Influence of in-wheel motors on the ride comfort of electric vehicles

R. Vos
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Master’s thesis

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Abstract

This report describes the influence of in-wheel motors on the ride comfort and road holding of electric vehicles. To this end, on-road experiments are performed using an ICE vehicle and simulations are performed with a model representing a battery electric vehicle. The experiments and simulations are performed by adding a mass of 15 kg to each individual wheel. This is determined based on the assumption that the vehicle has to be equipped with drive motors that have a combined power of 30 kW in order to overcome the road load during normal driving and based on the assumption that a specific motor output of approximately 1 kW/kg can be considered to be an appropriate guideline for permanent magnet brushless dc motors.

The on-road experiments are performed using an ICE VW Lupo 3L that is currently being converted to a battery electric vehicle at the Eindhoven University of Technology. The experiments are conducted with the baseline configuration, a configuration with an increased unsprung mass at the front, and a configuration with an increased unsprung mass at the rear. All measurements are performed on four different road surface types with ascending severity: smooth asphalt, highway, cobblestones and Belgian blocks. The results of the experiments show that in-wheel motors, especially placed in the front wheels, result in high deterioration of the ride comfort. This negative effect increases as the severity of the road increases.

In order to validate the full-car model, simulation results are compared to experimental results. The input parameters of the model are all derived from the baseline Lupo 3L. To this end, the vehicle mass and weight distribution, the suspension parameters, stiffness of the anti-roll bars and the tyre parameters are determined experimentally. The comparison of the results show that there is a fair agreement between the simulations and experiments up to at least the validation frequency of 20 Hz. Overall, the simulations are able to give a good approximation of the influence of in-wheel motors on the vehicle ride comfort.

Using the validated model it has been found that the battery electric vehicle is approximately 14% more comfortable than the baseline ICE vehicle. In-wheel motors decrease this ride comfort again, although this increase does not lead to a less comfortable ride than the baseline ICE vehicle. Unfortunately, the dynamic wheel load is increased by approximately 40% and the suspension travel is increased with about 16%. Since the suspension system as implemented in the baseline vehicle has been found to be optimized, even for the battery electric vehicle, this increase can not be sufficiently reduced by changing the suspension parameters. Reducing the tyre pressure or using a dynamic vibration absorber are found to work insufficiently as well.

Furthermore, possible improvements in ride comfort, road holding and suspension travel using a (semi-)active electromagnetic suspension system have been investigated. Two control techniques are examined: the skyhook and hybrid control, both semi-active and full active. Under the assumption that an improvement in the dynamic wheel load is not accompanied with a deterioration in the ride comfort, the semi-active skyhook controller is able to decrease the dynamic wheel load the most, up to 9%. Without the above assumption, it is best to use the active hybrid controller, since this controller is able to decrease the dynamic wheel load up to 18%, although at the expense of the ride comfort. Moreover, it has been found that only low power levels are needed to control the system.
Samenvatting

In dit verslag worden de invloeden van in-wiel motoren op de comfort en veiligheid van een elektrisch voertuig beschreven. Om dit in kaart te brengen zijn er experimenten uitgevoerd op de weg met een diesel auto en zijn er simulaties uitgevoerd met een model dat een representatie is van een batterij elektrisch voertuig. Zowel de experimenten en de simulaties zijn uitgevoerd uitgaande van het toevoegen van 30 kg aan de onafgeveerde massa. Deze waarde is gebaseerd op de aannemer dat de auto moet worden voorzien van motors met een gecombineerd vermogen van 30 kg en op de aannemer dat de in-wiel motor een specifieke output heeft van 1 kW/kg. Dit laatste is namelijk een goede leidraad voor permanent magnet brushless dc motors.

De experimenten zijn uitgevoerd met behulp van een VW Lupo 3L. Deze auto wordt op dit moment omgebouwd tot een batterij elektrisch voertuig op de Technische Universiteit Eindhoven. De experimenten zijn uitgevoerd met 3 verschillende configuraties: de onveranderde configuratie, een configuratie met verzwaarde voorwielen en een configuratie met verzwaarde achterwielen. Alle metingen zijn uitgevoerd op 4 verschillende weg types: glad wegdek, snelweg, klinkers en Belgische kinderkopjes. Uit de resultaten blijkt dat in-wiel motoren resulteren in een flinke verlaging van de rijcomfort, vooral als ze geplaatst worden in de voorwielen. Dit negatieve effect is het grootst op de Belgische kinderkopjes.

Om het simulatie model te valideren zijn simulatie resultaten vergeleken met experimentele resultaten. Alle invoer parameters van het model zijn afgeleid van de VW Lupo 3L. Daarvoor is de massa van de auto, de gewichtsverdeling, de parameters van de ophangsystemen, de stijfheid van de anti-rol barren en de parameters van de banden experimenteel bepaald. Uit de vergelijking blijkt dat het simulatie model sterk lijkt op het echte voertuig tot een frequentie van 25 Hz. Hieruit kan worden geconcludeerd dat aan de hand van simulaties een goede schatting kan worden gegeven met betrekking tot de invloeden van in-wiel motoren.

Uit de simulaties blijkt dat een batterij elektrisch voertuig ongeveer 14% comfortabeler rijdt dan de oorspronkelijke VW Lupo 3L. In-wiel motoren resulteren weer in een verslechtering van deze rijcomfort, maar niet dusdanige dat het elektrisch voertuig minder comfortable rijdt dan de oorspronkelijke VW Lupo 3L. Helaas nemen de dynamische wielkrachten wel met ongeveer 40% toe en de veerweg met ongeveer 16%. Gezien geconcludeerd kan worden dat de ophangsystemen van de Lupo optimaal zijn, kunnen deze toenames niet worden verklaard door het anpassen van de parameters van de ophangsystemen. Het aanpassen van de bandenspanning of door gebruik van een zogenaamde 'dynamic vibration absorber' kunnen deze krachten ook niet efficiënt genoeg worden verlaagd.

Verder is onderzocht wat een (semi-) actief elektromagnetisch ophangsysteem kan betekenen om de wielkrachten te verlagen. Twee control technieken zijn gebruikt voor de aansturing van het systeem: de skyhook en hybride control, beide semi-actief en volledig actief. Onder de aannemer dat een verbetering in de wielkrachten niet samen gaat met een verslechtering in de rijcomfort, kan het beste de semi-actieve skyhook control worden gebruikt om de wielkrachten te verlagen met 9%. Zonder deze aannemer kan het beste gebruik worden gemaakt van de actieve hybrid controller. Deze is namelijk in staat om de wielkrachten te verlagen met 18%, al gaat dit wel gepaard met een hoge verslechtering van de rijcomfort. Het benodigde vermogen voor de aansturing van het systeem is laag.
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<td>( l )</td>
<td>m</td>
<td>Vehicle wheelbase</td>
</tr>
<tr>
<td>( w )</td>
<td>m</td>
<td>Vehicle track width</td>
</tr>
<tr>
<td>( L )</td>
<td>m</td>
<td>Vehicle length</td>
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<td>( W )</td>
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<td>( m_v )</td>
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<td>( a )</td>
<td>m</td>
<td>Distance of center of gravity to front wheels</td>
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<tr>
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<td>m</td>
<td>Distance of center of gravity to rear wheels</td>
</tr>
<tr>
<td>( A )</td>
<td>m²</td>
<td>Vehicle frontal area</td>
</tr>
<tr>
<td>( c_d )</td>
<td>-</td>
<td>Vehicle air drag coefficient</td>
</tr>
<tr>
<td>( c_r )</td>
<td>-</td>
<td>Tyre rolling resistance coefficient</td>
</tr>
<tr>
<td>( F_z )</td>
<td>N</td>
<td>Vertical force</td>
</tr>
<tr>
<td>( q_{F,v}^2 )</td>
<td>-</td>
<td>Tyre stiffness increase with velocity</td>
</tr>
<tr>
<td>( R_0 )</td>
<td>m</td>
<td>Non-rolling free tyre radius</td>
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<tr>
<td>( q_{F,cx} )</td>
<td>-</td>
<td>Vertical sinking of the tyre due to longitudinal forces</td>
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<tr>
<td>( q_{F,cy} )</td>
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<td>( q_{F,c1} )</td>
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<td>Tyre force deflection characteristic 1</td>
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<td>( q_{F,c2} )</td>
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<td>Tyre force deflection characteristic 2</td>
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<td>Influence of the tyre inflation pressure</td>
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<td>( F_{z0} )</td>
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<td>( dp_i )</td>
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<td>Pressure increment</td>
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<td>( D_{reff} )</td>
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</tr>
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</tr>
<tr>
<td>( F_{reff} )</td>
<td>-</td>
<td>Tyre model parameter</td>
</tr>
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<td>N/m</td>
<td>Tyre vertical stiffness</td>
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<tr>
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<td>m/s²</td>
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<tr>
<td>( J_w )</td>
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<td>( F_{air} )</td>
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<td>( F_r )</td>
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<td>( F_g )</td>
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<tr>
<td>( F_i )</td>
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<td>Internal force</td>
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<tr>
<td>( T_m )</td>
<td>Nm</td>
<td>Motor torque</td>
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<tr>
<td>( P_m )</td>
<td>W</td>
<td>Motor power</td>
</tr>
<tr>
<td>( V_0 )</td>
<td>m/s</td>
<td>Maximum speed</td>
</tr>
<tr>
<td>( t_0 )</td>
<td>s</td>
<td>Acceleration time</td>
</tr>
<tr>
<td>Symbol</td>
<td>Unit</td>
<td>Description</td>
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<td>---------</td>
<td>------------</td>
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<td>T</td>
<td>Average airgap flux-density</td>
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<tr>
<td>$Q$</td>
<td>A/m</td>
<td>Specific electrical loading</td>
</tr>
<tr>
<td>$D_{out}$</td>
<td>m</td>
<td>Rotor outer diameter</td>
</tr>
<tr>
<td>$D_{in}$</td>
<td>m</td>
<td>Rotor inner diameter</td>
</tr>
<tr>
<td>$I_{xx}$</td>
<td>kgm$^2$</td>
<td>Moment of inertia wrt the x-axis</td>
</tr>
<tr>
<td>$I_{yy}$</td>
<td>kgm$^2$</td>
<td>Moment of inertia wrt the y-axis</td>
</tr>
<tr>
<td>$I_{zz}$</td>
<td>kgm$^2$</td>
<td>Moment of inertia wrt the z-axis</td>
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<td>$a_{overall}$</td>
<td>m/s$^2$</td>
<td>Overall ride comfort index</td>
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<td>$acc_x$</td>
<td>m/s$^2$</td>
<td>Weighted RMS value of the longitudinal acceleration</td>
</tr>
<tr>
<td>$acc_y$</td>
<td>m/s$^2$</td>
<td>Weighted RMS value of the lateral acceleration</td>
</tr>
<tr>
<td>$acc_z$</td>
<td>m/s$^2$</td>
<td>Weighted RMS value of the vertical acceleration</td>
</tr>
<tr>
<td>$f_{null}$</td>
<td>Hz</td>
<td>Null points</td>
</tr>
<tr>
<td>$V$</td>
<td>m/s</td>
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</tr>
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<td>Height of center of gravity above ground</td>
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<tr>
<td>$dF_z$</td>
<td>N</td>
<td>Dynamic wheel load</td>
</tr>
<tr>
<td>$dz$</td>
<td>mm</td>
<td>Suspension travel</td>
</tr>
<tr>
<td>$\tau$</td>
<td>s</td>
<td>Switching time constant</td>
</tr>
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<td>N</td>
<td>Skyhook force</td>
</tr>
<tr>
<td>$d_{sky}$</td>
<td>N/m</td>
<td>Skyhook damping constant</td>
</tr>
<tr>
<td>$\dot{z}_s$</td>
<td>m/s</td>
<td>Vertical velocity of the sprung mass</td>
</tr>
<tr>
<td>$\dot{z}_a$</td>
<td>m/s</td>
<td>Vertical velocity of the unsprung mass</td>
</tr>
<tr>
<td>$F_{gnd}$</td>
<td>N</td>
<td>Groundhook force</td>
</tr>
<tr>
<td>$d_{gnd}$</td>
<td>N/m</td>
<td>Groundhook damping constant</td>
</tr>
<tr>
<td>$F_h$</td>
<td>N</td>
<td>Hybrid force</td>
</tr>
<tr>
<td>$\alpha$</td>
<td></td>
<td>Relative ratio between skyhook and groundhook force</td>
</tr>
<tr>
<td>$P_{cu}$</td>
<td>W</td>
<td>Copper losses</td>
</tr>
<tr>
<td>$t_e$</td>
<td>s</td>
<td>Time</td>
</tr>
<tr>
<td>$R_{ph}$</td>
<td>$\Omega$</td>
<td>Phase resistance</td>
</tr>
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<td>$\tilde{i}$</td>
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<td>Three phase commutated current</td>
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<tr>
<td>$F_{act}$</td>
<td>N</td>
<td>Actuator force</td>
</tr>
<tr>
<td>$K_i$</td>
<td>N/A</td>
<td>Actuator motor constant</td>
</tr>
<tr>
<td>$M$</td>
<td>W</td>
<td>Motor power</td>
</tr>
<tr>
<td>$R$</td>
<td>W</td>
<td>Regenerated power</td>
</tr>
<tr>
<td>$P_{cu,M}$</td>
<td>W</td>
<td>Copper losses in motor mode</td>
</tr>
<tr>
<td>$P_{cu,R}$</td>
<td>W</td>
<td>Copper losses in regeneration mode</td>
</tr>
<tr>
<td>$T$</td>
<td>W</td>
<td>Total power</td>
</tr>
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## Acronyms

<table>
<thead>
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<th>Description</th>
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<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>VW</td>
<td>Volkswagen</td>
</tr>
<tr>
<td>ABS</td>
<td>Anti-lock Braking System</td>
</tr>
<tr>
<td>TCS</td>
<td>Traction Control System</td>
</tr>
<tr>
<td>BEV</td>
<td>Battery Electric Vehicle</td>
</tr>
<tr>
<td>PSD</td>
<td>Power Spectral Density</td>
</tr>
<tr>
<td>CG</td>
<td>Center of gravity</td>
</tr>
<tr>
<td>DC</td>
<td>Direct Current</td>
</tr>
<tr>
<td>AC</td>
<td>Alternating Current</td>
</tr>
<tr>
<td>PMS</td>
<td>Permanent Magnet Synchronous</td>
</tr>
<tr>
<td>PMBDC</td>
<td>Permanent Magnet Brushless Direct Current</td>
</tr>
<tr>
<td>SR</td>
<td>Switch Reluctance</td>
</tr>
<tr>
<td>M</td>
<td>Motor</td>
</tr>
<tr>
<td>GB</td>
<td>Gearbox</td>
</tr>
<tr>
<td>FG</td>
<td>Fixed Gearing</td>
</tr>
<tr>
<td>RMS</td>
<td>Root Mean Square</td>
</tr>
<tr>
<td>iw-front</td>
<td>Vehicle with front in-wheel motors</td>
</tr>
<tr>
<td>iw-rear</td>
<td>Vehicle with rear in-wheel motors</td>
</tr>
<tr>
<td>iw-four</td>
<td>Vehicle with four in-wheel motors</td>
</tr>
<tr>
<td>GPS</td>
<td>Global Positioning System</td>
</tr>
<tr>
<td>SUV</td>
<td>Sports utility vehicle</td>
</tr>
<tr>
<td>RFPFM</td>
<td>Radial Flux Permanent Magnet</td>
</tr>
<tr>
<td>AFPM</td>
<td>Axial Flux Permanent Magnet</td>
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<tr>
<td>dva</td>
<td>dynamic vibration absorber</td>
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Chapter 1

Introduction

In 2005, the Netherlands Society for Nature and Environment ("Stichting Natuur en Milieu") initiated the so-called c,m,n project. This project challenged the three technical universities of the Netherlands (Eindhoven, Delft and Twente) to design a sustainable car for the year 2020 [1, 2]. Due to the concerns about global warming by CO\textsubscript{2} emissions of internal combustion engines (ICEs), this vehicle has to be much more efficient and cleaner than vehicles currently on the market. Since the beginning of the project, two different cars, the c,m,n 1.0 and 2.0, have been presented at the Amsterdam Motor Show AutoRAI in 2007 and 2009, respectively.

In 2007, the exterior of the car, based on a specific vision of automobility in the year 2020, was shown together with a fuel cell/super capacitor drive train. This drive train consists of a fuel cell to convert hydrogen into electrical energy and an electric motor to convert the electrical energy into mechanical energy. The super capacitors are used to store the brake energy recuperated by the electric motor. For more information about the drive train the reader is referred to paper [56]. In 2009, the car has been equipped with a battery electric drive train and thus transformed to a battery electric vehicle (BEV), consisting of an electric motor for the propulsion of the vehicle and batteries as energy source. The recuperated brake energy can be stored directly in the batteries. The choice of using this type of drive train was made, since it features less energy conversion steps than the fuel cell drive train and consists of components that all work at high efficiency, resulting in a high overall well-to-wheel efficiency. More information about the drive train of the c,m,n 2.0 can be found in [59].

Besides the advantages of being able to recuperate braking energy, propelling the vehicle by an electric motor offers other advantages over an ICE vehicle: the torque generation is very quick and accurate and can be controlled much more precisely with a shorter control period. The torque response is in the order of several milliseconds, 10-100 times faster than an internal combustion engine or conventional braking system [38]. All this makes high acceleration of the vehicle possible and results in control systems, like the anti-lock braking system (ABS) and the traction control system (TCS), that are more effective than found in ICE vehicles. Furthermore, electric motors come in many different sizes and weights that are smaller and lower than conventional ICEs.

As a result of the substantial choices in electric motors, multiple propulsion configurations can be adopted, each with its own advantages and disadvantages. The configuration in which the motors are implemented in the wheels, the in-wheel motor, seems to be one of the most simple, interesting and innovative option. For example, this configuration offers the opportunity to control each wheel independently, resulting in potentially the maximum stable controlled vehicle. Moreover, several parts like the drive shafts and differentials become redundant, which results in an overall decrease of weight and in a more efficient vehicle. However, placing the motors in the wheels also result in an increase in unsprung mass, which results in a decrease in ride comfort and road holding of the vehicle.
Chapter 1. Introduction

1.1 Research objectives

Considering the many advantages of the in-wheel motors, extensive research has been performed on potential designs and control systems. However, although the disadvantages with respect to ride comfort and road holding are often shortly noted in many papers, research within this specific field has not been reported in the open literature. Since passenger comfort and safety is an ever increasing demand in the automotive industry, knowledge of the severity of the disadvantages will play an important part in the successful implementation of in-wheel motors in future vehicles.

Some of the negative effects can possibly be reduced or even eliminated by optimization of the passive suspension system or by using a semi-active or active suspension system. Note however that, as the available energy in current BEVs is limited, a semi-active or active suspension system can only be used if it does not consume a lot of energy.

From these considerations, the objectives of this thesis are therefore:

- Investigate the influence of in-wheel motors on the ride comfort and road holding capability of vehicles
- Investigate the possible improvements in ride comfort and safety of electric vehicles equipped with in-wheel motors by:
  - optimization of the passive suspension system
  - using a low-energy semi-active or active suspension system

The first investigation is performed by on-road experiments using an ICE vehicle that is currently being converted to a BEV at the Eindhoven University of Technology. Using the experimental results, a full-car model, made within the software program Matlab/Simulink, is validated and used to investigate the influence of in-wheel motors on a BEV. The parameters of the model are therefore adjusted to reflect the converted BEV. Using this adapted simulation model, the possible improvements by optimization of the suspension system or by the use of a semi-active or active suspension system are investigated.

In the next section, the outline of this thesis is presented.

1.2 Thesis outline

Chapter 2 discusses the design choices of the main parts of the drive train of a BEV: the batteries and electric motors. Also included are six typical propulsion configurations used for a BEV and the advantages and disadvantages of these configurations. The in-wheel motor configuration is discussed in more detail at the end of this chapter.

The important parameters of the ICE vehicle used for the experiments are given in Chapter 3. The performance requirements of the converted BEV are defined, which results in specific requirements of the in-wheel motors to be placed in this vehicle. Based on these requirements, the dimensions and weight of the in-wheel motors are specified and used as guideline throughout the investigations performed in this thesis.

Chapter 4 describes the experimental setup and the different roads encountered during the experiments. Moreover, it will show the influence of in-wheel motors on the vehicle ride comfort, using them either at the front or rear of the vehicle. Since further investigations are only performed by simulations, the model representing the ICE vehicle is validated by comparing simulation results with experimental results. This validation is presented in Chapter 5.

The conversion of the ICE vehicle to a BEV is briefly described in Chapter 6. The influence of in-wheel motors on both the ride comfort and safety of this vehicle will also be addressed. This chapter shows
the possible improvements that can be obtained by optimization of the suspension system. Semi-active and active suspension systems, that can be used to further improve the suspension system, are addressed in Chapter 7. Furthermore, this chapter gives information about controller development and shows the improvements that can be achieved in the ride comfort and safety of the vehicle using an electromagnetic suspension system.

Chapter 8 will conclude the findings of the investigations and will provide recommendations for future research.

The project presented in this report has resulted in a paper for the 10th International Symposium on Advanced Vehicle Control (AVEC 10). This paper is given in Appendix F.
Chapter 1. Introduction
Chapter 2

Battery electric vehicle drive train

Compared to a conventional ICE vehicle, the drive train of a BEV is much more flexible. The increased flexibility is mainly due to the energy flow via electrical wires rather than mechanical shafts. These flexible wires make it possible to distribute the subsystems of the drive train more freely. The flexibility is increased due to the many options in the two major subsystems of the drive train, the energy source system and the electric propulsion system. This increase in flexibility leads to several propulsion configurations that can be used for a BEV, each with its own advantages and disadvantages. This chapter will discuss these possible propulsion configurations and their (dis-)advantages.

The main components of the two subsystems of the drive train, the batteries and the electric motors, are discussed more in detail in Section 2.1. Several options of propulsion configurations are discussed in Section 2.2. One specific configuration that has gained thorough attention throughout the literature, the one in which the drive motors are mounted inside the wheels, are discussed in more detail in Section 2.3.

2.1 Subsystems

This section discusses the subsystems of the BEV drive train and gives an overview of choices that can be made within the main parts of the two major subsystems.

The two major subsystems of the BEV drive train are the energy source system and the electric propulsion system. The energy source system comprises the batteries, energy management unit and battery charging unit. The electric propulsion system consists of an electronic controller, power converter, electric motor and possibly a mechanical transmission. Moreover, the drive train consists of a third subsystem that comprises the power steering unit, temperature control unit and auxiliary power supply [23]. An overview of the topology of an electric vehicle containing the first two subsystems is shown in Fig.2.1. The black arrows illustrate an electrical link, while the white arrows illustrate control links.

The increase in flexibility in the drive train compared to an ICE vehicle attributed to the energy source system and the propulsion system are the different weights, shapes and sizes of the batteries and electric motors. Both are discussed in more detail next.
2.1.1 Batteries

A battery is a device that can transform stored chemical energy into electrical energy and vice versa. The most commonly electrochemical batteries used in automotive applications are:

- Lead-acid
- Nickel-cadmium (NiCd)
- Sodium-Nickel (ZEBRA)
- Nickel-metal hydride (NiMH)
- Lithium-ion (Li-ion)

The most important specifications of a battery are the specific energy in Wh/kg and the specific power in W/kg. This specific energy is defined as the ratio of the battery’s available energy to its weight, and thus determines the overall weight of the battery pack for a given energy content. The specific power is defined as the ratio of power available from the battery to its weight, and therefore determines the maximum amount of power that can be delivered to the electric motor. The battery types listed in Table 2.1 are compared based on these two values, together with the cycle life and costs. Do note that these values can vary slightly, depending on the type and application of the battery. For now, only average values are given to give a general idea of the specifications of the batteries.

The lead-acid battery is widely used within the 12V network of conventional vehicles. They are considered to be cheap, robust and reliable. The specific energy is about 35 Wh/kg and the specific power is around 175 W/kg. The very low cycle life of around 400 cycles is the downside of this battery. The NiCd battery has a higher specific energy, around 50 Wh/kg and has about twice the cycle life of the lead-acid battery. However, it also has a higher purchase price and a lower specific power of around 125 W/kg, making it less suitable for electric vehicles. The ZEBRA battery has an specific energy of around 110 Wh/kg and a specific power of around 160 W/kg. The downside of this battery is the high working temperature of around 300 degrees Celsius, which leads to standby energy losses of around 100 W. These batteries are for example used in the THINK City battery electric vehicle [3]. The NiMH battery contains around the same energy as the NiCd battery, around 55 Wh/kg, and has about the same cycle life an purchase price. On the other hand, the specific power is a lot higher, varying between 250 and 1000 W/kg, making it an interesting battery for automotive applications. The NiMH is already successfully used in production electric vehicles like the Toyota Prius [4]. The amount of specific energy and specific power of the Li-ion battery depends on the chemistry of the battery. The most common chemistries used in the automotive sector are the Lithium Iron Phosphate (LiFePO4) and the Lithium Cobalt Oxide (LiCoO2). The specific energy and the specific
power of the LiFePO4 are about twice as much as for the NiMH battery, about 110 Wh/kg and 1100 W/kg. The LiCoO2 battery has a slightly higher specific energy of around 125 Wh/kg and a higher specific power of around 1250 W/kg. These batteries are therefore already used in high performance prototype electric vehicles, like the Tesla Roadster [5]. However, they are still expensive and further development is needed to increase the robustness of the battery. The latest development is the Lithium-Ion Polymer battery, which has a specific energy ranging from 130 - 200 Wh/kg and a specific power up to 3000 W/kg. The downside of this battery is the high purchase price and poor safety rating. A general comparison of the discussed batteries is shown in Table 2.1 [35, 45, 23, 41, 25].

Table 2.1: General comparison of batteries on basis of specific energy, specific power, cycle life and costs.

