CZECH TECHNICAL UNIVERSITY IN PRAGUE
FACULTY OF MECHANICAL ENGINEERING

Department of Automotive, Combustion Engine and Railway
Engineering

Master Thesis

Planetary Gearbox for Electric Vehicle

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MASTER'S THESIS ASSIGNMENT

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Branch of study: **Advanced Powertrains**

II. Master's thesis details

Master's thesis title in English:

Planetary gearbox for electric vehicle

Master's thesis title in Czech:

Planetová převodovka pro eletrické vozidlo

Guidelines:

1) Make a review of powertrains of serially produced electric vehicles. Also provide a review of patented solution of EV powertrains.
2) Choose the parameters of passenger EV, for which you will design the transmission. Choose right energy source.
3) Propose several variants of transmission and make a guess of efficiency, mass and dimensions. Minimally one variant has to be planetary one.
4) For chosen planetary variant prepare a CAD model and basic necessary strength calculations. Make a fabrication drawing of one part chosen by the tutor.

Bibliography / sources:

[1] Lino Guzzela, Antoniol Sciarretta:
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Springer-Verlag Berlin Heidelberg, 2007
ISBN 978-3-540-74691-1
[2] Harald Naunheimer, Bernd Bertsche, Joachim Ryborz, Wolfgang Novak
Automotive Transmissions, Fundamentals, Selection, Design and Application Second Edition
Springer Heidelberg Dordrecht London New York 2011

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Assignment valid until: _____

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III. Assignment receipt

The student acknowledges that the master's thesis is an individual work. The student must produce his thesis without the assistance of others, with the exception of provided consultations. Within the master's thesis, the author must state the names of consultants and include a list of references.

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Annotation

Author	: Samed Ali Akköse
Title	: Planetary Gearbox for Electric Vehicle
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Major	: Advanced Powertrains
Supervisor	: doc. Dr. Ing. Gabriela Achtenová

Abstract : This thesis consists of research on battery electric vehicles currently available on the market and their powertrains, reviews of patented solutions of electric vehicle powertrains, calculations of longitudinal dynamics of the selected vehicle and it is design parameters, three variants of powertrains and their mass, efficiency and dimension estimations, calculation of a planetary gearbox system and the strength calculation results. The thesis also includes a 2D CAD model of the assembly and fabrication drawing of the sun shaft , 3D CAD model of the planetary gearbox and results of strength calculations on Ricardo Company's SABR/GEAR software.

Keywords : Electric Vehicle, Powertrain, Vehicle Dynamics, Planetary Gearset, Computer Aided Design (CAD)

Number of Pages	: 91
Number of Figures	: 66
Number of Tables	: 18

Declaration

I hereby declare that the thesis 'Planetary Gearbox for Electric Vehicle' is based on my own work. All ideas, major views and data from different resources by other authors were only used as a reference and/or research purpose. I am the sole author of the thesis and that neither any part of this thesis nor the whole of the thesis has been submitted for a degree to any other University or Institution.

In Prague …………   Samed Ali Akköse

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Finally, my special thanks belong to my parents for their encouragement, love and their belief to me.

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Acronyms

AC   Alternating Current

AWD   All-Wheel Drive

BEV   Battery Electric Vehicle

CPC   Cooperative Patent Classification

EDU   Electric Drive Unit

EV   Electric Vehicle

FD   Final Drive

IPC   International Patent Classification

ISO   International Organization for Standardization

LOAMA   Load of Action Misalignment

PCD   Pitch Circle Diameter

PGS   Planetary Gear Set

PMSM   Permanent Magnet Synchronous Motor

RWD   Rear Wheel Drive

SUV   Sport Utility Vehicle

1. Introduction

General Overview to Electric Vehicles

An electric vehicle (EV) uses an electric motor as a power source. After the invention of the storage battery in 1850´s and the improvements in the area made the production of the electric vehicle possible and the first application used in Paris in 1881.

In the early years of the automotive industry, the electric vehicles were in a competition with petroleum-fueled cars but their low top speeds, high costs, the short range because of the inefficiency of the batteries and the discovery of the large amount of fossil fuels in 1920´s caused a reduction of the prices in the gasoline. As a result, the use of the electric vehicles declined.

In the recent years, the electric vehicles become an important research subject. Especially in the last decade the electric vehicles started to be more dominant on the market again. Decreasing of the fossil fuels and increase in the gas prices, recent improvements in the battery and power electronic technologies and since the battery electric vehicles have zero tailpipe emissions in comparison with the petroleum-fueled cars can be shown as the main reasons for the explanation why the electric vehicles are becoming more popular.
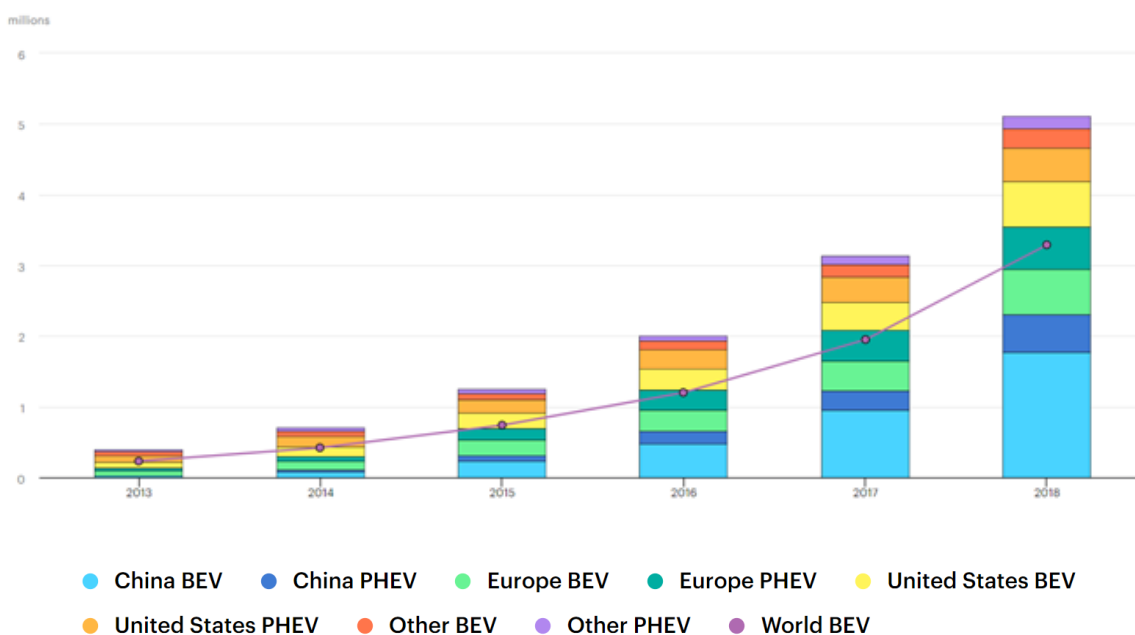
Fig. 1   Electric car Deployment in Selected Countries, 2013-2018 [1]

From the actual sales data, the electric car deployment is increasing [1]. The Figure 1 shows that, in 2018 there was a 63% increase in the electric vehicle deployment in the world. Four-wheel, battery electric vehicle (BEV) worldwide, amounted to approximately three million units. Since it is compared, this number is coming to less than 3% of the global total automobile sales. Also, it could be seen from the Figure 2 that by the year 2030 it is expected that the Battery Electric Vehicle deployment would reach up to 25 million units.
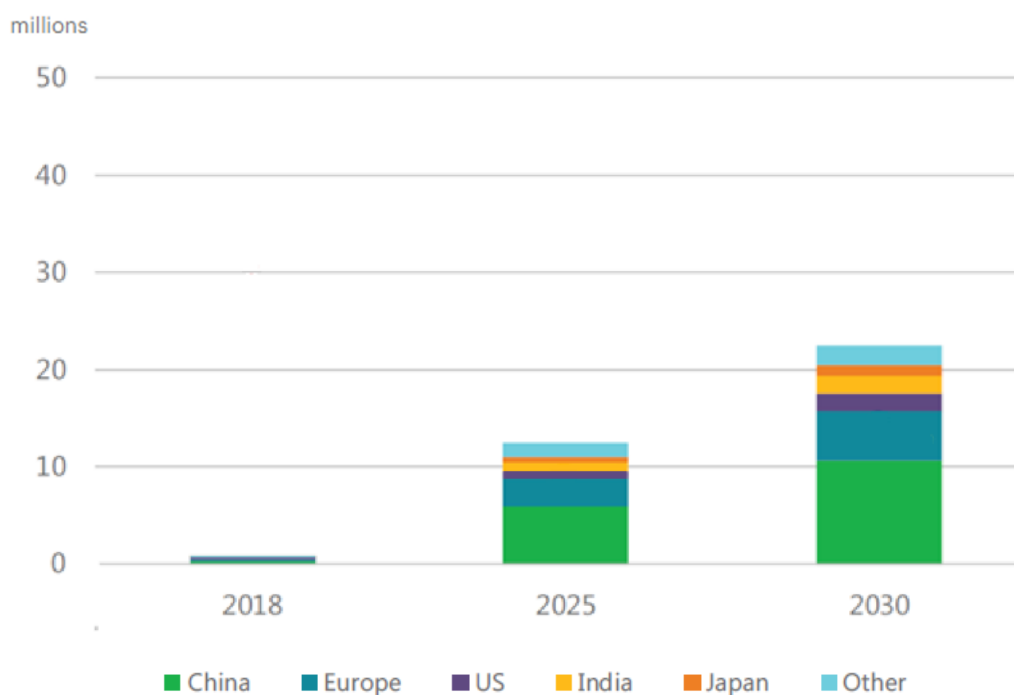
Fig. 2   Expected BEV Sales in the years 2025 and 2030 [1]

Since it is seen from the actual sales data and the future estimations, the battery electric vehicles will be more frequent on the market and in the streets in global scale. Since the vehicles are running with the electricity, the efficiency of the powertrain parts has critical importance for the battery life.

2.   Review of Powertrains of Serially Produced EV´s

The electric vehicles which are using only the batteries as the power source are called as Battery Electric Vehicle (BEV). Compared with the conventional combustion engine vehicles, the battery electric vehicles (BEV) are simpler and easier to operate because they do not have complex transmissions. Most battery electric vehicles have single speed transmission. Also, the BEV´s does not have exhaust systems. It means no exhaust after treatment device needed in a battery electric vehicle. The battery electric vehicles are consisting of;

a. Battery Pack: This is the energy source of the vehicle which it will be used in drive, heat & cooling operations and running the light of accessories. Lithium-ion batteries are found in most of the modern EV´s today. [3]

b. Power Inverter: Since the electric vehicle has AC motor, the energy from the battery should be converted into AC to be used. The inverter also works during the regenerative braking to modify the alternative current into direct current.

c. Electric Motor : This is the electric machine which converts the electrical energy into mechanical energy to use in drive.

d. Transmission : This is the machine which provides controlled application of the power. In the battery electric vehicles mostly Single Speed Transmissions are used. Single speed transmission is more advantageous because there will be no need for shifting of gears and no need of a clutch.

e. Charging Port : Charging port is the location where the electrical energy enters the car, much like the access port to the gas tank in an internal combustion engine (ICE).
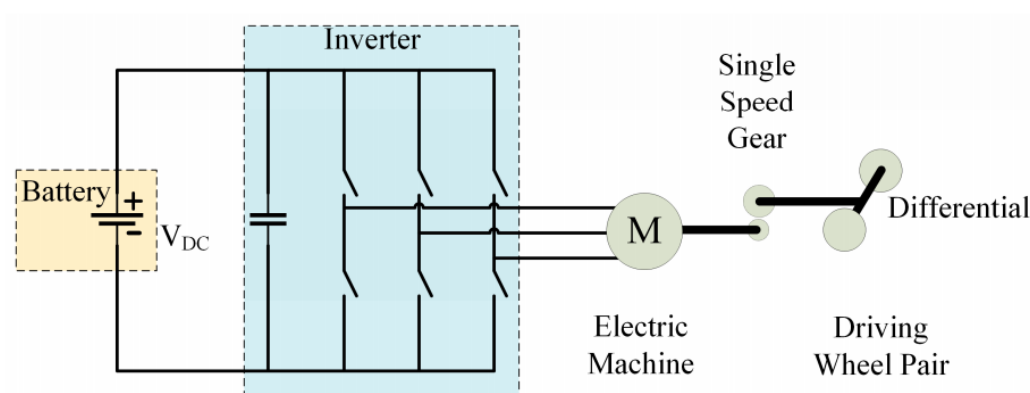
Fig. 3   BEV Configuration [4]
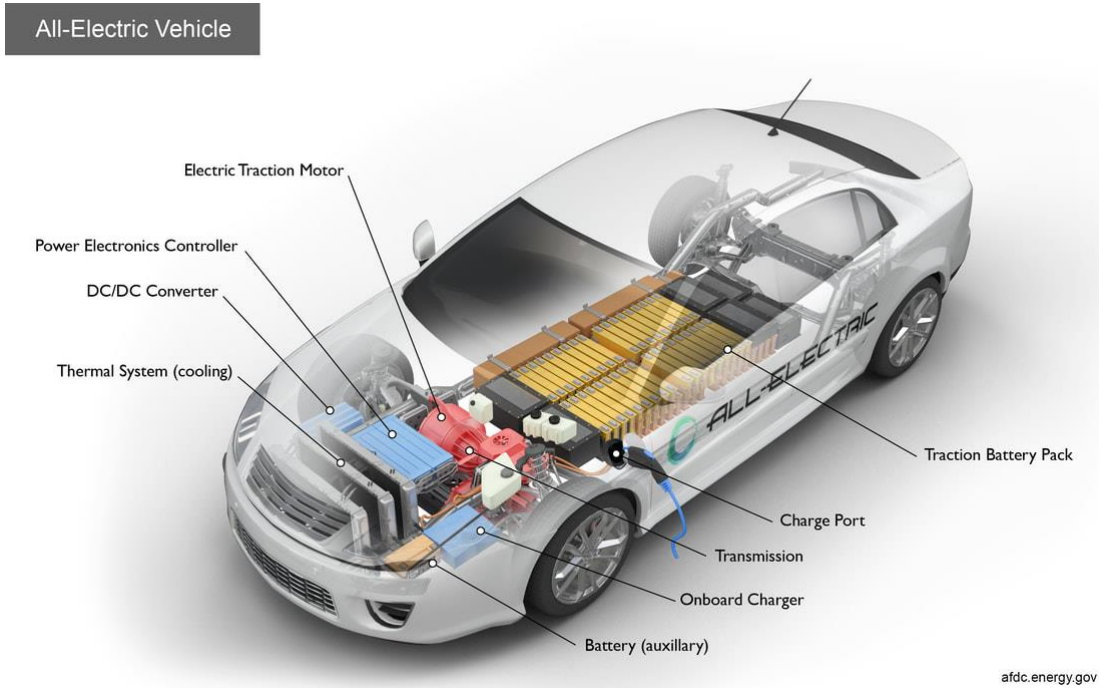
Fig. 4   BEV Anatomy [U.S Department of Energy]

In the last decades so many automotive manufacturers started to introduce EV´s and they have entered to the automotive market. Those vehicles range are related with their battery capacity. Typical range of the BEV is 100km to 250km regarding on the driving and the environmental conditions [2].

2.1.   Battery Electric Vehicles Currently Available on the market

In this project, to be able to make a design of a powertrain, it is necessary to make a research on the vehicles which are currently available on the market. Each car has different aspects that with the data available the inspiration could be helpful for the vehicle parameters selection.

The vehicles are selected according to their market sales and sorted alphabetically.

2.1.1. Audi e-tron

Since the electric car industry improves, each brand joins the car market with a new alternative. Audi e-tron is the all-electric SUV, the first mass production electric car of the company. It is being produced since 2018.

Audi e-tron uses planetary reduction on the front axle with additional simple parallel gear step to differential.

Fig. 5   Audi e-tron (left),Transmission System(right) 2018 [5] [6]

2.1.2. BMW i3

BMW i3 is the first mass produced electric vehicle of the company and it is launched by the BMW´s subcompany BMW i [7]. It is being produced since 2013 by the BMW Group in Plant Leipzig. In BMW i3 two stage parallel shaft with the compound gears on the countershaft is used. This design has rear wheel drive (RWD) layout.

Fig. 6   BMW i3 (left), Transmission System (right) [6] [8]

2.1.3. Citroën C-Zero

This model is the rebadged version of the Mitsubishi i-MiEV (Mitsubishi innovative Electric Vehicle). This vehicle is produced in Europe under the name of Citroën C-Zero and Peugeot iOn for the customers. Similar to BMW i3 design, also in this model a two stage parallel shaft with the compound gears on the countershaft is used.

Fig. 7   Citroën C-Zero (left) and it´s transmission (right) [6] [9]

2.1.4. Chevrolet Bolt EV

Chevrolet Bolt EV is the full electric production of the GM company. It is being produced since 2016. The vehicle is rebadged for the continental Europe and sold under the name Opel Ampera-e . In this model, two stage parallel shafts with compound gears on countershaft is used. Motor is coaxial with differential. It means the half shaft goes through the motor.
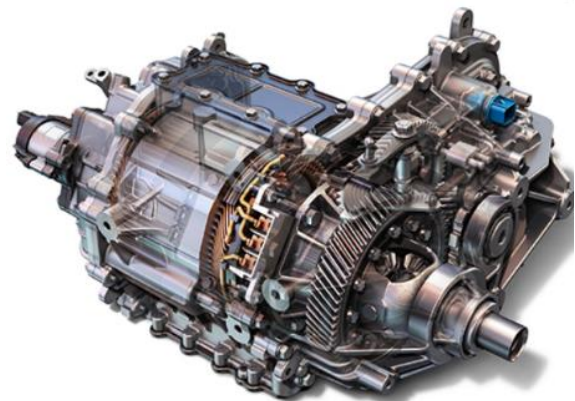
Fig. 8   Chevrolet Bolt EV (left) the Electric Drive Unit (right) [6][10]

2.1.5. Tesla

Tesla Inc. can be named as the pioneer of the electric car market. The company is focused on producing high performance electric vehicles. As it be can seen from their products on the market, they have three popular models. They are Model 3, Model X and Model S.

2.1.5.1. Tesla Model 3

Tesla Model 3 is the most popular model of the brand. It is produced since 2017.

In this model the two-stage parallel shaft with compound gears on the countershaft is used. This design has all-wheel-drive layout.

Fig. 9   Tesla Model 3 (left) , Transmission System (right) [6] [11]

2.1.5.2. Tesla Model X and Model S

Tesla Model S is the all-electric sedan and Tesla Model X is the luxury all electric SUV designed and produced by Tesla corporation. Both models have the all-wheel drive layout and has two electric motors in both rear and front axles. Also, in both Tesla S and Tesla X, same type of transmission single speed two stage parallel shaft is used.

Fig. 10   Transmission System (middle), Model X(left), Model S(right) [6] [12] [13]

2.1.6. Nissan LEAF

Nissan LEAF is the battery electric vehicle alternative of the company. It is produced since 2010 by the Japanese company Nissan Motor Company. Like it is competitors, Nissan LEAF also have single speed two stage parallel shaft with compound gears on the countershaft.

Fig. 11   Nissan LEAF (left), Transmission System (right) [6] [14]

2.1.7. Jaguar I-Pace

Jaguar I-Pace is the battery electric vehicle SUV which is produced by the company Jaguar Land Rover (JLR). Jaguar I-Pace is the first all-electric car of the company and produced since 2018. This vehicle uses two identical powertrains on both rear and front axles. The motor is coaxial with differential and there is a planetary gearset reduction.

Fig. 12   Jaguar I-Pace (left) , Transmission System (right) [6] [15]

2.1.8. Volkswagen e-Golf

Volkswagen e-Golf is the all-electric version of the model VW Golf. It is produced since 2014. VW e-Golf also uses the system like their competitors BMW, Citroen, Nissan and Tesla; single speed two stage parallel shafts with compound gears on the countershaft.

