Design and evaluation of powertrain architectures for battery electric vehicles

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Master of Science Thesis

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Abstract

The continued increase in greenhouse emissions has recently raised serious environmental, economic and social concerns, giving electric propulsion an important role in future road transportation. The greater flexibility of electric vehicles versus conventional ones in terms of powertrain configurations has promoted the development of highly differentiated architecture topologies.

This work will investigate two electric powertrains with respect to energy consumption and efficiency: the standard single-motor architecture, derived from conventional internal combustion engine vehicles, equipped with a high-speed electric motor and a mechanical reduction system and the novel in-wheel topology with direct drives. The potential benefits of a two-speed transmission to improve performance and driving range of battery electric vehicles is also studied.

The initial part of the thesis is dedicated to modelling the components responsible of most of the losses in electric drivetrains. Vehicle simulations are usually executed by complex software which lack of traceability and transparency; in this project models are derived directly from mathematical formulas, in order to allow the reader to easily track and understand the flow of energy through a drivetrain and the distribution of losses.

In the second part of this work the acceleration performance, energy consumption and efficiency of the vehicle concepts are determined and analyzed over various drive cycles. Particular focus is given to the influence of driving style on the loss sources.
Acknowledgements

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# Contents

**Abstract**

**Acknowledgements**

## 1 Introduction 1

1.1 Background .................................................. 1  
1.2 Thesis objective ........................................... 2  
1.3 Methodology and thesis outline ......................... 2  
1.4 Contributions .............................................. 3  

## 2 Literature review 5

2.1 Drivetrain architectures .................................... 5  
2.2 Electric motors ............................................. 6  
2.3 Transmission ............................................... 8  
2.4 In-wheel drives ............................................ 10  
2.5 Conclusion .................................................. 12  

## 3 Modeling 15

3.1 Powertrain architectures ................................... 15  
3.1.1 Single-motor topology ................................ 16  
3.1.2 In-wheel topology ..................................... 16  
3.2 Glider Model ................................................. 18  
3.3 Transmission Model ........................................ 20  
3.3.1 Introduction ........................................... 20  
3.3.2 Design Principles .................................... 20  
3.3.3 Clutch .................................................... 23  
3.3.4 Gear pair ............................................... 23  
3.3.5 Bearings ................................................ 30  
3.3.6 Lubrication .............................................. 34  
3.4 Electrical drives ........................................... 35  
3.4.1 Induction machine operation ....................... 35  
3.4.2 PM synchronous machine operation ............... 37  
3.4.3 Power losses .......................................... 39
Chapter 1

Introduction

This chapter serves as introduction to the work of this research. Section 1.1 describes the role and development of electric vehicle in recent mobility. Section 1.2 illustrates the objectives of this thesis work. Section 1.3 describes the approach taken and the structure of the survey. Section 1.4 gives the main contributions achieved in this research.

1.1 Background

Vehicles constitute a crucial component of road transportation and they have become an integral part of modern life. The recent rise of serious environmental, economic and social concerns related to the increase of emissions of greenhouse gases is forcing the automotive industry to rethink the design of future cars.

The electrification represents the most promising solution to convert sustainable energy into drive energy and to satisfy the demands for higher fuel economy and lower emissions. The high efficiency and the instantaneous torque make electric motors a clean and powerful alternative to internal combustion engines (ICEs). Low noise and low maintenance required are also excellent qualities for traction application.

The large degree of flexibility of electric drive systems allows different level of electrification and numerous possible configurations in electric powertrains. While hybrid vehicles have already become a mass-market product, pure or battery electric vehicles (BEVs) are facing new complex challenges which are greatly restraining their diffusion. The main drawbacks are related to the energy storage: the low energy and power densities of current battery technology can’t provide a driving range comparable to conventional vehicles at a reasonable weight and price. The battery also dominates the overall cost break-down of BEVs and it can come up to almost 60% of the total production costs. A high initial cost introduces a market barrier which doesn’t make them attractive to potential customers interested on money savings as well as pollution. Large investments from governments and industry are promoting a rapid growth of the technology related to electric vehicle (EV) in the areas of lightweight materials, high-energy-density storage devices and electric propulsion motor drives. Hence, there is the hope for BEVs to take up in the near future the commercial competition with fuel energized cars on a large
scale.

1.2 Thesis objective

This research is driven by the ever growing requirement of the automotive industry to reduce toxic emissions. BEVs have the potential to replace conventional ICE vehicles, but their short driving range often induces a feeling of "range anxiety" in the driver which greatly limits their diffusion.

Researchers are focusing on overcoming the technical barriers caused by the present battery technology: innovative and differentiated powertrain architectures which promise lower energy consumption are being constantly investigated. Multi-speed transmissions are often adopted to improve the efficiency or the energy recovery capability of EVs, but the theoretical benefits are still controversial and not always clear.

Most of the studies which evaluate the performance of electric vehicles have dealt with complex simulation tools but without mathematical analysis and therefore they are often devoid of explicit correlations between vehicle dynamics and the power loss sources.

The aim of this thesis is to implement a transparent and traceable model of an electric vehicle and investigate the effects on energy consumption that different drivetrain designs have. The specific objectives of this work are defined below:

- Develop an electric vehicle model based on mathematical equations which can predict the power losses of the main mechanical and electrical systems
- Compare the energy consumption over driving cycles for two transmission layouts: a fixed reduction gear and a two-speed transmission
- Compare the energy consumption over driving cycles for two powertrain architectures: the single-motor configuration derived from conventional vehicles and the novel in-wheel drivetrain
- Provide an illustrative and detailed picture of the energy flow through an electric powertrain

This work was conducted using Matlab/Simulink environment as it has high flexibility to implement the models and it affords the ability to handle all the parameters and variables.

1.3 Methodology and thesis outline

This thesis consists of five chapters, including the current first chapter which contains an explanation into the motivation behind this research, the objectives it aims for and the methodology used to reach them.
CHAPTER 1. INTRODUCTION

In Chapter 2 an extensive review of the current literature pertaining to this field of research is carried out. Different powertrain architectures of BEVs are presented. An overview of the motor technologies most suited for tractive applications and their design choices is given. The effects of mechanical transmissions on performance and efficiency of electric vehicles are discussed. Through reviewing these studies, the author concludes the areas which could most benefit from further research.

Chapter 3 describes the vehicle models implemented. The behaviour of the main electrical and mechanical systems are derived from the characteristic governing equations. This section aims to perform an accurate calculation of the power losses that occur in each component.

Chapter 4 illustrates the performance requirements of a small BEV designed to be used mainly in urban areas and traffic conditions. Based on this specifications, the sizing of the powertrain components is carried out.

In Chapter 5 the energy consumption of the vehicles is evaluated during the simulation of standard driving cycles. First, the theoretical performance benefits of a two-speed transmission over a fixed reduction gear are verified. After this, the classic single-motor powertrain architecture is compared with the novel in-wheel configuration.

In Chapter 6 final considerations are drawn based on the results of the simulations. The main limitations of the models implemented are remarked and the potential consequences are speculated. The main contributions and recommendations for future work are discussed

1.4 Contributions

The purpose of this research is to quantify the potential benefits that the powertrain configuration has on energy consumption of battery electric vehicles. The main contributions are summarized below:

- An analytical model of an electric vehicle is proposed, based on mathematical expressions, which gives an in-depth look at energy flow throughout an electric powertrain and illustrates the interactions between its components

- Power losses in the main electrical and mechanical systems are calculated with empirical formulas during dynamic conditions, to accurately predict energy consumption and efficiency and examine the influence of vehicle dynamics on the loss sources

- Two diametrically opposed electric powertrain layouts are compared over differing driving scenarios: the standard ICE derived architecture, which dominates the BEV market, and the innovative in-wheel powertrain, considered very promising by current research yet without practical application in mass produced cars

- The uncertain efficiency and performance benefits of a two-speed transmission versus a fixed reduction gear on BEVs are verified over differing driving scenarios
Chapter 2

Literature review

This chapter is dedicated to a literature review of the main technologies pertaining to this field of research. Section 1 illustrates the typical vehicle configurations that exist for pure electric propulsion. Section 2 examines the features of the electric drives mostly adopted for traction applications. In section 3 the transmission systems specifically designed for electric vehicles are discussed. Section 4 focuses on the current status of the in-wheel motor technology. The aim of this review is to examine the most interesting areas of research and investigate for possible gaps in knowledge which rise questions for further analyses.

2.1 Drivetrain architectures

The large number of components involved in automotive powertrains and the high degree of flexibility compared to conventional ICE vehicles promoted the development of several architectures greatly diversified for battery electric vehicles.

Guang et al. propose a comprehensive review of the powertrain architectures of electrified vehicles, including both pure electric and hybrid vehicles, and illustrate the three fundamental solutions. The simplest layout consists of a single electric motor.
connected to the differential through a single-speed reduction gear, illustrated in Figure 2.1 (b) and (c). This architecture is the favourite among manufacturers; the absence of a gearbox is cost-effective, beneficial for installation space and it decreases driveline losses.

A similar configuration employs a multi-speed gearbox, Figure 2.1 (a). The additional gear ratios can extend the range of available torque and speed at the wheels, with significant improvements of vehicle performance and driving experience. The electric motor could be even downsized, with a reduction of costs and weight. The second major advantage from adopting a multiple speed transmission is that the drivetrain can theoretically operate in a higher efficiency region for a larger portion of a driving cycle. However, the effective implications on fuel economy are still controversial and they are analysed more in detail in Section 3.

The last powertrain architecture discussed by Gang et al [2] involves two or more driving units. This group covers many possible layouts, which differ for the number of motors, their location and the connection to the wheels. Innovative systems capable to combine more power sources are being studied: electric motors can drive independently front and rear axle or can be coupled to the same differential through a planetary gear set to generate different driving operation modes.

Recently a similar architecture is becoming very attractive, Figure 2.1 (e) and (f), where motors are installed inside the wheel with or without the presence of a fixed gearing.

2.2 Electric motors

Electrical machines and drives are a key component for electric vehicles since they determine vehicle performance and efficiency. Recent research describes the induction machine, the switched reluctance machine and the permanent magnet synchronous machine as the best suited for EVs.

![Figure 2.2: Desired torque and power curves for traction applications](image_url)
Zhu Z. Q. et al. [4] examine the basic operational and design features employed to achieve the typical torque/power-speed characteristic required for traction applications, depicted in Figure 2.2, i.e., high torque at low speed and constant power over a wide speed range. Permanent magnet (PM) brushless machines, especially the permanent magnet synchronous machine (PMSM), are currently the most attractive motor drives for electric vehicle propulsion for their excellent performance. They are generally classified as being either sinusoidal or trapezoidal back-EMF machines. Different topologies are illustrated, which differ for the location and the shape of the magnet material. Rotor permanent magnet machines are described as the most used configuration in automotive for the high power density and the wide constant-power operation range. Particular attention is given to the demagnetization withstand capability of machines. Magnets are extremely vulnerable to partial irreversible demagnetization at high operating temperature, a typical scenario in traction applications considering the elevated angular velocities. An effective thermal management and flux barriers are useful methods to cope with this problem. Despite the extensive use of these motors, there is a great concern about the availability of rare earth-based magnets and their increasing price, responsible for manufacturing costs often unacceptable for the automotive market. Recently, companies and researchers are investigating other technologies that do not use rare magnet material but achieve similar performance.

The induction machine (IM) is presented as the most mature and common technology, thanks to its several advantages in terms of reliability, robustness, and costs. High starting torque and a wide constant power region can easily be achieved with an inverter, to the detriment of the pull-out torque at higher speeds. High losses and low efficiency, particularly when compared to permanent magnet synchronous motors, are the main drawbacks.

The last technology presented by Zhu et al. [4] is the switched reluctance machine (SRM); modern SR motor drive is becoming attractive for electric vehicles because of its design simplicity, robustness, and low costs. Specific rotor and stator poles design and current phase advancing method allow the machine to extend the constant power operation up to 3-7 times the base speed. Acoustic noise, vibration, and torque ripple are significant and they remain, together with control complexity, a major issue of this technology.

Trovao et al. [5] conducted a comparative analysis of the performance of IMs and PMSMs for daily use EVs. The two drives are modelled on Matlab/Simulink and tested along two urban driving cycles: the ECE-15 and the NEDC. The PMSM performs significantly better than the induction machine, both from the point of view of speed response and energy consumption, extending the driving range up to 35%. However, the authors observe that for a typical urban utilization, where torque and range demands are limited, the IM drive represents a valid alternative, given its cost effectiveness and negligible maintenance.

Goss et al. [6] investigated the possibility to replace a PMSM with a copper rotor IM. The CR-IM was designed to achieve the same continuous performance as the 50 kW interior permanent magnet motor adopted on the 2004 Toyota Prius Hybrid System.
CHAPTER 2. LITERATURE REVIEW

The efficiencies and losses of both designs are evaluated across the US06, UDDS and HWFET driving cycles. The analysis concludes that the induction machine is a low cost alternative to permanent magnet motors, but its lower efficiency and hence higher losses can influence the design of other components, such as increased battery capacity and inverter rating, and thus they can result in higher vehicle running costs. The authors suggest that the choice of the motor technology requires a careful evaluation of all the trade-offs of the specific application.

Table II: Material quantities and costs

<table>
<thead>
<tr>
<th></th>
<th>IPM</th>
<th>CR-IM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper (kg)</td>
<td>4.5</td>
<td>9.1</td>
</tr>
<tr>
<td>Steel (kg)</td>
<td>23.9</td>
<td>24</td>
</tr>
<tr>
<td>Permanent Magnet/Rotor Cage (kg)</td>
<td>1.3</td>
<td>8.4</td>
</tr>
<tr>
<td>Total (kg)</td>
<td>29.7</td>
<td>41.5</td>
</tr>
<tr>
<td>Copper ($)</td>
<td>31.3</td>
<td>63.8</td>
</tr>
<tr>
<td>Steel($)</td>
<td>23.9</td>
<td>24.0</td>
</tr>
<tr>
<td>Permanent Magnet/Rotor Cage ($)</td>
<td>536</td>
<td>240</td>
</tr>
<tr>
<td>2011 NeFeB Price ($/kg)</td>
<td>536</td>
<td>240</td>
</tr>
<tr>
<td>2013 NeFeB Price ($/kg)</td>
<td>201</td>
<td>58.6</td>
</tr>
<tr>
<td>Total ($)</td>
<td>256-591</td>
<td>146</td>
</tr>
<tr>
<td>Reduction in Consumer Purchase Price with a CR-IM ($)</td>
<td>275-1112</td>
<td></td>
</tr>
</tbody>
</table>

Table V: Cumulative losses over the vehicle lifetime

<table>
<thead>
<tr>
<th></th>
<th>IPM</th>
<th>CR-IM</th>
</tr>
</thead>
<tbody>
<tr>
<td>UDDS (kWh) over lifetime</td>
<td>1274</td>
<td>2235</td>
</tr>
<tr>
<td>US06 (kWh) over lifetime</td>
<td>1430</td>
<td>2514</td>
</tr>
<tr>
<td>HWY (kWh) over lifetime</td>
<td>609</td>
<td>1248</td>
</tr>
<tr>
<td>Combined Average</td>
<td>1104</td>
<td>1999</td>
</tr>
<tr>
<td>Additional Energy Cost from baseline @ $0.25/kWh</td>
<td>0</td>
<td>$224</td>
</tr>
<tr>
<td>@ $0.294/kWh (From ICE)</td>
<td>0</td>
<td>$263</td>
</tr>
</tbody>
</table>

Figure 2.3: Estimated costs associated with each motor design compared by Goss [6]. Table V gives the energy costs due to cumulative losses in the electric drive over a 120000 miles vehicle lifetime.

Pellegrino et al [7] also found that, despite being penalized by cage losses, the induction machine has excellent overload capabilities and performance not far from PMSMs.

2.3 Transmission

The torque characteristics of electric motors lend well to single-speed transmissions. The high torque region from rest can provide enough torque for pull-off and inclines whilst the constant power region is extended for a large speed range. The additional advantages from having electric motor, fixed gearing and mechanical differential in a single compact assembly result in most of the battery electric vehicles currently on the market adopting this configuration.

In 2014 Chevrolet conducted a survey [8] to select the most suitable drive unit for a small electric city car, the Spark EV. The company selected a high torque EM paired with a single-speed planetary gearing, designed to optimize the package space, noise, mass, manufacturability, flexibility and efficiency. The designed unit allowed to use the components of other GM vehicle for more than 75%; this has benefits on vehicle reliability, testing processes and costs, but it can also explain the conservative choice of most of the automotive companies for the single-motor architecture over more innovative powertrain configurations.
The effort to pursue every possible avenue for minor energy saving drives a key area of current research in electric mobility, which investigates the potential advantages of multi-speed gearboxes to improve driving range and drivability.

Musio et al. [9] investigate the effects of different transmission systems on EVs, comparing a two-speed transmission with a fixed gear reduction. Simulations prove that, over the NEDC cycle, the adoption of a two-speed transmission can extend the estimated driving range by 9.3%. The assumptions of constant efficiency of the transmission and of the power electronic converter rise many questions regarding the validity of these results.

Ganji et al. [10] compared three transmission technologies: a fixed gear, a continuously variable transmission (CVT) and a five speed transmission. The simulations included an urban driving cycle, the JC08, an extra-urban schedule, the NEDC, and a highway cycle, the HWFET. The five speed configuration always achieves the longest driving range, with a larger difference over the single-speed during highway driving (+23.8%) than in the urban cycle (+0.6%). The authors remarks the limits of their work, which disregards driving dynamics due to acceleration and deceleration and it doesn’t consider speed losses in the transmissions. These assumptions could have a different impact on the results of the systems reviewed.

The research presented by Ren et al. [11] analyses three transmissions: a fixed reduction, a four speed gearbox and a CVT. They developed a simple EV model in Matlab using predefined libraries available in the QSS Toolkit collection to calculate the energy consumption over different driving cycles. The multi-speed and the CVT systems result in a lower energy consumption over the single speed drivetrain, with gains provided by the four step gearbox ranging from 4.5% to 11%. This work has several limitations, as the model uses a generic set of equations to predict the losses of the electric motor and it doesn’t consider the efficiency of the transmission, of the inverter and the battery. The authors conclude that it is worthy to conduct further research with a more sophisticated driveline model to investigate whether these gains would be lost through the additional losses.

