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Efficiency estimation of one- and two-speed gear transmissions for the electric powertrain

Supervisor
Prof. Nicola Amati
Co-supervisor
Prof. Sanjarbek Ruzimov

Candidate
Monica Santapaola

A Rosa,
ho continuato a volare
come avevi promesso

Abstract

In the last decades, the automotive industry developed towards battery electric vehicles (BEVs) and this trend is expected to grow. Hence, it is necessary to focus on the EV powertrain for an efficient transmission of the power stored in the battery pack to the wheels of the vehicle.
This thesis is organised in two parts. In the first one, an overall description of the EV powertrain, its main components, benefits and challenges is presented with specific details on the transmission's state-of-the-art. Indeed, the researches' attention is currently drawn by the possible introduction of a multi-speed gearbox in electric vehicles to enhance the economic and dynamic performances. Successively, the second section of the work centres on the development of a one-speed and a two-speed transmission models with the help of KISSsoft development tool. This software also introduces the efficiency computation of the designed layouts for several combinations of rotating speed and torque of the motor shaft, allowing the construction of the efficiency maps for the two solutions.

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Introduction

In the last decade, the automotive industry developed towards battery electric vehicle (BEVs) and this trend is expected to grow. Indeed, the increase of oil prices, the development of new technologies of lithium-ion batteries, the action of several governments providing fiscal incentives to meet more environmentally-friendly energy and reduce fossil fuel consumption and CO2 emission are pushing the research for a new kind of mobility. Among all the alternatives, electric energy is the most attractive one, enhancing the competition in BEVs development for the transport sector.
Electric vehicles submit sustainability, low maintenance, and low cost energy with respect to the fuel based counterpart. On the other side, many challenges regarding battery charging infrastructure, battery capacity, sufficient driving range, vehicle weight and cost must be overcame to make this alternative the solution of the future.

Electric energy consumption is typically measured in kWh/100 km or Wh/km and it depends on the efficiency of the powertrain components, mainly of the electric machine, the battery, and the transmission [1].
Hence, before the energy density of battery to be significantly increased, improving the overall efficiency of powertrain is a practical and cost-effective way to increase the driving range of EVs. In particular, the improvement of transmission efficiency for EVs enables less electricity energy consumption.
The efficiency of the gearbox is defined by the power losses that occur in the several elements of transmission. Mostly, the loss maps are based on measurements, while an explicit computation of gearbox losses is done rarely.
In the literature, it is possible to find several analytical approaches to estimate the power losses of the transmission, contributing to the computation of the powertrain efficiency. One of the main advantages of having a reliable analytical estimation of the powertrain performance is the possibility to select the suitable energy density of battery skipping the practical measurements.

This project aims to design two models of the single and double speed transmission to build the corresponding efficiency maps. In order to do that, the use of the software KISSsoft 03-2017 is required for the design and the power losses computation. Moreover, to validate the outcomes, an analytical model for the overall efficiency estimation of the EVs transmission is used as a reference.
Indeed, the greatest part of electric vehicles uses single-speed transmission to take advantage of electric motor's high power density and to reduce mass and complexity of the driveline. Although this solution is widely appreciated and applied by almost all car manufacturers, some of them discuss the possibility to introduce two-speed transmission to improve the vehicle's performance. It follows that the models and results obtained for the comparison are also used to give on overview on the principle aspects characterizing the two solutions in EVs powertrain, and transmission in details.

Part I

1 Introduction to the Electric vehicle

1.1 Structure of the EV

The development of the vehicle in general is based on the research of a means of transport [1]:

- autonomous in terms of energy supply;
- with a short refueling time with respect to the driving time;
- able to move an acceptable load/number of passengers;
- qualified to accelerate in an adequate timeframe and drive uphill in a variable grade path.

As a consequence, any vehicle strictly requires an on-board energy storage system with a power pick larger than the power needed for most driving conditions [1]. For the Internal Combustion Engine (ICE) vehicles, this task is accomplished by the fuel tank, while in the Electric Vehicle (EV) the same function is ascribed to the battery pack.
Although the storage system is not the only big difference between the two technologies, indeed the electrification of the drivetrain introduces the electric motor as a prime mover, the energy path is roughly comparable. As shown in the Figure 1.1, the power flow starts from the energy stored in the battery, which represents one of the main challenge for the e-mobility nowdays, goes through the inverter that regulates the current flow to send to the motor and arrives to the final major component, the transmission, which conveys the power to the wheels of the vehicle.

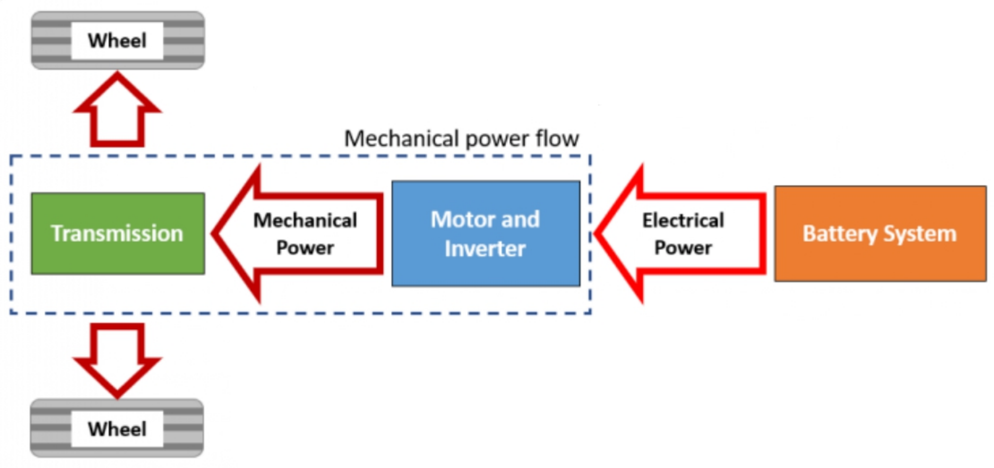

Fig. 1.1: *Electric vehicle power flow [2]*

The powertain of a vehicle is the equipment provided to the vehicle to supply the required power for driving. Briefly, the EV powertrain consists of the battery pack, the electric motor, the DC/AC converter, the Electronic Control Unit (ECU) and the transmission system. In the same way, the ICE powertrain is equipped with the fuel tank, the engine, the transmission and the driveshaft. So, what does make the EV powertrain truly different from its counterpart?
Excluding the evident divergence of the energy sources, the performance of the electric motor as the "engine" of the vehicle strongly affects the subsequent structure of the drivetrain, and the transmission in particular. Indeed, the characteristic of the motor shows profitable qualities: wide consistent power band (almost from 0 to 20 000 rpm), high torque capability at low speed and ability to rotate in both directions.
As a consequence, a multi-speed transmission is not strictly necessary as it happens for the traditional

powertrain, where the gearbox adapts the motor traction force to maintain the working point in the narrow efficient range of the ICE speed. Hence, a one-speed transmission is sufficient for an efficient performance of the vehicle and the complexity of the powertrain's architecture is significantly reduced.
In addition to the simplification of the structure thanks to the single ratio, the Vehicle Electronic Control Unit (VECU) becomes predominant in the EVs and further contributes to the minimization of the components' complexity. In terms of design, the architecture becomes more flexible with respect to the almost-fixed layout of the ICE vehicle, introducing the opportunity to work on different EVs' configurations, described in Section 1.2.4, to meet all the economic and dynamic requirements summed up in the Figure 1.2.
Lastly, the EV can achieve higher fuel consumption efficiency than conventional vehicles [3]. Indeed, the EV powertrain can rely on the great advantage of the regenerative breaking which gives the possibility to absorb the mechanical power during the breaking phase and convert it in electricity to recharge the battery.

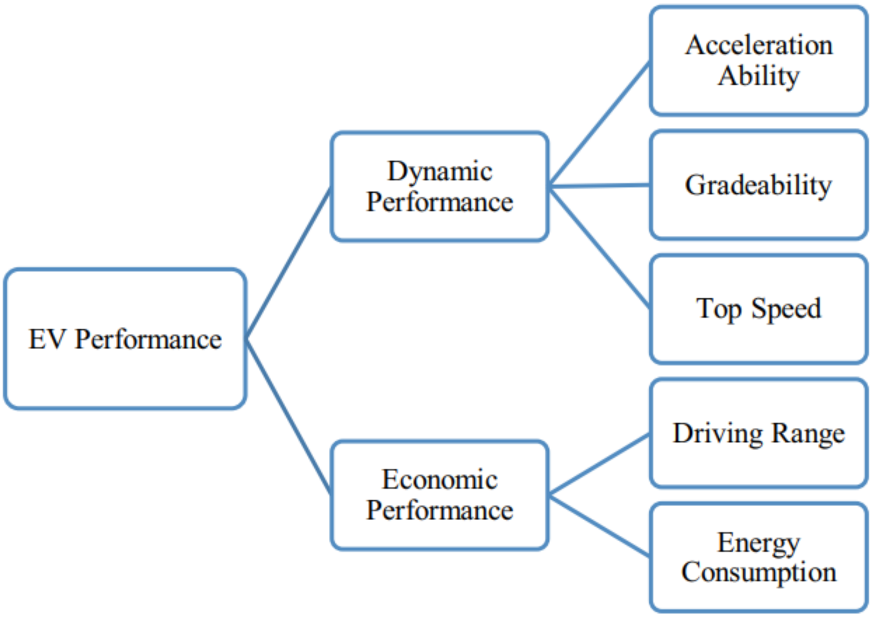

Fig. 1.2: *Criteria for EV performance evaluation [4]*

1.2 Configurations of the EV

Extending the notions exposed in the previous Section 1.1, the representation 1.1 is not univocal for all the possible configurations of an electric propulsion system. Actually, in the automotive sector, the electric motor can contribute partially or totally for the Hybrid-Electric Vehicles (HEV) or the Battery-Electric Vehicle (BEV), respectively.
Another important clarification concerns the type of energy source used in the power flow. As a matter of fact, the fundamental requirement for a vehicle to be included in the EVs' category is the presence of an electrochemical or electrostatic energy storage system [1]. Although the battery is the fundamental element in all the configurations, the charging process is variable and relies on different technologies, like the hydrogen in the Fuel Cell Hybrid Electric Vehicle (FCHEV) or the dedicated refueling hub for the Plug-In Hybrid Electric Vehicle (PHEV).

1.2.1 *BEVs*

The purely electric design is freely identified with the acronym EV or BEV and it is characterized by the simplest structure comprising only the previously cited fundamental components represented in Figure 1.3.
Its main advantage is the fully green energy source made of the battery pack which eliminates any CO_2

emission. At the same time, this element represents the greatest limit to the vehicle's performance in terms of driving range, especially. For example, a BEV can cover approximately 100–250 km on a single charge depending on the vehicle specifications, with an energy consumption rate of 15–20 kWh for 100 km [5]. To increase the driving range, a larger battery pack is needed and, as a consequence, longer charging time and higher vehicle weight would impact the performance and the cost of the car.
To overcome this issue, several solutions have been considered such us the development of an efficient Energy Management System (EMS) which exploits the regenerative breaking or the consideration of the driving strategies [3].

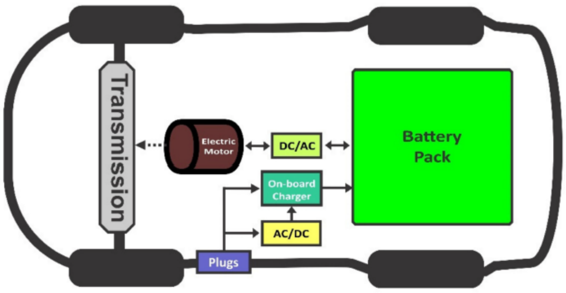

Fig. 1.3: *Configuration of a BEV [3]*

1.2.2 *HEVs*

For the exposed reasons, the EVs' market largely benefits from the distribution of the HEVs. This term is used for any solution characterized by the combination of more than one power storage system where at least one power source is an electric one. Typically, the electric motor is supported by a thermal engine that contributes with its high energy and power density [1].
The link between the two prime movers is not fixed and varies according to the requested type of collaboration. As shown in Figure 1.4, there are two mains configurations:

- Series hybrid: only the e-motor is mechanically connected to the transmission and the two engines works simultaneously;
- Parallel hybrid: the two motors are both independently connected to the driveshaft and they can also work individually.

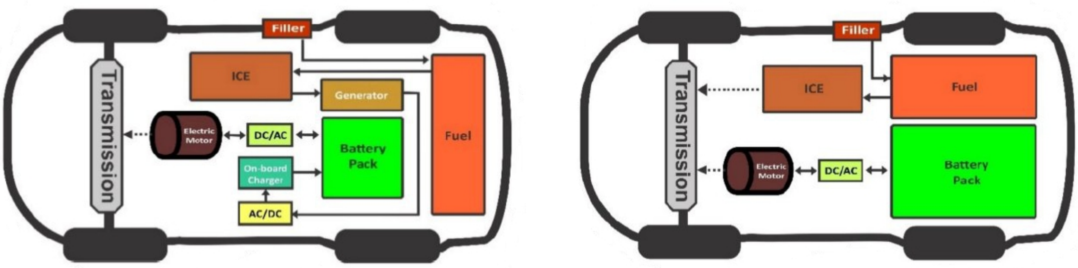

Fig. 1.4: *Configuration of a series HEV (left) and a parallel HEV (right) [3]*

Regardless of the configuration, the main benefit of the hybrid solution is the possibility to integrate to the conventional vehicle the great advantages of the electric motor such as the regenerative breaking, the reduction of the engine size, the decrease of the power losses in the engaging phase. On the other side, the HEV has more components and a heavier structure.
In the figure 1.5 it is possible to appreciate also a further classification of the hybrid vehicle (micro-, mild- and full-hybrid) according to the increasing number of functionalities in charge of the electric counterpart.

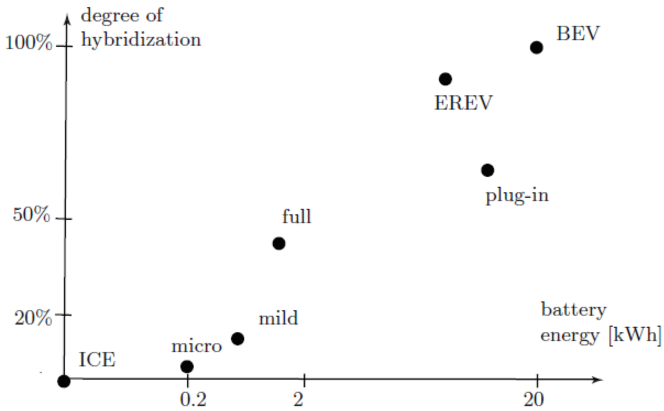

Fig. 1.5: *Degree of hybridization for HEV [1]*

1.2.3 *PHEVs*

A further growth of the driving race can be guarantee by the opportunity to directly recharge the battery pack thanks to a dedicated hub. Basically a Plug-in Hybrid Vehicle is a trade-off between the two previous solutions: the propulsion is supported by the ICE and the e-motor, anyway the car is also allowed to operate in a fully electric mode.
Since the architecture approaches the BEV solution, the issue of the battery pack and the recharging time rises again, together with the limitations on the driving range. For this purpose, the Extended-Range Electric Vehicle (EREV) was designed.

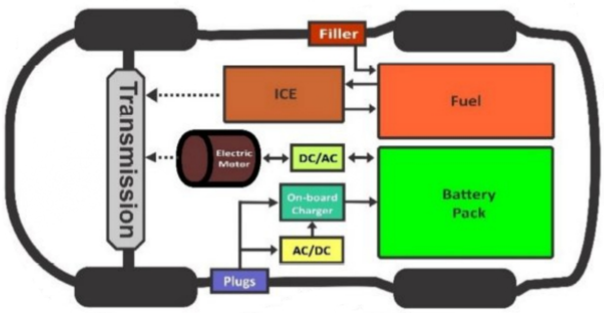

Fig. 1.6: *Configuration of a parallel PHEV [3]*

1.2.4 *FCHEVs*

The greteast result for the fuel cells-equipped EV is the zero emission technology combined with the optimal performances, short refueling time and high efficiency. Despite the promising characteristic of this solution, the FCHEV is not as attractive as the other EVs' techologies because of the high cost in the market.

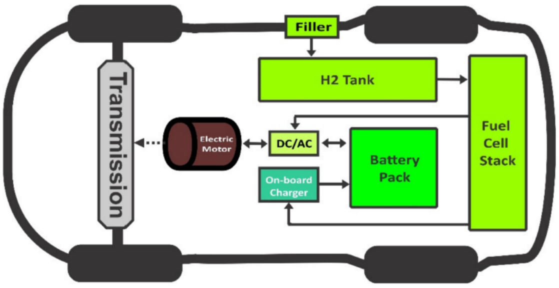

Fig. 1.7: *Configuration of a parallel FCHEV [3]*

1.3 Description the EV powertrain components

In the Section 1.1, the principal components of the electric powertrain have been introduced. For sake of simplicity, the description is circumscribed to the pure electric configuration (BEV) which includes the main elements needed in all the solutions discussed in the Section 1.2.4.
The EV powertrain is primarily composed by **the battery**, **the electric motor** and **the transmission**. The components are described below in details.

1.3.1 The battery

The battery is the device that stores chemical energy and transform it in electrical energy during the drive. Since the regenerative breaking is a function normally implemented in BEVs, the conversion process of the battery has to be reversible. This component plays a fundamental role in the EV powertrain because the high-voltage battery ensures that the performance requirements such us power density, traction torque, drive range, etc, managed by the following powertrain's components are achievable.
Nowdays, multiple technologies for the battery pack are available combining different materials for the electrodes and the medium.
To characterize the performance of a battery, the common parameters are the specific energy [Wh/kg] which is the amount of useful energy that can be stored in the battery per unit mass and it is important to define the range of the vehicle, the specific power [W/kg] similarly determines the achievable acceleration and top speed, the nominal capacity [Ah] that expresses the integral of the current that could be delivered by a full battery when completely discharged under certain reference conditions and the State of charge (SoC) which describes the capacity remaining in the battery as a percentage of its nominal capacity.
The necessary attributes for a good traction battery are high specific energy, high specific power, long calendar and cycle life, low initial and replacement costs, high reliability, wide range of operating temperatures, and high robustness [1]. In the Table I are summed up the main requirements for a suitable battery pack.

Battery Attributes	**Main Requirements**
Energy Densities	750 Wh/L and 350 Wh/kg for cells
Cost	$100/kWh for cells
Fast Charge and Power	80% of ΔSOC in 15 min
Life	15 years

TABLE I
Main Requirements for an EV's battery pack [6]

Briefly, a battery is made of two combined electrochemical cells, at least. The single cell consist of a positive cathode and a negative anode associated with the electrolyte and the compound response between these elements produces the necessary power [7]. The solution actually affirmed in the market are listed below.

- **Lead/Acid**
  The cathode is lead oxide, while the anode is a sponge metallic lead and they are both immerse in aqueous sulfuric acid which operates at ambient-temperature. In the release phase from chemical to electric energy, the products of the reaction are lead sulfate, water and energy.
  This specific category is the most established because it is robust, reliable and low cost among all. On the other hand it guarantees a specific energy only about 40 Wh/kg [7] which can be considered low if compared to the other alternatives and low cycle life.

- **Nickel -Metal Hydride**
  In this case, the cathode e is nickel oxyhydroxide, the anode is made of nickel alloys and they are drenched in potassium hydroxide. This type of battery overcomes the drawback of the previous one because it has higher energy and power density and longer life. Unfortunately, the best performance are given by rare earth–nickel alloy which makes the technology expensive. Moreover, under defined conditions, the battery can experience the so called "memory effect" when a cell is recharged without being fully discharged, which can cause the battery life to be shortened.