<table>
<thead>
<tr>
<th>Battery type</th>
<th>Specific energy [Wh/kg]</th>
<th>Specific power [W/kg]</th>
<th>Cycle life [cycles]</th>
<th>Estimated costs [US$/kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lead-Acid</td>
<td>30-40</td>
<td>150-200</td>
<td>300-500</td>
<td>100-150</td>
</tr>
<tr>
<td>NiCd</td>
<td>40-60</td>
<td>100-150</td>
<td>1000-2000</td>
<td>250-350</td>
</tr>
<tr>
<td>Zebra</td>
<td>90-120</td>
<td>130-180</td>
<td>1000-1500</td>
<td>250-350</td>
</tr>
<tr>
<td>NiMh</td>
<td>30-80</td>
<td>250-1000</td>
<td>1000-2000</td>
<td>200-350</td>
</tr>
<tr>
<td>LiFePO4</td>
<td>80-120</td>
<td>800-1400</td>
<td>&gt; 2000</td>
<td>&gt; 300</td>
</tr>
<tr>
<td>LiCoO2</td>
<td>110-140</td>
<td>1100-1400</td>
<td>&gt; 800</td>
<td>&gt; 500</td>
</tr>
<tr>
<td>Lithium-ion Polymer</td>
<td>130-200</td>
<td>3000</td>
<td>&gt; 1000</td>
<td>&gt; 1000</td>
</tr>
</tbody>
</table>

2.1.2 Electric motors

The motor in an electric vehicle is used to convert the electrical energy from the battery into mechanical energy to drive the wheels. Additionally, it can convert mechanical energy, gained by regenerative braking, into electrical energy to recharge the battery. The electric motor can be classified into two categories:

- the direct-current (DC) motor
- the alternating-current (AC) motor

Both motors consist of a stationary part, the stator, and rotating part, the rotor. The stator is connected to the vehicle body, while the rotor is connected direct or indirect by means of drive shafts and/or differentials to the wheels of the vehicle. The working principle of both categories is described below.

When a voltage is applied to the rotor windings in a DC-motor through contact between the brushes and the commutator, a magnetic field is generated. At the same time, a stationary magnetic field is created in the stator by permanent magnets. The interaction of these two fields causes the rotor to turn. The direction of current through the rotor windings is changed by the commutator, reversing the magnetic field and keeping the rotation of the rotor going. In AC motors a magnetic field is generated by applying a voltage to the stator. The orientation of this field is changed according to the sign of the current flowing through the stator, resulting in a rotating magnetic field. The interaction of the rotor with the magnetic field of the rotor makes it revolve around its axis.

The creation of the magnetic field in the rotor of an AC-motor depends on the type of AC-motor:

- induction
- permanent magnet synchronous (PMS)
- permanent magnet brushless DC (PMBDC)
- switched reluctance (SR)
Chapter 2. Battery electric vehicle drive train

The first two AC-motors are fed by a sinusoidal supply, while the last two are fed by a rectangular supply. In an induction motor a rotating magnetic field is created by a current in the stator, which induces current in the rotor conductors. This current interacts with the magnetic field of the stator, which causes a rotation of the rotor. In order to produce torque the speed of the rotor must differ from the speed of the rotating magnetic field. Because of this they are also often called asynchronous AC motors. In a synchronous motor the rotor has its own magnetic field, often achieved by permanent magnets. The rotor operates at the same speed as the rotating magnetic field of the stator. The creation of an magnetic field in the rotor of an PMBDC motor is the same as in the PMS motor. In fact, the only difference between the two is the way of voltage supply, which is rectangular or sinusoidal, respectively. The term brushless DC stems from the PMBDC approximating the operation of a DC motor. In a SR motor the stator and rotor consist of salient poles (or "teeth"). Each stator pole carries an excitation coil, forming one phase with an opposite coil, while the rotor does not have any windings or magnets. This means that the rotor does not have a magnetic field. When a voltage is supplied to a phase, the rotor will rotate in order to minimize the reluctance of the magnetic path \[41, 22, 35\]. Schematics of these motors and the schematic of a DC-motor can be found in Fig. 2.2. In the 1970s, several electric vehicle prototypes used a DC-motor because they were more easy to control. An example of this is the Citicar produced by the short-lived company Sebring Vanguard. However, the AC-motor was found to have a better efficiency, specific power, robustness and reliability and needs less maintenance due to the absence of brushes. These advantages and advancements in power electronic devices, new materials and modern control algorithms, resulted in the fact that more recent electric vehicles use AC-motors. Although the purchase price of an AC-motor is often lower, they do require more sophisticated control electronics, making them overall more expensive than DC-motors \[41, 22, 35\].

Comparing the AC-type motors it can be noted that the induction motor generally has a higher specific power than the permanent magnet motors. However, the induction motor has a lower peak efficiency as a result of the losses caused by the induced current in the rotor. In permanent magnet motors the power losses are only found in the stator, making heat removal also somewhat more simple. On the contrary, induction motors are generally able to bear higher rotational speeds. Due to these reasons, induction motors are mostly preferred for high power applications, while the permanent magnet motors are mostly preferred for vehicles where the efficiency is a bigger issue. For example, the high performance Tesla Roadster uses an induction motor to drive the rear wheels, while the efficiency oriented Mitsubishi i MiEV uses a PMS motor \[5, 44\]. The PMBDC motors usually have a higher specific power than the PMS motors. This is because the interaction between rectangular field and rectangular current in the motor can produce a higher torque than that produced by a sinusoidal field and sinusoidal current. On the other hand, the torque produced in the synchronous motor is much smoother. The peak efficiency of the SR motor is slightly below the peak efficiency of the permanent magnet motors, but their efficiency is maintained over a wider range of speed and torque than the other motors. The SR motor is gaining more interest due to its simple and cost effective construction. However, they are more difficult to control, have some problems with acoustic noise and have a non-uniformity of operation due to torque ripples \[41, 22, 35, 45, 58, 54\]. A general comparison of the motor weight, overall efficiency of motor and power electronics and cost can be found in Table 2.2 [63].

![Schematics of four types of electric motors.](image)

**Figure 2.2:** Schematics of four types of electric motors.
2.2. Electric vehicle propulsion configurations

The number 5 represents the lowest weight, highest efficiency and the lowest cost.

Table 2.2: General comparison of electric motors on basis of weight, efficiency and costs [63]. The number 5 represents the lowest weight, highest efficiency and the lowest cost.

<table>
<thead>
<tr>
<th>Machine type</th>
<th>Weight</th>
<th>Efficiency</th>
<th>Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>DC</td>
<td>2</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>AC induction</td>
<td>4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>PM</td>
<td>4.5</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>SRM</td>
<td>5</td>
<td>4.5</td>
<td>4</td>
</tr>
</tbody>
</table>

Due to these options and the fact that electric motors have a higher torque response it may be possible to develop lighter, more compact and more efficient propulsion configurations. The possible propulsion configurations are described below.

2.2 Electric vehicle propulsion configurations

The six typical electric vehicle configurations that exist due to the variations in electric propulsion systems are shown in Fig. 2.3 [23]. Note that these configurations all consist of front wheel drive. Rear wheel drive or four wheel drive can also be adopted.

The first configuration is directly derived from the conventional ICE vehicle and consists of an electric motor, a clutch, a gearbox and a differential. The use of a gearbox in combination with a clutch gives the opportunity to control the amount of torque exerted by the wheels on the road. The differential allows the wheels to rotate at different speeds during cornering. Since it is directly derived from a conventional ICE configuration, the implementation is easy. However, the clutch as used in conventional ICE vehicles becomes redundant, since electric motors have the ability to generate torque at zero speed and can operate efficiently over a wide speed range. To match the high speed of the motor with the speed of the wheels a single gear ratio is sufficient. This is taken into account in configuration b, which consists of only an electric motor, a fixed gear and a differential. By integrating the electric motor, fixed gearing and a differential into a single assembly, the more compact configuration c arises, allowing a more flexible design for the rest of
Chapter 2. Battery electric vehicle drive train

The configuration resembles the transverse front-engine front-wheel drive system of an ICE vehicle. A fixed gear transmission is stronger, more compact and has a higher efficiency than the variable gear transmission. Combined with the removal of the clutch, the system is therefore reduced in complexity, weight, size and costs and has less torque interrupts [23]. These advantages result in most electric vehicles adopting the configuration with the electric motor, fixed gearing and differential integrated into a single assembly. For example, the Tesla Roadster and Mini-Cooper E use the AC induction motor in combination with a single speed fixed gear transmission to drive the rear or front wheels, respectively [5, 6]. The Mitsubishi i MiEV use a differential gear, which also functions as a fixed-ratio reduction gear, to transmit the torque generated by the PMS motor to the rear wheels [44]. Honda equips the Honda FCX Clarity Fuel Cell Electric Vehicle with the compact 100 kW PMS motor as shown in Fig.2.4 [7].

![Figure 2.4: Clarity motor of the Honda FCX Clarity Fuel Cell electric vehicle.](image)

In configuration d the differential is removed and replaced by another electric motor with fixed gearing. Now each wheel is driven by its own electric motor. The differential action is electronically controlled instead of using mechanical means, resulting in a higher overall efficiency. However, the reliability of the electronic differential is less compared to a purely mechanical construction and should be taken into account, although this reliability has greatly improved in recent years [23, 41]. This configuration is for example used by Tahami et al. [57]. They converted an ICE vehicle to an electric vehicle for experimental purposes. The propulsion system consisted of two permanent magnet synchronous motors, connected to the sprung mass, driving the front wheels by a reduction gear and drive shafts.

Configuration e and f are the so-called indirect-drive and direct-drive in-wheel configurations, with or without the presence of a fixed gearing, respectively. The in-wheel configuration with fixed gearing is used in combination with a high-speed permanent magnet inner-rotor type. The second configuration is used for the low-speed permanent magnet outer-rotor type. An example of both configurations is shown in Fig.2.5 [23]. The advantage of using the high-speed inner-rotor configuration is that it allows the motor to operate at much higher speeds for a given vehicle speed. Since only low torque is required at higher speeds, the motor has a smaller size, lower weight and lower costs. Using a low-speed outer-rotor motor, the motor speed is equivalent to the speed of the wheels, resulting in a bigger size and weight of the motor. However, the absence of the fixed gearing reduces the complexity, improves the overall reliability and increases the efficiency due to the lack of transmission losses [23, 27]. As example, the last configuration is used by Terashima et al.. They developed a high-performance electric motor vehicle, called the IZA. This vehicle uses four direct-driven AC synchronous in-wheel motors [58].

Combinations of configurations can also be found. As example, the "UOT Electric March II", built by the Tokyo University, remodeled a Nissan March, using four permanent-magnet motors with a built-in
2.3. In-wheel motors: advantages and disadvantages

In-wheel motors can be categorized into two main configurations:

(a) Indirect-drive in-wheel motor with gear reduction
(b) Direct-drive in-wheel motor without gear reduction

Figure 2.5: Two different in-wheel motor configurations.

The centralized configuration appears to be one of the most popular configurations due to its similarity with existing systems. However, the direct-drive in-wheel motor configuration is less complex and more efficient due to the absence of gears. The increased efficiency is especially of great importance for a sustainable BEV. Additional advantages that the in-wheel motor configuration offers over the centralized configuration are explained in the next section. However, the use of in-wheel motors do increase the weight of the unsprung mass, which has a negative effect on the ride comfort and road holding capability of the vehicle. These disadvantages are discussed in more detail in the next section.

2.3 In-wheel motors: advantages and disadvantages

In addition to the increased simplicity and efficiency as described above, in-wheel motors give the opportunity to provide torque to each wheel independently. This can be used as a type of steering control input, making stability control possible. Control of the vehicle using this method, usually called Direct Yaw-moment Control, seems to be more effective in enhancing vehicle stability than four wheel steering. Existing systems like ABS and TCS can be implemented easily at low cost. Moreover, drive shafts, supporting systems and differentials become unnecessary, resulting in an overall decrease of size, weight and reduction gear. At the front the motors are placed at the ends of each driving shaft and attached to the base chassis. At the rear, in-wheel motors are used. The front and rear motors are shown in Fig.2.6 [38].

Figure 2.6: Front and rear motors of the UOT Electric March II built by the Tokyo University [38].
Chapter 2. Battery electric vehicle drive train

transmission losses. This does not only result in a higher overall efficiency, but this also creates more space for the interior of the vehicle [57].

These advantages lead to the fact that extensive research within the field of in-wheel motor design is being performed [54, 27, 62, 21]. For example, companies like E-traction and PML Flightlink are designing in-wheel motors. E-traction designed TheWheel™, which is a direct-drive outer-rotor motor [8]. Meanwhile, PML Flightlink equipped several vehicles with their designed direct-drive in-wheel motors called Hi-Pa Drive™. These include a four wheel drive, plug-in hybrid Mini called Mini QED, a four wheel drive, full electric Ford 150 called Ford Hi-Pa Drive and a four wheel driven, plug-in hybrid Volvo C30 called Volvo C30 Re-Charge [9]. The implementation of the TheWheel™ and the Hi-Pa Drive in-wheel motors are shown in Fig. 2.7 and Fig. 2.8, respectively.

However, these advantages are accompanied with one big disadvantage: in-wheel motors increase the vehicle unsprung mass. This results in a decrease in ride comfort, in a decrease of road holding capability and in an increase in suspension travel. This can be explained as follows. Due to the increase in unsprung mass, the unsprung-to-sprung mass ratio $\left(\frac{m_{us}}{m_{s}}\right)$ of the vehicle changes. This change in ratio has in turn an effect on the transmissibility ratio, suspension travel ratio and dynamic tyre deflection ratio. The transmissibility ratio gives the response of the sprung mass to the excitation from the road and can be used for assessing the ride comfort of a vehicle. This will become clearer further in this report. The suspension travel ratio is defined as the ratio of the maximum relative displacement between the sprung and unsprung mass to the amplitude of the road input. It can therefore be used for assessing the space required to accommodate the suspension spring. The dynamic tyre deflection ratio is defined as the ratio of the maximum relative displacement between the unsprung mass and the road surface to the amplitude of the road input. This ratio determines the amount of dynamic wheel load, which is a measure for the road holding capability: a high dynamic wheel load means more vertical load variations on the wheels, resulting in a loss of average side force of the tyres. This loss in side force can be divided into a ‘static’ loss and ‘dynamic’ loss. The static loss arises due to a diminishing average cornering stiffness. The dynamic loss is attributed to the rate of change of the relaxation length with wheel load: each time the vertical load changes, the tyre has to travel some distance again before the maximum side force can be obtained. More information about these effects can be found in Chapter 8.2 of [52].

The general effect of an increased unsprung-to-sprung mass ratio on the three ratios for a two-degree-of-freedom system can be found in Fig. 2.9 [61]. A decrease in all three ratios can be considered a positive change, while an increase is considered to be negative. As shown, an increase in unsprung-to-sprung mass ratio has almost no effect in the frequency range below the natural frequency of the sprung mass (around 1 Hz). However, in the frequency range between the natural frequency of the sprung mass and the natural frequency of the unsprung mass, it can be noted that all three ratios are increased. Above the natural frequency of the unsprung mass the transmissibility ratio and suspension travel ratio are slightly decreased,
2.3. In-wheel motors: advantages and disadvantages

while it has a relatively insignificant effect on the dynamic tyre deflection. Hence, the in-wheel motors result in a decrease in ride comfort and dynamic wheel load of the vehicle and in an increase in suspension travel.

Figure 2.9: Effect of unsprung-to-sprung mass ratio on the transmissibility, suspension travel and dynamic tyre deflection ratios [61].

Due to the many advantages of in-wheel motors, several companies are working on in-wheel motor designs that minimize these disadvantages. For example, Michelin equipped the Heuliez Will electric vehicle with its own designed Active Wheel Drive system shown in Fig. 2.10 (left) [10]. This system consists of a fixed gear in-wheel motor with an integrated active suspension system. This active suspension is operated by a second electric motor via a gear rack and pinion. A single lower control arm suspension arrangement attaches the wheel to the vehicle body. Fig. 2.10 (right) shows the eCorner module created by Siemens. This module consists of an electric drive motor, an active suspension system and Siemens electronic wedge brake-by-wire system [11]. These systems replace the conventional wheel suspension, hydraulic shock absorbers, mechanical steering, hydraulic brakes and the ICE. Especially the use of an active suspension system can possibly decrease the negative effects of the increased unsprung mass.

Figure 2.10: Two existing in-wheel motor configurations: 1 - rim, 2 - electric motor, 3 - brake, 4 - suspension spring, 5 - active suspension, 6 - electrical suspension motor, 7 - electronic steering.

Bridgestone has invented the so called Dynamic-Damping In-wheel Motor Drive System as shown in Fig.2.11 to deal with the negative side-effects of in-wheel motors [12]. The system comprises a shaft-
less direct-drive motor that is connected to the wheel by a flexible coupling. Four coil springs and two tubular dampers insulate the motor from the unsprung mass. Bridgestone claims that the motor works as a vibration damper, since the vibration of the motor cancels the vibration input from the road irregularities. This results in better road-holding performance and a more comfortable ride with respect to conventional in-wheel systems and single-motor electric vehicles.

![Figure 2.11: Bridgestone’s Dynamic-Damping In-wheel Motor Drive System [12].](image)

### 2.4 Conclusion

In this chapter the main components of the BEV drive train are described and the advantages and disadvantages of six possible propulsion configurations for a BEV are discussed. It has become clear that the in-wheel configuration is a very interesting and innovative configuration for a BEV, that offers the following advantages over the centralized configuration:

- A possible more stable controlled vehicle by independently driving and braking the wheels
- Easier and less expensive implementation of existing systems like ABS and TCS
- Transmission, drive shafts, differentials and supporting systems are redundant, which results in:
  - a lower overall weight
  - a higher efficiency
  - a less complex system
  - more interior space

However, due to the increase in unsprung mass, the use of in-wheel motors leads to a deterioration of the ride comfort and load holding of the vehicle. Knowledge of the severity of these disadvantages will play an important part in the successful implementation of in-wheel motors in future vehicles. Therefore, the goal of this thesis is to investigate the severity of the negative effects of an increased unsprung mass. This investigation is described in the following chapters of this thesis.

The parameters of the test vehicle and in-wheel motors are discussed first in the next chapter.
Chapter 3

Vehicle and in-wheel motor parameters

In the previous chapter it has become clear that the use of in-wheel motors is a very interesting and innovative configuration to be used for a BEV. However, it does decrease the ride comfort and road holding of the vehicle as a result of the increase in unsprung mass. In order to perform experiments and simulations with a validated model to investigate these negative effects, several parameters of the test vehicle and the in-wheel motors need to be determined. This is done in this chapter.

Due to simplicity and cost, the test vehicle is not equipped with real in-wheel motors during the experiments. They are only emulated by adding a set of weights, representing the motor stator and rotor, to the unsprung mass. The important parameters of these weights, i.e. dimensions, weight and inertia, depend on the amount of torque and power they have to produce. These depend in turn on the performance demands of the BEV. These demands and the requested torque and power are therefore also described in this chapter.

Section 3.1 provides the parameters of the test vehicle. These are important for the simulation model and for determining the requirements for the motors. The motor requirements are given in Section 3.2.1, based on which the main parameters of the in-wheel motors are determined as described in Section 3.2.2. This chapter ends with a discussion, given in Section 3.3.

3.1 Vehicle parameters

The vehicle used for the experiments is the Volkswagen (VW) Lupo 3L. This vehicle is currently being converted into a BEV at the Eindhoven University of Technology to gain practical experience in the field of electric vehicles. The Lupo 3L is chosen because it matches the dimensions and weight of the car design. The resemblance of the Lupo 3L with the car can be clearly seen in Fig. 3.1, illustrating the dimensions of the car with the overlapping dimensions of the Lupo 3L. The important dimensions and other known parameters of this vehicle are provided in Table 3.1. The cargo represents the mass of a driver and passenger, which was also the occupation of the vehicle during the on-road experiments. More specific information on the weight distribution is given in Appendix A.1.

There are still some additional important unknown parameters, such as the damping and stiffness characteristics of the suspension system of the Lupo. These parameters are important factors determining the comfort and road holding capability of the vehicle. For example, a good ride comfort is obtained by using a soft suspension system to isolate the sprung mass from road disturbances. However, good road holding is obtained by a stiffer suspension system. This means that the parameters have a high influence on the simulation results and as such, they need to be determined accurately. These suspension parameters are described in the next section. The parameters of the tyres also play an important role in the simulation results. For example, a lower vertical stiffness usually absorbs more vibrations from the road input, resulting in a better comfort. The effective rolling radius of the tyre also needs to be known, since this determines...
Figure 3.1: Design of the car of the year 2020 with the overlapping dimensions of the VW Lupo 3L.

Table 3.1: Parameters of the VW Lupo 3L.

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheelbase</td>
<td>( l )</td>
<td>2.321</td>
<td>m</td>
</tr>
<tr>
<td>Track width</td>
<td>( w )</td>
<td>1.425</td>
<td>m</td>
</tr>
<tr>
<td>Total height</td>
<td>( h )</td>
<td>1.484</td>
<td>m</td>
</tr>
<tr>
<td>Vehicle mass unloaded</td>
<td>( m )</td>
<td>870</td>
<td>kg</td>
</tr>
<tr>
<td>Cargo</td>
<td>( m_c )</td>
<td>160</td>
<td>kg</td>
</tr>
<tr>
<td>Distance of CG to front wheels</td>
<td>( a )</td>
<td>0.892</td>
<td>m</td>
</tr>
<tr>
<td>Distance of CG to rear wheels</td>
<td>( b )</td>
<td>1.429</td>
<td>m</td>
</tr>
<tr>
<td>Frontal area</td>
<td>( A )</td>
<td>1.97</td>
<td>( m^2 )</td>
</tr>
<tr>
<td>Air drag coefficient</td>
<td>( c_d )</td>
<td>0.29</td>
<td>-</td>
</tr>
<tr>
<td>Tyre rolling resistance coefficient</td>
<td>( c_r )</td>
<td>0.0085</td>
<td>-</td>
</tr>
</tbody>
</table>
the amount of torque necessary to apply a certain amount of horizontal force on the ground. Moreover, it
determines the rotation speed of the motor for specific forward velocities of the vehicle. The important tyre
parameters are described in Section 3.1.2.

3.1.1 Suspension parameters

The VW Lupo 3L is equipped with an independent McPherson suspension at the front and a semi-independent
suspension at the rear. The stiffness and damping characteristics of both suspension systems are determined
using the quarter-car test setup available at the Department of Electrical Engineering at the Eindhoven Uni-
versity of Technology. The stiffness of the springs is determined by measuring the force created by slowly
compressing and extending the spring. The damping characteristics are determined by exciting the damper
sinusoidally around a specific equilibrium point. This point coincide with the nominal working point of
the damper at nominal load. By varying the frequency of the sine, the velocity of the damper when it
crosses the equilibrium point can be changed. By determining the force at each velocity, the damping force
characteristics can be established. More information about this can be found in Appendix A.2. The charac-
teristics of the spring and damper in the McPherson strut are illustrated in Fig. 3.2 (a) and (b), respectively.
The corresponding characteristics of the rear suspension are shown in Fig. 3.3 (a) and (b), respectively. The
resulting spring curves and damping data points are fitted numerically by drawing straight lines through
the measurement data. Only in the characteristics of the front spring an exponential curve is fitted through
the measurement data at a suspension travel of around 0.04 m. The derivatives of these fitted lines give the
spring stiffness and the local damping constant of the suspension systems.

As displayed in Fig.3.2 and Fig.3.3 (a), a small force difference exists between the inward and outward
stroke of the spring. This is due to hysteresis in the system. Fig.3.3 (b) shows that for the rear springs a
linear relation exists between the suspension travel and force throughout the whole measured range. For
the front springs this linear relation only holds up to a suspension travel of 0.02 m. For a higher suspension
travel the force increases rapidly due to the bump stop. This bump stop is implemented in the system to
prevent the strut from reaching the end of its stroke when highly compressed. The spring stiffness in the
linear area is calculated to be 23.5 kN/m at the front and 13.9 kN/m at the rear. However, due to the design
of the suspension system, this spring stiffness is not equal to the effective spring stiffness at the wheels
[24]. This effective stiffness depends on the the installation ratio, which gives the ratio of the wheel travel
to the shock travel. The effect of this ratio on the effective spring stiffness is proportional to the installation
ratio squared. The installation ratio of the rear suspension spring in the Lupo is 350 : 1 , which gives a
vertical stiffness at the wheel center of 10.2 kN/m. At the front the installation ratio is close to 1:1.

Fig.3.2 and Fig.3.3 (b) show that both dampers have a highly non-linear characteristic, a low damping coeff-
icient on compression and a high damping coefficient on extension. This non-linear damping characteristic
Chapter 3. Vehicle and in-wheel motor parameters

(a) Spring force characteristics

(b) Damping force characteristics

Figure 3.3: Characteristics of the rear suspension system.

is common in conventional passenger vehicles [19]. The effective damping coefficient at the wheels also depend on the installation ratio, which in the case of the rear dampers is \( \frac{445}{410} : 1 \). At the front the damper installation ratio is close to 1:1.

3.1.2 Tyre parameters

Several parameters of the tyres are determined by experiments on the flat-plank tyre test setup available at the Department of Mechanical Engineering at the Eindhoven University of Technology. The inflation pressure of the tyre is chosen to be 2.4 and 3.0 bar for the experiments. A pressure of 2.4 bar is chosen because this is the tyre pressure recommended by the vehicle manufacturer. However, since the Lupo 3L is specifically designed for economical driving, most owners use a relatively high pressure of 3.0 bar. At this higher pressure, the bending and shearing of the tyre’s tread is decreased and the rolling resistance is reduced. Therefore, the tyre is also tested at a pressure of 3.0 bar. This pressure is also used in the tyres during the on-road experiments.