Fig. 13 Volkswagen e-Golf (left), Electric Drive Unit (right) [6] [16]
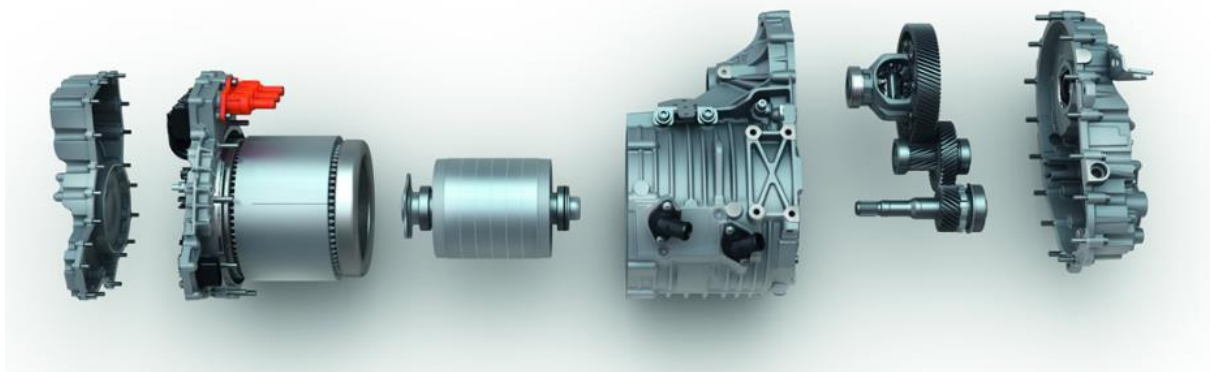
Fig. 14 Electric Drive Unit Disassembly of VW e-Golf [6]
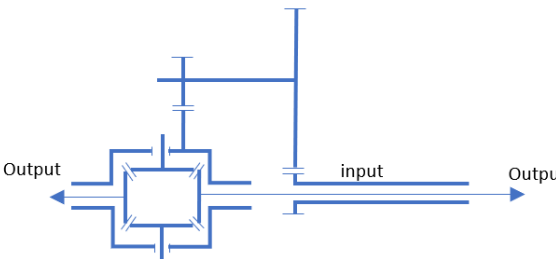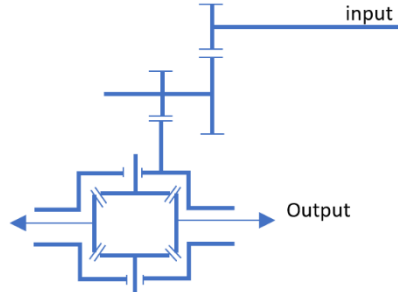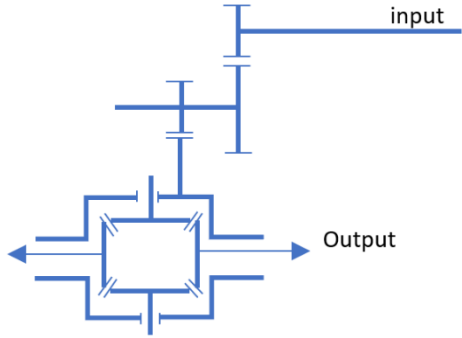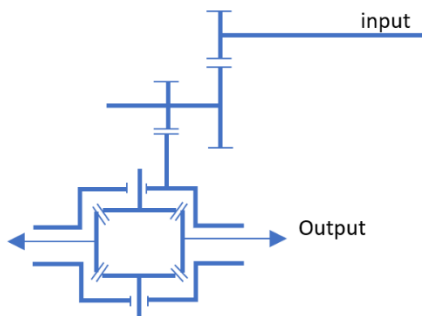
2.2. General Review of the Transmissions

Tab. 1 *Kinematic Schemes of the Transmissions Regarding to the Model*

Brand/Model	Type of Transmission	Kinematic Scheme
Audi e-tron	Planetary Single Speed reduction. Sun is input, carrier is output, ring is stationary. There is additional simple parallel gear step to differential.	
BMW i3	Two stage parallel shaft with compound gears on the layshaft.	
Citroen C-Zero	Two stage parallel shaft with compound gears on the layshaft.	

Chevrolet Bolt EV	Two stage parallel shaft with compound gears on the layshaft. Motor is coaxial with differential. Half shaft goes through with the electric motor.	
Tesla Model 3	Two stage parallel shaft with compound gears on the layshaft.	
Tesla Model S-X	Two stage parallel shaft with compound gears on the layshaft.	
Nissan LEAF	Two stage parallel shaft with compound gears on the layshaft.	

Jaguar I-Pace	Planetary gear reductor with compound planets is used. Sun is input, carrier is output, ring is stationary. Sun is meshing with bigger planet and ring is meshing with smaller planets. Carrier is mounted to differential cage.	input Output Output
VW e-Golf	Two stage parallel shaft with compound gears on the layshaft.	input Output

2.3. Patented Solutions of Electric Vehicle Powertrains

Before making a proper design for the powertrains which are going to be used in electric vehicles, the patented solutions and the new technological trends from the companies are also highly recommended . Because there is the risk of re-inventing the same design. Before making the research, it is important to know patent classification systems. By the help of the patent classification system, it is easier to make the research for the inventions which is the patent is correlated. To utilize the system the patent offices developed the hierarchical classification systems.

2.3.1. International Patent Classification (IPC)

The international Patent Classification serve as an instrument for orderly arrangement of the patent documents and helps the researcher to find the contemporary solutions of the related search field. All the International Patent Organisation Member companies' patents can be found. IPC revisions take place yearly.[14]

2.3.2. Cooperative Patent Classification System (CPC)

Cooperative Patent Classification System is developed by European Patent Office (EPO) and the United States Patent and Trademark office (USPTO). CPC system based on the IPC system although it is more detailed, and the CPC revisions take place monthly. Patent distributions are appointed in at least one grouping term and extra classification terms are used to give added specifications of the patent.

In the classification structure each term represents the category of the patent. Each patent has the specific symbol of "X00Y00/00". The first letter "X" represents the "section symbol" comprises the letters from "A (Human Necessities)" to "H (Electricity)" and "Y (Emerging Cross-Sectional Technologies)" . After this letter the two-digit number gives the "Class Symbol". The final letter "Y" represents the "subclass symbol". After the subclass symbol it follows a one to four-digit "group" number, and at last, it follows with an at least two-digits "main group" number. This hierarchy helps the researcher to find similar concept technological solutions patents. [15]

Ranking of CPC

- Section (one letter A to H, and Y)
  - Class (two digits)
    - Subclass (one letter)
      - Group (one to four digits)
        - Main Group (at least three digits)

For the powertrain research the CPC ranking **"F16H2200/2005"** is used.

In this system it means;

- Section **F** (one letter A to H, and Y)
  - Class **16** (two digits)
    - Subclass **H** (one letter)
      - Group **2200** (one to four digits)
        - Main Group **2005**(at least three digits)

It can be seen the representation of the letters from the Fig. 15;

Fig. 15 Classification Research for Planetary Transmission [16]

2.3.3. KR101971189B1 , Transmission of Electric Vehicle

The patent is assigned by Valeo Kapec Co.,Ltd.

The invention comprises a planetary gearset transmission. It includes planetary gearset elements sun gear, ring(annulus) gear, planet (pinion) gears and a carrier (spider). The motor through an input shaft is connected to the sun gear. The gear reduction is provided by the stationary ring and the carrier is the output.

This is a two-speed gearbox. There are two electromagnets, one of them is between the ring and the casing, which makes the ring stationary when engaged and the planetary gearset works as a redactor. The second electromagnet is between the ring and the carrier, which makes the planetary gearset working as a direct drive when engaged. There cannot be any free rotation of the ring. Otherwise the planetary gearset become a differential with two degrees of freedom and no power will be transmitted.
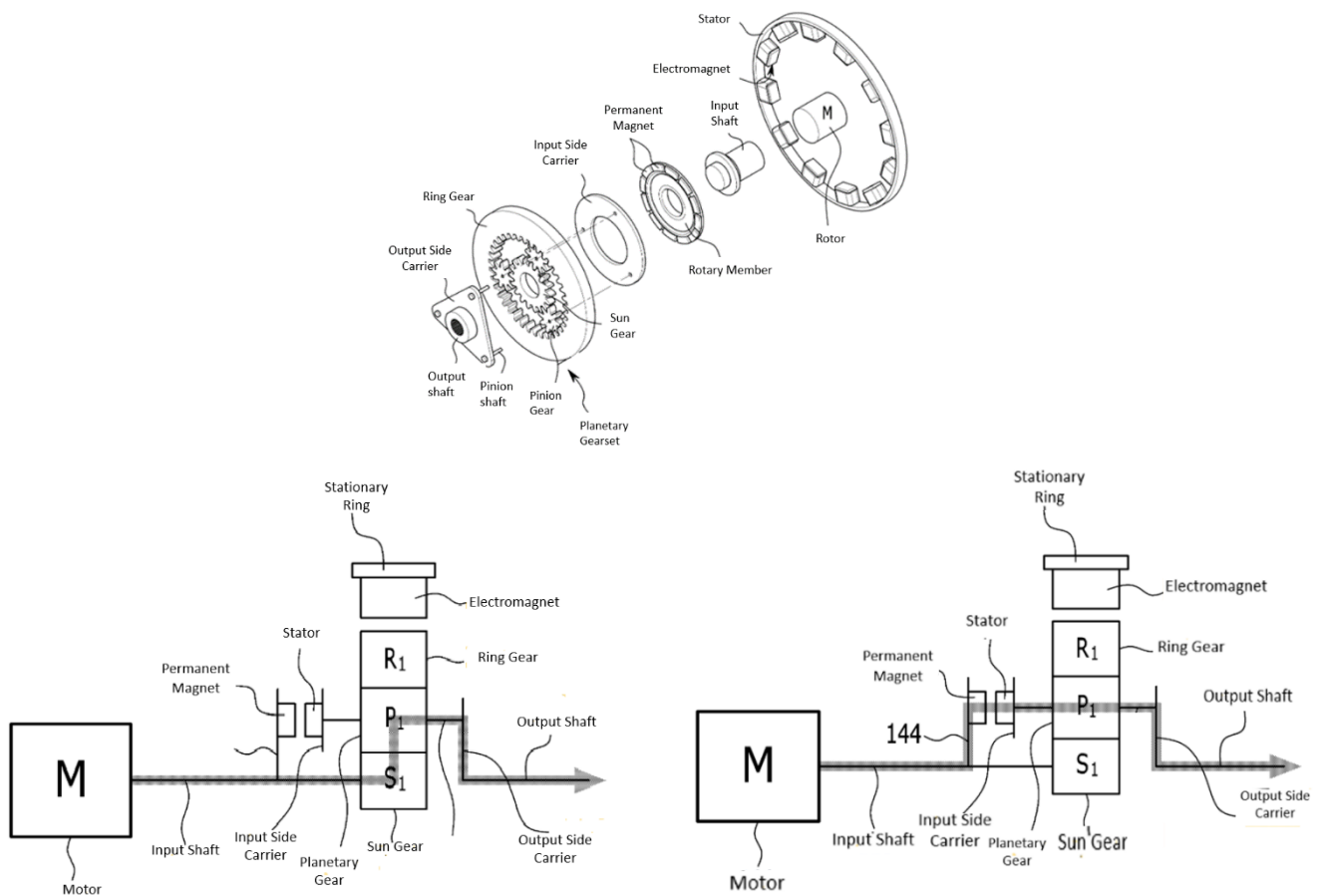
Fig. 16   The Patented Transmission For Electric Vehicle and Power Flow[17]

2.3.4. WO2019214995A1 , Multi-speed Planetary Transmission

This patent is assigned by Robert Bosch GmbH.

This invention consists of a drive shaft, planetary gearset, a clutch which provides a connection between input and output shaft, a freewheel and a locking element which is a dog clutch. In this multi speed planetary gearset the speed shift is done by the clutch engagement .

When the first gear is engaged the ring gear is blocked, and the clutch is not applied, as a result speed of the input shaft reduced. When the second gear engaged, the clutch is applied, and the ring gear is released as a result all the planetary gearset parts rotate together and create a high-speed output.
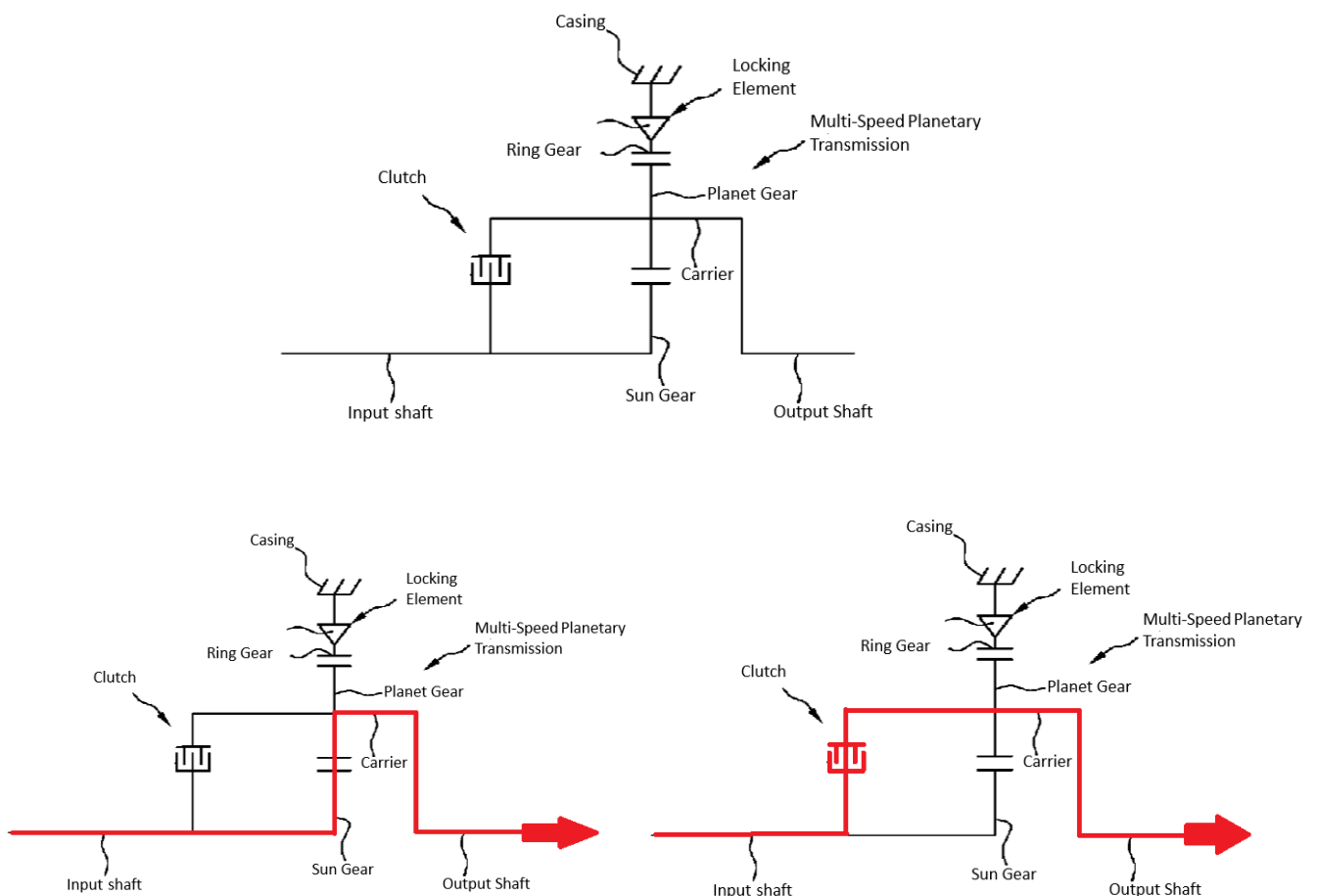
Fig. 17   Multi Speed Planetary Transmission for Vehicle and Power Flow [18]

2.3.5. WO2019115204A1, Electric Drive for a Motor Vehicle

The patent is assigned by Daimler AG.

This invention consists of an electric vehicle, a planetary gearbox includes two planetary sets, a braking element, a clutch and a differential. For the first planetary gearset, sun gear is input, and carrier is output, the ring gear is fixed. For the second planetary gearset the ring(annulus) is input and carrier of the second set is output.

When the first speed is shifted, braking element is activated and the sun gear in the second planetary gearset is fixed. The power which is coming from the first gearset is directed to ring gear of the second set and from there directed to differential & output shaft. When the second gear is shifted the clutch is applied, the ring and the sun of the second planetary gearset will rotate with the same speed, as a result the power comes from the first planetary gearset is directed to differential & output shaft.
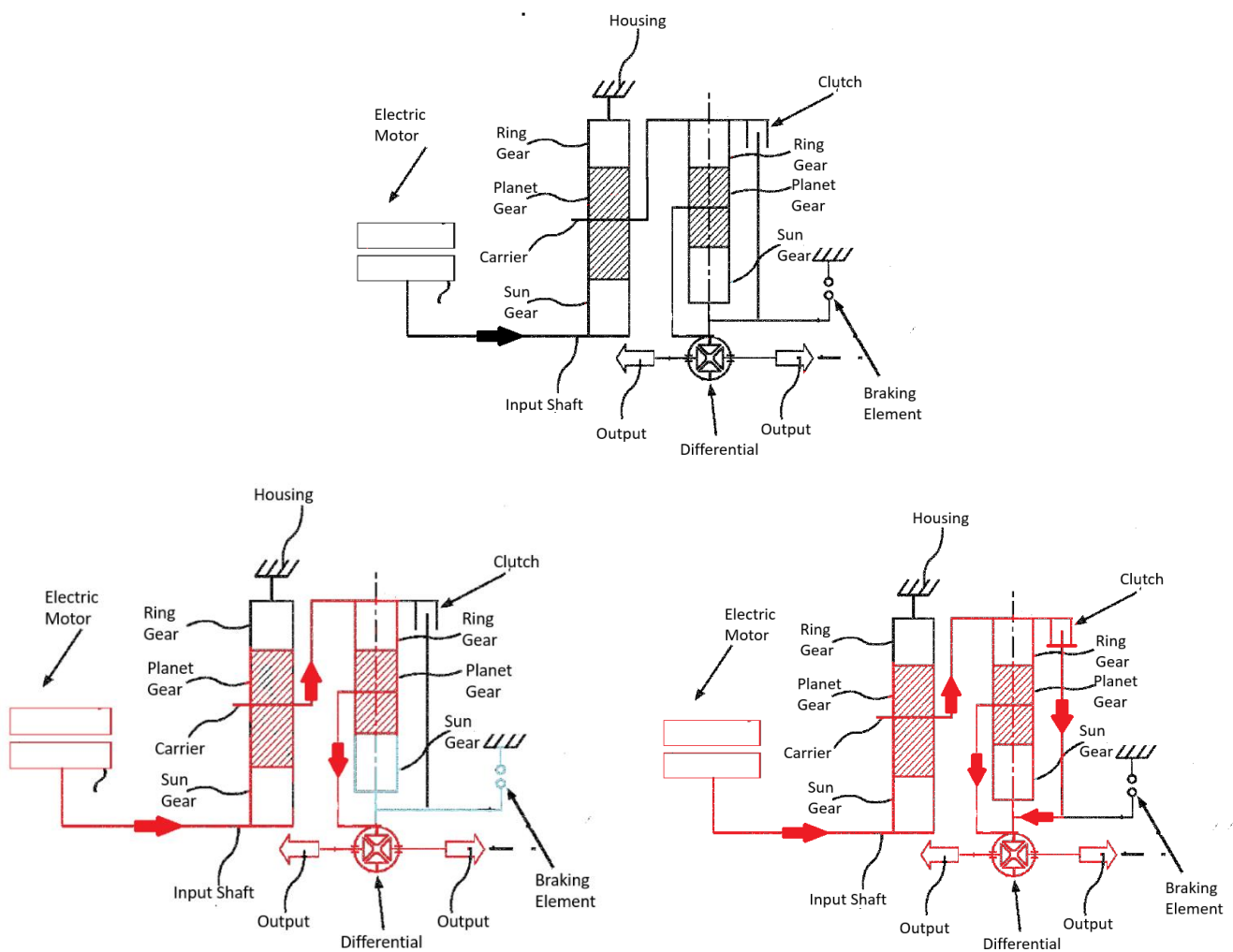
Fig. 18 WO2019115204A1 Patent Design and the Power Flow [19]

2.3.6. WO2019091719A1 , Transmission for an Electric Vehicle

This patent is assigned by ZF Friedrichshafen AG.