- **Lithium-Ion**
  This type of battery consist of a carbon-based anode, like graphite, with lithium ions as interstitional defects and a lithium/cobalt-based cathode, even if the second metal can be substituted by manganese or nickel to enhance specific parameters. While the acqueous medium is usually a lithiated liquid solution.
  The Lithium-Ion solution is nowdays the most exploited and it represented the revolution for BEV manufacturers thanks to the high specific energy that ranges from 60 to 250 Wh/kg while the power density can be as high as 2000 W/kg [7]. This is the reason why so many variations are currently available and why the researches are focused in further developments of this technology.

- **Metal Air**
  This solution slightly differs from the initial explanation of the working principle since it consist of a open system: it uses a catalytic air cathode, where the active material is oxygen from the air, in combination with an electrolyte and a lithium anode. The attraction of this battery is that it only requires one electrode, so the weight of the element is reduced which justifies the higher specif energy around 220 Wh/kg [7]. The main challenges for its development concern the recharging ability and the cyclability. Anyway, the battery packs in EVs include not only cells but also other components like busbars, thermal components, and battery management systems. Unfortunately, the adding of new elements to the hardware further reduces the overall pack-level specific energy. Thus, improving both cell design and pack efficiency is critical to increase energy densities of EV batteries.

1.3.2 The electric motor

While in the conventional vehicles the electric motor is used only for the auxiliary tasks, namely as a starter and alternator, in the BEV this component has a key role. Firstly, the machine has to transform the power stored in the battery into mechanical power to drive the wheels and, secondly, the regenerative breaking requires the ability to recuperate the mechanical power from the drivetrain to recharge the battery. Consequently, the electric machine has to work both as a motor and a generator according to the driving condition.
Besides the environmentally friendly aspect, electric motors have several advantages with respect to the conventional ICE: first of all the efficiency is normally around 65–80% with a limit of 90–97% [8], they are smaller, lighter, cheaper and, above all, they can provide instant torque at almost any speed. The latter is, indeed, the reason why the trasmission system, and the powertrain as a consequence, are much reduced in number of components, size, weight and complexity. Furthermore, an efficient downstream power flow benefits the battery pack design.
The types of electric machines actually employed in the automotive sector are roughly organised in two big groups, the alternating-current (AC) and direct-current (DC) motors. In the Figure 1.8, a graphical representation of the electric machines' family is given. It is worth to notice that the most investigated

technologies are the the DC motors (both separately excited and brushless with permanent magnets) which were used in the earlier electric vehicles above all, the Permanent Magnet Synchronous Motor (PMSM), the Three Phase AC Induction Motors and the Switched Reluctance Motors (SRM).

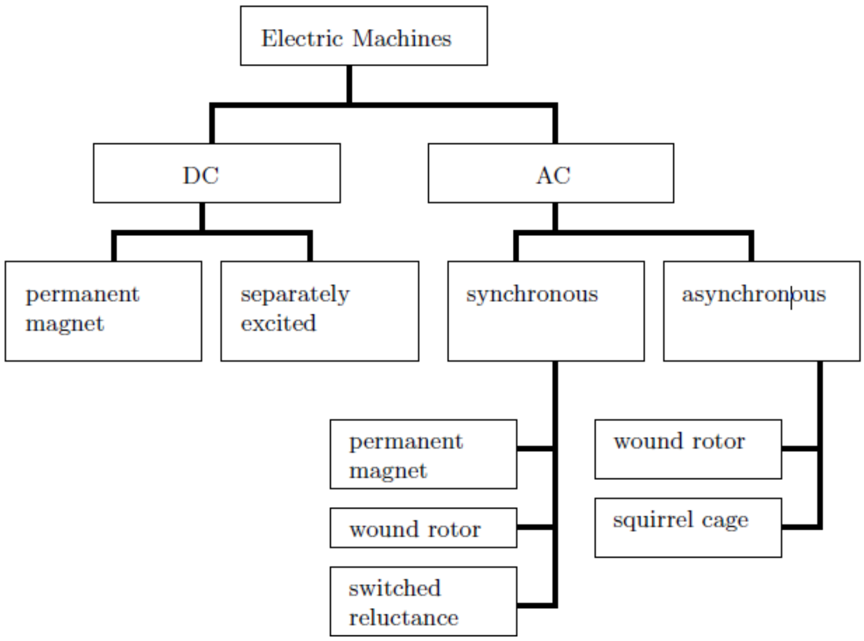

Fig. 1.8: *Family tree of electric machines [1]*

Even though the DC motor is a suitable option for the traction application, it also requires high maintenance due to the wear of the brushes and commutators, while the brushless version (BLDC) is still a valid alternative to the AC motors listed above. Additional details on the technologies and their applications in the current market follow.

Brushless DC Motors (BLDC)
This type of machine can be considered an evolution of the conventional DC motor where the system of brushes is substituted by an electronic commutator. In such a way, the wearing losses are removed and the efficiency of the motor increased at the expenses of the simple design and the low cost of the brushed DC Motors. Moreover, the BLDC machines own significant advantages like high speed range, faster dynamic, suitable characteristic for high performance and high ratio of delivered torque over the size of the motor. Looking at the main disadvantages, the BLDC has a higher cost due to the permanent magnets which introduces a further drawbacks: they could be damaged by the thermal conditions in case of motor overloading for an extended period over the limit (demagnetization).

Permanent Magnet Synchronous Motors (PMSM)
In general, AC motor can boast higher power density and efficiency than the DC motor, but also an increase of costs. Similarly to the BLDC motor, the PMSM are equipped with the permanent magnets on the rotor and the stator windings that, in this case, are supplied by the alternating current which create a rotating magnetic field. Even though the two technologies can be presented as versions of a generic brushless motor with different waveform of back electromotive force (see Table II), the PMSM is characterized by higher efficiency and power ratings which make this solution the most adopted by the automotive manufacturers. For example, Toyota Prius, Chevrolet Bolt EV, Ford Focus Electric, zero motorcycles S/SR, Nissan Leaf, Hinda Accord, BMW i3, etc use PMSM motor for propulsion.

Three-Phase AC Induction Motors
This type of machine is also know simply as the Asynchronous AC motor because the rotor does not turn at the same speed of the magnetic field unlike the Synchronous motor. Both motors use a three-phase design, but the Induction motor does not guarantee the same performance in terms of efficiency, range, torque and power density. Regardless of that, this technology is interesting for the EV manufacturers thanks to some benefits like the easy control of the speed and torque, the self-starting ability which is

missing in the PMSM, the relatively low cost of production since rare-earth material are not required for the magnets. On the other hand, lower weight and compactness is given by the PMSM and mostly high-performance electric cars are still powered by this kind of motor. A possible option is to combine the two variants of motor in the same car model: for example, the asynchronous induction motors are on Tesla Model S, 3, X and Y on front axles while the Permanent magnet motors are used on the rear axles of the same models.

Switched Reluctance Motors (SRM)
Recently, more attention is given to the Switched Reluctance Motor (SRM) for its simple and robust construction. With respect to the solutions described above, the working principle is completely different because it is not equipped with any windings or magnets and more similar to the structure of a stepper motor. In SR motors, both stator and rotor are designed with "teeth" called poles and each stator pole carries an excitation coil. Opposite coils are connected to form one "phase," while the rotor has no windings. When DC voltage is supplied to a phase, the rotor rotates in order to minimize the reluctance of the magnetic path [1].
This solution is particularly attracting thanks to the wider speed range, the insensitivity to high temperature and high efficiency. On the other side, some obstacles like noise, complex power electronics and torque ripples are not overcome, yet.

BLDL	**PMSM**
Synchronous machine	
DC	AC
Trapezoidal back emf	Sinusoidal back emf
Torque ripples at the commutation	No torque ripples
High core losses	Low core losses
Less switching losses	High switching losses
Simple control algorithms	Complex control algorithms
Better for low speed	Higher maximum speed
Noisy	Less noisy

TABLE II
Comparison of BLDM and PMSM attributes [9]

The previous sections gave an idea of the working principle, features, main advantages and drawbacks of the most widespread alternatives for the electric motors. Despite the fact that some solutions clearly better fit the performance requirements, this aspect is not the only one to be taken in consideration during the selection of the suitable motor. Indeed, the operating conditions and the final cost of the vehicle can affect this decision, so that more alternatives can be offered in the automotive market.

1.3.3 The transmission

Generally, any type of vehicles owns a gearbox. The latter is defined as mechanical system able to adapt the transmission ratio between engine and wheels to the need of the vehicle and to the deficiencies of the engine [10]. This necessity is dictated by the fact that the engine alone is not always able to supply the required combination of speed and torque to wheels or because it transmits the power inefficiently.
According to David A. Crolla [11], the functions ascribed to the transmission are:

- Allowing the vehicle to start from rest, with the engine running continuously.
- Let the vehicle stop by disconnecting the drive when appropriate.
- Enabling the vehicle to start at varied rates, under a controlled manner.
- Varying the speed ratio between the engine and wheels.
- Allowing this ratio to change when required.
- Transmitting the drive torque to the required wheels.

According to this list, the transmission is strictly required in conventional vehicles since the ICE is not able to produce torque at zero speed, the maximum torque engine is limited to a certain speed range, the fuel efficiency strongly depends on the operating speed and torque and the motor shaft can only rotate in one way (hence, a reverse gear must be integrated). On the other hand, the same necessity is not applicable to the electric BEVs thanks to the characteristic of the electric motor which has a wider power band (almost 0-20 000 rpm), can produce the peak torque even at zero rpm and its shaft can rotate in both directions.
Even though the electric motor is much more efficient than ICE, the generic motor efficiency map of the Figure 1.9 shows that there are zones at higher efficiency where it is better to set the operating point of the motor, independently of the torque and speed requested by the driver. As mentioned in the previous Sections, the efficiency of the EV powertrain is not only a matter of energy consumption for the final purchaser or for the environmental impact, but an important factor to determine the size of the battery pack and the feasibility of the vehicle project. It is for this reason that the research on EV transmissions is drawing the attention of the researchers.

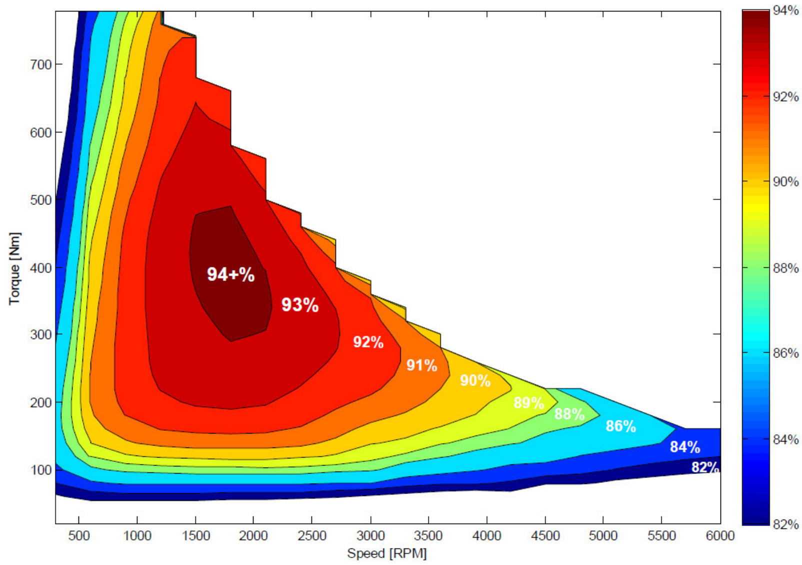

Fig. 1.9: *Permanently Excited Synchronous motor (PMSM) efficiency map, source: x-engineer.org*

In the Figure 1.10 is reported the conceptual evolution in the market and the expected development of the transmission system for ICE, HEV and BEV. Currently, Automated Manual Transmission (AMTs) with single and multi-speed options and the Continuous Variable Transmission (CVT) are considered suitable for the electric vehicles. Despite the first option is widely adopted by the car manufacturers, many researchers are prone to demonstrate the potential improvement of the energy consumption and the driver's experience thanks to the other two alternatives.

Since a deeper analysis of the discussion about the single and multi-speed gear transmissions will be given in the next chapter, only a brief description of the CVT and its applications is here reported for a more complete overview on the EV transmission's state-of-art.

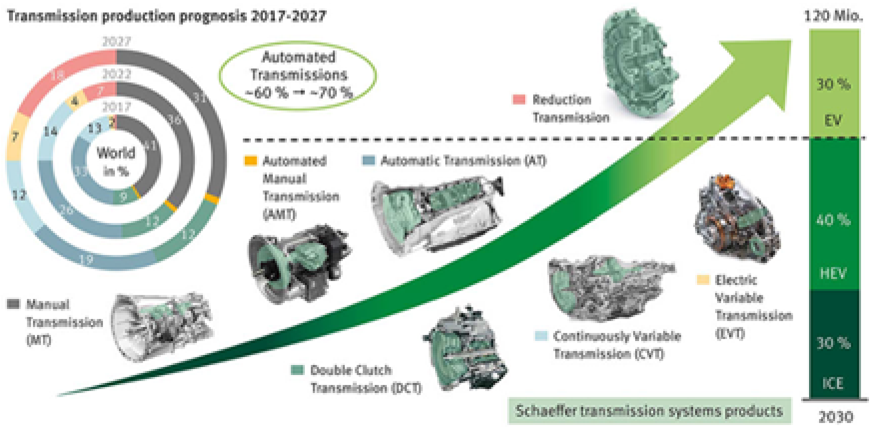

Fig. 1.10: *Automation trend in transmission system and percentage of powertrain concepts in the overall market in Schaeffler scenario for 2030 [12]*

1.3.4 Continuous Variable Transmission (CVT)

The CVT system consist of a two variable diameter pulleys kept at a fixed distance apart and connected by a power-transmitting device, e.g. belt or chain. The belt/chain can undergo both radial and tangential motions depending on the torque loading conditions and the axial forces on the pulleys. This consequently causes continuous variations in the transmission ratio [13]. Despite the slip between the parts determines a lower efficiency of the system with respect to a conventional gearbox, the CVT solution can shift the gear ratio continuously and optimise the operating point of the engine. Moreover, it assures a smooth and comfortable behaviour unlike any other conventional solution.
Today, Bosch Transmission Technology is one of the companies who bet on this technology and introduced CVT4EV in the market promising improved performance, lower energy consumption, high modularity and enhanced user experience [14].

Fig. 1.11: *Design of the CVT4EV introduced by Bosch Mobility Solutions [14]*

1.4 Architectures of the EVs' powertrain

One of the main innovations in the electrification of the vehicle is the introduction of a simple and flexible structure, as briefly mentioned in the Section 1.1. This advantage is not negligible since it gives more freedom to the designers for BEVs with respect to conventional vehicle, where all the car manufacturers are basically constrained to similar architectures because of the volume of the ICE and the supporting components.
In EVs, the vehicle layout adopted, namely the number and the location of the electric machines, affects

the transmission itself and the necessary controls and interconnections, for this reason a small introduction to the possible architectures is given.
A primary distinction is made between the distributed drive system and the centralised drive system (Figure 1.12). In the first case, each wheel is individually powered by a dedicated motor, while in the second one a single or dual motor power unit is coupled to the transmission system that distributes the power to the wheels, similarly to what happens in an ICE solution. Moreover, for the traditional vehicle the choice of a particular layout is determined manly according to the target market sector and brand image [11], whereas the centralized or distributed drive is used in EVs to create favorable conditions for the dynamic control [15].

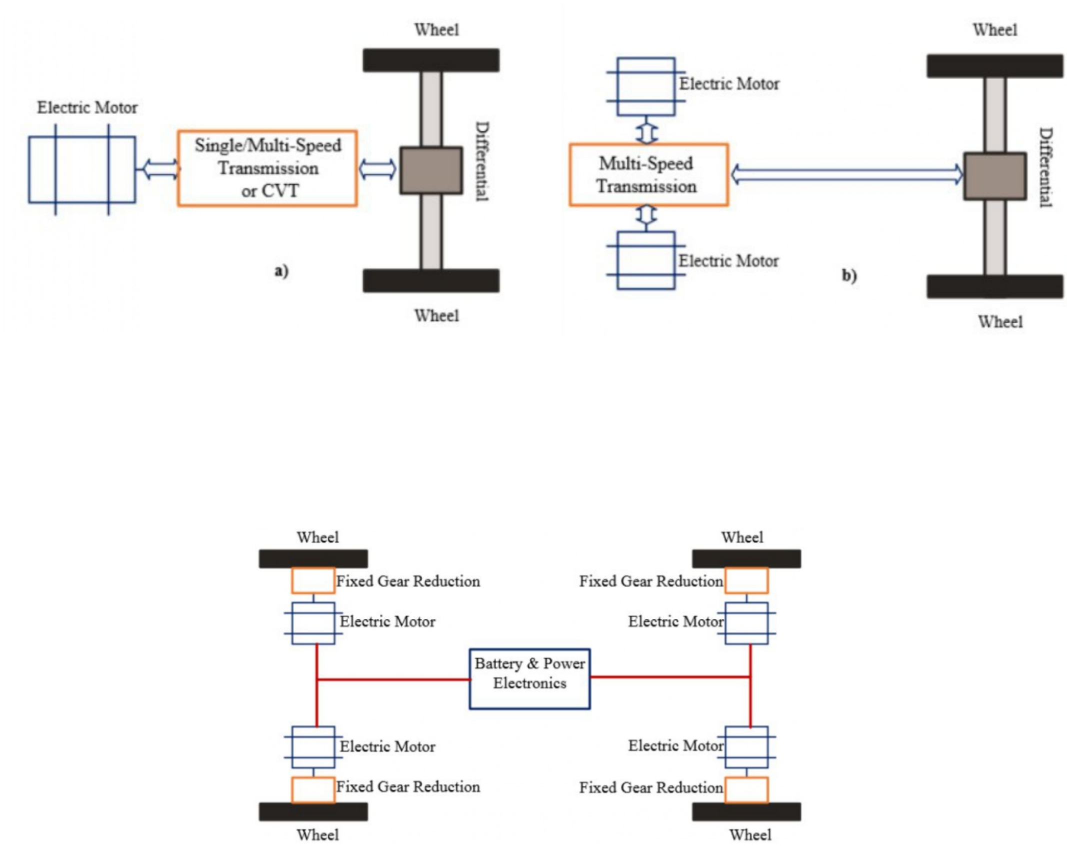

Fig. 1.12: *Centralised (top) and distributed (bottom) drive systems [4]*

Nowdays, the centralised drive system in the single or dual motor arrangement is the most adopted for EVs and its main drawbacks is an higher power loss for the torque distribution together with the need of a large torque output capability. As a consequence, the research is currently focused on the development of innovative multi-speed transmission that can optimise the power distribution and ensure the required performance at any driving conditions.
On the other hand, the decentralised drive solution fix the problem of the energy waste because the system can control the traction torque for each wheel and a single speed transmission is sufficient for the coupling. Unluckily, in critical condition of low adhesion, for example, the wheels are not able to fully exploit the power provided to the powertrain [15].

2 The transmission system in the electric powertrain

2.1 Introduction to the gear transmissions

Despite the advantages of the infinite gear ratios provided by the CVT, the discrete gear transmission is widely confirmed as the leading technology actually implemented in the BEVs market.
In the Section 1.3.4, the description of the typical electric motor's characteristic is useful to explain theoretically why the single speed gearbox is sufficient for an EV: the machine is able to provide a consistent torque at any speed in a wide range. At the same time, even if the electric motor is an high efficient machine, its efficiency can vary between 80-95% [8] and it would be in favor of the limited capacity of the battery to reduce this span and guarantee the dynamic and economic performance.

