_w	Wheel diameter
F	Force

1 INTRODUCTION

Drivetrain is an important part of any moving vehicle. This system connects the power source to the wheels moving the vehicle. With good design the drivetrain can minimize the losses in power transition and improve the performance of the vehicle

The goal of this thesis is to design a planetary gear system for Tampere Formula Student TFS25 vehicle hybrid drivetrain. The drivetrain will be used in competitions for the 2025 competition season and the preceding testing season. The developed drivetrain needs to fulfill goals created for it and create a basis for future design of electric vehicle drivetrains.

PICTURE 1. TFS25 vehicle concept

1.1 Formula Student

Formula Student or FSAE depending on the event, is an international design competition created for university students. The goal of the competition is to design and manufacture a new formula-style racecar each year.

Formula Student teams are classified into two categories. Combustion, where the vehicles are powered by internal combustion engines and Electric for electric powered vehicles. Depending on the competition, there can also be a different class for Driverless, in which the vehicle is self-driving and for Hydrogen, where the vehicle is either powered by hydrogen internal combustion engine or fuel cell technology.

The competitions include dynamic and static events in which the teams from different universities compete in different courses. Static events measure the ability of the team to design and develop vehicles, as well as their cost and manufacturing knowledge. Additionally, the teams are responsible for creating a business plan involving some aspects of their car. Dynamic events include different racing scenarios for the developed vehicles in the forms of acceleration, autocross, skid pad and endurance.

Formula Student is governed by FSG (Formula Student Germany) rulebook, which main goal is to limit the power achievable by the teams and guide the design of the vehicles towards safety.

1.2 Tampere Formula Student

Tampere Formula Student is the Formula Student team of Tampere University of Applied Sciences. The team works under its own association TAMK Formula Student Ry founded in 2006.

The team is made up of students from multiple different fields of study interested in motorsports. The main purpose of the team is to gain knowledge that can be helpful in each of the members’ studies as well as the following careers.

The team created its first working vehicle in 2008 in the form of FS008. The team has been active after this and created a new vehicle each year except for 2020, when the COVID-19 pandemic halted the manufacturing of the car.

PICTURE 2. FS008 vehicle (Tampere Formula Student n.d.)

Throughout the course of the 17 active year in Formula Student, the team has always had a combustion car. In recent years the team has been trying to create a hybrid vehicle where both electric and combustion vehicles get combined.

2 THEORY

Gears are components that are used for transitioning power from one axle to another. In addition, these gears can be used to change the rotational speed and torque affecting the axle. This effect is called gear ratio. The gear ratio is the ratio of teeth in driven gear to driver gear. The gear ratio of a single gear pair can be calculated with formula 1 (KHK Gears n.d.)

$$i = \frac{z_2}{z_1} \tag{1}$$

Gears are a versatile component that can be used to create complicated drive systems with multiple inputs, multiple outputs or high gear ratios. As the system structure gets more complicated, also the design process and needed tools change.

This chapter touches on gears and drivetrains that use them. The focus is on different drive systems commonly used in Formula Student.

2.1 Gear terminology

Gears can be thought of as just friction wheels as they work by turning each other on tangent reference diameter, known as the pitch diameter (d). In gears case the power transitioning is just done by structural teeth and not by friction. This pitch circle and pitch diameter can be seen in figure 1.

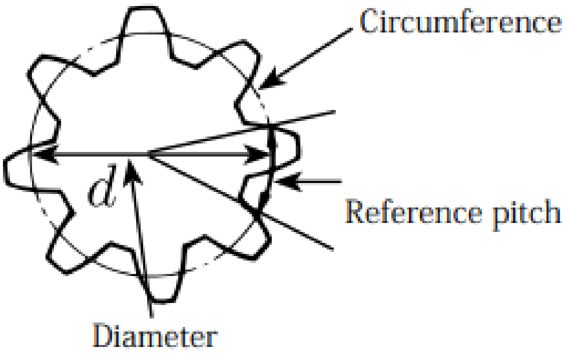


FIGURE 1. Pitch and pitch diameter (Cranor 2009)

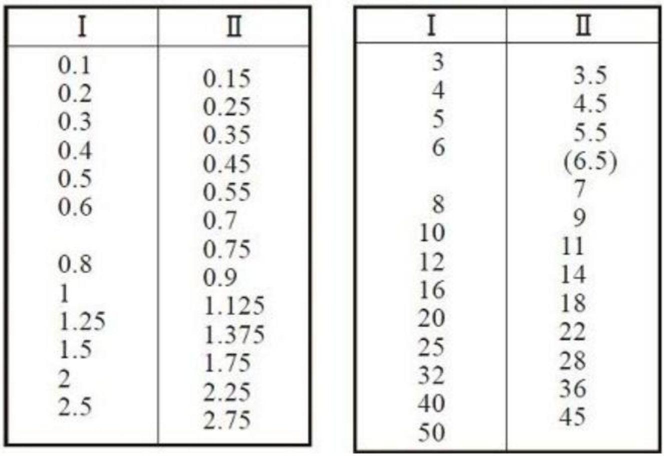
Circular pitch (p) is the distance between corresponding points in adjacent teeth measured on pitch circle. Two gears with different pitches cannot mesh properly, so gear pairs need to have the same pitch as the other. Pitch is defined by formula 2 (KHK gears n.d.).

$$p = \frac{\pi \cdot d}{z} \tag{2}$$

Pitch is often not used in defining the teeth sizes as using π in multiple calculations is not ideal. For this purpose, there is an ISO defined unit module (m). Two gears with different modules cannot mesh with each other. As pitch (p) represents the distance between teeth and larger pitch means larger tooth size, it is directly linked to module (KHK gears, n.d.). Module of gears teeth can be calculated by formula 3.

$$m = \frac{d}{z} \tag{3}$$

As with pitch, the larger module corresponds to larger teeth. For design purposes the values for gear modules have been defined by JIS standard. The standard defines used modules and divides them into two categories. The standard also recommends the use of row one and to avoid the module 6,5.



I	II
0.1	0.15
0.2	0.25
0.3	0.35
0.4	0.45
0.5	0.55
0.6	0.7
	0.75
0.8	0.9
1	1.125
1.25	1.375
1.5	1.75
2	2.25
2.5	2.75

I	II
3	3.5
4	4.5
5	5.5
6	(6.5)
	7
8	9
10	11
12	14
16	18
20	22
25	28
32	36
40	45
50	

FIGURE 2. Gear modules (KHK gears n.d.)

The tooth pressure angle (α) is defined as the angle between a radial line and a line tangential to the tooth profile at pitch point. At the pitch point the pressure angle defines the direction normal to tooth profile. Standard values for pressure angle are usually 14,5 and 20 degrees. Pressure angle of 14,5 degrees was extensively used in the past as it provides more power for transmission and less forces for bearings, but smaller pressure angle teeth are weaker. Now the pressure angle of 20 degrees is more commonly used. Picture 3 shows the pressure angle with gears (A) and (B), and reference circles (2) and (3).

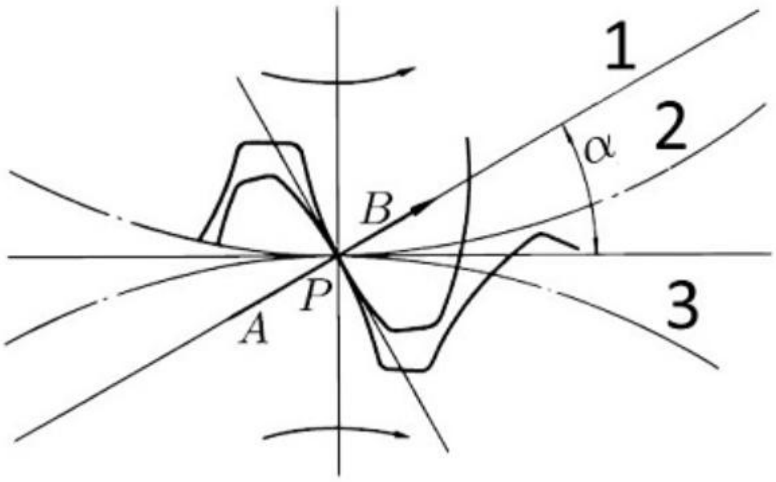


FIGURE 3. Pressure angle (KHK GEARS n.d.)

Tooth depth is the distance from tip circle to the root circle. This depth can be divided into two parts. Addendum is the distance from pitch circle to tip circle and dedendum is the distance from pitch circle to root circle. Tooth thickness is the Distance between two pitch points on opposite sides of a tooth. Face width of a gear indicates the width of the tooth measured on a parallel axis to that of the gears center axle.

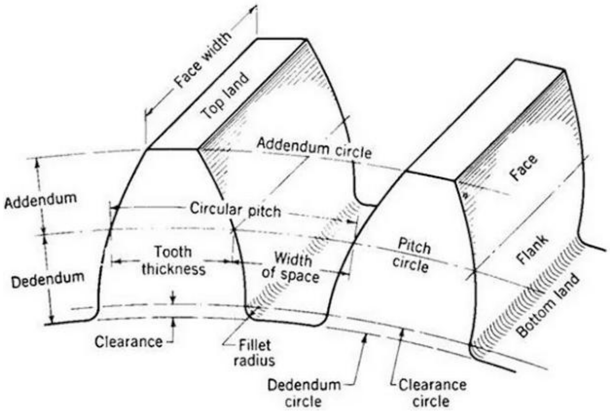


FIGURE 4. Gear terminology (Magalhães 2024)

The size of the gear can be visualized by tip diameter (d_a) and root diameter (d_f). These diameters form the above-mentioned virtual circles. Tip diameter can be calculated with formula 4 and root diameter by formula 5.

$$d_a = d + 2 \cdot m \tag{4}$$

$$d_f = d - 2{,}5 \cdot m \tag{5}$$

When gears are put together, it is called a gear pair. The distance between two gear axis is center distance (a) and can be calculated as half of the sum of their pitch diameters. When two gears are paired there should be a proper gap between the teeth to allow for proper meshing without extra friction or locking. The gap between mating surfaces in meshing gears is called backlash. Tip and root clearance is the distance between meshing gears tip circle and root circle.

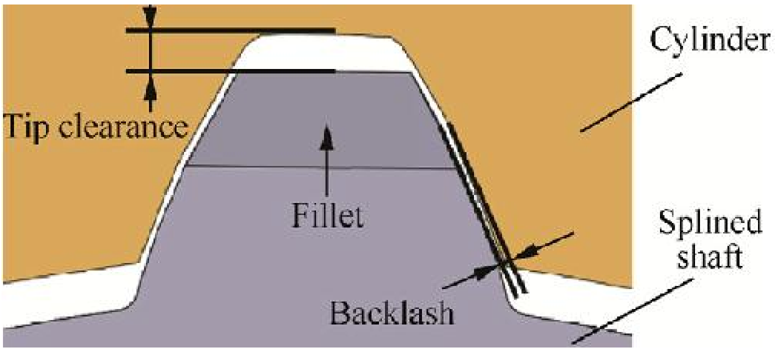


FIGURE 5. Backlash and tip clearance (Zhang 2018)

A simple gear profile for cylindrical gear is often displayed as a spur gear. This type of gear has straight cut teeth that are parallel to the gears center axle. This type of gear is easy to manufacture to high precision and in theory does not cause any axial loads. The main downside of a spur gear pair is the high noise emitted by their meshing.

In assemblies requiring more quiet solutions with cylindrical gear a gear type called helical gear is used. These gears have their teeth cut at an angle called helix angle (β) around the center axle of the gear. These gears are often quieter than spur gears and due to the angled teeth having a higher contact ratio. Because of this they often have less vibrations and can transmit large forces, but their manufacturing is harder and more expensive. These gears also require thrust bearing for the axial forces created.

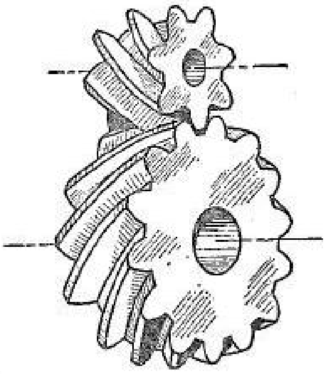


FIGURE 6. Helical gears (KHK gears n.d.)

Sometimes the assembly needs to be quiet, but no axial forces can be present and for this the herring bone gears or double helical gears could be used. These gears are a combination of two opposite helix angles on the gear surface eliminating the axial loads.

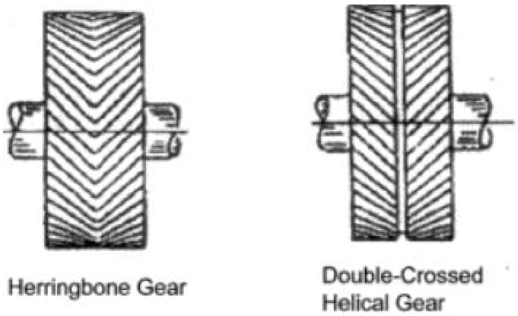


FIGURE 7. Herringbone and double-crossed gears (Collins 2017)

2.2 Gear strength and durability

When designing gear systems, the most important part after the gear ratio is the strength and durability of the said system. This can usually be divided into two different categories. These are bending strength and surface durability.

Bending strength can be thought as the root strength of the gear tooth as the failure case is a stress fracture at tooth root fillet. The gears need to have sufficient bending strength to avoid these types of failures. Allowable tangential force for spur gears can be calculated with the following formula according to JGMA401-01 (KHK Gears n.d.).

$$F_{tlim} = \sigma_{Flim} \cdot \frac{m \cdot b}{Y_F \cdot Y_\varepsilon} \cdot \frac{K_L \cdot K_{FX}}{K_V \cdot K_O} \cdot \frac{1}{S_F} \tag{6}$$

Often a problem with bending strength of the teeth can occur if the gears are too small. This causes the tooth profile to get thin at the root. This is where profile shifting of the teeth can be introduced. With profile shift the profile of teeth can be modified to increase bending strength.

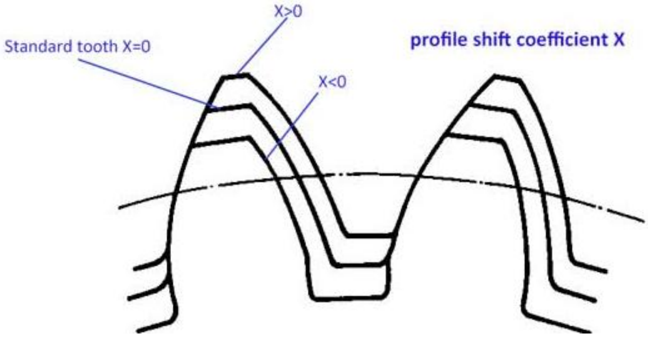


FIGURE 8. Tooth profile shift (Boyan manufacturing solutions n.d.)