The invention includes an electric motor, a dual clutch element, a layshaft transmission system and an oil pump. Oil pump is connected to electric motor, which is always driven regardless from the clutches. When the first clutch (Clutch1) is engaged, the power follows the Shaft1, and there will be a speed reduction in the output shaft. When the second clutch is engaged; the power follows the Shaft2 respectively.
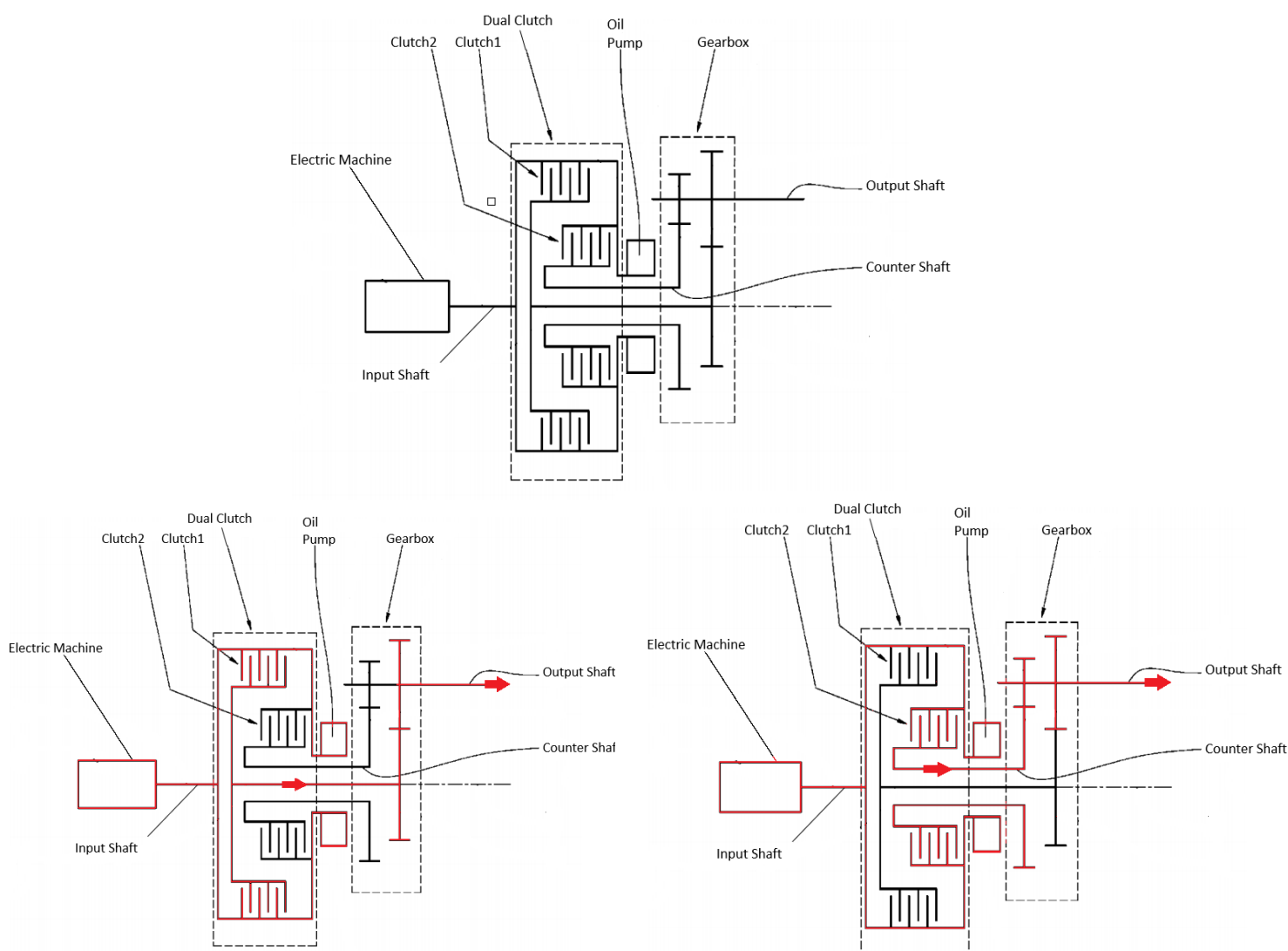
Fig. 19 WO201909179A1 Patent Design and Power Flow [20]

2.3.7. US20160003351A1, Two-Speed Transmission for Vehicle

This patent is assigned by Hyundai Motor Co. Ltd.

The invention includes an electric motor, a planetary gearset which the sun gear is connected to the input and the carrier is connected to the output shaft, clutch elements which is a friction clutch, used in the connection of the sun gear and the carrier and a dog clutch which controls the movement of the ring gear.

When the first speed is engaged, the dog clutch connects the ring gear with the housing and the ring gear is fixed. The friction clutch is not engaged so, there is no connection between the sun gear and the planet carrier. As a result, the power from the motor will be transmitted to sun gear and through the sun gear to the planet carrier respectively and speed is reduced regarding to the gear ratio. When the second gear is engaged the dog, clutch releases the ring gear, the friction clutch create the connection between the carrier and the sun gear. As a result, all of the planetary gear parts will rotate with the same speed and it becomes direct drive.
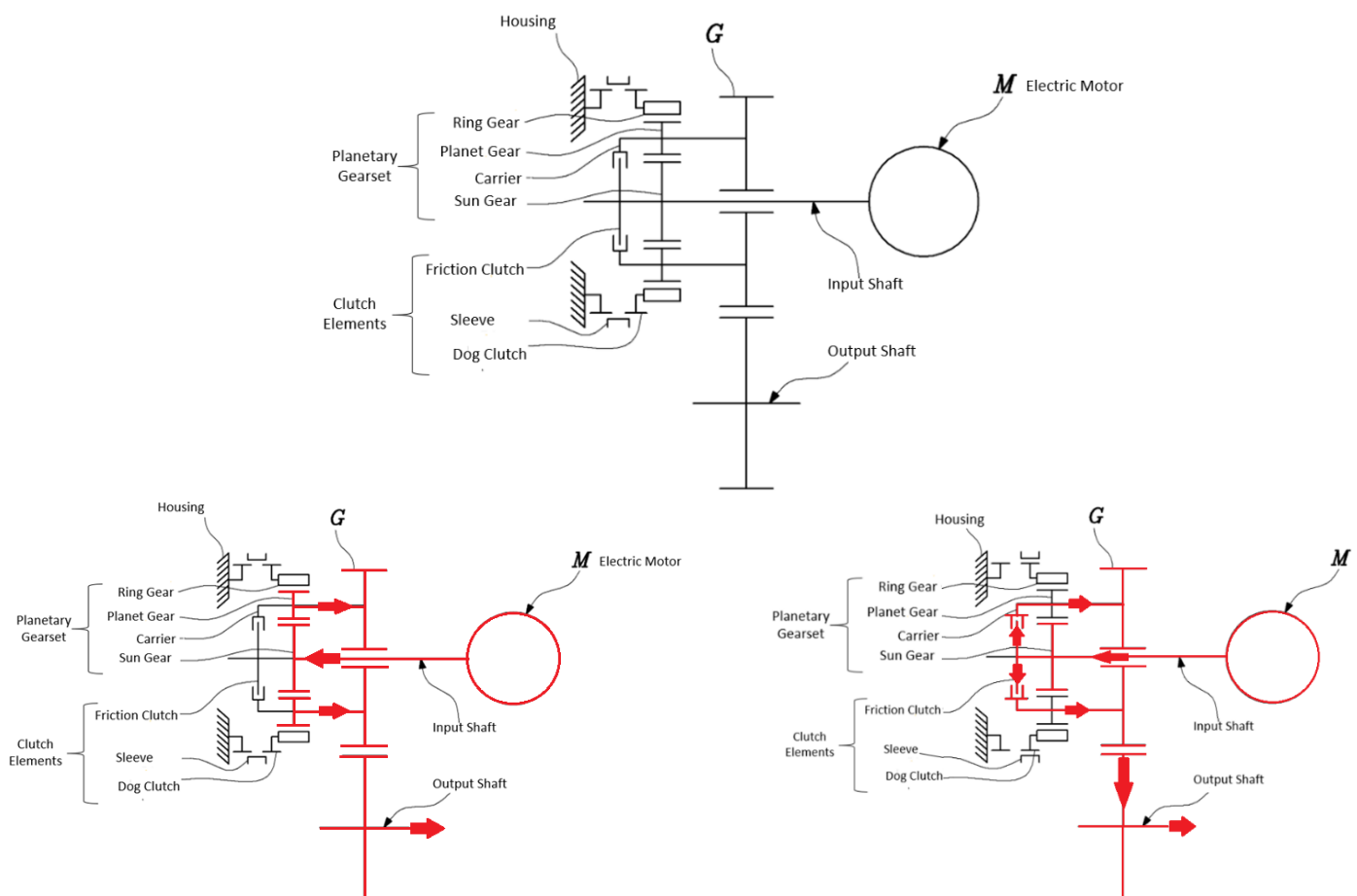
Fig. 20 US20160003351A1 Patent Design and Power Flow [21]

3. Vehicle Parameters and Energy Source Selection

3.1. Parameters of Chevrolet Bolt EV

To start with the design of the vehicle transmission, the most crucial part is determining the parameters of the vehicle and from the data calculating the power requirements of the vehicle. As it can be seen from the current state of art of the vehicles are on the market, the most of them are compact, city car designs. The objective of the thesis is working on a city car which the range of the electric vehicle is adequate for daily urban activities.

From that information the vehicle parameters of the Chevrolet Bolt EV is going to be used for the transmission design.

The specifications are inserted below :

Tab. 2   Specifications of Chevrolet Bolt 2020 [22]

Chevrolet Bolt EV 2020	
Curb Weight (kg)	1616
Width (mm)	1765
Height (mm)	1595

Fig. 21   Chevrolet Bolt 2020 Front View [22]

3.2. Longitudinal Dynamics Calculation

Since the car parameters has been chosen, the longitudinal dynamics properties and the electric motor selection can be done. The main task is to achieve the maximum velocity, which is a required input parameter. So, the maximum velocity is selected as 160 km/h (44.4m/s). With the data of the maximum velocity, the power requirement for the highest speed could be calculated. Driving resistance calculations normally assumed from straight running on horizontal and dry surface.

The traction force required at the drive wheels is called as sum of the acting forces to vehicle.

It is calculated as ;

$$Fres = Fr + Fd + Fa + Fst \qquad \text{Eq. 1}$$

The Fr represents the rolling resistance, Fd represents the air drag, Fa represents the acceleration resistance and Fst represents the gradient resistance. Let's have a detailed look in every term in the formulation.

3.2.1. Rolling Resistance

The rolling resistance is calculated by ;

$$Fr = F_N * f = m * g * f * \cos(\alpha) \qquad \text{Eq. 2}$$

Where , m is the weight of the vehicle, g is the gravity constant and f is the rolling resistance coefficient. In our calculations the rolling resistance coefficient will be accepted as a constant f=0.015 [23].

3.2.2. Air Drag

The air passes through the vehicle in high velocities and creates a drag force.

The air drag is calculated by ;

$$Fd = 0.5 * \rho_{AIR} * C_W * A * v^2 \qquad \text{Eq. 3}$$

In this formula ρ is the density of air. At the pressure 1.013 bar with a relative humidity of 60% at 20°C temperature the ρair =1.2 kg/m3 [23].”Cw” is the drag coefficient, and the Chevrolet Bolt EV's drag coefficient is Cw =0.32[24]. A is the cross-sectional area of the vehicle. The recommended value of the cross-sectional area by Ricardo is A=1.95 m^2.

3.2.3. Acceleration Resistance

When the vehicle velocity increases or decreases, additional inertia forces take place. The acceleration resistance is calculated by ;

$$Fa = m * a * \delta \qquad \qquad \text{Eq. 4}$$

In the formula m represents the weight of the vehicle, a represents the acceleration and the term δ represents the inertial forces of the all rotating masses in the vehicle. The recommended value is provided as δ=1.2 [25].

3.2.4. Gradient Resistance

When the vehicle is going uphill, the gradient resistance takes place. The gradient resistance acting on the vehicle is calculated by ;

$$Fst = m * g * \sin(\alpha) \qquad \qquad \text{Eq. 5}$$

In the formula m represents the weight of the vehicle, g is the gravitational force and α is the angle of the slope of the road. The acting forces on the vehicle going uphill are represented in the figure, Fig.22.
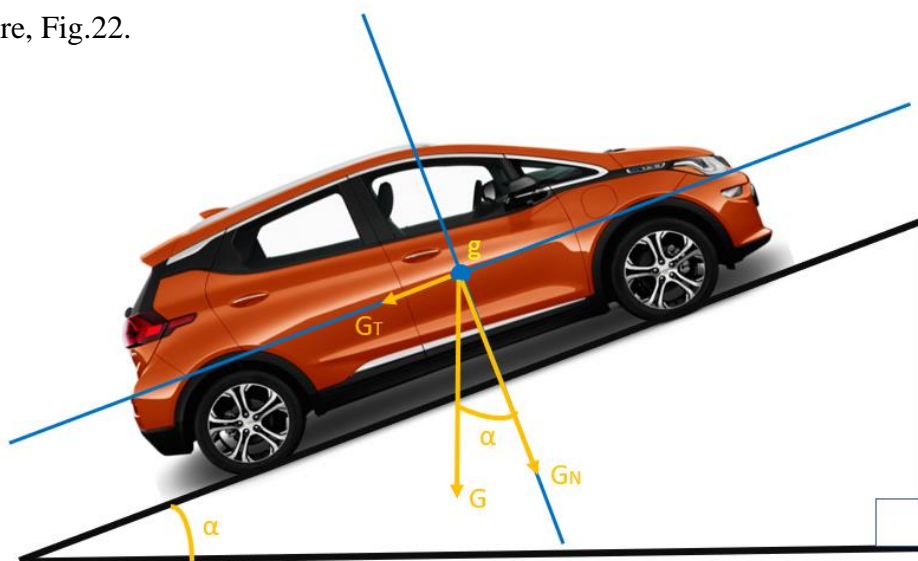
Fig. 22   The Forces Acting on the Vehicle, Uphill Movement

3.3. Total Driving Resistance

In the previous section, the tractive force calculation described in the equation 1.

If the equation would be described in extended version, the equation 1 becomes like ;

$$Fres = Fr + Fd + Fa = m * g * f + 0.5 * \rho_{AIR} * C_W * A * v^2 + m * a * \delta$$

Tab. 3 Input Parameters for Calculation of Total Driving Resistance

Curb Weight [22]	m	1616 kg
Weight of Driver	md	70 kg
Coefficient of Rolling Resistance	F	0.015
Drag Coefficient [24]	Cw	0.32
Cross Sectional Area	A	1.95 m^2
Kinematic Radius of Tire	Rd	0.301 m
Efficiency of Driveline	ηt	0.90
Air Density [23]	ρ air	1.2 kg. m^{-3}
Gravitation Constant	g	9.81m.s^{-2}
Inertia of the Rotating Masses	δ	1.2

When the driving resistance for the maximum velocity calculated, since the car is going with maximum speed, there is no acceleration resistance (a= 0). And the formula would be ;

$$Fres = Fr + Fd = m * g * f * \cos(\alpha) + 0.5 * \rho_{AIR} * C_W * A * v^2 \qquad \text{Eq. 6}$$

$$Fres = Fr + Fd = 1686 * 9{,}81 * 0{,}015 * 1 + 0{,}5 * 1.2 * 0{,}32 * 1{,}95 * \left(\frac{160}{3.6}\right)^2$$

$$Fres = Fr + Fd = 987.6 \, N$$

The power which is transferred to the wheels is equal to multiplication of resistance forces and velocity. From that equation, the cubic equation of the velocity is obtained with power is the unknown value. With that result, the estimation of the power requirement can be observed, an adequate electric motor can be selected.

The power which is transferred to the wheels is calculated by ;

$$P * \eta t = (Fr + Fd) * v \qquad \text{Eq. 7}$$

$$P * \eta t = (m * g * f + 0.5 * \rho_{AIR} * C_W * A * v^2) * v$$

$$P.\eta t = m * g * f * v + 0.5 * \rho_{AIR} * C_W * A * v^3$$

$$P * 0.90 = 1686 * 9.81 * 0.015 * \left(\frac{160}{3.6}\right) + 0.5 * \rho_{AIR} * C_W * A * \left(\frac{160}{3.6}\right)^3$$

$$P = 54635 \, W = 54.6 \, kW$$

As a result, the power requirement of the electric motor is approximately 54.6 kW. With the data calculated, the electric motor selection can be done.

3.3.1. YASA P-400 RS Electric Motor

To be able to make an adequate selection for the electric machine, the required power and torque of the electric machine should be considered. When the market research for the electric motors done the model YASA P-400 RS meets the requirements for the power.

The electric motor P-400 RS provides up to 94% motor efficiency and high maximum speed (up to 8000 rpm) [28].

Fig. 23   YASA P-400 RS PMSMElectric Motor [26]

The specifications of the YASA P-400 RS are listed below :

Tab. 4   YASA P-400 RS Specifications

Mechanical		Electrical	
Casing Diameter	305mm	Peak Torque	Up to 370 Nm
Axial length	106.7mm	Continuous Torque	Up to 200 Nm
Dry Mass	28.2 kg	Continuous Power	Up to 60 kW
Stator Cooling	Oil	Maximum Speed	8000 rpm
Mounting	294mm PCD	Peak Power	160kW
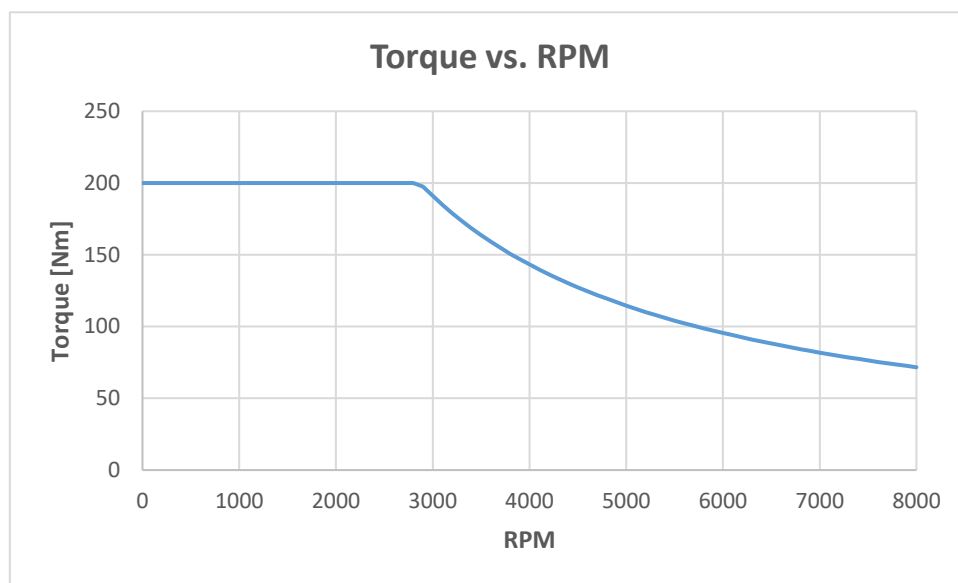
Fig. 24   The Choosen YASA Motor Characteristics

To get the tractive force on the wheels the following formula is used;

$$Ftr = \frac{Me * i * \eta tr}{rd} \qquad \text{Eq. 8}$$

In this formula Me represents the torque of the electric motor, i is the gear reduction ratio, ηtr is the efficiency of the transmission system and rd is the dynamic radius of the vehicle. The only unknown value here in the equation 8 is the gear reduction ratio i, and it can be found from the maximum velocity of the vehicle (160km/h=44.4m/s).

