Eindhoven University of Technology

MASTER

Analysis and synthesis of hybrid truck energy management

Arts, G.J.C.M.

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Analysis and Synthesis of Hybrid Truck Energy Management

G.J.C.M. Arts
DCT 2007-081

Master’s thesis

Coach(es): dr.ir. A.F.A Serrarens
Supervisor: prof.dr.ir. M. Steinbuch

Technische Universiteit Eindhoven
Department Mechanical Engineering
Dynamics and Control Technology Group

Eindhoven, August, 2007
Preface

The drive for people to overcome their limitations has always been present; the need for a solution for people to move fast from one place to another dates back ages ago. With the invention of the steam engine by James Watt in 1765 the basis of the first self powered vehicle, actually built in 1769 by Nicholas Joseph Cugnot, was created. This vehicle had such excessive weight that it could therefore only be used on a road made of steel rails. From that moment many attempts were made to develop a vehicle that did not need these rails. Since then until the late 19th century steam cars dominated the automotive landscape.

As an alternative on the heavy steam cars, electric propulsion started to become available from 1839 when Robert Anderson built the first electric vehicle. Electricity was known since 1646 already and the first battery like device is said to be invented back in 1651. The basic technology for such vehicles was thus already available, long before the inventory of the steam engine. Because of the poor driving speed and range of electric vehicles, an alternative power source was found with the invention of the first practical gas engine, patented by Etienne Lenoir in 1860. As a disadvantage internal combustion powered cars were quite unreliable at that time, however the action radius and the performance was much better.

By combining both the advantages of the reliability of an electric vehicle, the driving range, and speed of an internal combustion powered car, the series hybrid electric drivetrain came into existence already in 1898. One of the first vehicles equipped with this technology was Ferdinand Porsche’s hybrid. This vehicle uses an internal combustion engine to spin a generator that provides power for electric motors located in the wheel hubs. Two years later, in the year 1900, a Belgian car maker named Pieper introduced his “voiturette” in which a small gasoline engine was mated to an electric motor under the seat. When the car was at cruising speed its electric motor operated as a generator, recharging the batteries. When the car was climbing a grade, the electric motor, which was mounted coaxially with the gas engine, gave it a boost. This concept is patented [EspaceNet, 1906] as the first parallel hybrid drivetrain and became commercially available from 1906 till 1912. At that time American car companies made 1681 steam, 1575 electric and 936 gasoline cars. In a poll conducted at the first National Automobile Show in New York City (1900), people favoured electric powered cars as their first choice, followed closely by steam cars.

In 1904 Henry Ford overcame the challenges posed by gasoline powered cars (noise, vibration, and odour) and began assembly-line production of low-priced, lightweight, gas-powered vehicles. With his invention Ford made cars accessible for ordinary people, and from that time the electric vehicle slowly disappeared from the streets. With the advent of the self-starter in 1913, steamers and electrics were almost completely wiped out. In this year, sales of electric cars dropped to
6000 vehicles, while Ford sold 30 times as many Model T gasoline cars. The smaller and less expensive automobile, with an internal combustion engine provided a new avenue of interest that was much more personal than the steam engine car. Driving an automobile required mechanical skill and special clothing including hat, gloves, duster coat, goggles and boots. Everyone in town knew who owned what car and they soon became each individual's token of identity. Compared to gasoline cars, hybrid cars on the market were expensive and difficult to service, which resulted in a fall for those vehicles as well. The period from 1920 till 1965 happened to be a dormant period for mass-produced electric and hybrid cars. So-called alternative cars became the province of backyard thinkers and small-time entrepreneurs. Meanwhile, internal combustion powered vehicles became more and more reliable and powerful.

Yet in 1966 the U.S. Congress introduced the first bills recommending the use of electric vehicles as a means of reducing air pollution. Nowadays, hybrid and electric vehicles are introduced for both the decrease of fuel consumption and emissions. The time has come to make a change in powering vehicles; (hybrid) electric vehicles do have the potential! Concluding from a short lesson in history and the success of Ford's vision of car manufacturing, this technology should therefore be made affordable and accessible for anyone. Hopefully, this report can contribute to that.

source: [Berman, 2007], [Blinkhorne, 2007], [Bottorff, 2006]
Figure 1: 'electric powered vehicles will also become common sense'
Contents

Preface 2
Summary 11
Samenvatting 13

1 Introduction 15

2 Heavy Duty Hybrids 17
  2.1 Hybrid drivetrains 17
  2.1.1 Hybrid drivetrain functionality 17
  2.1.2 Hybrid drivetrain topologies 18
  2.1.3 Energy accumulators 22
  2.1.4 Hybridisation rate 22
  2.2 Hybrid drivetrains used in heavy duty vehicles 22
  2.3 Electric energy accumulators 23
  2.4 Duty classes 26
  2.5 Drivetrain configuration versus duty class 26
  2.6 Fuel consumption improvement 30
  2.7 Additional cost for hybridisation 30
  2.8 Hybrid drivetrain developers and manufacturers 30
  2.9 Conclusions 32
  2.10 Final word 34

3 Energy Management Strategies 35
  3.1 Introduction 35
  3.2 Problem formulation 36
  3.3 Accumulator energy flow control 36
  3.4 Drivetrain energy flow control 37
  3.4.1 Intelligence control 37
  3.4.2 Advanced control 38
  3.4.3 Optimisation algorithms 40
  3.5 Combining methods 43
  3.6 Analysis versus synthesis 43
  3.7 Conclusions 44
## 4 Vehicle Modelling

4.1 Introduction ................................................................. 45
   4.1.1 Reference vehicle .................................................. 45
   4.1.2 Hybrid vehicle ..................................................... 45
4.2 Power flow ................................................................. 46
4.3 Component models ....................................................... 47
   4.3.1 General vehicle components ...................................... 47
   4.3.2 Conventional drive unit ......................................... 52
   4.3.3 Hybrid drive unit ................................................. 53
   4.3.4 Non-vehicle related parameters ................................ 59
4.4 Drive cycles ............................................................... 60
4.5 Vehicle models and supervisory control ............................ 63
   4.5.1 Reference vehicle model ......................................... 63
   4.5.2 Reference vehicle supervisory control ......................... 64
   4.5.3 Reference vehicle validation .................................... 67
   4.5.4 Simulation results reference vehicle ......................... 69
   4.5.5 Hybrid vehicle model ............................................ 70
   4.5.6 Hybrid vehicle supervisory control ........................... 70
4.6 Conclusions ............................................................... 72
4.7 Recommendations ....................................................... 72

## 5 Analysis and Synthesis of the EMS

5.1 Introduction ............................................................... 73
5.2 Energy management analysis: Dynamic Programming .............. 73
   5.2.1 Dynamic Programming working principle ..................... 73
   5.2.2 Results ........................................................... 78
   5.2.3 Analysis of the obtained results ............................... 81
5.3 Energy management synthesis: Rule Based control ................. 84
   5.3.1 Torque split control ............................................. 84
   5.3.2 Control rules ................................................... 86
   5.3.3 Results ........................................................ 91
5.4 Conclusions ............................................................. 91
5.5 Recommendations and discussion ................................... 92