Fig.3.4 shows the tyre deflection versus the vertical tyre force for a pressure of 2.4 and 3.0 bar. As shown, the stiffness of the tyre is highly pressure dependent. There is also a non-linear relation between the vertical tyre force and the tyre deflection. According to Besselink et al. [20] this non-linear relationship can be

Figure 3.4: Vertical tyre force as function of tyre deflection.
3.1. Vehicle parameters

modeled as:

\[
F_z = \left( 1 + q_{v2} \frac{R_0}{V_0} |Ω| \right) \left( \frac{q_{Fcx} F_x}{F_{z0}} \right) ^2 - \left( \frac{q_{Fcy} F_y}{F_{z0}} \right) ^2 \left( q_{Fz1} \frac{ρ}{R_0} + q_{Fz2} \left( \frac{ρ}{R_0} \right) ^2 \right) (1 + p_{Fz1} dp_i) F_{z0}
\]

(3.1)

with \( q_{v2} \) a stiffness increase with velocity, \( q_{Fcx} \) and \( q_{Fcy} \) the vertical sinking of the tyre due to longitudinal and lateral forces, \( q_{Fz1} \) and \( q_{Fz2} \) characterize the force deflection, \( R_0 \) is the non-rolling free tyre radius, \( p_{Fz1} \) the influence of the tyre inflation pressure, \( F_{z0} \) the nominal load and \( dp_i \) the non-dimensional pressure increment calculated by:

\[
dp_i = \frac{p_i - p_{i0}}{p_{i0}}
\]

(3.2)

with \( p_{i0} \) the nominal tyre inflation pressure. However, the stiffness increase with velocity and the vertical sinking of the tyre due to longitudinal and lateral forces are unknown. Therefore, Eq. (3.1) is reduced to:

\[
F_z = \left( q_{Fz1} \frac{ρ}{R_0} + q_{Fz2} \left( \frac{ρ}{R_0} \right) ^2 \right) (1 + p_{Fz1} dp_i) F_{z0}
\]

(3.3)

Using the measurement results, a value of 4000 N for the nominal load \( F_{z0} \) and 2.4 bar for the nominal tyre inflation pressure \( p_{i0} \), the following values are found: \( q_{Fz1} = 11.6455 \), \( q_{Fz2} = 22.058 \) and \( p_{Fz1} = 0.53 \). The resulting fit of this equation can also be found in Fig. 3.4. Note that it is hard to understand from these values what the vertical stiffness of the tyres is in N/m. However, for loads between 1000 N and 6000 N, an almost linear relationship can be found between the vertical load and deflection of the tyres. The vertical stiffness of the tyre based on this approach are given in Table 3.2.

<table>
<thead>
<tr>
<th>Tyre pressure</th>
<th>Vertical stiffness (k(_t))</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.4 bar</td>
<td>2.07e5 N/m</td>
</tr>
<tr>
<td>3.0 bar</td>
<td>2.35e5 N/m</td>
</tr>
</tbody>
</table>

The effective rolling radius, relating the angular velocity of a freely rolling wheel (\( ω \)) with the forward velocity of the vehicle (\( V_x \)), can be calculated by:

\[
Re = R_0 - \frac{F_{z0}}{c_z} \left( D_{reff} \arctan \left( B_{reff} \frac{F_x}{F_{z0}} \right) + F_{reff} \frac{F_x}{F_{z0}} \right)
\]

(3.4)

Where \( D_{reff} \), \( B_{reff} \) and \( F_{reff} \) are model parameters and \( c_z \) the vertical stiffness at the nominal vertical load, nominal inflation pressure, no tangential forces and zero forward velocity, given by:

\[
c_z = \frac{F_{z0}}{R_0} \sqrt{q_{Fz1}^2 + 4q_{Fz2}^2} \cdot (1 + p_{Fz1} dp_i)
\]

(3.5)

Using the measurement results the following values are found: \( c_z = 2.13e5 N/m \), \( D_{reff} = 0.2555 \), \( B_{reff} = 4.9884 \) and \( F_{reff} = 0.001 \). The results are shown in Fig.3.4. Additional tyre parameters playing a role in the handling of the vehicle are the relaxation length, cornering stiffness and pneumatic trail. These values can be found in Appendix A.3.
Chapter 3. Vehicle and in-wheel motor parameters

3.2 Electric propulsion system

Now that the important parameters of the vehicle and tyres are known, the important parameters of the in-wheel motors can be determined. These are based on the amount of torque and power they have to produce in order to meet the performance demands of the BEV. These requirements are discussed initially, followed by the determination of the motor parameters.

As discussed in Chapter 2, a direct-drive motor or a motor with gear reduction can be used as in-wheel motor. The specifications of the motor are therefore depending on the type selected. The direct-drive type is chosen, because, although it tends to be bigger and heavier, it is less complex, more reliable and more efficient. Especially the efficiency is of crucial importance for a BEV. Furthermore, it will represent a worse case scenario (maximum additional mass).

3.2.1 Motor requirements

There are six important vehicle requirements that are used to determine the performance of the in-wheel motors:

- A top speed of 130 km/h
- Continuously drive a slope of 10% at 80 km/h
- Continuously drive a slope of 15% at 50 km/h
- Drive a slope of 30% at 15 km/h for a short period of time
- Accelerate from 0 to 100 km/h in less than 14 seconds
- Drive up a curb from standstill

The first requirement is due to the fact that 130 km/h is the maximum allowed vehicle speed throughout Europe. Although it should be able to drive at this speed, it should also be limited to 130 km/h due to constraints in the capacity of the battery. The second and third requirement are setup since not all roads are perfectly horizontal. According to Mitschke [50], the steepest permissible slope on public road is 15% which the vehicle needs to be able to drive continuously. The fourth requirement is needed to drive up slopes often found in for example parking garages. The fifth requirement stems from the assumption that the vehicle has to be competitive with existing vehicles of comparable size and weight in order to appeal...
3.2. Electric propulsion system

to the public. Since the magazine "Autoweek" showed that for 7 compact-class vehicles the average acceleration time from 0 to 100 km/h is 14 seconds [13], the vehicle is required to have the same performance. The last requirement is setup since curbs can be encountered during for example parking.

The motor requirements can now be determined taking into account the external longitudinal forces (road load) acting on the vehicle during driving. These forces include aerodynamic drag forces, longitudinal tyre forces, rolling resistance forces, gravitational forces and internal friction forces. The longitudinal dynamics of the vehicle are given by the following equation:

\[
(m_v + \frac{4J_w}{R_e^2}) \cdot a = F_t - F_{aero} - F_r - F_g - F_e
\]  

(3.6)

with \( m_v \) the total mass of the vehicle \([\text{kg}]\), \( J_w \) the moment of inertia of a rotating wheel \([\text{kgm}^2]\), \( R_e \) the effective rolling radius of the wheel \([\text{m}]\), \( a \) the acceleration of the vehicle \([\text{m/s}^2]\), \( F_t \) the combined traction force generated by the electric motors \([\text{N}]\), \( F_{aero} \) the external force due to aerodynamic friction \([\text{N}]\), \( F_r \) the external force due to the rolling resistance of the tyres \([\text{N}]\), \( F_g \) the force acting on the vehicle by gravity when driving on a non-horizontal road \([\text{N}]\) and \( F_e \) the internal forces due to friction in for example the wheel bearings or brake drag. \( F_{air}, F_r \) and \( F_g \) are given by:

\[
F_{aero} = \frac{1}{2} \rho A_f c_d v^2
\]

(3.7)

\[
F_r = c_r m_v g \cos(\alpha)
\]

(3.8)

\[
F_g = m_v g \sin(\alpha)
\]

(3.9)

with \( \rho \) the air density \([\text{kg/m}^3]\) at sea level and atmospheric pressure, \( A_f \) the frontal area of the vehicle \([\text{m}^2]\), \( c_d \) the air drag coefficient \([-]\), \( v \) the vehicle forward velocity \([\text{m/s}]\), \( c_r \) the roll resistance coefficient \([-]\), \( g \) the gravitational constant \([\text{m/s}^2]\) and \( \alpha \) the slope of the road \([\text{rad}]\). The internal forces \( F_e \) for passenger cars can be estimated to be around 50 N [49]. The torque and power that the electric motor has to deliver to overcome the road load can now be calculated by:

\[
T_m = F_t \cdot R_e
\]

(3.10)

\[
P_m = T_m \cdot \omega
\]

(3.11)

with \( \omega \) the rotational velocity of the wheels \([\text{rad/s}]\). The torque and power needed to accelerate from 0 to 100 km/h can be estimated by [56]:

\[
P_{max} = \frac{V_0^2 m}{t_0}
\]

(3.12)

with \( V_0 \) the maximum speed \([\text{m/s}]\) and \( t_0 \) the time needed for the acceleration to \( V_0 \) \([\text{s}]\). The amount of torque needed to drive up a curb from stand can be calculated by:

\[
T = m_f g a
\]

(3.13)

with \( m_f \) the mass at the front axle \([\text{kg}]\) and \( a \) the length of the force-arm as given in Fig.3.6. The length of the force-arm \( a \) depends on the height of the curb and radius of the wheel and can be calculated by:

\[
a = \sqrt{2hr - h^2}
\]

(3.14)

with \( h \) the height of the curb \([\text{m}]\) and \( r \) the radius of the tyre \([\text{m}]\).

Using the above equations and the parameters of the vehicle as stated in Table 3.1, the requirements of the in-wheel motors are determined. The results are listed in Table 3.3. Note that, due to the conversion of the Lupo 3L into a BEV, the unloaded mass of the vehicle will become 1030 kg instead of the 870 kg as stated in Table 3.1. In combination with a single driver, the mass used to determine the requirements is therefore 1110 kg. More information on the conversion is given in Chapter 6. The amount of torque needed to drive up a curb is based on the estimation that the average curb height in the Netherlands is approximately 12 cm.
Chapter 3. Vehicle and in-wheel motor parameters

Figure 3.6: Determination of the force-arm to calculate the torque needed to drive up a curb with height $h$.

Table 3.3: Vehicle demands and the requested torque and power of the motors.

<table>
<thead>
<tr>
<th>Vehicle demands</th>
<th>Torque</th>
<th>Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top speed of 130 km/h</td>
<td>160 Nm</td>
<td>21 kW</td>
</tr>
<tr>
<td>10% slope at 80 km/h</td>
<td>375 Nm</td>
<td>30 kW</td>
</tr>
<tr>
<td>15% slope at 50 km/h</td>
<td>490 Nm</td>
<td>25 kW</td>
</tr>
<tr>
<td>30% slope at 15 km/h</td>
<td>890 Nm</td>
<td>13.5 kW</td>
</tr>
<tr>
<td>Acceleration 0-100 km/h $&lt; 14$ s</td>
<td>-</td>
<td>60 kW</td>
</tr>
<tr>
<td>Curb of 12 cm high</td>
<td>1400 Nm</td>
<td>-</td>
</tr>
</tbody>
</table>

From Table 3.3, it can be concluded that the motor has to deliver a maximum torque of 1400 Nm and a power of 60 kW. However, these amounts are only needed for a short interval and not continuously. Since a well designed permanent magnet electric motor has a typical overloading capability of more than 200-300% during several seconds [27], an electric motor with a rated torque of 490 Nm and rated power of 30 kW is sufficient to meet all requirements. Fig.3.7 shows this motor torque and the road loads at different slopes. This figure also includes a 200% overloading torque that is used for full throttle acceleration as shown in Fig.3.8. The vehicle reaches a speed of 100 km/h within 10.8 seconds.

Figure 3.7: Motor torque and road load.  
Figure 3.8: Full throttle acceleration.

These motor requirements can now be used to size the in-wheel motors, as is described in the next section.
3.2. Electric propulsion system

3.2.2 In-wheel Motor parameters

Now that the requirements of the direct-drive in-wheel motors have been established, the weight, size and inertia can be determined. The direct-drive motor can be divided in a radial flux permanent magnet (RFPM) and axial flux permanent magnet (AFPM) type. Examples of both topologies are given in Fig. 3.9. The axial type is considered to have a higher power density and a higher efficiency than the radial type. They also have a much larger diameter to length ratio, making them more compact and suitable to be placed in or close to each wheel of the vehicle. In addition, it is easier to remove heat from this lay-out [30, 62, 64, 54]. Taking these factors into account, the parameters for an AFPM are determined.

![Figure 3.9: Topologies of the radial and axial flux permanent magnet motors.](image)

Within the AFPM motors there are three different topologies: the single-sided, the double-sided and the multi-stage topology. An example of a double-sided machine is shown in Fig. 3.10.

![Figure 3.10: Topology of the double-sided slotless machine with internal stator and twin permanent magnet rotor.](image)

The mass of the rotor and stator depend on the topology of the motor. The choice of topology depends on the requested torque the motor needs to deliver. For a single-sided AFPM machine the average electromagnetic torque can be calculated by the following equation [30]:

$$T_{single} = \pi B_g Q \left( \frac{D_{out}}{2} \right)^3 \left( \frac{D_{in}}{D_{out}} \right) - \left( \frac{D_{in}}{D_{out}} \right)^3$$  \hspace{1cm} (3.15)
and for the double-sided AFPM machine:

$$T_{\text{double}} = 2 \cdot T_{\text{single}}$$  \hspace{1cm} (3.16)

with $D_{\text{in}}$ and $D_{\text{out}}$ the inner and outer diameter of the permanent magnet rotor [m], $B_g$ the average airgap flux-density [T] and $Q$ the specific electrical loading [A/m].

The diameter of the motor is restricted by the size of the rim of the Lupo 3L tyre. Therefore, the maximum diameter and length of the in-wheel motor which to fit in the wheels is 300 mm and 120 mm, respectively. However, the total diameter of the motor depends not only on the outer diameter of the rotor but also on the overlap of stator windings, on the size of the disc holding the permanent magnets and the thickness of the casing (number 2, 3 and 5 in Fig.3.10, respectively). It is estimated that this represents 10% of the total motor diameter, which results in a maximum outer diameter of the permanent magnet rotor of 270 mm. The inner diameter of the permanent magnet rotor is determined using the optimum ratio between the inner and outer diameter of the magnets. Gieras et al. [30] reports that the maximum torque is found using a ratio of $\sqrt{3}$. Therefore a value of $\sqrt{3}$ is taken for the calculations of the inner diameter of the permanent magnet rotor, which results in an inner diameter of 156 mm.

Using Eq. (3.15) in combination with these parameters, an average airgap flux-density $B_g$ and a specific electrical loading $Q$ of a PMBDC motor, 0.9 T and 50000 A/m respectively, the torque for a single-sided AFPM is calculated to be 134 Nm. As a total continuous torque of 490 Nm is requested and since it needs to be delivered by at least 2 in-wheel electric motors, each motor has to be able to deliver a continuous torque of 245 Nm. This means that a double-sided AFPM motor needs to be chosen, which will be able to deliver 268 Nm.

The total weight of the motor depends on the combined weights of the stator core and windings, the rotor, permanent magnets, bearings and shaft. The weights of these components depend on their size and material. Since the exact size of each part is not known, the exact weight of the motor can not be obtained easily. However, a specific output of approximately 1 kW/kg can be considered to be a good guideline for PMBDC motors [46]. For example, at Oxford University a water jacket cooled brushless DC in-wheel electric motor has been designed with a peak power of 50 kW, weighing just 13 kg [62]. Taken this into account, a specific weight of 1 kW/kg is chosen for the electric motor. Since a continuous power of 30 kW is necessary to meet the vehicle requirements, the combined weight of the in-wheel motors becomes 30 kg. Using 2 in-wheel motors, each motor will have a mass of 15 kg. For a double-sided AFPM motor as shown in Fig.3.10, the mass of the rotor accounts for about 50% of the total active mass [30]. Therefore the weight of the rotor and stator mass are each taken to be 7.5 kg. An overview of the parameters of the in-wheel motor is given in Table 3.4.

The maximum rotational speed of the rotor depends on the critical speed of the motor. This critical speed is the speed at which the combined mass of the rotor, load and shaft cause a deflection of the shaft. This deflection can lead to damages in the system. The critical speed depends on the masses of the rotor, load and shaft, on the modulus of elasticity of the materials, the area moment of inertia of cross-sectional areas, the length of the shaft and the location of the rotor discs on the shaft [30]. Since the determination of these values goes beyond the scope of this project, it is assumed that the maximum rotational speed of the motor is of no influence on the important parameters of the in-wheel motors.

The moment of inertia of the rotating parts of the machine depends on the mass and sizes of the permanent magnets and the mass and size of the rotor holding the magnets. Since these values are not accurately known, the moment of inertia with respect to the y-axis of the rotating parts is estimated by:

$$I_{yy} = \frac{1}{2} m r^2$$  \hspace{1cm} (3.17)
3.3. Discussion

Table 3.4: Parameters of the in-wheel electric motor.

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Airgap flux-density</td>
<td>$B_0$</td>
<td>0.9</td>
<td>T</td>
</tr>
<tr>
<td>Specific electrical loading</td>
<td>$Q$</td>
<td>50000</td>
<td>A/m</td>
</tr>
<tr>
<td>Rotor inner diameter</td>
<td>$D_{in}$</td>
<td>156</td>
<td>mm</td>
</tr>
<tr>
<td>Rotor outer diameter</td>
<td>$D_{out}$</td>
<td>270</td>
<td>mm</td>
</tr>
<tr>
<td>Length</td>
<td>$L_m$</td>
<td>120</td>
<td>mm</td>
</tr>
<tr>
<td>Torque</td>
<td>$T$</td>
<td>268</td>
<td>Nm</td>
</tr>
<tr>
<td>Power</td>
<td>$P$</td>
<td>15</td>
<td>kW</td>
</tr>
<tr>
<td>Rotor mass</td>
<td>$m_r$</td>
<td>7.5</td>
<td>kg</td>
</tr>
<tr>
<td>Stator mass</td>
<td>$m_s$</td>
<td>7.5</td>
<td>kg</td>
</tr>
</tbody>
</table>

with $m$ the total mass of the rotor and permanent magnets and $r$ the outside radius of the rotor. Using this equation, an outer radius of 135 mm and a mass of 7.5 kg, the moment of inertia of the rotating parts is calculated to be 0.068 kgm$^2$.

Based on the estimated mass and the moment of inertia with respect to the y-axis of the electric rotor, a disc, as shown in Fig.3.11, is fabricated to represent the mass and inertia of an in-wheel motor rotor. The mass used to emulate the motor stator is a simple piece of mass added to the suspension, as will be explained in the next chapter. Note that the dimensions of the disc do not coincide with the dimensions as stated in Table 3.4. This is due to additional constraints imposed by the brake and rim design of the Lupo. Also note that the moment of inertia with respect to the x-axis and z-axis are not taken into account in the disc design, since these values also depend on the stator. It is assumed that these factors do not influence the experimental results. However, it would be wise to take them into account when investigating other vehicle conditions, such as cornering behavior.

![Fabricated disc](image)

Figure 3.11: Fabricated disc and schematic representation of the discs emulating the rotor of the in-wheel electric motors.

3.3 Discussion

To meet specific performance demands, the BEV has to be equipped with in-wheel motors that can deliver a continuous torque of 490 Nm and a continuous power of 30 kW. Based on the assumption that in-wheel motors have a specific power output of 1 kW/kg, the weight of the in-wheel motors are estimated to be 15 kg each, assuming two-wheel drive. However, it should be noted that most in-wheel motors found in open literature still have a specific output of less than 1 kW/kg. Therefore it is very well possible that motors
with a specific output of 1 kW/kg are state of the art and are therefore very expensive to implement in present vehicles. Nevertheless, it is expected that in the future electric motors will improve. Because of this, it is expected that in the future the in-wheel motors able to deliver the requested power and torque will have a weight of less than 15 kg and will be more affordable.

Due to the placement of the motors in the wheels, parts such as the drive shafts are superfluous. Moreover, due to the regenerative braking ability of electric motors, braking discs and callipers can be smaller. All this can result in a decrease of the unsprung mass. As this decrease in mass is not exactly known, it is not taken into account in the investigations. However, this means that the investigation performed in this thesis by adding 15 kg to the unsprung mass of each individual wheel can be considered to be a worst case scenario.

The investigation on the negative effects of in-wheel motors by on-road experiments is described in the next chapter.
Chapter 4
On-road experiments with an ICE vehicle

In the previous chapter the dimensions, weight and inertia of in-wheel motors are determined based on vehicle requirements. Based on these parameters, steel discs are fabricated that represent the mass and inertia of an in-wheel motor rotor. These steel discs are used during the experiments to investigate the negative effects of in-wheel motors. The on-road experiments are described in this chapter. Due to the available budget and time, the on-road experiments are only used to investigate the effect on the ride comfort.

Section 4.1 will first briefly describe how the ride comfort of a vehicle can be assessed. The setup of the experiments is described in Section 4.2, followed by the experimental results as given in Section 4.3. Section 4.4 will briefly describe the subjective feeling of driver and passenger during the experiments. This chapter ends with a discussion of the performed experiments in Section 4.5.

4.1 Determination of ride comfort

The vehicle ride comfort can be determined by the intensity of the accelerations of the sprung body. The sensitivity of the human body to these accelerations is directional and frequency dependent. For example, a vertical vibration between 4 and 8 Hz, which is the vertical resonance of the abdomen, and a vibration between 1 and 2 Hz in the transverse direction, the resonance of the upper torso, are considered to be unpleasant [34, 43]. To account for this sensitivity, a weighting function according to ISO 2631:1-1997 is applied to the vertical, longitudinal and lateral accelerations. These weighting functions can be found in Appendix B. The ride (dis)comfort can now be assessed by the overall ride comfort index, calculated by:

\[ a_{\text{overall}} = \sqrt{a_x^2 + a_y^2 + a_z^2} \]  

(4.1)

with \(a_x\), \(a_y\) and \(a_z\) the frequency weighted root mean square (RMS) value of the acceleration in x, y and z direction, respectively [34]. The RMS values are calculated using the power spectral density (PSD), which shows the distribution of the power of the accelerations over the frequencies. The RMS value is given by the square root of the total area of the PSD for a certain frequency range. The PSD plots are also used throughout this thesis to show the experimental and simulation results in the frequency domain.

Over the years considerable research is conducted to come up with an overall comfort criteria. According to the ISO 2631:1-1997 and BS 6841:1987 guidelines the overall RMS value of the frequency-weighted acceleration can be ranked as listed in Table 4.1. As shown, a higher ride comfort index value is considered to be a deterioration of the ride comfort. Please take this in mind while reading this report. Note that the
Chapter 4. On-road experiments with an ICE vehicle

boundaries are difficult to determine exactly, since it varies between individuals and also depends on the state of the person (healthy, sick, tired).

Table 4.1: Scale of vibration discomfort as suggested in ISO 2631:1-1997.

<table>
<thead>
<tr>
<th>RMS weighted acceleration</th>
<th>Comfort</th>
</tr>
</thead>
<tbody>
<tr>
<td>Less than 0.315 m/s²</td>
<td>not uncomfortable</td>
</tr>
<tr>
<td>0.315 to 0.63 m/s²</td>
<td>a little uncomfortable</td>
</tr>
<tr>
<td>0.5 to 1 m/s²</td>
<td>fairly uncomfortable</td>
</tr>
<tr>
<td>0.8 to 1.6 m/s²</td>
<td>uncomfortable</td>
</tr>
<tr>
<td>1.25 to 2.5 m/s²</td>
<td>very uncomfortable</td>
</tr>
<tr>
<td>Greater than 2 m/s²</td>
<td>extremely uncomfortable</td>
</tr>
</tbody>
</table>

According to Griffin [34], reductions of 5 to 10% in weighted vibration magnitude are usually undetectable in subjective ratings. According to Mansfield et al. [48], a difference of approximately 13% is needed to feel a difference in ride comfort.

4.2 Experimental setup

In this section it will become clear which experiments are performed, how the in-wheel motors are exactly emulated and which quantities are measured during the experiments.

The experiments are conducted with three configurations of the VW Lupo 3L: the baseline configuration, a configuration with an increased unsprung mass at the front and a configuration with an increased unsprung mass at the rear of the vehicle. From this point on these configurations are indicated by 'baseline', 'iw-front' and 'iw-rear', respectively. To emulate the in-wheel motors at the front or rear, the vehicle is modified as shown in Fig. 4.1 (a) and (b), respectively. As displayed, the modification consists of a stationary part directly attached to the suspension system and a rotating part, the fabricated disc, directly attached to the wheel. Both masses weigh 7.5 kg, as determined in the previous chapter.

![Figure 4.1](image_url): Vehicle modifications at the right front and right rear wheel.

The test vehicle is equipped with five acceleration sensors and with one dual-axis gyro sensor, all connected to a dSpace data acquisition system. The locations of these sensors are as illustrated in Fig.4.2.
4.2. Experimental setup

These sensors measure the following signals during the experiments:

**Sensor 1**: longitudinal, lateral and vertical acceleration of the chassis

*(center-x, center-y, center-z)*

**Sensors 2 and 3**: vertical acceleration of the front and rear wheel

*(fusee-front/fusee-rear)*

**Sensors 4 and 5**: vertical acceleration of the top of the suspension spring at the front and rear *(top-front/top-rear)*

**Sensor 6**: roll and pitch velocity of the chassis

*(roll, pitch)*

The words in italic style are the names of each sensor as used throughout this thesis. More information about the sensors and the dSpace data acquisition system can be found in Appendix C.

The measurements are performed on four different road surface types with ascending severity: smooth asphalt, highway, cobblestones and Belgian blocks. These four road types are all found as public road in the neighborhood of Eindhoven. Smooth asphalt is found on the Eisenhowerlaan, the highway road is taken to be the A270, the cobblestones are found on the Broekdijk and the Belgian blocks are found on the Soeterbeekseweg. The exact locations of these roads are shown in Fig. 4.3. An impression of these roads is provided in Fig. 4.4. The vehicle forward velocity on smooth asphalt, cobblestones and Belgian blocks is around 50 km/h, while the velocity on the highway is around 120 km/h. Due to this difference in vehicle velocity and due to differences in length of the roads, the total measurement distance of each road varies, as presented in the measurement overview in Table 4.2.