The gear reduction ratio (i) is calculated in the condition of the maximum speed. So, the equation of the maximum speed at the engine is ;

$$Vmax = \omega * rd = \frac{2 * \pi * n}{60} * \frac{rd}{i}$$

In this equation n is the velocity of the motor in rpm , ω is the angular velocity of the motor in s-1, rd is the dynamic radius of the vehicle. Traction Limit Due to Tire Adhesion

If the equation would be converted for finding the gear reduction ratio i, it becomes ;

$$i = \frac{2 * \pi * n}{60} * \frac{rd}{v}$$

$$i = \frac{2 * \pi * 8000}{60} * \frac{0.301}{44.44}$$

$$i = 5.6737$$

The resulting final ratio is equals to **i =5.6737.**
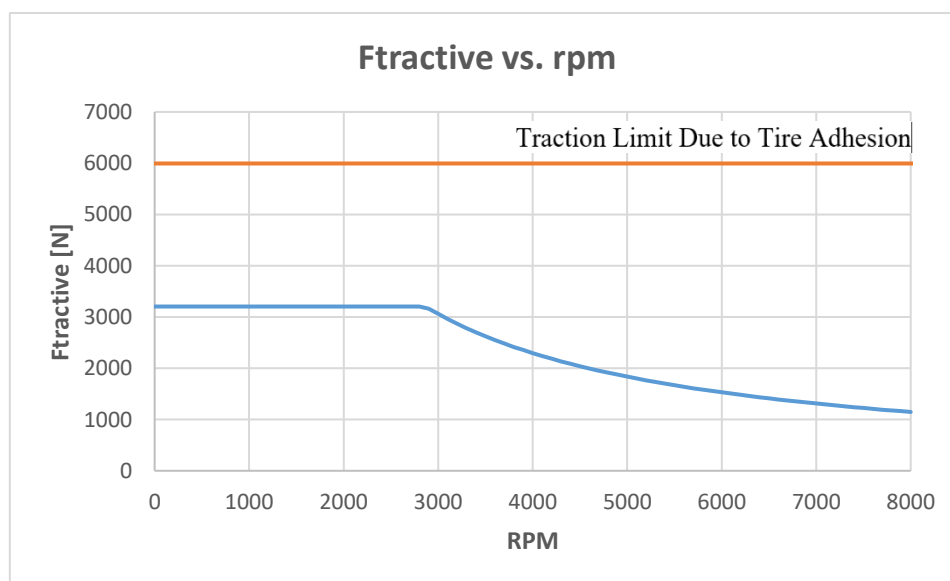
Fig. 25   Tractive Force vs. RPM

From the vehicle parameters given, in the enclosed Microsoft Excel file ;
Torque, Resistance Forces, Tractive Forces, distance taken have been calculated, and from the
results the graphs have obtained. These graphs are ;
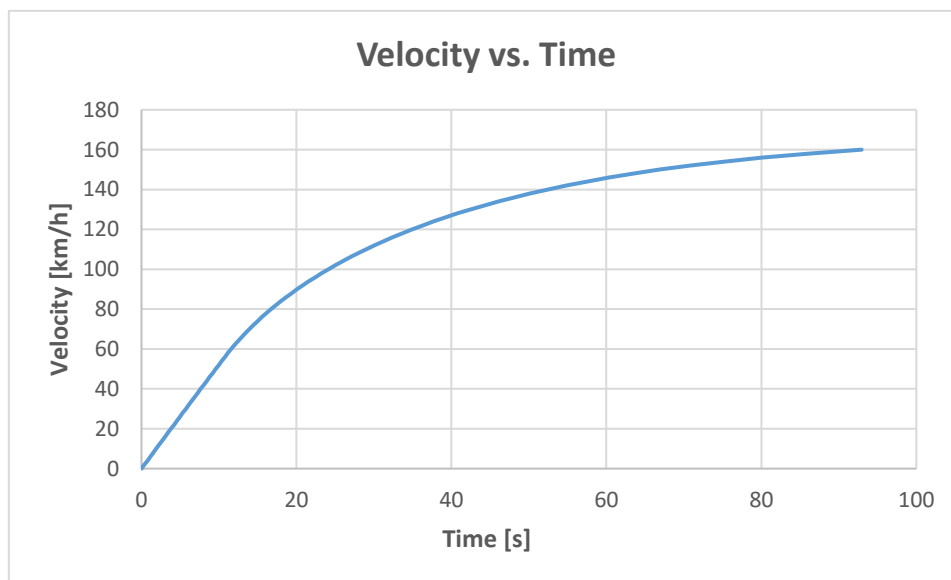
Velocity vs. Time

Fig. 26   Velocity vs. Time

Acceleration vs. Time
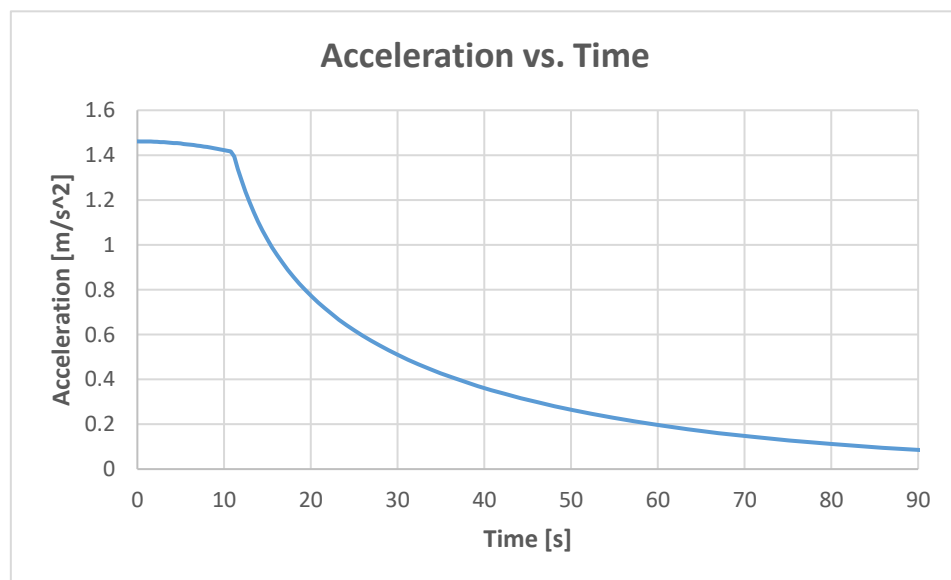
Fig. 27   Acceleration vs. Time
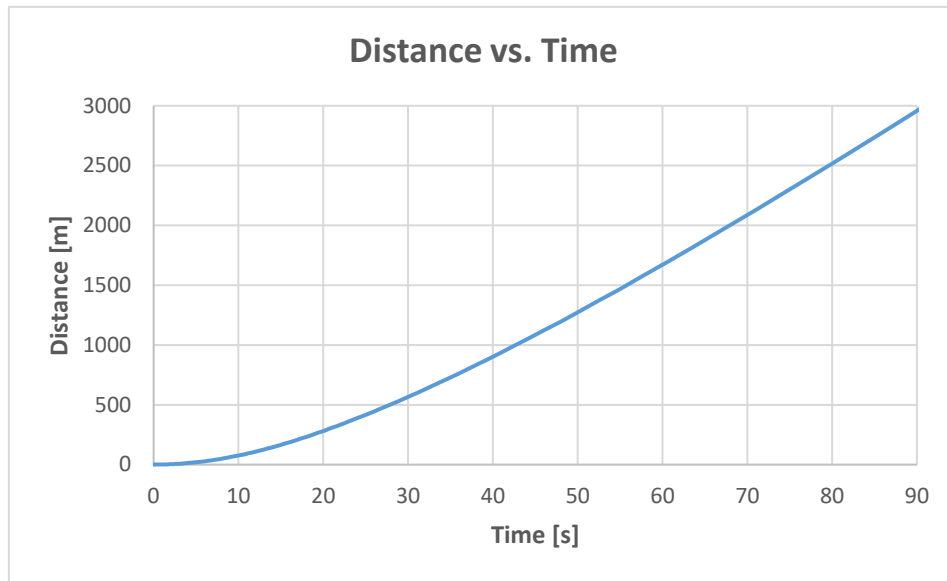
Distance vs. Time

Fig. 28   Distance vs. Time

Acceleration vs. Velocity
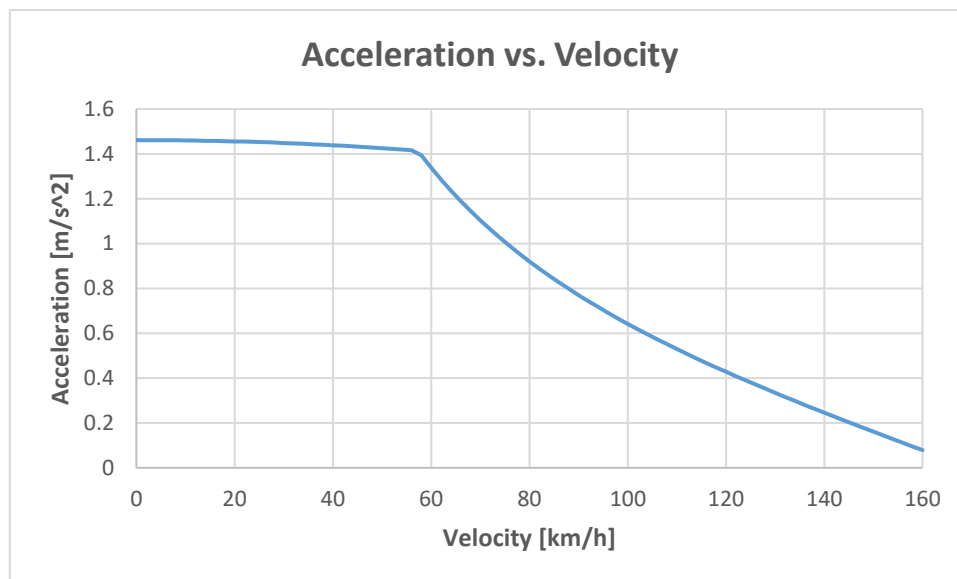
Fig. 29   Acceleration vs. Velocity
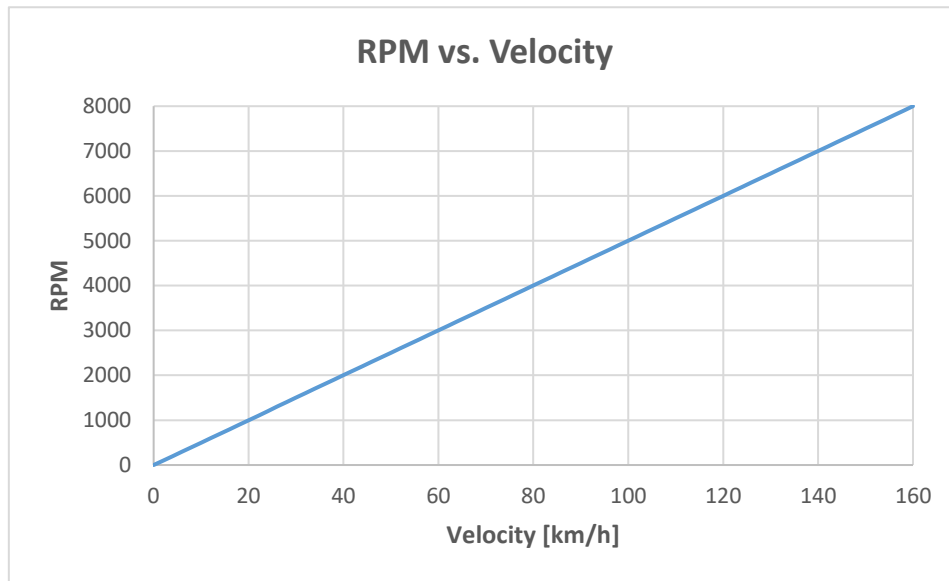
Fig. 30   RPM vs. Velocity
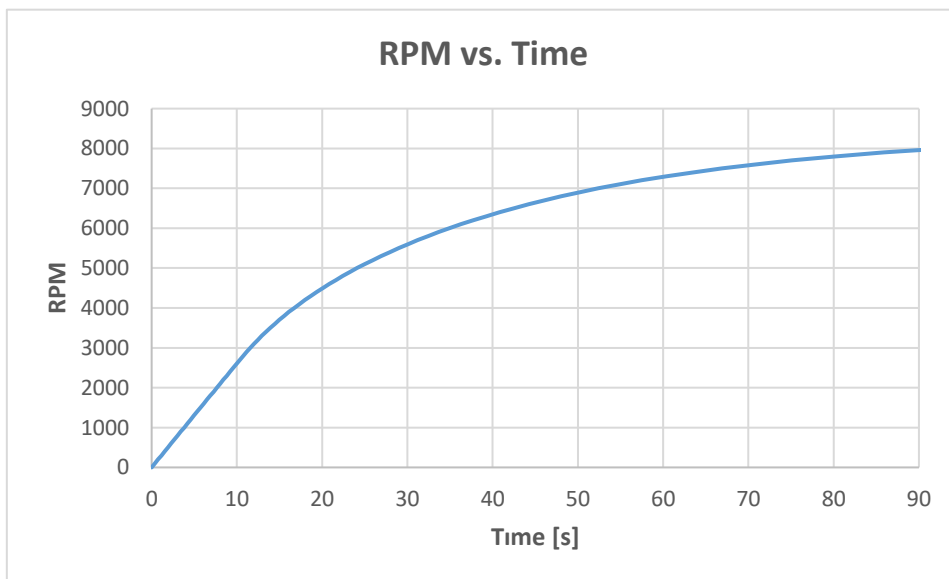
Fig. 31   RPM vs. Time

Also, we would like to know the how many percentages of slope our car could climb maximally.

To be able to calculate that, let's assume there are four passengers which is mpass=280kg and mcargo=50kg, which is mpayload = 330 kg in total.
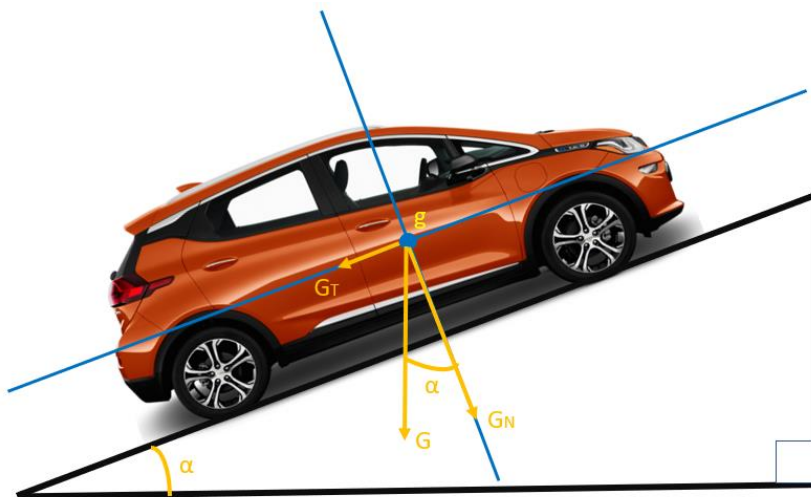
mcurb	1616 kg
mpass	280 kg
mcargo	50 kg
mtotal	1946 kg

To be able to find the maximum slope the car could climb uphill, we need to know the maximum torque which the electric motor can provide continuously, which can be found from the following equation ;

$$Fw = \frac{Me * i}{rd}$$
<div align="right">Eq. 11</div>

$$Fw = \frac{200N * 5.6737}{0.301}$$

The electric motor can provide maximum 1134.74 Nm that, the maximum resistance force which is applied to tires must not be exceed 3769.918 Newtons. The assumption is, the vehicle is in the beginning of the slope and just going upwards, and as a result the drag and acceleration resistance are taken as zero(Fd=0, Fa=0).

The resistance forces applied to vehicle while going uphill;

$$Fres = Fr + Fst \qquad\qquad \text{Eq.12}$$

$$Fres = m * g * f + m * g * \sin(\alpha)$$

$$3769.918 = 1946 * 9.81 * 0.015 + 1946 * 9.81 * \sin(\alpha)$$

$$\sin(\alpha) = 0.18247$$

$$\alpha = 10.51°$$

The maximum slope the car could climb uphill would be found as 10.51° and if it would be converted into a tangent slope in % ;

$$\tan(10.51) = 0.185 = 18.5\%$$

The vehicle could go up **maximum 18.5% slope** without a problem.

4. Efficiency, Mass and Dimension Comparison of the Transmission Systems

In the previous chapters, the transmission systems of the automobiles currently available on the market and the existing patent solutions were discussed. To proceed further, from the solutions three variants will be selected and their efficiency, mass and dimension comparison will be adressed.

The first variant of the transmission system is Audi e-tron's design ; planetary reduction with additional simple parallel gear step to differential.
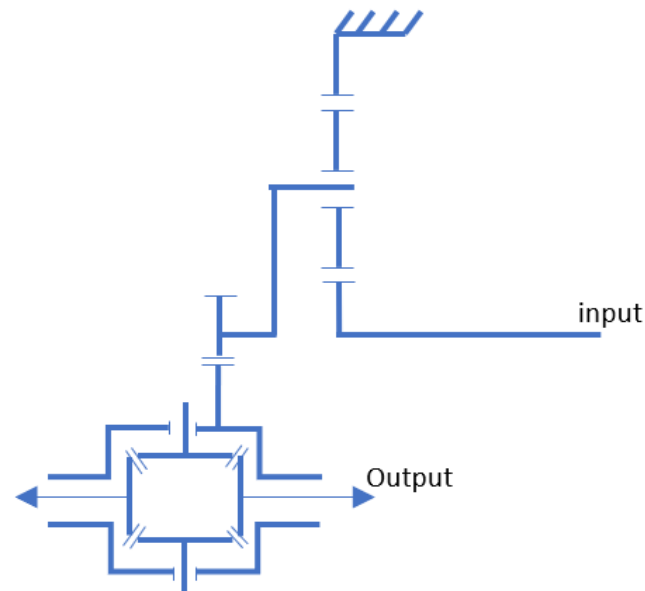
Fig. 32 Planetary Reduction With Additional Simple Parallel Gear Step

Second variant of the transmission system is the common layshaft transmission design which are used in many cars: Two stage parallel shaft with compound gears on the layshaft in the Figure 33.
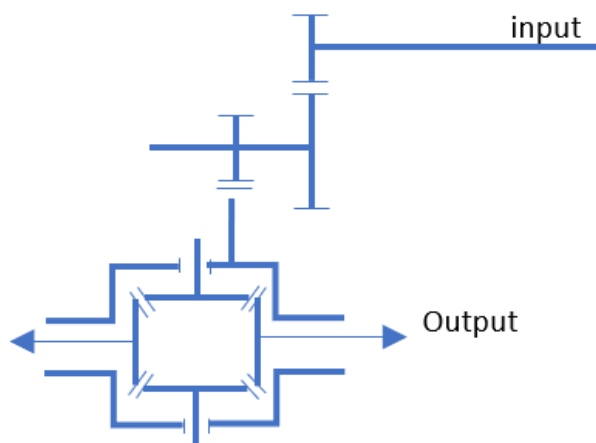
Fig. 33 Two Stage Parallel Shaft, Layshaft Transmission

The third variant of the transmission system is planetary gear reductor with compound planets, used in Jaguar I-Pace. In this design as mentioned, there are two gear meshes, in the first mesh, there is no ring gear and the planet of the first mesh is compound with the second planet.
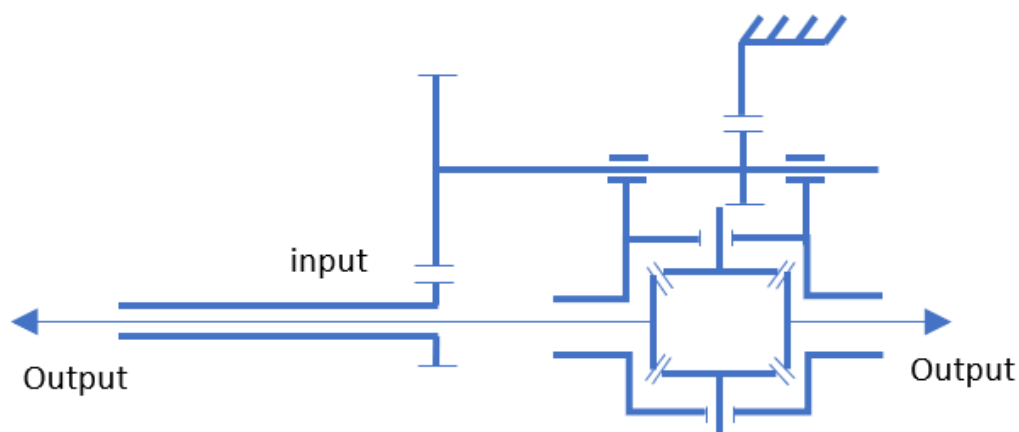
Fig. 34 Planetary gear reductor with Compound Planets

4.1. Efficiency and Losses

There is no device in the world which is working with 100% efficiency. This situation is also valid for the transmission systems. There are several losses taken into account when the transmissions are running. The efficiency is not constant due to unavoidable losses, and due to increase of the rotational speed, temperature of the oil, viscosity of the oil and temperature changes and it cause a decrease in output torque[27]. In this chapter all the losses in the efficiency are undertaken.

4.1.1. Lubrication Losses

In the current state of art automotive transmissions the oil splash system is being used. Since their effect on the losses can not be calculated analytically, by the help of experimentation method the lubrication losses are found. In the ordinary conditions, the lubrication losses are less than 1%. ($\zeta_{losses} < 1\%$)[28]. But, to be able to estimate their numerical value, the lubrication losses will be accepted as $\zeta_{losses} = 1\%$. This approximation is going to be used in all variants of the transmissions calculation.

4.1.2. Meshing (Tooth-Friction) Losses

The meshing losses are the most important part of the losses. The accuracy of the efficiency losses in meshings are difficult but engineers could provide some equation for their calculation. The approximate efficiency losses for helical gears are calculated as ;

$$\zeta helical = \frac{f *. \pi * \varepsilon}{4 * \cos \beta} * \left(\frac{1}{Z_P} \pm \frac{1}{Z_G} \right)$$
Eq. 13

In this equation f represents the coefficient of friction, ε represents transverse contact ratio and β represents the helix angle. Instead of making this long calculation, there is a table described in Mrs. Achtenova's book [29]. For simplification the mean value of the range of the table are going to be used.