2.1.1 Gear ratio(s) selection

According to the diagram in Figure 1.2, the powertrain must satisfy initial acceleration ability, gradeability and top speed, above all. Practically, to accelerate or to maintain a defined velocity, the vehicle has to overcome resistances that depend on the situation: required driving conditions, tire rolling resistance, aerodynamics, friction in powertrain and chassis, gravity, and inertia. In Figure 2.1, the main forces acting on a motor vehicle at constant speed in uphill drive are represented.
Looking at the equation 2.2, the *traction* force F_t has to balance the *rolling resistance force* (F_R) which is computed using the vehicle's mass $m_v$the rolling resistance coefficient C_R dependent from the actual velocity of the vehicle, the *slope resistance force* (F_{St}) that depends on the angle α and the *air drag force* (F_L) which depends on the aerodynamic parameter C_D of the vehicle, the air mass density ρ_L, the frontal area of the vehicle A_V the driving velocity $\dot{x}$ and the wind speed $\dot{x_W}$ against the direction of travel.

$$F_R = m_V g C_R \cos\alpha \qquad F_{St} = m_V g \sin\alpha \qquad F_L = \frac{1}{2} C_D A_V \rho_L (\dot{x} - \dot{x_W})^2 \tag{2.1}$$

$$F_t = F_R + F_{St} + F_L \tag{2.2}$$

Hence, the Electric Motor (EM) required power for the traction action at a certain vehicle's speed v is defined in 2.3

$$P_{EM} = F_t \cdot v \tag{2.3}$$

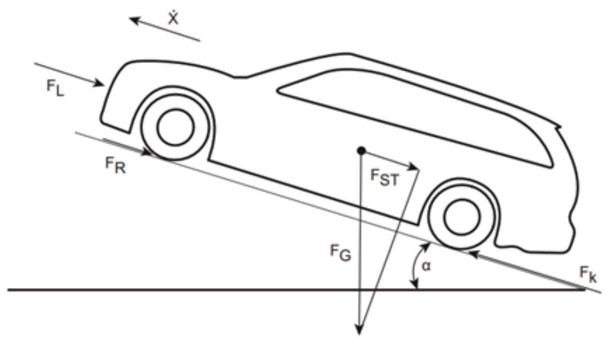

Fig. 2.1: *Forces on a vehicle driving at constant velocity [16]*

As a function of vehicle speed, the traction force is a hyperbolic curve (Figure 2.2) and represents the theoretically maximum tractive load delivered by the electric motor to the wheels. While the ICE is efficient in a narrow speed range, the electric machine is able to pursue the trend of the curve, but the friction and speed limit could prevent the vehicle to seek the desirable dynamic performance. Therefore, the gearbox is always provided to vehicle also to reduce the size and the power required for the motor. As a consequence, the main issues of the electric transmission is the choice of the number and the value of the gear ratios and the appropriate technology used to implement it.

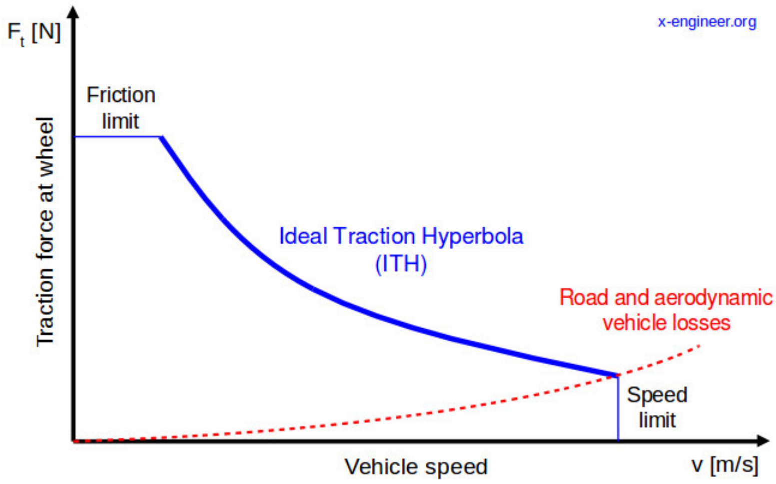

Fig. 2.2: *Ideal Traction Hyperbola [17]*

Focusing on the gradeability and top speed requisites, Walker at el. [18] proposed an analytic approach for the preliminary computation of the suitable range to individuate the optimal gear ratio(s).

- **Ratio Design for Grade**. The design of gear ratios for the capability to climb inclines is considered important for entering and leaving steep driveways and parking structures. The largest overall gear ratio required for the transmission is set according to the ratio of rolling resistance torque for a specified grade, usually equal to 30%, divided by the maximum motor torque T_{EM} multiplied by the overall powertrain efficiency η_{PT}. Faithfully, for low speeds the aerodynamic drag force is assumed to be zero.

$$i_{max} = \frac{(F_R + F_{St}) \times r_t}{(\eta_{PT} T_{EM})} = \frac{r_t m_v g (C_R \cos\alpha + \sin\alpha)}{(\eta_{PT} T_{EM})} \tag{2.4}$$

- **Ratio Design for Speed**. Vehicle top speed varies significantly depending on applications and it is reasonably important for consumer acceptance. The maximum speed achieved in the vehicle can then be used to determine the smallest possible ratio. It must consider the motor characteristics in terms of maximum rotating speed of the motor (N_m) and the ability of the motor torque to reach this top speed. The minimum ratio is defined by the maximum motor speed multiplied by the wheel radius r_t, converted to kph divided by the maximum vehicle speed v_{max}

$$i_{min,speed} = \frac{3.6\pi N_m r_t}{30 v_{max}} \tag{2.5}$$

To confirm the minimum value, the ratio has to be checked against the capability of the motor to supply torque at this speed on its characteristic by dividing the rolling resistance and aerodynamic drag toques by the maximum motor torque at its maximum speed and powertrain efficiency:

$$i_{min,torque} = \frac{(F_R + F_L) \times r_t}{(\eta_{PT} T_{EM,@maxRPM})} = \frac{(C_R m_v g \cos\alpha + 0.5 \times C_D \rho A_V V_V^2) \times r_t}{(\eta_{PT} T_{EM,@maxRPM})} \tag{2.6}$$

Where V_V is the actual linear speed of the vehicle. If this ratio is lower than the previous one, the minimum gear ratio is confirmed as the $i_{min,speed}$. Otherwise, the lowest ratio is given by $i_{min,torque}$.

2.2 Types of transmission

For an EV powertrain, several types of gear transmission system are available to choose such as automatic transmission (AT), dual-clutch transmission (DCT), inverse automatic mechanical transmission (IAMT), uninterrupted mechanical transmission (UMT), automatic mechanical transmission (AMT), etc [19].

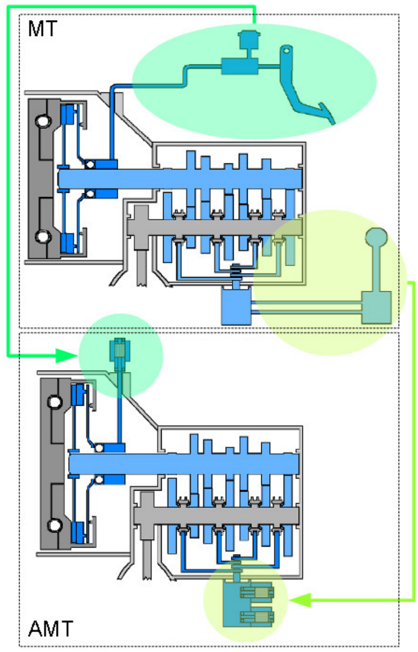

Fig. 2.3: *Conversion of the MT in AMT: in blue the replacement of the clutch pedal with an electrohydraulic / electrical actuator, in green the substitution of the shift lever with an electronic controlled gear actuation mechanism [17]*

2.2.1 ATs and AMTs

The main difference between the AMT and AT is the launch device: the fully automatic transmission uses a hydraulic torque converter and band brakes to replace the dry/wet clutches and smoothly shift the power, while the AMT was initially conceived as a conversion of the manual transmission in automatic manual transmission thanks to electronic controlled clutch and gear actuators. The AMTs maintain the torque interruption and wear issues typical of the MTs, but they are enhanced by an automated actuation system equipped with numerous sensors to activate the electrohydraulic / electric actuators for the clutch action and gear shifting, see Figure 2.3.

For applications where the primary objectives are fuel economy and efficient shifting operations, AMTs are being considered. Indeed, the gearshift is conducted through an actuator according to the command from the transmission control unit (TCU). In this way, it is possible to adjust the torque and speed between the rotor and input shaft of transmission during gear disengagement and engagement process and minimize the torque interruption at the vehicle wheels.
The one-speed transmission can be included in the category of automatic transmission even if there is just one gear ratio, while for the multi-speed solution the definition of the type of transmission is crucial. One of the first attempt to introduce the multi-speed transmission in an electric vehicle was conducted for the first generation of Tesla Roadster in 2006. The aim was to ensure quick acceleration and attractive top speed thanks to two gears, but the producers did not menage to supply a gearbox with satisfactory durability, efficiency and cost challenges [20]. So, Tesla had to delay the launch of the model and to redesign a new powertrain where the single speed transmission was introduced and, today, this solution is still the only one adopted by the EV leader manufacturer.

Model	Acc (0-100km/h) [s]	Top Speed [km/h]	WLTP Range [km]	Total Power [kW]
Audi e-tron GT	3.3	250	472	475
Porsche Taycan	3.2	260	452	500
Tesla Model S	2.5	261	580	639

TABLE I
Performance comparison of three main electric vehicles based on the EV database

Nowdays, top quality car manufacturers claim that the adoption of the multi-speed, and the two-speed gearbox in particular, has its target in the market and it can be competitive in terms of driver experience and investment cost. For example, in 2019, Audi selected a dual motor centralised configuration with a two-speed transmission only on the rear axle for the model E-tron GT, unlike the Porsche Taycan where the centralised architecture is combined with a pure double speed gearbox, respectively in Figure 2.4.

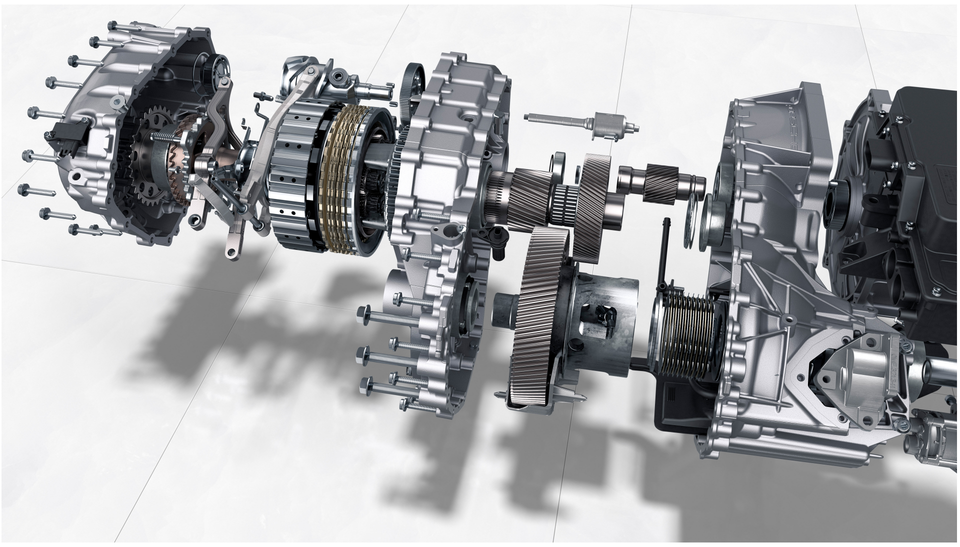

Fig. 2.4: *Two-speed transmission on the rear axle of Porsche Taycan [21]*

Hence, the choice to adopt the single or double speed transmission is a debate on the dynamic and economic performance, the cost, the maintenance and the efficiency of the transmission, above all.

2.2.2 One-speed transmission

As mentioned many times, the one-speed transmission is the most popular solution for the BEVs gearbox. The gear ratio design requires to achieve a balance between range, performance, and top speed in the limits individuated with the analytical computation presented in the Section 2.1.1.
To clarify the relevance of this choice, if a high gear ratio is used the car would not be able to accelerate fast enough, while for a lower gear ratio the vehicle would not achieve the maximum velocity needed. On the other hand, lower gear ratios offer higher potentials to minimize energy consumption, as it is less effected by sudden velocity changes (that occur in higher ratios) due to the fact the driver is limited to perform more smooth accelerations. For example, during the simulation for the evaluation of the effects of the ratio selection on the vehicle's energy consumption, Spanoudakis P. et al. [22] found out that compared to ratio 1:8, ratio 1:6 provided 2.6% lower mean consumption and ratio 1:10, 4.2% higher.
Due to the presence of a single gear ratio, the motor must deliver power over a very wide speed range, that is, high power at low speed to maximise acceleration performance and at high speed to overcome higher aerodynamic drag losses. As a consequence, the use of single speed or direct drive motors requires a larger motor with wide torque and speed ranges to achieve both of these objectives [18]. Moreover, the fixed ratio do not consent to move in more efficient zones of the motor characteristic accordingly to the velocity profile of the car.

Overlooking the limits of the simplest transmission configuration, the one-speed gearbox offers several advantages. First of all, it occupies reduced space lowering the total weight of the powertrain. Secondly, there is no need of clutches or extra gears which means less wear, power losses, noise and maintenance. Thirdly, the drive is very smooth due to the fact that any torque interruption is necessary for the gears shifting. Lastly, the design and manufacture of the system is simpler and less expensive than the other alternatives, as a consequence.

Fig. 2.5: *Single speed gearbox of Tesla Roadster's prototype 1.5 powertrain [23]*

All these benefits justify why mostly car manufacturers are still stuck on this technology. In the Figure2.5, it is possible to appreciate the simplicity of the single speed transmission, with a gear ratio (8.2752:1) that was adopted in Tesla Roadster 2008 after the failed attempt of the implementation of a two-speed gearbox in the previous generation.

2.2.3 Two-speed transmission

Even though the one-speed transmission is widely employed thanks to its simpleness, it obliges car manufacturers to find a trade-off for low-speed torque or high-speed efficiency in a single gear ratio. Instead, the double speed transmission gives the possibility to not renounce to any requests: it provides the first gear ratio to supply the acceleration and grade climbing capability at low speed, while the second gear ratio enhances the top speed requirements at cruising speed in an efficient range.
Recalling the disadvantages of the one-speed solution, it is immediate to recognise the complementary benefits of the innovating transmission. The two gear ratios allow the speed range of the car to be uncoupled from that of the drive motor, keep the motor working on its efficient operating region, improving the energy consumption and the recovery during the breaking phase, it comes closer to ICE driving experience thanks to the enhanced dynamic performance and it goes towards batteries' necessity to optimise the power management.
On the other side, the second gear brings with itself accentuate complexities and new challenges. In the first category it is included the issue of the gear ratios optimisation which was already discussed for the one-speed gearbox. In this case, the complication is given by the fact that the algorithm has to search the optimal ratios considering several factors. Indeed, the gap between the two values cannot be excessive to ensure the feasibility of the mechanical layout and the gear shifting can be affected by an high step ratio compromising the driving comfort. For this reason, the shift scheduling strategies are a central topic for the design of the two-speed transmission.
Further considerations about the multi-speed gearboxes are made on the addition of components to the transmission's design which leads to the increase of the weight, volume, power losses and costs.

2.2.3.1 Dual Clutch Transmission (DCT)

The greatest part of the examples available in the literature about the two-speed transmission is focused on the DCT, although this solution was born for the multi-speed gearbox of conventional ICE vehicles. It combines high efficiency of MTs with convenience of automatic transmissions by simultaneously changing between two primary clutches for gear changes. The two clutches have a common drum attached to the input shaft from the motor, and the friction plates are independently connected to the first and second gears, respectively.
With only two gear pairs and a final drive gear in the two-speed gearbox, it is a comparatively simple transmission, without the requirement to engage alternate gears using synchronizer mechanisms. For just two gear ratios, gear shifting is realized through dual clutch control alone. Consequently shift quality and driving comfort can be significantly improved in comparison to MTs, thus making DCTs well suited to applications in BEV powertrains [24].

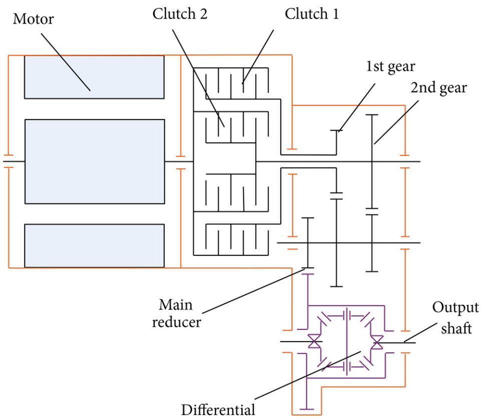

Fig. 2.6: *Representation of DCT for the EV powertrain [24]*

2.3 Power losses and efficiency computation

The transmission in the electric axle drives plays a key role in EVs because, in contrast to internal combustion engine powertrains, a significantly higher proportion of the total losses occur in the transmission [25].

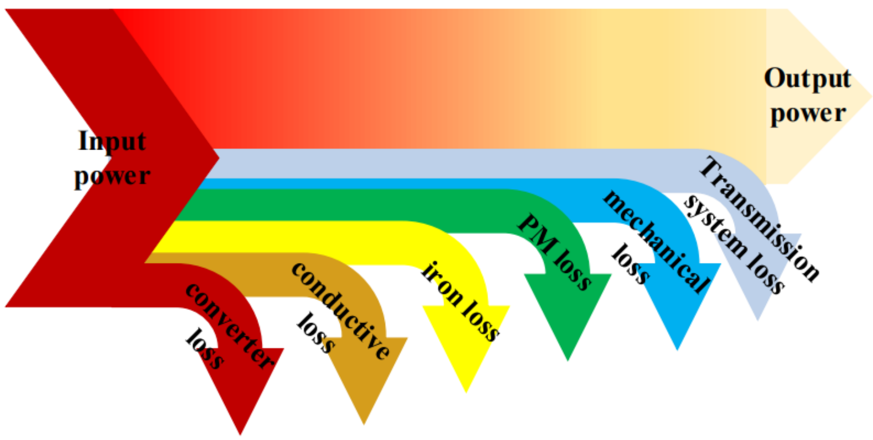

Fig. 2.7: *The various loss components of the propulsion system of EVs [26]*

The transmission efficiency is determined by the power losses occurring in the mechanical components of the gearbox, as defined in the Equation 2.7. Mostly, the loss maps are based on measurements, while an explicit computation of the gearbox losses is done rarely. In the literature, some analytical models are available to compute the power lost during the transmission, useful to estimate the efficiency of the unit itself. One of the main advantages of having a reliable analytical estimation of the powertrain performance is the possibility to select the suitable energy density of battery skipping the practical measurements.

$$\eta_{gearbox} = \frac{P_{input} - P_{loss}}{P_{input}} \tag{2.7}$$

Generally, the power losses of a gearbox are distinguished among losses generated by the gears (subscript Z), by the bearings (subscript L) and by the seals (subscript D). Furthermore, it is common to distinguish the losses that are directly depending on the transmitted power and the losses that are independent from the applied load (subscript 0). In particular, for what concern the gears, these are respectively identified as the meshing losses and gear churning losses [27].

$$P_V = P_{VZ} + P_{VZ0} + P_{VL} + P_{VL0} + P_{VD} \tag{2.8}$$

The meshing loss P_{VZ} is the gear load-dependent contribution to the losses due to the sliding between the tooth flanks under load, whereas the churning losses P_{VZ0} are speed-dependent. The former is defined

taking in consideration the geometry of the tooth, the average friction factor of the contact segment and the transmitted power of each gear pair, while the latter is determined by the resistance of the fluids like oil and air during the gears spinning action.
Similarly, bearing losses are distinguished in torque-dependent contribution P_{VL} due to the sliding and rolling friction and load-independent loss P_{VL0} caused by churning, splashing and windage phenomena [28]. Finally, the sealing losses P_{VD} depend on the oil viscosity and shaft's geometry.
To the losses previously described is necessary to include those ones coming from additional components such as synchronizers and clutches. Indeed, the multi-speed transmission implicates the equipment of the drivetrain with a shifting system to ensure the change of the gear ratios during the drive.
Although the presence of further shifting elements, the possible drop of the transmission efficiency is not a valid argument to exclude the two-speed gearbox from the EV powertrain concept. The reason why the debate is still open on the new option has to be found on the improvement of the fuel consumption of the multi-speed transmission. In the study [29], Walker et al. demonstrates that adding a second gear in the transmission could lead to improve the energy-utilizing efficiency by above the 10 %. On the other hand, it is important to examine if the reduced complexity and weight of the single speed solution can offset the fuel consumption advantages.