<table>
<thead>
<tr>
<th>road type</th>
<th>road name</th>
<th>velocity</th>
<th>distance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth asphalt</td>
<td>Eisenhowerlaan</td>
<td>50 km/h</td>
<td>350 m</td>
</tr>
<tr>
<td>Highway</td>
<td>A270</td>
<td>120 km/h</td>
<td>2000 m</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>Broekdijk</td>
<td>50 km/h</td>
<td>1200 m</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>Soeterbeekseweg</td>
<td>50 km/h</td>
<td>1550 m</td>
</tr>
</tbody>
</table>

Each measurement is performed twice. This allows to analyse the variation between measurements and provides confidence with respect to repeatability. In order to repeat the same measurement on open road and in order to compare results obtained with a vehicle with and without increased unsprung mass, it is important that the same road segments are traveled with approximately the same vehicle velocity. To measure the location and velocity a Parallax GPS Receiver module is used. More information about this module can be found in Appendix C.2.
Chapter 4. On-road experiments with an ICE vehicle

4.3 Experimental results

This section discusses the results obtained by the on-road experiments. However, before the results can be assessed, it is important to note that part of the measured accelerations consists of engine vibrations. As the results obtained on different road types are compared to each other and since the vehicle velocity is different on some occasions, the influence of the engine vibrations on the results will differ. Moreover, the results obtained with the experiments are employed to validate the model used for the simulations. Since the engine is not modeled, these engine vibrations are also unwanted. Because of these reasons it is preferable to examine the results up to a frequency such that the engine vibrations do not play an important role in the results. To determine this frequency, the excitation frequencies of the Lupo 3L engine are theoretically and experimentally determined.

Theoretically, in an ICE there are disturbances due to the explosion of the fuel in the cylinders and disturbances due to the inertia force and torque caused by the pistons, connecting rods and crankshaft [65]. The frequencies of these disturbances are found at the so called fundamental frequency and its harmonics \((x2, x3, x4, \text{etc})\). The fundamental frequency depends on the engine speed, the number of cylinders in the engine and the number of strokes, according to:

\[
f = \frac{N}{2 \cdot 60} \cdot n_{\text{cyl}} \cdot n \quad \text{with} \quad n = 1, 2, 3, 4, \ldots
\]

with \(N\) the rotational speed of the engine [rpm] and \(n_{\text{cyl}}\) the number of cylinders of the engine. Since the 3-cylinder engine of the Lupo 3L has a stationary speed of 830 rpm, the disturbance frequencies are theoretically found at 20.75, 41.5, 62.25, 83, 103.75 Hz, etc.

To investigate the excitation frequencies of the engine experimentally, the accelerations of the chassis at the top of the suspension-spring at the front and rear of the vehicle (top-front, top-rear) and at the left side of the driver’s seat (center-z) are measured during stand still with the engine running idle. The PSD of these accelerations are shown in Fig.4.5. The disturbance frequencies of the motor can be distinguished at approximately 14, 21, 28, 42, 62, 83, 104, 125 and 145 Hz. Although some of these frequencies coincide with the theoretically fundamental frequencies, the disturbance frequencies are also found at the frequency that coincide with the rotational velocity of the engine at stationary speed, approximately 14 Hz, and its harmonics. The certain small amount of power at all other frequencies visible in the figure is caused by noise present in the system.

Taking the above results into account, the lowest disturbance frequency during the experiments can now be determined, depending on the vehicle forward velocity. At 50 km/h and at 120 km/h the engine of the vehicle rotates with a speed of around 1300 and 2500 rpm, respectively, resulting in a lowest disturbance
4.3. Experimental results

The disturbance frequencies of the Lupo’s diesel engine are shown in Figure 4.5. The frequencies range from 10−8 Hz to 103 Hz. The magnitude of the acceleration in the z-direction is shown for top-front, center-z, and top-rear positions.

From this figure, it can be concluded that the difference between two repeated measurements are below 5% for experiments taken on the highway, on cobblestones and on Belgian blocks. However, on smooth asphalt the difference can be as high as 30%, which can be explained by the fact that the measured accelerations are very low and are therefore dominated by the noise in the system. The small differences on the other roads can be explained by the fact that the measurements are done in dynamic experiments in which the
forward velocity and traveled route are never exactly the same. These differences are also found with all other sensors. Taking these results into account, it can be concluded that the results on Belgian blocks, cobblestones and highway can be trusted. The results found on the smooth asphalt differ substantially and as such, these results have to be handled with some care. Because of this reason, all results found on the smooth pavement will be presented in italic style.

**Center sensor**

The absolute values of the vertical acceleration obtained with the center sensor and the associated overall ride comfort index for all three configurations are listed in Table 4.3. The relative differences of the iw-front and iw-rear with the baseline vehicle are also given. The absolute values of the overall ride comfort index are also illustrated in Fig. 4.7.

<table>
<thead>
<tr>
<th>Road type</th>
<th>baseline (a_{overall}) [m/s²]</th>
<th>iw-front (a_{overall}) [m/s²]</th>
<th>iw-rear (a_{overall}) [m/s²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth asphalt</td>
<td>1.142</td>
<td>1.433</td>
<td>1.210</td>
</tr>
<tr>
<td>Highway</td>
<td>0.402</td>
<td>0.437</td>
<td>0.406</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>0.611</td>
<td>0.702</td>
<td>0.659</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>1.142</td>
<td>1.433</td>
<td>1.210</td>
</tr>
</tbody>
</table>

Fig. 4.7 clearly shows the difference in severity of the various road inputs. A low overall ride comfort index is found for the smooth asphalt and the highest index is found for the Belgian blocks. From Table 4.3 and Fig. 4.7 it can be concluded that increasing the unsprung mass results in an increase in overall ride comfort index and thus in a deterioration of the ride comfort. For the iw-rear, this increase stays below the 10% for all roads, while for the iw-front it increases even up to 25% on Belgian blocks. The relative difference increases as the severity of the road input increases.

Based on the assumption that an increase in overall ride comfort index between 5% and 10% is necessary to feel a difference in comfort, it can be concluded that front in-wheel motors will definitely be perceptible for the vehicle occupations, but this is questionable for rear in-wheel motors. Note however that, according to the scale of vibration discomfort as stated in Table 4.1, the overall feeling of the ride only changes
on Belgian blocks. The feeling in the baseline vehicle on Belgian blocks is already considered to be uncomfortable, but with an increase in unsprung mass at the front the feeling can be considered as very uncomfortable.

**Top-front / top-rear sensor**

The vertical accelerations of the chassis measured with the top-front and top-rear sensor are investigated next. The absolute values and the relative difference of the in-wheel configurations with the baseline vehicle obtained with these sensors are shown in Table 4.4. Note that the top-front sensor is used only to compare the iw-front with the baseline and that the top-rear sensor is used only to compare the iw-rear with the baseline vehicle.

Table 4.4: Absolute values and relative differences of the top-front and top-rear sensor for the iw-front and iw-rear compared to the baseline.

<table>
<thead>
<tr>
<th>sensor</th>
<th>road type</th>
<th>baseline (RMS) [m/s²]</th>
<th>iw-front (RMS) [m/s²]</th>
<th>relative difference</th>
<th>iw-rear (RMS) [m/s²]</th>
<th>relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>top-front</td>
<td>Smooth asphalt</td>
<td>0.168</td>
<td>0.199</td>
<td>+ 18.6 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>0.698</td>
<td>0.817</td>
<td>+ 17.0 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>1.030</td>
<td>1.210</td>
<td>+ 17.4 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>2.002</td>
<td>2.418</td>
<td>+ 20.8 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>top-rear</td>
<td>Smooth asphalt</td>
<td>0.159</td>
<td>-</td>
<td>-</td>
<td>0.201</td>
<td>+ 26.4 %</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>0.716</td>
<td>-</td>
<td>-</td>
<td>0.650</td>
<td>- 9.3 %</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>1.077</td>
<td>-</td>
<td>-</td>
<td>1.256</td>
<td>+ 16.6 %</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>1.693</td>
<td>-</td>
<td>-</td>
<td>1.820</td>
<td>+ 7.5 %</td>
</tr>
</tbody>
</table>

From Table 4.4 it can be concluded that the added mass at the front or rear results in an increase in vertical acceleration at the front or rear of the vehicle, respectively, for all road types. Only the results obtained with the iw-rear on the highway constitutes an exception. All vertical accelerations of the top-front and top-rear are higher than the vertical acceleration of the center of the chassis, independent of the vehicle configuration. Furthermore, in-wheel motors at the front have a higher influence on the acceleration of the chassis at the front, than in-wheel motors at the rear do on the accelerations of the chassis at the rear. This complies with the results found with the center sensor. Because of this it can be concluded that, although the relative differences do not exactly match with the relative differences as found for the center sensor, the trend of the results are in line with the trends found with the center sensor.

**Fusee-front / fusee-rear sensor**

To investigate the effect of in-wheel motors on the vertical accelerations of the wheels, the influence of the increase in unsprung mass on the eigenfrequencies of the wheels is determined first. The PSD of the front wheel obtained with the baseline and iw-front and the PSD of the rear wheel obtained with the baseline and the iw-rear are shown in Fig.4.8 (a) and (b), respectively.

From these figures it can be concluded that the eigenfrequencies of the front and rear wheel are both around 18.5 Hz. Due to the increase in unsprung mass, these frequencies change to around 13.5 Hz. This information will be used to determine the unsprung masses, as explained in the next chapter.

The absolute values of the vertical accelerations of the front and rear wheel and the relative difference of the in-wheel configurations with the baseline can be found in Table 4.5. Note that the fusee-front sensor is used only to compare the iw-front with the baseline and that the fusee-rear sensor is used only to compare the iw-rear with the baseline vehicle. These results clearly show that an increase in unsprung mass results in an increase of the accelerations of the wheels. The relative difference depends on the severity of the road and the configurations, as is also the case for the vertical accelerations of the chassis. However, in this case it is found that the increase in accelerations is much higher for the iw-rear than for the iw-front. A possible explanation for this is the fact that the suspension at the rear is less stiff and slightly less damped than the suspension at the front.
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Figure 4.8: Shift in eigenfrequency of the front and rear wheel due to the increase in unsprung mass.

Table 4.5: Absolute values and relative differences of the fusee-front and fusee-rear sensor for the iw-front and iw-rear compared to the baseline.

<table>
<thead>
<tr>
<th>sensor</th>
<th>road type</th>
<th>baseline (RMS) [m/s²]</th>
<th>iw-front (RMS) [m/s²]</th>
<th>relative difference</th>
<th>iw-rear (RMS) [m/s²]</th>
<th>relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>top-front</td>
<td>Smooth asphalt</td>
<td>0.360</td>
<td>0.442</td>
<td>+ 22.9 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>4.714</td>
<td>4.853</td>
<td>+ 3.0 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>6.849</td>
<td>7.345</td>
<td>+ 7.2 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>16.042</td>
<td>19.390</td>
<td>+ 20.9 %</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>top-rear</td>
<td>Smooth asphalt</td>
<td>0.684</td>
<td>-</td>
<td>-</td>
<td>1.038</td>
<td>+ 52.0 %</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>6.854</td>
<td>-</td>
<td>-</td>
<td>7.370</td>
<td>+ 7.5 %</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>6.925</td>
<td>-</td>
<td>-</td>
<td>9.881</td>
<td>+ 42.7 %</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>16.493</td>
<td>-</td>
<td>-</td>
<td>25.861</td>
<td>+ 56.8 %</td>
</tr>
</tbody>
</table>
4.3. Experimental results

Roll and pitch sensor

The absolute values of the roll and pitch velocity of the chassis and the relative differences of the in-wheel configurations with the baseline are given in Table 4.6. The absolute values of the roll and pitch velocity are also illustrated in Fig. 4.9 (a) and (b), respectively.

Table 4.6: Relative difference of the roll velocity of the chassis for the iw-front and iw-rear compared to the baseline.

<table>
<thead>
<tr>
<th>sensor</th>
<th>road type</th>
<th>baseline velocity (RMS) [m/s]</th>
<th>iw-front velocity (RMS) [m/s]</th>
<th>relative difference</th>
<th>iw-rear velocity (RMS) [m/s]</th>
<th>relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>roll</td>
<td>Smooth asphalt</td>
<td>0.207</td>
<td>0.235</td>
<td>+13.8 %</td>
<td>0.242</td>
<td>+17.1 %</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>0.636</td>
<td>0.631</td>
<td>-0.7 %</td>
<td>0.652</td>
<td>+2.5 %</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>1.483</td>
<td>1.629</td>
<td>+9.8 %</td>
<td>1.468</td>
<td>-1.1 %</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>2.182</td>
<td>2.415</td>
<td>+10.7 %</td>
<td>2.291</td>
<td>+5 %</td>
</tr>
<tr>
<td>pitch</td>
<td>Smooth asphalt</td>
<td>0.301</td>
<td>0.323</td>
<td>+7.3 %</td>
<td>0.314</td>
<td>+4.4 %</td>
</tr>
<tr>
<td></td>
<td>Highway</td>
<td>0.627</td>
<td>0.621</td>
<td>-1.0 %</td>
<td>0.646</td>
<td>+3.0 %</td>
</tr>
<tr>
<td></td>
<td>Cobblestones</td>
<td>1.759</td>
<td>1.787</td>
<td>+1.6 %</td>
<td>1.911</td>
<td>+8.6 %</td>
</tr>
<tr>
<td></td>
<td>Belgian blocks</td>
<td>2.512</td>
<td>2.581</td>
<td>+2.7 %</td>
<td>2.596</td>
<td>+3.3 %</td>
</tr>
</tbody>
</table>

From the relative differences as given in Table 4.6, it can be concluded that overall, with some exceptions, the increase in unsprung mass also results in an increase in roll and pitch velocity. The increase in roll velocity is higher for the iw-front, while the increase in pitch velocity is higher for the iw-rear. The higher increase in roll velocity found for the iw-front is probably due to the fact that the anti-roll bar is stiffer at the front than at the rear, which can result in more interaction between the left and rear wheels. The higher increase in pitch velocity found for the iw-rear is probably due to the fact that the center of gravity of the vehicle is located more at the front. The in-wheel motors at the rear can therefore create a higher momentum around the center of gravity.

Influence in the frequency domain

Next, the differences between the three vehicle configurations in the frequency domain are investigated. Fig. 4.10 shows the PSD of the vertical accelerations of the chassis measured by the center sensor for all three vehicle configurations on Belgian cobblestones. In this figure it is visible that in the mid frequency range, between 1 and 10 Hz, the accelerations are slightly increased. In the area between 10-20 Hz a clear increase in acceleration is found for both the iw-front and the iw-rear. This increase is the highest for the iw-front. At frequencies above the eigenfrequency of the wheels, the accelerations are slightly decreased. It has been found that these conclusions also hold for the results found on the other road types. Moreover, these results comply with the theoretically effect of a change in unsprung-to-sprung mass as described in
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Figure 4.10: PSD of the vertical acceleration of the chassis on Belgian blocks for all vehicle configurations.

4.4 Subjective rating

In this section the subjective feelings of both the driver and the passenger are discussed briefly.

The increase in unsprung mass, either at the front or at the rear, was not really noticeable by both the driver and the passenger during driving on the smooth asphalt and the highway. However, when driving on cobblestones and especially on Belgian blocks the presence of the masses was definitely perceptible. On these roads the increase in unsprung mass at the front was experienced to be the worst. Moreover, the increase in unsprung mass at the front resulted in vibrating of the steering wheel, which was considered to be very annoying. However, taking thresholds with the iw-rear resulted in an excessive pitch of the vehicle when the threshold was hit by the rear wheels. This was experienced to be very uncomfortable. Though, overall, the subjective feelings comply with the measurement results as described in this chapter.

During normal driving the cornering behavior was not affected perceptibly, but this was not the case during more excessive cornering maneuvers like a J-turn. During these maneuvers more oversteer with the iw-rear vehicle and more understeer with the iw-front vehicle was clearly noticeable.

4.5 Discussion

The experimental results clearly show that in-wheel motors do decrease the ride comfort of the vehicle. By placing the motors in the front wheels, the ride comfort decreases up to 25%, depending on the severity of the road input, while by placing the motors in the rear wheels this decrease is less than 10%. Take in mind that, due to the limited time available for the time consuming experiments, the measurements are only repeated once. In order to have a high accuracy of the experimental results, it is advisable to increase the number of measurements. Nevertheless, as most of the repeated measurements do not differ more than 5 %, the experimental results are considered to be reliable and can be used to get a good impression of the effect of in-wheel motors on the ride comfort.

To make sure that the vibrations of the engine do not play an important role in the results, all measurements are compared with each other up to a frequency of 20 Hz. However, as several figures show, the increase in unsprung mass also has an influence on the accelerations at higher frequencies. On the other hand, it can
also be seen that the energy of these accelerations are lower than found for the frequencies below 20 Hz and due to the comfort weighting functions applied to these accelerations, these frequencies are attenuated even more. Because of this, the influence of in-wheel motors at higher frequencies is estimated to be very small.

Since the Lupo is not equipped with cruise control, it is very hard for a normal driver to keep the vehicle at a constant speed. Although the vehicle forward velocity is kept as steady as possible during the experiments, the vehicle velocity of each measurement can differ up to 5%. It is assumed that these differences do not have a significant effect on the outcome of the research.

In this chapter the effect of in-wheel motors on the ride comfort of an ICE vehicle is investigated based on on-road experiments. Next up is to investigate the effect of in-wheel motors on the ride comfort and road holding using a BEV. Since a BEV is not available for on-road experiments, this investigation is only performed by simulations. In order to trust these simulation results, the simulation model has to be validated first. This model validation is therefore described in the next chapter.
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Chapter 5

Model validation

The previous chapter shows the effect of in-wheel motors on the ride comfort based on on-road experiments. As these experiments are performed with an ICE vehicle, the effect of in-wheel motors on the ride comfort of a BEV is still unknown. Furthermore, the effect of in-wheel motors on the road holding capability are also still unknown. In order to investigate these effects, simulations are performed with the software program Matlab/Simulink using the multi-body toolbox SimMechanics. To have confidence in the results obtained with this program, the model used for the simulations has to be an accurate representation of the real vehicle. Hence, the model has to be validated. This is dealt with in this chapter.

Information about the model used for the simulations is given in Section 5.1. The validation of the model, done by the comparison of the model behavior to the measurements obtained with the VW Lupo 3L, will be described in Section 5.2.

5.1 Model

The vehicle model used for the simulations is a 10 degree of freedom full-vehicle model as shown in Fig.5.1. It consists of a vehicle body representing the sprung mass and four corner modules. Each corner module, consisting of a body representing the unsprung mass, is connected to the sprung body by a vertical spring-damper system. Each of the corner bodies is connected to the MF-Swift 6.1.0 tyre model, as developed by TNO Automotive. The tyre forces and moments in this model are described by the Magic Formula. This tyre model is accurate in the frequency range where the bending modes of the tyre belt can be neglected, which is at least up to the evaluation frequency of 20 Hz. The tyre parameters of the Lupo tyre as given in Chapter 3 and in Appendix A.3 are used as input for this model.

The vehicle body has 6 degrees of freedom (longitudinal, lateral, vertical, roll, pitch, yaw) and each corner module has one vertical degree of freedom. The distance of the center of gravity to the front and rear of the vehicle, \( a \) and \( b \) respectively, and the track width \( w \) are listed in Table 3.1 in Chapter 3. The damping constant \( d_s \) and the spring stiffness \( k_s \) are the damping and spring characteristics as determined in Chapter 3, implemented in the model by look-up tables. The unsprung masses, \( m_{a1} \) to \( m_{a4} \), which also defines the mass of the sprung body \( m_s \), will be determined in the next section.

Other important parameters that still need to be determined are the height of the center of gravity above the ground \( (h) \) and the vehicle moment of inertia around the x-, y- and z-axis. Since they can not be measured easily, these parameters are estimated based on the rules of thumb according to the National Highway Traffic Safety Administration database of cars, vans, SUV and pickup trucks [37]. The estimated parameters are given in Table 5.1. In addition, the model is equipped with an anti-roll bar at the front \( (c_{φf}) \) and at the rear \( (c_{φr}) \). The stiffness of these anti-roll bars are determined experimentally and are 1800 and 800 Nm/rad at the front and rear, respectively. A steady state cornering simulation shows that
a roll angle of about 5 degrees is reached at a lateral acceleration of 8 m/s², which is not uncommon for a real vehicle [19]. The anti-roll bars are implemented by applying an amount of force to the wheels that depends on the relative suspension travel between the left and rear side of the vehicle. The weight of the unsprung masses and the validation of the model is described in the next section.

Table 5.1: Additional estimated parameters of the VW Lupo 3L.

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height of center of gravity above ground</td>
<td>( h )</td>
<td>0.55</td>
<td>m</td>
</tr>
<tr>
<td>Moment of inertia w.r.t. x-axis</td>
<td>( I_{xx} )</td>
<td>290</td>
<td>kgm²</td>
</tr>
<tr>
<td>Moment of inertia w.r.t. y-axis</td>
<td>( I_{yy} )</td>
<td>1120</td>
<td>kgm²</td>
</tr>
<tr>
<td>Moment of inertia w.r.t. z-axis</td>
<td>( I_{zz} )</td>
<td>1250</td>
<td>kgm²</td>
</tr>
<tr>
<td>Stiffness anti-roll bar front</td>
<td>( c_{\phi f} )</td>
<td>1800</td>
<td>Nm/rad</td>
</tr>
<tr>
<td>Stiffness anti-roll bar rear</td>
<td>( c_{\phi r} )</td>
<td>800</td>
<td>Nm/rad</td>
</tr>
</tbody>
</table>

The road profile height used as input in the simulations is a road profile measured by DAF, only scaled such that overall ride comfort index is comparable to those found with the on-road experiments. To resemble the smooth asphalt, highway, cobblestones and Belgian blocks, the road profile is scaled with a factor of 0.005, 0.20, 0.35 and 0.8, respectively. The displacement PSD of these various road profiles are found in Fig. D.1 in Appendix D.1.

5.2 Validation

In this section the simulation results are compared to the experimental results in order to validate the model. First, the results obtained by simulations using the baseline vehicle are compared to the associated results obtained by the experiments. This comparison is only performed in the frequency domain. Second, the relative differences of the in-wheel configurations found by simulations are compared to those found by the experiments.

For the first comparison only the results obtained on Belgian blocks are used, since on this road the effect of an increase in unsprung mass on the ride comfort index is the largest. For the second comparison, all road types are taken into account.

The vehicle velocity for the simulations is taken to be the same as the average vehicle velocity found during the experiments. However, to take the variation of the velocity as found in the experiments into account, these velocities are varied between the average velocity ± 5 km/h. The vehicle velocities are therefore chosen as stated in Table 5.2.
Table 5.2: Vehicle forward velocity during the experiments on all roads for all configurations.

<table>
<thead>
<tr>
<th>Road Type</th>
<th>Baseline velocities (km/h)</th>
<th>iW-front velocities (km/h)</th>
<th>iW-rear velocities (km/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth asphalt</td>
<td>43 - 53</td>
<td>45 - 55</td>
<td>44 - 54</td>
</tr>
<tr>
<td>Highway</td>
<td>114 - 124</td>
<td>114 - 124</td>
<td>115 - 125</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>43 - 52</td>
<td>45 - 55</td>
<td>44 - 54</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>41 - 51</td>
<td>42 - 52</td>
<td>44 - 54</td>
</tr>
</tbody>
</table>

5.2.1 Baseline comparison

First, the experimental results of the baseline are used to estimate the weights of the unsprung masses of the vehicle. In the previous chapter it has been determined that the eigenfrequencies of the front and rear wheel are around 18.5 Hz. Using this information an approximation of the unsprung masses can be found by using the equation:

\[
\omega_{ma} = \sqrt{\frac{k_s + k_t}{m_a}}
\]  

with \(\omega_{ma}\) the natural eigenfrequency of the unsprung mass [rad/s]. Taking this into account the weight of the total unsprung mass at the front and rear of the vehicle are estimated to be 35 kg each. The power spectral density of the experimental results and the simulation results, given in Fig.5.2 (a) and (b) for the front and rear wheel respectively, show the correctness of these estimations. Overall, there is a good similarity between the experimental results and the simulations results. However, at low frequencies (<1 Hz) and at higher frequencies (>40 Hz) the figures show small differences between the experimental and simulation results. This can be explained by the fact that the road input as used for the simulations may be different at these frequencies. For example, the road input used for the simulations shows a very large decrease in road amplitude at around 40 Hz, which is clearly reflected in the simulation results. Note that the road profile height used to obtain these simulation results are scaled such that the simulation results nicely coincide with the experimental results. This is purely done for clarification. The scaling factors used to resemble the various road types actually result in PSD plots that lie somewhat higher compared to the experimental results.

Figure 5.2: PSD of the acceleration of the front and rear wheels obtained by experiments and simulations.

Next, the vertical acceleration of the chassis found with the experiments and the simulations are compared first by analyzing the PSD of the accelerations, given in Fig.5.3.

As can be seen, both the experiments and the simulations show several peaks with almost the same amplitude and location up to a frequency of around 25 Hz. The peaks around 1.5 Hz and 18.5 Hz belong to
the natural frequency of the sprung and unsprung body, respectively. The other peaks and the fact that for certain frequencies the power goes to a certain minimum, the so called null points, are caused by an effect called wheelbase filtering. This effect is caused by the fact that the rear wheels encounter the same obstacles as the front wheels, only with a certain time delay. The location of the ’null points’ depends on this delay, which in turn is dependent on the forward velocity and wheelbase. Theoretically the null points are found at:

\[ f_{\text{null}} = \frac{(2N - 1)V}{2l} \quad \text{with} \quad N = 1, 2, 3, \ldots \]  

(5.2)

With \( V \) the forward velocity [m/s] and \( l \) the wheelbase [m]. With a forward vehicle velocity of 50 km/h, this gives vertical displacement minima at 2.8, 8.4, 14.1, 19.7, 25.3, 30.9, 36.6, 42.2 and 47.8 Hz. Both the simulation results and the experimental results do not exactly show their null points at these locations, but that is due to the variations of the forward velocity.

The differences found at frequencies higher than 25 Hz are partly due to the differences in road inputs, as already explained above, and partly due to the fact that the first bending mode of the vehicle, which normally lies around 30 Hz, has not been taken into account in the model. The differences found in the amplitude of the power at frequencies below 25 Hz are most likely due to the differences in vehicle parameters, for example the inertia, due to the differences in the real and modeled suspension system and due to the fact that the drive train components, which can also cause vibrations in the 10 to 20 Hz area, are not modeled.

Next, the chassis accelerations in the longitudinal and lateral direction obtained by experiments and simulations are compared to each other. The PSD of both are found in Fig. 5.4 and Fig. 5.5, respectively.