## 6 Analysis and Synthesis of Hybrid Truck Applications

6.1 GVW sensitivity ........................................................... 95
6.2 Drive cycle sensitivity .................................................. 97
6.3 Hybrid synergies: electrified power take-off ....................... 102
   6.3.1 Refrigeration system ............................................. 103
   6.3.2 Vehicle integration of the ePTO ................................. 104
   6.3.3 Modelling the CS-PTO ........................................... 104
   6.3.4 Modelling the ePTO ............................................. 105
   6.3.5 Simulation results .............................................. 105
6.4 Conclusions ............................................................. 108
6.5 Recommendations ...................................................... 108
CONTENTS

7  Marketing the Hybrid Electric Distribution Truck 109
   7.1  SWOT analysis .............................................. 109
   7.2  Internal analysis .............................................. 111
       7.2.1  Strengths .................................................. 111
       7.2.2  Weaknesses ............................................... 113
   7.3  External analysis .............................................. 115
       7.3.1  Opportunities ............................................. 115
       7.3.2  Threads .................................................... 116
   7.4  Market confrontation ......................................... 118
       7.4.1  Strengths to defend ...................................... 118
       7.4.2  Weaknesses to improve/restructure to strengths .... 118
       7.4.3  Opportunities to exploit ................................ 119
       7.4.4  Threats to avoid/withdraw ................................ 119
   7.5  Conclusions .................................................... 120

8  Conclusions and Recommendations 121
   8.1  Conclusions .................................................... 121
   8.2  Recommendations ............................................. 124
       8.2.1  Future research ........................................... 124
       8.2.2  Technological improvements ............................... 126

Dankwoord 127
Nomenclature 129
Bibliography 135

A  Effective tyre roll radius 139
B  Wheel inertia calculation 141
C  DAF V40 simulation data 143
D  Simulation results 145
E  AES Heavy Duty Chassis Dynamometer 151
Summary

Hybrid vehicles are equipped with at least two power sources instead of one. Hybrid vehicles are not new to the world, already back in the 19th century hybrid drivetrains were developed which had similarities with the drivetrain studied in this report. The drive for developing those vehicles nowadays is not to provide reliable or fast transportation in the first place. The main target today is to strongly reduce the consumption of fossil fuels and tailpipe emissions. DAF Trucks NV together with the TU/e and TNO therefore participate in a joined research project to obtain application knowledge for the hybridisation of distribution trucks. The research described in this report contributes to this project by providing an analysis and synthesis tool for the energy management of a parallel hybrid electric truck.

Starting with two literature studies, the first literature study in this report deals with an investigation of the various hybrid trucks and their drivetrain technologies developed worldwide. It is concluded that the parallel hybrid drivetrain topology is the most suitable for the application in distribution trucks. Next literature study discusses the various ways of developing an Energy Management Strategy (EMS) which is considered to be the most important step when achieving the goals of a hybrid vehicle. The Dynamic Programming (DP) optimisation routine is chosen for the analysis of the EMS since it provides the global optimal control solution. For an online implementation of the EMS a Rule Based (RB) controller is found to be most interesting to use.

Vehicle and component models of the base line and hybrid electric truck as well as their supervisory control are built, and serve as a basis for the EMS analysis. These mathematical models represent the studied vehicles and are used to determine their longitudinal dynamic behaviour. The analysis and synthesis tool includes the integrated DP optimisation routine and the vehicle models. The tool is used to provide knowledge for the various hybrid electric truck applications as well as the data required for the synthesis of the RB EMS.

For changing Gross Vehicle Weight (GVW) and drive cycles, various simulations have been made. It is concluded that the various truck applications are of great influence on the resulting fuel consumption and the EMS itself. Additional fuel consumption improvements can be gained when the drive cycle is changed slightly and when the component sizing is optimised. Also the synergetic integration of an electrified Power Take-Off (ePTO) system enables additional fuel savings.

Finally, the current market position for hybrid electric distribution trucks is discussed. This is done with the help of a Strengths, Weakness, Opportunities, and Threats (SWOT) analysis and the so-called confrontation matrix. It is concluded that hybrid electric trucks certainly have a chance of success at the truck market of the near future.
Samenvatting

Hybride voertuigen zijn uitgerust met tenminste twee in plaats van een enkele krachtbron. Hybride voertuigen zijn zeker niet nieuw, al in de 19de eeuw werden er aandrijflijnen ontwikkeld die sterke gelijkenissen vertoonden met de aandrijving welke is beschreven in dit onderzoek. De huidige hybride aandrijvingen worden echter niet meer in de eerste plaats ontwikkeld om te voorzien in snel en betrouwbaar transport. Vandaag de dag is de belangrijkste reden het terugdringen van fossiel brandstof gebruik en uitlaat emissies. Om deze redenen neemt DAF Trucks NV in samenwerking met de TU/e en TNO deel aan een gezamenlijk onderzoeksproject voor het verwerven van toepassingsgerichte kennis over de hybridisering van distributie vrachtwagens. Het onderzoek dat is beschreven in dit rapport draagt daaraan bij middels de ontwikkeling van een applicatie voor de analyse en synthese van het energie management van de parallel elektrisch hybride vrachtwagen.

Het eerste literatuuronderzoek in dit rapport bespreekt de verzameling van de verschillende hybride vrachtwagens en hun aandrijflijnen welke in de afgelopen jaren over de wereld zijn ontwikkeld. De belangrijkste conclusie vormt het feit dat de parallel hybride topologie het meest geschikt is voor toepassing in distributie voertuigen. Het daaropvolgende literatuuronderzoek behandeld de verschillende methoden om tot een Energie Management Strategie (EMS) te komen. Deze wordt beschouwd als de belangrijkste schakel in het behalen van de gewenste resultaten. Voor de analyse van een dergelijke EMS is gekozen voor de toepassing van het zogenaamde Dynamisch Programmeren (DP) omdat dit de globale optimale regelstrategie oplevert. Voor de implementatie van de verkregen strategie worden de gegevens omgezet in een zogenaamde Rule Based (RB) regelaar.

Voor het onderzoek zijn voertuig- en componentmodellen gebouwd van het referentievoertuig en het hybride voertuig evenals de bijbehorende overkoepelende regeling. Deze mathematische modellen representeren de geanalyseerde voertuigen en worden gebruikt om het longitudinale dynamische gedrag te bepalen. De analyse en synthese applicatie bevat een geïntegreerd DP algoritme en de voertuig modellen en voorziet in kennis aangaande verscheidene hybride vrachtwagen toepassingen. Daarnaast voorziet het ook in de noodzakelijke gegevens voor de synthese van de RB EMS.

De analyse en synthese applicatie is gebruikt om de invloed van veranderende voertuig massa (GVW) en rit cyclussen op de EMS te bepalen. Er wordt geconcludeerd dat deze veranderingen significant van invloed zijn op het resulterende brandstof verbruik en de EMS zelf. Om het behaalde verbruik verder te verlagen wordt een voorstel gedaan om de rit cyclus enigszins aan te passen. Tevens levert het optimaliseren van de grootte van de hybride componenten een belangrijke bijdrage in het verder terugdringen van het brandstof verbruik. Verdere verbeteringen zijn
nog te verkrijgen door de synergie van de hybride aandrijving met de aandrijving voor bijvoorbeeld koelsystemen door middel van een zogenaamde electrified Power Take-Off (ePTO).