Part II

3 One-speed gear transmission

3.1 Description of the model

Typically, one-speed transmissions are realised in two stages, as represented in Figure 3.1. Anyway, this configuration is not unique.

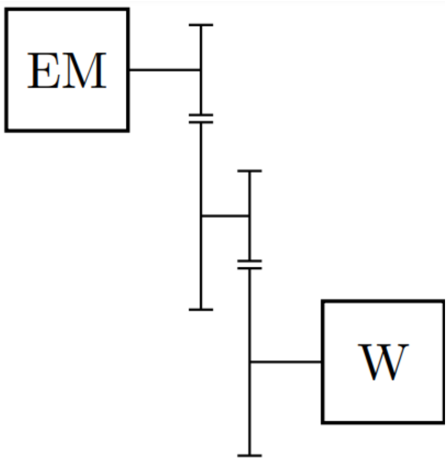

Fig. 3.1: *Stick diagram of a one-speed transmission [30]*

The gearbox of the Volkswagen ID.3 Pro Performance is chosen as a reference to model the one-speed transmission for the reasons explained in the Paragraph 3.1.1. In the Figure 3.2 is represented the dual stage gearbox: the motor input shaft transmits the power to the countershaft through the first gear pair, while the second engagement transfers the power to the output shaft.
The two-stages layout gives more freedom to achieve the required gear ratio in a more compact solution since it possible to play with the gear ratios of the single pairs to get the overall ratio as required.

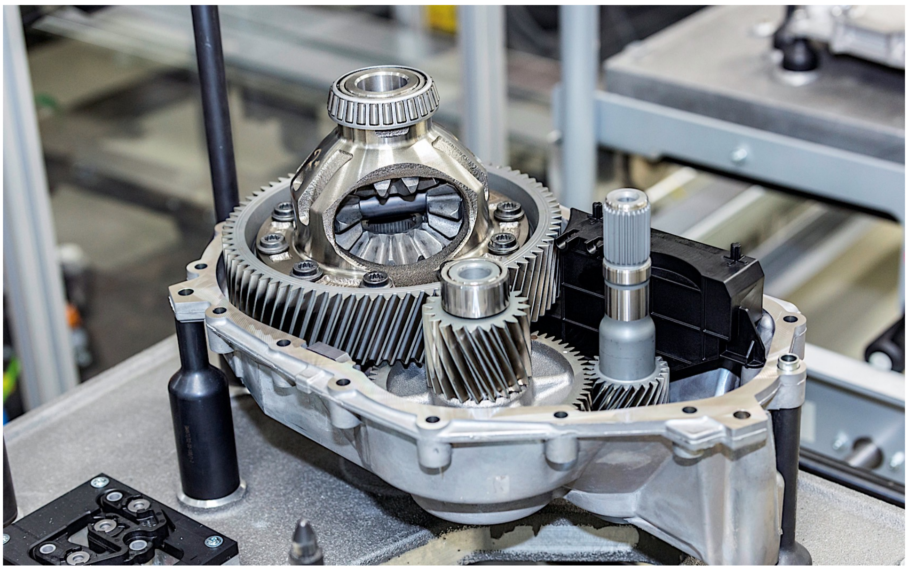

Fig. 3.2: *Gearbox of VW ID.3 Pro Performance [31]*

3.1.1 Information on the Volkswagen ID.3 Pro Performance

For the aim of the project, the Volkswagen ID.3 Pro Performance in Figure 3.3 is the reference vehicle to get the necessary information about engine's characteristics, vehicle performance, powertrain structure, etc.

Fig. 3.3: *Volkswagen ID.3 Pro Performance [31]*

The Volkswagen ID.3 is rear-wheel-drive with motor and gearbox located on the rear axle. The powertrain is made of an electric motor powered by Lithium-ion battery and the transmission consist of a direct drive single speed gearbox with a gear ratio equal to 10.

Motor maximum power	P_{max}	150 kW
Motor maximum torque	T_{max}	310 Nm
Motor speed @ T_{max}	N_m	6000 rpm
Vehicle top speed	v_{max}	160 km/h
Batteries' net energy capacity		58 kWh
Electric range (WLTP)		426 km
Acceleration (0-100km/h)		7.3 sec
Vehicle consumption		166 Wh/km
Transmission gear ratio	i_{tot}	10

TABLE I
Motor and vehicle's performance

The choice fell on this vehicle thanks to the good availability of the useful parameters for the design of a reliable transmission model.

3.2 KISSsoft design

KISSsoft is the tool used to design the transmission. Practically, the gearbox is modeled in KISSsys, while the computations are made in KISSsoft's calculation sheets.
In Figure 3.4 is reported the KISSsys model tree where it is possible to distinguish the main components and their sub-elements. The main folder "GB" contains the whole transmission and the following elements are added:

- *Shaft1* (Input shaft), *Shaft2* (CounterShaft) and *Shaft3* (Output Shaft).
- Two rolling bearings to bear each shaft.
- Helical gears *z1* (pinion), *z2*, *z3* and *z4* (output gear).
- *Coupling1* and *Coupling2* in Shaft1 and Shaft2 to define the boundary conditions of input and output, respectively.
- Gear pair constraints *GearPair12* and *GearPair34* to define the connection between the matching gears.

- Blue symbols for the shafts and the connections are the computation sheets that link the KISSsys model to the KISSsoft calculation window, where is possible to define the design parameters(diameters, length, tooth, pressure angle, etc...)

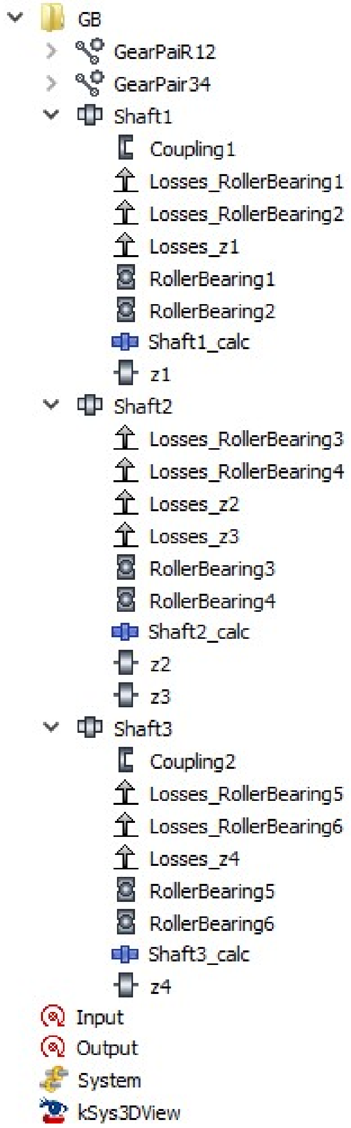

Fig. 3.4: *KISSsys model tree of the one-speed transmission*

The input is defined using the parameters of the motor and the vehicle's performance. The available information are shown in the Table I.

To clarify the power flow in the transmission, in Figure 3.5 is represent the diagram supplied by KISSsys. While in the Figure 3.6

3.2.1 Gear Sizing

By opening the KISSsoft computation sheets, it is possible to proceed with the sizing of the gears. The same procedure is repeated for both the engagements and it consist of two sizing steps.
In the **rough sizing** phase, the most important parameters to define are the required ratio and its allowed variation in percentage. Then, it is also possible to insert the approximated value of the helix angle in case of helical gears.
The gear ratios of each stage are represented in Table II. These values permit to have a balanced distribution of the power in each shaft and to avoid any interference between the parts (i.e. shafts and

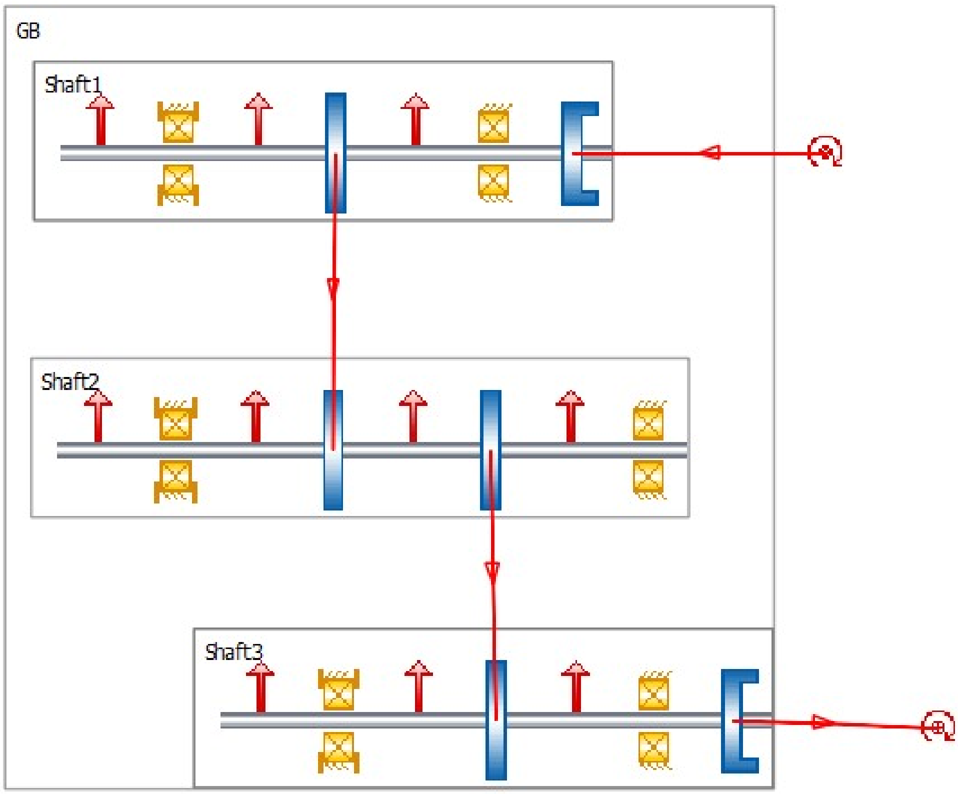

Fig. 3.5: *KISSsys diagram of the one-speed transmission*

bearings of different stages). The total gear ratio diverges from the theoretical transmission ratio (i.e., 10) with an error of 7.23 %, which is the closest achievable number with the given data.

First stage gear ratio	i_1	3.84
Second stage gear ratio	i_2	2.43
Actual total gear ratio	i_{tot}	9.33

TABLE II
Gear ratios for the one-speed transmission

In the **fine sizing** phase, it is possible to define more details as inputs for the iterative computation. For the design of the model, the interesting parameters are, again, the required ratio and its permissible error, the pressure angle, the acceptable range of the helix angle, center distance, normal module and profile shift coefficient. For the latter, in particular, there is the option to fix the value for one of the two gears. Given that the Matlab model that has to be validated does not include the contribution of this parameter, it is necessary to set at zero the profile shift coefficient of one of the two gears at least and, then, among the possible solutions listed after the iterations, to chose the one with the lowest coefficient for the second gear.
The Table III summarises all the information of the four gears' geometry.

3.2.2 Bearings selection

The selection of the bearings is useful for the definition of the shafts dimensions. Indeed, the lack of information about the transmission model of the reference vehicle leads to an atypical procedure for the design of this transmission model.
Firstly, the method consist of several iteration to find the suitable combination of geometry (i.e., the diameter of the shaft) and bearing selection for the input shaft. Secondly, the rest of the transmission is

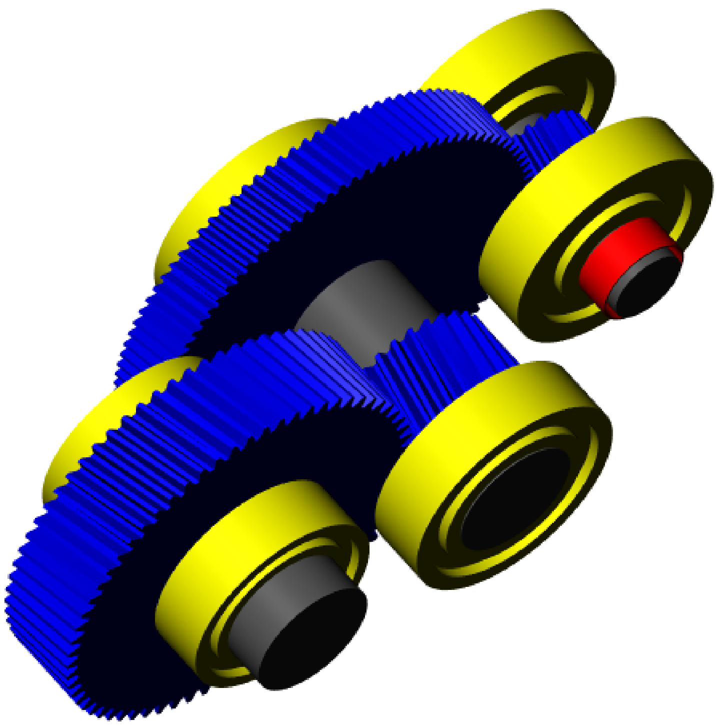

Fig. 3.6: *KISSsys diagram of the one-speed transmission*

Gear	Pitch radius r [mm]	Number of tooth z [-]	Module m [mm]	Pressure angle α [°]	Helix angle β [°]	Band width b [mm]	Center distance a [mm]	Profile shifting x [mm]
1	44.57	25	1.75	20	11	23.9	107.90	0.000
2	171.15	96	1.75	20	11	23.9	107.9	0.002
3	63.75	28	2.25	20	12	37.1	109.28	0.000
4	154.81	67	2.25	20	12	37.1	109.28	0.008

TABLE III
Gear specifications for the one-speed transmission

sized proportionally using the Figure 3.2.

For the first step, the *SKF Bearing Select* tool available online in SKF official website is used. For each iteration, these are the followed stages:

1. The forces acting on the shaft are computed, knowing the geometry features of the gears defined in the previous Section *Gear Sizing*.
2. A possible couple of bearings is chosen from the catalogue.
3. The rating life of the two bearings is verified.

The required bearing service life is esteemed to be around 8000 hours and the available information about the motor performance refer to the maximum torque and its corresponding engine speed.
Given that the motor is not supposed to operate continuously in the above-mentioned working point, it is sufficient to verify that the rating life of the chosen bearings is included in a minimum range of 5-10 % of the required rating life of the transmission. Moreover, for sake of simplicity, two bearings of the same type support each shaft.
The Table IV summarises the selected bearings and the corresponding rating life.

Shaft	Bearing type	Rating life L_{10mh} [h] (left/right)	Rating life L_{10mh} [%] (left/right)
1	6406	3270/5130	40.9/64.1
2	32210	522/934	6.5/11.7
3	33010	949/1890	11.9/23.6

TABLE IV
Bearing selection for the one-speed-transmission

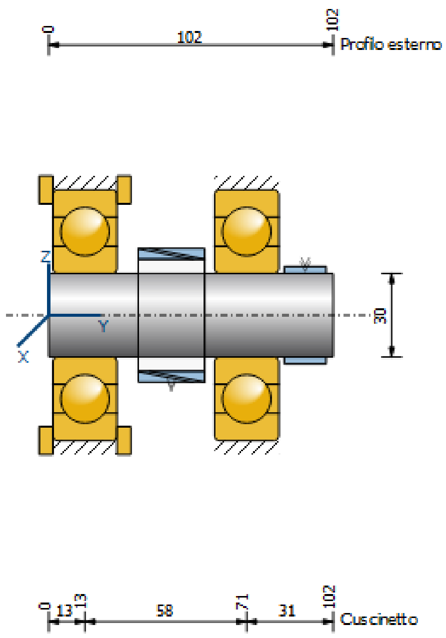

Fig. 3.7: *Sketch of the input shaft for the one-speed transmission*

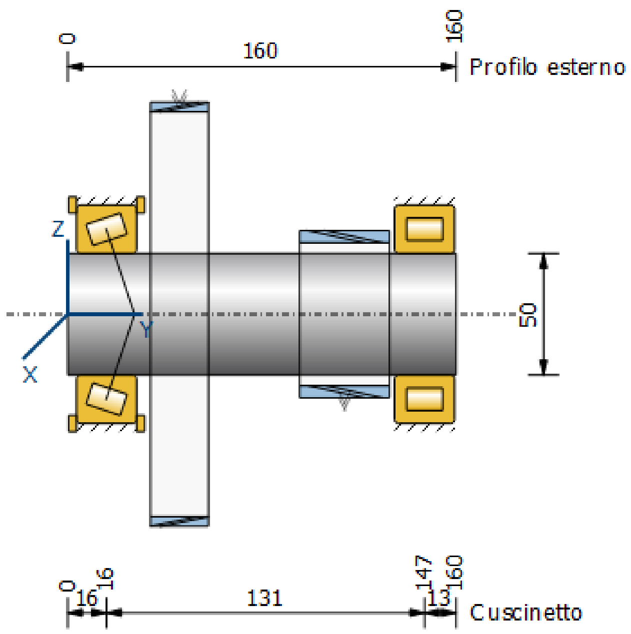

Fig. 3.8: *Sketch of the countershaft for the one-speed transmission*

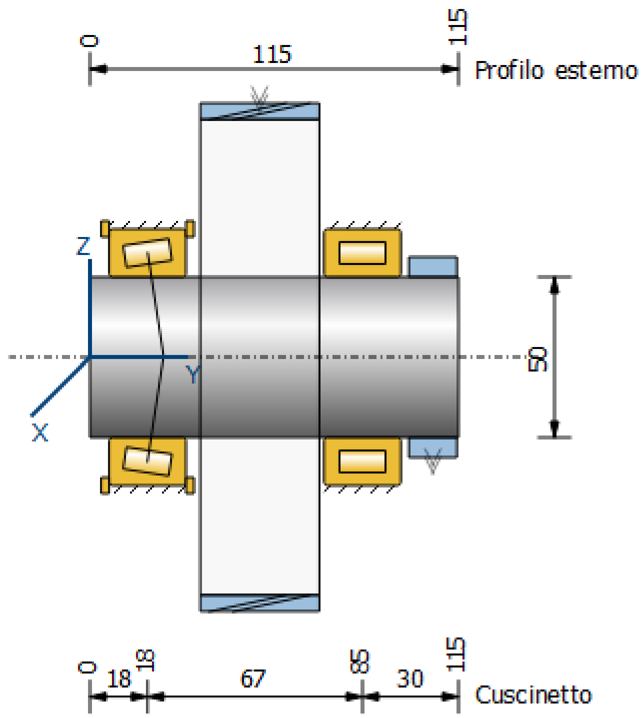

Fig. 3.9: *Sketch of the output shaft for the one-speed transmission*

3.2.3 Load spectrum

The constant input is not the actual working condition of the gearbox, hence it is necessary to introduce the load spectrum to simulate a reliable driving pattern.
To harmonize the test procedure among different countries the WLTP (Worldwide Harmonized Light Duty Test Procedure) was developed [1]. It consist of four parts with different average speeds: low, medium, high and extra high, as can be seen in Figure 3.10.