As can be seen, the similarity of the simulation results with the experimental results is lot less than found for the vertical accelerations. These differences can, besides the reasons as mentioned above, possibly be explained by the fact that during the experiments the vehicle velocity is not totally constant, small accelerations and decelerations exist, and lateral movements are made during corners and traffic avoidance maneuvers. However, only the accelerations around a frequency between 1 and 2 Hz, are important to assess the ride comfort of the vehicle. Around these frequencies the simulations do not differ that much from the experiments.

Finally, the roll and pitch velocity obtained with the simulations are compared to the experiments. The PSD of both velocities can be found in Fig. 5.6 and Fig. 5.7, respectively.

From these figures it can be concluded that there is a good agreement in roll and pitch velocity between the simulations and the experiments up to a frequency of around 25 Hz. However, the eigenfrequency of the roll and especially the eigenfrequency of the pitch do not exactly match those found in the experiments.
5.2. Validation

Figure 5.4: PSD of the longitudinal acceleration of the chassis obtained by experiments and simulation.

Figure 5.5: PSD of the lateral acceleration of the chassis obtained by experiments and simulation.

Figure 5.6: PSD of the roll velocity obtained by experiments and simulation.

Figure 5.7: PSD of the pitch velocity obtained by experiments and simulation.
Chapter 5. Model validation

Besides the differences in vehicle parameters like the inertia, this difference is most likely due to the fact that the engine and drive train components and their mounts are not taken into account into the model.

Taking all above results into account, it can be concluded that, up to a frequency of 25 Hz, the simulation results show a high degree of similarity with the experimental results. Therefore it is concluded that the baseline vehicle can be considered to be validated. Next up is to investigate if the effect of in-wheel motors is the same found for the simulations as found for the experiments.

5.2.2 Configuration comparison

In order to determine whether the effect of increasing the unsprung mass is the same for the simulations as found in the experiments, the absolute values of the overall ride comfort index, vertical accelerations of the wheels and the roll and pitch velocity found by the simulations for all configurations are compared to those found in the experiments. Moreover, the relative differences of the iw-front and iw-rear configurations with the baseline are also compared on all four points.

First it is inspected if the increase in mass is correctly implemented in the simulation model. It is found that this is indeed the case, since the eigenfrequencies drops to approximately 13.5 Hz. Proof of this is found in Fig. D.2 in Appendix D.2. Furthermore, it is inspected if the change in power spectral density of the vertical chassis acceleration due to the increase in unsprung mass complies with those found in the experiments, as described in Section 4.3. It is concluded that this is indeed the case: almost no change at frequencies below 1 Hz, a small increase around 6 Hz, a high increase in the 10-20 Hz are and a small decrease at higher frequencies for both the iw-front and iw-rear. Proof of this can be found in Fig. D.3 in Appendix D.2.

The absolute values of the overall ride comfort index found by experiments and simulations for all four roads and all configurations are shown in Fig. 5.8 (a) and (b), respectively. The relative differences between the configurations found by both are given in Table 5.3.

![Figure 5.8](image)

The figures clearly show that the absolute value of the overall ride comfort index found with the simulations are almost the same as found for the experiments for all four road types. The trend in increase in overall ride comfort index due to the increase in severity of the roads is almost the same. Regarding the relative differences, it can be concluded that the difference found on the highway and on the cobblestones by simulations seem to be quite similar to those found in the experiments, but this is not the case for the results found on Belgian blocks. Especially the increase of the ride comfort index found for the iw-front differs strongly. Note that this is also the case for the results found on smooth pavement, but since these results found in the experiments can not be fully trusted, they will not be further taken into account. The higher differences found on Belgian blocks are maybe because the vibrations of the drive train and the
5.2. Validation

Table 5.3: Relative differences in overall ride comfort index for all roads and for all vehicle configurations compared to the baseline found by experiments and by simulation.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Experiments</th>
<th>Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>iw-front</td>
<td>iw-rear</td>
</tr>
<tr>
<td>Smooth asphalt</td>
<td>+ 13.4 %</td>
<td>+ 31.3 %</td>
</tr>
<tr>
<td>Highway</td>
<td>+ 8.8 %</td>
<td>+ 1.1 %</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>+ 14.7 %</td>
<td>+ 7.8 %</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>+ 25.5 %</td>
<td>+ 5.9 %</td>
</tr>
</tbody>
</table>

bending of the chassis play a bigger role in the accelerations of the chassis on a severe road like Belgian blocks, than on a less severe road like the highway. Since this is not modeled, this is not reflected in the simulation results. Nevertheless, the overall conclusion that the increase in unsprung weight at the front results in a higher overall ride comfort index than the increase at the rear is also substantiated in the simulation results.

The absolute values of the vertical accelerations of the wheels found in the experiments and the simulations for all four roads are shown in Fig.5.9 (a) and (b), respectively. The relative differences between the configurations are given in Table 5.4. For this comparison the increase of the acceleration of the front wheel found for the iw-front is only compared to the acceleration of the front wheel found for the baseline and increase of the acceleration of the rear wheel found for the iw-rear is only compared to the acceleration of the rear wheel of the baseline.

Figure 5.9: Absolute values of the vertical accelerations of the front and rear wheel on all roads for all vehicle configurations found by experiments and by simulations.

Table 5.4: Relative differences of the vertical accelerations of the front and rear wheel for all roads and for all vehicle configurations compared to the baseline found by experiments and by simulations.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Experiments</th>
<th>Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>iw-front</td>
<td>iw-rear</td>
</tr>
<tr>
<td>Smooth asphalt</td>
<td>+ 22.9 %</td>
<td>+ 52.0 %</td>
</tr>
<tr>
<td>Highway</td>
<td>+ 3.0 %</td>
<td>+ 7.5 %</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>+ 7.2 %</td>
<td>+ 42.7 %</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>+ 20.0 %</td>
<td>+ 56.8 %</td>
</tr>
</tbody>
</table>

Fig.5.9 (a) and (b) clearly show that there is a difference in magnitude between the accelerations of the wheels found by experiments and simulations. The relative differences also clearly differ, particularly at the rear. The difference in magnitude is possible due to the difference in suspension system between the model
and the real vehicle and due to the absence of rubber bushes in the model. However, the trend of increase in vertical accelerations due to the increase in severity of the road input is also found for the simulations. Furthermore, it can be concluded that in both the experiments and the simulations the acceleration of the rear wheels are higher than the accelerations of the front wheels, which is possible due to the fact that the suspension at the rear is less stiff and slightly less damped than the suspension at the front.

The absolute values of the roll velocity found in the experiments and the simulations for all four roads are shown in Fig. 5.10 (a) and (b), respectively. The relative differences between the configurations are given in Table 5.5.

![Figure 5.10](image)

Table 5.5: Relative differences of the roll velocity for all roads and for all vehicle configurations compared to the baseline found by experiments and by simulation.

<table>
<thead>
<tr>
<th>configuration</th>
<th>Experiments</th>
<th>Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>iw-front</td>
<td>iw-rear</td>
</tr>
<tr>
<td>Smooth asphalt</td>
<td>+ 13.8 %</td>
<td>+ 17.1 %</td>
</tr>
<tr>
<td>Highway</td>
<td>- 0.7 %</td>
<td>+ 2.5 %</td>
</tr>
<tr>
<td>Cobblestones</td>
<td>+ 9.8 %</td>
<td>- 1.1 %</td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>+ 10.7 %</td>
<td>+ 5 %</td>
</tr>
</tbody>
</table>

As presented in the figure, the absolute values of the roll velocity found for the simulations are slightly higher than found for the experiments for all roads, except for the cobblestones. A possible explanation for this is the fact that the real road used for the experiments is somewhat oblique at the sides, causing extra roll of the vehicle. This is not the case for the simulated road. The relative differences in configurations do not match exactly with the experiments. However, the overall conclusion that the roll velocity increases more due to the placement of the in-wheel motors at the front is also confirmed by the simulations. As noted before, this is most likely due to the fact that the anti-roll bar at the front is stiffer than at the rear.

The absolute values of the pitch velocity found in the experiments and the simulations for all four roads are shown in Fig. 5.11 (a) and (b), respectively. The relative differences between the configurations are given in Table 5.6.

As displayed, the difference in absolute values of the pitch velocity for the different roads is almost the same as found for the experiments, except again for the results found on the cobblestones. A possible explanation for this could be that the real road has a somewhat higher amplitude at low frequencies. The relative differences do not match exactly, however, the overall conclusion that the pitch velocity increases more due to the placement of the in-wheel motors at the rear is also present in the simulations. As noted before, this is most likely due to the fact that the center of gravity of the vehicle is located more at the front.
5.3 Discussion

By comparing simulation results with experimental results, it has been found that there is a fairly good qualitative agreement between the results of the simulations and the experiments up to a frequency of 25 Hz. Although the model is not able to exactly predict what happens if the unsprung mass is increased, it is able to give a good indication of the trends found in the overall ride comfort index, accelerations of the wheels and the vehicle roll and pitch velocity. Therefore, the model can be considered to be valid.

Several suggestions can be made to improve the model and to get a better match with the experiments. For example, the semi-independent suspension system at the rear and the McPherson suspension at the front can be implemented in the model. Several vehicle parameters, like the height of the center of gravity and the inertia can be measured instead of estimated. In addition, bending of the chassis can be taken into account and the engine (and mount) and other drive train components can be implemented in the model. Note however, that this is not easy due to the complexity and unknown parameters of each component and the unknown bending of the chassis. Moreover, it can result in a much more complex model. This is not always wanted, since it increases the computing time and the accuracy of the model will not necessarily become better.

This chapter illustrates that the model represents the real vehicle with a fair degree of accuracy. Therefore, further investigations can be performed by the use of simulations. On that account, the effect of in-wheel motors on the ride comfort and safety of a BEV is investigated by the performance of simulations in the next chapter.
Chapter 6

Battery electric vehicle analysis

In the previous chapters the effect of in-wheel motors on the ride comfort of an ICE vehicle based on on-road experiments has been described and a simulation model has been developed and validated. This chapter explicates the ride comfort of the BEV equipped with in-wheel motors based on the validated model. The effect on the dynamic wheel load (road holding) and suspension travel is also investigated. This chapter also describes the possible improvements in ride comfort and road holding of the BEV, equipped without or with in-wheel motors, by modification of the suspension system.

The idea of these additional investigations is that due to the conversion of the ICE vehicle into a BEV, several parts of the vehicle are removed or replaced and several battery packs are added. This results in a change of mass, weight distribution and inertia of the vehicle. As such, the ride comfort and road holding of the vehicle change, the effect of in-wheel motors will be different from the ICE vehicle and the damping coefficient and spring stiffness of the suspension systems may need to be altered.

Information about the vehicle conversion and possible improvements by modification of the suspension system is given in Section 6.1. In Section 6.2 the effects of in-wheel motors on a BEV are described. Several options for improvements in the suspension system for a BEV with in-wheel motors are evaluated in Section 6.3. This chapter ends with a discussion in Section 6.4.

6.1 Vehicle conversion and suspension modification

The change in vehicle mass, weight distribution and inertia is addressed first. Hereafter, the possible improvements by modification of the suspension system is described.

6.1.1 Battery electric vehicle conversion

With the conversion of the Lupo 3L into a BEV, a number of vehicle parts, such as the engine, the exhaust-pipe and the fuel tank, are removed and parts such as an electric motor, inverter and batteries are added. Mainly due to the placement of heavy battery packs of 273 kg, containing a total energy of 27 kWh, this conversion results in an increase in vehicle mass and in a change of weight distribution. Table 6.1 shows the vehicle mass and weight distribution of both the baseline vehicle and the BEV including 2 persons, each weighing 80 kg. The total vehicle mass is increased by a total of 160 kg, from which 135 kg is placed at the rear. This increase in mass is especially due to the placement of the battery packs, which contain a total energy of 27.3 kWh and have a combined weight of 273 kg.

Due to the increase in mass and change of the weight distribution, the moment of inertia with respect to the x-, y- and z-axis changes. Using the Huygens-Steiner theorem it has been estimated that the moment of inertia $I_{xx}$ increases by 16%, while both $I_{yy}$ and $I_{zz}$ increase by 14%.

49
Table 6.1: Mass distribution of the baseline ICE and the to a BEV converted Lupo 3L.

<table>
<thead>
<tr>
<th></th>
<th>Baseline</th>
<th>BEV</th>
<th>Increase</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front axle</td>
<td>634 kg</td>
<td>659 kg</td>
<td>25 kg</td>
</tr>
<tr>
<td>Rear axle</td>
<td>396 kg</td>
<td>531 kg</td>
<td>135 kg</td>
</tr>
<tr>
<td>Total</td>
<td>1030</td>
<td>1190</td>
<td>160 kg</td>
</tr>
</tbody>
</table>

In order to investigate the effect of these changes on ride comfort, a simulation is performed with these new model parameters on Belgian blocks with a forward vehicle velocity of 50 km/h. The resulting overall ride comfort index is shown in Table 6.2. This table also compares the overall ride comfort index to the baseline ICE Lupo 3L model. As the results are obtained by simulations, the dynamic wheel load (dFz) and suspension travel (dz) can also be determined and are provided in the table as well.

Table 6.2: Change in overall ride comfort, dynamic wheel load and suspension travel due to the conversion of the ICE vehicle into a BEV.

<table>
<thead>
<tr>
<th></th>
<th>Baseline</th>
<th>BEV</th>
<th>Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_{overall} ) [m/s²]</td>
<td>1.132</td>
<td>0.977</td>
<td>-13.7 %</td>
</tr>
<tr>
<td>dFz (RMS) [N]</td>
<td>796</td>
<td>812</td>
<td>+2.1 %</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>10.6</td>
<td>11.2</td>
<td>+5.7 %</td>
</tr>
</tbody>
</table>

Based on the calculated differences between the BEV and baseline vehicle it can be concluded that a ride with the BEV is around 14% more comfortable. However, this increase in comfort is at the expense of a minor increase in dynamic wheel load and suspension travel.

### 6.1.2 Suspension modification

As the weight of the vehicle especially increases at the rear, the height of the rear will become undesirably low if the original Lupo springs are maintained. Therefore, different springs will be implemented in the real BEV, that are either stiffer or longer. In order to decide if they need to be stiffer or longer, the effect of the rear spring stiffness on the ride comfort index, dynamic wheel load and suspension travel is investigated. Furthermore, since the dynamic wheel load and suspension travel slightly increase in the conversion, the question arises whether the rear damper can be optimized to reduce these undesirable effects. Therefore this is also investigated.

For these investigations, simulations are performed on Belgian blocks with a vehicle forward velocity of 50 km/h. The Belgian blocks are chosen, since it is expected that the effect of the modifications will be most clearly visible on this severe road.

#### Spring modification

The effect of the spring stiffness is investigated by multiplying the force of the original stiffness characteristics of the rear springs, as determined in Chapter 3, by a value between 0.1 and 3. This range will proof to be more than sufficient to illustrate the main effects. The results are shown in Fig. 6.1. The ride comfort index, dynamic wheel load and suspension travel found for the baseline vehicle are, for ease of comparison, also indicated in this figure by a diamond.

As is demonstrated, the ride comfort index increases as the stiffness of the spring increases, while the dynamic wheel load only decreases slightly. The suspension travel initially rapidly decreases as the stiffness increases, but subsequently decreases at a slow rate.

If the main goal is to delete the increase in dynamic wheel load and suspension travel, it is recommended to use a spring stiffness of about twice the old stiffness. This results in the same dynamic wheel load and suspension travel as found for the ICE vehicle, although the ride comfort is then only decreased by 11.5% instead of 13.7% with respect to the ICE vehicle. More improvements in either dynamic wheel load or
6.1. Vehicle conversion and suspension modification

The overall ride comfort index (RMS) can not be obtained by changing the stiffness. If a high ride comfort is the main goal, it is recommended to use the same stiffness as used for the ICE vehicle. This does not result in any changes in the dynamic wheel load and suspension travel, but since the dynamic wheel load and suspension travel are only increased slightly due to the conversion, this is not considered to be a problem.

**Damper modification**

To investigate the effect of modified characteristics of the damper, the force of the original rear non-linear damper characteristic, as also determined in Chapter 3, is multiplied by a value between 0.1 and 3. The results are shown in Fig. 6.2.

From this figure it can be concluded that the ride comfort index increases rapidly as the damping constant increases. The dynamic wheel load and suspension travel initially decrease rapidly, then increase slightly again for the higher multiplication factors. The minimum ride comfort index is found at a multiplication factor of 1.5.
factor of approximately 0.3, while the minimum dynamic wheel load is found for a factor of approximately 1.5. As the ride comfort index increases rapidly and the dynamic wheel load decreases only slightly for higher damper factors, it is recommended to use the same dampers for the BEV as implemented in the baseline vehicle.

So far, the effect of changing the damper or spring characteristics are examined separately. The effect of changing both characteristics simultaneously has also been investigated. However, this did not show major improvements. Therefore, it can be concluded that the suspension of the Lupo 3L can be considered to be optimal for the BEV and the parameters of the suspension system do not have to be altered, taking into account that the small increase in dynamic wheel load and suspension travel due to the conversion do not impose a problem. Note that in this case this means that the rear springs of the real BEV have to be lengthened to level the car. If the small increase in dynamic wheel load and suspension travel is considered to be a problem, the stiffness of the rear springs can best be increased by a factor 2.

6.2 Influence of in-wheel motors

As previously discussed, the BEV becomes 14% more comfortable than the baseline vehicle due to the increase in sprung mass. Assuming that the ride of the baseline vehicle is comfortable, this increase in comfort offers a bigger margin to deal with the negative effects of the in-wheel motors. Because of this, in-wheel motors might still be an option for a BEV. Therefore, the effect of in-wheel motors on the ride comfort, dynamic wheel load and suspension travel of a BEV are investigated and described in this section.

The investigation is performed for three vehicle configurations: a configuration with in-wheel motors at the front (BEV-front), a configuration with in-wheel motors at the rear (BEV-rear) and a configuration with in-wheel motors in all the four wheels (BEV-four). The total increase in unsprung mass is 30 kg for all three configurations, since this value is also used for the experiments as described in Chapter 4. Since the increase in unsprung mass resembles the in-wheel motors replacing the central motor at the front, the increase in unsprung mass is subtracted from the sprung mass. The absolute values of the ride comfort index, dynamic wheel load and suspension travel for the baseline vehicle, the BEV, BEV-front, BEV-rear and BEV-four are given in Fig. 6.3. The dynamic wheel load and suspension travel are divided into the front side and rear side of the vehicle to show the distribution of these values over the front and rear wheels. The average of these values and the relative differences of the BEV-front, BEV-rear and BEV-four compared to the BEV are given in Table 6.3.

<table>
<thead>
<tr>
<th></th>
<th>$\alpha_{overall}$</th>
<th>dFz</th>
<th>$dz$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline</td>
<td>1.132 m/s²</td>
<td>796 N</td>
<td>10.6 mm</td>
</tr>
<tr>
<td>BEV</td>
<td>$-13.7%$</td>
<td>$+2.1%$</td>
<td>$+5.7%$</td>
</tr>
<tr>
<td>BEV-front</td>
<td>$-1.5%$</td>
<td>$+20.2%$</td>
<td>$+7.5%$</td>
</tr>
<tr>
<td>BEV-rear</td>
<td>$-6.3%$</td>
<td>$+21.2%$</td>
<td>$+8.5%$</td>
</tr>
<tr>
<td>BEV-four</td>
<td>$-5.6%$</td>
<td>$+22.7%$</td>
<td>$+8.5%$</td>
</tr>
</tbody>
</table>

From these results it can be concluded that the increase of the ride comfort index is the highest for the BEV-front, as to be expected. The increase of the ride comfort index for the BEV-rear and BEV-four are comparable. However, the highest ride comfort index is still found for the baseline vehicle, which means that the BEV with in-wheel motors will be as comfortable as the ICE Lupo 3L. Unfortunately, the average dynamic wheel load and suspension travel increase by approximately 20% and 8%, respectively. The figure clearly shows that the dynamic wheel load and suspension travel are especially increased at the wheels equipped with in-wheel motors. This means that for the iw-front the dynamic wheel load at the front increases with around 40%, which results in understeer of the vehicle. For the iw-rear this means that the dynamic wheel load at the rear increases with around 40%, which results in oversteer of the vehicle. For the BEV-four the dynamic wheel load at the front is increased by 20%, as is also the case for the rear. Since
6.3 BEV-front suspension optimization

As noted in Chapter 4, the increase in under- or oversteer of the vehicle was perceptible during excessive cornering maneuvers like a J-turn. Therefore, it is desired to diminish the increase in dynamic wheel caused by in-wheel motors. Three possible solutions are investigated and described in this section: changing the suspension parameters, changing the tyre stiffness and using a dynamic vibration absorber.

The previous section demonstrates that a good ride comfort is obtained by using a soft suspension system to isolate the sprung mass from road disturbances, while good road holding is obtained by a stiffer suspension system. This means that an improvement in one area is obtained at the expense of a deterioration in the other area. Taking this and Fig. 6.3 into account it would be obvious to optimize the suspension of the BEV-rear or the BEV-four, since they have the lowest ride comfort index. However, equipping the vehicle with rear in-wheel motors also means that regenerative braking would be applied to the rear wheels. This is not advisable, since this can result in unstable braking. Equipping the vehicle with four in-wheel motors results in undesirable high costs and complexity. Therefore, it is chosen to investigate the possible solutions using the BEV-front configuration. All simulations are performed on Belgian blocks with a forward velocity of 50 km/h.

Spring and damper modification

The effect of the spring and damper characteristics are investigated by multiplying the original front spring stiffness and non-linear damper characteristics of the front dampers by a value between 0.1 and 3, respectively. Both results are shown in Fig. 6.4.

This figure shows almost the same results as found in Section 6.1.2. The stiffness of the spring has a significant effect on the ride comfort index, but not on the dynamic wheel load and suspension travel. Changing the damping characteristics can lead to a decrease in dynamic wheel load and suspension travel, but only at the expense of the ride comfort. Using a damping multiplication factor of 2, the lowest dynamic

---

Figure 6.3: Change in ride comfort, dynamic wheel load and suspension travel for the baseline vehicle, the BEV and the BEV equipped with in-wheel motors.

they balance each other out, this does not necessarily have to lead to an increase in under- or oversteer, but could lead to a reduction of the maximum lateral acceleration of the vehicle.

In order to decrease or even eliminate the negative effects of the in-wheel motors completely, the improvements by optimization of the suspension system are investigated in the next section.
wheel load is found. However, at this point it is still around 20% higher compared to the baseline vehicle, while the overall ride comfort index is increased by approximately 28%. Since this is highly undesirable, changing the suspension characteristics is not the ideal option to lower the dynamic wheel load.

**Tyre pressure**
A lower tyre vertical stiffness is achieved by lowering the tyre inflation pressure. The effect of the tyre inflation pressure on the ride comfort, dynamic wheel load and suspension travel is shown in Fig. 6.5.

This figure clearly shows the decrease in dynamic wheel load by the decrease in tyre pressure. The ride comfort index and suspension travel are only slightly influenced by the tyre pressure and therefore by the tyre vertical stiffness. At a pressure of 2 bar, the ride comfort index is still low, the suspension travel is almost unchanged and the dynamic wheel load is comparable to the baseline vehicle. Based on these results it can be concluded that decreasing the tyre inflation pressure is a good option to lower the dynamic wheel load.
load. However, it should be noted that changing the pressure affects other important tyre characteristics like wear, sound, rolling resistance and handling, which makes it a less suitable option.

**Dynamic vibration absorber**

A dynamic vibration absorber (dva) is an additional mass that is connected to the unsprung mass by a damper and spring as shown in Fig. 6.6. This principle is already used in the Dynamic-Damping In-wheel Motor Drive System designed by Bridgestone [12].

![Figure 6.6: Representation of the dynamic vibration absorber.](image)

In order to determine the effect of a dva, the simulation model is equipped with two absorbers, one on each front suspension, and the dva damper constant is changed between 0 and 1000 Ns/m. The mass of the absorber is taken to be 5 kg, which is 10% of the unsprung mass as is also taken by Hrovat [39] in order to globally study the benefits of a dva. The stiffness of the springs are chosen such that the eigenfrequency of the absorber lies around the wheel-hop frequency (13.5 Hz), which is achieved with a stiffness of 33400 N/m. The results of this investigation can be found in Fig. 6.7. The values found for the BEV-front without a dva and for the baseline are also given (at 0 Ns/m) for ease of comparison.

![Figure 6.7: Influence of the damping constant of a dynamic vibration absorbers on the ride comfort index, dynamic wheel load and suspension travel of the BEV-front.](image)

This figure shows that a lowest dynamic wheel load occurs at a damping constant of 150 Ns/m. This damping constant leads to a 10% reduction in the dynamic wheel load and a 5% reduction in the ride
comfort index compared to the BEV-front. However, this still means that the dynamic wheel load is around 30% higher than the baseline vehicle.

Please note that for this investigation the dva spring stiffness has a specific constant value. It is also an option to change the damper constant, as done above, for a dva spring stiffness other than the chosen one. However, it has been found that this does not lead to much improvement.

As the comfort of the drive is increased by approximately 5% in combination with a decrease in dynamic wheel load of about 10% it can be concluded that the dva has potential to improve the ride and road holding of the vehicle. However, this decrease in dynamic wheel load is not enough to eliminate the increase due to the in-wheel motors. Besides, using dva’s also adds mass to the vehicle and they are not easy to implement due to the packaging requirements. This leads to design and development challenges that have to be resolved in order for the system to be used. This is possibly also the main reason why there has been only one vehicle, the Citroen 2 CV (1949), which used an absorber in order to improve the ride comfort [39].

6.4 Discussion

In this chapter the conversion of the ICE vehicle into a BEV is addressed. It is shown that due to the increase in vehicle mass, the vehicle becomes approximately 14% more comfortable, while it only slightly increase the dynamic wheel load and suspension travel. Increasing the damping of the suspension system decreases the dynamic wheel load and suspension travel insignificantly. As such, the current suspension system in the ICE Lupo can be considered as optimal for the BEV. To account for the increased weight at the rear of the vehicle, the rear springs have to be stiffened or lengthened.