Larger profile shifts are often needed in solutions requiring small gears as the smaller the gear gets the thinner the teeth are and undercut becomes present. With profile shift undercut can be partly or completely terminated. For other solutions gears need to have different center distance than standard, and this can be achieved with the same means.

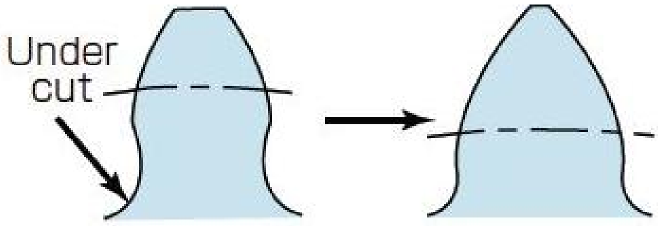


FIGURE 9 Undercut correction with profile shift (KHK Gears n.d.)

Main characteristics of positive profile shifting are increased bending strength and smaller contact ratios. For negative shifting they are decreased bending strength, increased contact ratio and possibility of undercut.

The other discussed factor in gear strength and durability is surface durability. Surface durability considers the stresses affecting the tooth contact surface. The strength against surface damage known as pitting can be calculated by hand with formula 7 given by KHK gears according to JGMA402-01.

$$F_{tlim} = {\sigma_{Hlim}}^2 \cdot d \cdot b \cdot \frac{i}{i \pm 1} \cdot \left(\frac{K_{HL} \cdot Z_L \cdot Z_R \cdot Z_V \cdot Z_W \cdot K_{HX}}{Z_H \cdot Z_M \cdot Z_\varepsilon}\right)^2 \cdot \frac{1}{K_{H\beta} \cdot K_V \cdot K_O} \cdot \frac{1}{{S_F}^2} \quad (7)$$

The factors used in previous calculations are explained in the table below.

TABLE 1. Symbols for stress calculation factors (KHK gears n.d.)

σ_{Flim}	Allowable bending stress	Z_H	Zone factor
σ_{Hlim}	Allowable Hertz stress	Z_M	Material factor
Y_F	Tooth profile factor	Z_ε	Contact ratio factor
Y_ε	Load sharing factor	Z_{HL}	Life factor
K_L	Life factor	Z_L	Lubrication factor
K_{FX}	size factor of root stress	Z_R	Surface roughness factor
K_V	Dynamic load factor	Z_V	Lubrication speed factor
K_O	Overload factor	Z_W	Hardness ratio factor
K_{HX}	Size factor	$K_{H\beta}$	Longitudinal load distribution factor

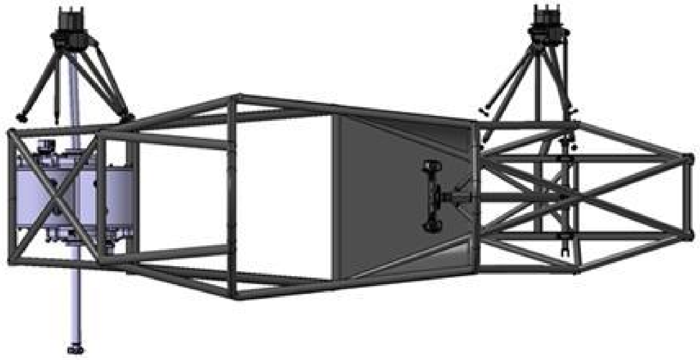
When talking about the transmitting load through the gears one of the main factors is the contact ratio. Contact ratio defines the average amount of teeth in contact during the meshing of the gear pair. The transverse contact ratio for spur gears can be calculated with formula 8 (Dengel 2021).

$$\varepsilon_\alpha = \frac{\sqrt{\frac{d_{a1}}{2}^2 - \frac{d_1}{2}^2} + \sqrt{\frac{d_{a2}}{2}^2 - \frac{d_2}{2}^2} - a \cdot \sin(\alpha_b)}{\pi \cdot m \cdot \cos(\alpha)} \tag{8}$$

If the contact ratio is 1 only one tooth is in contact. This shortens the lifetime of the gears considerably. Usually, a contact ratio higher than 1,2 is desired (Dengel 2021). The contact ratio can be increased with profile shifting, increasing the number of teeth or reducing the backlash.

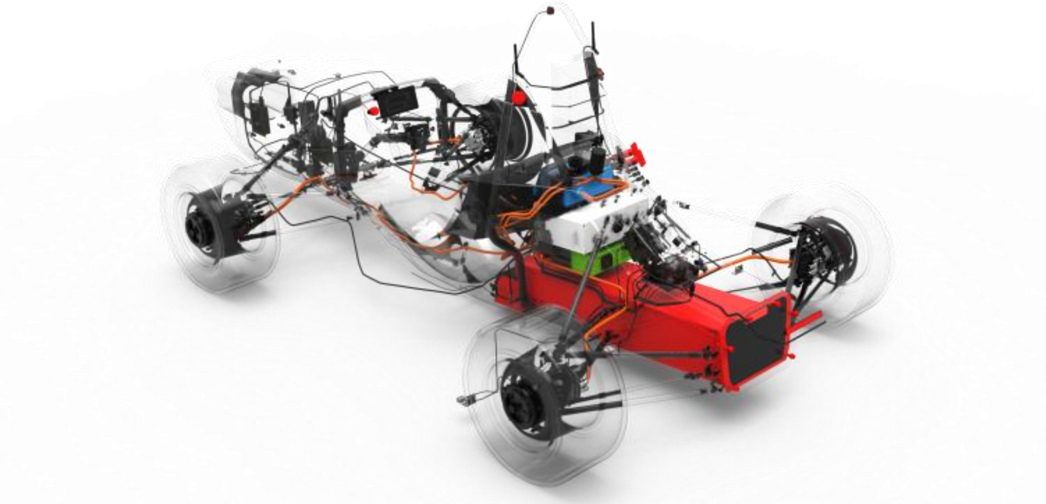
2.3 Powertrains in Formula Student

As mentioned previously, Formula Student cars usually fit into two categories, Combustion and Electric. Combustion cars mostly use gearboxes integrated into the engine itself and a chain drive comprising from front and rear sprocket, chain and a differential. Electric cars on the other hand usually have a self-made gearbox in between the motor and wheel. Some teams have made electric drivetrains with inboard motors that either have gearboxes fitted to them or a chain drive fitted with differential. Usually these are cheaper to make.



PICTURE 3. Inboard drivetrain of HPF019 (Asplund 2019)

Most electric Formula Student vehicles have one motor and gearbox for each of the four wheels. This way the vehicle can achieve greater dynamic performance as torque going to each tire can be individually controlled in what is known as torque vectoring.



PICTURE 4. Four-wheel drive outboard system (Tufast Racing Team n.d.)

Gearboxes used for these types of vehicles are each different, but some types of gearboxes have become more common than others. The simple version of a planetary gear system consists of a sun gear, multiple planet gears and a ring gear.

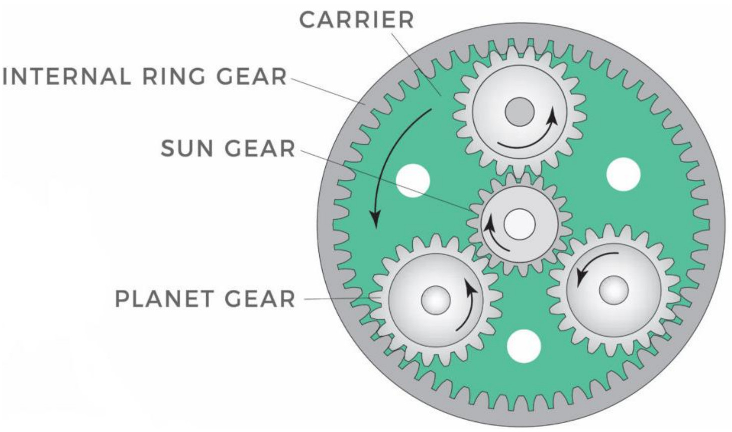


FIGURE 10. Simple planetary gear system (Collins n.d.)

When ring gear is held stationary in the above-mentioned planetary gearbox, sun gear works as input and carrier as output. The gear ratio can be calculated by formula 9 (KHK Gears n.d.).

$$i = \frac{z_{ring}}{z_{sun}} + 1 \tag{9}$$

Maybe the most common type of gearbox used in formula student is a compound planetary with stepped planets. This type of gearbox has input on the sun shaft, the planets on different stages are fixed to each other, the ring gear meshes with second stage planets and is fixed to the casing, and the output of the transmission is planetary carrier.

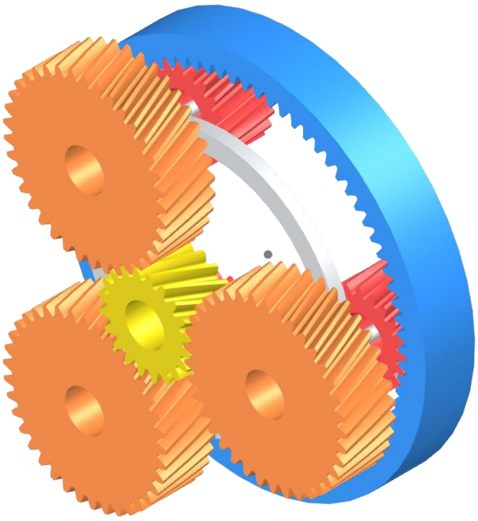


FIGURE 11. Compound planetary with staged planets (Glinsky 2020)

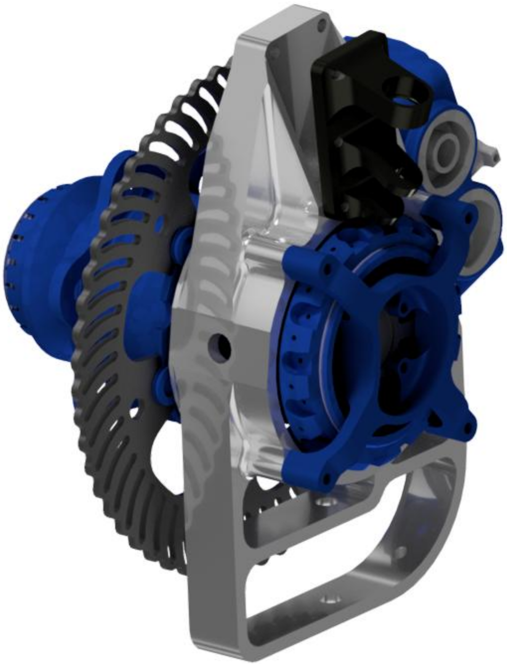
This type of gearbox has the advantage of fitting within a small axial space, for example between the wheel bearings while the ring gear is mounted to the upright. Another advantage is the somewhat high gear ratios achieved as the gear ratios commonly used revolve around 10:1 to 16:1. According to AGMA 6123-B06 the gear ratio of this system can be calculated with formula 10 (Cova 2020).

$$i = \frac{z_{planet2} \cdot z_{sun} + z_{planet1} \cdot z_{ring}}{z_{planet2} \cdot z_{sun}} \qquad (10)$$

3 DESIGN STARTING POINT

3.1 Previous designs

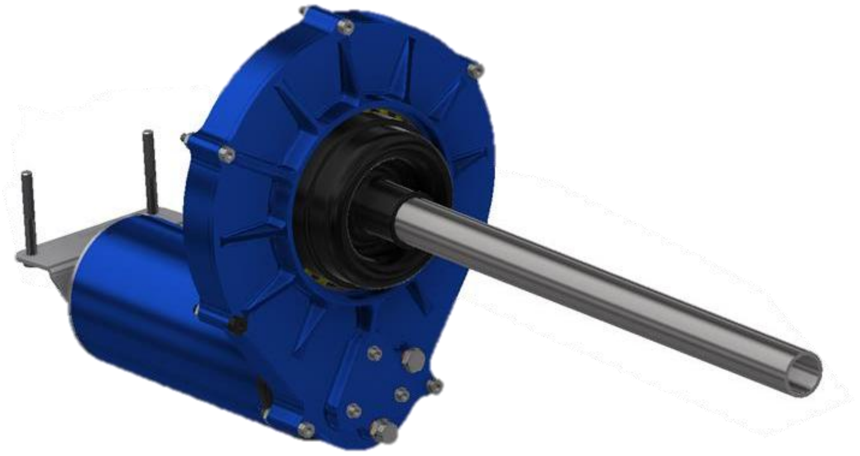
The first iteration of a hybrid powertrain in Tampere Formula Students vehicle was made for TFS22 in 2022. The drivetrain had electric motors directly mounted to front wheel hubs, without any gearboxes. The design was not tested very thoroughly due to problems with the battery package. More thorough investigation after the competitions showed multiple problems with the whole design and prompted an immediate change for a new concept.



PICTURE 5. TFS22 hybrid drivetrain

The cars for 2023 and 2024 already had a different drivetrain concept. The electric motors were mounted under the vehicles frame and the gearboxes attached to the sides of the frame. Power was transmitted to the wheels by means of drive shafts. This drivetrain has the advantage of limiting the extra unsuspended weight added to the wheel carrier package as well as being quite simple in design.

Problems with this drivetrain were mostly in the electrical systems, but also the mechanical side had its disadvantages. The system was heavy, and the gear ratio limited by gearbox size, as the gear boxes only had one gear pair. Also, as the front wheels need to turn, the drive shafts need to have constant velocity joints in the outer ends. This created a problem for hub design.



PICTURE 6. TFS23-24 hybrid drivetrain

3.2 Goals and limitations

As the previous drivetrain concepts have been failures, a completely new design concept is needed. The team has been contemplating a change for an electric vehicle, but the knowledge on electric drivetrains is still lacking. This new concept could have an opportunity to be a new testing ground for the future electric vehicle project.

The main goals for the new hybrid drivetrain are to gain experience with EV-relevant technologies, improved acceleration performance, more compact packaging and lower system mass. These goals are set for the whole hybrid system, but they are relevant in each part.

Main limitations for the project include limited space in wheel carriers, limited or non-existent budget for drivetrains and unknown manufacturing capabilities. The design should also accommodate regular wheel hubs, in case a working hybrid powertrain is not achieved. The already limited space in a 13-inch wheel is further reduced due to a change for a 10-inch wheel.

As the Formula Student series is governed by FSG rules, the system should also be designed to acknowledge these. The rules dictate different things regarding the design, for example the fasteners used for mounting and scatter shield thickness.

3.3 Gear ratio

The gear ratio for the drivetrain is chosen based on lap time simulation and Excel spreadsheet calculations, that consider the different motors and battery packages achievable. The gear ratio choice is not done as a part of this thesis and is therefore only briefly discussed.

The electric motor series chosen is Lehner Motoren Technik LMT 3080, as this motor is compact, available and cost efficient. Also, it has a high current rating, good efficiency and can be customized. The motor series offers a wide range of windings that correspond to different running speeds and torques.

Battery pack is modeled to be made according to FSG rules, which mainly dictate the maximum voltage and cell weight. The design goals have been to maximize the electrical power and energy of the package.