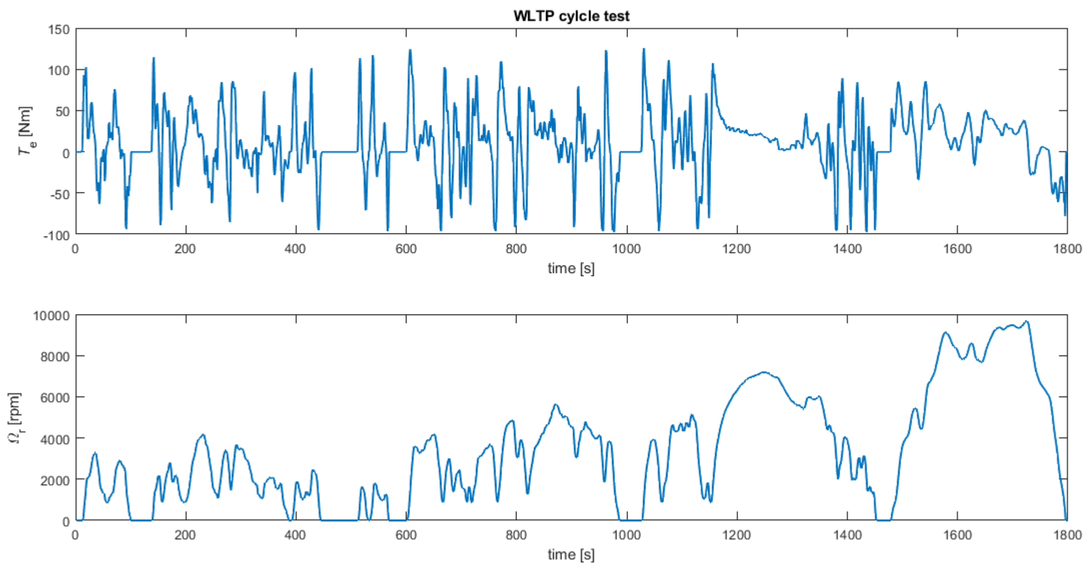

Fig. 3.10: *WLTP test cycle*

In KISSsoft, the load spectrum is usually defined as a combination of frequency, motor torque and rotational speed to define the working point for each bin, see Figure 3.11. Given that the standard test cycle lasts 1800 seconds, some computation errors occurred for the definition of the frequency and the load spectrum was defined through time in hours, instead of time fractions. In order to that, the rain-flow approach was used to calculate the amount of hours that characterized each bin for the entire required service life of the gearbox (8000 hours).

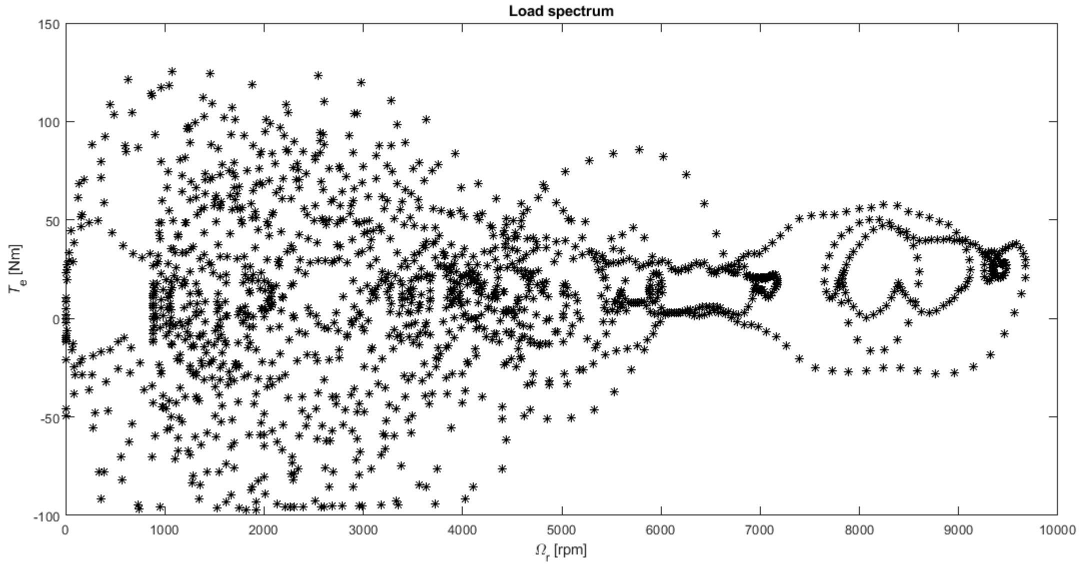

Fig. 3.11: *WLTP test cycle*

4 Two-speed transmission

4.1 Description of the model

The double-speed gearbox can be realized in different solutions. In Figure 4.1 is illustrated the stick diagram of a two-speed AMT with the typical dual stages configuration and the synchronizer. Anyway, this solution is not unique since a dual planetary gear transmission (PGT) which is one of the preferred layout to aim compactness and high efficiency in the ATs. Indeed, PGTs offer more options for generating transmissions ratios, more compact, space and weight saving design, noise reduction, higher efficiency, more favorable load distribution and higher load carrying capacity in comparison to conventional transmissions [32]. Consequently, the complexity, the cost and the efficiency of the power-train is strongly affected by the features of the PGTs design and control.

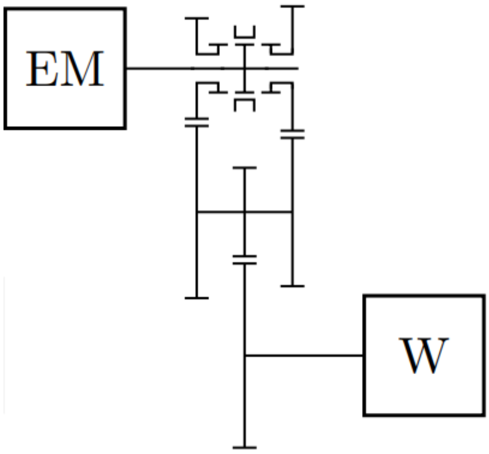

Fig. 4.1: *Stick diagram of a two-speed ATM transmission [30]*

The **single planetary** gear set consists of a central gear (sun), planet gears supported by a planet carrier and an internal gear (ring), as represented in Figure 4.2.
Starting from the simple configuration, it is possible to define **complex compound PGTs** with the introduction of stepped and/or meshing planets, reverted layout with double suns or open PGTs. Modifications of this type offer a larger number of degrees of freedom, and transmission ratios as a consequence.
The same objective can be reached using **coupled PGTs**, consisting of multiple sets of single and/or compound PGTs connected trough the central shafts in various way. In this case, the complexity of the design and manufacturing processes decreases with respect to the compound configuration, but the number of parts increase at the expense of the transmission's weight and compactness.

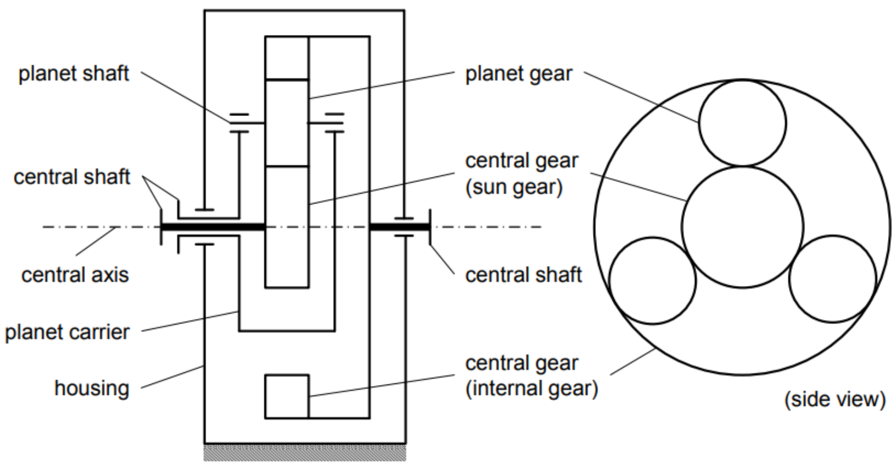

Fig. 4.2: *Example of a single PGT [32]*

For this project, the design proposed in the article *Design 2-Speed Transmission for Compact Electric Vehicle Using Dual Brake System* [33] was chosen as a reference for the model of the two-speed transmission. It consists of a compound reverted PGT with the two central gears connected to two separated brakes, three stepped planets meshing with the ring only on one gear while the other one engages just with the bigger sun. Shown in Figure 4.3, this is a clear example of the typical compactness of this category of transmission, since the carrier element is shared by the two stages, one ring gear is omitted and the two planets are merged in a single pin shaft.
Looking at complete configuration, the power flows in input from the motor shaft to the gearbox through the external gear of the casing and it is transferred from the planetary gear unit to the differential in output thanks to an external gear on the carrier.

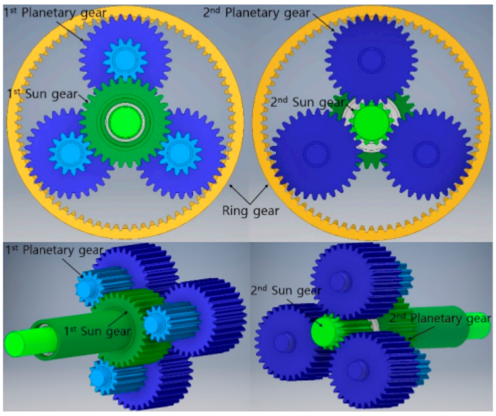

Fig. 4.3: *3D representation of the proposed two-speeds transmission [33]*

The Figure 4.4 illustrates the working principle of the gearbox in the two configurations realised fixing one sun st time with the two-disc braking system. Whereas, the stick diagrams in the Figure **??** are used to define the equations 4.1 and 4.2 that express the value of the two gear ratios.

$$i_{CR,1} = \frac{\alpha_C}{\alpha_R} = \frac{P1 \times R}{P1 \times R + S1 \times P2} \tag{4.1}$$

$$i_{CR,2} = \frac{\alpha_C}{\alpha_R} = \frac{R}{R + S2} \tag{4.2}$$

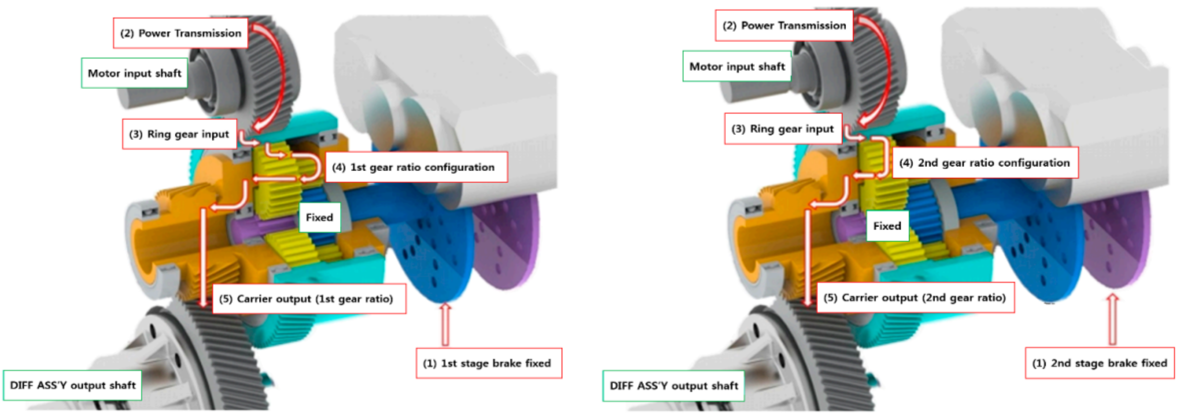

Fig. 4.4: *Working condition for the first and second gear ratio realization, respectively [33]*

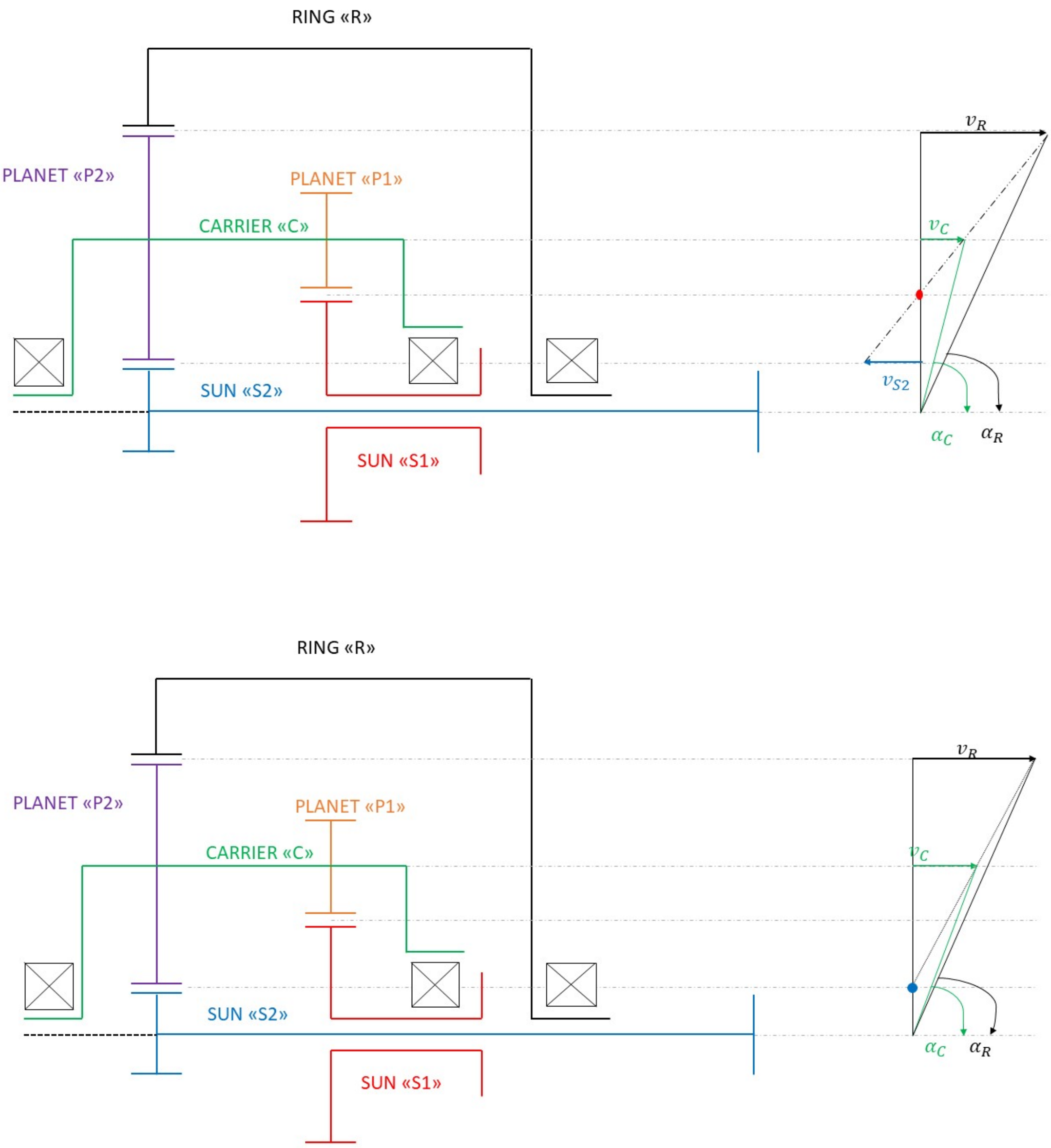

Fig. 4.5: *Stick diagrams of the first gear ratio $i_{CR,1}$ and the second gear ratio $i_{CR,2}$*

4.2 Gear ratios computation

The purpose for the design of the second model is to maintain the same motor characteristic presented in the Section 4.1 and to adapt the two-speed transmission to the performance of the Volkswagen ID.3 introduced in Section 3.1.1.
In order to preserve the dynamic performance of the reference vehicle, it is necessary to compute the new gear ratios using the information displayed in the Table I and the computation procedure explained in the Paragraph 2.1.1. As a matter of fact, the aim of this project is the design of a reliable model to build up an efficiency map and not the definition of the optimal gear ratios which is a complex issue as mentioned in Paragraph 2.1.1.
Hence, in the Table I are showed parameters used for the computation of the two gear ratios with the analytical approach, while in the Table II are reported the resulting ratios of the Equations 4.1 and 4.2.

Vehicle maximum speed	V_V	160 km/h
Gross Vehicle Weight	m_v	2270 kg
Wheel radius	r_t	0.26 m
Drag coefficient	C_D	0.267 -
Rolling resistance	C_R	0.016 -
Frontal Area	A_V	2.36 m^2
Road slope	θ	30°

TABLE I
Parameters for the gear ratios computation

First gear ratio	$i_{1,min,speed}$	9.40
Second gear ratio	$i_{2,min,torque}$	3.60

TABLE II
Required gear ratios for the two-speed transmission

To verify if the obtained gear ratios are compliant with the layout introduced in the Section 4.1, a first estimation of the gear sizes is carried out in a Matlab code implementing the Equations 4.1 and 4.2. Starting from an estimated total volume of the gearbox, the approximated diameters of the two suns and of the ring are the inputs, while the dimension of the stepped planets is the output. In addition, during the iterations for the computation of the overall gear ratio, the input and output gear ratios are considered variable so that additional degrees of freedom can help to achieve the required ratios with feasible gear sizes.
Looking at the Table III, after several attempts, it is evident that there is a noticeable disparity between the diameters of the two planet gears. Since these two are splined on the same pin shaft, it is not possible to arrange the gear ratio of the Table II with the proposed configuration.

	First attempt	Second attempt	Third attempt	Forth attempt
GEAR	Diameters [mm]			
Ring **R**	119.5	120	80	110
Sun, first stage **S1**	61	62	40	57
Sun, second stage **S2**	23	24	15	24
Planet, first stage **P1**	10.25	10	7.5	10
Planet, second stage **P2**	48.25	48	32.5	43
Carrier **C**	71.25	72	47.5	67
GEAR RATIO	[-]			
Input stage i_{in}	1.7	1.9	2	1.8
Output stagei_{out}	1.625	1.45	1.5	1.625
First overall i_1	9.4	9.6	9.5	9.4
Second overall i_2	3.3	3.3	3.6	3.6

TABLE III
Bearing selection for the one-speed-transmission

Since the gear sizing is not practically possible with the computed gear ratios presented in Table II, an alternative must be found.
For sake of simplicity, the gear ratios implemented in the reference article [33] are taken in consideration and shown in the Table IV.

First overall gear ratio	$i_{1,ref}$	9.953
Second overall gear ratio	$i_{2,ref}$	5.687

TABLE IV
Required gear ratios for the two-speed transmission

The reference ratios are acceptable because they overcome the minimum values recommended by the analytical computation. In particular, the first gear ratio is very similar to the lower limit ensuring the

acceleration time 0-100 km/h being less than 7.3 seconds as indicated in Table I, while for the second gear ratio further examinations are needed. Indeed, the difference between the minimum value and the reference one is around the 36,7% and the operating point on the motor characteristic could be very distant to the recommended one.
The Figure 4.6 represents the efficiency map of the Volkswagen ID3's motor to individuate the operating point for the second gear ratio. Using the Equation 2.6, the coordinates in Table V are computed and displayed on the map. In this way, it is evident that the working point is in a high-efficiency area and, as a consequence, it is possible to proceed to the KISSsoft design with the gear ratios on the Table IV for the considered vehicle.