Tot slot wordt de huidige marktpositie van de elektrisch hybride distributie vrachtwagen bepaald met behulp van een Strengths, Weakness, Opportunities, and Threats (SWOT) analyse en de zogenaamde confrontatie matrix. Er wordt geconcludeerd dat elektrisch hybride vrachtwagens een succesvolle toekomst tegemoet kunnen gaan op de vrachtwagen markt van de nabije toekomst.
Chapter 1

Introduction

The Technische Universiteit Eindhoven (TU/e) and TNO (Dutch organisation for applied scientific research) together with DAF Trucks NV are participating in a joint research and development project named ‘ADAPT-HEV’. Driven by emission regulations and raising fuel prices, the main target of this project is to obtain application knowledge for the hybridisation of distribution trucks. Hybridised vehicles have a drivetrain which consists of at least two power sources which act together in propelling and braking the vehicle. A hybrid drivetrain has the potential to improve vehicle behaviour in the sense of performance, emissions and fuel consumption.

Both governments and fleet owners are interested to take advantage of these potential improvements, however with different interest. Sometimes compensated by incentives (VaMil/MIA\(^1\) [Van Geel, 2006]), governments are mainly interested in emission reductions (emission standards: [DieselNet, 2006], [VROM, 2007] and anti-idling regulations: [EPA, 2003]). Fleet owners, on the other hand, may take advantage of operational savings but not without investing in a new and more expensive vehicle. This investment needs to be compensated by the operational savings within a certain period of time while productivity gains need to be better or equal compared to a conventional vehicle. The current state-of-the-art hybrid vehicles still have a high on-cost. Nevertheless, their potential is improving due to improved energy storage technologies and smarter control strategies.

Problem formulation

Within the ADAPT-HEV project a parallel hybrid electric drivetrain is studied in simulations and actual implementation in a distribution truck. This studied parallel hybrid electric drivetrain has two energy sources; an electrochemical battery and the petrochemical (diesel) fuel. Unlike the petrochemical fuel source, the battery enables power transfer in two directions: charging and discharging. An Energy Management Strategy (EMS) can control the use of the two sources. The EMS of a hybrid vehicle is mainly decisive to optimise the global drivetrain energy efficiency by optimising the energy flow in the system.

Energy management optimisation can be done using a Dynamic Programming (DP) routine which requires full knowledge of the particular drive cycle. Further improvement in fuel eco-

\(^1\)Dutch fiscal regulations for the investment in environmental friendly goods
nomy can be reached by extending the EMS for the control of electrified onboard auxiliaries and accessories. One can think of typical truck accessories like cargo cooling systems. Another aspect where distribution vehicles deal with is the variation in Gross Vehicle Weight (GVW) during trips. Sensitivity of the EMS for GVW variations will also be analysed as well as influences of the imposed drive cycle. The contribution of this work within the ADAPT-HEV project is two-fold. First, a tool for the (sensitivity) analysis of the above mentioned applications is developed. Secondly, by the analysis of these results a Rule Based (RB) energy management controller is synthesised for online implementation.

Report layout

This thesis covers eight chapters which analyse the effectiveness of using hybrid electric distribution vehicles. In the first two chapters two literature studies are performed. The first chapter deals with an investigation made of the current (till 2005) existing hybrid trucks. The list of vehicles includes prototype vehicles and production models which are compared on different fields of interest. In the second literature study an investigation and comparison of different principles to develop the EMS is made.

Analysis of the considered truck applications will be done by determining the numerical optimal control path which is obtained by using a dynamic programming optimisation routine. The design objectives taken into account are minimisation of fuel consumption, maintain or enhance vehicle performance, and equalising the battery State Of Energy (SOE) during an a priori known drive cycle. The RB EMS is then derived from the optimal control path. This is done in Chapter 5. Because an EMS cannot be developed and evaluated without proper simulation models, a vehicle model of the specified hybrid and reference truck are built and validated in Chapter 4.

Furthermore, when considering trucks instead of (for example) cars new aspects enter the research. Trucks are for example subjected to enormous weight changes during their daily driving routes. These routes also vary from highways with constant high speeds to urban environments with varying speeds up to the regulated speed limits. Besides all this, trucks do have a significantly large auxiliary power take-off when for example freight refrigeration systems are applied. The influence of integration of such power take-off in the hybrid drivetrain on fuel consumption is analysed in Chapter 6 as well.

By performing a SWOT analysis in Chapter 7 one can conclude whether hybridisation of distribution vehicles is effective or even realistic. Finally in Chapter 8 the conclusions and recommendations for future work are given.

Besides the research described in this report a heavy duty chassis dynamometer for road load simulations is developed and realised in the Automotive Engineering Science (AES) laboratory of the Technische Universiteit Eindhoven. This test facility will be used to analyse the hybrid electric truck on performance and fuel economy. The TU/e chassis dynamometer is capable of testing 7.5t of rolling mass in combination with 2.5 $[m/s^2]$ accelerations and decelerations up to vehicle speeds of 90 $[km/h]$. More information concerning the chassis dynamometer can be found in Appendix E.
Chapter 2

Heavy Duty Hybrids

In this chapter different ‘hybrid vehicle’ concepts and their most important design aspects as well as a comparison of all known hybrid truck projects world wide are presented. Finally conclusions are drawn according to the findings.

This chapter is separated into two parts; in the first part hybrid power trains in general are discussed. Three important design aspects are highlighted; drivetrain configuration, hybridisation rate and types of energy storage device. Furthermore, in the second part of this chapter more specific subjects related to the list of hybrid truck projects will be treated. Items like energy accumulators, drivetrain configurations versus duty class, fuel consumption reduction, additional costs for hybridisation, and finally a list of hybrid drive train manufacturers will be reviewed.

2.1 Hybrid drivetrains

When considering a hybrid drivetrain, a primary power source and at least one secondary power source mostly connected to an energy accumulator can be distinguished. This secondary power source is used to prevent the primary power source from operating in inefficient operating points resulting in decreasing energy consumption by the vehicle. Hybrid drivetrains can also feature a start-stop functionality which enables shutting down the primary power source to prevent it from idling and further decrease fuel consumption.

2.1.1 Hybrid drivetrain functionality

A conventional vehicle with a single power source, e.g. an internal combustion engine, can only be driven by using its primary power source. A hybrid vehicle with both a primary and secondary power source however has $2^2 = 4$ possibilities to drive the vehicle. These possibilities are called hybrid modes and are determined by the combination of, and the way how the various power sources are cooperating.