The tire chosen for TFS25 has also a high impact on gear ratio choice. The tire chosen is Hoosier 16x7,5-10 R20 FSAE tire and it has an outside radius of 16 inches.

The battery design gives us the maximum electrical power and the capability of motor controllers is already known. The maximum speed wanted for the car is 110 km/h. With these parameters the optimum transmission gear ratio is chosen to be around 24:1. At the same time the final motor configuration is chosen to be LMT 3080/9.

3.4 Geartrain layout concept

One of the projects’ main goals is to gain knowledge for the possible EV transition in the future, so the drivetrain should serve this goal. Now most teams use a four-wheel drive system with outboard motors and planetary gearboxes in each wheel, and this also seems to be the most reasonable choice for the future of the team. This then limits the drivetrain concept to an outboard system with planetary gearing.

The most often used layout for an electric Formula Student cars gearbox, as stated before, is a dual stage layout with staged planets. This layout is simple to make and consists of very few parts but is limited in its gear ratio for this application. As the needed gear ratio is quite high compared to typical electric Formula Student vehicles, the gearbox would become too big to fit in the wheel carrier packaging.

Basic planetary gear system would be easiest to design and manufacture as it requires the least parts. These gearboxes are quite limited in how big the ratio can be as defined previously. Also, as the gear ratio increases, the size of the gearbox increases.

Preliminary research brought multiple ideas and as the originally planned gear ratio was relatively high, a compound planetary set known as a Wolfrom gearset became a thought option. These planetary gear sets are most used in low power transmissions and robotics, as they can achieve high gear ratios. Wolfrom gearset is relatively like the previously mentioned two stage layouts, but in this case also the first stage has a fixed ring gear, and the second stages ring gear works as the output.

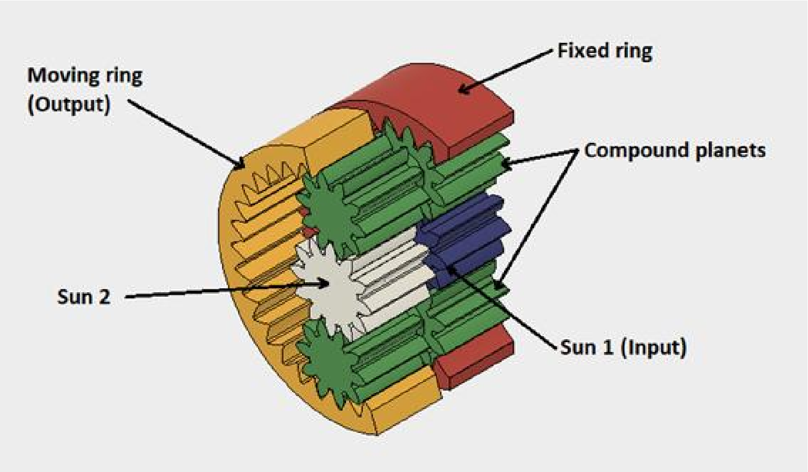
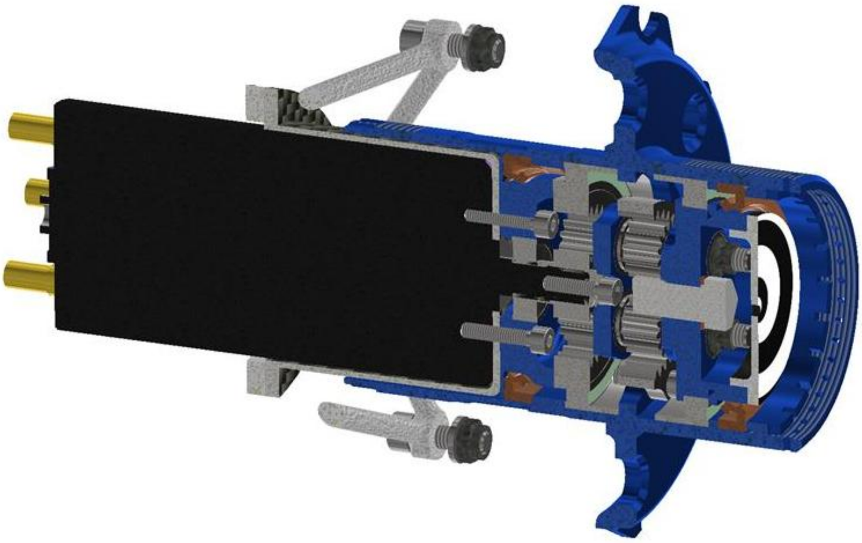


FIGURE 12. Wolfrom gearset (Juangg projects 2018)

The design was started with a Wolfrom set, but as the design went further, efficiency of the drivetrain became a problem. These types of gearings are known for relatively bad efficiency and as it became apparent in this instance, the layout was rejected.



PICTURE 7. Modeled Wolfrom gearset

Ultimately the final layout choice is a dual stage planetary gear system with two independent stages connected from first stages carrier to the second stages sun gear. The first stage sun shaft works as input and the second stages planetary carrier works as the output. The ring gear is fixed to casing.

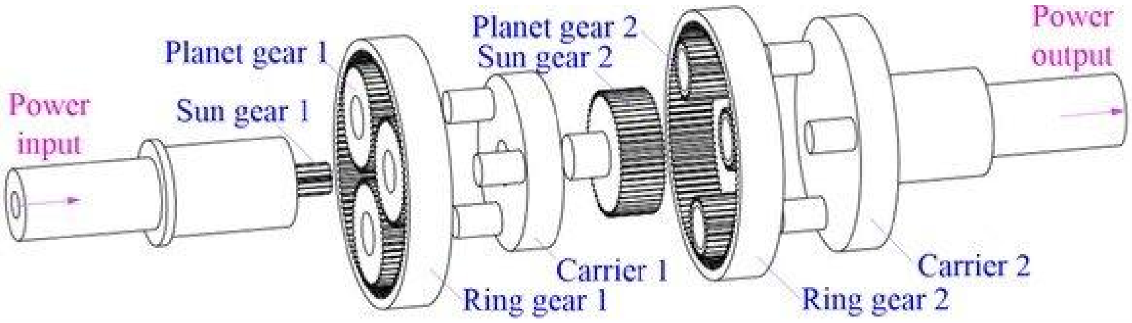


FIGURE 13. Gearbox layout choice (Zhang 2017)

This layout is simple to design, as the planetary stages are working independently. Therefore, the stages could be designed independently, and the second stage can use the first stages output values as input. Downsides are complicated multipart assembly and a long layout. Otherwise, the layout serves the goals and limitations as it can be designed as small in diameter, quite light and to work with a normal wheel hub.

The manufacturing costs can also be limited by creating planetary stages in a way that they have the same gear ratio. Therefore, the ring gear could be a single component spanning the length of the gearbox. Also, the planetary- and sun gears could be with the same profile for both stages and only change in face width. This is not optimal for the gear life, but as the lifetime of a formula student vehicle and for hybrid systems especially is short, this could be possible.

4 DESIGN

4.1 Geometric limitations

The first part of the design is to create a preliminary 3D model of the whole wheel carrier assembly to see how much space would be available for the gearbox. This includes wheel, kinematic points to create a mockup upright, wheel bearings, brakes, and a wheel hub.

Suspension kinematics are designed by the team and can be used for the creation of mockup upright. Carbon fiber wheels have been designed to fit with future electric vehicles, and their design will be a limiting factor. One of the main system goals was to have a wheel carrier package able to perform without the hybrid drivetrain attached to it. For this the mockup model has a preliminary wheel hub.

4.1.1 Wheel bearings

Before the diameter limitations can be measured, the wheel bearings need to be decided on. Due to limited space the choice is SKF super precision angular contact ball bearings. These bearings have a low friction and high tolerance for both axial and radial forces. SKF has a tool for bearing dimensioning that will be used extensively throughout this thesis, but it does not have a calculation module for angular contact ball bearings. For this the dimensioning will be done by hand.

The first part is to calculate the forces affecting the wheel bearings. These forces can be calculated from Forces affecting the wheel, which have been simulated by the team with Optimum G Optimum kinematics forces module. The forces used for the calculation are the maximum values and their effect can be scaled in the process. The forces are calculated for each wheel in multiple different load cases, with the worst being corner brake curb with 2,4 G of lateral acceleration, 4 G of vertical acceleration and -1 G of longitudinal acceleration. These accelerations cause the following forces affecting the wheel contact patch in braking, cornering and the before mentioned extreme load case.

TABLE 2. Wheel contact patch loads

	Vertical load (N)	lateral load (N)	Longitudinal load (N)
Braking	1674	0	2177
Cornering	1527	993	0
Corner brake curb	7871	2634	1463

After the forces have been calculated, the sufficiency of bearings can be determined. SKF has an extensive guide for dimensioning different types of bearings. In this thesis the bearings are mainly calculated by using this tool, but in this case the bearings wanted are super precision bearings and SKF does not provide a calculation tool for these bearings. Therefore, the calculations need to be done by hand.

Firstly, the dimensions of the hub need to be determined, as this will contribute to how the forces affect the wheel bearings. This is done according to figure 14.

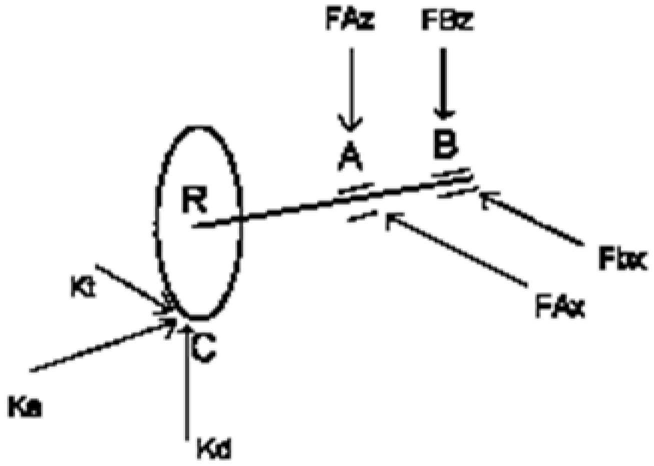


FIGURE 14. Forces affecting wheel bearings

As seen in the picture above, the lateral force is shown as K_a, the vertical force as K_d and longitudinal force as K_t. R is the wheel center point that is used as a reference point. A is the location of first bearing and B is the location of the inner bearing. Forces affecting them are also shown in the diagram. The preliminary dimensions used for the calculation of hub assembly are as follows. Distance AB is 45 mm, distance RA is 77,5 mm and distance RB is 122,5mm. The tire used in the TFS25 vehicle is Hoosier 16x7,5-10 R20 FSAE tire that has an outside diameter (D_w) of 406,4 mm and the calculations will use the radius of the tire (R_w).

At this point the forces can be calculated the forces are calculated for each of the determined load cases and the highest bearing forces are taken for bearing calculations. load affecting bearing A from upside view can be calculated with formula 11.

$$F_{AX} = \frac{K_t \cdot RB}{AB} \tag{11}$$

The load from side view is calculated similarly with the formula below.

$$F_{AZ} = \frac{K_a \cdot R_w - K_d \cdot RB}{AB} \tag{12}$$

The forces affecting bearing B are calculated with the following formulas. Load from upside view with formula 13 and from side view with formula 14.

$$F_{BX} = K_t - F_{AX} \tag{13}$$

$$F_{BZ} = K_d - F_{AZ} \tag{14}$$

The results of these calculations for the three determined load cases are shown in table 3 below.

TABLE 3. Calculated bearing forces

$F_{AXcorner}$	0 N	$F_{AZcorner}$	327 N
$F_{AXbrake}$	5926 N	$F_{AZbrake}$	4557 N
F_{AXCBC}	3982 N	F_{AZCBC}	9532 N
$F_{BXcorner}$	0 N	$F_{BZcorner}$	1199 N
$F_{BXbrake}$	3749 N	$F_{BZbrake}$	2883 N
F_{BXCBC}	2519 N	F_{BZCBC}	1661 N

Resultants can be calculated with formula 15.

$$F_{corner} = \sqrt{F_{Xcorner}{}^{2} + F_{Zcorner}{}^{2}} \tag{15}$$

The calculations are done for all load cases and both bearings. The results can be seen in table 4.

TABLE 4. Calculated force results

$F_{Acorner}$	327 N		$F_{Bcorner}$	1199 N
F_{Abrake}	7475 N		F_{Bbrake}	4729 N
F_{ACBC}	10331 N		F_{BCBC}	3018 N

The biggest radial force (F_r) is shown in the table above as 10331 N. The biggest axial force (F_a) as shown before is 2634 N. These values can be used in the following life calculations.

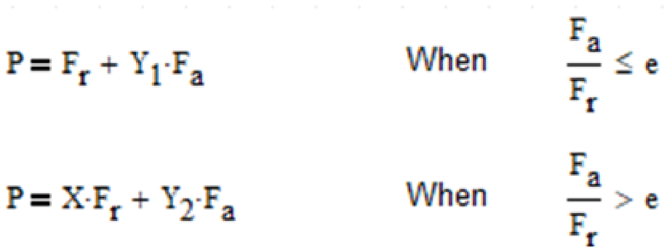
Preliminary bearing was set to SKF 71816 ACD/P4, as the previous cars have used similar bearings with success. These bearings are also ideal size for the assembly. The dimensions of the bearings are inner diameter (d) 80 mm, outer diameter (D) 100 mm and width (B) 10 mm. Basic dynamic load rating (C) is 14,6 kN and basic static load rating (C_0) is 18,3 kN. Axle preload is set to class C of 212 N for this bearing. Therefore, total axial force (F_{areal}) is the sum of this and maximum force which comes to 2846 N.

Life calculations are done according to SKF bearing rating life calculation section, and therefore all formulas are supplied by them. The calculation factors needed are also given by SKF and are as follows.

TABLE 5. Calculation factors for angular contact bearings (SKF n.d.)

e	0,68
X	0,67
Y_1	0,92
Y_2	1,41
Y_0	0,76

For the calculation of equivalent dynamic bearing load the following picture shows the correct formula.



$$P = F_r + Y_1 \cdot F_a \qquad \text{When} \quad \frac{F_a}{F_r} \leq e$$

$$P = X \cdot F_r + Y_2 \cdot F_a \qquad \text{When} \quad \frac{F_a}{F_r} > e$$

FIGURE 15. Equivalent dynamic bearing load calculation. (SKF n.d.)

in our case the upper formula is used. The calculation of the equivalent dynamic load gives the result of 12494 N.

SKF gives a formula for calculating the bearing life in millions of revolutions according to ISO 281. With minor modifications the formula can be modified to tell the driven length in kilometers. The final formula is as follows.