Gear ratio	$i_{2,ref}$	5.687	-
Motor maximum rotating speed	N_m	7334	rpm
Motor maximum torque @ maximum speed	$T_{EM,@maxRPM}$	62	Nm

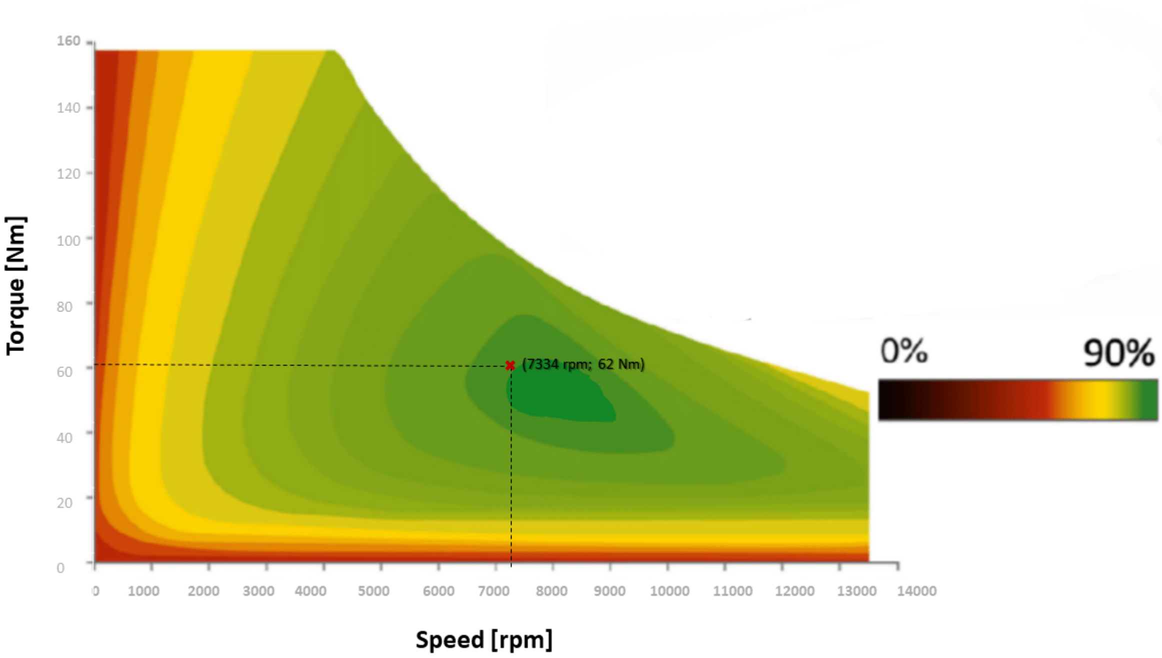

Fig. 4.6: *Motor efficiency map with the operating point for* $i_{2,ref}$ *[34]*

4.3 KISSsoft design

The design in KISSsys is organized in three folders: the *GroupBox*, the *Input group* and the *Output group*. The first one is the main group which contains the fundamental elements of the PGT, while the input and output groups respectively collect the motor shaft and gear engaging the external ring gear and the differential gear meshing with the external carrier gear.

Fig. 4.7: *KISSsys model tree of the two-speed transmission*

The Group box is made of five coaxial shafts:

- the **Carrier Shaft** which constitutes the carrier element containing the *Roller Bearings* and the *Planet* folder that defines the structure of the three stepped planets, where only the second gear engages the ring gear;
- the **Casing Shaft** where the external ring to engage the motor gear is located. It is rigidly connected to the Ring Shaft trough the *ConnectionRingCasing*;
- the **RingShaft** that provides the internal gear to the simple planetary stage;
- the **SunShaft 1** which support the first sun gear that engages the first planet gear;
- the **SunShaft 2** which support the second sun gear that engages the second planet gear;

The computation sheets in blue follows are the *PlanetShaftCalc* which defines the geometry of the planets (Figure 4.8) and the *MainLineShaftCalc* that includes the sizing of all the coaxial shafts mentioned previously (Figure 4.9). The constraints for the gear meshing are the two *PlanetGearPair* between the second planet gear and the ring, and the planet gear and the second sun which are collected in the *PlanetaryGearSetcalc* sheet (Figure) for the gear sizing of the planetary stage. On the other hand, the constraint first sun gear and the correspondent planet gear is taken into account with the *SunPlanetConstraint 1* sheet (Figure). Similarly, the sizing of the shaft and the gear of the motor and the differential shaft are defined in the *GearPair Input* and *GearPair Output* constraints.
The boundary conditions are the following: on the Ring is set the input torque and speed, the Carrier is free from any power constraints, while on the Sun is selected which of the two central gear has to be fixed to realise the requested gear ratio (the first sun is fixed for the first gear ratio and the second sun is fixed for the smallest gear ratio), as shown in Figure 4.10.

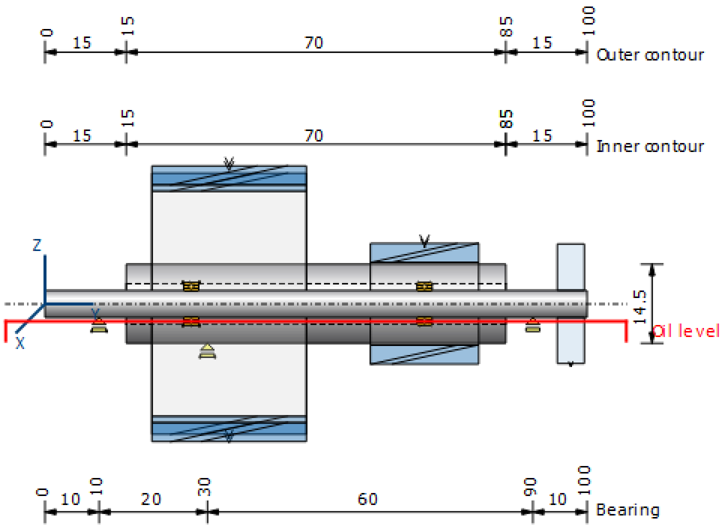

Fig. 4.8: *Sketch of the stepped planet shaft for the two-speed transmission*

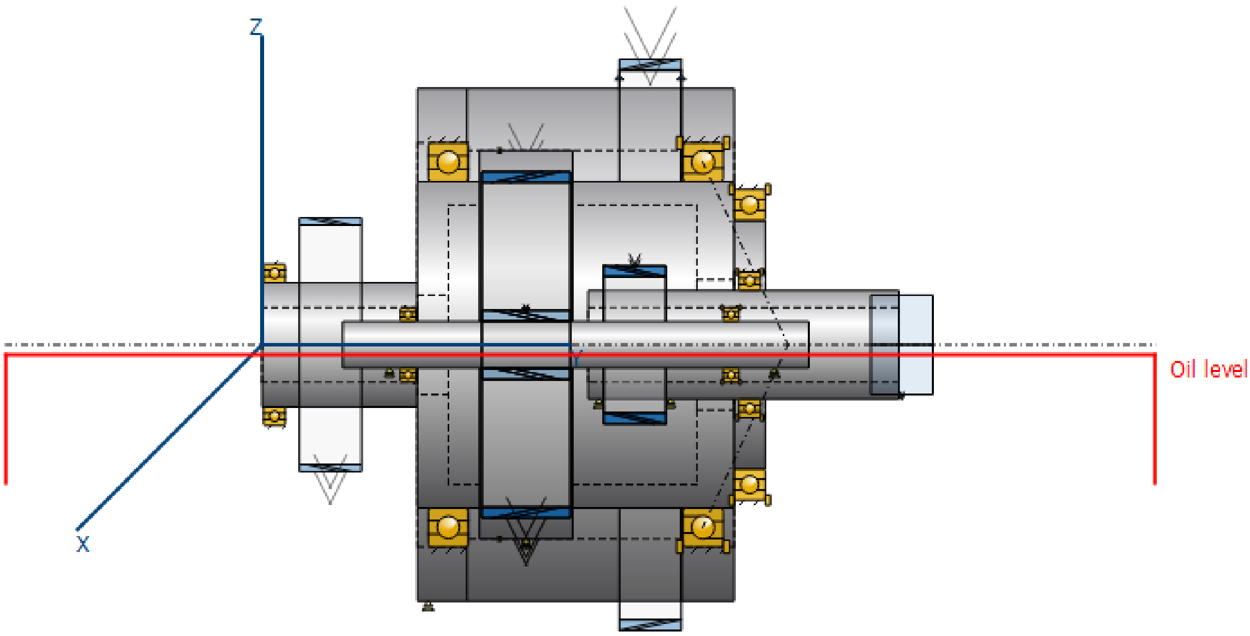

Fig. 4.9: *Sketch of the coaxial shafts group for the two-speed transmission*

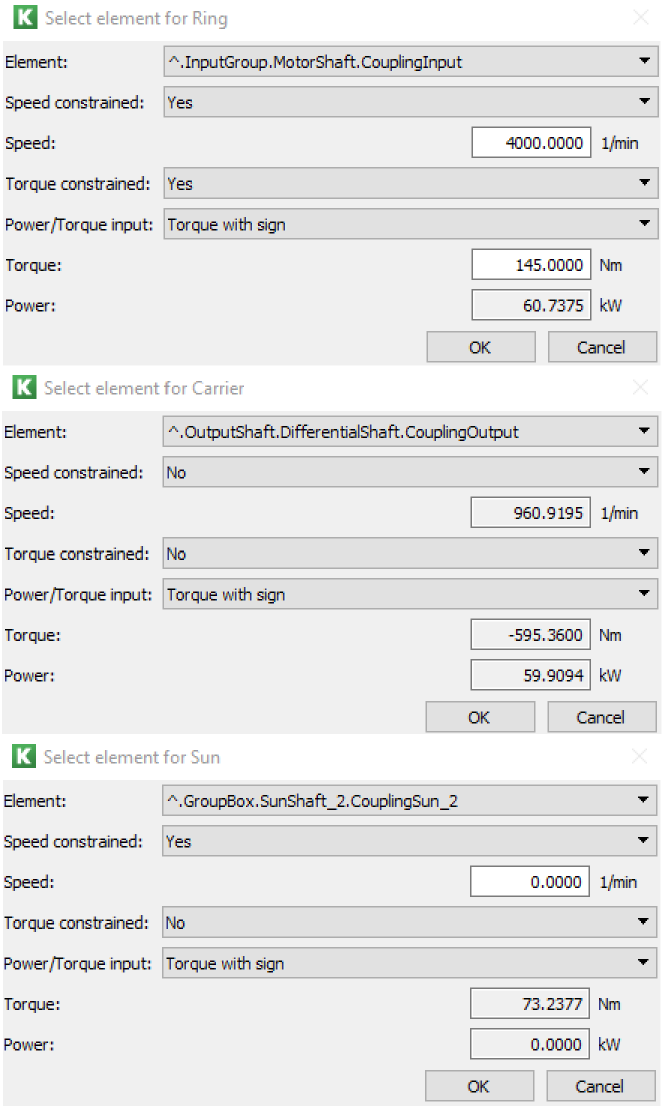

Fig. 4.10: *Settings of the boundary conditions for the second gear ratio configuration in the two-speed transmission*

In Figure 4.11 is reported the diagram to understand the imposed constraints and the resulting power flow of the entire transmission system.

4.3.1 Gear Sizing

In Paragraph 4.2, the requested values for the two gear ratios are presented. Recalling the table IV, the gear sizing of the two stages is carried out with the help of the information contained in the reference article [33].
The results the sizing of the two planetary units is resumed in Table VI. The selection of the geometrical parameters is accomplished with the tools *Rough sizing* and *Fine sizing* offered by KISSsoft. During the iterations, the values locked are the nominal ratio and its deviation, the module, the pressure and helix angle, while the number of teeth and the profile shifting factors are let free. Finally, the combination with the previously established number of teeth is preferred. The same procedure is applied to the sizing of all the gear pairs included the couple matching between motor shaft (M)/external ring(ER) and external carrier(EC)/differential shaft(D), whose result are shown in Table VII.

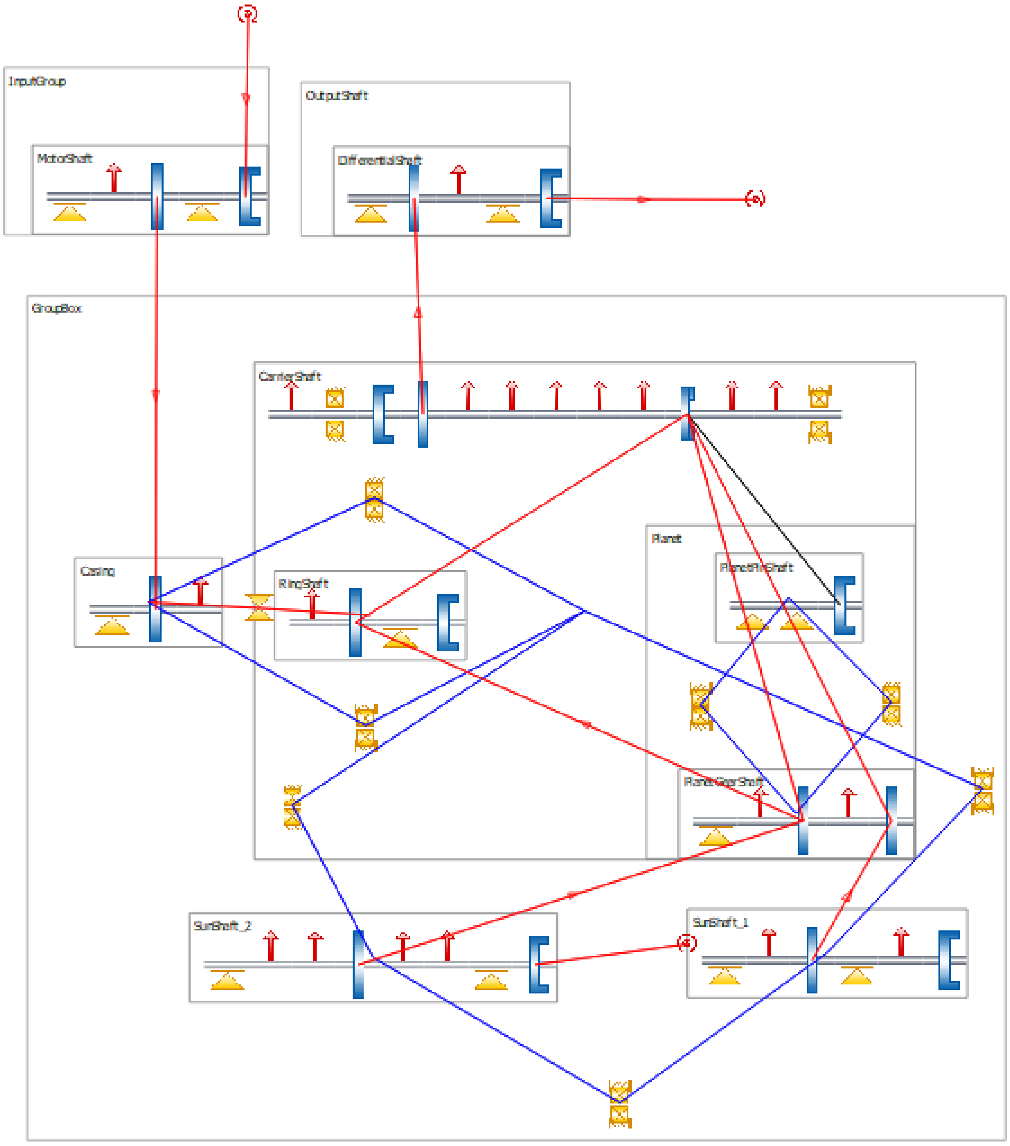

Fig. 4.11: *KISSsys diagram of the two-speed transmission*

Calculation item	**Sign**	**S1**	**P1**	**S2**	**P2**	**R**
Teeth	Z	30	12	12	30	72
Reduction ratio	i	2.042		1.166		
Helix angle	β	12 °				
Pressure angle (normal)	α_n	20 °				
Module (normal)	m_n	1.5 mm				
Facewidth	b	20.000 mm		28.5371 mm		
Shift profile factor (normal)	x_n^*	0.0000	0.8881	0.5361	0.3521	-1.2402

TABLE VI
Gears geometry of the planetary units in the two-speed transmission

Calculation item	Sign	M	ER	EC	D
Teeth	Z	38	116	77	126
Reduction ratio	i	3.0526		1.6364	
Helix angle	β	12 °			
Pressure angle (normal)	α_n	20 °			
Module (normal)	m_n	1.5 mm		1.0 mm	
Facewidth	b	20.000 mm			
Shift profile factor (normal)	x_n^*	0.4898	0.8613	0.3374	0.9458

TABLE VII

Gears geometry of the input and output units in the two-speed transmission

Eventually, in Table VIII are disclosed the material and lubrication choices which are identical for all the gear pairs of the model.

Material	18CrNiMo7-6
Lubrication oil	ISO-VG 220
Lubrication method	Oil bath lubrication

TABLE VIII

Material and lubrication information for the gear sizing

Considering the parameters introduced in the Table VI and Table VII, the definitive ratios are reported in the Table IX.

First overall gear ratio	$i_{1,def}$	10.199
Second overall gear ratio	$i_{2,def}$	5.825

TABLE IX

Definitive gear ratios for the two-speed transmission

For complete information, the motor efficiency map to check the operating point for the definitive second gear ratio is represented in Figure 4.14.