Simulations show that using in-wheel motors in the BEV results in a decrease of the ride comfort, although this decrease does not lead to a less comfortable ride than the baseline ICE vehicle. The in-wheel motors do increase the dynamic wheel load with approximately 40% and the suspension travel with approximately 16%. Especially the increase in dynamic wheel load is considered to be high. By changing the suspension parameters or by the use of a dynamic vibration absorber, this increase in dynamic wheel load can only be reduced to an increase of approximately 20%, which is still considered to be high. It can further be decreased by lowering the tyre pressure, but this negatively effects other important tyre characteristics.

Options that remain to lower the dynamic wheel load are the use of semi-active or active suspension systems. The possible improvements in ride comfort and road holding of the vehicle of such systems are described in the next chapter.
Chapter 7

Semi-active and active suspension control

In the previous chapter it has become clear that using in-wheel motors in a BEV does not lead to a less comfortable ride than the baseline ICE vehicle. However, simulation results show that the dynamic wheel load and suspension travel do increase by approximately 40% and 16%, respectively. Furthermore, the results show that adapting the parameters of the passive system does not lead to significant improvements, neither does the use of a dynamic vibration absorber. Because of this, other options have to be explored to eliminate the increase in dynamic wheel load and suspension travel. Most preferably, this should be accompanied by an increase of the ride comfort. Therefore, the possible improvements by the use of a (semi-) active suspension system are investigated in this chapter.

Two control strategies are considered: the skyhook damping control and the hybrid damping control. Both are investigated under the assumption that it can work semi-active or fully active. The investigation is performed using an electromagnetic suspension system, which is designed at the Department of Electrical Engineering at the Eindhoven University of Technology [36].

First, existing semi- and active systems are described in Section 7.1. More information about the electromagnetic system and its limitations is given in Section 7.2. The control strategies and the performance of these controllers are illustrated in Section 7.4. The energy consumption of the system is investigated and discussed in Section 7.5, as a possible downside of a semi- or active system is that it uses a certain amount of energy.

7.1 Existing semi- and active suspension systems

This section will address the existing semi-active and active suspension systems and their advantages over a passive system. While a passive suspension system is only able to store energy via a spring and to dissipate energy via the damper, a semi-active suspension system is able to adapt some of its characteristics to achieve a controllable energy dissipation. This is also the case for an active system, but this system can also introduce energy, and thus a controlled force, into the system.

The existing semi-active suspension systems and their advantages and disadvantages are addressed first, followed by examples of the existing active suspension systems.
7.1.1 Semi-active suspension systems

There are several semi-active suspension systems, knowing:

- hydraulic or pneumatic continuously variable damper systems
- Magneto- and Electro-rheological damper systems
- hydropneumatic systems
- Frequency Selective Damping systems

The continuously variable damper system consists of a proportional damping valve. This controls the liquid or air resistance of the damper, resulting in a continuously variable damper adjustment. Only a limited amount of power is needed to actuate the valves. This actuation can be achieved by an electric motor or by magnet valves [14, 29].

The electro-rheological and magneto-rheological damper systems contain fluids which are able to change their rheological properties, like elasticity, plasticity and viscosity, upon application of an electric or magnetic field. The electro-rheological fluids have a fast response to an electric field resulting in a high control bandwidth, but the rheological changes are relatively small and exhibit an extreme property change with temperature. Although the power requirements of both systems are approximately the same, the magneto-rheological fluids require small voltages and currents, while the electro-rheological fluids require very large voltages and very small currents. The magneto-rheological damper system is already used on a number of high-end cars like the Audi R8 and the Ferrari 599GTB. The reliability of both damper systems is high, since it changes into a passive damper system in case of a power breakdown [33, 29].

In a hydropneumatic system, invented by Citroën, each of the four suspension struts contain a sphere on top. Each sphere consists of two compartments, separated by a flexible rubber membrane. The top of the sphere is filled with high pressurized nitrogen and the bottom, openly connected to the suspension strut, is filled with oil. Due to irregularities in the road, the suspension rod pushes oil into the sphere, which pressurizes the nitrogen even more. By doing so the gas works as a spring. The damping of the system is provided by valves in the orifice between the strut and sphere. Due to the properties of gas (half its volume doubles its pressure) the suspension is very soft around its initial working point and becomes more stiff as it is compressed. This means that during normal driving the stiffness is low, while at cornering it is very stiff. The most well known hydropneumatic system in current vehicles is the so called Hydractive found in several Citroën vehicles. Besides a sphere on each of the four suspension struts, extra spheres are added at the center of each axle. Each center sphere links the fluid between the left and right sphere. Due to this interconnection, the negative effects of a mechanical anti-rollbar, observed when only one wheel hits a bump, are eliminated. By opening or closing the valves in the center spheres, a quick and smooth transition between a soft or hard suspension system can be obtained. The whole circuit is pressurized by a pressure pump powered by the engine [15, 31]. A representation of the system is given in Fig.7.1.

In a Frequency Selective Damper the damping characteristics are not only defined by the oil flow through the conventional piston valves, but also by an additional valve that controls an oil flow parallel to the first one. The control of the second valve gives rise in damping force almost linear to the time that the piston is moving in one direction. This means that at low frequencies the damping force is high and handling is improved, while at high frequencies the damping is low and comfort is improved [16].
7.1. Existing semi- and active suspension systems

Figure 7.1: Hydractive suspension system designed by Citroën: 1 - Integrated hydrotronic unit, 2 - front suspension struts, 3 - front stiffness regulator, 4 - front electronic position sensor, 5 - rear hydropneumatic cylinders, 6 - rear stiffness regulator, 7 - rear electronic position sensor, 8 - interface, 9 - steering wheel sensor, 10 - hydraulic fluid reservoir, 11 - accelerator-pedal and brake-pedal [15].

7.1.2 Active suspension systems

There are several active suspension systems, knowing:

- hydraulic or pneumatic systems
- electro-magnetic actuators
- variable geometry force actuator

A hydraulic (or pneumatic) active suspension system comprises of an actuator, an electro-hydraulic control valve and a pump. The actuator consists of a piston sealed in a cylinder with valves on either ends that lead to the power spool valve. By positioning the power spool valve, fluid pressurized by the pump flows into one cylinder chamber, while the fluid in the other cylinder chamber is allowed to flow into the pump reservoir. This creates a pressure difference between the two chambers and active suspension force is created. The spool valve can be slided electronically into different positions to either pressurize the bottom or the top chamber, creating a force in upward or downward direction. Often, also a bypass valve is added to control the flow resistance between the two chambers of the actuator [18]. The Lotus F1 cars in the early 1980's were fully actively suspended by hydraulics, although they did consume a lot of energy [42].

Another example of the use of hydraulic actuators is the Active Body Control system developed by Sachs for Daimler. This system is standard on all Mercedes-Benz SL and CL models. In this system a hydraulic actuator is placed in series with the conventional steel spring. During cornering, the springs on the outer side of the corner will be compressed, while the springs on the outer side of the corner will be stretched. In order to maintain the vehicle in a level position, the hydraulic chamber on the outer side of the corner will be filled to compensate for the compression of the spring, while the chambers on the inside of the corner will be emptied in order to compensate for the extension of the spring [14]. These hydraulic active systems have a very high force density, are easy to control, are commercial available, have a commercial maturity and are reliable. However, the bandwidth of these active systems are mostly limited to approximately 5 Hz due to pressure loss and flexible hoses and they are considered to be inefficient due to the required pressure (high power consumption). Hydraulic active systems are also causing environmental pollution due to hose leaks and ruptures, since the hydraulic fluids are toxic. These active systems can work very well to lower the roll and pitch motion of the vehicle, but due to the low bandwidth (up to about 5 Hz), they are not able to reduce the accelerations of the sprung mass significantly [40].

In an electromagnetic suspension system forces are generated by means of linear electric motors. This active system is bidirectional, meaning that it can work as a motor as well as a generator. In motor mode
the system is able to supply a force on the sprung mass to eliminate body roll or pitch. In generator mode the system is able to act as a damper to absorb road vibrations, where the absorbed energy can be fed back into the batteries. These systems are considered to have a high force density due to the tubular structure and have a relatively high bandwidth up to 50 Hz. No continuous power is needed since mechanical springs support the sprung mass resulting in a minimization of energy consumption. The force density is lower than that of a hydraulic system and therefore the mass of the system is larger. However, the overall weight added to the vehicle body could be lower, due to the absence of hydraulic pipes, fluids and pumps. This also results in less pollution and the absence of acoustic noise. Due to the placement of a system at each corner of the vehicle, no anti-roll bar is required and the disadvantages of the anti-roll bar are no longer present. The system combines active suspension and active roll control. The limiting factor for the electromagnetic actuator is the maximum actuator temperature. The temperature rise is a direct consequence of the dissipated power. Examples of electromagnetic suspensions are the system designed by Gysen et al. [36] and the system designed by Bose implemented in the Lexus 400 LS. An impression of the Bose system is given in Fig. 7.2 [17].

![Electromagnetic suspension system as designed by Bose](image)

Figure 7.2: Electromagnetic suspension system as designed by Bose [17].

In a Variable Geometry Force Actuator system, designed by Evers et al. [26], the force acting on the sprung mass is changed by rotating or moving several parts of the suspension system, which give rise to variable forces acting on the sprung mass. This system, shown in Fig.7.3, consists of a wishbone with length $l$ connected through a rotational joint with the unsprung mass. An electric motor, attached to the wishbone, controls the orientation of an actuator arm with length $h$. At the end of this arm a rod is connected through a rotational joint and attached to a preloaded spring. The force within this spring gives rise to a force at the end of the wishbone. By rotating the actuator arm a variable force can be created at the end of the wishbone. The performance increase with this system is about the same as that of linear actuator systems, but the peak required power is ten to eighty times lower. For more information the reader is referred to the description of the variable geometry active suspension system as given in [26].

Another possible active suspension system is the Dynamic Drive System of BMW, designed by Sachs. This system comprises a hydraulic rotation actuator placed in the anti-rollbar. However, this system is only able to control the roll motion of the vehicle, resulting in an improved vehicle steering. Vibrations of the body due to the road input and the vehicle pitch motion are not influenced by this system. Therefore the vehicle is also equipped with a semi-active damper system. This makes it a promising trade-off between complexity, costs and energy demand [14, 29].

Although semi-active and active systems can be used to both increase the drive comfort and handling behavior of the vehicle, these systems are also more complex, have a higher installation and maintenance costs, and have a reduced reliability compared to passive suspension systems [40].
7.2. Electromagnetic system specifications

This section describes the specifications of the electromagnetic suspension system as designed by Gysen et al. and includes the limitations of the system that have to be taken into account.

The electromagnetic suspension system, as shown in Fig. 7.4, consists of a brushless tubular permanent magnet actuator parallel with a passive coil spring. The passive coil spring is added to support the sprung mass without the need of continuous electric power.

The force specifications of this system are determined based on measurements taken on the Nürburgring. From these measurements it is concluded that the system has to be able to apply a mean force of 1 kN with a peak force of 4 kN. The volume specifications are chosen such that the system can easily replace an existing passive suspension system. Moreover, mass and thermal constraints are taken into account during the design. By a 2-D finite element analysis it is shown that the final design is capable of delivering a continuous force of 755 N and a peak force of 4 kN (at 0.1 m/s). A higher force level results in a temperature higher than the maximum allowable temperature of 120 degrees.
Since the system has to be safe when a power breakdown occurs, it needs to have a certain amount of passive damping. Passive damping can be incorporated in the system electromagnetically by means of eddy currents or by the placement of an external passive damper. Gysen et al. have chosen to use eddy currents. The exact working principle of eddy currents and the resulting passive damping goes beyond the scope of this investigation and is therefore not further explained. However, it is noted that with the use of eddy currents the amount of passive damping force can be designed upon the wish of the designer. Because of this, and since it is expected that the passive damping system has an influence on the performance of the system, the investigations performed in this chapter are carried out for several passive damping constants.

One example of the passive damping that can be obtained by the use of eddy currents is the average damping of the Lupo damper as shown in Fig. 7.5. In this figure the non-linear damper characteristics of the Lupo damper (solid lines) are shown together with the average damping (dashed line), which thus resembles the passive damping of the electromagnetic system. In reality, the passive damping created using eddy currents will not be totally linear, but saturation will occur at higher velocities. As this effect is not clearly known, this saturation is not taken into account.

As stated above, the system is limited to a continuous force of 755 N. This means that, due to the presence of the passive damping, the system is able to apply a total force of 755 N on top of the force delivered by the passive damping. The force constraints of the system are therefore given by adding or subtracting the maximum force from this passive damping force, as is also indicated in Fig. 7.5 by the dotted lines. Although the actuator is theoretically able to supply or regenerate a force of 755 N, it is decided to take a maximum force of 500 N to be on the conservative side. The peak force of 4 kN is not taken into account, since it is unknown how long the system is able to supply this force. The working area of the actuator is divided into four segments: 2 segments where the actuator works in the motor mode (M) and two segments where the actuator works in regenerative mode (R). This is due to the fact that the actuator, on top of the passive damping force, works in motor mode when it has to supply a force in the same direction as the movement of the actuator, while it works in generation mode when it has to supply a force in the opposite direction.

It should also be taken into account that it requires some computational time to measure states of interest (for example the velocity of the sprung body) and that the actuator is not able to instantaneous deliver the forces at high frequencies. As such, the switching dynamics of the actuator are taken into account by means of a first order system:

$$H(s) = \frac{1}{\tau s + 1}$$  (7.1)

with $\tau$ the switching time constant [s]. As is claimed by the designers that the system has a bandwidth of at least 100 Hz, the switching time $\tau$ is taken to be 0.0016 seconds.

### 7.3 Semi-active and active suspension control

This section describes the theory of the control techniques that are used to investigate the possible improvements with the use of the electromagnetic suspension system.

In literature, many different types of control techniques are found that can be used to control the suspension system: e.g. Linear Optimal control, preview control, adaptive control or robust control [60]. Mantaras et al. [47] compared seven different active and semi-active control strategies in order to research the improvement in ride comfort and handling stability compared with a passive system. The seven control techniques are: Linear Optimal control, Robust control, Kalman filter, Skyhook damper, Pole-assignment, Neural network and Fuzzy logic. They concluded that significant improvements can be obtained in the ride comfort as compared to a passive system. However, the contradiction between an improvement in ride comfort and a deterioration of the dynamic wheel load is still present: the active control methodologies that improve the ride comfort the most have the lowest road holding capability.

According to Rajamani [55], the most simple control, the skyhook control, provides almost all the performance improvements that can be achieved by more complex control techniques like Linear optimal
control. Therefore, it is chosen to analyse the skyhook control. However, it is expected that when using the skyhook control, the motion of the unsprung mass becomes highly undamped, resulting in high dynamic wheel loads. Since the main purpose of the system is to lower the dynamic wheel load, this is undesired. Combining the skyhook control with the groundhook control, the so called hybrid control, can possibly lead to better results. Therefore, improvements of the hybrid control are investigated also.

**Skyhook control**

Theoretically, the ideal skyhook damping control is based on connecting the vehicle sprung mass to an inertial reference in the sky by a damper, as can be seen in the schematic representation of a 2 DOF model as given in Fig. 7.6 (a) [32]. In this configuration the sprung mass becomes more resistant to vertical accelerations, independent of the influence of the unsprung mass. Note that this is impossible to realize in practice directly. However, the skyhook damper force can be generated by implementing an actuator between the sprung and unsprung mass. The modified model is shown in Fig. 7.6 (b).

The amount of skyhook force that needs to be delivered depends on whether the system is fully active or semi-active. In case of the active skyhook control strategy, the skyhook force is given by:

\[ F_{sky} = -d_{sky} \cdot \dot{z}_s \]  \hspace{1cm} (7.2)

with \( d_{sky} \) the skyhook damping constant and \( \dot{z}_s \) the vertical velocity of the sprung mass.

In case of the semi-active skyhook control strategy, the system can only generate a damping force when the velocity of the sprung mass (\( \dot{z}_s \)) multiplied by the velocity of the unsprung mass relative to the sprung mass (\( \dot{z}_a - \dot{z}_s \)) is lower than zero. This can be explained as follows. If for example the sprung mass moves upwards with a positive velocity and the unsprung mass moves downwards, then the system is extending and the actuator can apply a damper force opposite to the sprung velocity. If now the sprung mass moves downwards with a velocity less than the downward velocity of the unsprung mass, the system is still extending, meaning that the actuator can only apply a force downward, while an upward force is requested. As the system is not able to supply a force, the best that can be achieved is to minimize the damping force. As such, the semi-active skyhook force is given by:

\[
\begin{align*}
F_{sky} &= -d_{sky} \cdot \dot{z}_s & \text{if } \dot{z}_a (\dot{z}_a - \dot{z}_s) < 0 \\
F_{sky} &= 0 & \text{if } \dot{z}_a (\dot{z}_a - \dot{z}_s) > 0
\end{align*}
\]  \hspace{1cm} (7.3)

with \( \dot{z}_a \) the vertical velocity of the unsprung mass.
The total force delivered by the electromagnetic suspension system consists of the skyhook force in combination with the passive damping force:

\[ F_{act} = F_{sky} + d_p \cdot (\dot{z}_a - \dot{z}_s) \]  

(7.4)

The Simulink model of the semi-active skyhook controller is given in Fig. 7.7. The skyhook force is limited to 500 N by means of a saturation block. By deleting the switch in the Simulink model the active skyhook controller can be obtained.

**Figure 7.7: Simulink model of the semi-active skyhook controller.**

**Hybrid control**

The hybrid controller is a combination of the above described skyhook controller with a groundhook controller [32]. Where the skyhook controller is especially used to control the motion of the sprung mass, the groundhook controller aims to control the motion of the unsprung mass.

The groundhook controller works almost the same as the skyhook controller, however, instead of connecting the sprung mass to an inertial reference, the unsprung mass is connected to the inertial reference. The amount of groundhook force depends on whether the system is fully active or semi-active. In case of the active system the groundhook force is given by:

\[ F_{gnd} = -d_{gnd} \cdot \dot{z}_a \]  

(7.5)

while in the case of the semi-active system the groundhook force is given by:

\[
\begin{align*}
F_{gnd} &= 0 & \text{if } \dot{z}_a(\dot{z}_s - \dot{z}_a) > 0 \\
F_{gnd} &= -d_{gnd} \cdot \dot{z}_a & \text{if } \dot{z}_a(\dot{z}_s - \dot{z}_a) < 0
\end{align*}
\]  

(7.6)

with \(d_{gnd}\) the groundhook damping constant. The hybrid force is given by:

\[ F_h = \alpha F_{sky} + (1 - \alpha)F_{gnd} \]  

(7.7)

with \(\alpha\) the relative ratio between the skyhook force and the groundhook force. In the case when \(\alpha\) is 1, the control approach becomes a pure skyhook controller. When \(\alpha\) is 0, the control approach reduces to the pure groundhook controller. The total force delivered by the electromagnetic suspension system is given by the passive damping forces added to the hybrid force:

\[ F_{act} = F_h + d_p \cdot (\dot{z}_a - \dot{z}_s) \]  

(7.8)

It is expected that the best improvements can be found when the skyhook and groundhook forces, given by Eq. (7.2) and Eq. (7.5) for the active case, are considered as the desired total actuator force. Since part of this desired force will be supplied by the passive damping force, the passive damping force is subtracted.
from the desired skyhook and groundhook force to obtain the force that is delivered by the skyhook or groundhook controller. This working principle is visible in the Simulink model of the semi-active hybrid controller, as shown in Fig. 7.8. The active hybrid controller is obtained by deleting the switches (Switch and Switch1) in the Simulink model.

Figure 7.8: Simulink model of the semi-active hybrid controller.

The performance of the two control approaches, both semi-active and active, is discussed in the next section.

### 7.4 Control performances

This section shows the possible improvements in comfort, dynamic wheel load and suspension travel using skyhook control and hybrid control. Both are investigated for the semi-active and active case.

For reasons of simplicity and calculation time, these investigations are performed using the quarter car model as shown in Fig. 7.6. This model captures many essential characteristics of a real suspension system and as such, the insights gained with this model are also applicable to the full car model. Since the active suspension system is used to delete the negative effects of in-wheel motors, all simulations are performed with the parameters of the BEV-front. The simulations are performed by driving on Belgian blocks with a vehicle forward velocity of 50 km/h. Moreover, as it has been found that the amount of passive damping has an influence on the results, the control strategies are investigated for a passive damping value of 500, 700, 900 and 1100 Ns/m. The values of the ride comfort index, dynamic wheel load and suspension travel for these passive damping values are provided in Table E.1 in Appendix E.

First, the simulation results using semi-active and active skyhook control are shown, followed by the results using semi-active and active hybrid control. Hereafter, the improvements of the control techniques are compared to each other. The simulation results are demonstrated by plotting the ride comfort index versus the dynamic wheel load and by plotting the ride comfort index versus the suspension travel. Take in mind during the viewing of the results that it is undesirable that an improvement in one area is accompanied by a deterioration in another area. Therefore, the ride comfort index, dynamic wheel load and suspension travel found for the quarter BEV-front are implemented in the plots by solid vertical and horizontal lines. Every point in the down left square given by the solid lines is an improvement in both ride comfort and dynamic wheel load or in both the ride comfort and suspension travel. The values found for the quarter BEV without in-wheel motors are also implemented by dotted vertical and horizontal lines. Preferably, the controllers should be able to decrease the ride comfort index and dynamic wheel up to or beyond this point.
Skyhook control performance

The performance of the semi-active and active skyhook controller is investigated by varying the skyhook damping constant $d_{sky}$ between 250 and 40000 Ns/m. Fig. 7.9 (a) and (b) show the results using semi-active skyhook control and Fig. 7.9 (c) and (d) show the results using active skyhook control.

From these figures it is concluded that the improvements of the skyhook controller strongly depend on the passive damping of the system. Increasing the passive damping results in a lower dynamic wheel load but also in a higher ride comfort index. The lowest ride comfort index without (high) deterioration of dynamic wheel load, indicated by encirclement 1, is found for a passive damping of 700 Ns/m in case of semi-active control or 900 Ns/m in case of active control. The lowest dynamic wheel load without deterioration of the ride comfort index, encirclement 2, is found for a passive damping of 900 Ns/m or 1100 Ns/m for the semi-active and active case, respectively. The suspension travel is in both cases decreased, even beyond the value found for the BEV. This is also the case for the ride comfort index, but this is accompanied with a high deterioration of the dynamic wheel load. The dynamic wheel load can not be decreased beyond the value found for the BEV.

The absolute values of the ride comfort index, dynamic wheel load and suspension travel found at the encircled points are provided in Table E.2 and Table E.3 in Appendix E for the semi-active and active case, respectively. The relative differences with the BEV-front at these points, and thus the improvements of the controllers, are discussed parallel to the improvements of the hybrid controller at the end of this section.
Hybrid control performance

In the case of hybrid control, the total actuator force supplied by the system depends on four parameters (see Section 7.3): the skyhook damper value \( d_{sky} \), the groundhook damper value \( d_{gnd} \), the passive damping value \( d_p \), and the relative ratio between the skyhook force and the groundhook force \( \alpha \). To decrease the amount of combinations, the value \( d_{sky} \) and \( d_{gnd} \) are chosen to be fixed for each passive damping value. The value \( d_{sky} \) is chosen such that the lowest ride comfort index is found when only pure skyhook control is considered \( (\alpha = 1) \), and the value \( d_{gnd} \) is chosen such that the lowest dynamic wheel load is found when only pure groundhook control is considered \( (\alpha = 0) \). The ride comfort index, dynamic wheel load and suspension travel for each combination of values are subsequently determined for an \( \alpha \) ranging between 0 and 1.

Fig.7.10 (a) and (b) show respectively the ride comfort index versus the dynamic wheel load and the ride comfort index versus the suspension travel using semi-active hybrid control. The results found for the active case are shown in Fig.7.10 (c) and (d).

![Figure 7.10: Influence of the relative ratio \( \alpha \) on the ride comfort, dynamic wheel load and suspension travel of the BEV-front for four different values of the passive damping. a) and b): semi-active hybrid control, c) and d): active hybrid control.](image)

From these figures it can be concluded that the ride comfort and dynamic wheel load do not strongly depend on the amount of passive damping: the differences are less than 1%. The only difference is their ability to emphasize one specific aspect like ride comfort or dynamic wheel load. Using a high passive damping the dynamic wheel load can be more emphasized, while using a low passive damping the ride comfort can be more emphasized. The difference between semi-active and active hybrid control is that the active controller
is able to decrease the ride comfort index even beyond the ride comfort index of the BEV without in-wheel motors. The semi-active controller can only achieve this with a low passive damping value of 500 Ns/m. Although the hybrid controller is able to decrease the suspension travel significantly, it is unable to decrease the dynamic wheel load with respect to the BEV.

The ride comfort index, dynamic wheel load and suspension travel found at the encircled points can be found in Appendix E in Table E.4 and in Table E.5 for the semi-active and active case, respectively. The improvements of the hybrid control approach are discussed next in combination with the improvements found using skyhook control.

**Comparison of controllers**

To compare the control techniques, the relative difference of the ride comfort index, dynamic wheel load and suspension travel with respect to the BEV-front, found for each control approach at the encirclements, are determined and given in Table 7.1.

<table>
<thead>
<tr>
<th>Skyhook semi-active</th>
<th>Hybrid semi-active</th>
<th>Skyhook active</th>
<th>Hybrid active</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comfort improvements (encirclement 1)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$a_{overall}$</td>
<td>$-13.6%$</td>
<td>$-12.7%$</td>
<td>$-11.8%$</td>
</tr>
<tr>
<td>$dFz$</td>
<td>$+0.9%$</td>
<td>$+0.8%$</td>
<td>$-0.0%$</td>
</tr>
<tr>
<td>$dz$</td>
<td>$-27.0%$</td>
<td>$-27.0%$</td>
<td>$-23.8%$</td>
</tr>
<tr>
<td>Dynamic wheel load improvements (encirclement 2)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$a_{overall}$</td>
<td>$-4.5%$</td>
<td>$-0.1%$</td>
<td>$-0.0%$</td>
</tr>
<tr>
<td>$dFz$</td>
<td>$-5.6%$</td>
<td>$-9.0%$</td>
<td>$-7.6%$</td>
</tr>
<tr>
<td>$dz$</td>
<td>$-28.6%$</td>
<td>$-27.8%$</td>
<td>$-26.2%$</td>
</tr>
</tbody>
</table>

From this table it can be concluded that the semi-active skyhook controller manages to lower the ride comfort index the most. The dynamic wheel load is decreased most by the active skyhook controller. All control approaches are able to reduce the suspension travel with a similar level. Taking these results into account it is best to use active skyhook control for the electromagnetic suspension system. However, the improvements made by the hybrid controller can possibly be increased slightly by using different combinations of $d_{sky}$ and $d_{gnd}$. It is expected that this will lead to similar results as found with the skyhook controller.