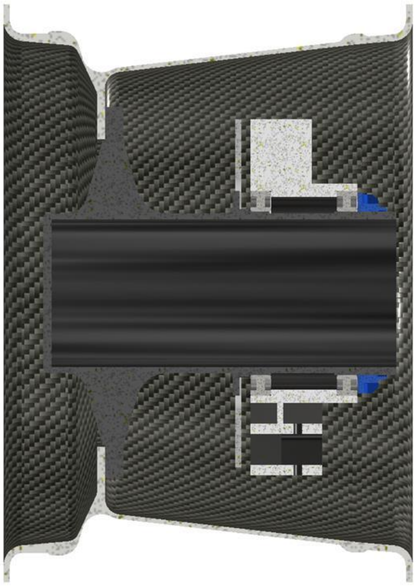
$$L_{10} = \left(\frac{C}{P}\right)^{p} \cdot 10^{6} \cdot (2 \cdot \pi \cdot R_w) \cdot 10^{-3} \tag{16}$$

The total driven distance comes to 2025 kilometers. This distance is much longer than the needed, but as the pretension is not always checked accordingly and other factors can shorten the lifecycle, these bearings are deemed good for this case.

4.1.2 Preliminary layout model

With the now determined wheel bearings, the rest of the wheel carrier package can be predesigned. The upright needs to be able to accommodate all the suspension kinematic points. Wheel hub works as a connection point between the upright and wheel, and in this case will be extremely long as the suspension kinematic points are located towards the inner edge of the wheel. For the gearbox this is ideal as the layout of the gearbox is inevitably long.

FSG rules bring a limitation for gearbox location as it is stated that nothing can extend outside of the outer edge of the wheel. (FSG rules 2025 T2.1.3). At this point the correct length dimensions for the gearbox and hub are not critical and the more important part is to know the maximum usable outer diameter. With considering the determined limitations and rules the preliminary model can be made and needed limiting dimensions measured.



PICTURE 8. Preliminary layout

With this model the limiting dimension can be determined. The main limiting factor is the maximum diameter of the geartrain assembly, and in this case the inside diameter of hub is 72mm. Lenth of the hub is 168 mm, but this should not limit the gearbox design as the electric motor can extend inwards from the hub. The inside diameter of 72mm should also accommodate at least 1mm of radial clearance between gearbox and hub.

4.2 Gear life calculation

The first part of gearbox design is determining the gear profiles. This step sets the starting point for the whole gearbox design as the design cannot exceed without knowing the dimensions of the gears. Gear profile design is important for a working gearbox that can withstand loads applied to it and perform in competitions.

Gear life calculation can be done by hand calculations, but for more complex assemblies like planetary gear trains, simulation tools can be used. As the geartrain itself is complex and needs to be very limited in size the design is chosen to be done with simulation software. For this a machine element design tool KISSsoft is used.

KISSsoft is an extensive software that can also be used for other purposes like axle, bearing and sealing design. In this thesis these features will not be used and all calculations for these components will be done with different tools and hand calculations. Also, for more complex assemblies KISSsoft offers different modules like KISSsys that can easily model extensive geartrains with casings, bearings, axles and even shifting systems.

4.2.1 Calculation setup

KISSsoft can simulate a single gear pair, or more complex gear sets. For this research KISSsoft offers a readymade planetary gear calculation module. As the planetary system considered in this work consists of two gear sets only connected by axle from the first carrier to second sun, they can be simulated independently from each other.

The goal of this part is to design gearsets that can use the same ring gear, so it can be manufactured as a single piece part. If other gears in the system could be made with same profiles between stages, it would be ideal for manufacturing purposes as the number of different parts is reduced. This design choice has its disadvantages with the gear ratio being the same on both stages. Therefore, the second stage cannot have more planets than the first stage due to space limitations, and additional strength must be made with increased face width. This adds extra weight to the gear set, but as the gears are quite small, the added weight should not be significant.

The first step to gear calculation is setting the base values for the system. This includes the dimensions of gears, number of planets, gear quality, material and lubrication. Some of these are hard values that cannot be compared with one simulation and in that case need to be changed between simulations.

To reduce axial forces and manufacturing expenses the gears are chosen to be simple spur gears and therefore have no helix angle. The normal pressure angle of 20 degrees is chosen for better efficiency. Also, as it is the standard pressure angle it is cheaper to manufacture.

The number of planets is set to 3. More planets could be more beneficial for load distribution as there would be more teeth in contact at any point in time. Due to the minimal size of the gearbox housing it is unfortunately not possible to fit extra planets with the desired gear ratio.

Quality level according to ISO 1328:1995 of the gears is preliminarily set to 6 as the manufacturers capabilities on this are unknown at this point. When the manufacturer of the gears is clear, the simulations should be redone with correct quality settings to make sure they withstand the forces applied. At that point final gear profile changes can be made.

A case hardening steel 18CrNiMo7-6 is chosen for all gears as it is widely available in different sizes. It also has sufficient material properties for gears. The properties can be seen in table 6.

TABLE 6. 18CrNiMo7-6 material properties (Fuhong forge n.d.)

Tensile strength	1200 MPa
Bending fatigue limit	580 MPa
Fatigue life (x10^6	100
Core hardness (HRC)	35-45
Surface hardness (HRC)	58-63
Youngs modulus	210 GPa

As the gearbox is fitted inside a wheel hub it is impossible to make a spray lubrication system. For this the lubrication is set to oil bath lubrication like it will be in the final design and oil type is set to ISO-VG (68). This choice is later reviewed in section 4.5.

Reference profile of the gears is set to ISO 53:1998 profile A. With this profile the Dedendum coefficient is 1,25, addendum coefficient is 1, and root radius coefficient is 0,38. The tooth thickness tolerance is set to DIN 3967 cd25, as it is suitable for small modules ranging from 0,5 to 3.

As the previous hybrid systems have not been tested extensively, a real-life load spectrum is missing. For this the simulations are done with standardized load spectrum inside KISSsoft that scales the peak load values for torque and rotational speed. As the battery capacity of the hybrid system is limited, it is not driven continuously. This limits the experienced loads and usage times. For this the service life has been set to 30 hours. As the loads are not always uniform and the real load case is unknown, the application factor is set to 1,25.

Other factors affecting root and flank calculations are set according to calculation method. The calculation method for factors is set to AGMA 2101-D04 (metric), and Tooth flank fracture is set to ISO/TS 6336-4. Micro pitting is calculated according to ISO/TS 6336-22 and reliability calculation is set to be done by Bertsche method.

Finaly the wanted gear ratio can be input. In this case the wanted gear ratio for one stage is 4,9:1 so the total gear ratio would become around 24:1. A 5 percent allowed deviation is set for the nominal ratio. The normal module is set to be between 0,8 and 1,2. Any smaller gears than this became hard to manufacture with reliable tolerances and larger teeth would result in gears not fitting inside the specified dimensions. Additionally, planets are set to be evenly pitched and the minimum distance between two planets is set to 0,5 mm.

4.2.2 Results

After the simulations, the following results were chosen for gears. Normal pressure angle of 20 degrees was chosen for manufacturing and strength purposes. A normal module of 0,8 was chosen as a larger module would have made the ring gear extend beyond the determined maximum diameter, and smaller gears would be hard to manufacture with acceptable tolerances. The final center distance between planets and sun is 17,5 mm. Main dimensions are shown in table 7, and other manufacturing data can be seen in appendices 1, 2 and 3.

TABLE 7. Dimensions of designed gears

	Sun	Planet	Ring
Number of teeth	18	24	69
Face width stage 1	11 mm	10 mm	“11” mm
Face width stage 2	25 mm	24 mm	“25” mm
Profile shift coefficient	0,5129	0,4833	0,0684

The ring gear is a lot larger than indicated by the table as it is made in one piece, but in relation to the other gears it is only 0,5 mm wider in each direction. The sun and ring gear a little wider than planets to reduce bending stresses. As the number of teeth on sun and planets is divisible by a common number, they can be assembled freely in any relation to each other. This makes the assembly process easier as the gears don’t need to be aligned. Manufacturing can also be made cheaper as the gears don’t need any aligning marks and can be, for example, made with only EDM cutting.

The gear ratio of 4,9:1 was not achieved with these results, but the gear ratio resulted in 4,83:1 for one stage. This is acceptable as maximum speed achievable does not become too slow. With the acquired gear ratio for one stage the total ratio is 23,3:1.

When the gearbox is exposed to forces the teeth are slightly deformed. This can cause the teeth to make contact in wrong places, even though the simulation doesn’t show this. This effect can be prevented with profile modifications. These modifications usually target the tip and root of the teeth but in this case only tip relief is discussed and applied. A well-designed tip relief can reduce irregularity of forces and therefore increase efficiency, noise and reliability of the gear system. The tip relief is usually applied to both driving gear and driving gear as it can affect impacts on entry and exit of the tooth contact.

Optimal tip relief is calculated with KISSsoft and in this case the calculation method is chosen as linear. Finaly a tip relief of 3 µm is chosen for sun and planets and a tip relief of 2 µm is chosen for ring gear.

KISSsoft is also able to show a visual representation of the gears meshing. Below the meshing of sun and planets can be seen in figure 16 and meshing between planets and ring can be seen in figure 17.

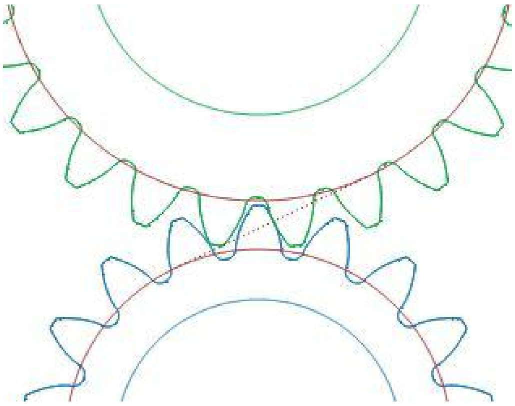


FIGURE 16. Sun and planets meshing

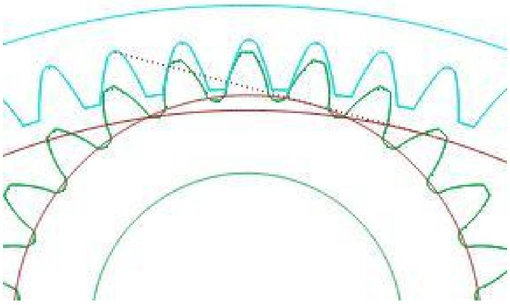


FIGURE 17. Planets and ring meshing

For safety side, the simulations can show the safety factors for root and flank safety. These results can be seen in table 8.

TABLE 8. Calculated safety factors

	Sun	Planet	Ring
Root safety stage 1	1,612	1,727	1,524
Root safety stage 2	1.034	1,101	1,009
Flank safety stage 1	1,388	1,433	2,721
Flank safety stage 2	1,034	1,068	2,040

From the result it can be seen especially the root safety for stage 2 are quite low. This is expected due to limitations in design from the two stages being made with the same geometry and the size of the gearbox. These safety factors, even though small, can be accepted as all of them still exceed the minimum of 1.

A case hardening depth of 0,16mm was chosen for all gears after a proposition by KISSsoft according to Niemann.

Contact ratio for sun gear and planet gear contact is 1,269 and for planet gear to ring gear 1,546. The planet to ring contact ratio is good but as expressed in chapter 2, the larger the contact ratio, the better. This makes the contact ratio of sun to planet contact lie on the edge of acceptable as the minimum contact ratio accepted would be 1,2. This is not ideal for the load distribution on teeth, but as it is over the minimum value and the life cycle of the gearing is not long, it can be accepted.

The software can also estimate the gearboxes’ efficiency with the given parameters. These results can give a valuable estimate on the direction of the efficiency. The results should not be considered as absolute truth as the gearbox casing and therefore the amount of oil and its behavior is missing from the calculation. The simulated efficiency for the system is 0,96. For more accurate reading the gearbox could be further simulated with different calculation modules. This could give an estimation in a graph format where efficiency could be read in different conditions.

Finaly KISSsoft can show a 3D representation of the gearing. This geometry can be directly exported in a 3D format for later design work. This representation can be seen in the figure 18.

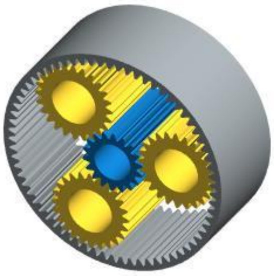
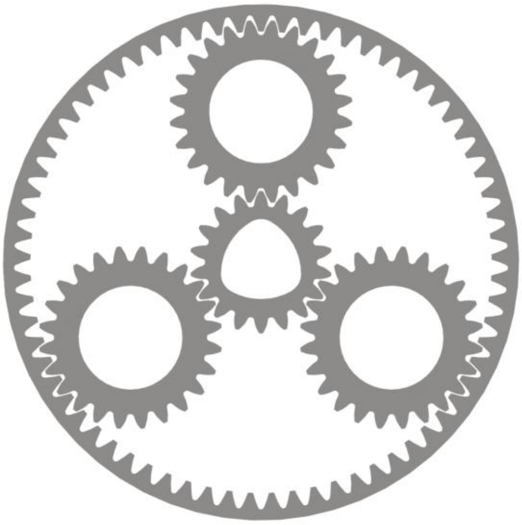


FIGURE 18. 3d representation

4.3 Mechanical design

After the final gear profiles have been determined, it is time for the mechanical design of the gearbox. The design starts with the design of the individual planetary stages and extends to casing and mounting from there. As the gears have the same profile on both stages, they use a same base model where just the face width is changed. The base planetary stage model can be viewed below.



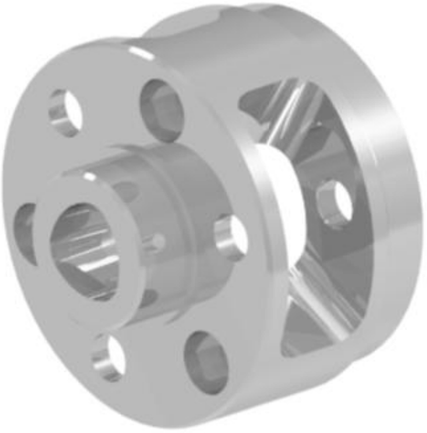
PICTURE 9. Planetary base stage

4.3.1 First stage

First stage is the nearest to the electric motor. As the sun gear will work as the input gear, it needs to be connected to the electric motor. This will be done with a custom P3G shape lock axle that will be added to the motor by the manufacturer. This shape lock is optimal for this use as it almost eliminates stress risers in the connection due to its rounded shape. Axial locking for the sun gear is done with washer that is locked on with M4 socket head bolt. Axle surface will be machined to manufacturers specified tolerance of g6. The other end will have an indent for an encoder magnet.

PICTURE 10. P3G axle

The planetary carrier is made from two independent pieces, for ease of assembly and simplicity of the gear connections. Carrier is made from aluminum to reduce the weight of the system. Carrier is split from middle and the only difference between the two parts is the bearing placements and P3G locking shape on the outer part. The outer part has a seat for bearing connecting the carriers together and the inner part has a seat for supporting bearing of the whole assembly.