Tab. 5   Reference Values of Meshing Friction Losses [29]

Type of Gear	Range of Coefficient of losses	Coefficient of losses used
Spur or Helical Gears (External Gears)	0.015-0.025	**0.02**
Annular Gears (Spur of Helical)	0.005-0.015	**0.01**
Bevel Gears	0.02-0.03	**0.025**
Hypoid Gears	0.025-0.04	**0.03**

4.1.3. Losses in Bearings

The power loss in the bearings is calculated from the formula;

$$|P_{\zeta_{bearing}}| = |M_\zeta| * \omega_P \qquad \text{Eq. 14}$$

$$|M_\zeta| = F * f * r_m \qquad \text{Eq. 15}$$

In these equations, M_ζ represents frictional torque, F is the load of bearing and f is the coefficient of friction and r_m is the main radius on which the elements are rolling. Also for simplification of the estimation and since there is no numbers for calculation, Ricardo suggests that, each shaft requires an existance of two bearings and each bearing cause approximately 1% lose in the efficiency.

The components which are supposed to be the same in all considered variants are not described in the table(differentials, half shafts etc.).

With those approximations, let's analyse all three variants ;

Variant 1 ;

Planetary reduction with additional simple parallel gear step to differential.
Since it is a planetary gearbox the efficiency calculation is done by the Willis formulas and the torque equilibrium in the system.
In this variant the number of teeth of the ring gear is z_R =108, and number of teeth of the sun gear is $z_S = 69$.

The basic ratio is found as;

$$i_{Basic} = \frac{Z_R}{Z_S} = \frac{108}{69} = 1.565 \qquad \text{Eq. 16}$$

$$i_{PGS} = 1 + i_{Basic} = 2.565 \qquad \text{Eq. 17}$$

With the internal and external meshes the efficiency coefficient is ;

$$\eta = 1 - 0.2 - 0.1 = 0.97 \qquad \text{Eq. 18}$$

Willis formulas provides the speed equation ;

$$n_S - \left(1 + \frac{z_R}{z_S}\right) * n_c + \frac{z_R}{z_S} * n_R = 0 \qquad \text{Eq. 19}$$

In the equation ns represents the speed of the sun , nc represents the speed of the carrier and nr represents the speed of the ring . It is known that the variant one has stationary ring, so the nr = 0. And the sun gear speed is equal to the speed of the electric motor.

The torque equilibrium represents;

$$T_s + T_R + T_c = 0 \qquad \text{Eq. 20}$$

Without Loses ;

$$\frac{z_R}{z_S} * T_S - T_R = 0 \qquad \text{Eq. 21}$$

With Losses ;

$$\frac{z_R}{z_S} * \eta^{Sign(P_S)} * T_S - T_R = 0 \qquad \text{Eq. 22}$$

Powers are calculated by the multiplication of the torques and the rotational speeds ;

$$P = T * n \qquad \text{Eq. 23}$$

With the matrix method the relative speeds of the all rotationg elements and their powers with and without losses can be found;

				PG ratio =	2.56522								
zS/zR	1.56522												
Eta	0.97												
			nS	nC	nR				nS =	1			
		Willis formula	1	-2.5652	1.56522		0		nC =	0.38983			
		fixed Ring	0	0	1		0		nR =	0			
		Sun is input	1	0	0		1						
		Without Losses	TS	TC	TR				TS =	1	PS =	1	
		Torque relationship	1.56522	0	-1		0		TC =	-2.5652	PC =	-1	
		Torque eqilibrium	1	1	1		0		TR =	1.56522	PR =	0	
		Torque input	1	0	0		1						
		With Losses	TS	TC	TR				TS =	1	PS =	1	
		Torque relationship	1.51826	0	-1		0		TC =	-2.5183	PC =	-0.9817	
		Torque eqilibrium	1	1	1		0		TR =	1.51826	PR =	0	
		Torque input	1	0	0		1						

Fig. 35   Efficiency Calculation of The Variant One

As it can be seen from the Figure 35, since the input efficiency is 100%, the 98.17% of the power can be provided from the output. The amount of the loss is approximately 2% in the planetary gear set.

Variant 2 :

Two stage parallel shaft with compound gears on the layshaft.

The transmission has ;

- Input shaft : 2 bearings = **2%**
- Counter Shaft : 2 bearings = **2%**
- 2 external meshes = **4%**
- Churning of the oil = **1%**
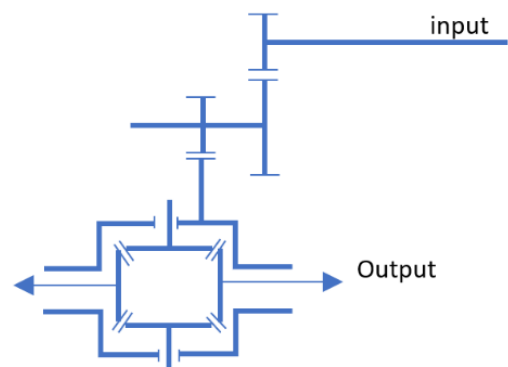
As a result with the simplified efficiency estimation total loss would be 9% and total efficiency would be 91%.

Variant 3 :

Planetary gear reductor with compound planets.

With the matrix method, the relative speeds of the all rotating elements and their powers with and without losses can be found (the same method applied to the variant 1) ;

			PG ratio :	5.54087							
zS/zPS*zPR/zR	4.54087										
Eta	0.97										
			nS	nC	nR						
		Willis formula	1	-5.5409	4.54087		0	nS =	1		
		fixed Ring	0	0	1		0	nC =	0.18048		
		Sun is input	1	0	0		1	nR =	0		
		Without Losses	TS	TC	TR						
		Torque relationship	4.5409	0	-1		0	TS =	1	PS =	1
		Torque eqilibrium	1	1	1		0	TC =	-5.5409	PC =	-1
		Torque input	1	0	0		1	TR =	4.54087	PR =	0
		with losses	TS	TC	TR						
		Torque relationship	4.4046	0	-1		0	TS =	1	PS =	1
		Torque eqilibrium	1	1	1		0	TC =	-5.4046	PC =	-0.9754
		Torque input	1	0	0		1	TR =	4.40464	PR =	0

Fig. 36 Efficiency Calculation of the Variant Three
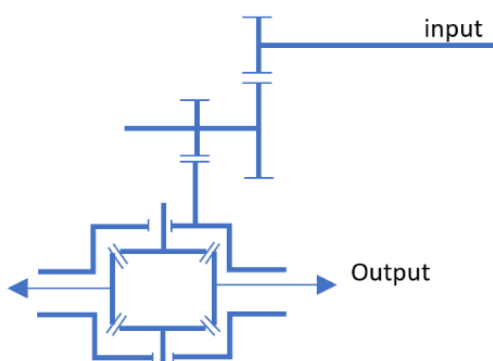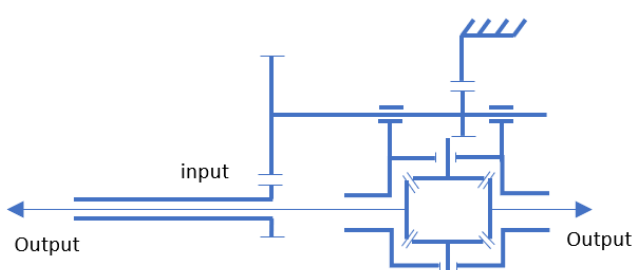
As it can be seen from the Figure 36, since the input efficiency is 100%, the 97.54% of the power can be provided from the output. The amount of the loss is approximately 2.46%. In the planetary gear set.

Tab. 6   Efficiency Estimation of the Variant 1

Variant 1 :	
	• Input shaft : 2 bearings = **2%** • Carrier Shaft : 2 bearings =**2%** • The loss in the Planetary Gearbox **≈2%** • External Mesh of Final Drive=**2%** • Churning of the oil **: 1%**
	TOTAL LOSS : 9% **TOTAL EFFICIENCY : 91%**

Tab. 7 Efficiency Estimations of Variant 2 and 3

Variant 2 :	
	• Input shaft : 2 bearings = **2%** • Counter Shaft : 2 bearings =**2%** • 2 external meshes =**4%** • Churning of the oil = **1%**
	TOTAL LOSS : 9% **TOTAL EFFICIENCY : 91%**
Variant 3 :	
	• Input shaft : 2 bearings = **2%** • The loss in the Planetary Gearbox **≈** 2.5**%** • Churning of the oil = **1%**
	TOTAL LOSS : 5.5% **TOTAL EFFICIENCY : 94.5%**

As it can seen from the Tables 6 and 7, it is expected that the third variant is more efficient than the first and the second variants.

$$\eta_{VARIANT\ 3} > \eta_{VARIANT\ 1} > \eta_{VARIANT\ 2}$$

4.2. Mass and Dimension Estimation

Since there are no parameters are known with the powertrains the mass estimation will be done by comparing the number of parts in the transmission system and the comparison will be prepared.

For the dimension comparison, it will be done from the transmission's cross sectional drawings and by that their dimensions will be estimated.

4.2.1. Mass Comparison of the Selected Transmission Systems

The first variant; planetary reduction with simple parallel gear step consists of :

- Differential
- Input Shaft
- One set of planet system
- Electric Motor (YASA P 400-RS)
- One parallel gear step (two gear system)
- Half shafts
- Medium Size Medium Weight Casing

The second variant ; two stage parallel shaft, layshaft transmission consists of :

- Differential
- An input shaft with one gear system
- A counter shaft with two gear system
- Electric Motor (YASA P 400-RS)
- Half Shafts
- Large Heavy Casing

The third variant ; Planetary gear reductor with Compound Planets consists of

- An hollow input shaft
- Differential
- First planetary reduction system without ring gear
- Second planetary reduction set without sun gear
- Half Shafts
- Electric Motor (YASA P 400-RS)
- Small Light Compact Casing

It can seen that with the given variants the differential and the electric motor and half shafts are exist on all variants and their effects on the mass are same.

The first variant has single set planetary gear has input shaft, one set of planetary system and one parallel gear step and a medium size medium weight casing which is different from the other variants.

The second variant has an input shaft with one gear pair and a counter shaft with two gear pairs with a large and heavy casing. .

The third variant has an hollow input shaft the first planetary gear set without ring gear and second planetary reduction set without sun gear and a small light and compact casing.

It is expected that the second variant is heavier than the first variant and the third variant and the lightest variant seems the third one.

$$m_{VARIANT\ 2} > m_{VARIANT\ 1} > m_{VARIANT\ 3}$$

4.2.2. Dimension Comparison of the Selected Transmission Systems

To be able to make a dimension comparison of the selected comparison systems, the configuration of the systems must be known.

The first variant is planetary single speed reduction. Sun is input, carrier is output, ring is stationary. There is additional simple parallel gear step to differential. It is assumed that the length of the transmission system is A, which can be seen in the Figure 35.

Fig. 37 Configuration of the First Variant of the Selected Transmissions

The second variant is two stage parallel shaft with compound gears on the layshaft.

It is assumed that the length of the transmission system is B, which can be seen in the Figure 36.
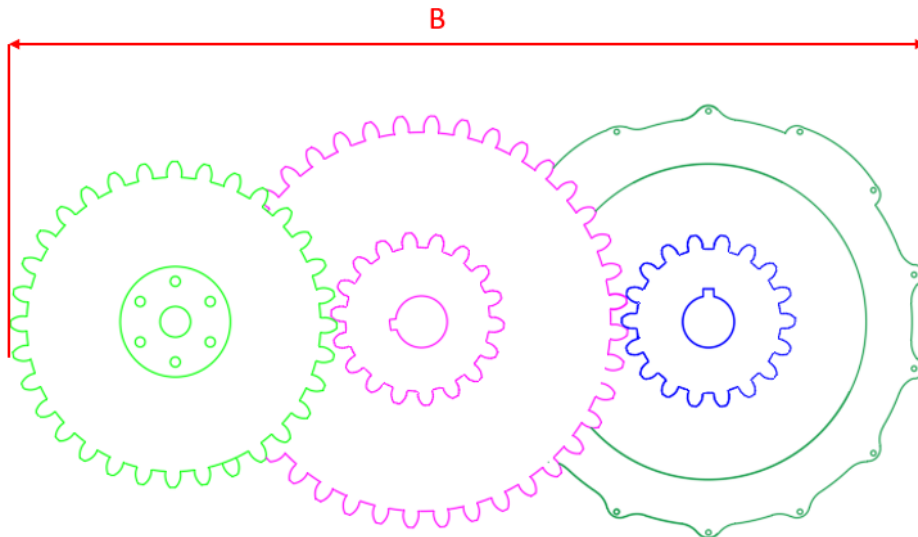
Fig. 38 Configuration of the Second Variant of the Selected Transmissions

The third variant is Planetary gear redactor with compound planets. Sun is input, carrier is output, ring is stationary. Sun is meshing with bigger planet and ring is meshing with smaller planets. It is assumed that the length of the transmission system is C, which can be seen in Figure 37.
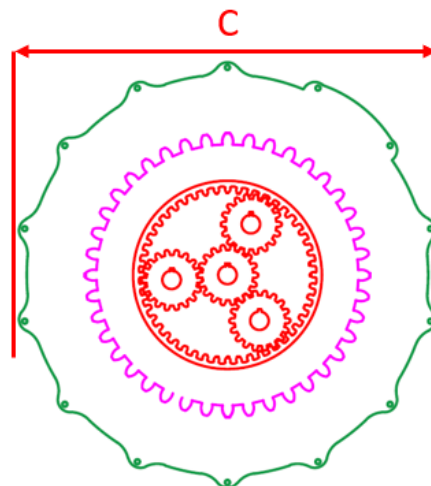
Fig. 39 Configuration of the Third Variant of the Selected Transmissions

As a result, in dimensional point of view, it is expected that length A is larger than length C and smaller than length A. In other words, the second variant occupy more space than the variant two and variant one, respectively.

$$d_{VARIANT\,2} > d_{VARIANT\,1} > d_{VARIANT\,3}$$

5.   The Planetary Gearset Design

In the previous chapters, the designs used in the vehicles on the market, patents and solutions have been discussed, the car have been selected and from the given vehicle parameters, resistances and the total ratio requirement have been calculated.

In this chapter, for the requirement of the design, the diameters of the parts, number of teeth of each element, the requirement of the assembly will be discussed. Also with the help of the Ricardo's engineering softwares SABR/GEAR, the duty cycle analysis will be done and the stresses which are affecting the gears during operation and their values will be calculated and the results will be discussed.

5.1.   Selection : Planetary Single Speed Reduction

In the previous chapter, since there are more efficient alternatives are exist instead of the selected gearbox for calculation, the requirement for this thesis to work on a design of a planetary gearset, not a layshaft transmission.

Also, if the design variants number one and three are compared, it can be seen the variant three transmission, only the transmission extracted from the powertrain seems more efficient than the selected variant. However, the drivetrain including the homokinetic joints at variant three might be subject of high loses and premature wear. Because the ground clearance in our application have to be gapped and only way to keep the ground clearance with variant three, is keeping the halfshafts highly misaligned.

Variant three is a coaxial design. In order to keep the axes from the differential towards from one of the wheels the rotor of the electric motor should be hollow. It requires a modification in the YASA E-Motor's rotor. Since it is not known, Variant three requires modification of YASA rotor to be hollow. In order to keep the axes from the differential towards from one of the wheels.
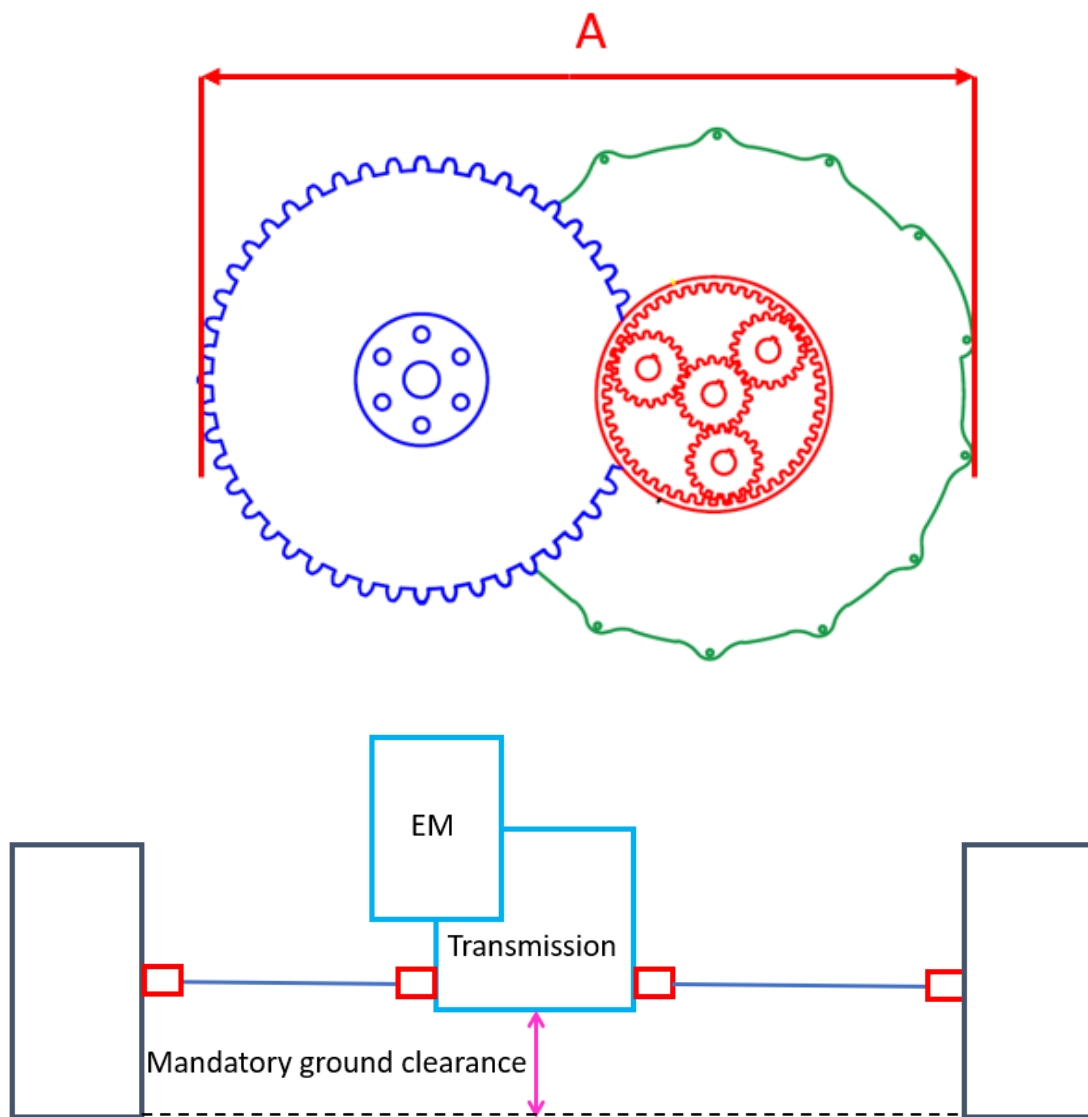
Fig. 41 Configuration of the Selected Variant
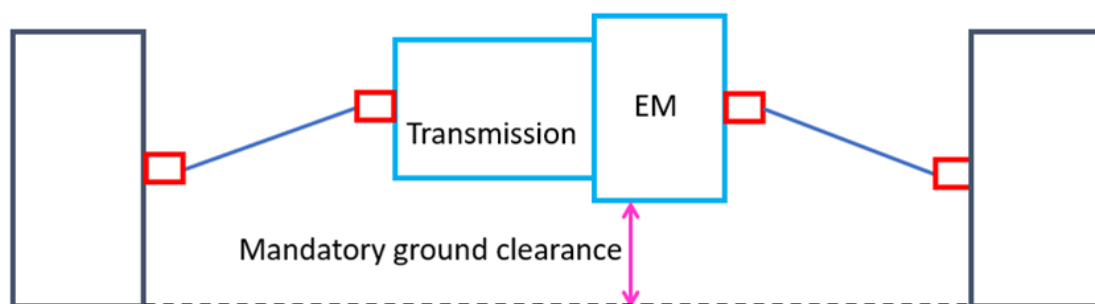
Fig. 40 Configuration of Variant Three

5.2. The Ratio and The Planet Diameters Calculation

In chapter three, with the given vehicle parameters the total ratio calculation have been done and it founded as i=5.6737. This ratio is total transmission ratio including the ratio of planetary gear set and the final drive ratio. The total transmission ratio is equal to the multiplication of the planetary gearset ratio and the final drive ratio.

$$i_{TOTAL} = i_{sun.carrier}^{ring} * i_{FD}$$

Eq.24

To be able to find the number of teeth of the planetary gearset components, the planetary gearset ratio must be found. The only known value is the total ratio, and as an engineering approach the final drive ratio could be estimated. The final drive ratio, can be choosen in a single step as $2 \leq i_{FD} \leq 7$ [30]. Ricardo suggests that the optimal final drive ratio varies between 2.1 to 2.5. As a result, the final drive ratio have been selected as $i_{FD} = 2.2$ for the calculations.