<table>
<thead>
<tr>
<th>Table 2.1: hybrid modes</th>
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<tbody>
<tr>
<td>Primary</td>
</tr>
<tr>
<td>Primary</td>
</tr>
<tr>
<td>Secondary</td>
</tr>
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</table>
The first hybrid mode (Table 2.1, 1) is the Engine only (E) mode which is the conventional drive mode. This mode can be activated when a mechanical link is made between the combustion engine and the wheels. The second hybrid mode (Table 2.1, 2) is called the Motor Assist (MA) mode; in this case the secondary power source assists the primary power source. The MA mode is used to prevent the primary power source from running in inefficient operating areas or to supply additional power when boosting. When the secondary power source takes energy from the primary power source the system is in Charge (CH) mode (Table 2.1, 3). This mode is used when cheap energy can be stored in the energy accumulator for later use. Finally, when applying the fourth mode in Table 2.1 the secondary power source is used without the primary one. Two hybrid modes are found here since the secondary power source is able to add energy to and take energy from the drive line. In the Motor only (M) mode the secondary power source accelerates the vehicle and in Brake Energy Recovery (BER) mode the secondary power source decelerates the vehicle. The BER mode is the most important mode because, while braking, free energy is stored into the accumulator. This energy can be used later to assist the engine in the MA mode or to propel the vehicle in the M mode for driving a limited distance without having emissions from the primary power source or for launching the vehicle at stand still.

### 2.1.2 Hybrid drivetrain topologies

Connecting a primary and a secondary power source can be done in several ways. All these different topologies [Chau, 2001] have different advantages and disadvantages and are thus suitable for different purposes. All known hybrid topologies are discussed below and they are compared to a conventional drivetrain.

**Series hybrid**

In the series hybrid configuration, (Figure 2.1) the primary power source drives a generator which provides energy for the secondary power source which propels the vehicle. Typically, for a series hybrid topology there is no mechanical connection between the driven wheels and the primary power source. In some cases recovered brake energy is stored in a battery or a capacitor. The best known example of a series hybrid vehicle is the diesel powered locomotive.

![Figure 2.1: series hybrid drivetrain topology](image)
2.1. HYBRID DRIVETRAINS

Table 2.2: advantages and disadvantages of the series hybrid

| + | No energy accumulator required |
|   | Or else downscaled primary power source possible |
|   | Flexible location for primary power source and generator |
| - | Structural modifications on vehicle chassis required |
|   | Additional generator required |
|   | Large electric components |

Table 2.3: hybrid drivetrain topology legend

| P  | Primary power source |
| S  | Secondary power source |
| A  | Energy accumulator   |
| G  | Generator            |
| L  | Vehicle road load    |
| ↔  | Bidirectional non-mechanical link |
| ↔  | Bidirectional mechanical link |
| →  | Unidirectional non-mechanical link |
| →  | Unidirectional mechanical link |

Parallel hybrid

Contrary to the series topology, the parallel topology (Figure 2.2) has a direct mechanical link between the primary power source and the driven wheels. Of all hybrid topologies the parallel topology has most similarities compared to a conventional drivetrain. The secondary power source is in most cases mechanically linked between the primary power source and the driven wheels (pre- or post-transmission) in such way that this can drive the wheels parallel to the primary power source. A clutch is mounted either before or after the secondary power source, or even two clutches might be applied, enabling the charging mode without any connection to the wheels.

Figure 2.2: parallel hybrid drivetrain topology
Table 2.4: advantages and disadvantages of the parallel hybrid

+ Less modifications required; less expensive to install
  Electrification of auxiliaries possible
  Downscaled primary power source possible
  Single secondary source serves as motor and generator
- Limited performance in M mode, depending on component size of A and S

Series-parallel hybrid

When combining the series and parallel topology the series-parallel or stringear topology (Figure 2.3) will arise. This topology appears to be like the parallel topology but with an additional generator, and like the series topology with an additional mechanical link between the secondary power source and the generator. One can distinguish an engine-heavy and an electric-heavy series-parallel hybrid drivetrain depending on the size and activity of the secondary and primary power source.

![Figure 2.3: series-parallel hybrid drivetrain topology](image)

Table 2.5: advantages and disadvantages of the series-parallel hybrid

+ Possesses advantages of both series and parallel topology
- Additional power source required (generator) compared to parallel topology
  More complicated system compared to series or parallel topology
  More expensive compared to series or parallel topology

Complex hybrid

On top of the series-parallel topology a fourth topology can be identified named the complex hybrid (Figure 2.4). In contradiction to the series-parallel topology, this topology includes a planetary gear set to connect the secondary power source and the generator to the primary power source. This system can offer additional operating modes compared to the series-parallel hybrid since the third power source, the generator, can be used in a bidirectional way and thus be used for propulsion as well.
2.1. HYBRID DRIVETRAINS

Table 2.6: advantages and disadvantages of the complex hybrid

| + Versatile operating modes possible |
| Usually simple mechanical transmission layout |
| - Difficult to be more effective than parallel topology |
| More complicated system compared to series or parallel topology |
| More expensive compared to series or parallel topology |

Plug-in hybrid

All hybrid drivetrains with an electrochemical accumulator can be converted to a "plug-in hybrid". A plug-in hybrid (Figure 2.5) simply charges the accumulator from a third party power source like the electrical power grid. In this way renewable energy is available for driving and the primary power source can remain inactive for a longer period of time.

Table 2.7: advantages and disadvantages of the plug-in hybrid

| + Additional fuel savings possible |
| Zero emission driving possible |
| Renewable energy (from the grid) can be used for driving |
| Many of these vehicles can peak-share power demands during night time |
| - Cheap/renewable energy from third party required |
| Vehicle needs to be plugged to source |
| Long charge time intervals |
2.1.3 Energy accumulators

All of above hybrid topologies, except from some series configurations, use an energy accumulator to store energy. As known, energy can be stored in various ways. Practical solutions for hybrid drivetrains include electrochemical, electrical, and mechanical systems. Thermal systems which store heat are less interesting for use in vehicles because of their slow dynamics and excessive dimensions. Potential energy storage systems make use of gravity and are for this reason also not very interesting for vehicle applications.

Electrochemical and electric storage systems like batteries and supercapacitors respectively are widely used in hybrid drivetrains. More information can be found in the second part of this chapter. Storage systems like flywheels, springs, hydraulics, and pneumatics can also be used, further analysis of these systems is however beyond the scope of this research.

2.1.4 Hybridisation rate

Besides the drivetrain topology another important design aspect, the rate of hybridisation, is known. The hybridisation rate HR (Equation 2.1) is the quantification of the relative amount of power available from the secondary power source compared to the total available power in the drivetrain [Liao, 2004]. The hybridisation rate is important because it determines whether the vehicle is a mild, semi or full hybrid (Table 2.1). Mild-hybrid vehicles only feature start-stop and regenerative braking while a semi-hybrid uses the available electric power to accelerate and boost the vehicle. Also regenerative braking is possible. A full hybrid on the other hand has enough electric power to drive purely electrically for a limited distance and with most times a limited speed. Regenerative braking is also possible. Semi- and full hybrids also feature the start-stop functionality of the primary power source. A fourth class of hybrid vehicles is the micro hybrid which has a smart energy management [Koot, 2006] for powering onboard auxiliaries through a generator with a typical size of 2 to 5 [kW]. A micro hybrid features brake energy recovery in a very limited sense.

\[
HR = \frac{P_{\text{secondary}}}{P_{\text{primary}} + P_{\text{secondary}}} \times 100\,\% \quad (2.1)
\]