PICTURE 11. First stage carrier

Gears are mounted to carriers with steel studs and between the studs and gears are HK0801 needle bearings. There are also brass shims on each side to work as sliding surfaces. These shims will also reduce wear on the aluminum carriers. Studs have holes going through and a radial hole in the middle to supply oil to the needle bearing. Studs have smaller diameter on both ends that fit to the holes on carrier to lock it on place.

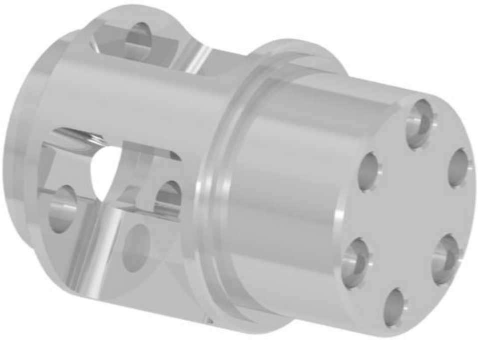
PICTURE 12. First stage planet assembly

The first stage carrier assembly is completed by putting all the parts together and then locking the carrier pieces together with three M5 bolts. Carrier pieces have indents to fit these bolts inside the bounding box to reduce the size of the assembly.

PICTURE 13. First stage

4.3.2 Second stage

The second stage consists of similar components to the first stage, but due to the carrier mounting to the hub, it needs to be made from one piece. Carrier has an extension piece that works as a bearing- and sealing surface. The two pieces are mounted together with the same bolts that connect the carrier to the wheel hub. Planet studs go into their places from through holes under the extension piece and are locked into place by the said extension. Carrier extension has extra threaded holes for oil change purposes. These holes are closed with bolts and O-rings to prevent leakages.

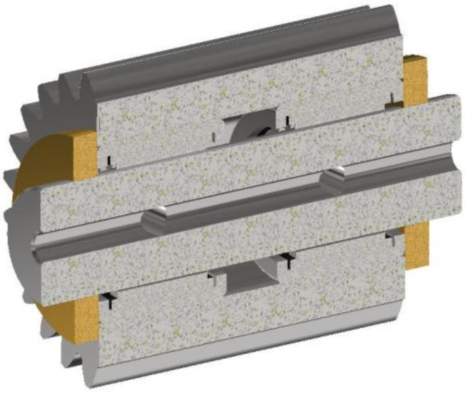


PICTURE 14. Second stage carrier

Input to the second sun gear is done with a small axle connecting to the first carrier. The axle has shape lock connection for both components and as the first stages carrier is made of aluminum and the sun gear from steel, the shape lock for carrier will be larger in diameter to prevent deformation. The axle itself is made from steel, and between the shape lock surfaces is a limiting part to keep it locked in its correct position. The outer end of the axle has a seat for bearing that limits movement in axial direction.

PICTURE 15. Stage connection axle

As the planet gears on the second stage are longer than the first stage, they have two needle bearings inside. This will reduce the misalignment angles affecting the gears. Studs have similar design to first stage by having oil supply holes, but the outer profile is straight due to different mounting method on carrier.



PICTURE 16. Second stage planet assembly

Carrier assembly is finished by putting the gears and shim plates on their places and sliding studs onto their holes. The extension piece is then added to prevent pieces from falling out. extension piece does not mount with bolts at this point, but the clearance between pieces is designed to hold them together with sufficient force.

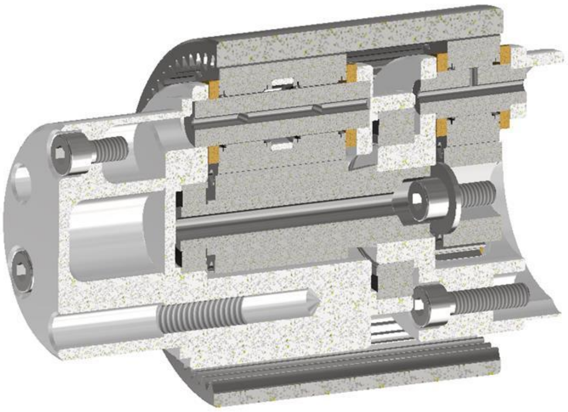
PICTURE 17. Second stage

4.3.3 Casing and mounting

Carriers attach with a deep groove ball bearing. This bearing lets the carriers spin around their axis at different speeds but supports them and the whole assembly. The bearing is not designed to experience high axial loads, as the gears are straight cut, and axial forces are therefore limited to minimal. The bearing chosen is SKF 61804 which can be acquired with different sealing solutions if needed

PICTURE 18. Both stages assembled

As already stated, one of the advantages of making the different stages from same gears is the ring gear. Due to this point of design, the ring gear profile is same for both stages and can be made from single piece. This makes it easier to mount the ring gear and requires fewer parts. A half section view of the completed carrier assembly with ring gear can be seen below.



PICTURE 19. Ring gear

The ring gear is stationary, while the carriers spin at different speeds. This means that the ring gear needs to be rigidly mounted to gearbox casing. The casing needs to eclose the whole geartrain, as it is inside the wheel hub, that spins with tires. The casing also holds the gearbox oil, so it needs to be sealed. Backside of the wheel hub will be open, so the casing can extend beyond the hub for mounting on the stationary upright. Material choice for casing is aluminum, as the diameter is large enough to withstand the torsion created by gearbox even with thin walls.

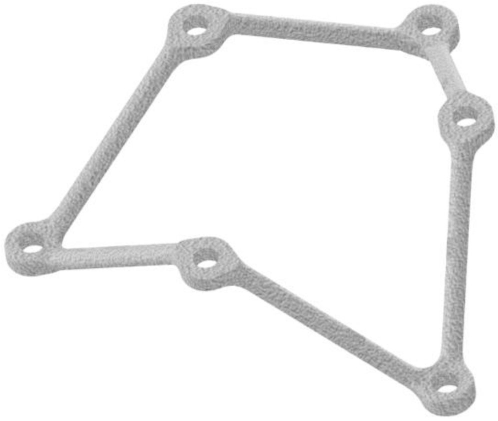
As the electric motor itself is not sealed for oil contact and the gearbox needs to be leakproof, the hole where the motor axle goes through needs to be sealed. This is done by creating a seat for a small oil seal in the wall between the gearbox and electric motor. This wall also works as the connection point for the motor, which is held in place with four M4 bolts. The ring gear is mounted to the casing with six countersunk M4 bolts, to keep the diameter of assembly as small as possible. The mounting point is in the middle of the connection between stages, so holes in the gear profile are not a problem.

PICTURE 20. Gearbox casing

To keep all parts easy to manufacture, the mounting of casing to upright is done with a separate component. This brings an additional connection to the assembly, but small misalignment in mounting should not cause any issues. Another reason for this design choice is the hub design that means the mounting must go around the hub nut.

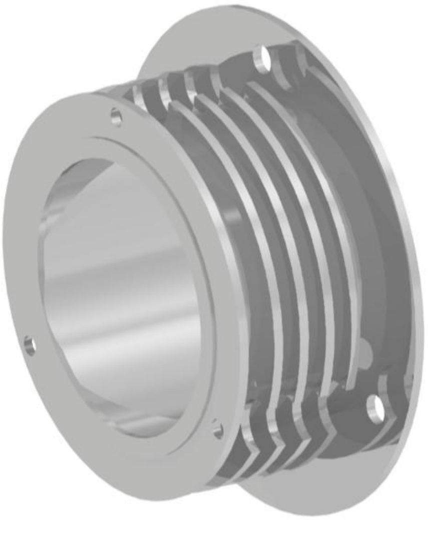
With traditional manufacturing methods the mount would become hard to manufacture, heavy, and unnecessarily large. For this reason, the mount is designed to be 3D-printed. This way the mount can be designed with a level of freedom not achievable by milling or turning. The mount has a spider-like design with three mounting points on both sides and rods between them. By itself the mount is fragile, but when connected to rigid parts like casing flange and upright, it becomes quite structural.

The torque affecting mounting points is not large enough to require M6 bolts, but the FSG rules state that all drivetrain shields and guards need to be mounted with 6 mm metric grade 8.8 or stronger bolts (FSG rules 2025 T 7.3.2). As the casing and mounting also works as an electric motor scatter shield, it needs to be mounted in a way that satisfies this rule.



PICTURE 21. Mounting spider

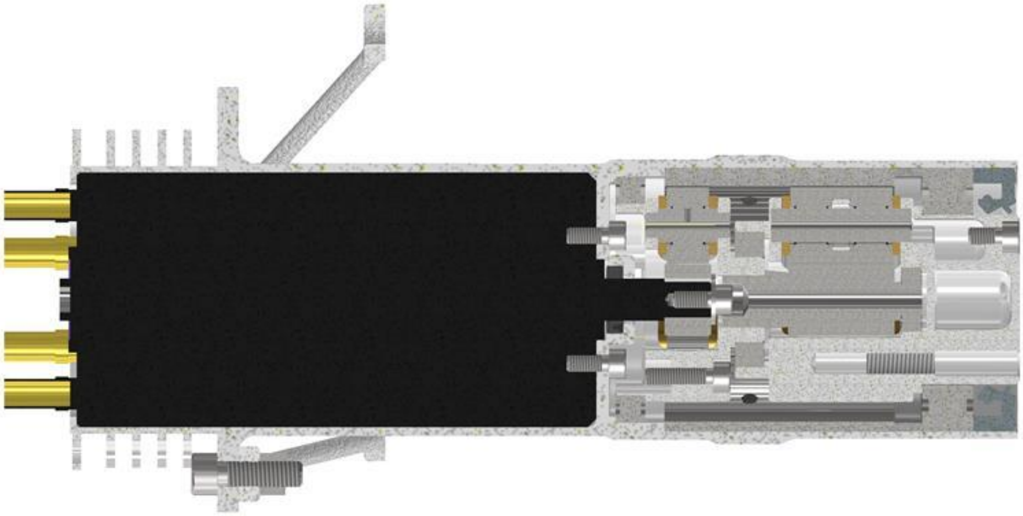
The final part of the gearbox casing is a cooling jacket for the open end of electric motor. This component is not a critical part of the structure but helps in keeping all parts away from dirt and moisture. The cooling jacket is designed to have fins on the outside surface that help in exchanging the heat from the motor. It is mounted with the same bolts that connect the casing to the mounting spider.



PICTURE 22. Cooling jacket

The whole gearbox can be assembled before connecting it to the hub or upright. Assembly starts by inserting the oil seal between compartment in its seat and sliding the motor into its place. Now the four connection bolts can be attached and fastened through the wall. The FSG rules state that all electric motor mountings must meet T10 (FSG rules 2025 EV 2.1.2). Section T10 of FSG rules Consists of rules for fasteners and in this case the rules state that motor mountings need to be critical fasteners. This rule will be satisfied by installing a safety wire between the four bolts.

Carriers themselves are supported from the inner end by a ball bearing that is inserted before carriers. This bearing is SKF 61809. After this has been mounted, the ring gear is inserted in its place and fixed with countersunk bolts. Thread locker is applied to the bolts, to limit possible failure points. After the ring gear is mounted, the already assembled first stage carrier assembly can be slid onto the ring gear and the support bearing. Before attaching the second stage to the first, sun gear needs to be fixed by a bolt onto the motor axle. Then the second stage can be mounted and carrier extension put to the end. Finaly the outer support bearing SKF 61908 is mounted, and the oil seal is placed after it. Last part before assembly to the wheel carrier is attaching the cooling jacket and mounting spider to the casings mounting flange. Below is a picture of the completed assembly. Lastly the correct amount of oil is applied through one of the oil change holes and bolts sealed from leakages. An exploded view of the gearbox can be seen in appendix 4 and parts list with main components in appendix 5.



PICTURE 23. Gearbox assembly

At this point the gearbox assembly can be mounted to the hub. This requires some persuasion as the carrier extension can be misaligned with the carrier. After the mounting bolts have been tightened, the package can be fastened to the upright. Finaly the whole package is assembled and ready to be attached to the vehicle.

PICTURE 24. Gearbox assembled with a wheel carrier

4.4 Structural design

Structural design of all components is important. This step can show if the components are too weak to withstand the forces affecting them and changes can be made. Also, if done correctly, the components can be optimized to be light while having acceptable stress and deformation under maximum loads. Also, material fatigue should be considered as a factor in the structural design process.

This section shows the structural design process of the geartrain components, without going into detail on every component. The design process is similar on most of the components and can be expected to follow the principles shown on a few of the components.

4.4.1 Hand calculation

For bolts the calculations are done manually by hand and the source for strength properties of metric bolts is Eurocode Applied. For more complex parts finite element analysis is used.

There are three sets of critical bolts in the system that have high loads applied to them. These bolts are the ones connecting the second stage carrier to hub, the ones connecting the casing to spider mount and the ones connecting the complete package upright. As all bolt sets are similar with all of them having the same number of bolts, the same calculation can be applied to them. The shear stress applied to each bolt is calculated with formula 17.

$$\tau = \frac{\frac{\frac{Torque\ applied}{Radius\ of\ bolt\ pattern}}{Shear\ area\ of\ threaded\ part}}{number\ of\ bolts} \tag{17}$$

The calculation is based on the situation where all the bolts have been loosened, which is the worst-case scenario. The radius of carrier bolt pattern is 15 mm, the radius of spider pattern is 41 mm and the radius for upright mounting bolts is 63 mm. The theoretical maximum torque applied to the bolts is 165 Nm. The Shear area for bolts threaded for M6 bolt is 20,1 mm^2 (Eurocode Applied n.d.)

With these values the shear stress affecting each bolt can be calculated. For the carrier set a stress applied to each bolt is 182,5 MPa, for the spider set it is 66,7 MPa and for the upright set it is 43,5 MPa. If the bolts used are grade 8.8 metric bolts, the yield strength of the bolts is 640 MPa. Also, Eurocode Applied gives us shear resistance of the bolts and for M6 grade 8.8 metric bolt this is 7,72 kN. From the previous calculation the maximum force for the carrier bolts can be calculated and it is 3,67 kN. Therefore, the bolts still would have a safety factor of 2,12. All the calculated values are acceptable.

Planet studs on the first stage are subjected to lower forces as the gear ratio at this point is only 4,82. This means that the max stress affecting a first stage stud is only 20 MPa. For the second stage, as the gear ratio is higher the stress rises to 195MPa. The stress on second stage planet studs is quite high, but as the lifetime of the gearbox is not expected to last more than 30 hours, it is not unacceptable.

4.4.2 Finite element analysis

Finite element method is a mathematical method of predicting the stress and deformation that components experience when loads are applied to them (Ansys n.d.). With finite element method it is possible to simulate complicated structures without high expenses or work that goes into building prototypes. For this reason, it is used extensively in this work to design components with lightweight design while remaining strong.

With finite element analysis it is possible to analyze complex structures with complex loading and supporting. This also means that the results are only as good as the input that the user gives the software.

In this work the simulations are done for smaller assemblies. The simulated assemblies are first stage carrier and connecting axle with second stage sun, Second stage carrier and connection to hub and gearbox casing with spider mount.