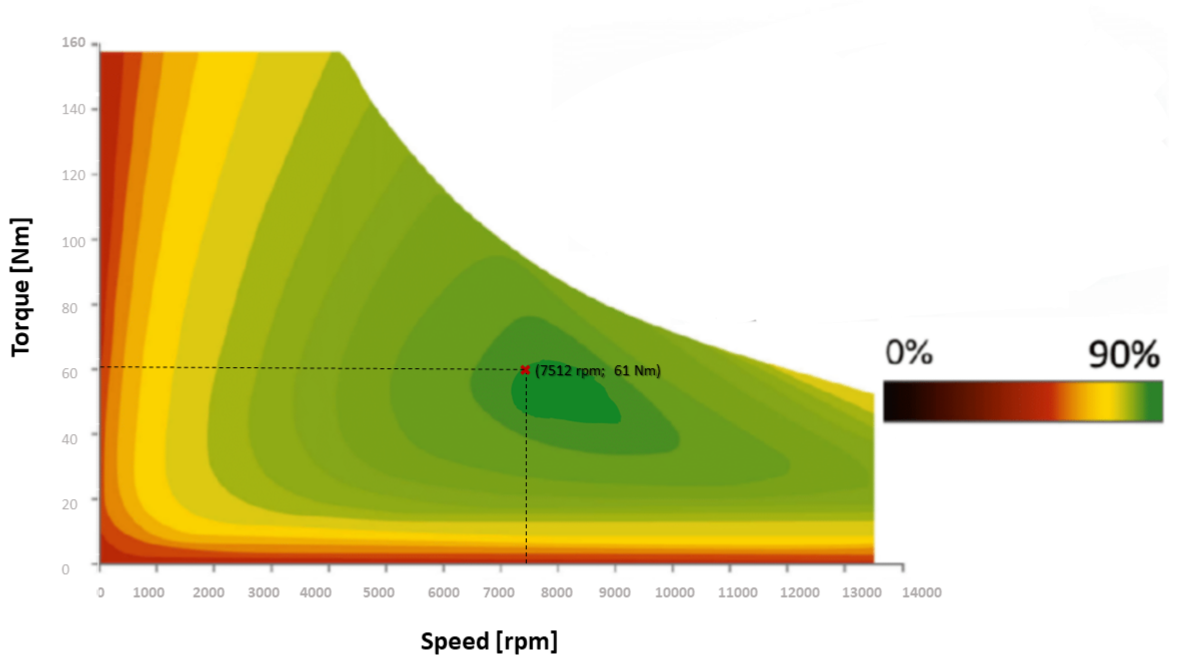

Fig. 4.14: *Motor efficiency map with the operating point for $i_{2,def}$ [34]*

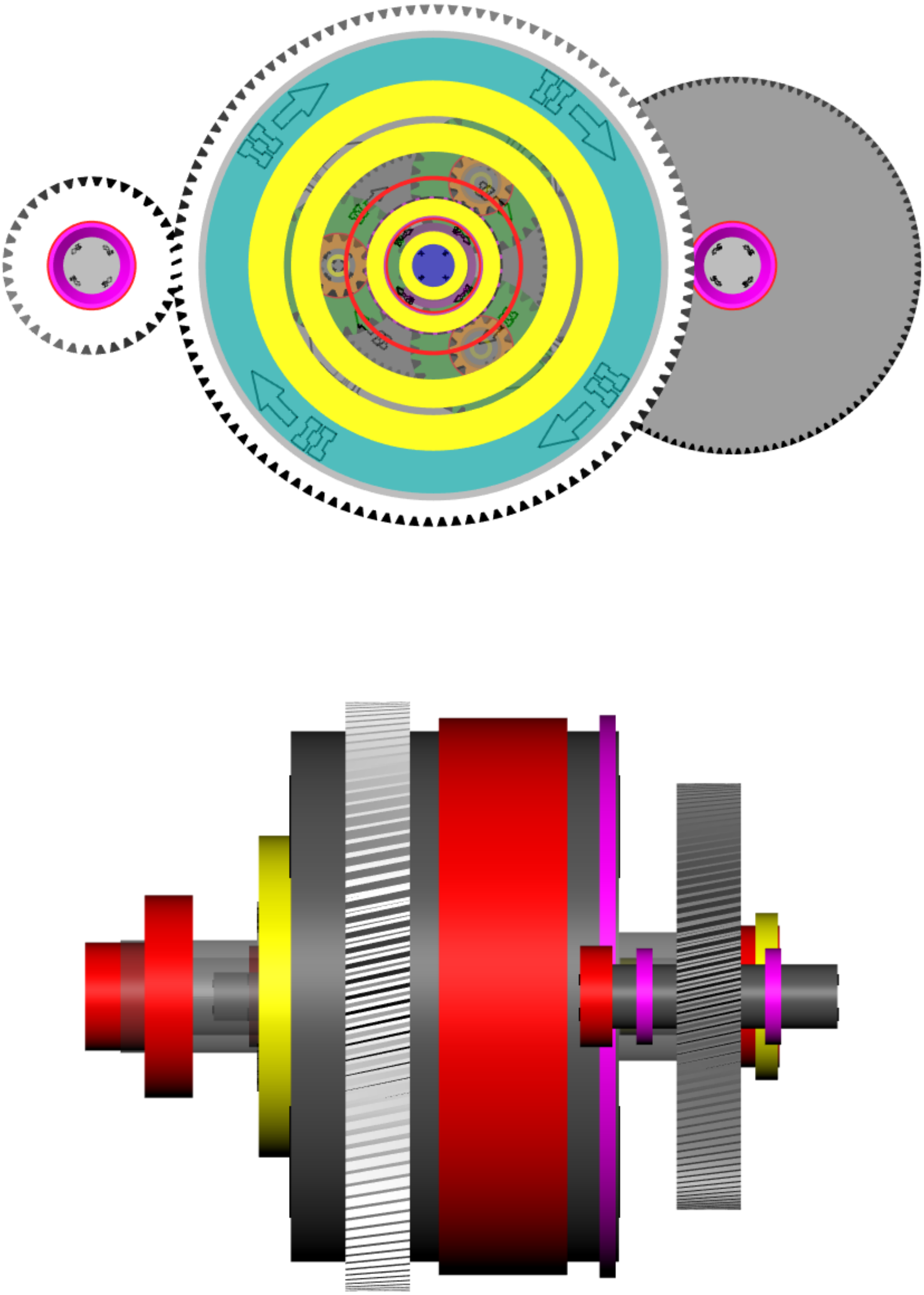

Fig. 4.12: *Frontal and left views of the two-speed model in KISSsys*

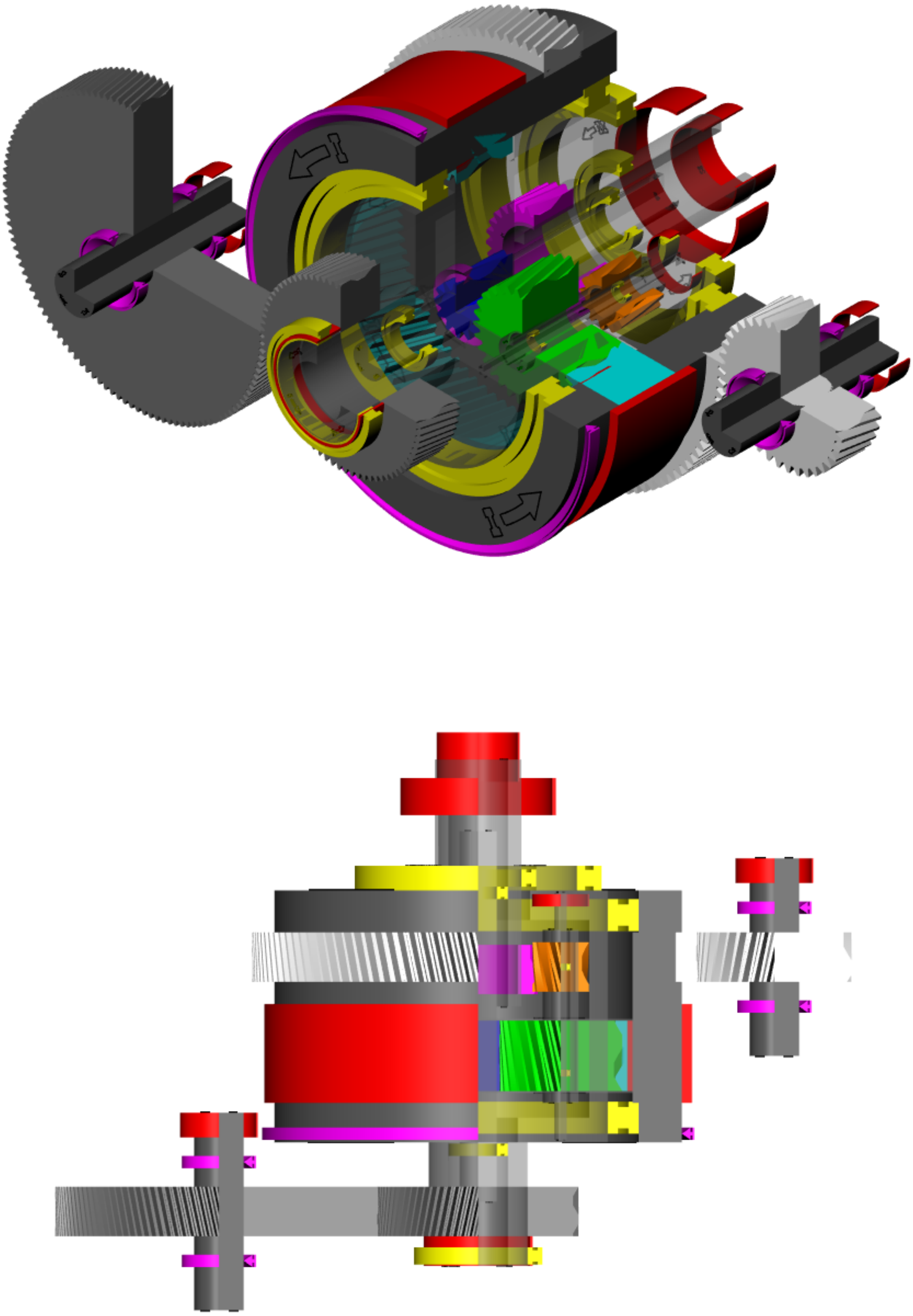

Fig. 4.13: *Front/left and top cut views of the two-speed model in KISSsys*

5 Efficiency of the gearbox

5.1 Efficiency computation methods

To compute the power losses and the effciency values, the software KISSsoft utilises the norm ISO TR 14179-1, while the analytical model proceeds with a slightly different approach. A brief description of the two alternatives follows to eventually justify any discrepancy during the comparison of the two results.

5.1.1 ISO TR 14179-1/2

The ISO/TR 14179 consists of two parts and the following descriptions are directly taken by the official published standard [35].
The ISO TR 14179-1 procedure is based on the calculation method presented in AGMA (American Gear Manufacturers Association) Technical Paper 96FTM9. The bearing losses are calculated from catalogue information supplied by bearing manufacturers, which in turn can be traced to the work of Palmgren. The gear windage and churning loss formulations originally appeared in work presented by Dudley, and have been modified to account for the effects of changes in lubricant viscosity and amount of gear submergence. The gear load losses are derived from the early investigators of rolling and sliding friction who approximated gear tooth action by means of disk testers. The coefficients in the load loss equation were then developed from a multiple parameter regression analysis of experimental data from a large population of tests in typical industrial gear drives. These gear drives were subjected to testing which varied operating conditions over a wide range. Operating condition parameters in the test matrix included speed, power, direction of rotation and amount of lubricant. The formulation has been verified by cross checking predicted results to experimental data for various gear drive configurations from several manufacturers.
The ISO/TR 14179-2 is based on a German proposal whereby the thermal equilibrium between power loss and dissipated heat is calculated. From this equilibrium, the expected gear oil sump temperature for a given transmitted power, as well as the maximum transmittable power for a given maximum oil sump temperature, can be calculated. For spray lubrication, it is also possible to calculate the amount of external cooling necessary for maintaining a given oil inlet temperature. The calculation is an iterative method.

$$\eta_{gearbox} = \frac{P_{input} - P_{loss}}{P_{input}} = \frac{P_{input} - (P_{loss,gears} + P_{loss,bearings})}{P_{input}} \tag{5.1}$$

5.1.2 Analytical model

Power losses in gears are mainly caused by four types of friction, namely: gear mesh sliding and rolling friction, oil churning and windage losses [36]. In addition, bearings' losses are computed according the widely accepted SKF method, while losses caused by clutches, synchronizers, actuators and oil pump are computed following the approach explained in detail in [36].
In the end, the total power loss is defined as:

$$P_{loss} = \sum P_b + \sum P_g + \sum P_{cl} + \sum P_{syn} + \sum P_a + Pp$$

In this phase of the project, the one-speed transmission does not require the introduction of the mechanical components considered in the last paragraph. As a consequence, the power losses due to clutches, synchronizers, actuators and oil pump are null, likewise those related to the sealing losses in the gears.

5.2 Efficiency maps

5.2.1 One-speed transmission

The one-speed model's efficiency map obtained in KISSsoft is shown in Figure 5.1a). With the help of the values displayed on the Table II, it is possible to appreciate that the results perfectly fit the typical

range of this kind of transmission showed on the Table I.
The difference between the manual and automatic gearbox is ascribed to the computation of the only bearings and gears losses in the first case [28]. This observation justifies the lower efficiency of the automatic gearbox where the clutches losses and the shifting system components produce further losses. Dealing with a single speed transmission, the presence of any shifting elements (synchronizers, clutches, oil pumps) allows to prefer the manual transmission range to validate the efficiency range.
Moreover, the result is coherent to what expected according to the definitions introduced in the Section 5.1.2. Indeed, the efficiency reduces along the x-axis because the speed-dependent gear losses rise proportionally to the increase of the speed of the motor shaft. Along the y-axis, an other characteristic behaviour can be identified: the efficiency is lower with smaller torque values because the bearing viscous losses are bigger compared to the input power; at lowest speed the total efficiency is higher because the bearing viscous losses are negligible [27].
In Figure 5.2 the behaviour of the total bearing losses is represented.

Transmission type	**Efficiency range**
Manual gearbox	92 - 97 %
Automatic gearbox	90 – 95 %
Mechanical CVT	87 – 93 %
Hydrostatic CVT	80 – 86 %

TABLE I
A rough survey of typical overall efficiencies of vehicle gearboxes under full load [37]

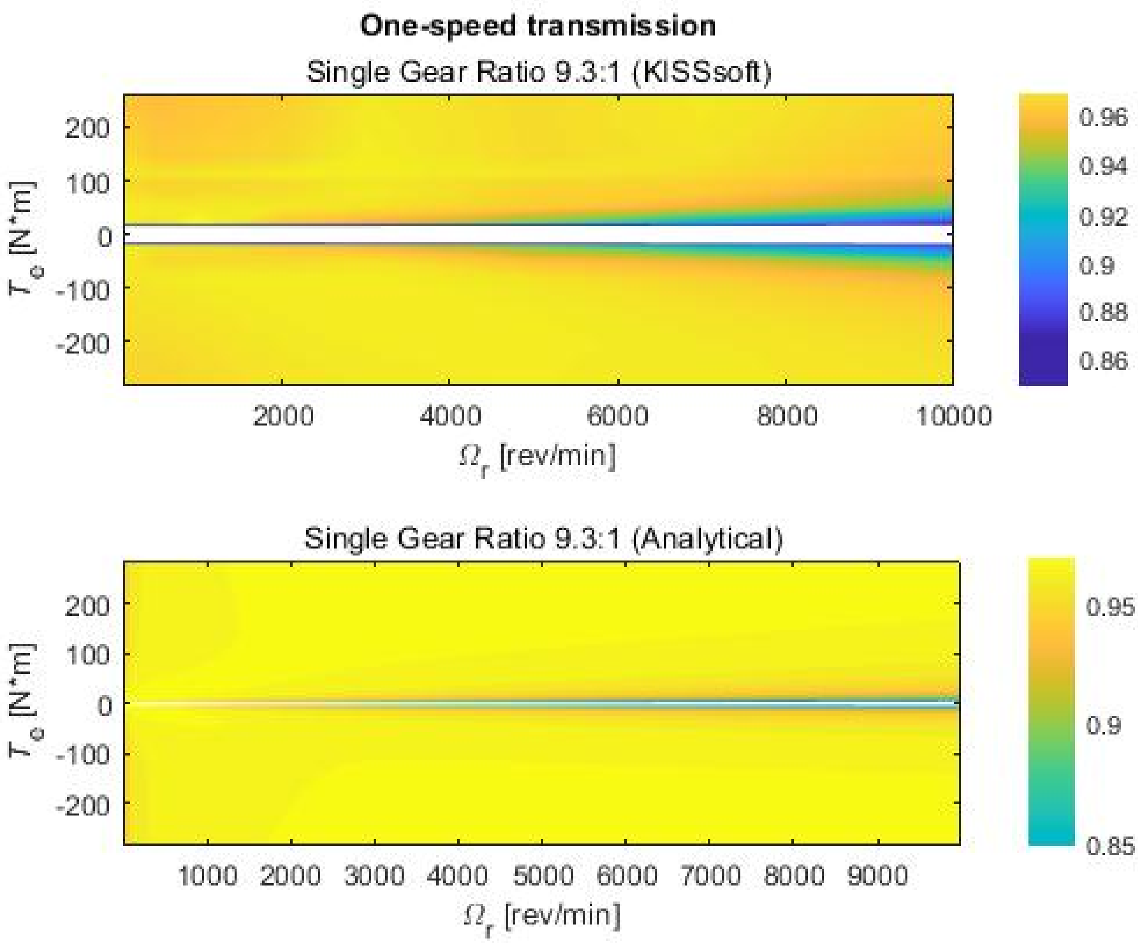

Fig. 5.1: *Efficiency maps of one-speed transmission*

Torque [Nm]	**Speed** [rpm]	Gear losses P_{VZ0} [W]	Gear losses P_{VZ}[W]	Bearing losses P_{VL}+P_{VL0} [W]	Total efficiency $\eta_{gearbox}$ [%]
20	5000	190.1	153.8	301	93.84
80	100	0	24.3	2.7	96.78
110	5000	190.1	988.4	436	97.2
145	2000	12.2	817.9	149.3	96.77
250	3000	41.1	1764.4	372.2	97.23

TABLE II

Losses and efficiency values in some points of the one-speed transmission's map

The two efficiency maps obtained with the different software are shown in the Figures 5.1. It is worth to notice that the shape of the two maps is consistent and comparable.
In the table III are shown the values of four points in the maps to compare the results numerically.
The faint differences can be ascribed to different factors, for example:

– The number of data to build the first efficiency map 5.1a) is higher than in the second one 5.1b), so the image is smoother in the first case, while a poorer quality characterizes the KISSsoft map.

– The profile shifting factor in the KISSsoft model cannot be completely eliminated, while the Analytical model does not take in account this parameter. So, the computed gear losses are not exactly overlapped.

– The bearing losses do not coincide in the two methods as shown in the Table III. Despite that, this kind of loss is negligible compared to the gear losses.

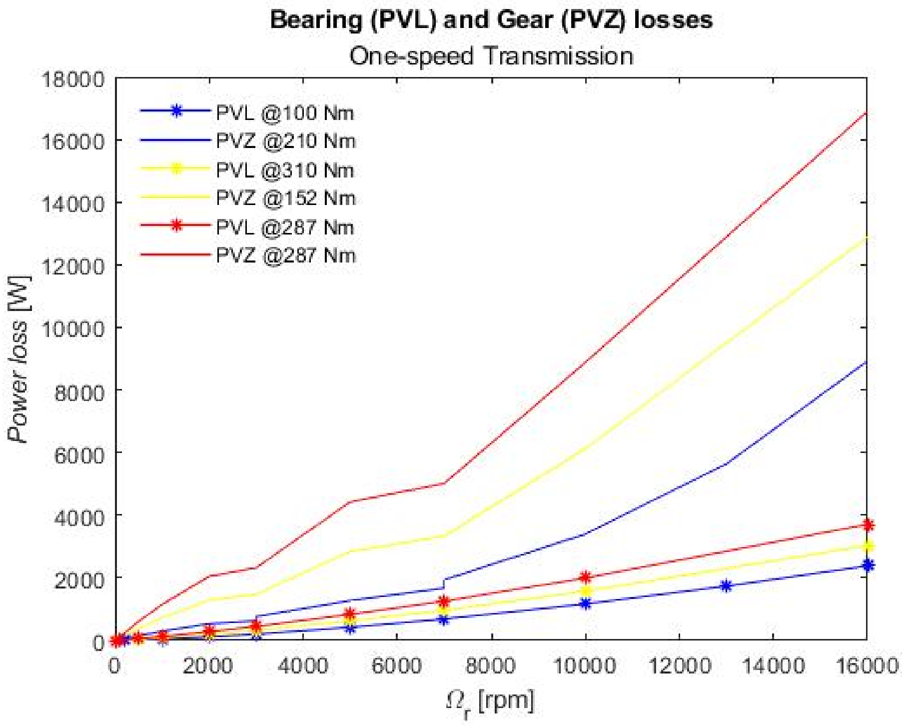

Fig. 5.2: *Total bearing losses P_{VL}+P_{VL0} compared to the total gear losses P_{VZ}+P_{VZ0} for the one-speed transmission*

5.2.2 Two-speed transmission

In Figure 5.3 are displayed the efficiency maps of the first and second gear ratios elaborated with the software KISSsoft. While in the Table IV are presented the loss and efficiency values for four operating points.

		Gear losses P_{VZ0}+P_{VZ}[W]		Bearing losses P_{VL}+P_{VL0} [W]		Total efficiency $\eta_{gearbox}$ [%]	
Torque [Nm]	**Speed [rpm]**	KISSsoft	Analytical	KISSsoft	Analytical	KISSsoft	Analytical
50	1000	110.4	131.6	31.7	0.26	97.29	97.48
50	7000	1036.8	1090.7	575.9	17.63	95.6	96.98
210	1000	722.3	629.6	88.2	0.4	96.31	97.14
210	7000	3334.9	3969.8	965.3	20.48	97.21	97.41

TABLE III

Losses comparison for four points in one-speed transmission efficiency

Based on the Table I, the results obtained are acceptable for the typical range. Indeed, clutch losses are the main elements contributing to the increased losses in the automatic gearbox which justify a lower efficiency range with respect to the manual one. Although the gearboxes considered for the project development are typically implemented with an automatic technology, for the design in KISSsoft any clutches component is included so that it is preferable to compare the maps' values with the manual efficiency range.