<table>
<thead>
<tr>
<th>Hybridisation rate</th>
<th>Classification</th>
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<tbody>
<tr>
<td>HR&lt;23</td>
<td>Mild hybrid</td>
</tr>
<tr>
<td>23&lt;HR&lt;38</td>
<td>Semi hybrid</td>
</tr>
<tr>
<td>38&lt;HR</td>
<td>Strong/full hybrid</td>
</tr>
</tbody>
</table>

2.2 Hybrid drivetrains used in heavy duty vehicles

In the past decade several heavy duty hybrid vehicles have been produced as a prototype and as commercially available vehicles. In total 184 known heavy duty hybrid vehicles, build till December 2005, are documented. These vehicles include buses, coaches, trucks, and military vehicles. They all have in common that they make use of at least two different power sources which propel
2.3. ELECTRIC ENERGY ACCUMULATORS

The primary power source is an internal combustion engine which is running on normal diesel or alternative fuels like CNG, LPG or Bio-Diesel. The secondary power sources are electric motors, generators, hydraulic actuators combined with several different types of energy accumulators. The 184 heavy duty hybrids include 49 hybrid trucks which are analysed in this report. Buses, coaches and other passenger transportation vehicles are documented but not taken into account since the overall research only focuses on cargo transportation.

2.3 Electric energy accumulators

Most of the hybrid heavy duty vehicles feature regenerative braking. With this system, brake energy is recuperated and to be stored into an energy accumulator for later use. All documented hybrid trucks use electrochemical or electrical accumulators or a combination of both.

The distribution of all documented electric energy storage devices is plotted as a percentage of the total amount of the documented storage devices. As an exception in this investigation all heavy-duty hybrid electric vehicles developed in the last ten years are taken into account, including passenger transportation vehicles. This is done because the development of storage technologies is the same for cargo and passenger transportation.

From Figure 2.6 it can be determined that lead acid batteries are most widely used over the last decade. The main reason for this is the price per [kWh] storage capacity for these batteries; it is the cheapest but as a disadvantage also the heaviest energy storage device. A trend can be observed towards new battery technologies to break through, one of those technologies is the Lithium-ion (Li-ion) battery. This technology is not widely used for heavy duty hybrid vehicle applications at the moment, however the number of applications is growing. Two reasons can be given: firstly the cost of Li-ion technology becomes lower and secondly the demand for a low specific weight energy storage device grows rapidly. An intermediate step towards Li-ion, the Nickel Metal Hydride battery, can be identified as the second most used technology. Nickel based batteries, including the Nickel-Cadmium battery, have a charge rate which is much higher than that of lead acid batteries. However the older Ni-Cd battery suffers from the memory effect and contains highly toxic materials. Capacitors, also called super- or ultracapacitors when built for high capacities, do not store power by using a chemical reaction but store power as static electricity. The main

![Figure 2.6: electric energy storage device distribution](image)
advantage is that faster charging and discharging is possible at a high efficiency. A disadvantage however is their low capacity and cost-to-weight ratio. To boost the performance of for example Lead Acid batteries, capacitors can be connected in parallel to them. In this way advantages of both technologies can be exploited and a high capacity accumulator with improved charge and discharge rates arises. As a second advantage also the lifetime increases as of improved energy management. An overview of accumulator specifications are summarised in Table 2.9. Besides the mentioned storage devices other less common devices are found in some heavy duty hybrid electric vehicles. These technologies are however beyond the scope of this research.

- Nickel Cadmium Chloride batteries
- Nickel Metal Chloride batteries
- Nickel Zinc/ultracapacitors battery/capacitor combination
- Zinc Air batteries
# 2.3. ELECTRIC ENERGY ACCUMULATORS

<table>
<thead>
<tr>
<th></th>
<th>Lead Acid</th>
<th>Nickel Metal Hydride</th>
<th>Nickel Cadmium</th>
<th>Super-capacitors</th>
<th>Lithium Ion (Mn type)</th>
<th>Lead Acid + Supercaps</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific energy [Wh/kg]</td>
<td>35 (30-50)</td>
<td>60 (60-120)</td>
<td>50 (45-80)</td>
<td>3-5</td>
<td>135 (100-135)</td>
<td>35-55</td>
</tr>
<tr>
<td>Power level [kW/kg]</td>
<td>0.18 (0.4)</td>
<td>0.2 (0.2)</td>
<td>0.12 (0.1)</td>
<td>1</td>
<td>0.43 (0.3)</td>
<td>0.6</td>
</tr>
<tr>
<td>Specific cost [$/kWh]</td>
<td>250</td>
<td>900</td>
<td>600</td>
<td>10260</td>
<td>1200</td>
<td>5250</td>
</tr>
<tr>
<td>Specific cost [$/MJ]</td>
<td>70</td>
<td>250</td>
<td>170</td>
<td>2850</td>
<td>330</td>
<td>1460</td>
</tr>
<tr>
<td>$R_i$ [mΩ]</td>
<td>100 (12 [V] pack)</td>
<td>200-300 (6 [V] pack)</td>
<td>100-200 (6 [V] pack)</td>
<td>&lt;1</td>
<td>25-75 (per cell)</td>
<td>&lt;100</td>
</tr>
<tr>
<td>η (relative)</td>
<td>o</td>
<td>––</td>
<td>o</td>
<td>++</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Charge time [h]</td>
<td>8-16</td>
<td>2-4</td>
<td>1</td>
<td>&lt;&lt;0.001</td>
<td>&lt;1</td>
<td>0.001-16</td>
</tr>
<tr>
<td>Expected lifetime [−] (cycles up to 80 [%] SOE)</td>
<td>600</td>
<td>1000</td>
<td>1500</td>
<td>&gt; 1.10^9</td>
<td>1200</td>
<td>1200</td>
</tr>
<tr>
<td>Usage for heavy duty purposes</td>
<td>35 [%]</td>
<td>20 [%]</td>
<td>12 [%]</td>
<td>11 [%]</td>
<td>10 [%]</td>
<td>7 [%]</td>
</tr>
<tr>
<td>Environmental impact</td>
<td>Toxic lead and acid, harmful to environment</td>
<td>Low toxicity, should be recycled</td>
<td>Highly toxic, harmful to environment</td>
<td>Low toxicity</td>
<td>Low toxicity</td>
<td>Toxic lead and acid, harmful to environment</td>
</tr>
</tbody>
</table>

Table 2.9: typical performance of accumulators used in heavy duty hybrid electric vehicles

2.4 Duty classes

Trucks come in different sizes and can be divided into different classes and duty classes depending on their Gross Vehicle Weight (GVW). This vehicle mass is the total maximum vehicle mass when driving on the road, including payload, and separates trucks in 8 different classes and 3 different duty classes (light; LD, medium; MD, and heavy duty; HD respectively) as can be seen in Table 2.10 [TTO, 2007] and Figure 2.7 [Woodroffe, 2000].