The first assembly for simulation is the first stage. The simulation model has bolts connecting the carrier pieces together. The torque of 27 Nm is applied to the carrier from planet stud holes. The assembly is supported with cylindrical support from bearing locations on both sides of the carrier and on the outer end of the connecting axle. The assembly is fixed in place from two of the sun gears teeth to show the correct results for other parts as the tooth profile has been validated before. Al contacts between parts that can slide in relation to each other have been set to frictional. the coefficient t of friction between steel and aluminum is set to 0,2 and for aluminum-aluminum contact is set to 0,3. (Engineering toolbox, n.d.). Also, a preload of 9000 N is applied to the three mounting bolts. The simulation setup can be seen in figure 19.

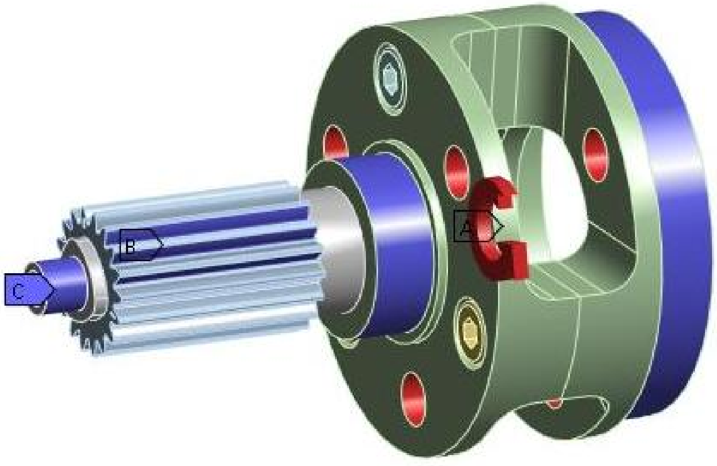


FIGURE 19. First stage setup

Normal mesh size for elements is set to 1mm and mesh refinements are set for the shape locs as 0,2 mm. The mesh and refinements can be seen in figure 20.

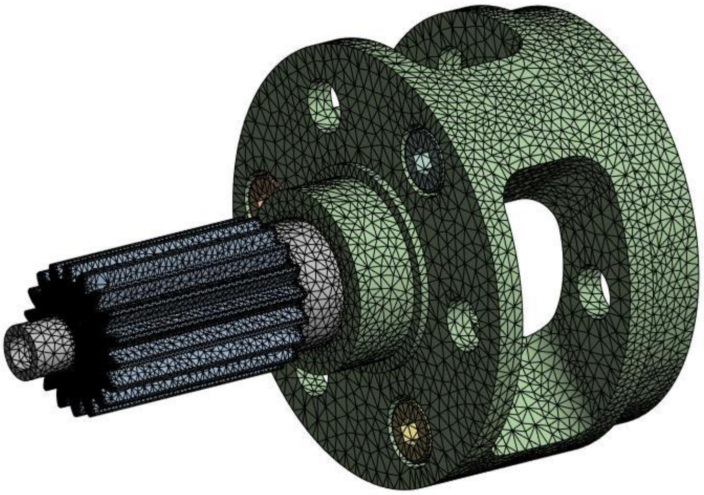


FIGURE 20 First stage mesh

After the setup the simulation can be done. Simulations show the result for multiple requested things, but in this case only the equivalent stress and maximum deformations are considered. Below the maximum stress can be seen.

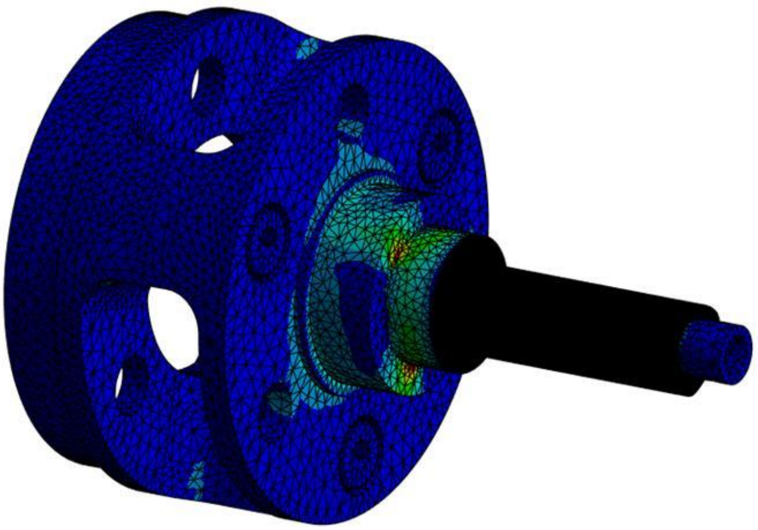


FIGURE 21. First stage maximum stress

Maximum stress on the entire part is 263 MPa. The stress is on top of the P3G triangle shape on the corner. Other parts of the carrier have more conservative stresses. As this is the only high-rise stress point it can be considered acceptable and a changeable component if needed. The first stage sun connection P3G is not simulated as it is made as a same profile as the second stage sun.

The second simulated assembly is the second stage with the hub connection. The simulation is done with similar mesh setup and connection setup. The assembly is set to be cylindrically supported from bearing surfaces and to be fixed in place from hub. The connection bolts have a pretension of 9000N.

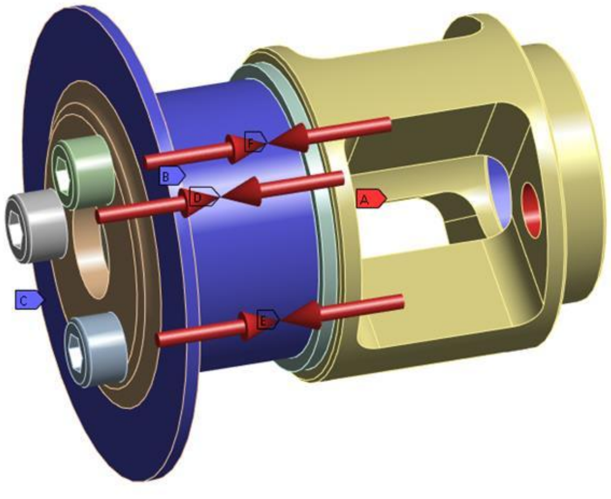


FIGURE 22. Second stage setup

As with the previous stage, the mesh is set with a similar setup. The torque is set to come from planet studs, and the torque is 165 Nm. The stresses can be seen below.

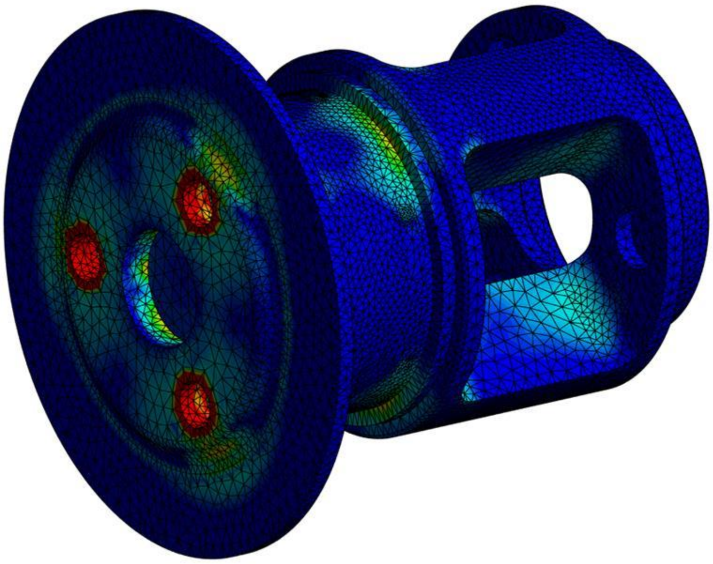


FIGURE 23. Second stage maximum stress

The maximum stress is concentrated in the connecting points as expected and it is 263MPa. This is high, but as the material is aluminum 7075 T6 it is not as high as the yield strength it is not unacceptable. Otherwise, the assembly does not have any high stress points that would need to be concentrated upon.

The final simulation concentrates on the casing and mounting. The setup follows the same path as previous simulations. The support is set from ring gear mounting holes and the torque is set from the spider mount upright holes. Assembly is cylindrically supported from the bearing surfaces of the casing. Mounting bolts between the spider and casing have a pretension force of 9000 N. The setup can be seen below.

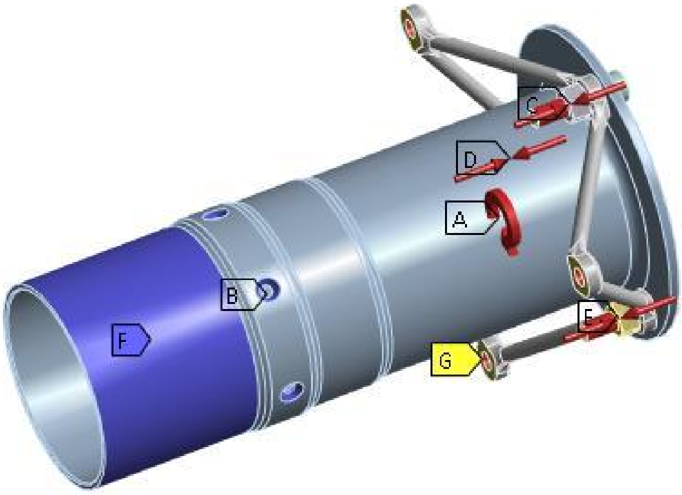


FIGURE 24. Mounting setup

As shown in the picture above. The assembly is also supported with directional support from the spider legs as they mount onto the upright. The mesh and connections are set similarly to previous simulations. The simulation results for stresses can be seen below.

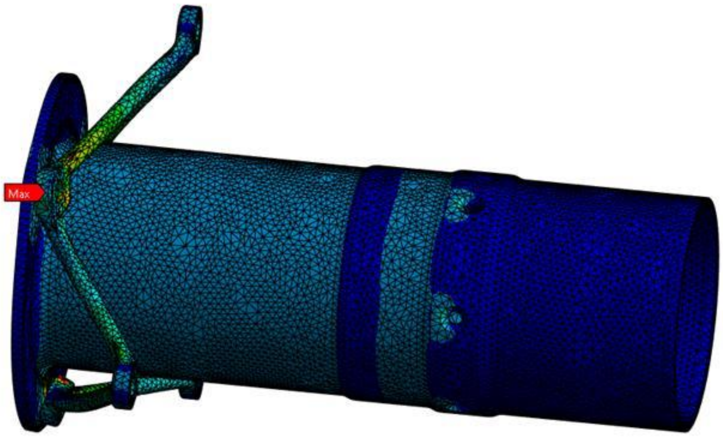


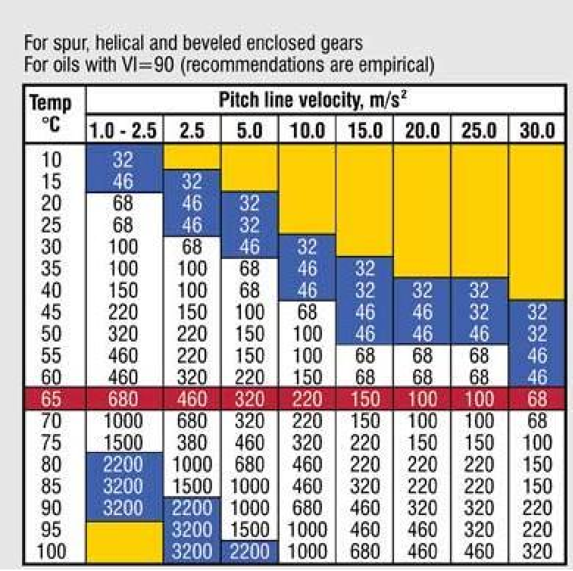
FIGURE 25. Mounting maximum stress

The maximum stress applied to the mounting and casing is in the mounting point between the two parts. As the maximum stress is still under 200 MPa and otherwise the structure does not experience any high stresses, they can be accepted.

4.5 Lubrication and sealing

The gearbox oil has a few different duties. These include the lubrication of gears and bearings, dirt removal and protection of corrosion. Normally the thickness of lubricant film on gear teeth can be thought of as being adequate if it is thicker than the average surface roughness. A high viscosity for oil is often chosen to have adequate thickness at lower rotational speeds. Resistance of gears for scuffing and pitting is also improved with higher oil viscosity. The downside of higher viscosity is the increased frictional losses and therefore overall gearbox efficiency is reduced.

The peripheral velocity for second stage is 34.2 m/s and. As the high loads of the gearbox are applied on this stage, the oil is decided upon it. As shown in the following diagram as the expected working temperature is 65 degrees, the oil chosen is ISO VG 68.



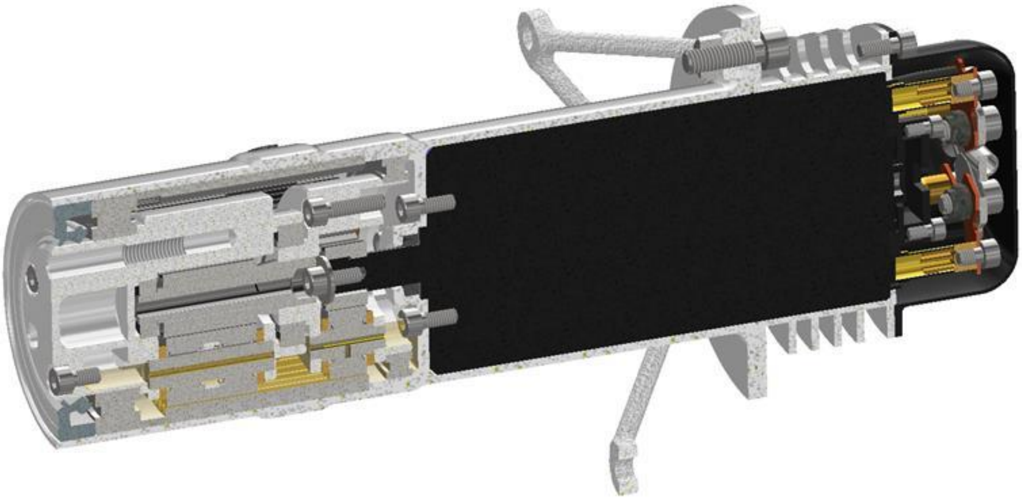
For spur, helical and beveled enclosed gears
For oils with VI=90 (recommendations are empirical)

Temp °C	Pitch line velocity, m/s^2							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
10	32							
15	46	32						
20	68	46	32					
25	68	46	32					
30	100	68	46	32				
35	100	100	68	46	32			
40	150	100	68	46	32	32	32	
45	220	150	100	68	46	46	32	32
50	320	220	150	100	46	46	46	32
55	460	220	150	100	68	68	68	46
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1000	680	320	220	150	100	100	68
75	1500	380	460	320	220	150	150	100
80	2200	1000	680	460	220	220	220	150
85	3200	1500	1000	460	320	220	220	150
90	3200	2200	1000	680	460	320	320	220
95		3200	1500	1000	460	460	320	220
100		3200	2200	1000	680	460	460	320

FIGURE 26. Oil viscosity (Machinery lubrication 2006)

The oil chosen for the gearbox is Castrol Manual Transaxle 75W-90 oil synthetic oil. This oil can be widely acquired and in special cases other similar oils can be used.