The three studies presented above convey that there may be an advantage to downsizing the electric motor when adopting a multiple-speed transmission. This marries up with the work of Knodel [12] which compares a single-speed drivetrain powered by a high-torque low-speed electric motor and a two-speed transmission connected to a low-torque high-speed drive. The space and weight saving achieved through adopting a smaller motor and a multi-speed gearbox is significant as the high-speed drive system is 53 kg (37 %) lighter than the 143 kg high-torque drive. The simulations used complex physical models developed in-house by Getrag, which include drag forces, moments of inertia and efficiency maps that account for the variation on temperature when necessary. It was found that the two-speed drivetrain has a 18% lower energy consumption over the single-speed for the NEDC. The authors also found a 5 – 10% improvement in favour of the two-speed system when using the same electric motor. Furthermore a 0 – 100 km/h acceleration test was conducted, where the two-speed powertrain achieved better performance (7.4 s vs 8.4 s). Despite the paper shows clear benefits from adopting a multi-speed gearbox, it lacks many details regarding the simulation process and the
models used and it only considers a single driving cycle, which limits these results for a specific driving scenario.

![Efficiency maps](image1.png)

(a) Efficiency maps with operating points of the single-speed high torque drive as a function of wheel torque and speed

![Efficiency maps](image2.png)

(b) Efficiency maps with operating points of the two-speed high speed drive as a function of wheel torque and speed

Figure 2.4: Comparison of the operating points of the two drives analysed by Knodel [12] over the NEDC drive cycle. The efficiency maps account for the losses in the electric machine, in the transmission unit, in the inverter and in the battery. The two-speed drive provides a much larger area where the efficiency is highest (green)

Recently, automotive research is moving towards more complex transmission systems that can improve the short driving range of BEVs. Bottiglione et al. [13] propose a comparison of the energy efficiency performance of six different transmission architectures for EVs: a single gear, a two-speed gearbox, full and half toroidal CVTs and two infinitely variable transmissions, novel structures originally designed to couple ICE and EM on HEVs. The aim of the study is to investigate the potential benefits that a wide speed ratio range could provide, by maintaining the operation of the EM close to the most efficient region. A backward facing model is developed on Matlab/Simulink to evaluate the performance of the drivetrain configurations in steady-state, i.e. when the vehicle is coasting at constant speed and constant load, and under two urban driving cycles: the UDC and the Japanese 10-15. As expected, at constant speed a variable gear ratio doesn’t provide any benefits respect to the single and the two-speed units, which exhibit the best performance among all the transmission systems analysed thanks to their higher efficiency. During the driving cycles the CVTs achieve the lowest energy consumption, outperforming the stepped transmissions and the more complicated and expensive IVTs. Furthermore, it was found that the energy consumption can decrease up to 10% when a two-speed transmission is employed rather than a fixed gear, with a minimum impact on vehicle weight, layout complexity and costs.

2.4 In-wheel drives

Traditional powertrain layouts, such as engine-transmission-differential, determine decisively the shape of conventional vehicles which are designed according to the package requirements of these components, leading to a limited passenger compartment and suboptimal aerodynamics. In-wheel motors are of special interest for future transportation
as they provide completely new car concepts beyond the current limitations. The absence of many power transmission components gives several benefits in terms of flexibility: vehicle body can be shaped in such a way to improve passenger comfort, ergonomics and aerodynamics.

Furthermore, they give the opportunity to optimise the distribution of torque and speed to each wheel independently with a superior maneuverability, driving dynamics and active safety of the vehicle.

Given the many advantages, the list of automotive companies investing in in-wheel drives and exploring their adoption on electric vehicles is rapidly growing. Concept cars such as the Volvo C30 ReCharge [14], the Mitsubishi Concept-EZ MIEV [15] and the Ford Fiesta E-Wheel Drive [16] clearly show the increasing interest.

![In-wheel motor equipped on the Mercedes-Benz E-Class Brabus EV concept](image1)

![Exploded view of the in-wheel direct drive system](image2)

Figure 2.5: The in-wheel direct drive system designed by Protean Electric [17] employs an outer rotor PM motor located around the suspension and a novel inside-out friction brake.

The packaging and integration processes are delicate challenges during the design of in-wheel systems [18]: the electric motor must be accommodated together with the brake, the suspension and, in case of indirect configuration, even with a reduction gear into the limited space available inside a wheel. This unusual position sets new demanding requirements for the drive unit, such as high stiffness to endure external forces and low weight to limit the unsprung mass. The influence of the increased unsprung mass on ride and handling behaviour is being largely studied [19], but the effective consequences are still ambiguous and they are beyond the scope of this research.

Protean Electric [17] investigated different solutions to find the best layout in terms of performance, use of space and compatibility with the rest of the drivetrain. The final concept uses an outer rotor BLDC motor with a ring shape and an inside-out friction brake specifically designed. The whole package can be accommodated around the suspension with minimal impact on the pre-existent structure.

In their paper Jain et al. [20] analyze the characteristics of various motor technologies to select the most viable for in-wheel powertrains. Induction machines are commonly adopted on electric vehicles, but due to the low power density and the low efficiency they are not optimal candidates as in-wheel motors. Currently the PM machine is the
most popular choice as it easily meets the power density and efficiency requirements. The outer rotor configuration can provide high torque capability at low speed as it maximizes the lever arm for the torque production in the machine air gap, but it suffers from a heavier and bulky package compared to the lighter high speed counterpart. As the rotor is directly exposed to outer air, the cooling conditions are improved for the magnets and the resistance to high temperature demagnetization is enhanced. High speed drives integrate a gearbox inside the limited space of a wheel rim, with negative consequences on heat dissipation and permanent demagnetization. Axial flux machines have advantages over radial flux machines in terms of power per unit active volume and power per unit active weight, but they suffer from high magnet material employment and complex manufacturing and assembly processes. Hence, they are restricted to high performance low volume applications. Switched reluctance motors can provide sufficient performance with higher efficiency than induction motors and lower cost than PM machines, but the aforementioned large torque ripple and noise exclude their use for in-wheel drive-trains. Recently these issues are in process of being solved by developing superior control technologies. Thus they could represent a practicable option in the future.

Reis et al. [21] compare three in-wheel concepts: a low speed direct drive PMSM and two high speed indirect drives, a PMSM and an IM. The units were modelled and simulated during the NEDC cycle. The outer rotor arrangement is used for the direct drive, while the high speed motors adopt a conventional inner rotor configuration with independent reduction gears to transfer power to the wheels. As the electromagnetic torque is proportional to the machine volume, the direct drive is heavier and bigger than the high speed machines, which use the fixed reduction gear to adjust the motor speed and torque at the wheels. On the other hand, the low speed machine doesn’t require a mechanical reduction system which could nullify or even reverse the advantages of high speed units in terms of lower volume and weight. According to the simulations the direct drive provides the lowest energy consumption, mainly because no gearbox losses occur during torque production and regenerative breaking. The mass differences between the concepts have significant consequences on the material costs too. As reported by authors’ estimations, the outer rotor machine is the most expensive solution due to the larger utilization of rare magnet material. But the cheaper high speed motors suffer from the additional costs of the transmission, which reduces also their advantage in terms of money. Readers should consider that other manufacturing processes disregarded by the authors could lead to completely different conclusions.

2.5 Conclusion

This chapter provides a review of the current literature pertaining to this field of research, including the diversified topologies of electric drivetrains, the electric motor technologies most suited for traction applications and the transmission systems specifically designed for electric vehicles. Based on their findings, some conclusions can be drawn.

It is clear that the in-wheel powertrain is a very interesting and innovative configu-
ration for a BEV, with substantial advantages over the standard centralized topology. The exposed and critical position of the motors and the increased unsprung mass set the main challenges which are currently excluding its use in the commercial vehicles market on a large scale. The literature with exhaustive analyses of this architecture regarding efficiency and energy saving is limited but it is promising and it encourages further investigation.

Current research describes permanent magnet synchronous machines as the best technology for traction in terms of power density, efficiency and dynamic response. However, the fluctuation in the price of rare permanent magnets is a crucial issue and it often prohibits the adoption of these motors in the automotive field. The papers advise to explore cheaper electric drive systems, such as induction or synchronous reluctance machines, when performance requirements are not too stringent.

The previously mentioned literature suggests that the adoption of a multiple-speed transmission has potential benefits for a fully electric vehicle. But most of the works lack in depth models considering all factors/efficiencies and optimising each parameter of the drivetrains. The papers all state that there is scope to further the research in this area using more complex models.

In conclusion this review shows that there are interesting topics to add to the field of electric vehicle research, in particularly relating to the impact of drivetrain configurations on reducing energy consumption and improving driving range. The upcoming sections describe in detail the mathematical model of an electric vehicle powertrain and give the reader the knowledge needed to understand the properties of the main loss sources.
Chapter 3
Modeling

In this chapter the mathematical model of an electric vehicle is derived from the governing equations during steady-state conditions. First the drivetrain architectures examined in this research are introduced and described. Then the road load theory is presented to describe the forces applied on the vehicle body as it moves. Finally, the main components responsible of losses in electric powertrains are discussed. The section focuses on modeling power losses in the transmission systems and in the electrical drives, calculated using Coulomb’s laws, fluid mechanics and equations based on experimental tests. The aim of this approach is to develop a rigorous methodology to predict energy consumption and efficiency of electric powertrains and to analyse the major sources of loss during changing driving conditions.

3.1 Powertrain architectures

The powertrain of a battery electric vehicle consists of an electric propulsion system with a battery serving as an energy buffer. Often there is only one electric machine connected to the wheel shaft via a gearbox and a differential. However some applications may use several motors which drives independently front and rear axles or each wheels.
The energy is stored chemically in the battery, which is electrically connected to the machine via a DC/AC power electronic converter accompanied by a control system. The control system controls the frequency and magnitude of voltage and current that are applied to the electric machine and these are depending on the driver’s present request, which is communicated via the acceleration and the brake pedal. The following paragraphs describes in more details the drivetrain layouts studied in this research.

3.1.1 Single-motor topology

The single-motor layout is currently dominating the market of BEVs, given the advantages for manufacturers to use most of the components already existing for ICE vehicles. In the front-engine front-wheel-drive configuration [22], power transfers to the wheel through a fixed reduction gearing and a mechanical differential. All the components are integrated in a single assembly on the front axle, thus resulting in a very light and compact structure, Figure 3.2.

Additionally, a multi-speed transmission can be introduced to increase the range of maximum available torque and speed at the wheels. Based on the findings of [13] and [12], a two-speed unit seems the best solution for EVs to improve the performance with a minimum impact on layout complexity, weight and costs. Both the single-speed and the two-speed configurations are modelled and evaluated.

The electric drive selected is the induction machine. Whereas it has a lower efficiency and power density than the permanent magnet synchronous machine it can offer adequate performance, particularly when considering urban utilization, at a much lower price and negligible maintenance, as found in [4], [5] and [6].

Figure 3.2: Conventional single-motor front-wheel drive architecture with a fixed reduction gear [22] (EM=Electric motor, FG=Fixed gear, D=Differential)

3.1.2 In-wheel topology

The in-wheel powertrain provides the tractive force through motors located inside a pair or all the wheels, Figure 3.3. Low speed drives can be directly connected to the wheels, while high speed inner rotor electric motors must be coupled to a fixed gearing...
to adjust torque and speed, with increased complexity and weight of the system and lower efficiency [21, 23]. Given the advantages of the direct drive type, the powertrain architecture proposed in this thesis employs low speed outer rotor permanent magnet machines directly connected to the front wheels. The outer rotor arrangement suits direct drive applications well as it maximizes the lever arm for the torque production and it can satisfy the high torque requirement at low speed. The selection of the permanent magnet technology, which has the highest power density, is common for in-wheel systems ([16], [17]) and it is driven by the aim to limit the negative effects that an increase of the unsprung mass could have on ride comfort and driving dynamics.

Figure 3.3: Direct drive in-wheel powertrain [22]
CHAPTER 3. MODELING

3.2 Glider Model

Vehicle dynamics describe the force applied on the vehicle body when it is moving. As with modeling any objects, the motion of a rolling vehicle can be expressed with various level of detail, depending on the phenomena of interest. For the type of studies conducted in this thesis, which disregards the effects on ride comfort and handling, it is reasonable to assume the vehicle body a single rigid mass, concentrated at its center of gravity [24]. Furthermore, only longitudinal motion is of interest. According to Newton’s second law, vehicle acceleration can be expressed as:

\[ m \frac{dv}{dt} = F_x - F_R \]  \hspace{1cm} (3.1)

where \( m \) is the mass of the vehicle and \( F_x \) is the tractive force. \( F_R \) is the sum of all the resistive forces applied to the vehicle and it includes the aerodynamic drag force, the rolling resistance force and the gradient force.

Figure 3.4: Vehicle longitudinal forces

The aerodynamic drag equals:

\[ F_a = \frac{1}{2} \rho_a C_d A_f (v - v_{\text{wind}})^2 \]  \hspace{1cm} (3.2)

where \( \rho_a \) is the air density, \( A_f \) is the effective front cross sectional area of the vehicle and \( v_{\text{wind}} \) is the wind speed moving in the direction of the car. The aerodynamic drag coefficient \( C_d \) is a dimensionless parameter that represents all the drag effect acting on the vehicle.

The rolling resistance is a result of the deformation of the wheel tyre as it rolls along the roadway. It is calculated as:
The rolling resistance coefficient $f_r$ is dependent on several factors such as the road surface, ambient temperature, driving speed and tyre pressure.


For the purpose of this thesis, which aims to compare different powertrain with the same glider body and during identical driving conditions, the air density $\rho_a$ and the rolling resistance coefficient $f_r$ are assumed constant.

Using Newton’s second law for rotation, we can derive the electromagnetic torque generated by the motor function of the vehicle speed [25]:

$$T_{em} = T_{loss} + \frac{r_w}{b} \left( m_{eff} \frac{dv}{dt} + F_R \right)$$

where $r_w$ is the wheel radius, $b$ is the total drive ratio of the powertrain and $T_{loss}$ takes into account all the losses from the motor to the wheels. Equation (3.5) is valid for all the three architectures, with the only difference that the in the in-wheel topology the torque requirement is split between two drives.

The effective mass $m_{eff}$ includes the vehicle mass and the rotating inertias $J$ and it differs for drivetrain layout. For the single-speed model it is calculated with

$$m_{eff} = m + \frac{1}{r_w^2} \left( b g f d (J_{em} + J_{1s}) + J_d + 2 (J_a + J_w) \right)$$

For the two-speed model

$$m_{eff} = m + \frac{1}{r_w^2} \left( b g f_d b (J_{em} + J_{1s}) + b g f_d J_{2s} + J_d + 2 (J_a + J_w) \right)$$

For the in-wheel powertrain

$$m_{eff} = m + 2 \left( \frac{J_w + J_{em}}{r_w^2} \right)$$

The subscripts refer to the various components of the vehicle: $J_{em}$ is the motor inertia, $J_{1s}$ and $J_{2s}$ are the inertias of transmission shafts, $J_d$ is the inertia of the mechanical differential, $J_a$ and $J_w$ are the inertias of one side axle shaft and of the wheel. $b_g$ and $b f_d$ are the gearbox and the final drive ratios. The formulation of all the steps for the analysis the torque transfer is detailed in the Appendix.

The computation of the effective mass shows that the inertias of the electric motor and of the primary shaft are the biggest contributors in a powertrain, due to the fact that they are multiplied by the square of the total drive ratio.
3.3 Transmission Model

3.3.1 Introduction

Power loss experienced by drivetrains has a great impact on fuel/energy consumption of vehicles. The efficiency of transmission systems is often regarded as a fixed value, but it varies significantly with the change of speed and torque. In order to have a valid evaluation of the efficiency of a powertrain, it is imperative to build a dynamic model which can predict its losses under different running conditions.

Drivetrain losses can be divided in two groups: torque dependent and torque independent losses. The first contribution originates from the sliding and rolling movements of the loaded mechanical components, mostly gears and bearings. Load independent losses, usually addressed as speed or spin losses, are related to the lubrication method and include windage and oil churning losses, which are present as a result of oil/air drag in the environment surrounding the gears.

A significant number of studies analyzed these losses and developed mathematical models to predict the efficiency of gear trains. When the majority of the factors involved are taken into account (gearbox and gears design, rotational speed, oil properties, oil level, operating temperature etc...), the nature of the environment surrounding the gears results in a hard task to formulate a fluid-mechanics based model. Therefore most of works published are based on dimensional analysis or experimental data. This means that such models will only be valid within very limited boundary conditions [26]. Given the excessive level of details required for an exact calculation of all the mechanical losses ([27], [28]), out of scope of this thesis, the aim is to present the major sources of losses in a gearbox, calculate them using the fundamentals of machine components design and simplified equations and derive a final model with reasonable accuracy and complexity which can predict the gear train efficiency during changing operating conditions.

In the following sections, the major contributors of these losses are discussed for each component of an automotive transmission.

3.3.2 Design Principles

The purpose of transmissions is to adjust torque and rotational speed of the engine to the demands of the driving wheels. The type of drive system proposed in this paper is called transaxle. Transmission and transaxle perform the same function, the only difference is that a transaxle combines the transmission gearing, differential and axle shaft connections in the same case housing. Constant-velocity (CV) joints at the ends of the drive shafts allow each wheel to move independently of the vehicle body and guarantee uniform transfer of torque. In the following sections, the terms transaxle and transmission are considered interchangeable. Figure 3.5 shows the typical arrangement of drivetrain components on front-wheel-drive cars.

The advantages it offers include a more compact and lighter package and a higher efficiency [29], provided by the ordinary parallel gear pair that replaces an hypoid or bevel final drive of conventional differential. Thus, transaxle systems are very common
on vehicles with the engine located at the same end as driven wheels.

Transmissions can vary widely. Their characteristics are based on several factors that are illuminated by the designs. The two systems discussed here are a single-speed reducer and a two-speed manual transmission. The manual type has been chosen for the high efficiency and the less complicated architecture [30], but it suffers from power interruption during the gear shift. Modern transmission technologies, which promise seamless gearshift with an efficiency comparable to the standard manual type, are being investigated but the complexity of their layouts requires specific works to model their kinematics ([31], [32] and [33]) and thus they have not been considered here.

![Figure 3.5: Front-wheel-drive powertrain with transaxle](image)

![Figure 3.6: Schematics of the transmission units adopted on the conventional single-motor powertrain](image)
The schematics of the configurations adopted on the single-motor vehicle are given in Figure 3.6 (a) and (b).

The single-speed gearbox consists of a single stage spur gear reduction, which incorporates the fixed reduction and the final drive into the same gear set.