		Gear losses P_{VZ0}+P_{VZ}[W]		Bearing losses P_{VL}+P_{VL0} [W]		Total efficiency $\eta_{gearbox}$ [%]	
Torque [Nm]	**Speed [rpm]**	i_1	i_2	i_1	i_2	i_1	i_2
31.6	1314	123.2	127.9	1.6	3.6	97.04	96.98
31.6	7475	1868.9	662.3	28.7	34.7	92.33	97.18
152.3	1314	2585.9	400.2	156.8	22.2	95.27	97.98
152.3	7475	5109.4	1817.5	369.3	141.3	96.05	98.36

TABLE IV

Losses for four operating points in the two-speed transmission efficiency map

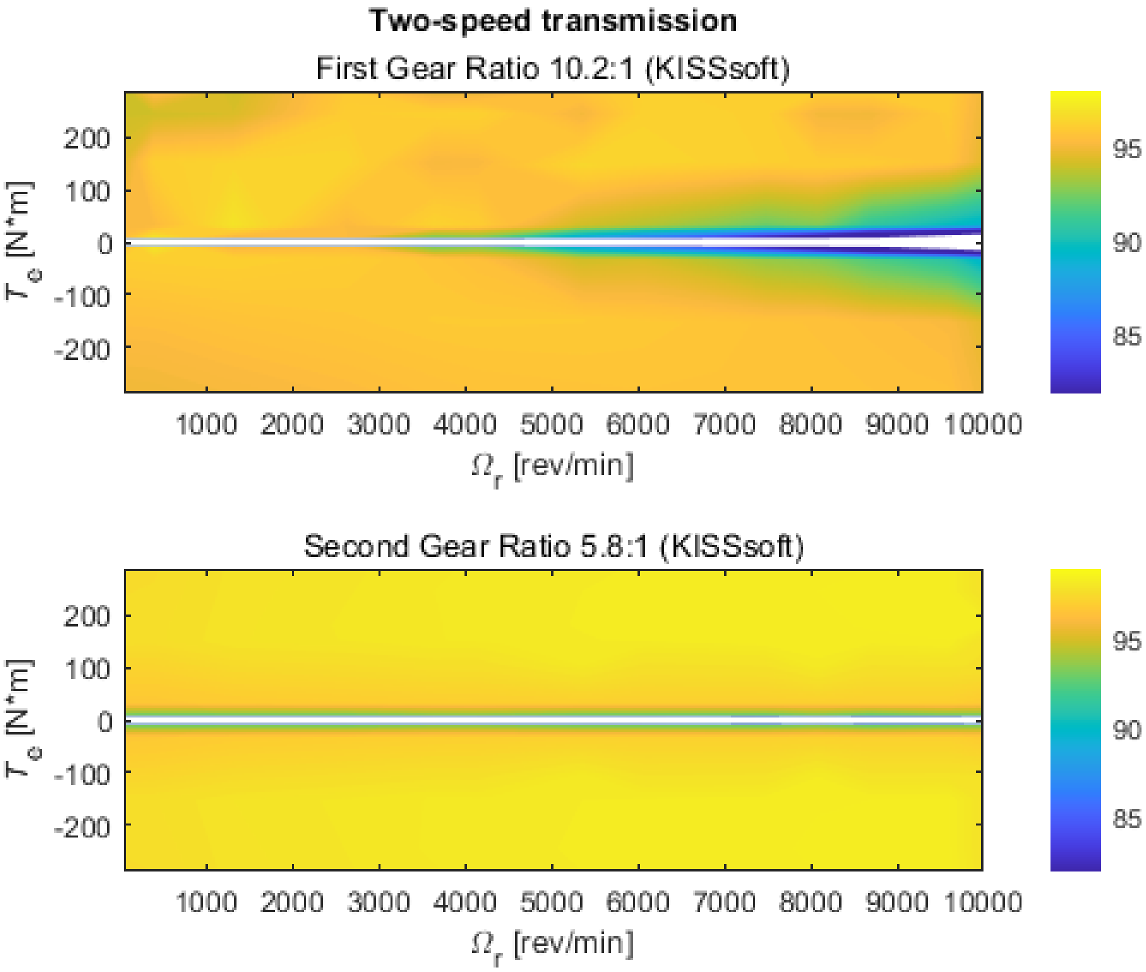

Fig. 5.3: *Efficiency maps of the two-speed model in KISSsoft*

The observations about the maps' trend presented in the Section 5.2.1 are still valid for these maps to approve the achieved outcomes. Fixing the torque value, the efficiency along the x-axis is expected to reduce because the speed-dependent gears and bearing losses increase. On the other side, fixing the speed value, the efficiency is higher at lower torques because the the bearing viscous losses are bigger compared to the input power.
It worth to notice that the smallest gear ratio has an overall efficiency higher than the largest gear ratio. To explain this behaviour, in Figure 5.4 are shown both the rends of the speed- and load- dependent gear losses. When the transmission is working with the largest gear ratio, it is more sensible to the speed-dependent gear losses PVZ_0. because the first ratio multiplies the speed motor shaft by a larger factor than the second one. On the other hand, the load dependent losses are still higher in the first configuration but in a less accentuate way.
As a consequence, it is evident that largest gear ratios imply lower efficiency accordingly to the recommendation of choosing the lowest possible gear ratio within the suitable range for the vehicle performance. [22]. A further observation deals with the behaviour of the transmission in the regenerative breaking configuration. Given that this feature is implemented in mostly EVs, the transmission has to work efficiently also when the electric machine works as a generator. For this reason, the maps in Figure 5.3 are built also for the negative torques.
In KISSsoft, the backward mode should be implemented by changing the boundary conditions showed in the Figure 4.10. In particular, the input power should be set on the differential gear shaft, while the motor output shaft should be free form any speed and torque constraints.
In the Table V, this procedure is indicated as a 'backward mode', while the 'forward mode' is the conventional one when the machine works as motor where the input is set on the Ring group (motor shaft + external ring) and the output one the Carrier group (external carrier + differential shaft). The results on the table prove that the backward outcomes do not differ from those computed in the forward mode

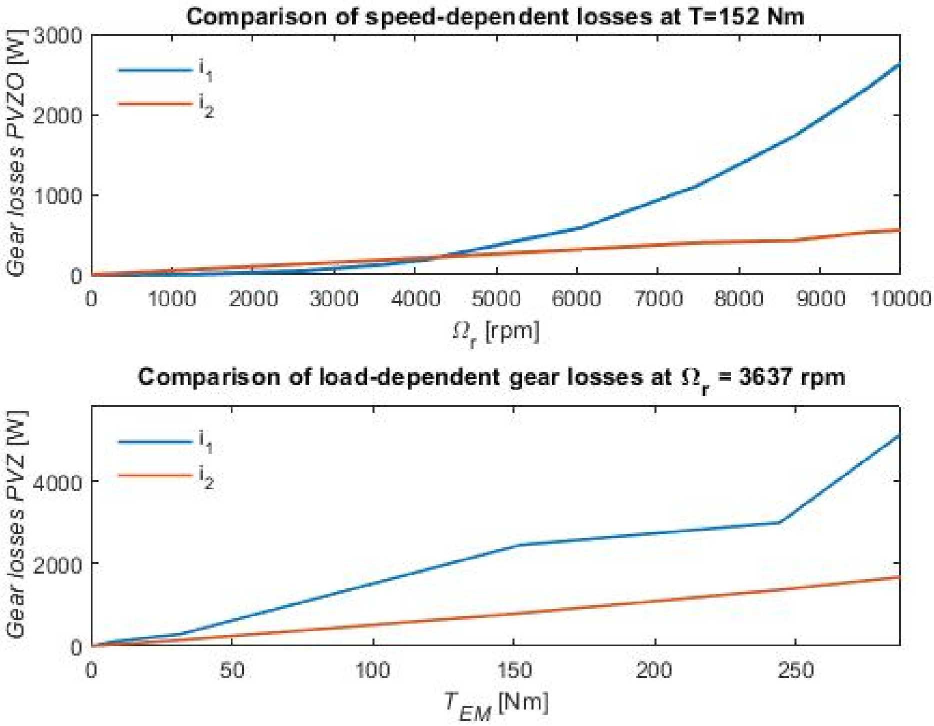

Fig. 5.4: *Comparison of the speed- and load-dependent gear loss for the two gear ratios*

with the corresponding negative torque.
Despite the results of the two modes for the regenerative breaking simulation are identical, it is worth to notice that in the same mode, torques with the opposite sign give slightly different results. Practically, it means that the efficiency of the transmission change when the electric machine works as a motor or a generator and it is lower in the first case.

	Ring group		**Carrier group**					
Mode	Torque [Nm]	Speed [rpm]	Torque [Nm]	Speed [rpm]	Gear losses P_{VZ0}[W]	Gear losses P_{VZ}[W]	Bearing losses $P_{VL}+P_{VL0}$ [W]	Tot Eff η_{GB} [%]
Forward	145	4000	-595.4	960.9	204.3	814.3	67.2	98.21
Forward	-145	4000	611.8	960.9	204.3	824.6	61.5	98.23
Backward	-145	4000	611.8	960.9	204.3	824.6	61.5	98.23
Forward	-169.5	6869	714.8	1650.1	362.7	1512.8	134.5	98.37
Backward	-169.5	6869	714.8	1650.1	362.7	1512.8	134.5	98.37
Forward	-31.6	42.4	10.2	135.2	2.1	2.4	0.1	96.81
Backward	-31.6	42.4	10.2	135.2	2.1	2.4	0.1	96.81

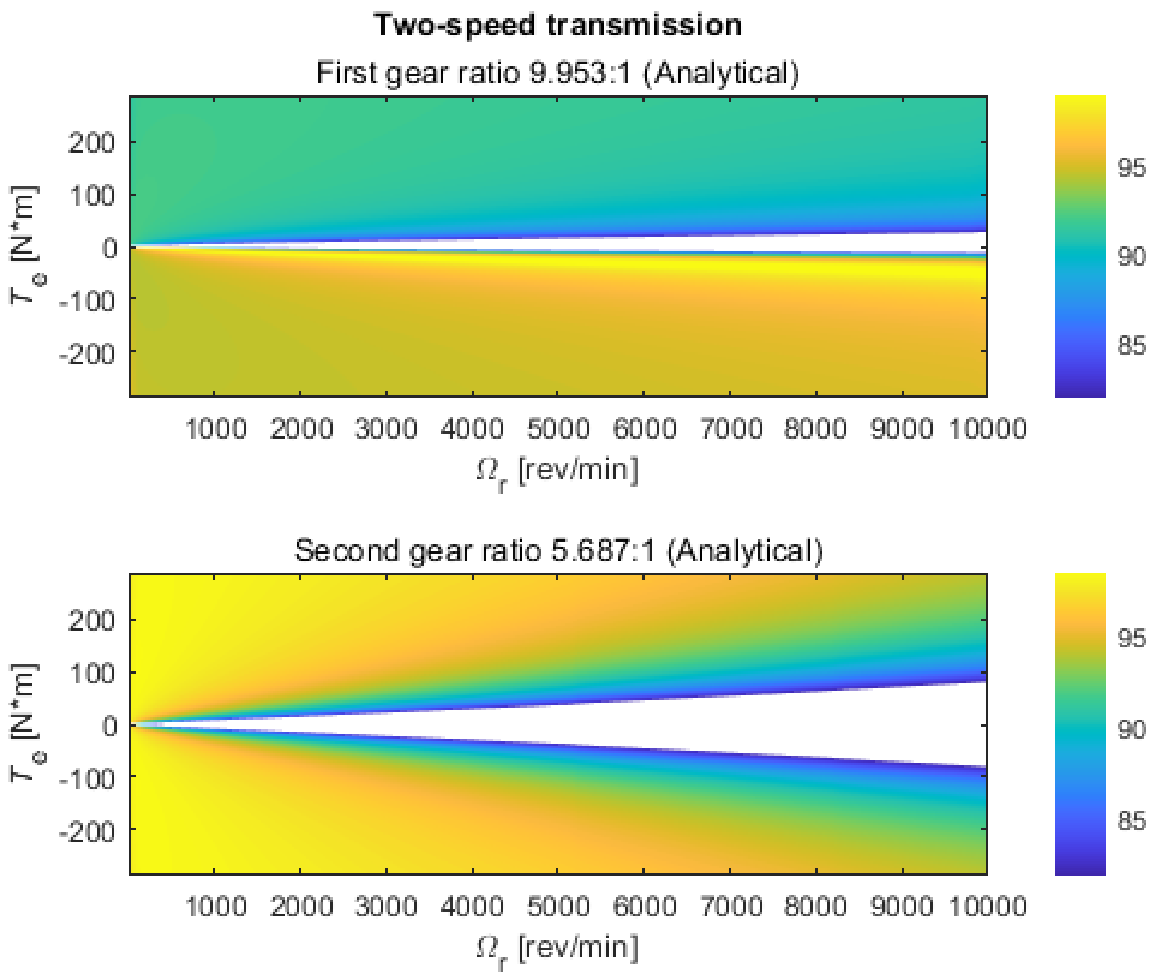

Fig. 5.5: *Efficiency maps of the two-speed model with the analytical approach*

In the Figure 5.5 are displayed the two efficiency maps of the analytical model for the losses' estimation. The maps immediately appear very different from the KISSsoft counterpart in many aspects: the efficiency values are significantly lower, the difference between the backward and forward modes is too accentuated, the overall trend is unexpected considering the previous models, included the one-speed transmission).

Moreover, the results of the same model applied for the efficiency estimation of a more commercial two-speed DCT are showed in the article [38]. In this case, the obtained maps are more similar to those analysed until now and, for this reasons, considered reliable instead of those showed in the Figure 5.5.

Focusing on the evident gap, some points from the different maps are compared on the Table **??**. Despite the fact that the analytical maps

Despite the fact that the the analytical method is not validate for this particular model, it is important to considerate the divergences to examine some issues present in the KISSsoft model. In particular, the Figure 5.6 underlines that the total efficiency of the latter are constantly above the values of the analytical model and, in many points, these values overcome the typical efficiency range of the transmission presented in the Table I. The explanation to this evident gap can be consistently justified by the following reasons :

- as in the one-speed model, the profile shifting factor is always null in the analytical model while in KISSsoft this modification of the profile is not preventable;
- the KISSsoft model is not able to compute the ring gear losses in any configuration or mode;
- the choice of the bearings is not reliable since it was not relevant for the realization of the two-speed transmission and, for the second gear ratio configuration in particular, the software does not take in account the losses of few bearings. In any case, the trend of the bearings losses is illustrated in the Figure 5.7 to prove that they slightly affect the final result.

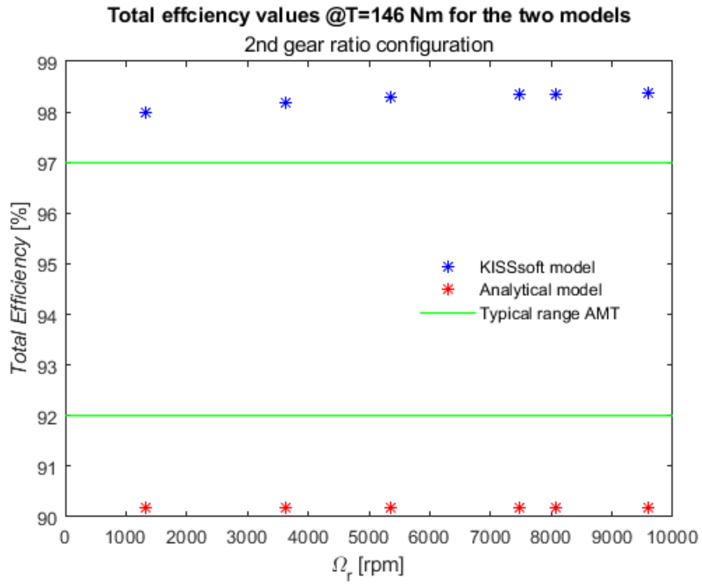

Fig. 5.6: *Comparison of few total efficiency values in the second gear configuration*

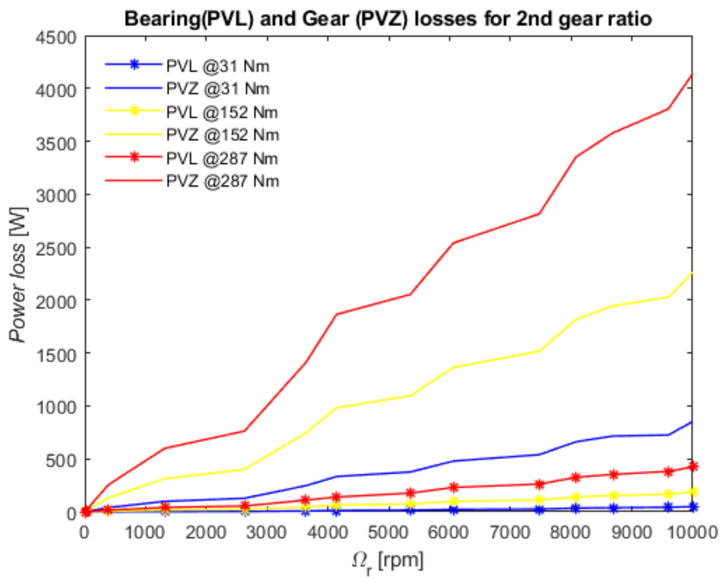

Fig. 5.7: *Bearing losses of the two-speed transmission for the second gear ratio*

Conclusions

The project was originally structured in two parts: the design of the two transmission models and the development of the efficiency maps for the comparison of the analytical model implemented in Matlab.
With the help of the available information of the vehicle Volkswagen ID.3 Pro Performance, the dual stage one-speed transmission's design is successfully completed and the efficiency estimation followed the expectations. The model designed in KISSsoft has reliable features and the obtained efficiency map is consistent and comparable with the one given by the analytical method. Some aspects superficially affecting the efficiency values, likewise the rough selection of the bearings, are listed to justify and approve the final outcome
Coherently to what is done in the greatest part of the studies concerning this topic, the design of the two-speed gearbox is based on the maintenance of the same motor characteristics. In this case, the choice fell on the design of a stepped planetary gearset layout presented in the Section 4.1. The computation of the two gear ratios followed several steps since the established range by the described procedure was not suitable for the proposed solution: indeed, a relevant gap between the two gear ratios implicated important geometrical differences between the elements of the same stage. As a consequence, a suitable alternative was verified on the motor efficiency map of the vehicle to reduce the gap and maintain the stepped configuration. Successively, the transmission's efficiency map in the first and second configuration was built and validate. In this case, the results obtained did not perfectly fit the typical efficiency range due to some limits of the model developed in KISSsoft such us the lack of the ring gear and few bearings losses computation. Moreover, the outcome did not agree with the results given by the analytical model since the latter exhibits some inconsistencies. Regardless, the application of the same analytical method to a commercial DCT gave reliable results which prove the correctness of the shape and the values of the KISSsoft map.
Based on the aforementioned considerations, the models designed on KISSsoft together with the corresponding efficiency maps can be considered accurate and reliable. Furthermore, some proposal on how to improve the obtained results are given. Firstly, it would be appreciable to correct the losses computation of some components, in the second model especially concerning the ring gear losses. Moreover, despite it was demonstrate that the choice of the bearings does not consistently affect the overall efficiency, additional attention could be given to the bearing selection for a more accurate model.

List of symbols

AT = Automatic Transmission
AMT = Automatic Mechanical Transmission
BEV = Battery-Electric Vehicle
BLDC = Brushless DC Motors
CVT = Continuously Variable Transmission
DCT = Dual Clutch Transmission
ECU = Electronic Control Unit
EM = Electric Motor
EMS = Energy Management System
EREV = Extended-Range Electric Vehicle
EV = Electric Vehicle
HEV = Hybrid-Electric Vehicle
FCHEV = Fuel Cell Hybrid Electric Vehicle
IAMT = Inverse Automatic Mechanical Transmission
ICE = Internal Combustion Engine
PEV = Pure Electric Vehicle
PGT = Planetary Gear Transmission
PMSM = Permanent Magnet Synchronous Motor
SOC = State Of Charge
SRM = Switched Reluctance Motors
TCU = Transmission Control Unit
UMT = Uninterrupted Mechanical Transmission
VECU = Vehicle Electronic Control Unit

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