The oil capacity can be figured with the previously made 3d model of the gearbox. A usual rule of thumb used in planetary gear systems is that the oil should come slightly higher than the midway point of the lowest planetary gear in stationary position. This helps with oil delivery to planet bearings. The oil is modeled into the 3d model, and the final capacity comes to be 21,5 ml.



PICTURE 25. Gearbox oil

Quick oil change can be done with the three bolt holes on the carrier extension. As these holes are not located in ideal placements and some oil will always stay left in the gearbox, it is recommended to always take the planetary carriers out of the gearbox. This also makes it possible to thoroughly clean all parts.

Sealing solutions are needed for four different locations. These are the mounting holes for ring gear, oil change bolts and both ends of the gearbox. The oil change bolts are easy to seal with O-rings placed under them. This should be enough to keep the oil inside. Ring gear mounting bolts are harder to seal in a lasting way, as there is no room for O-rings. For this the bolts are only mounted once, and silicone is applied between the bolt and casing. The silicone chosen for this is Loctite SI 5980. This silicone is designed for engine applications where oil and other liquids are present. For this it also works well in high temperatures.

The ends of the gearbox need to be sealed with proper oil seals as in the inner side there is the spinning motor axle and in the outer end the carrier extension. The diameter of the casing limits the choice for outer end seals and as the bearing for carrier support has already been chosen, options become limited. A radial shaft oil seal 40x62x10 HMSA10 RG is chosen, as it fits within the limiting dimensions. Its limiting speed is 3300 rpm and the rotational speed of the carrier at 120 km/h is around 1600 rpm. It also has a maximum operational temperature of 100 degrees and should not become a problem.

For the motor shaft sealing the solution is more complicated to find as the rotational speed is very high. For this the chosen seal is ETRA MZV 10x16x4 NBR oil seal that has a maximum circumferential speed of 30 m/s. For the axle the circumferential speed is only 19,2 m/s, so the seal should be good for this application. Also, the maximum operational temperature of the seal is 110 degrees and therefore sufficient.

5 CONCLUSIONS AND FUTURE PERSPECTIVE

The short-term goal of this thesis was to design a gear system to be used in the TFS25 vehicle. As a result of this thesis, a planetary gearbox and the components for its integration into the vehicle were designed. The longer-term goal of creating a basis for future electric vehicle drivetrain design was reached sufficiently as new tools and design principles were integrated into the existing repertoire of the drivetrain department. The design process and results were documented in a way that it can be used as a base for future development.

The completed gearbox designs mass with its mountings and motor comes to 3,2 kg. This is 1,9 kg lower than its 2024 counterpart and therefore reaches the goal set for the system mass. Regarding the cost effectiveness of the system, the conclusion cannot be made at this point. All system components were created to follow an easy-to-manufacture design philosophy but as the manufacturing of the system is still on going, the final costs are unknown at this point.

In the future the team should implement more tools for geartrain design in the form of KISSsys as this could be used to model the entire gearbox in a single simulation and be used to further analyze the drive systems behavior. With this a better understanding of power losses and lubrication could be gained. This could also help in time management of the design process as KISSsys calculation module could be used for bearing and axle design and time-consuming hand calculations could completely be replaced.

As the designed gear system is manufactured in the future, it should be tested extensively to create an understanding of its real capabilities and how it corresponds to the designed one. This is a crucial step in evaluating the design and how it could be improved in the future.

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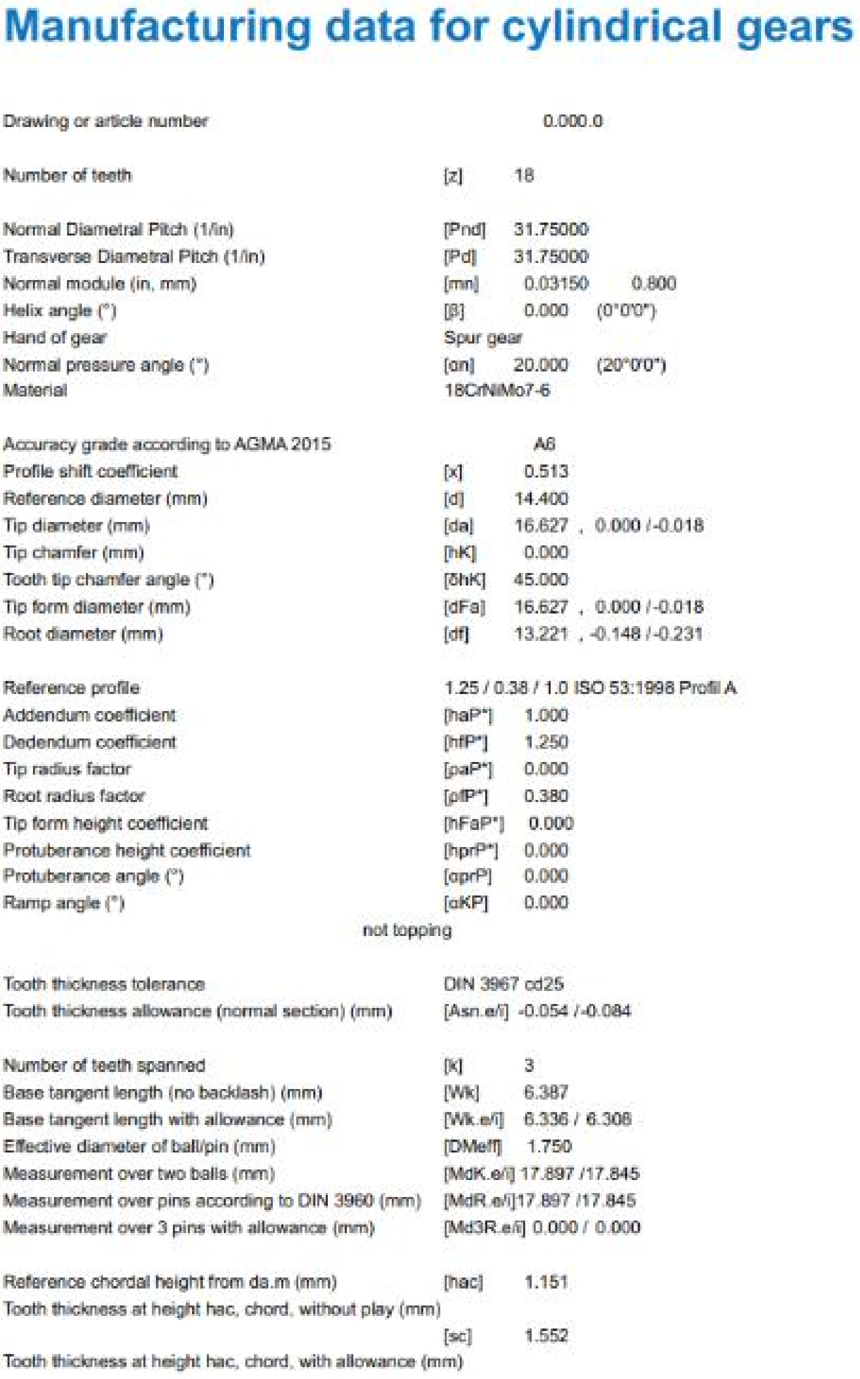
APPENDICES

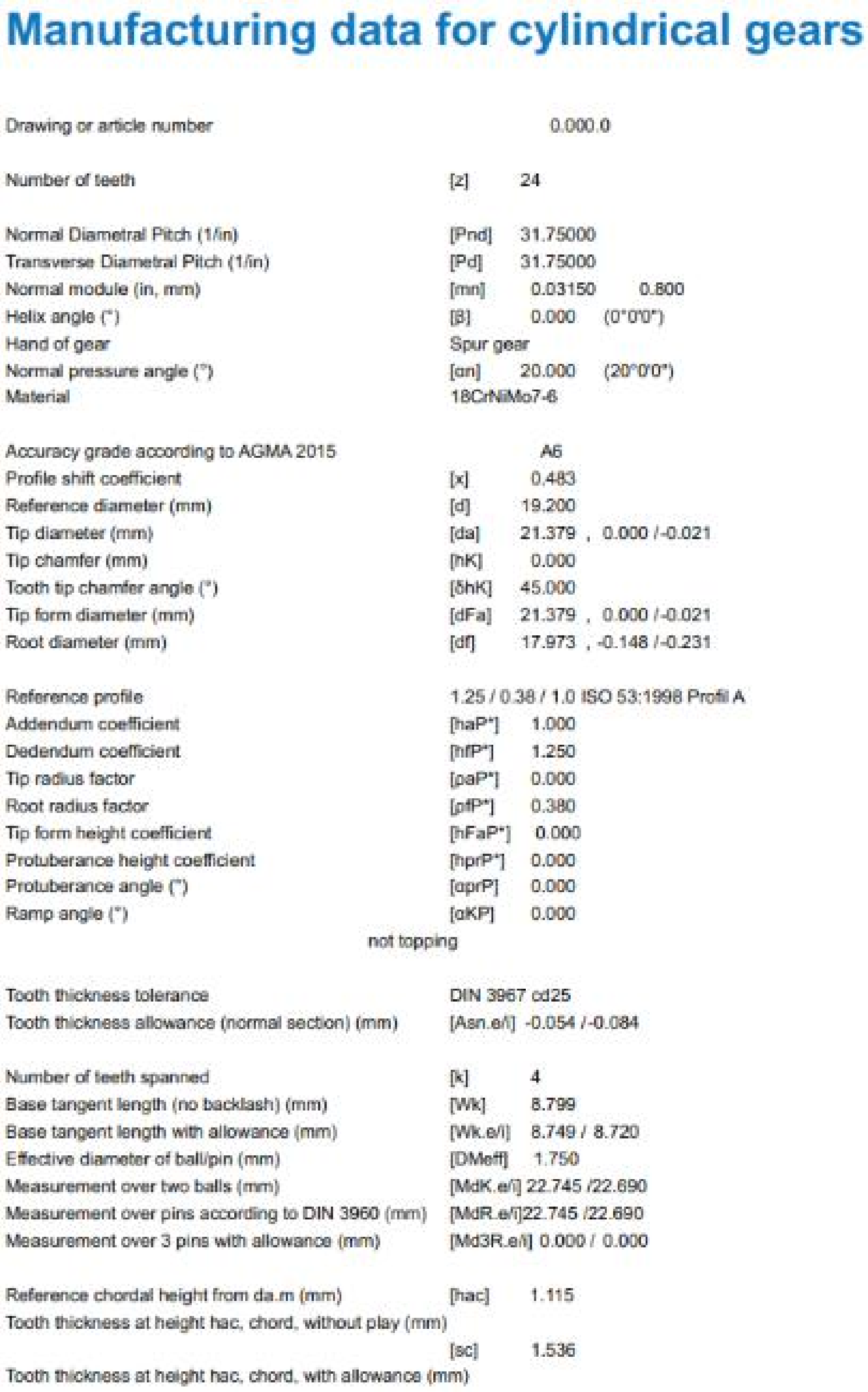
Appendix 1. Sun gear manufacturing data



Manufacturing data for cylindrical gears

Drawing or article number		0.000.0
Number of teeth	[z]	18
Normal Diametral Pitch (1/in)	[Pnd]	31.75000
Transverse Diametral Pitch (1/in)	[Pd]	31.75000
Normal module (in, mm)	[mn]	0.03150 0.800
Helix angle (°)	[β]	0.000 (0°0'0")
Hand of gear		Spur gear
Normal pressure angle (°)	[αn]	20.000 (20°0'0")
Material		18CrNiMo7-6
Accuracy grade according to AGMA 2015		A6
Profile shift coefficient	[x]	0.513
Reference diameter (mm)	[d]	14.400
Tip diameter (mm)	[da]	16.627 , 0.000 /-0.018
Tip chamfer (mm)	[hK]	0.000
Tooth tip chamfer angle (°)	[δhK]	45.000
Tip form diameter (mm)	[dFa]	16.627 , 0.000 /-0.018
Root diameter (mm)	[df]	13.221 , -0.148 /-0.231
Reference profile		1.25 / 0.38 / 1.0 ISO 53:1998 Profil A
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[ρaP*]	0.000
Root radius factor	[ρfP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[αprP]	0.000
Ramp angle (°)	[αKP]	0.000
	not topping	
Tooth thickness tolerance		DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[Asn.e/i]	-0.054 /-0.084
Number of teeth spanned	[k]	3
Base tangent length (no backlash) (mm)	[Wk]	6.387
Base tangent length with allowance (mm)	[Wk.e/i]	6.336 / 6.308
Effective diameter of ball/pin (mm)	[DMeff]	1.750
Measurement over two balls (mm)	[MdK.e/i]	17.897 /17.845
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	17.897 /17.845
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	0.000 / 0.000
Reference chordal height from da.m (mm)	[hac]	1.151
Tooth thickness at height hac, chord, without play (mm)	[sc]	1.552
Tooth thickness at height hac, chord, with allowance (mm)		