The two-speed transmission adopts a dual stage layout. The electric motor is directly connected to the primary shaft of the gearbox, which has the input gears casted on it. On the secondary shaft the output gears are supported on needle bearings, which allow them to spin freely when they are not engaged. The transmission of torque through one of the gear pair is effected by the dog clutch in the middle and the synchronizer unit.

Several designs exist for automotive differentials. Manufacturers have begun to use the epicyclic configuration, which provides high efficiencies with a very axially compact arrangement.

An ordinary epicyclic gear set is a coaxial mechanism that consists of the ring gear, an outer gear with internal teeth, the sun gear, a central gear with external teeth, and the planet gears which mesh simultaneously with the ring and the sun gears. A carrier ensures that the planets don’t fall out of the assembly. This arrangement makes the planets capable to both rotate and orbitate around the sun gear and it allows the wheels to rotate at different speed if necessary.

When applied as mechanical differential, the ring gear also meshes with the pinion, the input gear which sits at the end of the output shaft of the gearbox, respectively $R_1$ and $R_5$ in Figure 3.6 (a) and (b), and it provides the final drive ratio. Two sun gears are used to transmit the motion to the axle shafts. In the absence of any outside force on the sun gears from the wheels, as the carrier forces the planet gears to orbit around the sun gear, both sun gears and wheels travel at the same speed. If a wheel is being forced to slow down or speed up, such as when turning a corner, the planet gears begin to rotate around their axes to accommodate the speed difference developed between the sun gears and between the wheels.
CHAPTER 3. MODELING

Under the assumptions of straight-line driving and total traction, the sun gears turn at the same speed of the ring gear and none of the complicated motions associated with the rotation of the planet gears occur. Under these conditions, no mechanical losses are generated in the epicyclic gearing \[34\]. Therefore the differential model discussed here only considers the final drive gear pair.

All the gear pairs modelled are helical type, which have wide applications in the automotive industry for their smoother and quieter operation. The transaxle units employ two cylindrical roller bearings to support each shaft of the gearbox and the differential shaft.

In modelling step gearboxes corresponding to each gear ratio, the characteristics of the engaged gear must be taken into account in the calculation of load dependent losses and, according to the architecture of gearbox, spin losses should be considered for the idling gears too. Losses in the synchronizer and in the needle bearing of the idling gear are not calculated. Furthermore, losses in the CV-joints are assumed to be negligible too.

3.3.3 Clutch

The clutch’s function is to control the power transmission between two shafts. When the clutch is engaged, its friction plates are brought together and torque and rotational speed are transmitted. When it is disengaged, the friction plates are separated from each other and the power flow is interrupted. There are two clutch systems for automotive applications \[30\]. The first system is the dry clutch, it has high efficiency but low torque capacity, therefore they are mainly used on small vehicles with limited engine torques. On the other hand, a wet clutch has a higher power-to-weight ratio and longer life, but it has a lower efficiency and it is of a technically higher order because of the oil supply required. Wet clutches are suited to vehicles with little installation space.

Given the low torque of the designed vehicles, mainly intended for city driving, a single plate dry clutch is the best choice for a two-speed manual transmission. Friction losses occur only during the start of the engagement, when the two plates are slipping, and they are so small in steady-state conditions that they are negligible \[35\]. Especially on electric vehicles, where the motor can provide torque from zero speed and the clutch is required only for gear synchronization during the gear shift.

3.3.4 Gear pair

**Torque dependent losses** As mentioned earlier torque dependent losses occur at contact between the teeth of two meshing gears, the pinion, the input or driving gear of a gear pair, and the gear, the output or driven gear. Gear meshing losses consist into a sliding and a rolling component. Sliding loss is the dominant component and it is function of the sliding velocity and the friction force, which depends on the instantaneous tooth load and the instantaneous coefficient of friction. Rolling loss have a minor impact as source of load dependent gear losses and in almost every case it is neglected, since an involute gear which is perfectly shaped and has low tooth flexibility has no rolling losses
Ohlendorf’s approach [36] is the most common method to calculate the sliding losses based on the tooth geometry of the gear. Power losses can be derived using the Coulomb Law

$$P_{fric} = F_{fric}(x)v_{red}(x) = F_N(x)\mu_{fric}(x)v_s(x)$$  \hspace{1cm} (3.9)$$

where $F_N$, $v_s$ and $\mu_{fric}$ are the instantaneous values of the normal load, the sliding velocity and the friction coefficient at the meshing point. Equation (3.9) can be rearranged as

$$P_{fric} = F_{N_{max}}\frac{F_N(x)}{F_{N_{max}}}\mu_{fric}(x)\frac{v_s(x)}{v}$$  \hspace{1cm} (3.10)$$

For helical gears the normal load applied to the gear is

$$F_{N_{max}} = \frac{F_1}{\cos(\alpha_n)} = \frac{F_t}{\cos(\alpha_n)\cos(\beta)} = \frac{T}{0.5d\cos(\alpha_n)\cos(\beta)}$$  \hspace{1cm} (3.11)$$

where $\alpha_n$ is the normal pressure angle, $\beta$ is the helix angle and $d$ is the pitch diameter of the gear.

Figure 3.8: Forces acting on a helical gear mesh

If we approximate the friction coefficient constant along the path of contact

$$P_{fric} = F_{N_{max}}\mu_{fric}v \frac{1}{p_b} \int_A^E F_N(x) \frac{v_s(x)}{v}$$  \hspace{1cm} (3.12)$$

where the interval [A, E] includes the different meshing zones along the path of contact represented in Figure 3.9 (a).
CHAPTER 3. MODELING

Figure 3.9: Meshing zone profile (a) and load distribution along path of contact (b) of a gear tooth

The formulation of the instantaneous load distribution $F_N(x)$ on gear tooth is complicated and there exists divergence among different schools for its computation. Without considering the influence of gear deformation, Ohlendorf’s approach [36] assumes that loads are distributed evenly on the line of contact and that load coefficient is 1.0 when one pair of teeth is involved and it is 0.5 when two pairs of teeth are involved, as depicted in Figure 3.9 (b). Equation 3.12 is solved by introducing the gear loss factor

$$H_v = \frac{\pi}{\cos(\beta_b)} \left( \frac{1}{z_1} + \frac{1}{z_2} \right) (1 - \epsilon_\alpha + \epsilon_1^2 + \epsilon_2^2) \tag{3.13}$$

where $z_1$ and $z_2$ are the number of teeth of gear and pinion. $\epsilon_1$, $\epsilon_2$ are the addendum contact ratios of gear and pinion and $\epsilon_\alpha = \epsilon_1 + \epsilon_2$ is the transverse contact ratio. Mesh losses can then be expressed as

$$P_{fric} = P_{in} \mu_{fric} H_v \tag{3.14}$$

All the formulations presented are valid for gear pairs without profile shift corrections. One method to reduce meshing losses is to introduce a positive profile shift to change and optimize load distribution and sliding speed. For these cases of profile modifications a more complex approach, not discussed here, is required [37].

The prediction of the coefficient of friction is still a challenging task. It involves many parameters and it depends on the lubrication regime and the meshing point along the path of contact. Models derived from EHL analysis are more accurate than the ones based on the empirical formulae, but they have the disadvantage of requiring excessive computational effort. The formula derived by Benedict and Kelley [38] is a good compromise between accuracy and simplicity

$$\mu_{fric} = 0.0127 \log_{10} \left( \frac{29.652 W_n(x)}{b_{vy} \nu_s(x) v_r^2(x)} \right) \tag{3.15}$$
where \( v_r(x) = \frac{v_1(x) + v_2(x)}{2} \) and \( v_s(x) = v_1(x) - v_2(x) \) are the rolling and the sliding velocities.

Figure 3.10: Instantaneous coefficient of friction calculated along the path of contact (a) and average value over speed (b). The formula published by Benedict and Kelley [38] predicts larger values of the coefficient of friction than the EHL model [39], especially at the pitch point (C) where we expect zero friction instead of a maximum. (gear ratio=1, number of teeth=50, pitch diameter=150 mm, input speed=1500 rpm, input torque=15 Nm)

Figure 3.11: Losses due to gear friction (gear ratio=1, number of teeth=50, pitch diameter=150 mm, input speed=1500 rpm, input torque=15 Nm)

This equation considers the variation of the load distribution, sliding speed and rolling speed during the meshing cycle, but it shows a singularity at the pitch point where it becomes infinite, Figure 3.10 (a). This is in conflict with the thermal elastohydro-
CHAPTER 3. MODELING

dynamical lubrication (EHL) analyses which found nearly zero friction near the pitch point [39], due to the fact that sliding velocity is zero. As a consequence the model of Benedict and Kelley [38] predicts higher sliding losses when compared to more complex EHL based models and a slightly lower efficiency for a mechanical gear pair (−0.2% in [39]), but this has a negligible effect in the total losses of a drivetrain.
In order to limit the effects of this error, the coefficient of friction is averaged over a mesh period. Since multiple gear teeth can be engaged at the same time, the formulation of total contact length for each meshing point is performed for the estimation of the load distribution, using the method presented in [40].

Figure 3.11 shows the influence of speed and load torque on gear friction losses.

Torque-independent losses
Spin power losses in gear pairs are directly related to the lubrication method. They are either due to churning if the gears are immersed in an oil bath or due to windage if an intermittent jet lubrication method is used. In both types of lubrication methods (dip or oil jet), the spin losses involve complex hydrodynamic phenomena which are very difficult to describe in analytical formulations. Given the lack of published papers regarding no-load power losses under oil jet lubrication when compared to the number of those addressing dip lubrication, the following section describes the oil churning phenomena typical of dip-oil lubricated gearboxes, based on the work of Seetharaman [27].

Oil churning losses Oil churning losses are caused by the spin movement of mechanical components in dip-oil lubricated conditions.

They can be classified in drag losses $P_d$ and pocketing losses $P_p$

$$P_s = P_d + P_p$$  \hspace{1cm} (3.16)

Spin losses due to drag forces are associated with the interaction of individual gears with the surrounding lubricant medium. They are expressed as the sum of three individual components

$$P_{di} = P_{dpi} + P_{dfi} + P_{rfi}$$  \hspace{1cm} (3.17)
where \( P_{dpi} \) is the power loss due to oil drag on the periphery of a gear, \( P_{df_i} \) is the power loss due to oil drag on the sides and \( P_{rf_i} \) originate from the swirling motion of the lubricant during the filling of the cavity between two meshing teeth. The subscript \( i \) indicates the rotating element of the gear pair. Total drag losses of a gear pair are found as

\[
P_d = P_{d1} + P_{d2} \tag{3.18}
\]

The formulation of each contribution is discussed in details in [27]. According to his analysis \( P_{df} \) and \( P_p \) are predominant in the calculation of oil churning losses. For this reason \( P_{dp} \) and \( P_{rf} \) are neglected.

In the derivation of \( P_{df_i} \) two flow regimes are distinguished and handled separately. At low to medium speeds, depending on the size of the individual gears, the flow can be assumed to be laminar, with transition to turbulence taking place at higher rotational speeds corresponding to a Reynolds number defined as

\[
Re = \frac{2\rho \omega r_{oi}^2}{\mu} < 10^5 \sim 10^6 \tag{3.19}
\]

with \( r_{oi} \) outer radius of the rotating element. In the laminar regime losses can be expressed as

\[
P_{df_i} = \left( \frac{0.41 \rho \nu^{0.5} \omega_i^{2.5} r_{oi}^4}{\sqrt{\sin (\cos^{-1} (1 - \bar{h}_i) - (1 - \bar{h}_i) \sqrt{\frac{\pi}{2} - \sin^{-1} (1 - \bar{h}_i) - (1 - \bar{h}_i) \sqrt{\bar{h}_i (2 - \bar{h}_i)}})}} \right) \tag{3.20}
\]

while for turbulent working operations

\[
P_{df_i} = \left( \frac{0.025 \rho \nu^{0.14} \omega_i^{2.86} r_{oi}^4}{\sqrt{\sin (\cos^{-1} (1 - \bar{h}_i) - (1 - \bar{h}_i) \sqrt{\bar{h}_i (2 - \bar{h}_i)}}^{0.14}} \right) \tag{3.21}
\]

\( \bar{h}_i \) is the immersion ratio of the gear and it is defined as

\[
\bar{h}_i = \frac{h_i}{r_{oi}} \tag{3.22}
\]

with \( h_i \) immersion depth of the gear in the lubricant.

Pocketing losses originate from the squeezing of the fluid out of the mesh space: as two adjacent teeth of a gear approach the surface of the mating gear, the volume of the cavity between them decreases and the lubricant is pumped out of the meshing zone through the openings. Since multiple cavities are squeezed simultaneously and volume variation depends on the angular position of the cavity, a simplified approach
approximates the rotational positions of the gear in a sequence of \( M = 4 \) increments. The detailed calculation of the volumes area for each increment is not applied here and it can be found in Seetharaman work [27]. In this thesis a further approximation estimates the volume at \( m = 0 \) \( V^{(0)} \) as half of the total volume of the cavity, at \( m = 1 \) \( V^{(1)} \) decreases to \( \frac{2}{3} \) of \( V^{(0)} \), at \( m = 2 \) \( V^{(2)} \) is only the bottom half circle remaining and at \( m = 3 \) \( V^{(3)} \) is reduced to half of \( V^{(2)} \).

Figure 3.13: Volume areas at each increment

Pocketing losses are defined as follow

\[
P^{(m)}_{p,i,j} = v^{(m)}_{b,i,j} F^{(m)}_{b,i,j} + 2v^{(m)}_{e,i,j} F^{(m)}_{e,i,j}
\]

(3.23)

where \( v \) and \( F \) are the velocities and the forces acting on the oil which escapes respectively through the backlash or the end flow openings. This represents only the contribution of the \( j \)-th cavity at the \( m \)-th rotational position for gear \( i \). The velocity of the flow escaping from one opening is defined as

\[
v = -\frac{1}{A} \frac{dV}{d\theta} \omega
\]

(3.24)

where \( V \) is the volume of an arbitrary cavity, \( A \) is the flow area and \( \theta \) defines the angular position of the gear. The negative sign is neutralized by the negative value of \( \frac{dV}{d\theta} \), as each successive volume is smaller than the previous during pocketing. The pressure of the control volume is derived as

\[
p^{(m)} = p^{(m-1)} + \frac{1}{2}\rho \left[ \left( v^{(m-1)} \right)^2 - \left( v^{(m)} \right)^2 \right]
\]

(3.25)

For the conservation of momentum principle, the force applied on the oil is

\[
F = -p A
\]

(3.26)
Total pocketing losses are finally obtained as sum of the components of each volume for both gears, then averaged over the number of increments

\[ P_p = \frac{1}{M} \sum_{m=0}^{M} \left( \sum_{j=1}^{P_1(m)} + \sum_{j=1}^{P_2(m)} \right) \]  

(3.27)

Figure 3.14: Comparison of oil churning losses for different face width values calculated with the volume approximations and with the complete formulations [27]. The approximated model shows sufficient accuracy \( \bar{h} = 1 \), gear ratio=1, number of teeth=40, pitch diameter=95.95 mm)

Figure 3.14 compares the oil churning losses calculated with the approximated model proposed here and the Seetharaman model [27]. This proves the validity of the assumptions taken, which greatly simplify the calculations of the volume areas of the cavities but they don’t affect the accuracy of the results.

3.3.5 Bearings

The purpose of bearings is to make the shafts rotate with as little friction as possible when external axial and radial forces acting on them. Power loss that originates from the friction mechanisms in rolling bearings consists of both torque dependent and spin contributions.

Figure 3.15 shows six different types of losses in a rolling contact bearing: rolling and sliding friction between rolling element and bearing rings (1 and 2), sliding friction between cage bar and bearing rings (3), sliding friction between cage and rolling element front surface (4), sliding friction between rolling element and outer ring (5), and friction between cage and rolling element (6). The new SKF model [41] is able to reproduce accurately these loss mechanisms in different lubrication conditions. This method expresses the total frictional torque as sum of four components:
Since the rolling bearings considered don’t have seals, the term $T_{seal}$ is not discussed in this section.

Rolling frictional moment  Rolling friction comes from the deformation and the rolling motion of the elements in the raceway of a bearing under normal loads. This loss is calculated as:

$$T_{rr} = \phi_{ish} \phi_{rs} G_{rr} (\nu n)^{0.6}$$  \hspace{1cm} (3.29)$$

where $\nu$ is the kinematic viscosity of the lubricant at the operating temperature in mm$^2$/s and $n$ is the rotational speed in rpm. $G_{rr}$ represents the influence of bearing load on the rolling resistance and depends on the bearing type, the bearing mean diameter and the axial load. For cylindrical roller bearings

$$G_{rr} = R_1 d_m^{2.41} F_r^{0.31}$$  \hspace{1cm} (3.30)$$

where $d_m$ is the bearing mean diameter in mm, $F_r$ is the radial load and $R_1$ is the rolling friction constant related to the bearing type. $\phi_{ish}$ and $\phi_{rs}$ are the inlet shear and the kinematic replenishment/starvation reduction factors. They consider the influence of rotational speed and lubricant viscosity on the oil film thickness at the contact area and consequently on rolling resistance.
\( \phi_{ish} = \frac{1}{1 + 1.84 \times 10^{-9} (n \ d_m)^{1.28 \nu^{0.64}}} \)

\( \phi_{rs} = \frac{1}{\exp \left[ K_{rs} \nu \ n(d + D) \sqrt{\frac{K_z}{2(D - d)}} \right]} \)

where \( d \) and \( D \) are the bearing internal and the outer diameters, \( K_{rs} \) is the replenishment/starvation coefficient and \( K_z \) is a geometric constant related to the bearing type.