From that ;

$$5.6737 = i_{sun.carrier}^{ring} * (2.2)$$

$$i_{sun.carrier}^{ring} = \mathbf{2.578}$$

The planetary gearset ratio have been found as $i_{sun.carrier}^{ring} = \mathbf{2.578}$.

The design engineers are using a distinct notation to determine the power flow through the planetary gear set. In the notification, subscripts and superscript are used in order to show input, output and fixed element in the PGS.

$$i_{x.y}^{Z}$$

Fixed Element

Output Element

Input Element
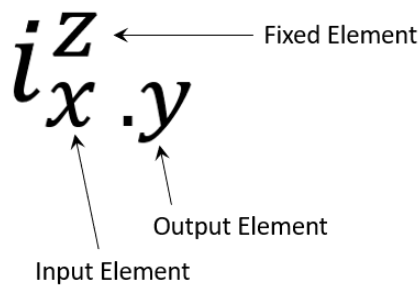
Fig. 42 Designation of Subscripts and Superscript

This ratio is the planetary gearset ratio when the sun is input, the carrier is output and the annulus(ring is stationary). So it could be written as ;

$$i_{PGS} = i_{sun.carrier}^{ring} = 2.578$$

5.2.1. The Basic Ratio of the Planetary Gearing

To be able to make a determination for the ratios, the geometrical parameters of the planetary gear systems are used. The determination is possible only for the mechanisms which have a fixed axis. In the analysis of the planetary gear drive trains, the carrier's rotation is restricted, by that way the drive train acts like a regular mechanism with fixed axis. By that way the pitch diameters or the number of teeths can be calculated. So the basic ratio is notated as ; $i_{sun.ring}^{r}$.

Now, to be able to continue the calculations; the basic ratio should be calculated. The basic ratio can be calculated by the help of the Willis formulas (sun is input, carrier is output and ring movement is restricted (ω_{ring}=0));

$$i_{sun.ring}^{carrier} = \frac{\omega_{sun} - \omega_r}{\omega_{ring} - \omega_r} = -\frac{\omega_{sun} - \omega_r}{\omega_r} = 1 - \frac{\omega_{sun}}{\omega_r} \qquad \text{Eq.25}$$

$$i_{sun.ring}^{carrier} = 1 - i_{sun.carrier}^{ring}$$

From that equation the basic ratio would be ;

$$i_{sun.ring}^{carrier} = 1 - 2.578$$

$$i_{sun.ring}^{carrier} = -1.578$$

The basic ratio is $i_{sun.ring}^{carrier}$ =1.578. The minus sign in the beginning means that the output rotates with a different sense of rotation than the input. It is called as negative ratio [31].

After the calculation of the basic ratio, the pitch diameter of the planetary gear system, and the numbers of teeth of each system can be calculated.

5.2.2. *Pitch Diameter and Teeth Number Calculation of the Planetary Gearset Components*

Since the basic ratio is known, now the pitch diameter of the components and the number of teeth of the components can be calculated.

To prevent any failure during operation, from comparison to other benchmarks; the diameter under bearing in the planet gear (pin diameter) must be higher than 10 milimeters. According to that information, the sun gear's pitch diameter is assumed as 110 milimeters (Fig 40. length A).
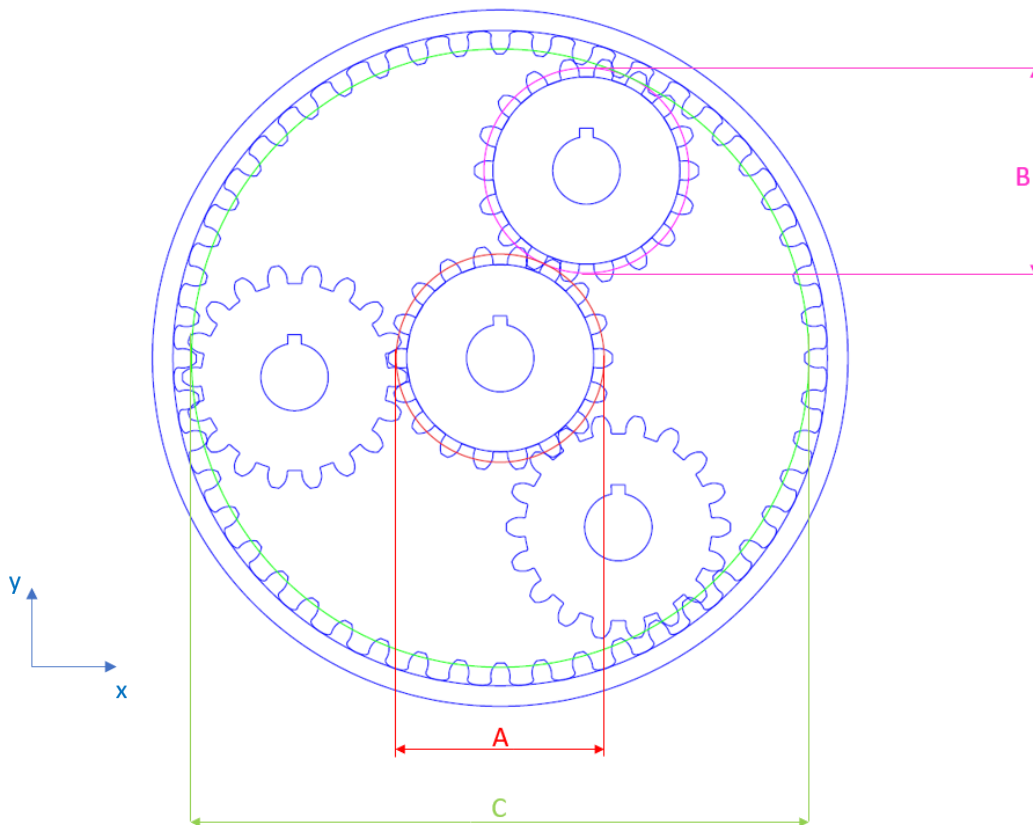
Fig. 43 Illustration for Showing the Pitch Diameters of the Components

The multiplication of basic ratio and the sun gear pitch diameter gives the pitch diameter of the ring gear. The result must be in the absolute value because ,the negative ratio only represents the direction of the output, which is the opposite direction of the input.

$$P_{ring} = \left| i_{sun.ring}^{carrier} \right| * P_{sun} \qquad \text{Eq.26}$$

From this calculation the pitch circle diameter of the ring(Fig.40 length C) will be found as;

$$\boldsymbol{P_{ring} = 173.685\ mm}$$

The difference between the ring pitch diameter and sun pitch diameter gives the pitch diameter of the planet gear(Fig. 40 length B) ;

$$P_{planet} = \frac{P_{ring} - P_{sun}}{2} \qquad \text{Eq. 27}$$

From this calculation the pitch circle diameter of the planet will be found as;

$$\boldsymbol{P_{planet} = 31.8425\ mm}$$

The module m is the ratio of the pitch diameter to the number of teeth. How big or small the gear is indicated by the module value. In this application the module m is accepted as 1.5 milimeters (m=1.5 mm).

The gears which are used in the design is helical gears. The main reason why the helical gears are used is the effect of noise. Since the electric vehicles does not have an internal combustion engine, which is a source of noise, it is wanted that the transmissions for an electric vehicle operates as silent as possible.

Since the gears are helical, there must be the helix angle identification. The helix angle is the angle between the helix and an axial line on its right [33]. The standard helix angle application is between 15° to 30° for helical gears[34]. For our application the selected number of the helix angle is **ß = 20°.**

The tooth height (h) and the dedendum (hf) are functions of module. They are calculated by ;

$$h = 2.5 * m \qquad \text{Eq. 28}$$

$$h_F = 1.5 * m \qquad \text{Eq. 29}$$

The dedendum and the tooth height are found as h=3.75mm and hf=2.25mm, respectively.

The root diameter of the planet can be found by the difference between the planet diameter and the deddendum diameter;

$$d_{ROOT} = P_{Planet} - 2 * h_F \qquad \text{Eq. 30}$$

The needle bearing diameter which is going to be used inside of the planetary gearset is calculated by ;

$$d_{BEARING} = d_{ROOT} - 2 * 1.2 * h \qquad \text{Eq. 31}$$

It is assumed that the roller in the needle bearing has size of 3 milimeters. And the diameter under the bearing is calculated by ;

$$d_{UB} = d_{BEARING} - 2 * d_{roller} \qquad \text{Eq. 32}$$

As a result the root diameter found as $d_{ROOT} = 27.34 \ mm$, the bearing diameter found as $d_{BEARING} = 18.34 \ mm$,and the diameter under the bearing is found as; $d_{UB} = 12.34 \ mm$ respectively. This calculation provides our necessary amount the diameter under the bearing must be higher than 10 milimeters. The bearing selection can be done according to this values calculated, from a catalog.

Since there are the pitch diameter and module values, the number of teeth of the each element can be calculated. The number of teeth can be found from the division of pitch diameter by the module. This calculation is valid for spur gears. For the helical gears the effect of the helix angle should be considered, and it changes the formula slightly as ;

$$z = \frac{P}{m} * \cos(\text{ß})$$

Eq. 33

When the formula is applied to the sun, planet and the ring gears they will be found as;

$$z_{sun} = 68.91 \; teeth$$

$$z_{planet} = 19.94 \; teeth$$

$$z_{ring} = 108.81 \; teeth$$

The founded numbers are not integer, the numbers must be integer and, the condition for assembly of the planets must be considered. This condition is given by ;

$$\frac{z_{ring} + z_{sun}}{\#p} = Integer$$

Eq. 34

This equation is must be correct for the number of planets are equally spaced inside the set. So, there can be small distribution placed for the number of teeth by rounding off the numbers to the close integer value. The effect of rounding off has a very small effect on ratio, that it is acceptable.

When the teeth numbers are rounded off to an integer value they will be founded as ;

$$z_{sun} = 69 \; teeth$$

$$z_{planet} = 20 \; teeth$$

$$z_{ring} = 108 \; teeth$$

If the condition is checked for the assemble ability of the planetary gearset with equispaced planets, when the aimed number of planets are using the system is going to be three ;

$$\frac{z_{ring} + z_{sun}}{\#p} = \frac{108 + 69}{3} = 59$$

The solution of the condition is 59, and thus it is an integer number, so the condition is fulfilled, and it is known that the assembly of the planetary gearset is possible.

5.2.3. Pitch Diameter and Teeth Number Calculations of the Final Drive

The final drive diameter and teeth number calculations are done with the same method as in the planetary gearset calculation. The only issue must be considered is, the center distance of the differential must be longer than the electric motor radius for prevent a collision and make the assembly possible.

The module and the helix angle selected differently for the Final Drive and the pinion pair; m_{FD} =4 mm, β_{FD}=28°. The reason behind this is to keep the contact ratio from 3 to 4 to operate the gearsets as silent as possible. To keep the contact ratio high the width of the gears selected high as well.

First of all, it is necessary to calculate the centre distance that, the pitch diameters of the final drive and the pinion calculation possible.

Tab. 8   Required Parameters for Centre Distance Calculation

Electric Motor Radius	147 mm
Output Shaft Radius	15 mm
Clearance	15 mm

The centre distance is calculated as the summation of all three parameters are described in the Table 8. As a result , the centre distance is found as 177 milimeters. With this information obtained, the pitch diameter calculation for both final drive and the pinion can be done.

The ratio, and the center distances are known; so it can be said that the multiplication of the final drive ratio and the pinion pitch diameter gives the final drive pitch diameter. It is calculated as ;

$$\frac{d_1 + d_2}{2} = 177 \ mm$$

$$d_1 * i_{FD} = d_2$$

From that calculation the pitch diameters of pinion and the final drive are calculated as;

$$d_1 = 110.625 \ mm$$

$$d_2 = 243.375 mm$$
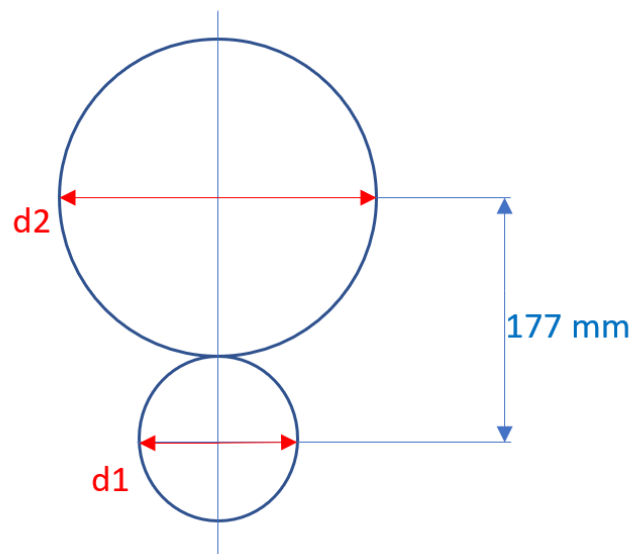
Fig. 44 Illustration of Final Drive and the Pinion Pitch Diameters

The number of teeth of pinion and the final drive is calculated from the equation 21, and from the equation the results are obtained as ;

$$z_{pinion} = 53.72 \ \approx 54 \ \text{teeth}$$

$$z_{FD} = 24.419 \ \approx 25 \ \text{teeth}$$

Tab. 9   Results of the Planetary Gearset Design Calculations

Planetary Gearset	Acronyms	-
Total Ratio	i_{TOTAL}	5.673
PGS Ratio	i_{PGS} or $i_{sun\,.carrier}^{ring}$	2.578
Basic Ratio	$i_{sun.ring}^{carrier}$	-1.579
Module	m	1.5 mm
Sun Diameter	P_{SUN}	110 mm
Ring Diameter	P_{RING}	173.685 mm
Planet Diameter	P_{PLANET}	31.8425 mm
Tooth Height	h	3.75 mm
Dedendum	h_F	2.25 mm
Root diameter	d_{ROOT}	27.342 mm
Bearing Diameter	$d_{BEARING}$	18.3425 mm
Roller Size	d_{ROLLER}	3 mm
Dia.Under Bearing	d_{UB}	12.342 mm
Helix Angle	ß	20 °
# of teeth of Sun	z_{SUN}	69 teeth
# of teeth of Planet	z_{PLANET}	20 teeth
# of teeth of Ring	z_{RING}	108 teeth
Final Drive	-	-
Final Drive Ratio	i_{FD}	2.2
Electric Motor Radius	r_{MOTOR}	147 mm
Output shaft diameter	d_{OUTPUT}	30 mm
Clearance	clearance	15 mm
Center Distance	CD	177 mm
Final Drive Pitch Diameter	D_2	243.375 mm
Pinion Pitch Diameter	D_1	110.625 mm
# of teeth of Final Drive	z_{FD}	54 teeth
# of teeth of Pinion	z_{PINION}	25 teeth
Module	m_{FD}	4 mm
Helix angle	$ß_F$	28 °

5.3. Strength Calculations of The Gear Meshes with SABR Software

After the geometrical properties are determined, the strength calculations of the planetary gearset prepared with the help of Ricardo Company's structural analysis software SABR/GEAR.

The software SABR/GEAR, checks the design's suitability to production, analyse and optimise the systems in terms of durability, friction, vibrations according to the standard: "ISO-6336 : Calculation of Load Capacity of Spur and Helical Gears''.

The software SABR/GEAR provides the user to enter the basic geometry parameters, stress calculation inputs (such as microgeometry, facewidth, the axial location of the mesh etc.),the amount of backlash and a duty cycle, and according to the material which is used, it calculates the amount of stress (bending and contact stress of the gears, the amount of damage) and possible lifetime when it is applied to the gears, shaft and the bearings.

This software is applied first for the gear geometry, to the meshes between Sun/Planet, Planet/Ring and the Final Drive/Pinion.

5.3.1. Duty Cycle

The duty cycle of a transmission is defined as the speed and load conditions over which the transmission operates in a specific application. Understanding the speed and load history and preparing the design according to the application have crucial importance for transmission designers. The designers' expectation is for the transmission is completeing their duty cycle in 250.000 kilometers accordingly. Ricardo uses „Speed-Up Duty Cycle" which is also called Factored Duty Cycle. The engineers in Ricardo Prague are provided with the measured Duty Cycles in the conditions of Torques +Shafts speeds + Durations (Times). In other words, for each component in the transmission, which has a crucial importance is tested in maximum nominal and peak torques for a specific time. For the Planetary Gearset for Electric Vehicle application, the suggested duty cycle strategy is by Ricardo is described in Table 10.

Tab. 10 Ricardo's Suggested Duty Cycle Strategy for the Current Gearbox

BENDING STRESS	21h	Maximum Nominal Torque
	0.05h	Maximum Peak Torque
CONTACT STRESS	41h	Maximum Nominal Torque
	0.1h	Maximum Peak Torque
BEARING STRESS	244h	Maximum Nominal Torque
	0.0001h	Maximum Peak Torque

Fig. 45   Screenshot of Ricardo Software SABR

5.3.2.   Material Properties of the Components

The aimed material for production is the case hardened steel 27MnCr5.

This alloyed heat treatable steel is used for parts requiring high surface wear resistance but having a soft core which absorb stresses without cracking. 27MnCr5 is widely used in automotive industry for many applications, such as in the production of axles, shafts, pinions and gears.

To be able to know the strength of the material during application, it is required to know the material properties. The properties are described in Table 11.

Tab. 11 Material Properties of 27MnCr5

Material Name	27MnCr5
Young's Modulus	206 GPa
Density	7800 kg/m^3
Bending Yield Stress	1150 MPa
Contact Yield Stress	2400 MPa
Bending Stress Limit	558 MPa
Contact Stress Limit	1190 MPa
Surface Hardness HRC	61 RC
Brinell / Vickers Hardness	670/720
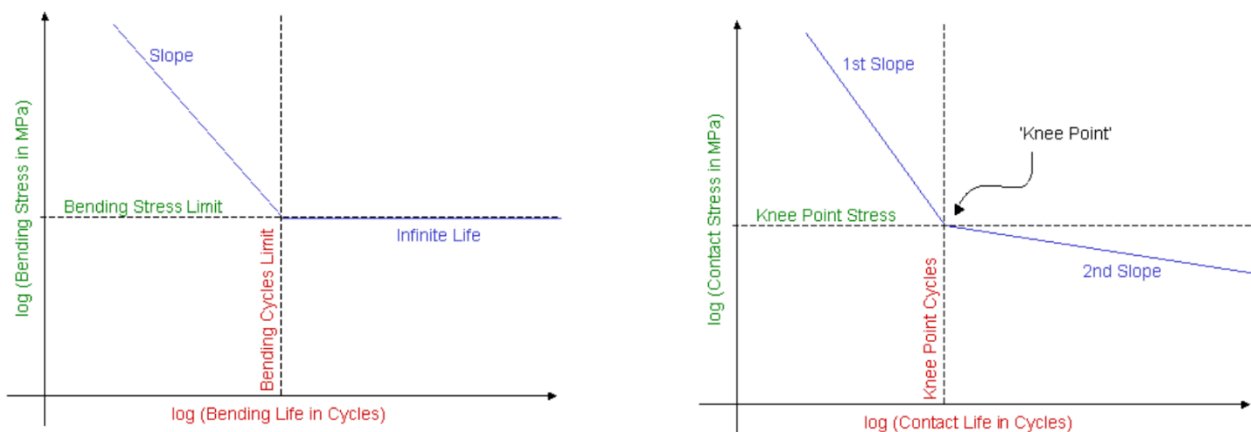
Fig. 46   S-N Curve of 27MnCr5 for Bending Stress and Contact Stress [34]

The aim is being in the zone which is lower than the stress limit to reach the infinite life. Since the material properties which is going to be used is known, the results of bending and the contact stresses which are affecting the gears are understandable. Each component mesh in the gearbox design is simulated with the recommended duty cycle strategy for this specific case by Ricardo; Bending Stress and Contact Stresses determined in the Maximum Nominal Torque and Maximum Peak Torque.

5.3.3. Sun/Planet Mesh Results

First application is done for the Sun/Planet arrangement.

The basic geometry parameters entered the software and the microgeometry features (such as max/min chamfer tip.) accepted as constant in all variants.