![Figure 2.7: heavy duty vehicle classes](image)

<table>
<thead>
<tr>
<th>Class</th>
<th>GVW [lb]</th>
<th>GVW [kg]</th>
<th>Duty class</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0-6.000</td>
<td>0-2.700</td>
<td>LD</td>
</tr>
<tr>
<td>2</td>
<td>6.001-10.000</td>
<td>2.701-4.500</td>
<td>LD</td>
</tr>
<tr>
<td>3</td>
<td>10.001-14.000</td>
<td>4.501-6.350</td>
<td>LD</td>
</tr>
<tr>
<td>4</td>
<td>14.001-16.000</td>
<td>6.351-7.300</td>
<td>MD</td>
</tr>
<tr>
<td>5</td>
<td>16.001-19.500</td>
<td>7.301-8.850</td>
<td>MD</td>
</tr>
<tr>
<td>6</td>
<td>19.501-26.000</td>
<td>8.851-11.800</td>
<td>MD</td>
</tr>
<tr>
<td>7</td>
<td>26.001-30.000</td>
<td>11.801-15.000</td>
<td>HD</td>
</tr>
<tr>
<td>8</td>
<td>33.000-150.000</td>
<td>15.001-68.000</td>
<td>HD</td>
</tr>
</tbody>
</table>

2.5 Drivetrain configuration versus duty class

On the following pages several plots are derived from a datasheet containing the different investigated hybrid trucks (including vans, trucks, refuse trucks, specialty trucks, and military applications) build in Europe & Japan and in the USA & Canada in the last decade. The differentiation between Europe & Japan on the one hand and the USA & Canada on the other hand is made
2.5. **DRIVETRAIN CONFIGURATION VERSUS DUTY CLASS**

because of the existing cultural automotive differences. Each different vehicle, no matter how many are built, is documented as a single project. The duty classes are categorised for the five different drivetrain concepts.

**USA & Canada**

These countries are the forerunners in the field of hybridised heavy duty vehicles. This has two reasons. In the first place governmental legislations in states like California are a motivation for developing clean heavy duty vehicles. These legislations enclose idle off [ARB, 2006], heavy duty vehicles may not run in idle mode longer than a certain period of time. Other emission based legislations are about \( CO_2 \), \( NO_x \), and PM emissions. A second reason for the early development of hybrid heavy duty vehicles is the United States Army which acts as a client for the research done in this field. The US Army is interested in alternatively propelled vehicles and fuel consumption reduction mainly because the price of one gallon of fuel costs up to $50,- on the battle field which is 25 times as expensive compared to urban native areas. Also performing stealth operations with electric vehicles and electric energy supply on the battle field are interesting items.

In the USA and Canada 114 hybrid heavy duty vehicles projects are documented, 26 from these are hybrid trucks, which is the most of all countries. These cargo vehicles include 6 different military applications.

Amongst the 26 hybrid trucks are 7 parallel drivetrains which are applied in the MD and HD class. The series configuration is used for 10 trucks, mostly in HD (refuse trucks) and LD class. The other 9 vehicles use series-parallel, plug-in, as well as hydraulic hybridisation. Plug-in hybrids are popular in the LD class; these vehicles are suitable for charging from the electricity grid because they don’t run out of electrical power as fast as the vehicles in the heavier truck classes.

The distribution of the several drivetrain concepts used for hybrid trucks in the USA & Canada can be found in Figure 2.8.

Summarised:

- More heavy duty hybrid vehicles and truck projects compared to Europe and Japan
- Military applications
- Variety of drivetrain concepts
- Mostly series, parallel comes second
- Class 7 and 8 vehicles are most popular for hybridisation
CHAPTER 2. HEAVY DUTY HYBRIDS

As a result of higher standards in internal combustion engine development, think of the diesel engine, as well as the use of lighter vehicles, the amount of hybrid heavy duty vehicles in Europe and Japan is a little less compared to the USA and Canada. Also customers like the defence industry are not (yet) common in this sector. For these reasons significantly less heavy duty hybrid vehicles are found; only 70, 23 of these are hybrid trucks. The parallel electric hybrid drivetrain is represented in 13 projects which is surprisingly more than in the USA and Canada. The parallel configuration is also used in all duty classes. The popularity of the parallel configuration is caused by its low cost and simple implementation. The system does not require many adaptations of the vehicle. Only 7 series electric trucks are known, these are only found in the LD and HD duty class.

The distribution of the several drivetrain concepts used for hybrid trucks in Europe & Japan can be found in Figure 2.9.

Summarised:

- Mostly parallel electric, series comes second
- Only one defence vehicle documented
- Less variety in different concepts (87 [%] is parallel or series)
- Light duty vehicle class most popular for hybridisation
2.5. DRIVETRAIN CONFIGURATION VERSUS DUTY CLASS

In the following graphs all vehicles in the sectors above are summed. From Figure 2.10 it can be concluded that generally series and parallel topologies are approximately equally distributed when taking military vehicles into account. Series-parallel, plug-in, and hydraulic hybrids are also equally distributed but are in a number of 4 vehicles per category. From the total amount of 49 hybrid trucks, hybridisation is mostly applied in the LD class (21 vehicles), followed by the HD class (16 vehicles) and the MD class (12 vehicles). From the 7 documented military vehicles 5 are equipped with the series electric drivetrain which brings the parallel hybrid electric vehicle as the number one applied concept for civil cargo transportation.
2.6 Fuel consumption improvement

It is clear that the main reasons for hybridisation are fuel economy improvement and emission reduction. Because it is hard to check whether a certain fuel consumption reduction or emission reduction is reached this aspect could not be investigated for each vehicle. Besides this, the amount of saved fuel is highly correlated with the type of drive cycle and especially the amount of vehicle stops where the engine can be shut off. An example is the hybrid electric boom-truck, a service truck for repairing for example highway lightning, which saves up to 50\% of fuel compared to the conventional truck. This effect was mainly caused due to a significant amount, a day period, of idle stop. However in general all hybrid heavy duty vehicle manufacturers claim a reduction of 20-50\% for both fuel consumption and \( \text{CO}_2 \) emission reduction.

2.7 Additional cost for hybridisation

Hybridisation of any vehicle needs an investment which has a return time depending on the amount of additional cost and the reached fuel consumption reduction. For a good market penetration this return time may not exceed 1.5 to 2 years in general. To create a benchmark for a future hybrid vehicle an investigation is made which encloses the additional cost for hybridisation of heavy duty hybrid vehicles. Unfortunately only three hybrid heavy duty vehicle manufacturers were able to inform about prices. Information about these vehicles is given in Table 2.11. Both the Solectria and Enova trucks are prototype vehicles and for that reason do not represent a developed market. The Toyota Hino parallel hybrid vehicle is already available on the market (2005) and costs an additional $11,000,- compared to the base line vehicle.