**Sliding frictional moment**  Sliding friction is calculated using

\[ T_{sl} = G_{sl} \mu_{sl} \]  \hspace{1cm} (3.31)

\( G_{sl} \) represents the influence of bearing load on the sliding resistance and depends on the bearing type, the bearing mean diameter and the axial and radial loads. For cylindrical roller bearings

\[ G_{sl} = S_1 d_m^{0.9} F_a + S_2 d_m F_r \]  \hspace{1cm} (3.32)

where \( F_a \) is the axial load, \( S_1 \) and \( S_2 \) are the sliding friction constants related to the bearing type. The sliding friction coefficient \( \mu_{sl} \) strongly depends on the lubrication conditions and it can be estimated using

\[ \mu_{sl} = \phi_{bl} \mu_{bl} + (1 - \phi_{bl}) \mu_{EHL} \]  \hspace{1cm} (3.33)

\( \phi_{bl} \) is a weighting factor which evaluates the transition of the friction coefficient from full film lubrication \( \mu_{EHL} \) to boundary lubrication \( \mu_{bl} \)

\[ \phi_{bl} = \frac{1}{\exp \left[ 2.6 \times 10^{-8} (n \ \nu^{1.4}) d_m \right]} \]  \hspace{1cm} (3.34)

**Drag friction**  The rotational movement of bearings submerged in oil bath generates drag frictions and consequently no load losses. The frictional moment of drag losses for cylindrical roller bearings can be calculated using

\[ T_{drag} = 4V_m \ K_{roll} \ C_w \ B \ d_m^4 \ n^2 + 1.093 \times 10^{-7} n^2 \ d_m^3 \left( \frac{n \ d_m^2 \ f_t}{\nu} \right)^{-1.379} \ R_s \]  \hspace{1cm} (3.35)
with

\[ K_{\text{roll}} = \frac{K_L K_Z (d + D)}{D - d} \times 10^{-12} \quad (3.36) \]

where \( B \) is the bearing width, \( K_L \) and \( K_Z \) are geometric constants related to the bearing geometry. The complete formulation of the remaining quantities \( C_w \), \( f_t \) and \( R_s \) is described in [41].

\[ \begin{align*}
\text{Rolling friction} & \quad \text{Sliding friction} \\
\text{Drag friction} & \quad \text{Total bearing friction}
\end{align*} \]

Figure 3.16: Friction loss mechanisms in cylindrical roller bearings over input speed @ \( T = 15 \) Nm calculated with [41]. Bearing geometry is given in the Appendix.

Figure 3.16 describe each loss contribution and the total losses of two bearings supporting a shaft at a constant load of \( T = 15 \) Nm. At low speed losses are mainly determined by sliding friction, due to the boundary lubrication conditions. As the speed increases the full-film layer formed restrains sliding mechanisms and drag friction becomes dominant.

At low speed the New SKF model predicts a friction torque very different from the one measured experimentally [28]. This happens because \( \mu_{bl} \) has a strong influence at low speed. For a correct prediction of the rolling bearing friction torque the adjusted
values given in [28] are used for the friction coefficients $\mu_{bl}$ and $\mu_{EHL}$.

### 3.3.6 Lubrication

Oil properties have a significant impact on gearbox performance [27]. A low viscosity lubricant can decrease no load losses, i.e. oil churning and bearing drag losses. A thicker oil can be used to reduce surface contact of load-elements (friction at the mating teeth and in the rolling elements of bearings) and as a consequence friction loss, but it requires more energy to move gears and bearing rollers through, leading to higher spin losses.

The mineral oil W211, typical lubricant for transmission systems, is used in the models. The main data are given in the Appendix.

Oil temperature variations are neglected during the simulations; it is assumed a constant operating temperature of 40° C at the gear proximity and of 100° C in the bearings.
3.4 Electrical drives

This section presents the basic principles of operation of the induction machine and the permanent magnet synchronous machine during steady state conditions. Transient behaviours are not considered due to the relatively slow dynamics of the driving cycles. Furthermore, thermal behaviour and saturation effects are disregarded.

3.4.1 Induction machine operation

There are two types of induction machines: the wound-rotor and squirrel-cage. Because of high cost, need for maintenance, and lack of sturdiness, the wound-rotor induction motor is less attractive than the squirrel-cage counterpart, especially for electric propulsion in electric vehicles [42]. Hence, this section discusses the model of a squirrel-cage induction motor.

![IEEE equivalent circuit of the induction machine](image)

Figure 3.17: IEEE equivalent circuit of the induction machine [43]

Figure 3.17 represents the IEEE recommended equivalent circuit of a single phase of the induction machine. $V_1$ is the stator voltage, $R_1$ and $R'_2$ are the resistances of the stator and the rotor windings, $X_1$ and $X'_2$ are the stator and rotor reactances and $X_m$ is the mutual reactance. The superscript indicates that the rotor quantities are referred to the stator side. This concept is useful in modeling induction machines for the purposes of studying their performance as seen from the stator terminals with a single equivalent circuit [44]. The slip $s$ defines the relative motion between the rotating electromagnetic field in the air gap and the rotor and it is expressed as

$$ s = \frac{\omega_s - \omega_r}{\omega_s} $$

(3.37)

where $\omega_s$ is the synchronous speed of the rotating field and $\omega_r$ is rotor angular speed.

The Thevenin’s network theorem is often applied to this circuit to simplify the computations of torque and power [43]. The equivalent circuit is described by the Thevenin
equivalent voltage $V_{th}$ and impedance $Z_{th} = R_{th} + jX_{th}$

$$V_{th} = V_1 \left( \frac{jX_m}{R_1 + j(X_1 + X_m)} \right)$$

(3.38)

$$Z_{th} = \frac{jX_m (R_1 + jX_1)}{R_1 + j(X_1 + X_m)}$$

The torque-speed curve of the induction machine is expressed by the following equation

$$T_{em} = \frac{3}{\omega_s} \left( \frac{V_{th}^2}{R_{th} + \frac{R_2^r}{s}} \right) + \frac{R_2^s}{s}(3.39)$$

Stator and rotor current are given by

$$I_1 = \frac{V_1}{Z_{in}} \quad I_2 = \frac{V_{th}}{Z_{th} + jX_2' + \frac{R_2^s}{s}}$$

(3.40)

with $Z_{in}$ input impedance

$$Z_{in} = R_1 + jX_1 + \frac{jX_m \left( \frac{R_2^r}{s} + jX_2' \right)}{\frac{R_2^r}{s} + j(X_m + X_2')}$$

(3.41)

The induction machine is fed by a voltage source inverter, which can control the stator frequency with a PWM switching scheme to extend the range of operation of the machine and to obtain the torque speed curve required for traction applications. Various speed control methods have been developed to adjust the motor speed of induction machines [42]. Since only steady-state operations of the electric motors are of interest, the variable-voltage variable-frequency control is preferred to the more complex field-oriented methods, suitable when dynamic performance are critical. The Voltage-to-frequency $V/Hz$ control is based on the following concepts:

- Below $f_{rat}$ constant torque is generated with a constant $V/f$ control. This is valid for most operating frequencies, where the induced voltage equals the stator voltage. At low frequency this approximation is not valid, hence a voltage boost is added to compensate the voltage drop across the stator impedance.

- Above $f_{rat}$ the voltage can’t increase further and it remains constant at the rated value, resulting in a field-weakening operation of the machine. Constant power is possible in this region.
CHAPTER 3. MODELING

3.4.2 PM synchronous machine operation

The PMSM originates from the synchronous motor with permanent magnets replacing the field circuit. This modification eliminates the rotor copper loss as well as the need for the maintenance of the field-excitation circuit. According to the placement and the arrangement of the magnets, different configurations exist for the permanent magnet synchronous machine. Among them, the surface mounted (SPM) and the interior (IPM) permanent magnet types are the most adopted for traction [7]. In SPM motors magnets are simply glued on the rotor surface, thus they offer the advantage of simple manufacturing processes at the expense of a limited operating speed due to the inadequate mechanical strength of the magnets [42]. IPM machines can be further classified in surface inset and interior buried. They are widely used because of the extra torque capability due reluctance and the inherent protection of buried magnets from demagnetization [42].

In the following sections the discussion will be based on the radial-field topology with permanent magnets mounted on the rotor surface. The principles of operation presented are valid both for the outer-rotor and the inner-rotor configurations.
Figure 3.20 shows the $d$-$q$ equivalent circuit of a synchronous machine, where $d$-$q$ refers to a reference frame with the direct or $d$-axis in the same direction of the magnetic flux and the quadrature or $q$-axis by 90 electrical degrees ahead the $d$-axis.

The stator voltage equations in the $d$-$q$ axis can be expressed as:

\[
\begin{align*}
  u_d &= R_s i_d + L_d \frac{di_d}{dt} - \omega_{el} L_q i_q \\
  u_q &= R_s i_q + L_q \frac{di_q}{dt} + \omega_{el} (L_d i_d \psi_m)
\end{align*}
\]  

(3.42)

where $R_s$ is the resistance of the stator winding, $\omega_{el}$ is the electrical speed, $L_d$ and $L_q$ are the direct and quadrature inductances and $\psi_m$ is the flux linkage related to the permanent magnets.

Under steady state for a non salient machine ($L_d = L_q = L_s$), Equation (3.42) simplifies into:

\[
\begin{align*}
  u_d &= R_s i_d - \omega_{el} L_s i_q \\
  u_q &= R_s i_q + \omega_{el} (L_s i_d + \psi_m)
\end{align*}
\]  

(3.43)

The electro-magnetic torque acting on the rotor is generated from the interaction between the PM flux and $q$-axis armature current

\[
T_{em} = \frac{3}{2} p \psi_m i_q
\]  

(3.44)

The rms value of the back electromotive force (back-EMF), the voltage induced in the stator windings due to the presence of a rotating magnetic field, is given as:

\[
E_{FRMS} = \frac{\omega_{el} \psi_m}{\sqrt{2}}
\]  

(3.45)

When controlling a non-salient PMSM the quadrature-axis current is uniquely determined by the desired torque. The only remaining control choice remains the desired value for the direct-axis current. The control strategy adopted here simply sets $i_d = 0$.  

38
CHAPTER 3. MODELING

It is common to supply a direct-axis current as to decrease the direct-axis flux linkage and thus limit the terminal voltage. This technique is commonly referred to as flux weakening and comes at the expense of increased armature current. A flux-weakening control has not been implemented in this thesis.

3.4.3 Power losses

In order to estimate the efficiency of the electrical machines accurately much attention is paid to the calculation of the losses. In general losses can be separated into winding, iron, mechanical and stray losses.

\[ P_{\text{loss}} = P_{\text{cu}} + P_{\text{fe}} + P_{\text{mech}} + P_{\text{stray}} \]  
(3.46)

Mechanical losses

Mechanical losses are caused by friction mechanisms between the rotating components. The major components are friction loss in the bearings and windage loss in the gas.

\[ P_{\text{mech}} = P_{\text{bear}} + P_{\text{wndg}} \]  
(3.47)

Bearing losses can be approximated as:

\[ P_{\text{bear}} = k_{\text{bear}} \omega_r \]  
(3.48)

where \( k_{\text{bear}} \) is a constant coefficient concerning the geometry of the bearing [41].

Windage loss represents the power absorbed by the liquid or the gas surrounding the rotor as a result of the relative motion of the rotor and the stator. Vrancik [46] performed experimental investigations on windage losses in electrical machines and he developed an equation valid under the following assumptions:

- no axial flow exists
- the airgap length is small compared to the radius
- the surface of rotor and stator has no irregularities

The resultant losses can be calculated with

\[ P_{\text{wndg}} = \pi k_{\text{fric}} \rho r_{\text{rotor}}^4 \omega_r^3 l_{\text{axial}} \]  
(3.49)

where \( \rho \) is the density of the gas or liquid in the airgap, \( r_{\text{rotor}} \) is the radius of the rotor and \( l_{\text{axial}} \) is the axial length of the machine. The skin friction coefficient \( k_{\text{fric}} \) is defined as:

\[ \frac{1}{\sqrt{k_{\text{fric}}}} = 2.04 + 1.768 \ln \left( Re \sqrt{k_{\text{fric}}} \right) \]  
(3.50)
The Reynold number $Re = \frac{\omega r_{\text{rotor}} \delta_{\text{ag}}}{\nu}$ describes the flow of the fluid for a rotating cylinder.

Figure 3.21 shows the analytical computation of windage and friction losses for two rotor designs. A rotor diameter of $r_{\text{rotor}} = 150$ mm is used to examine the mechanical losses in a high-speed inner rotor machine. A low-speed outer rotor type with $r_{\text{rotor}} = 350$ mm is analysed in contrast. The two concepts adopt the same axial length $l_{\text{axial}} = 170$ mm and air gap length $\delta_{\text{ag}} = 2$ mm. The air in the proximity of the rotor is assumed at a constant temperature of $T = 100^\circ C$.

![Figure 3.21](image)

(a) Mechanical losses in a high speed inner rotor machine

(b) Mechanical losses in a low speed outer rotor machine

Figure 3.21: Effect of speed on windage and bearing friction losses for two machine designs. The above figures show a larger magnitude of mechanical loss for high speed motors than low speed drives.

Copper losses Copper losses are caused by the flowing current into the windings of the machine. In a three phase system they amount to

$$P_{\text{cu}} = 3R_{\text{ph}}I^2$$

(3.51)

where $R_{\text{ph}}$ is the resistance of one phase winding and $I$ is the rms value of the flowing current. As mentioned before the adoption of permanent magnets instead of the excitation winding in PMSMs eliminates rotor copper losses, which are present in IMs.

At high frequencies, the skin effect and the proximity effect cause a nonuniform distribution of the current in the conductor and consequently higher losses due to the increase of the resistance. The effective resistance can be calculated using the method proposed in [47]:

$$R_{\text{ac}} = R_{\text{dc}} \left\{ \frac{h \sinh (h/\delta_c) + \sin (h/\delta_c)}{\delta_c \cosh (h/\delta_c) - \cos (h/\delta_c)} + \frac{2}{3} (d^2 - 1) \frac{h \sinh (h/\delta_c) - \sin (h/\delta_c)}{\delta_c \cosh (h/\delta_c) + \cos (h/\delta_c)} \right\}$$

(3.52)
The skin depth $\delta_c$ is defined as $\delta_c = \frac{1}{\sqrt{\pi f \mu \sigma}}$.

Figure 3.22 shows the distribution of the conductors in a slot and the approximation of a round conductor as a rectangular equivalent with an effective conductivity $\sigma = \sigma_c k_c$, where $\sigma_c$ is the material conductivity and $k_c$ is the slot filling factor.

The influence of skin and proximity effects on the resistance for different combinations of diameter $h$ and number of conductors per width $d$ is depicted in Figure 3.23 (a). It is evident that for frequencies below 150 Hz their impact is fairly small.

Figure 3.23: Influence of frequency on the effective resistance of conductors (a) and on iron losses (b)

It must be remarked that the resistivity of the conducting material changes with the temperature. However, the effects of temperature are not considered in this work.

Iron Losses  Iron losses, also called core losses, are caused by the changing magnetic field in the ferromagnetic parts of the machine. They can be separated into hysteresis and eddy currents losses.
The former are related to the hysteresis properties of the magnetic material and they are equal to the area enclosed by the hysteresis curve

\[ p_{Fe} = f \oint HdB \]  \hspace{1cm} (3.53)

Eddy currents originate from the attempt of the magnetic material to counteract the variation of the flux density and they lead to a broadening of the hysteresis curve and thus an increase of total losses. The standard loss separation model defines iron losses (W/kg) as

\[ p_{Fe} = k_h f^\alpha \hat{B}^\beta + k_e f^2 \hat{B}^2 \]  \hspace{1cm} (3.54)

where \( k_h \) and \( k_e \) are hysteresis and eddy current loss coefficients related on the type of material and laminations used. The M350-50A material is chosen for the stator iron. All the coefficients \( k_h, k_e, \alpha, \beta \) are extracted from the material loss data provided by the manufacturer through curve fitting and they are given in table 3.1. Figure 3.23 (b) shows the variation of iron loss density with frequency for the M350-50A using equation 3.54.

<table>
<thead>
<tr>
<th>Lamination coefficient</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( k_h )</td>
<td>( 6.41 \times 10^{-4} )</td>
</tr>
<tr>
<td>( k_e )</td>
<td>( 7.58 \times 10^{-6} )</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>1.795</td>
</tr>
<tr>
<td>( \beta )</td>
<td>2.53</td>
</tr>
</tbody>
</table>

Table 3.1: Material coefficients for M350-50A laminations

Core losses are calculated only in the stator laminations and they are disregarded for the rotor. In PMSMs rotor iron loss is caused by induced eddy currents and hysteresis in the iron core and the magnets. Under the assumption of operating speed equal or close to the synchronous speed, valid also for the induction machine at small slip values, core loss is negligible for the rotor.

**Stray losses** Stray load losses result from the leakage and the higher-harmonic components of the magnetic flux to which all the metallic parts of the machine are exposed to. Due to their nature, stray load losses are difficult to predict analytically and difficult to measure accurately [18]. The calorimetric method with segregation of losses is one the most used procedure: the stray load losses are determined as difference between the machine total losses measured with the calorimeter and the sum of the conventional losses (copper, iron and mechanical losses). According to the IEEE 112 standard [19], for induction machines with a power rating up to 90 kW stray load losses can be assumed as 1.8% of the output power. The same coefficient is used to calculate stray load losses in the PM synchronous motor.
3.5 Inverter Model

The exact calculation of power losses in a PWM inverter is very complex due to the shape of current and voltage waveforms and the non-linear characteristics of switches. The proposed IGBTs loss equations by [50] and [51] are used to predict analytically the semi-conductor power losses. The model developed considers the influence that temperature variations have on semiconductors performance, a necessary concern in automotive applications [52], [53]. This approach is based on two assumptions to simplify the calculations:

- Currents are perfect sinusoids, which is possible only with infinite pulses
- Voltage-current characteristics of switches are linearized

![Figure 3.24: Voltage and current waveforms for a power diode during a switching period](54)

Power dissipation in semiconductor devices of a 3-phase inverter is fairly generic and it can be divided in three groups:

- Conduction losses: when conducting the voltage drop across the semiconductor causes power dissipation
- Switching losses: turn-on and turn-off transitions are not instantaneous but they depend on rise and fall time of current and voltage
- Recovery losses: related to the reverse recovery characteristic of diodes

43
**Conduction losses** Instantaneous power losses during conduction are defined as

\[ p_c = u_{ce}(t)i_c(t) \quad \text{and} \quad p_d = u_d(t)i_c(t) \]

where the subscript \( c \) indicated the IGBT component and \( d \) the diode component.

Voltage is approximated by the following linear relation in order to simplify the calculations

\[ u = ri_c(t) + u_0 \quad (3.55) \]

where \( u_0 \) is the voltage threshold and \( r \) is the on-state resistance. These parameters are derived from the voltage-current graphs that can be read on the datasheet of the device. Conduction loss that occurs in one IGBT is expressed by

\[ P_c = \left( \frac{1}{8} + \frac{m_a}{3\pi} \right) r_{ce}\hat{I}_c^2 + \left( \frac{1}{2\pi} + \frac{m_a}{8} \cos(\phi) \right) u_{cea}\hat{I}_c \quad (3.56) \]

and in one diode

\[ P_d = \left( \frac{1}{8} - \frac{m_a}{3\pi} \right) r_d\hat{I}_c^2 + \left( \frac{1}{2\pi} - \frac{m_a}{8} \cos(\phi) \right) u_d\hat{I}_c \quad (3.57) \]

where \( m_a \) is the amplitude modulation index, \( \hat{I}_c \) is the peak value of the conducting current and \( \phi \) is the phase angle.