Fig. 47   Basic Geometry Features in Sun/Planet Configuration in SABR/GEAR

Fig. 48   The Inputs for Microgeometry and Backlash in SABR/GEAR

With the specific data entered for the Maximum Nominal and Maximum Peak Torque, the results are obtained for the four cases;

Tab. 12 Entered Duty Cycle Properties For Sun/Planet Mesh

Peak Torque on Sun Gear		370 Nm
Nominal Torque on Sun Gear		200 Nm
Sun Shaft Speed		8000 rpm
Carrier Speed		3103 rpm
BENDING	Nominal Torque Duration	21 hours
	Maximum Torque Duration	0.05 hours
CONTACT	Nominal Torque Duration	41 hours
	Maximum Torque Duration	0.1 hours

Fig. 49 Results for Bending in Nominal(Above) and Maximum Torque(Below)

	Sun			Planet	
ISO Nom. Bending Stress σFO	69.971	MPa		85.290	MPa
ISO Actual Bending Stress σF	267.612	MPa		326.202	MPa
ISO Permissible Stress σFP	837.801	MPa		579.472	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	3.131			1.776	
ISO Actual Contact Stress σH	1033.732	MPa		1033.732	MPa
ISO Permissible Stress σHP	1407.558	MPa		1389.848	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	1.362			1.344	

	Sun			Planet	
ISO Nom. Bending Stress σFO	129.446	MPa		157.787	MPa
ISO Actual Bending Stress σF	359.316	MPa		437.983	MPa
ISO Permissible Stress σFP	1335.180	MPa		844.277	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	3.716			1.928	
ISO Actual Contact Stress σH	1197.825	MPa		1197.825	MPa
ISO Permissible Stress σHP	2400.000	MPa		2397.048	MPa
ISO Contact Life	4186.318	hour		3640.277	hour
ISO Contact Damage	0.002	%		0.003	%
ISO Safety Factor SH	2.004			2.001	
ISO 6336 Edition		ISO 6336:2006 Corr. 2… ▼			

Fig. 50   Results for Contact in Nominal(Above) and Maximum Torque(Below)

•When the results obtained for the Bending Stress, the results are applicable and lower for the material's stress bending limit.

• The software shows that both Sun and Planet gears can be used theoretically in infinite lifetime, without a bending damage.

•When the results obtained for the Contact Stress, the results are applicable and low for the material's fatigue limit.

•The software shows that both Sun and Planet gears can be used theoretically in infinite lifetime in   nominal torque, but when the maximum peak torque applied, the contact life becomes determinable and micro damages occur in both Sun and Planet Gears.

5.3.4. Planet/Annulus(Ring) Mesh Results

Second application is done for Planet/Ring Mesh arrangement.

Tab. 13 Entered Duty Cycle Properties For Planet/Ring Mesh

Peak Torque on Ring		1276.5 Nm
Nominal Torque on Ring		690 Nm
Carrier Speed		3103 rpm
Ring(Annulus)Speed		0 rpm (stationary)
BENDING	Nominal Torque Duration	21 hours
	Maximum Torque Duration	0.05 hours
CONTACT	Nominal Torque Duration	41 hours
	Maximum Torque Duration	0.1 hours

With the specific data entered for the Maximum Nominal and Maximum Peak Torque, the results are obtained for the four cases;

Fig. 51   Results for Bending Stress in Nominal Torque

	Planet			Annulus	
ISO Nom. Bending Stress σFO	321.222	MPa		276.966	MPa
ISO Actual Bending Stress σF	637.301	MPa		549.498	MPa
ISO Permissible Stress σFP	948.500	MPa		1762.058	MPa
ISO Bending Life	2.081	hour		INFINITE	hour
ISO Bending Damage	2.403	%		0.000	%
ISO Safety Factor SF	1.488			3.207	
ISO Actual Contact Stress σH	1157.078	MPa		1157.078	MPa
ISO Permissible Stress σHP	2400.000	MPa		2400.000	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	2.074			2.074	
ISO 6336 Edition			ISO 6336:2006 Corr. 2... ▾		

Fig. 52   Results for Bending Stress in Maximum Torque

	Planet			Annulus	
ISO Nom. Bending Stress σFO	178.619	MPa		148.176	MPa
ISO Actual Bending Stress σF	477.365	MPa		396.003	MPa
ISO Permissible Stress σFP	580.328	MPa		858.580	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	1.216			2.168	
ISO Actual Contact Stress σH	970.291	MPa		970.291	MPa
ISO Permissible Stress σHP	1410.614	MPa		1485.723	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	1.454			1.531	

	Planet			Annulus	
ISO Nom. Bending Stress σFO	330.446	MPa		274.125	MPa
ISO Actual Bending Stress σF	651.693	MPa		540.618	MPa
ISO Permissible Stress σFP	858.932	MPa		1601.882	MPa
ISO Bending Life	1.582	hour		INFINITE	hour
ISO Bending Damage	6.323	%		0.000	%
ISO Safety Factor SF	1.318			2.963	
ISO Actual Contact Stress σH	1133.700	MPa		1133.700	MPa
ISO Permissible Stress σHP	2398.865	MPa		2400.000	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	2.116			2.117	
ISO 6336 Edition			ISO 6336:2006 Corr. 2... ▾		

Fig. 53   Results for Contact in Nominal(Above) and Maximum Torque(Below)

•When the results obtained for the Bending Stress, the results are applicable and lower for the material's stress bending limit in Nominal Torque. When the peak torque is applied, the bending stress limit is achieved for annulus gear, and ring is damaged 2.4 percentage.

• The software shows that both Ring and Planet gears can be used theoretically in infinite lifetime, without a bending damage in nominal torque, but in peak torque, ring has a finite life.

•The software shows that both Ring and Planet gears can be used theoretically in infinite lifetime in nominal torque, the contact stress limit is higher than the results.

5.3.5. Final Drive/Pinion Mesh Results

Third application is done for the Final Drive/Pinion mesh.

Tab. 14 Entered Duty Cycle Properties for Final Drive / Pinion Mesh

Peak Torque on Pinion		953 Nm
Nominal Torque on Pinion		515.6 Nm
Pinion Speed		3103 rpm
BENDING	Nominal Torque Duration	21 hours
	Maximum Torque Duration	0.05 hours
CONTACT	Nominal Torque Duration	41 hours
	Maximum Torque Duration	0.1 hours

With the specific data entered for the Maximum Nominal and Maximum Peak Torque, the results are obtained for the four cases;

	Pinion			FD	
ISO Nom. Bending Stress σFO	83.361	MPa		86.032	MPa
ISO Actual Bending Stress σF	169.886	MPa		175.330	MPa
ISO Permissible Stress σFP	871.838	MPa		924.496	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	5.132			5.273	
ISO Actual Contact Stress σH	829.136	MPa		829.136	MPa
ISO Permissible Stress σHP	1739.389	MPa		1861.099	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	2.098			2.245	
ISO 6336 Edition			ISO 6336:2006 Corr. 2... ▼		

	Pinion			FD	
ISO Nom. Bending Stress σFO	154.079	MPa		159.016	MPa
ISO Actual Bending Stress σF	251.081	MPa		259.127	MPa
ISO Permissible Stress σFP	1677.448	MPa		1772.198	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	6.681			6.839	
ISO Actual Contact Stress σH	1007.985	MPa		1007.985	MPa
ISO Permissible Stress σHP	2400.000	MPa		2400.000	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	2.381			2.381	
ISO 6336 Edition			ISO 6336:2006 Corr. 2... ▼		

Fig. 54   Results for Bending in Nominal(Above) and Maximum Torque(Below)

	Pinion			FD	
ISO Nom. Bending Stress σFO	83.361	MPa		86.032	MPa
ISO Actual Bending Stress σF	169.886	MPa		175.330	MPa
ISO Permissible Stress σFP	860.229	MPa		872.627	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	5.064			4.977	
ISO Actual Contact Stress σH	829.136	MPa		829.136	MPa
ISO Permissible Stress σHP	1640.132	MPa		1754.896	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	1.978			2.117	
ISO 6336 Edition			ISO 6336:2006 Corr. 2… ▾		

Fig. 55   Results for Contact Stress in Nominal Torque

	Pinion			FD	
ISO Nom. Bending Stress σFO	154.079	MPa		159.016	MPa
ISO Actual Bending Stress σF	251.081	MPa		259.127	MPa
ISO Permissible Stress σFP	1551.732	MPa		1644.681	MPa
ISO Bending Life	INFINITE	hour		INFINITE	hour
ISO Bending Damage	0.000	%		0.000	%
ISO Safety Factor SF	6.180			6.347	
ISO Actual Contact Stress σH	1007.985	MPa		1007.985	MPa
ISO Permissible Stress σHP	2400.000	MPa		2400.000	MPa
ISO Contact Life	INFINITE	hour		INFINITE	hour
ISO Contact Damage	0.000	%		0.000	%
ISO Safety Factor SH	2.381			2.381	
ISO 6336 Edition			ISO 6336:2006 Corr. 2… ▾		

Fig. 56   Results For Contact Stress in Maximum Torque

•When the results obtained for the Bending Stress, the results are applicable and lower for the material's stress bending limit in Nominal Torque and Peak Torque.

• The software shows that both Final Drive and Pinion gears can be used theoretically in infinite lifetime, without a bending damage in nominal torque and peak torque.

•The software shows that both Final Drive and Pinion gears can be used theoretically in infinite lifetime in nominal torque, the contact stress limit is higher than the results.

5.4. Strength Calculations of the Components with SABR Software

The strength calculations of the meshes for Bending and Contact Stresses are done with SABR software. In this section with SABR software, strength calculations of the assembly will be done.

In this chapter the strength results of the shafts and bearings will be discussed.
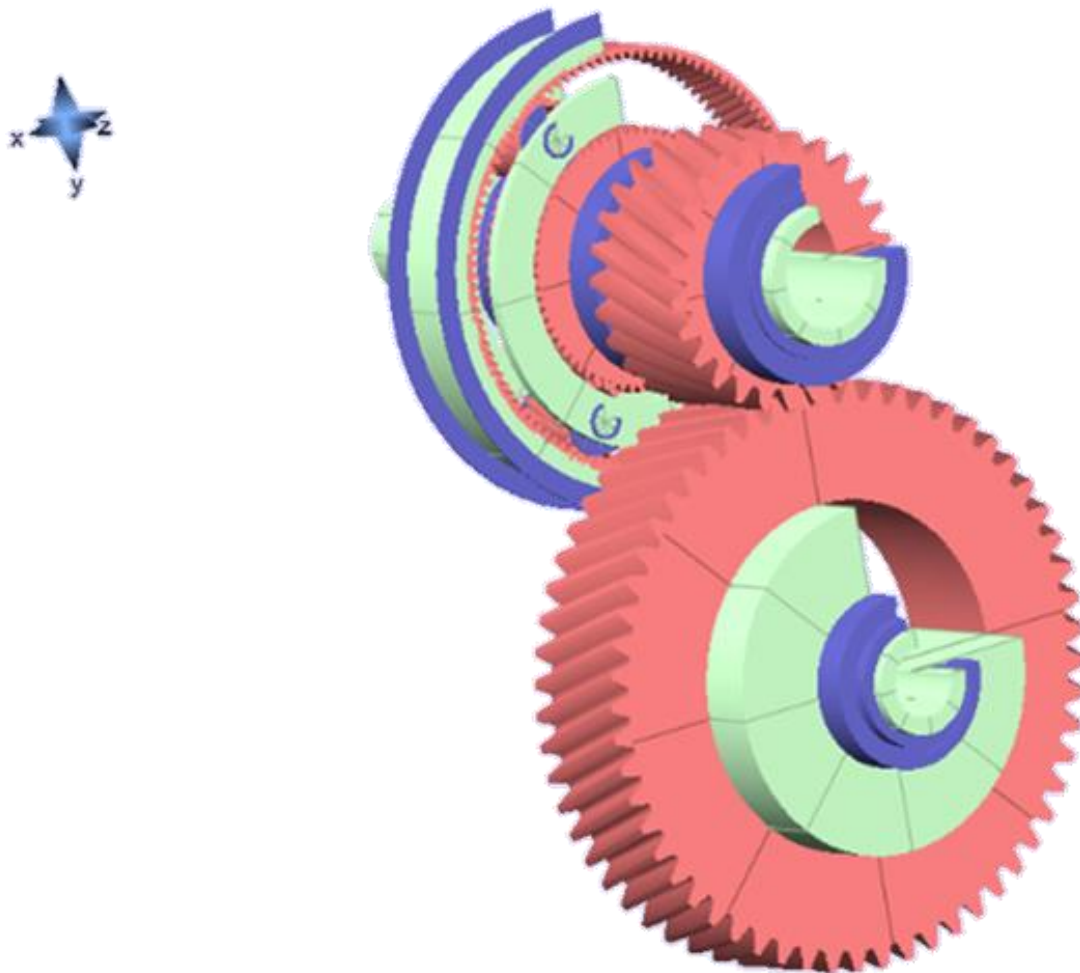
Fig. 57 Screenshot from the SABR Software

5.4.1. Bearing Selection

In the assembly used bearings are given in the Table 15.

Tab. 15 The Bearings Which are Used in The Assembly

Bearing Name In the SW	Bearing Catalog ID Name	Type of Bearing
BSun1	SKF-6209 [35]	Deep Groove Ball Bearing
BSun2	SKF-K455017 [36]	Needle Roller and Cage Bearing
Carrier Bearing1	SKF-6013Z [37]	Deep Groove Ball Bearing
Carrier Bearing2	SKF-6012 [38]	Deep Groove Ball Bearing
Bearing Differential1	SKF 32009 [39]	Tapered Roller Bearing
Bearing Differential2	SKF 32009 [39]	Tapered Roller Bearing
Planetary Gearset psa1	SKF K121812TN [40]	Needle Roller and Cage Bearing
Planetary Gearset Psa2	SKF K121812TN [40]	Needle Roller and Cage Bearing
Planetary Gearset psa3	SKF K121812TN [40]	Needle Roller and Cage Bearing
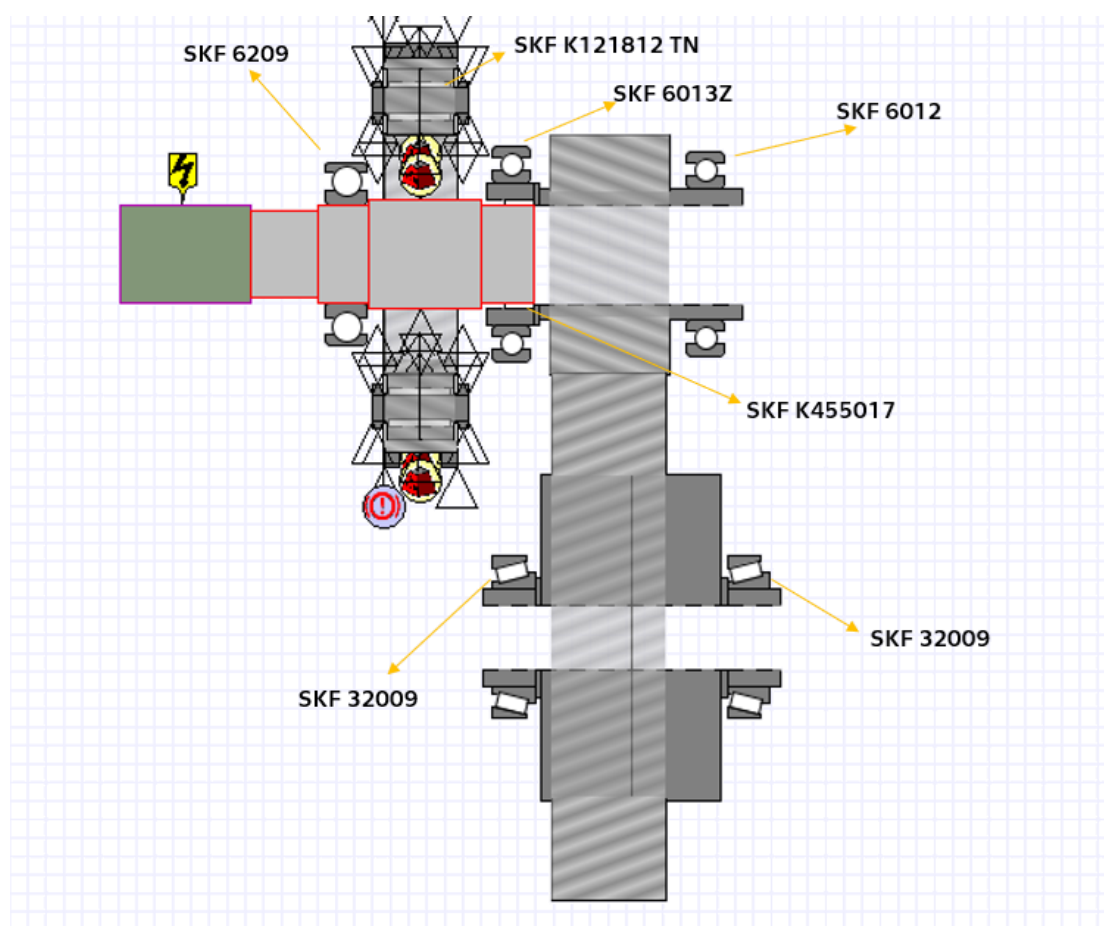
Fig. 58 Locations of The Bearings in the Assembly

5.4.2. Bearing Life and Temperature

Tab. 16 The Applied Stress/ Amount of Damage on Bearings

Bearing Name In the SW	Type	Life (hrs.)	Detailed Damage	Max. Stress (GPa)
BSun1	DG Ball	7083.6	3.4%	1.844
BSun2	CE Needle	INFINITE	0.0%	0.434
Carrier Bearing1	DG Ball	3223.4	7.6%	2.499
Carrier Bearing2	DG Ball	413.4	59%	3.221
Bearing Differential1	Tapered	7698.6	3.2%	2.628
Bearing Differential2	Tapered	68375.2	0.4%	2.041
Planetary Gearset psa1	CE Needle	673.3	36.2%	2.500
Planetary Gearset Psa2	CE Needle	648	37.7%	2.536
Planetary Gearset psa3	CE Needle	611.7	39.9%	2.539

As it can be seen from the Table 16, although the bearings are exposure to high stress levels, they will not have malfunction and can continue their operation.(see **Fig 57**) .

Fig. 59 Maximum Bearing Temperature Simulation

Increase of temperature on bearings is an unwanted situation and by the SABR software, the maximum temperature of the bearings could reach simulated accordingly. The bearing temperatures above 110°C is not desired that, for the bearings which are achieving the amount, the flow rate of the of the coolant oil is increased.

5.4.3. Deflections on Shaft

After the simulation is done ,the results are obtained for the deflection of the Shafts which can be seen in the Table 17.

Tab. 17 Maximum Radial Deflection By Load Case

Load Case/Shaft (μm)	Sun	Differential	Carrier	Pin1	Pin 2	Pin3
Nominal Torque	0.0	0.03	0.03	0.02	0.02	0.02
Peak Torque	0.0	0.05	0.05	0.03	0.03	0.03

By this part of the analysis, the areas which may have high stress can be detected. A deflection of 0.8 micron per mm deflection is acceptable by the Ricardo company that, all of the deflection values are within the limits.

For the nominal torque case we are concerned about the pinion and the wheel of the final drive. Therefore we checked the detailed figures for the center distance displacement as well as load of action misalignment (LOAMA) and the results are shown in the Table 18 and the pictures 57. As a result displacement is simulated as 0.069 microns and the slope is 0.259 mrad.

The deformations in the picture are 100 times magnified.

Fig. 60 LOMA and CD Displacement in FD and Pinion Mesh

The displacements of the differential and the carrier shaft are described on Figure 58 and Figure 59.