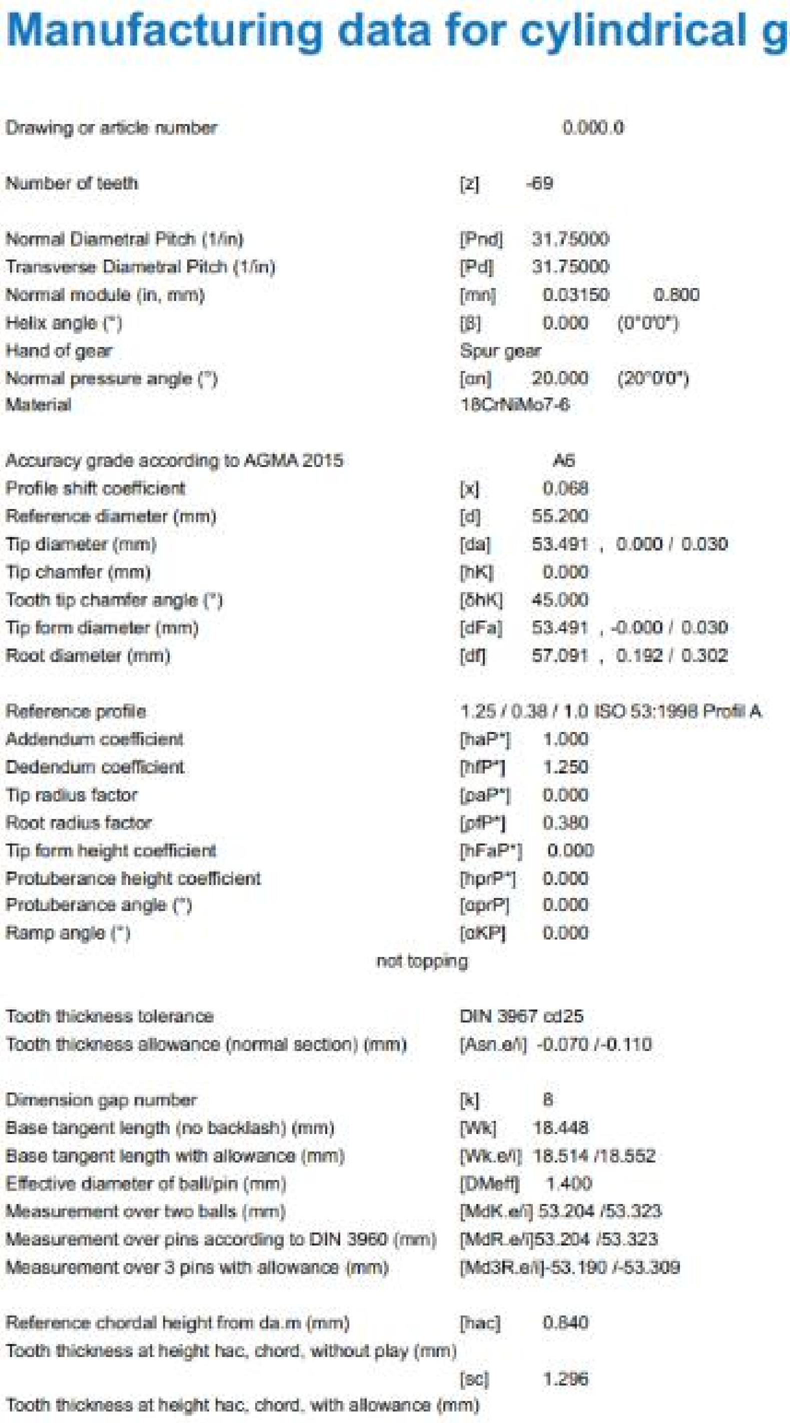
Appendix 2. Planet gears manufacturing data



Manufacturing data for cylindrical gears

Drawing or article number		0.000.0
Number of teeth	[z]	24
Normal Diametral Pitch (1/in)	[Pnd]	31.75000
Transverse Diametral Pitch (1/in)	[Pd]	31.75000
Normal module (in, mm)	[mn]	0.03150 0.800
Helix angle (°)	[β]	0.000 (0°0'0")
Hand of gear	Spur gear	
Normal pressure angle (°)	[αn]	20.000 (20°0'0")
Material	18CrNiMo7-6	
Accuracy grade according to AGMA 2015		A6
Profile shift coefficient	[x]	0.483
Reference diameter (mm)	[d]	19.200
Tip diameter (mm)	[da]	21.379 , 0.000 /-0.021
Tip chamfer (mm)	[hK]	0.000
Tooth tip chamfer angle (°)	[δhK]	45.000
Tip form diameter (mm)	[dFa]	21.379 , 0.000 /-0.021
Root diameter (mm)	[df]	17.973 , -0.148 /-0.231
Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A	
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[ρaP*]	0.000
Root radius factor	[ρfP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[αprP]	0.000
Ramp angle (°)	[αKP]	0.000
not topping		
Tooth thickness tolerance	DIN 3967 cd25	
Tooth thickness allowance (normal section) (mm)	[Asn.e/i]	-0.054 /-0.084
Number of teeth spanned	[k]	4
Base tangent length (no backlash) (mm)	[Wk]	8.799
Base tangent length with allowance (mm)	[Wk.e/i]	8.749 / 8.720
Effective diameter of ball/pin (mm)	[DMeff]	1.750
Measurement over two balls (mm)	[MdK.e/i]	22.745 /22.690
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	22.745 /22.690
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	0.000 / 0.000
Reference chordal height from da.m (mm)	[hac]	1.115
Tooth thickness at height hac, chord, without play (mm)	[sc]	1.536
Tooth thickness at height hac, chord, with allowance (mm)		

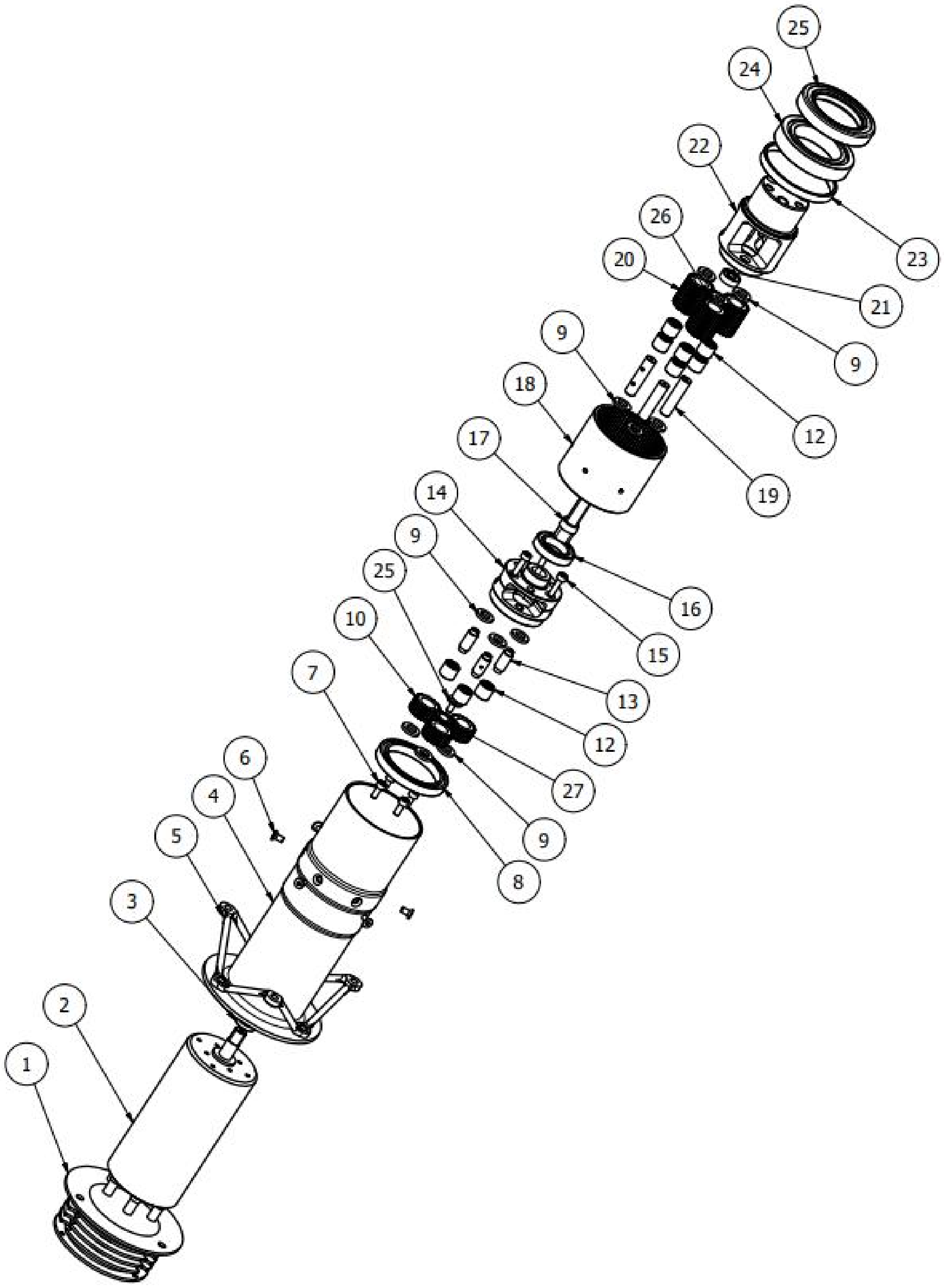
Appendix 3. Ring gear manufacturing data



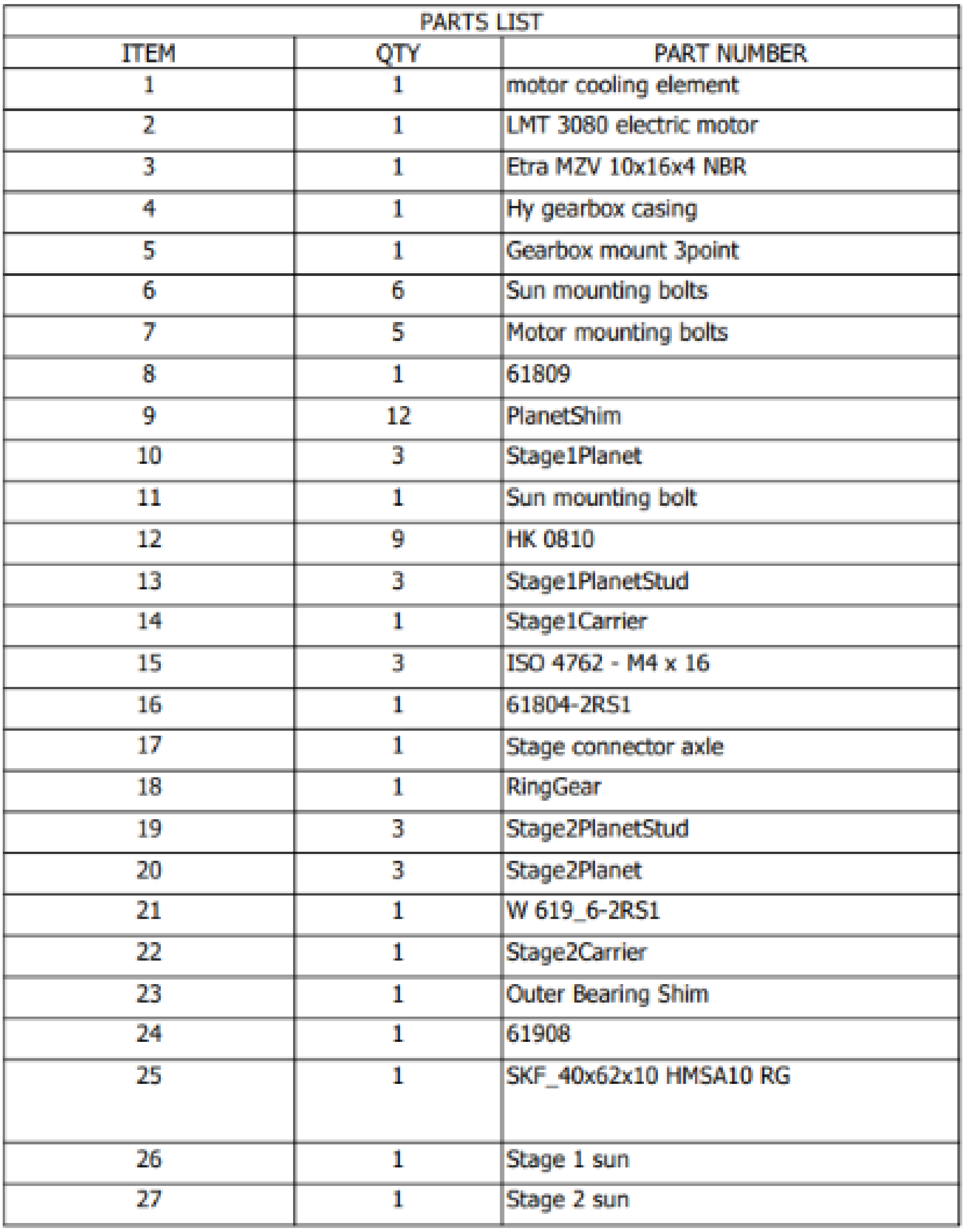
Manufacturing data for cylindrical gears

Drawing or article number		0.000.0
Number of teeth	[z]	-69
Normal Diametral Pitch (1/in)	[Pnd]	31.75000
Transverse Diametral Pitch (1/in)	[Pd]	31.75000
Normal module (in, mm)	[mn]	0.03150 0.800
Helix angle (°)	[β]	0.000 (0°0'0")
Hand of gear	Spur gear	
Normal pressure angle (°)	[αn]	20.000 (20°0'0")
Material	18CrNiMo7-6	
Accuracy grade according to AGMA 2015		A6
Profile shift coefficient	[x]	0.068
Reference diameter (mm)	[d]	55.200
Tip diameter (mm)	[da]	53.491 , 0.000 / 0.030
Tip chamfer (mm)	[hK]	0.000
Tooth tip chamfer angle (°)	[δhK]	45.000
Tip form diameter (mm)	[dFa]	53.491 , -0.000 / 0.030
Root diameter (mm)	[df]	57.091 , 0.192 / 0.302
Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A	
Addendum coefficient	[haP*]	1.000
Dedendum coefficient	[hfP*]	1.250
Tip radius factor	[ρaP*]	0.000
Root radius factor	[ρfP*]	0.380
Tip form height coefficient	[hFaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle (°)	[αprP]	0.000
Ramp angle (°)	[αKP]	0.000
	not topping	
Tooth thickness tolerance	DIN 3967 cd25	
Tooth thickness allowance (normal section) (mm)	[Asn.e/i]	-0.070 /-0.110
Dimension gap number	[k]	8
Base tangent length (no backlash) (mm)	[Wk]	18.448
Base tangent length with allowance (mm)	[Wk.e/i]	18.514 /18.552
Effective diameter of ball/pin (mm)	[DMeff]	1.400
Measurement over two balls (mm)	[MdK.e/i]	53.204 /53.323
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	53.204 /53.323
Measurement over 3 pins with allowance (mm)	[Md3R.e/i]	-53.190 /-53.309
Reference chordal height from da.m (mm)	[hac]	0.840
Tooth thickness at height hac, chord, without play (mm)	[sc]	1.296
Tooth thickness at height hac, chord, with allowance (mm)		

Appendix 4. Gearbox exploded view



Appendix 5. Parts list



PARTS LIST		
ITEM	QTY	PART NUMBER
1	1	motor cooling element
2	1	LMT 3080 electric motor
3	1	Etra MZV 10x16x4 NBR
4	1	Hy gearbox casing
5	1	Gearbox mount 3point
6	6	Sun mounting bolts
7	5	Motor mounting bolts
8	1	61809
9	12	PlanetShim
10	3	Stage1Planet
11	1	Sun mounting bolt
12	9	HK 0810
13	3	Stage1PlanetStud
14	1	Stage1Carrier
15	3	ISO 4762 - M4 x 16
16	1	61804-2RS1
17	1	Stage connector axle
18	1	RingGear
19	3	Stage2PlanetStud
20	3	Stage2Planet
21	1	W 619_6-2RS1
22	1	Stage2Carrier
23	1	Outer Bearing Shim
24	1	61908
25	1	SKF_40x62x10 HMSA10 RG
26	1	Stage 1 sun
27	1	Stage 2 sun