**Switching losses** Energy losses during switching transitions are calculated as

\[ E_{sw} = \frac{1}{2}V_{cc}\hat{I}_c(t_r + t_f) \quad (3.58) \]

with \( V_{cc} \) amplitude of the DC voltage. Due to the influence of operating conditions on switching waveforms, rise and fall times are not constant but they are assumed proportional to the flowing current \([50]\). Based on experimental data, they are defined as

\[ t_r = t_{rat}\frac{i_c}{I_{crat}} \quad \text{and} \quad t_f = t_{frat}\left( \frac{2}{3} + \frac{1}{3}\frac{i_c}{I_{crat}} \right) \]

where the subscript \( rat \) refers to the rated values written on datasheet.

\[ E_{sw} = \frac{1}{2}V_{cc}\left[ \frac{i_c^2}{I_{crat}}\left( t_{rat} + \frac{1}{3}t_{frat} \right) + \frac{2}{3}i_c t_f \right] \quad (3.59) \]
If the flowing current is considered a perfect sinusoid \( i_c = \dot{I}_c \sin(\omega t) \), the average energy loss of one switch during the whole period is

\[
E_{sw,av} = \frac{1}{2\pi} \int_0^\pi E_{sw} \quad (3.60)
\]

and power losses are

\[
P_{sw} = V_{cc} \dot{I}_c f_{sw} \left[ \frac{\dot{I}_c}{I_{c,rat}} \left( \frac{1}{8} t_{r,at} + \frac{1}{24} t_{f,at} \right) + \frac{1}{3\pi} t_{f,at} \right] \quad (3.61)
\]

**Recovery losses** The reverse recovery current required to turn off the diode leads to a current overshoot which generates additional losses [50]. They can be calculated with

\[
P_{rr} = V_{cc} f_{sw} \left[ 0.28 + \frac{0.38}{\pi} \frac{\dot{I}_c}{I_{c,rat}} + 0.015 \left( \frac{\dot{I}_c}{I_{c,rat}} \right)^2 \right] Q_{rr,at} + \left( \frac{0.8}{\pi} + 0.05 \frac{\dot{I}_c}{I_{c,rat}} \right) \left( \frac{2Q_{rr,at}}{I_{d,rr,at}} \right) \dot{I}_c \quad (3.62)
\]

where \( Q_{rr,at} \) and \( I_{d,rr,at} \) are the nominal values of the reverse recovery charge and current.

### 3.6 Battery Model

The basic equivalent circuit used to model the battery performance is shown in Figure 3.25. Dynamic behaviours of the battery are not considered, since they would have minimal contributions due to the relatively slow dynamics of the driving cycle.

![Battery equivalent circuit](image_url)

**Figure 3.25: Battery equivalent circuit**

The terminal voltage equation is calculated as

\[
V_T = V_{oc} - RI \quad (3.63)
\]
Both the open circuit voltage $V_{oc}$ and the internal resistance $R$ can vary depending on the state of charge level of the battery and the temperature. The influence of temperature is not considered in this research. The depth of discharge DOD can be calculated as

$$DOD_{batt} = \int \frac{I \, dt}{3600 \, C}$$

Due to the chemistry properties of batteries, the capacity $C$ depends on the discharging current and it decreases if the current is drawn more quickly [55]. The Peukert model is a simple method to predict the change in capacity at higher currents. The battery capacity at any discharge rate can be approximated as

$$C = C_{rat} \left( \frac{I_{rat}}{I} \right)^k \frac{I}{I_{rat}}$$

where the rated values refer to measurements at a specific discharge rate and $k$ is the Peukert coefficient.

Losses in the battery are caused by the current flowing in the internal resistance and they are calculated with

$$P_{loss} = RI^2$$

### 3.7 Conclusion

This chapter has explained the development of the model of an electric vehicle drivetrain. The powertrain components which lay the foundations for this research are described in this section: electrical drives, transmissions, batteries and power converters.

The electrical machines are implemented using an analytic method and their steady state operation is described using the governing equations. The layouts of the transmission units are presented and the mechanical losses in gear pairs are predicted using classic physics rules and empirical formulas based on experiments. Finally, the behaviour and the losses of power converters and of batteries are illustrated.

Given the hard task to replicate accurately all the loss mechanisms, some assumptions are made and discussed throughout this section and they are validated with the experimental data.

The vehicle model implemented is a significant achievement in itself as it goes well beyond the complexity of most of the models adopted in the literature related to electric vehicle powertrains. Furthermore it doesn’t focus on the analysis of one single component, but it attempts to investigate the interactions between several variables.

The three powertrain topologies studied in this comparison are derived from this general architecture: the single-speed single-motor, the two-speed single-motor and the in-wheel configuration. Vehicles will now be assigned a powertrain set-up, including sizing of the
electric motor, of the transmission, of the power converter and of the battery, to match
the basic requirements of typical city electric vehicles.
Chapter 4

Evaluation of Design Parameters

This chapter describes the specifications of the components discussed in the previous sections. The quantitative performance requirements of an electric vehicle mainly intended for urban utilization are established. Based on this data, a sizing study of the powertrain is conducted, without taking into account drivetrain losses. Then the operation and the efficiency of the single components are investigated.

4.1 Performance requirements

The performance requirements of the proposed vehicles are based on data of already existing BEVs of the same category on the market. The most relevant requirements are top speed, acceleration time, gradeability and driving range. For a small electric car designed for city driving, the following specifications are used:

- A top speed of 130 km/h on a flat road, based on the speed limit of highways in most of the countries
- An acceleration time from 0 – 100 km/h of 12 – 14 s
- Capability to drive a slope of 10% at a speed of 80 km/h

The basic design parameters of the concept vehicle used for the estimation of the road load and of the torque and power requirements are presented in table 4.1. These data are based around the top selling BEV models currently on the market.

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass</td>
<td>(m)</td>
<td>1200</td>
<td>Kg</td>
</tr>
<tr>
<td>Aerodynamic coefficient</td>
<td>(C_d)</td>
<td>0.3</td>
<td>-</td>
</tr>
<tr>
<td>Frontal area</td>
<td>(A_f)</td>
<td>2.05</td>
<td>m²</td>
</tr>
<tr>
<td>Tyre rolling coefficient</td>
<td>(\rho_r)</td>
<td>0.009</td>
<td>-</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>(r_w)</td>
<td>0.31</td>
<td>m</td>
</tr>
</tbody>
</table>

Table 4.1: Parameters of the concept vehicle
4.1.1 Wheel load analysis

Using the theory regarding vehicle dynamics presented in Chapter 3, the forces due to aerodynamic drag, rolling resistance, road grade and acceleration applied on a vehicle can be calculated.

In Figure 4.1 (b) the torque and power at the wheels are estimated for different driving conditions, based on the above stated performance requirements. Top speed and grading requirements demand respectively 21 kW and 32 kW. However, a more in-depth analysis is required to verify if these power levels also fulfil the acceleration time specification.

4.1.2 Acceleration performance

Figure 4.2 (a) shows the torque and speed characteristic that three electric motors which differ for the base speed would require to accelerate the vehicle from 0 – 100 km/h within 12 s. The three base speed levels have been chosen arbitrary. For all the design combinations the maximum power, that corresponds to 50.5 kW for the low base speed motor, 59.5 kW for the medium base speed and 72.5 kW for the high speed, is higher than the value determined by the top speed and the gradeability requirements.

Figure 4.2 (b) demonstrates that the lowest base speed is related to the lowest maximum power and the fastest acceleration at lower speeds. This simple evaluation assumes constant power across the entire speed range beyond the base value, which is not possible due to current limits, and it leads to overestimate the performance of Motor 1. For this reason, the power level of the high base speed machine is considered the design target for short time overload operations.
CHAPTER 4. EVALUATION OF DESIGN PARAMETERS

Figure 4.2: Torque characteristic at the wheels and vehicle speed over time during a full throttle test for three motor concepts designed to meet a $0 - 100$ km/h acceleration time in 12 s. For all the designs, the acceleration requirement results more strict with respect to the motor power than the top speed and the gradeability requirements.

4.2 Powertrain component sizing

4.2.1 Battery

The technology selected for the battery pack is the Lithium-ion. Li-ion batteries are becoming a common replacement of the lead acid types on recent electric vehicles given their superiority in terms of energy and power density. The cells used are manufactured by the Japanese company “Automotive Energy Supply Corporation (AESC)” and they are mounted on the Nissan Leaf models. The specifications are based on the measurements performed and published by the US Idaho National Laboratory in [56].

The cell terminal voltage $V_T$ as a function of the battery charge is obtained through curve fitting the data, see Figure 4.3. The open circuit voltage $V_{oc}$ has been estimated by assuming a constant voltage drop across the internal resistance during a C/3 discharge rate. The average open circuit voltage and capacity of a single cell are respectively 3.88 V and 28.8 Ah. The internal resistance is approximated to be independent of the state of charge and it equals 2.41 mΩ during charging and 2.88 mΩ during discharging operation.

In order to prolong the lifetime of the battery and avoid risks of excessive heating, it is recommended not to charge it to more than 90% of its capacity and not to discharge it below 10%. In this range, the assumption of constant internal resistance is consistent with the published datas [56].

The battery pack is designed to have an average open circuit voltage of 300 V, with a discharge power of 16.8 kW at constant C/3 rate and a pulse discharge power of 161 kW. Table 4.2 gives the main specifications of the battery pack.
4.2.2 Single-motor drive

Electric motor The AC drive mounted on the conventional single-motor powertrain is a high speed induction machine. Motor specifications are derived from the unit used in [57] and they are modified to meet the acceleration requirement previously established. Rated performance are calculated at the nominal slip of 2%. The geometry of the machine is estimated using the criteria proposed in [42]. Table 4.3 gives a summary of the main parameters and dimensions. The remaining data are given in the Appendix.

Figure 4.4 describes the torque speed characteristics and the efficiency map of the induction machine computed through the equations described in Chapter 3.4. The machine has the best performance close to the rated point, but it achieves a poor efficiency when overloaded. This is the consequence of the large copper losses that occur during high load operations.
CHAPTER 4. EVALUATION OF DESIGN PARAMETERS

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated power</td>
<td>$P_{\text{rat}}$</td>
<td>40</td>
<td>kW</td>
</tr>
<tr>
<td>Rated torque</td>
<td>$T_{\text{rat}}$</td>
<td>85</td>
<td>Nm</td>
</tr>
<tr>
<td>Rated synchronous speed</td>
<td>$\omega_{s,\text{rat}}$</td>
<td>4500</td>
<td>rpm</td>
</tr>
<tr>
<td>Rated slip</td>
<td>$s_{\text{rat}}$</td>
<td>2%</td>
<td>-</td>
</tr>
<tr>
<td>Rated phase voltage</td>
<td>$U_{\text{ratRMS}}$</td>
<td>170</td>
<td>V</td>
</tr>
<tr>
<td>Peak power</td>
<td>$P_{\text{max}}$</td>
<td>70</td>
<td>kW</td>
</tr>
<tr>
<td>Peak torque</td>
<td>$T_{\text{max}}$</td>
<td>160</td>
<td>Nm</td>
</tr>
<tr>
<td>Maximum phase current</td>
<td>$I_{\text{maxRMS}}$</td>
<td>264</td>
<td>A</td>
</tr>
<tr>
<td>Maximum speed</td>
<td>$\omega_{\text{em,max}}$</td>
<td>9000</td>
<td>rpm</td>
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<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
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</thead>
<tbody>
<tr>
<td>Stator inductance</td>
<td>$L_s$</td>
<td>0.472</td>
<td>mH</td>
</tr>
<tr>
<td>Rotor inductance</td>
<td>$L'_r$</td>
<td>0.472</td>
<td>mH</td>
</tr>
<tr>
<td>Mutual inductance</td>
<td>$M$</td>
<td>9.111</td>
<td>mH</td>
</tr>
<tr>
<td>Stator resistance</td>
<td>$R_s$</td>
<td>0.033</td>
<td>Ω</td>
</tr>
<tr>
<td>Rotor resistance</td>
<td>$R'_r$</td>
<td>0.033</td>
<td>Ω</td>
</tr>
</tbody>
</table>

Table 4.3: Induction machine specifications

Figure 4.4: Torque and power characteristic and efficiency map of the induction machine as a function of rotor speed and output torque. Three operation modes can be distinguished: constant torque operation below the rated speed, constant power operation above the rated speed and reduced power operation above the critical speed.

Figure 4.5 depicts the influence of speed and torque on the main loss mechanisms. As anticipated, copper loss results the major source of loss and it rapidly increases with the torque production. An increment of this loss component also occurs in the field weakening region at constant load, as the overloading capability of the machine progressively decreases with the growth of the speed.

53
Figure 4.5: Influence of speed and load torque on the loss components in the induction machine. Copper loss is responsible of most of the power losses.

**Single-speed transmission** As explained in Chapter 3.3 the single-speed transmission incorporates the fixed reduction ratio into the final drive. The drive ratio of the fixed reduction gear is based directly on the relationship between the top speed of the vehicle and the speed limit of the motor. The top speed of the vehicle is increased by a safety factor of 5% in order to guarantee the required performance even during not optimal driving conditions, e.g mild slope or opposing wind. The total drive ratio is obtained by:

$$ b_{tot} = \frac{\omega_{em_{max}} r_w}{1.05 \times v_{max}} $$  \hspace{1cm} (4.1) 

All the geometrical parameters have been designed based on the prototype studied in [33] and they can be found in the Appendix.

Figure 4.7 illustrates the efficiency of the reduction gear, calculated according to the theory presented in Chapter 3.3. The unit performs best at low speed high load, but it suffers from large pocketing loss as the speed increases. As expected, friction loss, a torque-dependent mechanism, dominates at high loads while pocketing loss, which is torque-independent, is the main source of loss at high speed.
CHAPTER 4. EVALUATION OF DESIGN PARAMETERS

Figure 4.6: Influence of input speed and input torque on the loss components in the fixed reduction gear. Pocketing loss dominates at high speed operations, friction loss is prevalent under high loads.

**Two-speed transmission** On the two-speed powertrain the first gear is designed to boost the wheel torque at low speed while a smaller second gear ratio can extend the top speed of the vehicle. The total drive ratio is determined by the product of the final drive ratio and the ratio of the engaged gear. The total drive ratios are respectively calculated as:

\[ b_{tot1} = \frac{T_{wheel_{max}}}{T_{em_{max}}} \]
\[ b_{tot2} = \frac{\omega_{em_{max}} r_w}{1.25 \times v_{max}} \]
CHAPTER 4. EVALUATION OF DESIGN PARAMETERS

Figure 4.8: Influence of input speed and input torque on the loss components in the two-speed transmission (continuous lines = first gear engaged; discontinuous lines = second gear engaged). The unit suffers from larger friction loss in first gear due to the higher transmitted torque, while pocketing loss increases more rapidly in second gear due to the higher output speed. Both of these observations are due to the differences in the gear ratios.

Figure 4.8 depicts the behaviour of losses over load and speed in the transmission. The first gear (continuous lines in the graphs) experiences larger friction loss, while when the second gear (discontinuous lines) is engaged pocketing loss is dominating. This comes from the choice to have a high first gear ratio to boost the torque transfer, at the expense of increased torque-dependent losses, and a lower ratio for the second gear to reach higher rotational speeds, with higher spin power losses.

Figure 4.9: Efficiency map of the two-speed transmission in first gear and second gear as a function of input speed and input torque.

The main design parameters of the two transmissions are described in table 4.4. According to the data of two comparable units developed by Oerlikon Graziano [58], the
difference between their masses is small when compared to the total mass of a vehicle. For this reason it is neglected in the simulations.

<table>
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<th>Definition</th>
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<th>Two-speed</th>
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<tr>
<td>First gear ratio</td>
<td>$b_{g1}$</td>
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<td>3.93</td>
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<tr>
<td>Second gear ratio</td>
<td>$b_{g2}$</td>
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<tr>
<td>Final drive ratio</td>
<td>$b_{fd}$</td>
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<td>3.70</td>
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<tr>
<td>Weight</td>
<td>$m_{trans}$</td>
<td>25 kg</td>
<td>38 kg</td>
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</table>

Table 4.4: Transmission systems specifications

Figure 4.10: Comparison of power losses and maximum available torque at the wheels with the two transmissions. In (a) continuous lines represent power losses in the single-speed unit, discontinuous lines refers the two-speed unit. The high first gear ratio of the two-speed transmission provides evident performance benefits versus the single-speed system, at the expense of higher losses. Pocketing loss is greatly reduced in the two-speed gearbox in second gear.

Figure 4.10 (b) demonstrates the performance benefits deriving from the adoption of a two-speed gearbox over fixed reduction gear. This is only a starting estimate because it doesn’t take into account the power losses through the powertrain. The maximum available torque at the wheels is almost two (2375 Nm) the peak value of the single-speed drive (1257 Nm) and the top speed that the vehicle can theoretically reach on a flat road increases from 136 km/h to 160 km/h.

We can observe that the two-speed torque curves have a discontinuous part, due to the limited performance of the motor. It is recommended to provide overlapping constant power regions to maintain an acceptable drivability during gearshifts, but this is beyond the scope of this thesis and it has no effects on the type of simulations conducted here.

The efficiency maps of the single-speed and two-speed drive systems, which include the electric motor and the transmission unit, are given in Figure 4.11 as a function of
wheel torque and vehicle speed. A shifting speed of 45 km/h is used for the two-speed drive.