<table>
<thead>
<tr>
<th>Vehicle</th>
<th>Drivetrain configuration</th>
<th>Duty class</th>
<th>Availability</th>
<th>Additional cost ($)</th>
<th>Total vehicle cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Toyota Hino</td>
<td>parallel</td>
<td>4</td>
<td>USA and Japan</td>
<td>11,000</td>
<td>55,000</td>
</tr>
<tr>
<td>Solectria Kenworth T300</td>
<td>series-parallel</td>
<td>7</td>
<td>USA</td>
<td>75,000</td>
<td>150,000</td>
</tr>
<tr>
<td>Enova HybridDrive post-transmission</td>
<td>parallel</td>
<td>4-6</td>
<td>USA</td>
<td>50,000</td>
<td>-</td>
</tr>
</tbody>
</table>

2.8 Hybrid drivetrain developers and manufacturers

As a final overview all known companies building hybrid drivetrains or parts for heavy duty applications together with their customer(s) are summed in Table 2.12. This list includes all companies involved in hybrid electric drivetrains development including drivetrains for buses. Most of these companies are located in the USA.


### Table 2.12: drivetrain manufacturers in hybrids (2005)

<table>
<thead>
<tr>
<th>Company</th>
<th>Customers (among others)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allison</td>
<td>GM, New Flyer, Orion, APTS</td>
</tr>
<tr>
<td>Azure Dynamics</td>
<td>Purolator, Canada Post, USPS, PAC-CAR, AM General, London Taxi</td>
</tr>
<tr>
<td>BAE</td>
<td>US Army, US urban transport companies</td>
</tr>
<tr>
<td>Capstone</td>
<td>Ebus, ISE Research</td>
</tr>
<tr>
<td>Eaton</td>
<td>FedEx, DAF</td>
</tr>
<tr>
<td>EPRI</td>
<td>DaimlerChrysler</td>
</tr>
<tr>
<td>e-Traction</td>
<td>-</td>
</tr>
<tr>
<td>Gemco</td>
<td>Bavaria</td>
</tr>
<tr>
<td>H-Power</td>
<td>Blue Bird</td>
</tr>
<tr>
<td>IDT/PEI</td>
<td>US Army</td>
</tr>
<tr>
<td>ISE Research</td>
<td>US urban transport companies</td>
</tr>
<tr>
<td>Permo-drive</td>
<td>US Army, DANA Corporation</td>
</tr>
<tr>
<td>ProPulse</td>
<td>US Army/Oshkosh</td>
</tr>
<tr>
<td>Solectria</td>
<td>part of Azure Dynamics</td>
</tr>
<tr>
<td>Spijkstaal</td>
<td>Bredamenarinibus, Gebr. Veldhuizen</td>
</tr>
<tr>
<td>UQM</td>
<td>Eaton</td>
</tr>
<tr>
<td>ZF</td>
<td>GM, DaimlerChrysler</td>
</tr>
</tbody>
</table>

Besides the companies which are developing drivetrains only, there are also truck manufacturers which develop their own drivetrains for mentioned applications. The following manufacturers can be distinguished:

- Volvo
- Toyota
- Mitsubishi Fuso
- Nissan Diesel
- GM
2.9 Conclusions

Many conclusions can be drawn on this investigation, a summary of the most important conclusions together with some recommendation are given.

Drivetrain topology

At first some words about the type of drivetrain topology, which is the first issue when hybridising a vehicle for fuel consumption and/or emission reduction. The kind of duty cycle and the application are of great influence on the topology chosen. For example a refuse truck is used in urban areas on drive cycles where many stops are involved. In this case a series topology (with a high rate of hybridisation) is preferred. On the other hand when developing a long haulage truck there is not very much fuel to win when using the electric motor for propulsion since constant vehicle speeds are involved most of the time. For this kind of vehicles a parallel topology with a low hybridisation rate and an idle stop system would be interesting, while electrified onboard auxiliaries are using power from the hybrid battery. Boost functionality could also be considered for overtaking slower traffic, or to downsize the internal combustion engine without performance penalties.

Storage devices

An energy storage device is required when recuperating energy while braking. The kind and size of this storage device depend on the application where the truck is used for. An electrochemical or electrical accumulator is preferred in the first place because it can be used in the broadest sense. A trend towards the use of light, maintenance free and less toxic electrochemical devices is observed. The latest technology is the Li-ion cell which already has these advantages. The size of the storage device is depending on the application and also the GVW of the vehicle. The main question here is: How much power and/or capacity is required when considering all power consuming devices?

Fuel economy improvement

Series hybrid drivetrains are used when many stops and go’s are involved in the drive cycle, while a parallel hybrid is a more general purpose concept. The series hybrid has the highest potential in fuel consumption improvement, when used for low demanding stop and go cycles, while the parallel concept has the highest potential when used for more demanding and varying high speed cycles. Also integration of electrified auxiliaries has an enormous influence on fuel economy. In general, fuel consumption reduction for heavy duty hybrids varies between 10 and 50 [%].

Cost

Cost is a very important factor while considering hybridisation and is expressed as cost of ownership. Investments in cargo vehicles should have a return on investment time of 1.5 to 2 years maximal. Thus an expensive series-parallel configuration requires higher fuel efficiency compared to a parallel electric hybrid where only a motor/generator and a battery are added to an existing vehicle.
2.9. CONCLUSIONS

Military applications

In this literature study a trend is observed for military vehicles to have series topologies most of the time while civil cargo vehicles mostly use parallel hybridisation. Reasons for this are due to budget and application. For military applications a larger budget is available to modify vehicles since fuel prices are much higher in battlefields and performing in stealth mode is a highly desired feature.

Future trend

What will be the trend for the near future? An answer to this question is in any case smart hybridisation; relatively high improvements with relatively less drivetrain adaptation. An example is found in the real-time optimal controller made for an alternator [Koot, 2006]. Electrification of vehicle components integrated in the energy management system of the drivetrain will also be an important issue. Besides smart hybridisation also the parallel drivetrain topology will gain more and more interest, it can be implemented at a much lower cost than a series drivetrain. Future hybrid electric trucks, on the market from 2009 onwards, will have this kind of topology.

DAF hybrid electric distribution truck

According to the research done in this chapter, a hybrid electric distribution truck would ideally consist of a parallel hybrid electric drivetrain and a Li-ion battery pack combined with supercapacitors. The battery size may vary depending on the number of integrated auxiliaries and accessories as well as the total capacity of the supercapacitors. The parallel concept is the most general purpose concept; it can be applied over the whole range of truck classes. These trucks will not only drive in urban environments but also on highways where an almost constant and high long-lasting power demand necessitates the use of an (almost) full size internal combustion engine. The (parallel) use of supercapacitors is preferred since highly dynamic vehicle speed in traffic results in short regenerative braking periods.
2.10 Final word

While finishing this report, a number of new hybrid electric heavy duty vehicles are being introduced as a production version or as a prototype. Zahnradfabrik Friedrichshafen (ZF) offers multiple hybrid technology concepts. Their latest parallel hybrid system offers a solution for 4.5t distribution trucks consisting of a 3 litre diesel engine and a 40 kW electric motor featuring start-stop and full electric launch. According to ZF the system increases fuel economy with 30 [%] [Goll, 2006]. Since July 2006 Mitsubishi Fuso offers the Canter Eco hybrid for the Japanese market. Powered with a 3 liter turbo diesel engine and a 35 kW parallel mounted electric machine the 5t vehicle is capable of 20 [%] increase in fuel efficiency. Another hybrid system comes from a global effort consisting of DaimlerChrysler, General Motors, and BMW called Global Hybrid Cooperation. The so called two-mode hybrid system will be available on the market for light duty vehicles in 2007. The two-mode hybrid consists of a powersplit transmission with two electric motors and four fixed speeds for mechanically transferring power from the internal combustion engine. This hybrid system is optimised for two operating modes, one for low and one for high speeds. Also available on the market, but in 2009, is the parallel hybrid electric system suitable for GVW greater than 7.5t developed by Volvo. Volvo claims a fuel economy improvement up to 35 [%].
Chapter 3

Energy Management Strategies

“The amount of fossil fuel saved by a single hybridised truck equals the amount of fuel saved by fifteen Toyota Priuses.” When keeping in mind that trucks are driving all day long, hybridisation could thus be very interesting and should be done in an optimal way.