Results: RESULTS_VALID **Solver:** solver converged after 123 iterations in 12ms; balance=0.2 error=0.00%

Radial Deflections

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
displacement (µm)	129.6	118.2	117.2	116.6	109.4	107.3	103.5	87.6	83.8	92.3	93.8	97.3	97.8	98.5	111.0

Deflections in XZ Plane

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
displacement (µm)	-76.5	-77.1	-77.1	-77.4	-79.9	-80.3	-80.7	-82.2	-83.7	-84.9	-84.9	-84.6	-84.6	-84.6	-85.3

Slopes in XZ Plane (about y axis)

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
slope (mrad)	-0.053	-0.053	-0.138	-0.224	-0.186	-0.105	-0.061	-0.056	-0.051	-0.027	0.020	0.047	0.004	-0.042	-0.042

Deflections in YZ Plane

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
displacement (µm)	104.6	89.6	88.3	87.2	74.7	71.2	64.8	30.3	-4.2	-36.1	-39.8	-48.0	-49.2	-50.4	-71.1

Slopes in YZ Plane (about x axis)

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
slope (mrad)	1.282	1.282	1.197	1.110	1.149	1.230	1.274	1.278	1.277	1.260	1.225	1.205	1.237	1.270	1.270

Axial Deflections (z axis)

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
displacement (µm)	347.4	347.4	347.4	347.4	347.3	347.3	347.2	347.2	347.2	347.3	347.3	347.3	347.3	347.3	347.3

All Deflections

location (mm)	170.0	181.8	182.8	183.8	195.0	198.0	203.0	230.0	257.0	282.0	285.0	291.8	292.8	293.8	310.0
Deflections in XZ Plane (um)	-76.5	-77.1	-77.1	-77.4	-79.9	-80.3	-80.7	-82.2	-83.7	-84.9	-84.9	-84.6	-84.6	-84.6	-85.3
Slopes in XZ Plane (about y axis) (mrad)	-0.053	-0.053	-0.138	-0.224	-0.186	-0.105	-0.061	-0.056	-0.051	-0.027	0.020	0.047	0.004	-0.042	-0.042
Deflections in YZ Plane (um)	104.6	89.6	88.3	87.2	74.7	71.2	64.8	30.3	-4.2	-36.1	-39.8	-48.0	-49.2	-50.4	-71.1
Slopes in YZ Plane (about x axis) (mrad)	1.282	1.282	1.197	1.110	1.149	1.230	1.274	1.278	1.277	1.260	1.225	1.205	1.237	1.270	1.270
Axial Deflections (z axis) (um)	347.4	347.4	347.4	347.4	347.3	347.3	347.2	347.2	347.2	347.3	347.3	347.3	347.3	347.3	347.3

Fig. 61 Slopes and Deflections of the Differential with Respect to Location

Results: RESULTS_VALID Solver: solver converged after 7784 iterations in 2424ms; balance=0.0 error=0.00%

Radial Deflections

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
displacement (µm)	14.4	14.1	11.7	11.8	12.2	13.1	13.4	15.1	15.7	16.7	21.5	25.9	26.8	27.0	30.3

Deflections in XZ Plane

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
displacement (µm)	6.5	6.7	10.8	11.0	12.0	13.1	13.4	15.0	15.4	16.3	19.3	21.3	20.9	20.9	22.4

Slopes in XZ Plane (about y axis)

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
slope (mrad)	0.107	0.107	0.107	0.107	0.107	0.107	0.173	0.165	0.167	0.142	0.090	0.073	-0.030	0.090	0.090

Deflections in YZ Plane

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
displacement (µm)	12.8	12.4	4.6	4.2	2.3	0.1	-0.3	-2.1	-2.7	-3.7	-9.5	-14.7	-16.8	-17.1	-20.4

Slopes in YZ Plane (about x axis)

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
slope (mrad)	0.206	0.206	0.206	0.206	0.206	0.206	0.196	0.198	0.197	0.201	0.201	0.194	0.116	0.207	0.207

Axial Deflections (z axis)

location (mm)	121.0	123.0	161.0	163.0	172.0	183.0	185.0	194.0	197.0	202.0	230.0	257.0	274.0	276.0	292.0
displacement (µm)	120.6	120.6	120.6	120.6	120.6	120.6	120.6	120.7	120.7	120.7	120.7	120.7	120.5	120.5	120.5

All Deflections

location (mm)	121.0	122.0	123.0	161.0	162.0	163.0	172.0	183.0	184.0	185.0	194.0	197.0	202.0	203.0	230.0	257.0	258.0	274.0	275.0	276.0	292.0
Deflections in XZ Plane (um)	6.5	6.6	6.7	10.8	10.9	11.0	12.0	13.1	13.2	13.4	15.0	15.4	16.3	16.4	19.3	21.3	21.4	20.9	20.8	20.9	22.4
Slopes in XZ Plane (about y axis) (mrad)	0.107	0.107	0.107	0.107	0.107	0.107	0.107	0.107	0.141	0.173	0.165	0.167	0.142	0.106	0.090	0.073	0.021	-0.030	0.031	0.090	0.090
Deflections in YZ Plane (um)	12.8	12.6	12.4	4.6	4.4	4.2	2.3	0.1	-0.1	-0.3	-2.1	-2.7	-3.7	-3.9	-9.5	-14.7	-14.9	-16.8	-16.9	-17.1	-20.4
Slopes in YZ Plane (about x axis) (mrad)	0.206	0.206	0.206	0.206	0.206	0.206	0.206	0.206	0.201	0.196	0.198	0.197	0.201	0.207	0.201	0.194	0.155	0.116	0.162	0.207	0.207
Axial Deflections (z axis) (um)	120.6	120.6	120.6	120.6	120.6	120.6	120.6	120.6	120.6	120.6	120.7	120.7	120.7	120.7	120.7	120.7	120.7	120.5	120.5	120.5	120.5

Fig. 62 Slopes and Deflections of the Carrier with Respect to Location

In this case the differential is a very stiff part as well as the carrier shaft, therefore the most deformation is determined by the clearance of the supporting bearings. The combined deformation are still very small , resulting 0.069 mm center distance difference and 0.259 mrad load of action misalignment (LOAMA).

5.4.4. Stresses Applied on Shaft
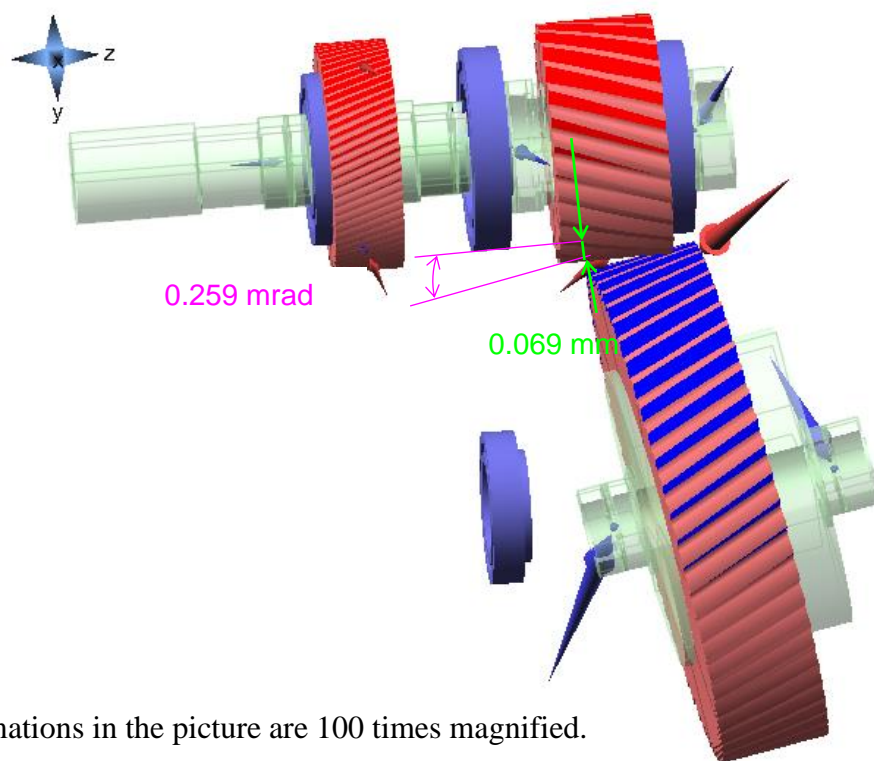
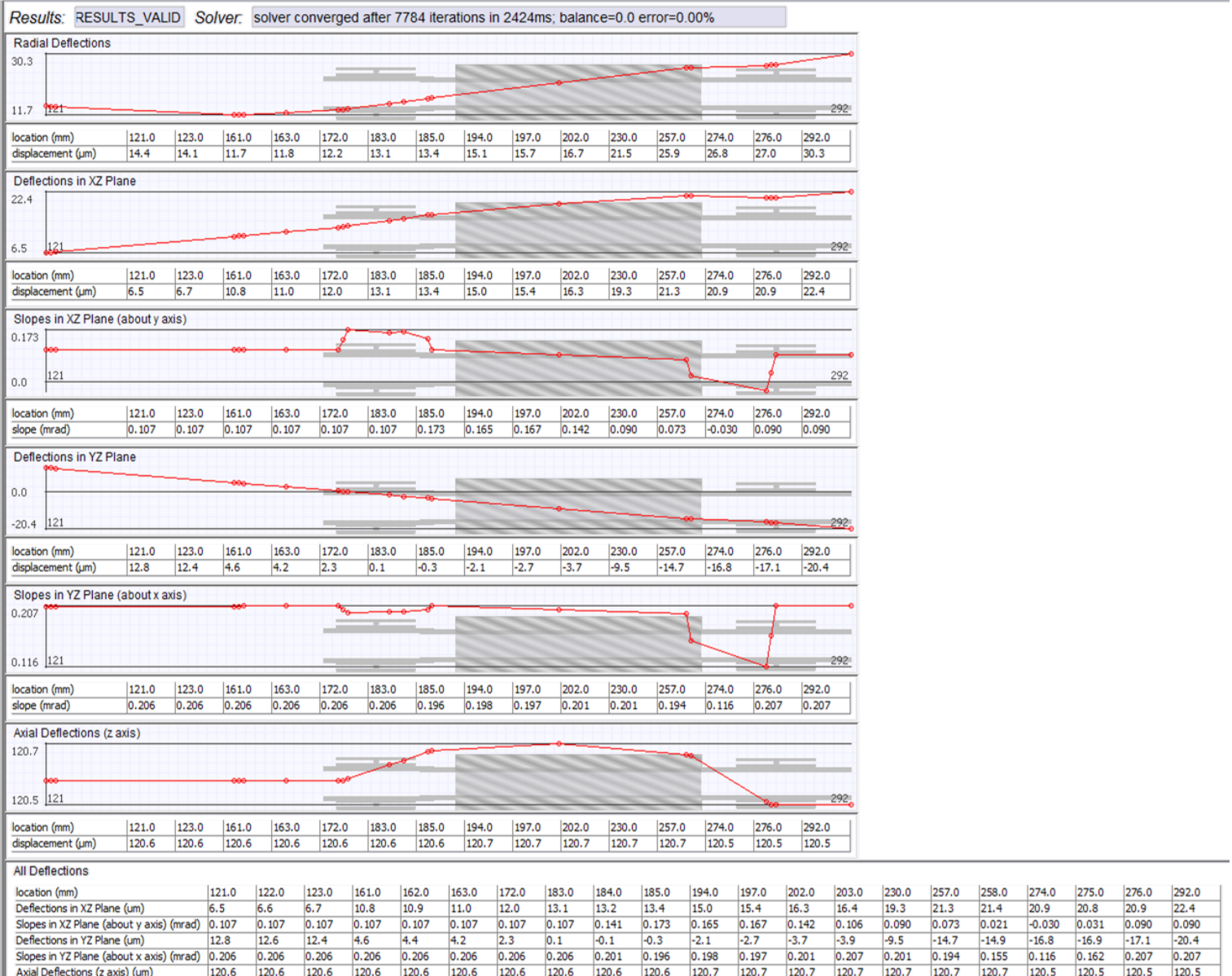After the simulation is done ,the results are obtained for the deflection of the Shafts which can be seen in the Table 18.

Tab. 18 The von Misses Stresses Applied on Shafts

Load Case/Shaft	Sun	Differential	Carrier	Planetary Gearset
Nominal Torque	28 MPa	30 MPa	44 MPa	119 MPa
Peak Torque	52 MPa	56 MPa	81 MPa	210 MPa

By this part of the analysis, the areas which may have high stress can be detected. The maximum von Misses stress is acceptable by the Ricardo is 800 MPa that, all of the stress values are within the limits.

After the simulations checked from the software SABR/GEAR, it can be said that the gearbox system is hypothetically ready for production. However, the next iterations would save a lot of weight, cost and reduce the risk of potential failures.
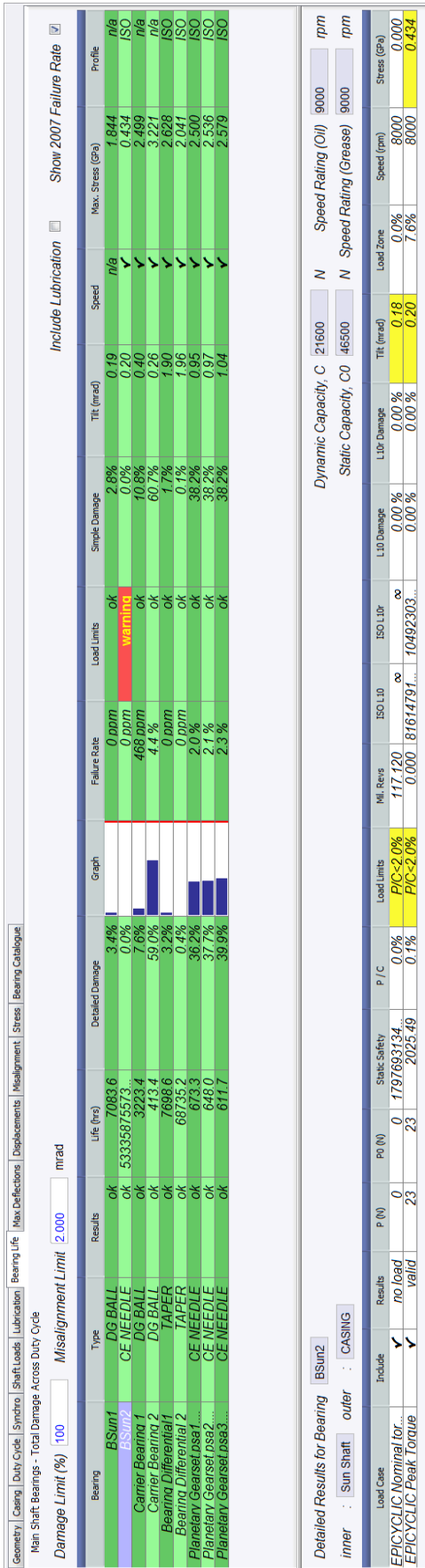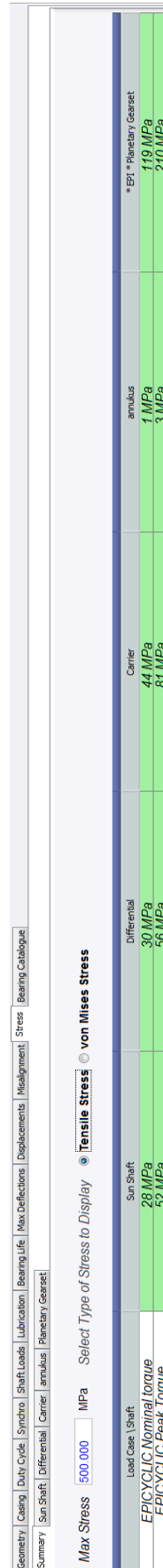
Fig. 64 Life Conditions of Bearings in SABR Software

Fig. 63 Results of Stresses Applied on Shafts in SABR Software

6. Conclusion

The task of the thesis was a design a planetary gearbox system for a city car which the range of the electric vehicle is adequate for daily urban activities.

Firstly, the research amongst battery electric vehicles dominating the market was made and their powertrains were analysed to find inspiration and observe the current state of art powertrains. Since the topic of the thesis is designing a planetary gearset for an electric vehicle, the designs with planetary gearsets become the main inspiration for the powertrain which is going to be designed. Also, to be able to observe more complex designs, the patent research was made, from there it was seen that how the planetary reductions are done.

After the battery electric vehicles and the patent research, vehicle parameters are selected, for the city car. For the parameter selection, the closest production vehicle was Chevrolet Bolt EV and the specifications of Chevrolet Bolt EV become the main inspiration vehicle for the design.

To be able to find the total ratio of the powertrain system, longitudinal dynamics calculations were made, the resistance forces have been found. Total gear reduction ratio (i) is calculated in the condition of the maximum speed that, in the vehicle specifications, maximum speed was given approximately 160 km/h, and the total ratio was found as i=5.6737.

Before the design process started, it was asked in the thesis to compare several powertrain systems in efficiency, mass and dimension point of view. Two of the designs were planetary gearsets and the third variant was a conventional layshaft transmission. From there the 'Variant 1' is selected as the powertrain to be designed. There are three motivations behind this decision that ; the ground clearance issue, wear of the homokinetic joints and modification of the rotor in the electric motor.

After the variant which is going to be designed is determined, the total ratio is divided between the Planetary Gearset (i=2.578) and the Final Drive (ifd=2.2). Pitch diameter, number of teeth and dimension calculations are done with the help of mechanical design tables.

The assembly condition of the planetary gearset is checked by the formula and the answer have been found integer, which is a proof of assembly condition was provided.

When the required calculation of the dimensions, the material has been selected for the production. The aimed material for production was the case hardened steel 27MnCr5. The material parameters have been entered to the softwares SABR and GEAR.

Company Ricardo Prague, have recommended a duty cycle for the specific powertrain, which was mentioned under the Chapter 5.3.1, Table 10. The duty cycle have applied in the GEAR Software simulation for determination of Bending and Contact Stress between the meshes, Sun/Planet, Planet/Ring and Pinion/Final Drive.

For the assembly, the bearings are selected on the shaft, which are described under Chapter 5.4.1, Table 15. With the SABR Software, the Bearing Life and Temperatures, Stresses Applied on shafts and Deflections on shafts have been determined in the simulation and the results were obtained.

In conclusion, all of the simultaion results are reasonable and in the limits. Those results shows that the production of the gearset system is possible.

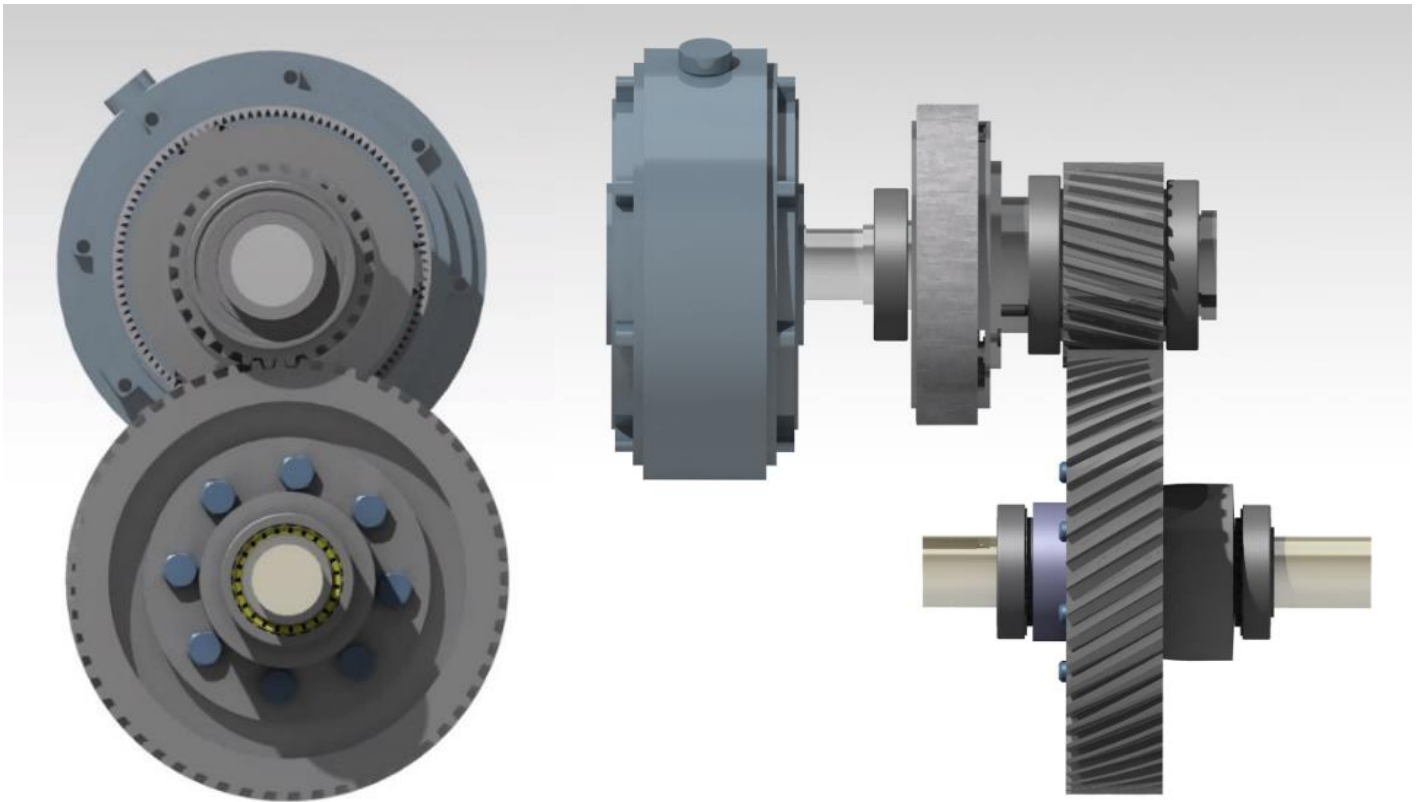*Fig. 65   The Planetary Gearbox For Electric Vehicle in CATIA V5*

Fig. 66   Front and Side View Of the Powertrain Design

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Attachments

1. CAD Model (2D Autocad, 3D CATIA)

2. Excel Calculations

3. Specifications of Bearings in the Assembly

4. SABR/GEAR Software Solutions

5. Electronic Version of the Thesis