Note that the above results are found under the assumption that an improvement in one area is not accompanied by a deterioration in another area. Without this assumption, the most improvements are found using active hybrid control. With this control, it is possible to decrease the dynamic wheel load by approximately 18% or to decrease the ride comfort index by around 45%, just by changing the value $\alpha$. However, this is accompanied by an increase in ride comfort index of 40% or an increase in dynamic wheel load of 38%, respectively. Nonetheless, the first setting can for example be used to enhance the vehicle behavior during cornering and the second setting can be used to enhance the comfort during straight driving.

As concluded in Chapter 6, the dynamic wheel load is due the placement of in-wheel motors increased by approximately 40%. Since this can only be diminished by 18% at most, the dynamic wheel load will still be approximately 22% higher compared to the BEV. This means that none of the investigated control approaches are able to diminish the increase in dynamic wheel load sufficiently.

### 7.5 Power consumption

As the available energy in a BEV is limited, the electromagnetic suspension should not use much energy. Because of this, the power consumption of the electromechanical system is investigated for both control approaches and described in this section.
7.5. Power consumption

In order to obtain reliable results, the energy losses in the batteries and in the bi-directional inverter and the energy losses in the system should be taken into account. The average efficiency of an inverter and batteries are both around 90% \[28\]. As such, the overall efficiency of the inverter and batteries are taken to be 81% for the investigation. The energy loss in the electromagnetic actuator is dictated by the amount of power that is lost due to copper losses. The average amount of copper losses is given by:

\[
P_{cu} = \frac{1}{t_e} \int_0^{t_e} \frac{3}{2} R_{ph} \hat{i}^2 dt
\]

(7.9)

with \(t_e\) the time [s], \(\hat{i}\) the amplitude of the three phase commutated current [A] and \(R_{ph}\) the resistance of the phase [\(\Omega\)]. The three phase commutated current \(\hat{i}\) depends on the output force of the actuator:

\[
\hat{i} = \frac{F_{net}}{K_i}
\]

(7.10)

with \(K_i\) the motor constant [N/A]. The resistance of the phase \((R_{ph})\) is taken to be 3.11 m\(\Omega\) and the motor constant is taken to be 5.99 N/A, as given in [36]. It should be noted that these values strongly depend on the supply voltage and the design of the actuator. However, the variations in these values by a change in supply voltage have a relatively small effect on the amount of copper losses. Therefore, it is assumed that the chosen values will give a good approximation of the copper losses. The energy losses found by the eddy currents are not considered as losses, since they contribute to the passive damping of the system.

In the case of skyhook control, the power consumption is determined for the specific combination of the passive damping and skyhook damping that result in the lowest dynamic wheel load and ride comfort index without deterioration of the other. In the case of hybrid control, the power consumption is determined for a passive damping of 900 and a value of \(\alpha\) that result in the lowest dynamic wheel load and ride comfort index without deterioration of the other. The passive damping of 900 Ns/m is chosen based on conclusions made in the previous section.

The consumed power \(M\), the regenerated power \(R\), the copper losses in both motor mode and regenerator mode \(P_{cu,M}, P_{cu,R}\) and the total consumed power \(T\) for the semi-active and active controllers while driving on Belgian blocks with a vehicle velocity of 50 km/h are given in Table 7.2. The efficiency of the batteries and inverter is already taken into account in the total consumed power \(T\). Note that these values are the consumed and regenerated power for one wheel. If the power consumption of the whole vehicle is requested, these values should be multiplied by the amount of actuators implemented in the vehicle. Note also that only the power consumption during driving on Belgian blocks is considered. It is assumed that the power consumption on other road types can be obtained by downsizing the found results.

<table>
<thead>
<tr>
<th>control</th>
<th>Dynamic wheel load optimized</th>
<th>Ride comfort optimized</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Skyhook</td>
<td>Hybrid</td>
</tr>
<tr>
<td></td>
<td>semi</td>
<td>active</td>
</tr>
<tr>
<td>M [W]</td>
<td>0</td>
<td>4.8</td>
</tr>
<tr>
<td>(P_{cu,M}) [W]</td>
<td>0</td>
<td>0.2</td>
</tr>
<tr>
<td>R [W]</td>
<td>26.6</td>
<td>10.9</td>
</tr>
<tr>
<td>(P_{cu,R}) [W]</td>
<td>2.1</td>
<td>0.3</td>
</tr>
<tr>
<td>T [W]</td>
<td>-19.8</td>
<td>-2.4</td>
</tr>
</tbody>
</table>

As can be seen, the semi-active controllers do not consume any energy, as it only works in regeneration mode. In the case of active control, the system does consume energy, although it regenerates more in most cases. Only the active hybrid control consumes a small amount if the ride comfort is optimized. In order to put the amount of energy consumed or regenerated by the system into perspective it can for example be compared to the power consumption of the auxiliaries in conventional vehicles: lights consume 200 W, heating consumes 3000 W and airconditioning consumes 3600 W [51]. Therefore, it can be concluded that
the system does not consume much energy, and as such, the system can be implemented in a BEV. The explanation for the low consumed or regenerated power is the fact that force is already partly delivered by the passive damping present in the system and that the force is limited to 500 N.

### 7.6 Discussion

In Chapter 6 it is concluded that the BEV with front in-wheel motors has almost the same ride comfort as the baseline vehicle, but the dynamic wheel load is increased by approximately 40%. Optimization of the passive system or using a dynamic vibration absorber does not lead to major improvements. Taking this into account, the active suspension system can best be used to decrease the dynamic wheel load. However, the simulation results show that the dynamic wheel load, under the assumption that the ride comfort does not deteriorate, can be decreased at most by 9% with respect to the BEV-front. This means that the investigated controllers are not sufficient enough to cancel the increase in dynamic wheel load due to the placement of in-wheel motors. Other control approaches might be more sufficient and should therefore be investigated.

It is shown that the system does not consume a large amount of energy on uneven road surfaces. However, the system can also be used to eliminate the roll and pitch behavior of a vehicle during cornering or braking to obtain better cornering behavior. In order to do so the system has to apply active forces on the sprung mass and as such, the system works in motor mode. The continuous forces that have to be delivered in this case can be very high, which leads to a higher energy consumption together with higher copper losses. In order to obtain a good representation of the consumed and regenerated power during normal driving, this should also be taken into account.

The influence of limitations of the electromagnetic suspension system on the performance of the two control approaches has not been investigated thoroughly. However, it has been found that the mean forces needed to improve the ride comfort or dynamic wheel load around the intersections with the BEV-front are less than 500 N. Therefore, it is expected that increasing the maximum force of the system has no big influence on the sufficiency of the controllers.

The most logical place of the switching dynamic blocks in the Simulink models of the controllers would be behind the force limitation block. However, in this case, the actuator occasionally works in motor mode, which is not permissible for the semi-active case. Therefore, the choice is made to place them at their current position. Moreover, since the switching dynamics are places on 100 Hz, their influence is expected to be small.
Chapter 8

Conclusions and recommendations

In this chapter, the conclusions and recommendations for further research are presented. The conclusions of the previous chapters are presented in Section 8.1 and the recommendations are given in Section 8.2.

8.1 Conclusions

In this report the influence of in-wheel motors on the vehicle ride comfort has been investigated by on-road experiments using a VW Lupo 3L. The experimental results are employed to validate a full-car simulation model made with the software program SimMechanics. This model is used to investigate the influence of in-wheel motors on the ride comfort and road holding of a battery electric vehicle. Moreover it has been used to investigate possible improvements in the ride comfort and road holding by optimization of the suspension system or using an electromagnetic suspension system, controlled by either a skyhook control or a hybrid control approach. The main conclusions of this thesis are summarized below:

- A battery electric vehicle should be equipped with two electric motors that both have a continuous power of 30 Kw to be able to compete with existing vehicles in its own size and weight class. Taking a specific output of 1 kW/kg, this means that in-wheel motors will add 15 kg to each individual wheel.

- Experiments show that an increase of 15 kg of both front wheels results in a decrease in ride comfort index of 13.4%, 8.8%, 14.7% and 25.5% on smooth asphalt, highway, cobblestones and Belgian blocks. Increasing the rear unsprung mass with a similar amount results in a decrease of 31.3%, 1.1%, 7.8% and 5.9%. Since a decrease in ride comfort index of 5 to 10% is considered to be a high deterioration of the ride comfort, it can be concluded that in-wheel motors, especially at the front, have a highly negative influence on the ride comfort.

- The simulation results obtained with the full-car model show a fair resemblance with the experimental results up to the valuation frequency of 20 Hz. Although the simulations are not able to exactly predict the influence of in-wheel motors, they do give a good indication of the trends found in the ride comfort, accelerations of the wheels and the vehicle’s roll and pitch velocity. As such, the model can be considered to represent the real vehicle.

- Due to the conversion of the baseline ICE VW Lupo 3L to a battery electric vehicle, the sprung mass is increased with 160 kg. Due to this increase, the ride comfort of the electric vehicle becomes approximately 14% more comfortable than the baseline vehicle on Belgian blocks. Although an increase in unsprung mass results in a decrease of the ride comfort, the comfort of the heavier electric vehicle with front, rear or four in-wheel motors is still 1.5%, 6.3% and 5.6% more comfortable than
the baseline vehicle on Belgian blocks. However, the dynamic wheel load and suspension travel are increased by approximately 40% and 16%, respectively.

- The suspension system as implemented in the real Lupo 3L can be considered to be optimized, even for the BEV. This means that changing the parameters of the suspension system does not lead to significant improvements. A dynamic vibration absorber only leads to small improvements.

- An electromagnetic suspension system, able to deliver a continuous force of 500 N, is unable to sufficiently decrease the dynamic wheel load when controlled by a (semi-)active skyhook or hybrid controller. Without a deterioration in the ride comfort, the dynamic wheel load can only be decreased by approximately 9% compared to the electric vehicle with in-wheel motors. If a deterioration of the ride comfort is allowed, the dynamic wheel load can be decreased up to approximately 18%. However, the suspension travel can be strongly reduced, even with respect to the electric vehicle without in-wheel motors.

8.2 Recommendations

Suggestions for further research are listed below.

- The full-car simulation model already resembles the real vehicle behavior quite nicely. However, improvements can still be made. For example, the height of the center of gravity of the vehicle and the inertia can be measured instead of estimated. The McPherson and the semi-independent suspension systems and the drive train components can be modeled and the bending of the chassis can be taken into account. It would be interesting to investigate if this results in a closer agreement with the experiments.

- The influence of in-wheel motors on the comfort and road holding of a vehicle are investigated based on an increase of the unsprung mass of 30 kg. This is based on the assumption that the motors have a specific power output of 1 kW/kg. However, since in-wheel motors found in open literature still tend to have a specific output of less than 1 kW/kg, the motors can even be slightly heavier. Though, in turn, taking into account that drive shafts can be deleted and brakes can be smaller due to the regenerative braking capability of the motors, the total increase in unsprung mass can be lower. More investigation in this field can be performed.

- The influence of in-wheel motors on the ride comfort and road holding of a BEV are so far only investigated by simulations. It would be very interesting to perform on-road experiments with the real battery electric vehicle to investigate the improvements in ride comfort due to the conversion and to investigate the influence of in-wheel motors in such a vehicle. These results can than be compared to the simulation results.

- The influence of in-wheel motors on the road holding of the vehicle are only assessed by looking at the dynamic wheel load. Simulating extreme driving maneuvers like the J-turn or a lane change maneuver could possibly give more insight in the effects of in-wheel motors on the road holding of the vehicle.

- Simulations show that the investigated techniques to control a (semi-) active electromagnetic suspension system, limited to a continuous force of 500 N, are only able to decrease the dynamic wheel load up to around 9%. Since the in-wheel motors in the battery electric vehicle result in an increase of dynamic wheel load of approximately 40%, this means that the investigated controllers, in combination with this limited force, are not sufficient enough to decrease this negative effect. Therefore, the influence of the limitations of the electromagnetic suspension system on the performance of the control approaches should be investigated. Moreover, other control techniques like full state feedback control have to be examined. Since the possible improvements of a semi-active and active suspension system are only investigated using a quarter-car model, it would be also be interesting to investigate the possible improvements by using the full-car model.
8.2. Recommendations

- At the time of the investigations of the possible improvements of the (semi-) active electromagnetic suspension system, the mass of the system was unknown and is therefore not been taken into account. Since this also increases the unsprung mass and thus influences the ride comfort and road holding of the vehicle, this should be taken into account in further investigations.
Bibliography


Appendix A

Additional vehicle and tyre information

A.1 Weight distribution

The weight distribution of the VW Lupo 3L with a full fuel tank over all four wheels with and without driver and passenger are given in Table A.1.

<table>
<thead>
<tr>
<th></th>
<th>Without driver</th>
<th>With driver + passenger</th>
</tr>
</thead>
<tbody>
<tr>
<td>weight left front</td>
<td>269.0 kg</td>
<td>323.5 kg</td>
</tr>
<tr>
<td>weight right front</td>
<td>280.5 kg</td>
<td>310.5 kg</td>
</tr>
<tr>
<td>weight left rear</td>
<td>163.0 kg</td>
<td>205.5 kg</td>
</tr>
<tr>
<td>weight right rear</td>
<td>157.0 kg</td>
<td>190.5 kg</td>
</tr>
<tr>
<td>Total weight</td>
<td>869.5 kg</td>
<td>1030 kg</td>
</tr>
</tbody>
</table>

A.2 Additional damper characterization information

The damping characteristics of the Lupo dampers are determined by exciting the damper sinusoidally at different frequencies. At each frequency the damper velocity and the damper force under extension and compression is measured. An example of the results of one measurement is given in Fig. A.1. This figure shows the position, the velocity and the force of the damper when excited with a sine wave with an amplitude of 40 mm and a frequency of 2.1 Hz. The vertical dotted lines in combination with the circles indicate the velocity and force under compression. The squares indicate the velocity and force under extension. As can be seen, the velocity is maximum when the damper goes through its equilibrium, i.e. the displacement $x = 0$. Using this information, two points in the damping force characteristic plot can be determined: one under compression and one under extension. Multiple points in the damping force characteristic plot are obtained by changing the oscillation frequency.
A.3 Additional tyre parameters

This appendix gives the additional parameters of the Lupo 3L tyres that are determined by experiments on the flatplank tyre test setup available at the Department of Mechanical Engineering at the Eindhoven University of Technology. The additional parameters are the relaxation length, the cornering stiffness, the overall lateral stiffness and the pneumatic trail.

Fig.A.2 (a) and (b) show the lateral tyre force versus the traveled distance under a slip angle of 1 degree for a tyre inflation pressures of 2.4 and 3.0 bar, respectively.

The relaxation curves can be fitted with the following equation:

\[ F_y = C_{F\alpha} \alpha (1 - e^{-\frac{s}{\sigma_y}}) \]  

(A.1)

with \( C_{F\alpha} \) the cornering stiffness [N/deg.], \( \alpha \) the slip angle [deg.], \( s \) the traveled distance [m] and \( \sigma_y \) the relaxation length [m]. The fitted curves are also shown in Fig.A.2.
The relaxation length and cornering stiffness for all three vertical loads can be found in Table A.2. From these numbers it can be concluded that the relaxation length is strongly dependent on the vertical force. Furthermore, a higher tyre inflation pressure results in a slightly lower relaxation length.

### Table A.2: Relaxation length and cornering stiffness.

<table>
<thead>
<tr>
<th>tyre pressure</th>
<th>p = 2.4 bar</th>
<th>p = 3 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical load</td>
<td>relaxation length</td>
<td>cornering stiffness</td>
</tr>
<tr>
<td>1000 N</td>
<td>0.15 m</td>
<td>206 N/deg.</td>
</tr>
<tr>
<td>3000 N</td>
<td>0.41 m</td>
<td>536 N/deg.</td>
</tr>
<tr>
<td>5000 N</td>
<td>0.48 m</td>
<td>634 N/deg.</td>
</tr>
</tbody>
</table>

The cornering stiffness can be calculated using the following equation:

$$C_{F\alpha} = p_{Ky} F_{z0} (1 + p_{py} dp_i) \sin \left(2 \arctan \left(\frac{F_z}{p_{Ky} (1 + p_{py} dp_i) F_{z0}}\right)\right)$$  \hspace{1cm} (A.2)

Using the measurement results the following values are found: $p_{Ky1} = 9.2179$, $p_{Ky2} = 1.4388$, $p_{py1} = 0.3463$ and $p_{py2} = 0.9719$. The results are shown in Fig. A.3.

### Figure A.3: Cornering stiffness as function of vertical force.

The overall lateral stiffness $e_y$ of the tyre and ground contact can be estimated by:

$$e_y = \frac{C_{F\alpha}}{\sigma_y} = c_{y0} (1 + p_{cy3} dp_i)$$ \hspace{1cm} (A.3)

With $C_{F\alpha}$ the cornering stiffness [N/rad] and $\sigma_y$ the relaxation length. Using the measurement results a value of 0.2396 is found for $p_{cy3}$.

The pneumatic trail $t$ can be calculated by:

$$t = D_t = (q_{Dz1} + q_{Dz2} df_z)(1 - p_{pz1} dp_i) F_z \frac{R_0}{F_{z0}}$$ \hspace{1cm} (A.4)

with $df_z$ the non-dimensional vertical force increment calculated by:

$$df_z = \frac{F_z - F_{z0}}{F_{z0}}$$ \hspace{1cm} (A.5)
Using the measurement results the following values are found: $q_{D1} = 0.0953$, $q_{D2} = 0.0048$ and $p_{p1} = 0.9298$. The results are shown in Fig. A.4.

Fig. A.5 and Fig. A.6 give the several characteristics of the tyre.

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Figure A.4: Pneumatic trail as function of vertical force.

Figure A.5: Tyre characteristics (1).
A.3. **Additional tyre parameters**

Figure A.6: Tyre characteristics (2).
Appendix A. Additional vehicle and tyre information
Appendix B

Comfort index

The acceleration frequency weighting functions for the whole-body vibrations applied to the accelerations of the chassis in order to obtain the ride comfort index, as used for the investigations performed in this report, are shown in Fig. B.1.

Figure B.1: Acceleration frequency weightings for whole-body vibration as used for the investigation performed in this thesis.
Appendix B. Comfort index
Appendix C

Additional measurement setup information

C.1 Acceleration sensors

To measure the accelerations of the chassis, three-axis ADXL330 accelerometers with a full-scale range of ± 3g are used. To measure the roll and pitch velocity the dual-axis gyro IDG-300 sensor is used with a full-scale range of ± 500°/s. To measure the acceleration of the wheels dual-axis ADXL321 accelerometer with a full-scale range of ± 18g are used. The accelerometers are able to measure dynamic accelerations and static acceleration (gravity). All the output signals are analog voltages proportional to acceleration or angular rate.

Specifications of the ADXL330, the IDG-300 and the ADXL321 are listed in Table C.1, Table C.2 and Table C.3, respectively.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement Range</td>
<td>±3.6 g</td>
<td>g</td>
</tr>
<tr>
<td>Nonlinearity</td>
<td>±0.3 %</td>
<td>%</td>
</tr>
<tr>
<td>Package Alignment Error</td>
<td>± 1 degree</td>
<td>degree</td>
</tr>
<tr>
<td>Alignment Error</td>
<td>±0.1 degree</td>
<td>degree</td>
</tr>
<tr>
<td>Cross Axis Sensitivity</td>
<td>± 1 %</td>
<td>%</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>300 mV/g</td>
<td></td>
</tr>
<tr>
<td>Sensitivity change due to temperature</td>
<td>±0.015 %/°C</td>
<td></td>
</tr>
<tr>
<td>0 g Voltage at X_{out}, Y_{out}</td>
<td>1.5 V</td>
<td></td>
</tr>
<tr>
<td>Noise Density X_{out}, Y_{out}</td>
<td>280 μg/√Hz rms</td>
<td></td>
</tr>
<tr>
<td>Noise Density Z_{out}</td>
<td>350 μg/√Hz rms</td>
<td></td>
</tr>
<tr>
<td>Sensor Resonant Frequency</td>
<td>5.5 kHz</td>
<td></td>
</tr>
</tbody>
</table>

All sensors are delivered on a fully assembled and tested breakout board. These boards are equipped with a 0.1μF filtering capacitor to implement low-pass filtering for anti-aliasing and noise reduction. With this filter capacitor selection the bandwidth of the ADXL sensors are placed at 50 Hz. This means that accelerations at frequencies above 50 Hz are attenuated.
Appendix C. Additional measurement setup information

Table C.2: Specifications of the IDG-300.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement Range</td>
<td>± 500</td>
<td>°/s</td>
</tr>
<tr>
<td>Nonlinearity</td>
<td>&lt; 1</td>
<td>% of FS</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>2.0</td>
<td>mV/°/s</td>
</tr>
<tr>
<td>0 g Voltage</td>
<td>1.5</td>
<td>V</td>
</tr>
<tr>
<td>High Frequency cutoff</td>
<td>140</td>
<td>Hz</td>
</tr>
<tr>
<td>Resonant Frequency x-axis</td>
<td>12</td>
<td>kHz</td>
</tr>
<tr>
<td>Resonant Frequency y-axis</td>
<td>15</td>
<td>kHz</td>
</tr>
<tr>
<td>Resonant Frequency z-axis</td>
<td>3</td>
<td>kHz</td>
</tr>
<tr>
<td>Noise Density</td>
<td>0.014</td>
<td>°/s/√Hz</td>
</tr>
</tbody>
</table>

Table C.3: Specifications of the ADXL321.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement Range</td>
<td>±18</td>
<td>g</td>
</tr>
<tr>
<td>Nonlinearity</td>
<td>±0.2</td>
<td>%</td>
</tr>
<tr>
<td>Package Alignment Error</td>
<td>±1</td>
<td>degree</td>
</tr>
<tr>
<td>Alignment Error</td>
<td>±0.1</td>
<td>degree</td>
</tr>
<tr>
<td>Cross Axis Sensitivity</td>
<td>±2</td>
<td>%</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>57</td>
<td>mV/g</td>
</tr>
<tr>
<td>Sensitivity change due to temperature</td>
<td>0.01</td>
<td>%/°C</td>
</tr>
<tr>
<td>0 g Voltage at X_{out}, Y_{out}</td>
<td>1.5</td>
<td>V</td>
</tr>
<tr>
<td>Noise Density</td>
<td>320</td>
<td>μg/√Hz rms</td>
</tr>
<tr>
<td>Sensor Resonant Frequency</td>
<td>5.5</td>
<td>kHz</td>
</tr>
</tbody>
</table>

C.2 Parallax GPS receiver module

To track the road during the experiments and to determine the vehicle forward velocity, a parallax GPS receiver module is used. This module transmits standard National Marine Association (NMEA) 0183 v2.2 strings, which is developed for data communications between marine instruments. The position accuracy of the module is approximately 5 meters and the vehicle velocity accuracy is approximately 0.1 m/s. The communication between satellites consists of a 4800bps, 8 data bits, no parity, 1 stop bit, non-inverted data stream. An example of a RMC string is given by:

\$GPRMC,090415,A,5126.8767,N,00529.3586,W,145.0,231.8,030909,004.2,W*70

with

090415 Fix taken at 09:04:15 UTC
A Status A=active or V=Void
5126.8767 Latitude 51 deg 26.8767' N
00529.3586 Longitude 5 deg 29.3586' W
145.0 Speed over ground in knots
231.8 Track angle in degrees
030909 Date - 2nd of September 2009
004.2 Magnetic Variation
W*70 Checksum data

An example of the output velocity is given in Fig. C.1.
The dSpace interface consists of a signal conditioner and an analog to digital converter. The conditioning system consists of a butterworth filter and an electromagnetic interference\(^1\) suppression filter. The analog to digital converter changes the analog input signal to a digital signal. The A/D converter has the following characteristics:

- 16-bit resolution
- \(\pm 10\) V input voltage range
- \(\pm 5\) mV offset error
- \(\pm 0.25\%\) gain error
- > 80 dB signal-to-noise ratio (SNR, ratio of a signal power to the noise power corrupting the signal)

The resolution of the converter is given by:

\[
\frac{20}{2^{16}} = 0.305\text{ mV}
\]

(C.1)

The resolution of the ADXL330 sensors become:

\[
\frac{20}{2^16} = 1.13 \cdot 10^{-3} \text{ g}
\]

(C.2)

and the resolution of the ADXL321 becomes:

\[
\frac{20}{2^16} = 5.3 \cdot 10^{-3} \text{ g}
\]

(C.3)

The anti-aliasing butterworth filter prevents high frequencies, in either the signal or noise, from introducing distortion into the digitized signal. The bode diagram of the butterworth filter is shown in Fig. C.2. As can be seen, the cutoff frequency is placed at 500 Hz, meaning that all frequencies above 500 Hz are attenuated.

To prevent aliasing a sample rate more than twice a signal’s highest frequency should be chosen. Therefore a sample frequency of 1000 Hz is chosen for the experiments.

---

\(^1\)electrical disturbance, interference or noise, in a system due to natural phenomena, low-frequency waves from electromechanical devices or high-frequency waves from ICs and other electronic devices
Figure C.2: Butterworth filter as used in the conditioning system.
Appendix D

Additional validation information

D.1 Road profiles

Fig. D.1 shows the displacement power spectral density of the four different road profiles as used for the simulations in this thesis. The road frequency $n$ is given by:

$$ n = \frac{1}{\lambda} = \frac{f}{V} \quad (D.1) $$

with $\lambda$ the wavelength [m], $f$ the frequency and $V$ the forward velocity of the vehicle [m/s]. The figure clearly shows a big decrease in road displacement at a wavelength of 0.33 m, which at a forward velocity of 50 km/h gives a frequency of around 42 Hz.