Figure 4.11: Efficiency maps of the single-speed drive system (motor and transmission) and of the two-speed drive system

### 4.2.3 In-wheel drives

The outer rotor permanent magnet machines are designed in such a way to be applied as direct drives and therefore to meet the performance requirements at the wheel without any mechanical transmission system. The machine geometry, dimensions and parameters are determined using the optimisation approach proposed in [60] and they are given in Table 4.5 The remaining specifications can be seen in the Appendix

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<th>Definition</th>
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<tr>
<td>Rated torque</td>
<td>$T_{\text{rat}}$</td>
<td>450</td>
<td>Nm</td>
</tr>
<tr>
<td>Rated synchronous speed</td>
<td>$\omega_s$</td>
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<td>RPM</td>
</tr>
<tr>
<td>Rated phase voltage</td>
<td>$U_{\text{rat},\text{RMS}}$</td>
<td>93</td>
<td>V</td>
</tr>
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<td>Peak power</td>
<td>$P_{\text{max}}$</td>
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<td>kW</td>
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<tr>
<td>Peak torque</td>
<td>$T_{\text{max}}$</td>
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<td>Nm</td>
</tr>
<tr>
<td>Maximum phase current</td>
<td>$I_{\text{max},\text{RMS}}$</td>
<td>105</td>
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<td>Maximum speed</td>
<td>$\omega_{\text{max}}$</td>
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<td>Leakage inductance</td>
<td>$L_s$</td>
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<td>Magnetizing inductance</td>
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<td>mH</td>
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<tr>
<td>Stator resistance</td>
<td>$R_s$</td>
<td>0.05</td>
<td>$\Omega$</td>
</tr>
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</table>

Table 4.5: In-wheel direct drives specifications

Figure 4.13 gives the power losses as a function of torque and speed. Copper loss is predominant at high loads and it grows in a quadratic fashion with the torque, due to the torque being linearly proportional to the armature current. As the speed increases core loss becomes the main source of loss.
Figure 4.12: Influence of speed and load torque on the loss components in the outer rotor PMSM. It is found that iron loss are dominant at high speed, while large copper loss occurs for torque levels above the rated value.

Figure 4.13: Efficiency map of the in-wheel motor as a function of output torque and rotor speed. The discontinuous lines represent the nominal and the maximum power levels, respectively at 20 kW and 35 kW.

We can observe that the in-wheel direct drives could suffer from a bulky package or expensive manufacturing processes due to the outer rotor configuration, but the absence of mechanical gearings contributes to the high efficiency of the system.

### 4.2.4 Power electronic converter

The power electronic system includes a DC-DC boost converter and a DC-AC inverter. The use of a DC-DC converter between the battery and the inverter is common practice on electrified vehicles. This configuration gives more flexibility to the drive system as it
can transform the variable output voltage of the battery into a regulated DC voltage. In this work, the vehicles are equipped with a boost converter to step up the battery voltage to obtain the level required by the DC-link. It is assumed that the DC-DC converter has an efficiency of 100%.

To select a proper DC-AC inverter module, the first thing to do is to define the voltage rating of the component. Owing to voltage overshoots that can happen during switching transitions, it is recommended to use a power module with a higher rating than the maximum voltage that could be applied during normal conditions \[45\]. In modern BEVs the voltage rating of the DC-link goes from 400V to 650 V \[45\]. A fixed DC-link voltage of 400 V is chosen for both the powertrains. This gives a desired voltage rating of the inverter in the order of 600 V. The current rating of the DC-AC converter is selected with respect to the maximum rms value of the phase current of the electric drives. It is assumed that it has to be $I_{\text{rat}} = \frac{3}{2} I_{\text{max\,RMS}}$ in order to have a reasonable margin. The Infineon FS400R07A1E3 H5 is used to drive the induction machine on the single motor powertrain, given its current rating of 400 A. For the in-wheel motors two converter modules with a current rating of 200 A are chosen (Infineon FS200R06KE3). Figure 4.14 (a) and (b) show the efficiency map of the inverter calculated using the data provided by the manufacturer in \[61\] and \[62\] at the temperature of 125°C.

![Efficiency map of the FS200R06KE3 converter](image)

![Efficiency map of the FS400R07A1E3 H5 converter](image)

Figure 4.14: Comparison of the efficiency map of the two converter modules selected as a function of the rms single-phase values of voltage and current. The efficiency is estimated using the analytic equations discussed in Chapter 3

### 4.2.5 Regenerative braking strategy

A unique feature of electric machine is their ability to transfer power in both directions, from the source to the load as a motor or in the opposite way as a generator. This characteristic promotes the implementation of a regenerative braking system that can recover and retain substantial amounts of braking energy that otherwise would be wasted. At present regenerative braking is one of the most effective approach to extend
CHAPTER 4. EVALUATION OF DESIGN PARAMETERS

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<th>Unit</th>
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<td>Voltage rating</td>
<td>$V_{ce}$</td>
<td>600</td>
<td>600</td>
<td>V</td>
</tr>
<tr>
<td>IGBT collector-emitter threshold voltage</td>
<td>$u_{ce0}$</td>
<td>0.7</td>
<td>0.7</td>
<td>V</td>
</tr>
<tr>
<td>Diode threshold voltage</td>
<td>$u_{d0}$</td>
<td>0.7</td>
<td>0.8</td>
<td>V</td>
</tr>
<tr>
<td>IGBT collector-emitter on-state resistance</td>
<td>$r_{ce}$</td>
<td>4.167</td>
<td>2.468</td>
<td>mΩ</td>
</tr>
<tr>
<td>Diode forward resistance</td>
<td>$r_d$</td>
<td>3.75</td>
<td>2</td>
<td>mΩ</td>
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<tr>
<td>Nominal current</td>
<td>$I_{cm}$</td>
<td>200</td>
<td>400</td>
<td>A</td>
</tr>
<tr>
<td>IGBT nominal rise time</td>
<td>$t_{rn}$</td>
<td>40</td>
<td>80</td>
<td>ns</td>
</tr>
<tr>
<td>IGBT nominal fall time</td>
<td>$t_{fn}$</td>
<td>70</td>
<td>70</td>
<td>ns</td>
</tr>
<tr>
<td>Reverse recovery charge</td>
<td>$Q_{rrn}$</td>
<td>17</td>
<td>30</td>
<td>µC</td>
</tr>
<tr>
<td>Reverse recovery current</td>
<td>$I_{rrn}$</td>
<td>230</td>
<td>280</td>
<td>A</td>
</tr>
</tbody>
</table>

Table 4.6: Specifications of the FS200R06KE3 and FS400R07A1E3 H5 inverter modules at 125°C

the driving range of EVs.

Generally the required braking torque of a vehicle exceeds the capability of the regenerative braking system, especially during emergencies. Therefore it is necessary to implement a braking control strategy which combines the forces of conventional friction brakes and of the motor. To ensure safety and smooth drivability, regeneration is deactivated at very low speed, below 5 km/h, and for demanded braking torques higher than 90% of the motor peak value. During these conditions only the friction brakes operate. The division of braking torque is set at the typical ratio of 70:30 between the front axle, where the regenerative and the hydraulic systems cooperate, and the rear axle, only equipped with friction brakes. The regenerative system is assumed to have an efficiency of 80%.
The total efficiency of the powertrain, including electric machine, power converter and transmission, is given in Figure 4.15 for each architecture topology.

Figure 4.15 shows that the in-wheel drivetrain achieves a significantly higher efficiency than both the single-motor layouts for almost every combination of torque and speed. It also suggests a more efficient operation of the single-motor drivetrain when equipped with a two-speed gearbox, in addition to the evident boost provided with regards to maximum wheel torque and speed. However, it is not possible to conclude whether these slight improvements will translate in an effective and evident lower energy consumption of the vehicle during the simulation of drive cycles.
4.3 Conclusion

The aim of this chapter is to carry out a sizing study of the powertrain components and obtain a final vehicle model designed to match a typical urban electric car. Power losses in each component are calculated with analytic formulas and discussed in different operating conditions to examine their relationships with vehicle dynamics such as tractive force and speed.

It was found that the acceleration requirements establish the most stringent power and torque requirements and that they can be satisfied with more than one combination of maximum wheel torque and power, depending on the base speed of the motor.

The analysis of the source of losses in the drivetrain components shows specific correlations between the loss mechanisms and the operating conditions. It was found that the performance of the induction machine is mainly constrained by large copper losses. In the outer rotor PMSMs copper loss dominates during high torque operations, while the drives suffer from large iron loss at high speed.

The two transmission designs result in similar loss mechanisms, with the two-speed gearbox being affected by larger torque-dependent and speed-dependent losses when running with the first gear than the fixed reduction. Pocketing loss can be significantly reduced with the two-speed unit in second gear. Both of these results are due to the differences in the gear ratios between the transmission systems.

The computation of the total powertrain efficiency indicates the in-wheel topology as the best configuration to optimise energy consumption. It was found that the two-speed gearbox can theoretically provide a significant boost to the vehicle performance, with regards to maximum wheel torque and top speed, and it can extend the operating area where the efficiency of the drivetrain is highest.

A more in-depth investigation of performance and energy consumption of the vehicles is carried out in the following chapter to verify whether these improvements provide effective advantages when even more variables are considered.
Chapter 5

Simulation results

This chapter tests performance and energy consumption of the three drivetrain architectures implemented. The chapter is organised in the following manner. Section 5.1 discusses the simulation approach adopted in this thesis. In section 5.2 an acceleration test is carried out to analyse the vehicle performance with each transmission system, the fixed reduction gear and the two-speed transmission, and quantify the benefits of multi-gear ratios. Section 5.3 explains the selection of the driving cycles and provides the simulation results of the three powertrains. Section 5.4 gives the concluding remarks.

5.1 Simulation approach

![Simplified block diagram of the EV models](image)

Figure 5.1: Simplified block diagram of the EV models

A simplified presentation of the vehicle model built in the Matlab/Simulink environment is given in figure 5.1. When developing models there is always a trade-off between accuracy, simulation time, size and complexity. When considering a model for simulating long driving cycles it would be beneficial to reduce the complexity of the model and consequently the simulation time at the expense of the accuracy of the results ([63], [35]). While this methodology is not suitable for more complex tasks such as controller
validation, it provides insight into the workings of a powertrain and help the user understand how the components are interacting.

The modelling approach proposed in this research is based on the backward facing methodology, where the speed profile of the driving cycle is used as input variable for the calculation of the traction force and the energy consumption. This method neglects all the transient phenomena and it assumes that the driving profile will be exactly followed, without checking if the vehicle is actually able to meet the the power requirement determined by the speed trace. Despite these limitations, the low complexity and the fast simulation runtimes make backward facing models suitable for the estimation of energy consumption and efficiency of a vehicle when the driving dynamics are not of interest.

All the simulations are executed with a fixed time step of 0.01 s. The gearshift strategy implemented uses the vehicle speed as input to determine when to change gear. The simplicity of this method is an additional variable which affects the results of the simulations. More complex and optimized shifting schedules which promise better performance and efficiencies are being examined in specific studies [31]. The shifting speed of the two-speed drivetrain is set at 45 km/h.

5.2 Performance

Multi-speed transmissions are often claimed as a superior technology than fixed reductions with regard to improving the performance of electric vehicles. The results from the acceleration test confirm this hypothesis, but the benefits are lower than what speculated in Chapter 4.

The test simulates a full throttle acceleration of the single-motor model with the fixed reduction and the two-speed gearbox for a period of time of 30 s.

Figure 5.2 shows that the two-speed transmission provides a faster acceleration rate at low speed, due to the first gear ratio being higher than the single-speed, with a 0 – 50 km/h time 27% faster than the opponent. The gain in the available wheel force when the first gear is engaged can be seen in figure 5.2 (b). Whereas, above the upshift speed the single-speed model can ‘catch up’ thanks to the reduction in the wheel torque capability of the two-speed powertrain when running in second gear. This translates in a smaller difference in the 0 – 100 km/h acceleration time between the two architectures, which corresponds to 16% pro the two-speed. As expected, the lower value of the second gear than the fixed reduction ratio extends the theoretical maximum speed of the car.

It can be noticed that the high ratio of the first gear provides an increment in the tractive force smaller than what expected in the design process (1750 Nm instead of 2375 Nm estimated in Chapter 4). Below the upshift speed, the total drive ratio of the two-speed model is approximately two times bigger than the fixed reduction ratio, respectively 14.6 and 7.7, but this doesn’t translate in a twice as high wheel force.
Figure 5.2: Comparison of the single-speed and the two-speed powertrains during a full throttle acceleration test. The two-speed vehicle can accelerate faster and it can reach a higher top speed thanks to the multiple gear ratios.

Figure 5.3: Transmission of the propulsive force through the powertrain of the conventional single-motor vehicles during the acceleration test. The force is calculated at the output of the main components that form the propulsion system. A great reduction is shown out of the motor shaft in the two-speed model in first gear.

The transmission of the propulsive force through the powertrain components, from the motor to the wheels, is described in Figures 5.3 (a) and (b). There is an evident reduction in the transmitted force out of the motor shaft in the two-speed and, in a much smaller degree, in the single-speed models. This missing energy is not only dissipated in losses but it is mostly stored in the powertrain inertias. When the vehicle is accelerating, part of the tractive power is used to spin the rotating elements of the drivetrain and
change their current state of motion. Due to the dependence of the inertia effect on the 
rate of change of the speed, this phenomenon is greater on the two-speed powertrain 
where the motor shaft reaches higher angular accelerations. Peaks correspond to 220 \( \text{rad/s}^2 \) in the two-speed model versus 64 \( \text{rad/s}^2 \) in the single-speed.

The inertial energy is not accounted as a drivetrain loss, since it can be theoretically 
fully recovered when the vehicle is decelerating. As found in Chapter 4, when compared 
with the single-speed unit, the two-speed transmission achieves a lower efficiency in first 
gear and a higher efficiency in second gear. This is mostly caused by the change of the 
speed-dependent losses, larger when running in the first gear and reduced in second gear.

Figure 5.4: Comparison of efficiency and power losses between the single-speed and the two-speed 
transmissions. The slightly lower efficiency is not the main responsible of the loss in propulsive 
force found in the two-speed unit in first gear.

In conclusion the single-speed vehicle achieves a 0 – 100 km/h time slightly slower 
than the value chosen in the design section, set at 12 s, whereas the two-speed model can 
meet the acceleration requirement thanks to the increased maximum available torque 
at the wheels. Although there are significant performance benefits deriving from the 
adoption of a multi-speed gearbox on electric vehicles, during a full pedal acceleration 
test they are limited by the dynamics of the powertrain inertias.
5.3 Driving cycles

The formulation of a methodology for robust and equal comparisons of competing vehicle designs is a major concern for the automotive industry. A driving cycle is a fixed schedule of driving operations conducted by the vehicle on a chassis dynamometer to assess its performance, mainly in terms of energy consumption and polluting emissions. This methodology allows test procedures which are repeatable and standardized for laboratory analysis of cars.

Vehicles of different categories are used during various driving conditions, by different types of drivers and for numerous purpose. Each of these scenarios demands for specific driving schedules and it is the reason for the vast database of existing cycles.

It is well known the great influence of the driving profile on the operating characteristics of EVs [64]. In their comparison, Wang at all [23] included eight different cycles with distinct driving dynamics, such as the HWFET originally created to simulate highway driving conditions and the NYCC, characterized by frequent stop and go situations typical of urban traffic. Simulations results prove the strong influence of the driving style on EVs design and operating conditions: variations up to 75% can occur in energy consumption during different schedules, which means that a BEV could achieve just half of the promised all electric range under certain driving conditions, thus amplifying a crucial deficiency of electric mobility. Hence, the necessity to identify those schedules which best reproduce the possible driving scenarios of a BEV, mainly designed for use in urban areas.

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<th>Driving cycle</th>
<th>Total distance (m)</th>
<th>Total time (s)</th>
<th>% of time driving (%)</th>
<th>% of time accelerating (%)</th>
<th>% of time cruising (%)</th>
<th>Ave. speed (km/h)</th>
<th>Max speed (km/h)</th>
<th>Max pos. acc. (m/s²)</th>
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Table 5.1: Driving cycles parameters

The selected driving cycle have been divided in three categories: Urban, Rural and Highway. The purpose of this arbitrary classification, based on the speed profile, is to evaluate the vehicle models under certain driving characteristics. The descriptive parameters of the cycles calculated according to the Art.Kinema method [65] are listed in table 5.1 the complete graphs of speed and acceleration over time for each cycle can be seen in Appendix.

The three vehicle concepts are tested over these schedules and they are compared with respects to discharging power, energy consumption and efficiencies of the total
CHAPTER 5. SIMULATION RESULTS

powertrain and of the individual mechanical and electrical systems.

5.3.1 Urban

<table>
<thead>
<tr>
<th></th>
<th>Average discharging power (kW)</th>
<th>Max discharging power (kW)</th>
<th>Total efficiency (%)</th>
<th>Mechanical efficiency (%)</th>
<th>Electrical efficiency (%)</th>
<th>Energy consumption (Wh)</th>
<th>Energy consumption w/o rb* (Wh)</th>
</tr>
</thead>
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<td><strong>ECE-15</strong></td>
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<td>90.5</td>
<td>(−4.4%)</td>
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<td>79.6</td>
<td>(9.3%)</td>
<td>133</td>
</tr>
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<td><strong>NYCC</strong></td>
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</tr>
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<td><strong>Artemis urban</strong></td>
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<td></td>
</tr>
<tr>
<td>Single-speed</td>
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<td>34.1</td>
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<td>92.1</td>
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<td>71.9</td>
<td>(10.4%)</td>
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</table>

Table 5.2: Simulation results of urban cycles - ([value]) is the percentage difference with respect to the single-speed - *energy consumption without the regenerative braking system

The Urban cycles attempt to reproduce driving conditions characterized by either aggressive and moderate accelerations at limited vehicle speed. Table 5.2 presents the results of each powertrain and the percentage change with respect to the single-speed, chosen as reference model.

Figure 5.5: Energy distribution in the vehicles over the Artemis urban. During the Urban cycles, the electric drive is responsible of most of the losses - (Diff=Differential, Gear=Gearbox, EM=Electric motor, Inv=Inverter, Batt=Battery, AUX=Auxiliary system, Tract=Traction)
The in-wheel drivetrain always achieves the lowest energy consumption and a significantly higher total efficiency than the other architectures. The absence of mechanical transmission components minimises losses due to energy conversion and gives great benefits regarding lower motor power demands and a superior energy utilization.

When looking at the main sources of energy loss in the path from the battery to the wheels inside an electric drivetrain, the electric motors account for the biggest contribution during urban driving. In-wheel motors outclass the induction machine in terms of efficiency within limited operating velocities, especially when high torque is demanded. This translates into evident better performance of the in-wheel configuration than the conventional single-motor when aggressive accelerations are frequently executed at low vehicle speed, typical during the NYCC and the Artemis Urban. This gap is reduced during less aggressive (slower) driving conditions.

![Figure 5.6: Average efficiency of the main components of each powertrain over an aggressive (a) and a low-performance (b) schedule. The influence of driving conditions on their operation is evident. In either the cases, the in-wheel drivetrain outperforms the conventional single-motor models](image)

Figure 5.7 shows the influence of the driving pattern on the loss mechanisms in the electrical drives. The NYCC driving cycle, which includes fast accelerations, is compared with a lower performance schedule, the ECE cycle. The first observation is that core loss is the main source of losses in the in-wheel motors, while they are reduced in the induction machines. Furthermore, it is evident that copper loss greatly depends on the magnitude of the output torque and it is much larger in the induction machine equipped on the single-speed model. A reduction of copper loss is found in the two-speed vehicle. This originates from the high ratio of the first gear which cuts the motor torque demand and decreases significantly copper loss. The related higher rotor speed intensifies core and friction losses than the fixed gear configuration.
CHAPTER 5. SIMULATION RESULTS

Figure 5.7: Energy loss mechanisms in the electrical drives. During urban driving, aggressive schedules (a) cause high copper loss, iron loss becomes more relevant with a less aggressive driving style (b).

Figure 5.8: Motor operating points on the conventional single-motor vehicles over the Artemis Urban. The adoption of a two-speed gearbox reduces significantly the level of torque demanded to the motor and it moves its operation towards a more efficient area. However the resultant efficiency gain is minimal.

Between the two conventional single-motor topologies, the single-speed model always achieves a lower energy consumption. The adoption of a two-speed gearbox improves the motor efficiency when the vehicle is starting and when medium-high torque levels are required at low velocities. Figure 5.8 shows that it can move the operating points of the
motor to the most efficient region during driving conditions typical of urban scenarios. However this small benefit is always compensated or even exceeded by the additional losses in the more complex transmission system. In conclusion no significant differences are found in the overall efficiency of the single-motor drivetrain between the single-speed and the two-speed configurations.

Figure 5.9: Energy loss mechanisms in the transmission systems. The two-speed transmission generates higher losses than the fixed reduction gear during city driving.

It is interesting to notice that the regenerative braking is most effective on the two-speed powertrain. This comes from the increased maximum torque at the wheels, and as a consequence increased maximum regenerative capability, when the first gear is engaged and the possibility to recover most of the energy stored in the powertrain inertias, which otherwise would be lost. We can observe that with no regenerative braking system the percentage change in energy consumption of the two-speed model versus the single-speed broadens due to larger mechanical losses and inertial phenomena.
5.3.2 Rural

<table>
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<th>Average discharging power (kW)</th>
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<th>Total efficiency (%)</th>
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<th>Electrical efficiency (%)</th>
<th>Energy consumption (Wh)</th>
<th>Energy consumption w/o rb* (Wh)</th>
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<td>(6.2%)</td>
<td>(2.8%)</td>
<td>(6.2%)</td>
<td>(2.8%)</td>
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<td>(−6.7%)</td>
<td>(−10.4%)</td>
<td>(−7.6%)</td>
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<td>(0.6%)</td>
<td>(0.4%)</td>
<td>(0.7%)</td>
<td>(0.8%)</td>
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<td>-</td>
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<td>(5.9%)</td>
<td>(2.0%)</td>
<td>(1.4%)</td>
<td>(0.5%)</td>
<td>(0.7%)</td>
<td>(0.3%)</td>
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<td>-</td>
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<tr>
<td></td>
<td>(1.3%)</td>
<td>(3.8%)</td>
<td>(1.2%)</td>
<td>(0.1%)</td>
<td>(0.7%)</td>
<td>(0.6%)</td>
<td>(0.8%)</td>
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<td>(−9.1%)</td>
<td>(8.8%)</td>
<td>(0.1%)</td>
<td>(−5.4%)</td>
<td>(−5.5%)</td>
<td>(−5.5%)</td>
</tr>
</tbody>
</table>

Table 5.3: Simulation results of rural cycles - ([value]) is the percentage difference with respect to the single-speed - *energy consumption without the regenerative braking system

Over the rural cycles the vehicle models are tested through a wider range of speed levels, including low, moderate and high speed conditions, to replicate different scenarios typical of extra-urban driving.

Simulation results show smaller improvements in energy consumption for the in-wheel architecture. The performance of wheel hub motors drops with the growth of vehicle speed. This pattern can be seen in Figure 5.10. When compared to the induction machines equipped on the single-motor layouts, the in-wheel direct drives achieve similar performance over those driving cycles with low-to-medium speed levels, such as the FTP and the NEDC. But during faster schedules, like the Artemis Road or the WLTP Class 3, they result in a lower efficiency and a higher overall dissipation of energy, because of extremely large core loss.

Whereas the electric drive is still the main source of loss in an electric drivetrain, a larger part of energy is dissipated in the transmission when compared to the Urban tests. As found in Chapter 4 speed-dependent losses, mainly due to pocketing mechanisms, have critical influence on the operation of the transmission systems, to a greater extent than torque-dependent friction losses. This explains the lower efficiency of the units.
CHAPTER 5. SIMULATION RESULTS

Figure 5.10: Average efficiency of the main components of each powertrain over a moderate speed (a) and a higher speed (b) rural schedule. The single-motor vehicles can reduce the efficiency gap with the in-wheel drivetrain observed in the urban cycles during faster driving conditions.

Figure 5.11: Energy loss mechanisms in the electric drives over a moderate speed (a) and a higher speed (b) rural schedule. The efficiency of the in-wheel direct drives drops with the vehicle speed, due to increasing core loss.

There are no significant differences in energy consumption and efficiency between the two conventional single-motor powertrains. The two-speed gearbox, with a second gear ratio lower than the fixed reduction ratio, reduces spin dependent power losses above the upshift speed, but the resultant benefits are so small that they do not provide any effective improvements with respect to total energy demand or total efficiency of the drivetrain.
CHAPTER 5. SIMULATION RESULTS

Figure 5.12: Energy loss mechanisms in the transmission systems. Speed-dependent pocketing loss dominates over the rural cycles. With a second gear ratio lower than the fixed ratio of the single-speed unit, the two-speed gearbox results in lower losses during faster driving conditions thanks to the reduced rotational speed of the internal mechanical components.

5.3.3 Highway

<table>
<thead>
<tr>
<th></th>
<th>Average discharging power (kW)</th>
<th>Max discharging power (kW)</th>
<th>Efficiency (%)</th>
<th>Mechanical efficiency (%)</th>
<th>Electrical efficiency (%)</th>
<th>Energy consumption (Wh)</th>
<th>Energy consumption w/o rb* (Wh)</th>
</tr>
</thead>
<tbody>
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<td>73.5</td>
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<td>89.3</td>
<td>84.0</td>
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<td>2117</td>
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<td>76.7</td>
<td>-</td>
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<td>28.4</td>
<td>76.2</td>
<td>89.3</td>
<td>84.0</td>
<td>2042</td>
<td>2117</td>
</tr>
<tr>
<td></td>
<td>(-2.7%)</td>
<td>(-9.5%)</td>
<td>(4.2%)</td>
<td>-</td>
<td>(-8.9%)</td>
<td>(-1.3%)</td>
<td>(-1.7%)</td>
</tr>
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<td><strong>In-wheel</strong></td>
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<td>79.4</td>
<td>-</td>
<td>79.4</td>
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<td>5611</td>
</tr>
</tbody>
</table>

Table 5.4: Simulation results of highway cycles - ([value]) is the percentage difference with respect to the single-speed - *energy consumption without the regenerative braking system

Among all the drive cycles included in this comparison the Highway schedules have the highest average vehicle speed.

The single-speed model has the largest energy consumption and worst energy utilization in all the highway driving cycles, while the two-speed and the in-wheel architectures achieve similar results. It is interesting to notice that, despite the poor efficiency of outer rotor electric drives under these driving conditions ($\approx -8\%$ than the induction machine), the in-wheel configuration can still provide good performance thanks to the absence of a mechanical transmission and the related energy losses.
CHAPTER 5. SIMULATION RESULTS

Figure 5.13: Energy distribution in the vehicles over the Artemis motorway. The amount of energy dissipated in the transmission systems becomes relevant over the Highway drive cycles - (Diff=Differential, Gear=Gearbox, EM=Electric motor, Inv=Inverter, Batt=Battery, AUX=Auxiliary system, Tract=Traction)

The performance of the conventional single-motor drivetrains is constrained by the low efficiency of the mechanical components. As anticipated in the rural tests, during highway driving the transmission becomes a major source of energy loss in the drivetrain due to the large speed-dependent losses. The lower rotational speed of the internal components in the two speed unit in second gear can limit spin power losses. This provides a higher efficiency of the system than the less complex fixed gear.

Figure 5.14: Average efficiency of the main components and energy loss mechanisms in the transmissions over the HWFET. The high-speed motor equipped on the conventional powertrains performs significantly better than the in-wheel direct drives under fast driving conditions

However, the adoption of a gearbox doesn’t provide any benefits with respect to the
motor operating points.

Figure 5.15: Motor operating points on the conventional single-motor vehicles over the Artemis motorway. The two-speed model can satisfy all the torque-speed combinations demanded to the motor during extreme driving conditions, precluded on the single-speed drivetrain.

It is found that the two-speed vehicle was able to satisfy power demands which exceeded the capability of the induction machine on the single-speed drivetrain during the Artemis Motorway schedule. This could have led to small errors in the results of the last-mentioned architecture over this drive cycle. As mentioned in the introduction of this chapter, it is one of the limits of the backward facing approach.
5.4 Conclusion

This section presented the simulation results of the powertrain models designed and sized in the previous chapters. An acceleration test was carried out to assess the performance benefits of a multi-speed gearbox on BEVs. Energy consumption and efficiency of the three vehicle concepts were then measured over standard driving cycles.

The novel in-wheel topology appears to be a superior architecture in terms of energy utilization. Outer rotor PM machines suit city driving well, a crucial achievement for cars designed to be used primarily in urban areas, and the absence of mechanical components seems an optimal solution to minimise losses due to energy conversion. The in-wheel drivetrain achieved satisfactory total efficiency, comparable to the other configurations, even over the highway schedules, which are the most adverse operating conditions for low-speed direct drives.

The adoption of a two-speed gearbox is a successful strategy to extend the range of available torque and speed at the wheels and increase the maximum performance of a battery electric vehicle. When compared to a fixed-gear system, the multiple gear ratios can influence the operation of the components of an electric drivetrain in terms of losses:

- when the high first gear is engaged, it reduces torque-dependent losses at the expense of speed-dependent losses
- with the lower second gear engaged, it reduces speed-dependent losses at the expense of torque-dependent losses

It should be noted that this pattern is fairly small and it is most visible in the two opposite scenarios, either high-torque low-speed or low-torque high-speed operations. In conclusion over an entire driving cycle that also includes less extreme operating conditions, it doesn’t provide any significant advantages with respect to the powertrain efficiency or the energy consumption. On the contrary, the two-speed configuration always achieves worse efficiency and energy consumption than the single-speed model during Urban cycles, which are key factors for city cars.

It was also found that driving patterns have a great influence on drivetrain performance and this confirms the importance of advanced knowledge and evaluation of typical driving habits when designing a vehicle.
Chapter 6

Conclusions and recommendations

6.1 Conclusion

Different powertrain architectures designed for electric vehicles have been modelled and evaluated with regards to performance and energy consumption.

The literature review presented the state-of-the-art research concerning drivetrain configurations, motors and transmission for battery electric vehicles. Although novel powertrains that promise higher efficiency and energy saving have been developed, the market of pure electric vehicles is still monopolized by the single-speed single-motor architecture, derived from conventional internal combustion engine vehicles. Thus the decision to compare the standard single-motor powertrain with the new and innovative in-wheel layout.

While the in-wheel prototype is equipped with high power density PM synchronous machines, the induction machine has been preferred for conventional architectures, given its cost benefits, a crucial parameter for automotive companies. Further research was conducted to investigate the potential and often controversial advantages of multi-speed transmissions on electric vehicles. Current literature suggests that a two-speed system is the best compromise between performance, layout complexity and costs.

Most studies which analyze electric vehicles energy consumption adopt complex simulation tools which don’t provide a clear picture of the relationships between vehicle dynamics and power losses. Here the main components which constitute electric drivetrains have been derived from the governing equations that describe operation and loss mechanisms. The proposed approach allows the user to track the flow of energy through the vehicle and to analyze the influence that dynamic operating conditions have on each loss type. This gives the reader insight into the workings of a powertrain and helps to understand how the components are interacting. Backward-facing simulations have been conducted in the Matlab/Simulink environment to evaluate the vehicle models over different driving scenarios.

When comparing the three powertrain alternatives, the in-wheel layout is a superior
technology to improve efficiency and energy utilization. It was found that this architecture excels in urban driving, but it also achieves good energy consumption during adverse operations, such as high speed highway conditions. The absence of mechanical components provides minimum losses due to energy conversion processes. The complexity and the cost of this technology, however, are crucial technical barriers, which are still limiting its diffusion.

The acceleration test executed shows that there are evident performance gains in using a two-speed gearbox on conventional single-motor drivetrains. These achievements are determined by the high first gear ratio designed in such a way to increase the available wheel torque. On the other hand, the simulations conducted in this research don’t show any significant reduction in the energy consumption over the single-speed for most of the cycles. The small benefits deriving from the more efficient motor operation are counterbalanced by higher mechanical losses of the transmission system.

Based on these results a fixed reduction gear seems the best option for an electric city vehicle with low performance. The in-wheel powertrain achieved outstanding results, especially during urban driving, but its adoption on commercial electric vehicles on a large scale is currently not feasible as previously explained. Further research and development of this technology are therefore strongly recommended to confirm the theoretical results achieved here and to overcome its current barriers.

6.2 Future work and recommendations

Over the course of this research, the assumptions made to solve problems give rise to a number of possible questions for further investigation. Recommendations for future research are given below.

- A more complex simulating approach could be adopted, such as the forward-facing type, to include transient phenomena into the analysis
- The vehicle model could be extended to a higher level of details. For example, by implementing a wheel slip model and by including more auxiliary components used on automotive powertrains
- Design and geometry of the transmission systems could be optimized to minimize losses
- Different and advanced gearshift schemes on the two-speed vehicle could be explored for possible efficiency improvements
- It would be interesting to investigate the effect of temperature by implementing a thermal model. The dissipation of heat is a major concern for electric vehicles; the operation of motors, especially with permanent magnets, and batteries is usually constrained in a temperature range to avoid deterioration and their efficiency drops with the raise of temperature
• Apart from energy consumption, efficiency and performance, aspects such as powertrain complexity, costs and lifecycle greenhouse gas emissions of the BEV design choices are of vital importance and could be investigated for an extended evaluation.
Appendices
Appendix A

Driving Cycles

The driving cycle data are collected from the simulation tools for use with Matlab/Simulink Driving Cycle [66] and ADVISOR [67].

A.1 Urban cycles

![ECE 15 Speed and Acceleration](image1)

![NYCC Speed and Acceleration](image2)

![Artemis Urban Speed and Acceleration](image3)
APPENDIX A. DRIVING CYCLES

A.2 Rural cycles

Figure A.2: Speed and acceleration over time of the Rural driving cycles
A.3 Highway cycles

Figure A.3: Speed and acceleration over time of the Highway driving cycles
Appendix B

Drivetrain Propulsion Modeling

This appendix deals with the calculation of the equations governing the powertrain for each configuration. The analysis of the torque transfer is carried out using the Newton’s second law for rotational systems [24]. The aim is to express the motor torque in terms of the vehicle speed and acceleration, input of the backward facing model implemented.

As described in Chapter 3.2, the longitudinal acceleration of a rigid mass representing the vehicle body can be expressed as:

\[ m \frac{dv}{dt} = F_x - F_R \]  \hspace{1cm} (B.1)

where \( m \) is the mass of the vehicle, \( F_x \) is the tractive force and \( F_R \) is the sum of all the road load forces. Since the three powertrains have the same vehicle body, this relation is valid for all the models.

B.1 Single-speed drivetrain

The schematic of the single-speed transmission considered in this thesis is depicted in Figure [5.1]. It consists of a single-stage spur gear reduction where the motor torque is transmitted from the main shaft to the axle shafts through the final drive gear pair, which determines the reduction ratio, and the differential.
The subscripts $em$, $fd$, $d$, $a$ and $w$ refer respectively to the electric motor, the final drive, the differential, the drive axle and the wheel. The torque balance equation at the primary shaft is calculated with:

$$T_{em} - T_{loss_{em}} - F_1 R_1 = (J_{em} + J_{1s}) \frac{d\omega_{em}}{dt}$$  \hfill (B.2)

$T_{em}$ and $\omega_{em}$ are the motor torque and speed, $J_{em}$ and $J_{1s}$ are the moments of inertia of the motor and of the primary shaft, $F_1$ is the force transmitted to the differential and $R_1$ is the radius of the pinion (input gear) of the final drive. $T_{loss_{em}}$ takes into account the mechanical losses, friction and windage, of the electric motor. The subscripts $em$, $d$, $a$ and $w$ refer respectively to the electric motor, the differential, the drive axle and the wheel. At the final drive shaft, the torque balance equation is:

$$F_1 R_2 - T_d - T_{loss_{fd}} = J_d \frac{d\omega_d}{dt}$$ \hfill (B.3)

$R_2$ is the radius of the output gear of the final drive and $T_{loss_{fd}}$ are the losses which occur in the final drive gearing. The torque is then splitted between the two axles and it is delivered to the wheels, according to

$$0.5T_d - T_w = (J_a + J_w) \frac{d\omega_w}{dt}$$ \hfill (B.4)

The angular speed and acceleration through the powertrain is expressed by the relation

$$\frac{d\omega_{em}}{dt} = b_{fd} \frac{d\omega_d}{dt} = b_{fd} \frac{d\omega_w}{dt} = \frac{b_{fd}}{r_w} \frac{dv}{dt}$$ \hfill (B.5)

where $b_{fd} = \frac{R_2}{R_1}$ is the final drive ratio and $r_w$ is the wheel radius. The required torque at each wheel is determined by the tractive force calculated in equation ??

$$T_w = 0.5F_x r_w$$ \hfill (B.6)
Finally, by combining the previous equations it is possible to derive the torque demand at the motor shaft function of the vehicle speed

\[
T_{em} = \frac{1}{b_{fd}} \left[ F_R r_w + \left( b_{fd} T_{loss,em} + T_{loss,fd} \right) + \frac{1}{r_w} \frac{dv}{dt} \left( b_{fd}^2 \left( J_{em} + J_{1s} \right) + J_d + 2 \left( J_a + J_w \right) + m r_w^2 \right) \right]
\]  
(B.7)

Equation B.7 can be reduced to

\[
T_{em} = \left( b_{fd} T_{loss,em} + T_{loss,fd} \right) + \frac{r_w}{b_{fd}} \left( m_{eff} \frac{dv}{dt} + F_R \right)
\]  
(B.8)