![Figure D.1: Displacement power spectral density of the road profiles as used for the simulations.](image)

D.2 Shift in eigenfrequency

Fig. D.2 (a) and (b) show the shift in eigenfrequencies of the front and rear wheel due to the added mass, respectively, found by experiments and simulations. The figures clearly show that the shifts found by
simulations are the same as found by the experiments.

![Figure D.2: Shift in eigenfrequency of the unsprung mass due to added mass.](image)

### D.3 Influence of in-wheel motors in the frequency domain

Fig.D.3 shows the PSD of the vertical chassis acceleration for all vehicle configurations, obtained by simulations. The differences noticeable are the same as found for the experiments.

![Figure D.3: PSD of the vertical acceleration of the chassis on Belgian blocks for all vehicle configurations found by simulations.](image)
Appendix E

Control performance tables

Table E.1 provides the ride comfort index, dynamic wheel load and suspension travel of the quarter BEV-front with different values of the passive damping constant.

<table>
<thead>
<tr>
<th>passive damping constant [Ns/m]</th>
<th>300</th>
<th>500</th>
<th>700</th>
<th>900</th>
<th>1100</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_{\text{overall}}$ (RMS) [m/s²]</td>
<td>1.145</td>
<td>1.058</td>
<td>1.079</td>
<td>1.154</td>
<td>1.248</td>
</tr>
<tr>
<td>$dFz$ (RMS) [N]</td>
<td>1628</td>
<td>1339</td>
<td>1169</td>
<td>1065</td>
<td>999</td>
</tr>
<tr>
<td>$dz$ (RMS) [mm]</td>
<td>18.9</td>
<td>15.3</td>
<td>13.3</td>
<td>11.9</td>
<td>11.0</td>
</tr>
</tbody>
</table>

The absolute values of the ride comfort index, dynamic wheel load and suspension travel found at the encircled points using semi-active and active skyhook control (see Fig.7.9) are provided in Table E.2 and Table E.3, respectively. The values found using the semi-active and active hybrid controller (see Fig.7.10) are given in Table E.4 and Table E.5, respectively.

Table E.2: Absolute values and relative difference compared to the baseline for the comfort optimized and dynamic wheel load optimized case using semi-active skyhook control.

<table>
<thead>
<tr>
<th>Passive damping [Ns/m]</th>
<th>Skyhook damping [Ns/m]</th>
<th>Quarter BEV-front</th>
<th>Comfort improvement (1)</th>
<th>Relative difference</th>
<th>Wheel load improvement (2)</th>
<th>Relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>non-lin.</td>
<td>700</td>
<td>5000</td>
<td>-</td>
<td></td>
<td>900</td>
<td>-</td>
</tr>
<tr>
<td>0</td>
<td>1.212</td>
<td>1.047</td>
<td>- 13.6 %</td>
<td>-</td>
<td>1.158</td>
<td>- 4.5 %</td>
</tr>
<tr>
<td>1098</td>
<td>1108</td>
<td>+ 0.9 %</td>
<td></td>
<td></td>
<td>1037</td>
<td>- 5.6 %</td>
</tr>
<tr>
<td>12.6</td>
<td>9.2</td>
<td>- 27.0 %</td>
<td></td>
<td></td>
<td>9.0</td>
<td>- 28.6 %</td>
</tr>
</tbody>
</table>
Table E.3: Absolute values and relative difference compared to the baseline for the comfort optimized and dynamic wheel load optimized case using active skyhook control.

<table>
<thead>
<tr>
<th>Quarter BEV-front</th>
<th>Comfort improvement (1)</th>
<th>Relative difference</th>
<th>Wheel load improvement (2)</th>
<th>Relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Passive damping [Ns/m]</td>
<td>non-lin.</td>
<td>900</td>
<td>-</td>
<td>1100</td>
</tr>
<tr>
<td>Skyhook damping [Ns/m]</td>
<td>5000</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$a_{overall}$ (RMS) [m/s²]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>dFz (RMS) [N]</td>
<td>1.212</td>
<td>1.059</td>
<td>-12.7%</td>
<td>1.214</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>1098</td>
<td>1107</td>
<td>+0.8%</td>
<td>999</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>12.6</td>
<td>9.2</td>
<td>-27.0%</td>
<td>9.1</td>
</tr>
</tbody>
</table>

Table E.4: Absolute values and relative difference compared to the baseline for the comfort optimized and dynamic wheel load optimized case using semi-active hybrid control.

<table>
<thead>
<tr>
<th>Quarter BEV-front</th>
<th>Comfort improvement (1)</th>
<th>Relative difference</th>
<th>Wheel load improvement (2)</th>
<th>Relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>skyhook / groundhook ratio (α) [-]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>passive damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Skyhook damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Groundhook damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$a_{overall}$ (RMS) [m/s²]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>dFz (RMS) [N]</td>
<td>1.212</td>
<td>1.067</td>
<td>-12.0%</td>
<td>1.212</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>1098</td>
<td>1098</td>
<td>-0.0%</td>
<td>1015</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>12.6</td>
<td>9.6</td>
<td>-23.8%</td>
<td>9.3</td>
</tr>
</tbody>
</table>

Table E.5: Absolute values and relative difference compared to the baseline for the comfort optimized and dynamic wheel load optimized case using active hybrid control.

<table>
<thead>
<tr>
<th>Quarter BEV-front</th>
<th>Comfort improvement (1)</th>
<th>Relative difference</th>
<th>Wheel load improvement (2)</th>
<th>Relative difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>skyhook / groundhook ratio (α) [-]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Passive damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Skyhook damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Groundhook damping [Ns/m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$a_{overall}$ (RMS) [m/s²]</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>dFz (RMS) [N]</td>
<td>1.212</td>
<td>1.091</td>
<td>-10.0%</td>
<td>1.212</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>1098</td>
<td>1098</td>
<td>-0.0%</td>
<td>1026</td>
</tr>
<tr>
<td>dz (RMS) [mm]</td>
<td>12.6</td>
<td>9.2</td>
<td>-27.0%</td>
<td>9.0</td>
</tr>
</tbody>
</table>
Appendix F

AVEC 10 paper

The paper submitted for the 10th International Symposium on Advanced Vehicle Control, organized in August 22-26 2010, is provided on the next page.
Influence of in-wheel motors on the ride comfort of electric vehicles

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An in-wheel electric motor is an interesting and innovative configuration for an electric vehicle, that compared to central motors offers benefits in the field of space, control and efficiency. However, it also increases the weight of the unsprung mass, which results in a decrease in vehicle ride comfort and safety. In order to investigate the severity of these disadvantages, on-road experiments are performed with an ICE vehicle and simulations are performed with a model representing a battery electric vehicle. As the largest disadvantage is found to be the increase in dynamic wheel load, possible improvements using an active electromagnetic suspension system are investigated also.

1. INTRODUCTION

In recent years, the electric vehicle has gained substantial interest as a possible solution to the environmental and energy problems caused by the conventional internal combustion engine (ICE) vehicles. Within the class of electric vehicles, several propulsion configurations can be adopted. One of the most popular configurations is the centralized motor drive with reduction gears and differential due to its similarity with existing systems. However, the in-wheel configuration, in which the drive motors are mounted inside the wheels, appears to be an interesting and innovative configuration that offers several benefits compared to the centralized configuration. It gives the opportunity to provide torque to each wheel independently, which offers significant potential to improve the vehicle stability. The transmission, drive shafts, differentials and supporting systems become redundant, resulting in a high overall vehicle efficiency. Moreover, more interior space is created due to the dense packaging.

All these advantages lead to the fact that research within the field of in-wheel motor design and control is performed extensively [1, 2, 3, 4]. Several companies are thereby engaged in the fabrication of in-wheel motors. For example, Michelin has designed the Active Wheel Drive in-wheel motor [5], Siemens has designed the eCorner [6] and Bridgestone has invented the Dynamic-Damping In-wheel Motor [7].

However, in-wheel motors do have one major disadvantage: they increase the vehicle unsprung mass. This results in a decrease in ride comfort and road holding capability (safety) and in an increase in suspension travel. Although this disadvantage is often briefly noted in many papers, extensive research within this specific field has not been reported in the open literature. Since passenger comfort and safety is an ever increasing demand in the automotive industry, the knowledge of the severity of the disadvantages will play an important part in the successful implementation of in-wheel motors in future electric vehicles.

Therefore, the goal is to investigate the severity of the negative effects of an increased unsprung mass, with a main focus on the ride comfort. To this end, on-road experiments are performed using an ICE vehicle and simulations are performed with a validated model. The model is further employed to investigate the negative effects in a battery electric vehicle (BEV) and to investigate possible improvements in ride comfort and safety using an active suspension system.

The experiments and simulations are performed by adding 15 kg to each individual wheel to emulate the mass of the in-wheel motors. This results from the assumption that the vehicle will be equipped with in-wheel motors with a combined continuous power of 30 kW and a specific motor output of 1 Kw/kg, which is considered to be an appropriate guideline for permanent magnet brushless DC motors [8].

The paper is organized as follows. The experimental setup is introduced first. Hereafter, informa-
tion about the simulation model is given, followed by the description of the model validation. The model is then used to investigate the negative effects of an increased unsprung mass in a BEV and to analyse the possible improvement in ride comfort and safety using an active electromagnetic suspension system.

2. Experimental setup

The experiments are performed using a VW Lupo 3L. This vehicle is currently being converted into a BEV at the Eindhoven University of Technology (TU/e) to gain practical experience in the field of electric vehicles. The experiments with the original ICE vehicle are conducted with three configurations: the unmodified configuration, a configuration with an increased unsprung mass at the front, and a configuration with an increased unsprung mass at the rear. In this paper, these configurations will be addressed by 'baseline', 'iw-front' and 'iw-rear', respectively. To emulate the stator and rotor of the in-wheel motors, the vehicle is modified by attaching a stationary part to the suspension system and a rotating part directly to the wheels. Both parts have a mass of 7.5 kg. As example, the iw-front modification is shown in Fig. 1.

The test vehicle is equipped with a GPS module to determine the traveled route and forward velocity, with five acceleration sensors and with one dual-axis gyro sensor, all connected to a dSpace data acquisition system. The sensors, placed in the vehicle as shown in Fig. 2, measure the following signals during the experiments:

- **Sensor 1**: longitudinal, lateral and vertical acceleration of the chassis
- **Sensors 2 and 3**: vertical acceleration of the front and rear wheel
- **Sensors 4 and 5**: vertical acceleration of the top of the suspension spring at the front and rear
- **Sensor 6**: roll and pitch velocity of the chassis

The dynamic wheel load and suspension travel are not measured during the experiments. The measurements are performed twice on four different road surface types with ascending severity: smooth asphalt, highway, cobblestones and Belgian blocks. An impression of these roads is provided in Fig. 3. These four road types are all found on open road in the neighborhood of Eindhoven. The vehicle forward velocity is 50 km/h on all roads, except for the 120 km/h on the highway. To minimize the influence of the vibrational disturbances of the engine, all results are evaluated up to the engine’s first disturbance frequency of 20 Hz (at 50 km/h).

3. Vehicle model

In order to investigate the severity of the negative effects of the increase in unsprung mass in a BEV, simulations are performed with the multi-body toolbox Matlab/SimMechanics using a full-car model as illustrated in Fig. 4. It consists of four unsprung bodies connected to a sprung body by a vertical spring-damper system. The sprung body has 6 degrees of freedom (longitudinal, lateral, vertical, roll, pitch, yaw) and each corner module has one vertical degree of freedom. Each unsprung body is connected to the MF-Swift 6.1.0 tyre model as developed by TNO Automotive. In this model, the tyre forces and moments are described by the Magic Formula. Furthermore, the model contains an anti-roll bar at the front and at the rear.
The input parameters of the model are provided in Table 1. These parameters are all derived from the VW Lupo 3L to represent this vehicle as accurate as possible. The track width and wheelbase are provided by the manufacturer. The total vehicle mass, mass distribution, stiffness of the anti-roll bars, the spring and damper characteristics of the front and rear suspension systems and the tyre parameters are all established by measurements. The height of the center of gravity and the inertia of the vehicle are estimated based on the rules of thumb according to the National Highway Traffic Safety Administration database.

The road profile height used as input in the simulations is a measured road profile, which is scaled such that it mimics the severity of each of the four experimental roads.

### Table 1: Parameters of the VW Lupo 3L.

<table>
<thead>
<tr>
<th>Definition</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Track width</td>
<td>(w)</td>
<td>1.425 m</td>
</tr>
<tr>
<td>Vehicle mass unloaded</td>
<td>(m)</td>
<td>880 kg</td>
</tr>
<tr>
<td>Cargo</td>
<td>(m_c)</td>
<td>150 kg</td>
</tr>
<tr>
<td>Distance CG to front wheels</td>
<td>(a)</td>
<td>0.892 m</td>
</tr>
<tr>
<td>Distance CG to rear wheels</td>
<td>(b)</td>
<td>1.429 m</td>
</tr>
<tr>
<td>Stiffness anti-roll bar front</td>
<td>(c_{of})</td>
<td>1800 N/m(\text{rad})</td>
</tr>
<tr>
<td>Stiffness anti-roll bar rear</td>
<td>(c_{or})</td>
<td>800 N/m(\text{rad})</td>
</tr>
<tr>
<td>Height of CG</td>
<td>(h)</td>
<td>0.55 m</td>
</tr>
<tr>
<td>Moment of inertia x-axis</td>
<td>(I_{xx})</td>
<td>290 kg(\text{m}^2)</td>
</tr>
<tr>
<td>Moment of inertia y-axis</td>
<td>(I_{yy})</td>
<td>1120 kg(\text{m}^2)</td>
</tr>
<tr>
<td>Moment of inertia z-axis</td>
<td>(I_{zz})</td>
<td>1250 kg(\text{m}^2)</td>
</tr>
</tbody>
</table>

### 4. Experimental and simulation results

In this section, the experimental and simulation results are assessed on the basis of the passengers ride (dis)comfort. This ride (dis)comfort during driving results are assessed on the basis of the passengers ride comfort index, calculated by:

\[
a_{\text{overall}} = \sqrt{a_x^2 + a_y^2 + a_z^2} \tag{1}
\]

with \(a_x\), \(a_y\) and \(a_z\) the frequency weighted\(^1\) root mean square (RMS) value of the longitudinal, lateral and vertical accelerations, respectively. A reduction of this value of 5 to 10% is usually considered to be a respectable improvement\([9]\).

Before the assessment is performed, two examinations have to be carried out. First, the repeatability of each measurement is investigated. It has been found that the differences in the overall ride comfort index between two repeated measurements are below 5% for all roads, except for on the smooth asphalt. On this road the results can differ up to 30%, and therefore these results have to be handled carefully. Second, the measured eigenfrequencies of the wheels are employed to determine the weight of the unsprung mass. Since the eigenfrequency of both the front and rear wheels are found at a relative high frequency of 18.5 Hz, the unsprung mass at the front and rear are approximated to have a relatively low value of 35 kg each. This low mass results especially from the light weight of the magnesium rims: one rim and tyre together weigh only 9.5 kg. Proof of the location of this eigenfrequency and the similarity between experimental and simulation results using this approximation are shown in the power spectral density (PSD) plot in Fig. 5. Only at frequencies below 1 Hz and above 30 Hz small differences exists, which are caused by the difference in road input. Due to the added weight of 15 kg to each unsprung mass, the eigenfrequency of the wheel shifts to 13.5 Hz. This result is also reproduced by the simulations.

![Fig. 5: PSD of the vertical acceleration of the front wheels. (50 km/h, Belgian blocks)](image)

Next, the vertical accelerations of the chassis found by experiments and simulations are investigated and compared with each other in the frequency domain. An example of the results found on Belgian blocks is shown in Fig. 6.

![Fig. 6: PSD of the vertical acceleration of the chassis. (50 km/h, Belgian blocks)](image)

This figure indicates that there is a high similarity between the experimental and simulation results up to at least the valuation frequency of 20 Hz. Both results thereby show both the eigenfrequency of the sprung and unsprung bodies as well as the effect of wheelbase filtering at around the same frequencies. The differences in magnitude of these locations are due to differences in vehicle parameters, differences in the suspension system and due to the fact that drive train components are not modeled. The small difference in locations of the peaks and null points are due to differences in the vehicle forward velocity. The differences visible above 20 Hz are probably

---

\(^1\)On the basis of ISO 2631:1997 and BS 6841:1987 guidelines
due to the difference in road input and due to the fact that the first bending mode of the vehicle is not taken into account in the vehicle model. The similarity of the accelerations of the chassis in longitudinal and lateral direction between the experiments and measurements, although not shown in this paper, are less good than the vertical accelerations. The differences are probably due to small accelerations and de-accelerations and lateral movements during cornering and traffic avoidance maneuvers during the experiments. However, around the frequency area of importance for the ride comfort, between 1 and 2 Hz, they do look similar.

The absolute values of the overall ride comfort index found by experiments and simulations for all three configurations and for the specified roads are provided in Fig. 7 and in Table 2. The relative differences between the configurations are also provided in this table.

It can be concluded that in-wheel motors do indeed result in an increase in overall ride comfort index and thus in a deterioration of the ride comfort. Both the experimental and simulation results show that this negative effect is clearly more severe for the iw-front than for the iw-rear. Overall, the negative effects increase as the severity of the road is increased.

Based on these results, it can be concluded that front or rear in-wheel motors will definitely be perceptible for the vehicle occupations on cobblestones and Belgian blocks. These conclusions also comply with the subjective feelings of the driver and passenger during the experiments: on cobblestones and especially on Belgian blocks the presence of the masses were definitely perceptible. The increase in unsprung mass at the front was experienced to be the worst.

According to the results described above it can be concluded that, although the relative differences found by simulations do not exactly match those found by experiments, the simulations are able to give a fair approximation of the effect of in-wheel motors on the vehicle ride comfort. The basic simulation model is thus validated and can be used for further investigations. Taking this into account, the model is used to give an idea of the magnitude of the dynamic wheel load (dFz) and suspension travel (dz) on different roads, as these are not measured during the experiments. The results are shown in Fig. 9.

5. Battery electric vehicle

To this point, all investigations are performed on the basis of an ICE vehicle. However, by the conversion of the ICE Lupo 3L into a BEV, the vehicle unloaded mass is increased with 160 kg to 1040 kg and the weight distribution is shifted from 62/38 to 55/45. These changes are caused by the placement of battery packs with a combined energy of 27 kWh, weighing 273 kg in particular.

This change in mass, weight distribution and inertia has an effect on the ride comfort index, dynamic

<table>
<thead>
<tr>
<th>Road type</th>
<th>Baseline</th>
<th>Baseline — front wheel</th>
<th>Baseline — rear wheel</th>
<th>Experiments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth asphalt</td>
<td>0.099 [m/s²]</td>
<td>+13.4%</td>
<td>+31.3%</td>
<td></td>
</tr>
<tr>
<td>Highway</td>
<td>0.402 [m/s²]</td>
<td>+8.8%</td>
<td>+1.1%</td>
<td></td>
</tr>
<tr>
<td>Cobblestones</td>
<td>0.611 [m/s²]</td>
<td>+14.7%</td>
<td>+7.8%</td>
<td></td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>1.142 [m/s²]</td>
<td>+25.5%</td>
<td>+5.9%</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Road type</th>
<th>Baseline</th>
<th>Baseline — front wheel</th>
<th>Baseline — rear wheel</th>
<th>Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth asphalt</td>
<td>0.078 [m/s²]</td>
<td>+11.0%</td>
<td>+6.6%</td>
<td></td>
</tr>
<tr>
<td>Highway</td>
<td>0.370 [m/s²]</td>
<td>+7.7%</td>
<td>+2.5%</td>
<td></td>
</tr>
<tr>
<td>Cobblestones</td>
<td>0.570 [m/s²]</td>
<td>+9.6%</td>
<td>+6.8%</td>
<td></td>
</tr>
<tr>
<td>Belgian blocks</td>
<td>1.132 [m/s²]</td>
<td>+11.5%</td>
<td>+8.0%</td>
<td></td>
</tr>
</tbody>
</table>
wheel load and suspension travel. This effect is displayed in Table 3 for the Belgian blocks. On this road, the ride of the BEV becomes around 14% more comfortable than the baseline ICE vehicle, while the dynamic wheel load and suspension travel only increase slightly.

The increase in comfort of 14% due to the conversion, provides more room to deal with the negative effects of the in-wheel motors. Therefore, the influence of in-wheel motors in a BEV on the ride comfort, dynamic wheel load and suspension travel are determined for the configurations with in-wheel motors in front (BEV-front), in-wheel motors in the rear (BEV-rear) and in-wheel motors in all four wheels (BEV-four). In the case of the BEV-four, only 7.5 kg is added to each wheel. The results are demonstrated in Fig. 10. The relative differences of these configurations compared to the baseline ICE vehicle are provided in Table 3.

From these results, it can be concluded that the ride comfort index increases in particular for the in-wheel motors, as is also the case as found for the ICE vehicle. However, it can be seen that this increase does not lead to a less comfortable drive compared to the baseline ICE vehicle. Unfortunately, the dynamic wheel load does increase significantly. The average wheel load, as stated in Table 3, increases with around 20%, which means that the dynamic wheel load of the wheels equipped with in-wheel motors increase with around 40% for the BEV-front and BEV-rear.

Simulation results show that the suspension system as implemented in the real Lupo 3L can be considered to be optimized, even for the BEV. This means that modification of the suspension parameters does not lead to a significant reduction of the dynamic wheel load. Changing the tyre pressure or using a dynamic vibration absorber are also found to work insufficient. For more information the reader is referred to [10]. Because of these reasons, possible improvements using an active suspension system are explored and described next.

6. Active suspension control

The possible improvements by an active suspension system are investigated based on the use of an electromagnetic system that is currently being designed by Gyssen et al. at the department of Electrical Engineering at the TU/e [11]. This system consists of a brushless tubular permanent magnet actuator in parallel with a conventional coil spring.

The system is able to supply forces but can also regenerate energy when it works in damper mode to absorb road vibrations. The amount of force it can supply or regenerate has been found to be 755 N using a 2-D finite element analysis. To guarantee a safe ride when power breakdown occurs, the system also has a certain amount of passive damping. In the investigation performed in this paper, the passive damping is 900 Ns/m.

The system is controlled using a semi-active and active control approach. In this paper only the active approach is described. For more information about the semi-active approach the reader is referred to [10]. The control technique used attempts to merge the performance benefits of the skyhook and groundhook control and is known as hybrid control [12]. The force that is delivered by this controller is given by:

\[ F_h = \alpha F_{sky} + (1 - \alpha) F_{gnd} \]  

with \( \alpha \) the relative ratio between the skyhook force \( F_{sky} \) and the groundhook force \( F_{gnd} \). By changing the value of \( \alpha \), the control policy amplifies either skyhook or groundhook control. In the case of active control, the skyhook and groundhook forces are given by:

\[ F_{sky} = -d_{sky} \cdot \dot{z}_s \]  
\[ F_{gnd} = -d_{gnd} \cdot \dot{z}_a \]

with \( d_{sky} \) and \( d_{gnd} \) the skyhook and groundhook damping constant, respectively, and \( \dot{z}_s \) and \( \dot{z}_a \) the vertical velocity of the sprung and unsprung mass, respectively.

The ride comfort index, dynamic wheel load and suspension travel are determined for an \( \alpha \) ranging...
between 0 and 1. The value of the skyhook and groundhook damper are chosen such that the lowest ride comfort index is found when only pure skyhook control is considered ($\alpha = 1$) and the lowest dynamic wheel load is found when only pure groundhook control is considered ($\alpha = 0$). The dynamic wheel load versus the ride comfort index, and the suspension travel versus ride comfort index are given in Fig. 11 a and b, respectively. The values found for the BEV-front are also implemented in the figures by vertical and horizontal lines for clarity of the possible improvements. Assuming that an improvement in one area is not accompanied with a deterioration in an other area, either the ride comfort index can be reduced with about 10% or the dynamic wheel load can be reduced with about 7%. The suspension travel is decreased with respect to the BEV-front.

Without the above assumption it is possible to decrease the dynamic wheel load with about 18% or to decrease the ride comfort index with about 45%, just by changing the value $\alpha$. Though, this is accompanied by an increase in ride comfort index of 40% or an increase in dynamic wheel load of 38%, respectively. Nevertheless, the first setting can for example be used to enhance the vehicle behavior during cornering and the second setting can be used to enhance the comfort during straight driving.

Similar results can be found with the semi-active control approach, though it is less able to emphasize one specific aspect like comfort or dynamic wheel load.

Overall it can be concluded that, since the dynamic wheel load can only be decreased with merely 18%, the hybrid controller is not sufficient enough to diminish the increase in dynamic wheel load caused by the in-wheel motors. Therefore, other control approaches may have to be investigated.

Since the available energy in a BEV is scarce, an active suspension system should only be placed if it does not consume a lot of energy. Taking into account the charge/discharge efficiency of the batteries and the bi-directional inverter and the energy losses due to copper losses, it is calculated that one system on average uses around 15 W on Belgian blocks. For less severe roads, the amount of power consumption is even less. Hence, the system does not consume a lot of energy and may be a feasible solution for a BEV.

### 7. Conclusions

In this paper the influence of in-wheel motors on the ride comfort of vehicles has been presented.

Experiments on open road with an ICE vehicle showed that motors in the front wheels decrease the ride comfort between 10% and 25%, depending on the severity of the road. Motors in the rear wheels decrease the ride comfort only between 1% and 8%.

Using a validated model, it is shown that due to the placement of heavy battery packs, the ride comfort of a BEV is 14% increased with respect to an ICE vehicle. The dynamic wheel load and suspension travel increase only slightly. A 160 kg heavier BEV with in-wheel motors has the same ride comfort as the original ICE vehicle. However, the motors do increase the dynamic wheel load up to 40%.

An active hybrid controlled electromagnetic suspension system is able to diminish the dynamic wheel load from 40% to around 20%. Since this is not enough to guarantee the safety of the vehicle, other control approaches have to be investigated.

### References