\(m_{eff}\) is the effective mass of the vehicle, including the effects of the rotating inertias

\[
m_{eff} = m + \frac{1}{r_w^2} \left( b_{fd}^2 \left( J_{em} + J_{1s} \right) + J_d + 2 \left( J_a + J_w \right) \right)
\]  
(B.9)

### B.2 Two-speed drivetrain

![Two-speed transmission schematic](image)

Figure B.2: Two-speed transmission schematic

Figure B.2 illustrates the layout of the dual stage two-speed transmission. The electric motor is directly connected to the primary shaft of the gearbox, which employs two gear pairs with different ratios and a dog clutch to connect the output gear to the secondary shaft and thus transmitting torque. When disconnected, the output gears at the secondary shaft can freely spin. Torque is then passed to the axle shafts through the final drive gearing and the differential.
APPENDIX B. DRIVETRAIN PROPULSION MODELING

The transmission of torque through the drivetrain is studied for the first gear; the same steps can be used when the second gear is engaged.
The torque balance equation at the primary shaft is

\[ T_{em} - T_{loss,em} - F_1 R_1 = (J_{em} + J_{1s}) \frac{d\omega_{1s}}{dt} \]  \hspace{1cm} (B.10)

\( F_1 \) is the force transmitted through the first gear set, \( R_1 \) is the radius of the first gear set input gear, \( J_{1s} \) and \( \omega_{1s} \) are the moment of inertia and the angular speed of the primary shaft.
The torque balance equation at the secondary shaft is equal to

\[ F_1 R_2 - T_{loss,gb} - F_3 R_5 = J_{2s} \frac{d\omega_{2s}}{dt} \]  \hspace{1cm} (B.11)

\( F_3 \) is the force transmitted by the final drive, \( R_2 \) is the radius of the first gear set output gear, \( R_5 \) is the radius of the input gear of the final drive, \( J_{2s} \) and \( \omega_{2s} \) are the moment of inertia and the angular speed of the secondary shaft. When running with the second gear, the force transmitted to the secondary shaft is \( F_2 \) and the radii of the second gear set \( R_2 \) and \( R_4 \) replace \( R_1 \) and \( R_3 \) in Equation (B.10) and (B.11).
At the differential shaft, the torque balance equation can be written with

\[ F_3 R_6 - T_d - T_{loss,fd} = J_d \frac{d\omega_d}{dt} \]  \hspace{1cm} (B.12)

\( R_6 \) is the output gear of the final drive.
Since the rest of the powertrain matches the single-speed model, Equations (B.4) and (B.6) are valid to describe the transmission of torque from the differential to the wheels.
The relationship between the angular acceleration of the shafts is given by

\[ \frac{d\omega_{1s}}{dt} = b_{g1} \frac{d\omega_{2s}}{dt} = b_{g1} b_{fd} \frac{d\omega_w}{dt} \]  \hspace{1cm} (B.13)

\( b_{g1} = \frac{R_3}{R_1} \) and \( b_{fd} = \frac{R_4}{R_2} \) are the ratios of the first gear and of the final drive.
Again, it is possible to express the motor torque in terms of the vehicle speed

\[ T_{em} = \frac{1}{b_{g1} b_{fd}} \left[ F_R r_w + (b_{g1} b_{fd} T_{loss,em} + b_{fd} T_{loss,gb} + T_{loss,fd}) + \frac{1}{r_w} \frac{dv}{dt} \left( (b_{g1} b_{fd})^2 (J_{em} + J_{1s}) + b_{fd}^2 J_{2s} + J_d + 2 (J_a + J_w) + mr^2_w \right) \right] \]  \hspace{1cm} (B.14)

The effective mass \( m_{eff} \) equals

\[ m_{eff} = m + \frac{1}{r_w^2} \left( (b_{g1} b_{fd})^2 (J_{em} + J_{1s}) + b_{fd}^2 J_{2s} + J_d + 2 (J_a + J_w) \right) \]  \hspace{1cm} (B.15)
B.3 In-wheel drivetrain

In the in-wheel architecture the absence of mechanical components for power transmission greatly reduces the complexity of the formulations. The required electromagnetic torque for each motor can be calculated with

\[ T_{em} = (T_{em_{loss}}) + \frac{r_w}{2} \left[ \left( m + 2 \left( \frac{J_w + J_{in-wheel}}{r_w^2} \right) \right) \frac{dv}{dt} + F_R \right] \]  

(B.16)

\( J_{in-wheel} \) is the inertia of the in-wheel motor. For the outer rotor arrangement it can be estimates as

\[ J_{in-wheel} = \frac{1}{2} m_{rotor} (r_i^2 + r_o^2) \]  

(B.17)

\( r_{in} \) and \( r_{out} \) are the inner and the outer radii of the rotor.
## Appendix C

### Parameters

#### C.1 Vehicle

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle mass (Kg)</td>
<td>$m$</td>
<td>1200</td>
</tr>
<tr>
<td>Aerodynamic coefficient</td>
<td>$C_d$</td>
<td>0.3</td>
</tr>
<tr>
<td>Frontal area (m$^2$)</td>
<td>$A_f$</td>
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</tr>
<tr>
<td>Tyre rolling coefficient</td>
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</tr>
<tr>
<td>Wheel radius (m)</td>
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<td>Wheel moment inertia (kgm$^2$)</td>
<td>$J_w$</td>
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<td>Axle half-shaft moment of inertia (kgm$^2$)</td>
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</tr>
<tr>
<td>Differential shaft moment of inertia (kgm$^2$)</td>
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<td>Gearbox primary shaft moment of inertia (kgm$^2$)</td>
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<td>Gearbox secondary shaft moment of inertia (kgm$^2$)</td>
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<td>Front-rear braking split</td>
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<td>70:30</td>
</tr>
<tr>
<td>Regenerative braking efficiency (%)</td>
<td></td>
<td>80</td>
</tr>
<tr>
<td>Air density at T=15$^\circ$C (kg/m$^3$)</td>
<td>$\rho_{a}$</td>
<td>1.204</td>
</tr>
<tr>
<td>Air density at T=100$^\circ$C (kg/m$^3$)</td>
<td>$\rho_{100}$</td>
<td>0.9413</td>
</tr>
<tr>
<td>Air dynamic viscosity at T=100$^\circ$C (kg/m s)</td>
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<tr>
<td>W211 oil density at T=40$^\circ$C (kg/m$^3$)</td>
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<td>879.8</td>
</tr>
<tr>
<td>W211 oil dynamic viscosity at T=40$^\circ$C (kg/m s)</td>
<td>$\mu_{40}$</td>
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<td>W211 oil density at T=100$^\circ$C (kg/m$^3$)</td>
<td>$\rho_{100}$</td>
<td>841.5</td>
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<tr>
<td>W211 oil dynamic viscosity at T=100$^\circ$C (kg/m s)</td>
<td>$\mu_{100}$</td>
<td>4.463 x 10$^{-3}$</td>
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</table>
**APPENDIX C. PARAMETERS**

Table C.2: Battery

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal open circuit voltage (V)</td>
<td>$V_{oc}$</td>
<td>299</td>
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<tr>
<td>Nominal energy capacity (kWh)</td>
<td>$E_{batt}$</td>
<td>16.84</td>
</tr>
<tr>
<td>Pulse discharge power (kW)</td>
<td>$P_{dis\ peak}$</td>
<td>161.2</td>
</tr>
<tr>
<td>Pulse charge power (kW)</td>
<td>$P_{ch\ peak}$</td>
<td>57.1</td>
</tr>
<tr>
<td>Charge resistance (mΩ)</td>
<td>$R_{ch}$</td>
<td>92.78</td>
</tr>
<tr>
<td>Discharge resistance (mΩ)</td>
<td>$R_{dis}$</td>
<td>110.88</td>
</tr>
<tr>
<td>Number of cells in series</td>
<td>-</td>
<td>77</td>
</tr>
<tr>
<td>Number of cells in parallel</td>
<td>-</td>
<td>2</td>
</tr>
<tr>
<td>Estimated pack weight (kg)</td>
<td>$m_{batt}$</td>
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</tbody>
</table>

C.2 Electric drives

Table C.3: Induction machine

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated power (kW)</td>
<td>$P_{rat}$</td>
<td>40</td>
</tr>
<tr>
<td>Rated torque (Nm)</td>
<td>$T_{rat}$</td>
<td>85</td>
</tr>
<tr>
<td>Synchronous speed (rpm)</td>
<td>$\omega_{syn}$</td>
<td>4500</td>
</tr>
<tr>
<td>Rated slip (%)</td>
<td>$s_{rat}$</td>
<td>2</td>
</tr>
<tr>
<td>Rated phase voltage (V)</td>
<td>$U_{rat RMS}$</td>
<td>170</td>
</tr>
<tr>
<td>Peak power (kW)</td>
<td>$P_{max}$</td>
<td>70</td>
</tr>
<tr>
<td>Peak torque (Nm)</td>
<td>$T_{max}$</td>
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</tr>
<tr>
<td>Maximum phase current (A)</td>
<td>$I_{max RMS}$</td>
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<td>Maximum speed (rpm)</td>
<td>$\omega_{em max}$</td>
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</tr>
<tr>
<td>Stator inductance (mH)</td>
<td>$L_s$</td>
<td>0.472</td>
</tr>
<tr>
<td>Rotor inductance (mH)</td>
<td>$L'_r$</td>
<td>0.472</td>
</tr>
<tr>
<td>Mutual inductance (mH)</td>
<td>$M$</td>
<td>9.111</td>
</tr>
<tr>
<td>Stator resistance (Ω)</td>
<td>$R_s$</td>
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</tr>
<tr>
<td>Rotor resistance (Ω)</td>
<td>$R'_r$</td>
<td>0.033</td>
</tr>
<tr>
<td>Pole pairs</td>
<td>$p$</td>
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<tr>
<td>Stator outer diameter (mm)</td>
<td>$D_{so}$</td>
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</tr>
<tr>
<td>Stator inner diameter (mm)</td>
<td>$D_{si}$</td>
<td>150</td>
</tr>
<tr>
<td>Rotor diameter (mm)</td>
<td>$D_r$</td>
<td>146</td>
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<tr>
<td>Axial length (mm)</td>
<td>$L_{ax}$</td>
<td>220</td>
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<tr>
<td>Air gap length (mm)</td>
<td>$\delta_{ag}$</td>
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</tr>
<tr>
<td>Weight (kg)</td>
<td>$m_{cm}$</td>
<td>50</td>
</tr>
<tr>
<td>Motor shaft moment of inertia (kgm²)</td>
<td>$J_{cm}$</td>
<td>0.15</td>
</tr>
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</table>
APPENDIX C. PARAMETERS

Table C.4: PM synchronous machine

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<tr>
<th>Definition</th>
<th>Symbol</th>
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</tr>
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<tbody>
<tr>
<td>Rated power (kW)</td>
<td>$P_{\text{rat}}$</td>
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</tr>
<tr>
<td>Rated torque (Nm)</td>
<td>$T_{\text{rat}}$</td>
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</tr>
<tr>
<td>Rated speed (rpm)</td>
<td>$\omega_{\text{syn}}$</td>
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<tr>
<td>Rated phase voltage (V)</td>
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<td>Peak power (kW)</td>
<td>$P_{\text{max}}$</td>
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<td>Peak torque (Nm)</td>
<td>$T_{\text{max}}$</td>
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</tr>
<tr>
<td>Maximum phase current (A)</td>
<td>$I_{\text{max}}$</td>
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<tr>
<td>Maximum speed (rpm)</td>
<td>$\omega_{\text{max}}$</td>
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<td>Leakage inductance (mH)</td>
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<tr>
<td>Magnetizing inductance (mH)</td>
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<td>Stator resistance (Ω)</td>
<td>$R_s$</td>
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</tr>
<tr>
<td>Pole pairs</td>
<td>$p$</td>
<td>12</td>
</tr>
<tr>
<td>Rotor outer diamater (mm)</td>
<td>$D_{\text{so}}$</td>
<td>360</td>
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<tr>
<td>Rotor inner diamater (mm)</td>
<td>$D_{\text{si}}$</td>
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</tr>
<tr>
<td>Stator inner diameter (mm)</td>
<td>$D_r$</td>
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<td>Axial length (mm)</td>
<td>$L_{\text{ax}}$</td>
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<tr>
<td>Air gap length (mm)</td>
<td>$\delta_{\text{ag}}$</td>
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<td>Rotor &amp; magnet mass (kg)</td>
<td>$m_{\text{rotor}}$</td>
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<td>Total weight (kg)</td>
<td>$m_{\text{em}}$</td>
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Table C.5: Inverter

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<tr>
<td>Voltage rating (V)</td>
<td>$V_{ce}$</td>
<td>600</td>
<td>600</td>
</tr>
<tr>
<td>IGBT collector-emitter threshold voltage (V)</td>
<td>$u_{ce0}$</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>Diode threshold voltage (V)</td>
<td>$u_{d0}$</td>
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<td>0.8</td>
</tr>
<tr>
<td>IGBT collector-emitter on-state resistance (mΩ)</td>
<td>$r_{ce}$</td>
<td>4.167</td>
<td>2.468</td>
</tr>
<tr>
<td>Diode forward resistance (mΩ)</td>
<td>$r_d$</td>
<td>3.75</td>
<td>2</td>
</tr>
<tr>
<td>Nominal current (A)</td>
<td>$I_{cn}$</td>
<td>200</td>
<td>400</td>
</tr>
<tr>
<td>IGBT nominal rise time (ns)</td>
<td>$t_{\text{rn}}$</td>
<td>40</td>
<td>80</td>
</tr>
<tr>
<td>IGBT nominal fall time (ns)</td>
<td>$t_{\text{fn}}$</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>Reverse recovery charge (µC)</td>
<td>$Q_{\text{rrn}}$</td>
<td>17</td>
<td>30</td>
</tr>
<tr>
<td>Reverse recovery current (A)</td>
<td>$I_{\text{rrn}}$</td>
<td>230</td>
<td>280</td>
</tr>
<tr>
<td>DC-link voltage (V)</td>
<td>$V_{DC}$</td>
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<td>400</td>
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<tr>
<td>Switching frequency (kHz)</td>
<td>$f_{\text{sw}}$</td>
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<td>10</td>
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C.3 Transmission

Table C.6: Differential

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<th>Symbol</th>
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<th>Two-speed</th>
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<tr>
<td>Final drive ratio</td>
<td>( b_{fd} )</td>
<td>7.705</td>
<td>3.7</td>
</tr>
<tr>
<td>Gear pitch diameter (mm)</td>
<td>( d_p )</td>
<td>212.5</td>
<td>212.5</td>
</tr>
<tr>
<td>Pinion pitch diameter (mm)</td>
<td>( d_p )</td>
<td>27.717</td>
<td>58.514</td>
</tr>
<tr>
<td>Gear number of teeth</td>
<td>( z_g )</td>
<td>69</td>
<td>69</td>
</tr>
<tr>
<td>Pinion number of teeth</td>
<td>( z_p )</td>
<td>9</td>
<td>19</td>
</tr>
<tr>
<td>Gear face width (mm)</td>
<td>( b_w )</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Helix angle (°)</td>
<td>( \beta )</td>
<td>32</td>
<td>32</td>
</tr>
<tr>
<td>Normal pressure angle (°)</td>
<td>( \alpha_n )</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Transverse module (mm)</td>
<td>( m_t )</td>
<td>3.1</td>
<td>3.1</td>
</tr>
</tbody>
</table>

Table C.7: Two-speed gearbox

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>First gear</th>
<th>Second gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear ratio</td>
<td>( b_g )</td>
<td>3.936</td>
<td>1.749</td>
</tr>
<tr>
<td>Gear pitch diameter (mm)</td>
<td>( d_p )</td>
<td>108.25</td>
<td>95.5</td>
</tr>
<tr>
<td>Pinion pitch diameter (mm)</td>
<td>( d_p )</td>
<td>26.461</td>
<td>55.178</td>
</tr>
<tr>
<td>Gear number of teeth</td>
<td>( z_g )</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>Pinion number of teeth</td>
<td>( z_p )</td>
<td>11</td>
<td>26</td>
</tr>
<tr>
<td>Gear face width (mm)</td>
<td>( b_w )</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Helix angle (°)</td>
<td>( \beta )</td>
<td>30</td>
<td>32</td>
</tr>
<tr>
<td>Normal pressure angle (°)</td>
<td>( \alpha_n )</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Transverse module (mm)</td>
<td>( m_t )</td>
<td>2.406</td>
<td>2.122</td>
</tr>
</tbody>
</table>

Table C.8: Cylindrical roller bearings

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Internal diameter (mm)</td>
<td>( d )</td>
<td>30</td>
</tr>
<tr>
<td>Outer diameter (mm)</td>
<td>( D )</td>
<td>50</td>
</tr>
<tr>
<td>Bearing width (mm)</td>
<td>( B )</td>
<td>13</td>
</tr>
<tr>
<td>Replenishment/starvation constant</td>
<td>( K_{rs} )</td>
<td>( 3 \times 10^{-8} )</td>
</tr>
<tr>
<td>Bearing type related geometric constant</td>
<td>( K_e )</td>
<td>5.1</td>
</tr>
<tr>
<td>Bearing type related geometric constant</td>
<td>( K_L )</td>
<td>0.65</td>
</tr>
<tr>
<td>Sliding friction coefficient in full lubrication conditions</td>
<td>( \mu_{EHL} )</td>
<td>0.02</td>
</tr>
<tr>
<td>Sliding friction coefficient in boundary conditions</td>
<td>( \mu_{bl} )</td>
<td>0.044</td>
</tr>
<tr>
<td>Geometric constant for rolling friction</td>
<td>( R_1 )</td>
<td>( 1.09 \times 10^{-6} )</td>
</tr>
<tr>
<td>Geometric constants for sliding friction</td>
<td>( S_1 )</td>
<td>0.16</td>
</tr>
<tr>
<td>Geometric constants for sliding friction</td>
<td>( S_2 )</td>
<td>0.0015</td>
</tr>
<tr>
<td>Drag loss factor</td>
<td>( V_m )</td>
<td>( 7 \times 10^{-4} )</td>
</tr>
</tbody>
</table>
Bibliography


[21] K. Reis and A. Binder, “Comparison of direct drive and high speed drive concepts for the use in wheel-hub-drives,”


[58] Oerlikon Graziano, “Transmission systems for full electric and hybrid vehicles.”.