3.1 Introduction

As explained in Chapter 2, a hybrid drivetrain configuration can be chosen in such way that it performs most effective for a certain duty on a certain drive cycle. A second step in developing a hybrid electric (heavy duty) vehicle is the development of the Energy Management Strategy (EMS) or supervisory drivetrain controller. An Energy Management Strategy determines the amount of energy taken from and send to the energy accumulator for each vehicle driving condition. The EMS is mainly decisive for the optimal energy flow through the drivetrain and thus mainly decisive for the performance of the total vehicle. This performance can be in the sense of maximal fuel economy, minimal emission, or optimal drivability or comfort. The development of an EMS can thus be seen as the most important step in achieving the actual fuel saving in a hybrid vehicle.

Figure 3.1: energy management development technologies

The flow chart of Figure 3.1 shows an overview of the technologies available for the development of an EMS discussed in this chapter. In this chapter it will be researched which EMS is most suitable for fuel consumption minimisation.
3.2 Problem formulation

In this study, the aim of the EMS is to provide the optimal control energy setpoint $E_{s,\text{opt}}(t)$ of the battery in order to minimise the cumulative fuel consumption $m_f$ of the hybrid vehicle with a power demand $P_v(t)$. This general optimisation problem for the fuel consumption minimisation of hybrid electric vehicles can be described by Equation 3.1. The optimisation constraints (Equations 3.5, 3.4, 3.3) describe the limited power $P_s(t)$ that can be send from and to the secondary power source and the limited energy $E_s(t)$ available from the battery which is calculated by Equation 3.2. A third constraint determines the maximum allowed energy content deviation $\Delta E_s$ over the complete cycle. Whether a charge sustaining or a charge depleting strategy is analysed this value is zero or non-zero respectively.

$$m_f(E_s(t)) = \min_{E_s(t)} \int_{t=0}^{t=t_{\text{end}}} m_f(E_s(t) \mid P_v(t)) \, dx$$ (3.1)

$$E_s(t) = E_s(0) + \int_0^t P_s(\tau) \, d\tau$$ (3.2)

$$E_s(t = t_{\text{end}}) - E_s(t = 0) = \Delta E_s$$ (3.3)

$$E_{s,\text{min}} \leq E_s(t) \leq E_{s,\text{max}}$$ (3.4)

$$P_{s,\text{min}} \leq P_s(t) \leq P_{s,\text{max}}$$ (3.5)

3.3 Accumulator energy flow control

In general two main EMS types can be distinguished: a charge sustaining and a charge depleting strategy. In case of a charge depleting strategy, an external power source like the electricity grid is applied. The use of energy from the grid can be preferable in the sense of efficiency or emissions from well to wheel, as well as in the sense of costs. A charge depleting strategy, when used in "plug-in" hybrids, will try to use as much energy from the battery as possible without using the primary power source [Frank, 2004]. This strategy is most suitable for vehicles which run on relatively short distance drive cycles.

A charge sustaining strategy will try to keep the amount of energy present in the battery at a constant level preferably where its efficiency is highest. Using a charge sustaining strategy can also be preferable for the extension of the battery life since regular deep discharging is not desired for some technologies. Especially Li-ion batteries show improved lifetime when not being subjected to deep discharge cycles. Besides all this, a charge sustaining EMS prevents the user from the hassle of an external charge system. In fact, a charge sustaining vehicle can be used in the same way as a conventional (non-hybrid) vehicle.

An additional advantage for the use of a depleting strategy is the possibility of battery recovery at night for some storage technologies which suffer from a memory effect. This effect decreases the battery capacity when a non-empty battery is re-charged, deep discharging followed by slow charging then recovers the battery. Li-ion batteries however do not suffer from this chemical
3.4 Drivetrain energy flow control

An EMS can be developed by using intelligence control or by using advanced control. Both ways intend to serve the same goal of maximising or minimising a certain performance criterion by taking certain constraints into account. For hybrid electric vehicles these constraints are prescribed by the torque and speed range of the drivetrain components as well as the battery capacity and performance. In the past decade the effectiveness of the EMS is increased due to new technologies and faster computing capability. This goes for the technology in the vehicle as well as for the technology used for the development trajectory of the EMS. Examples are various sensors and communication networks like CAN-bus (Controller Area Network) and FlexRay in the vehicle and faster computers and measurement devices in research facilities.

3.4.1 Intelligence control

The intelligence controllers described in this section are based on heuristics and make use of practical experience or experimental data. Heuristics, derived from the Greek word 'eureka' which means 'to find', usually provide a good solution but without proving it to be an optimal solution.

Rule Based control

The best known example of intelligence control is the Rule Based (RB) controller which consists of a set of heuristic rules which determine the energy flow through the hybrid drivetrain instantaneously. Typical controller rules for series hybrid drivetrains make use of the 'thermostat method' which turns the generator on and off depending on the energy content of the battery. Another strategy for series drivetrains is the 'load following method' where the generator provides all the required drive power and sends it directly to the wheels without the intervention of the battery [Van Mierlo, 2003]. Also for parallel configured hybrid vehicles various engineering intuitive controllers can be found in literature. Examples are based on the activation of hybrid modes depending on vehicle speed and the tracking of Optimal Operating Lines (OOL) or Optimal Operating Points (OOP) in the engine map. Also a power-split strategy where the amount of electric power assist is a function of the demanded power is a well known example [Rahman, 2000]. Unlimited combinations of these heuristic rules are possible, for example when adding a charge sustaining rule (like the thermostat method) to a powersplit rule for the prevention of battery damage. For "plug-in" hybrids [Frank, 2004] a charge depleting strategy is implemented in combination with a charge sustaining strategy to maximise the all-electric driving range.

Fuzzy Logic control

Another example of intelligence control is the Fuzzy Logic (FL) controller [Glenn, 2000]. This type of control is universally applicable for all sorts of complicated problems (e.g. Anti-lock Braking Systems) since it has its foundation on human intuition and experience. The basic idea of
a FL controller is to formulate human knowledge and reasoning, which can be represented as a collection of if-then rules, in a way tractable for computers. These rules are in fact linguistic rules like for the description of an error being ‘small-positive’ or ‘large-negative’ and can be seen as an extension of the conventional Boolean true/false logic [Hellgren, 2004].

**Neural Network control**

Also the Neural Network (NN) controller is an example of intelligence control and is in fact a statistic calculation model. Like the human brain, neural networks consist of neurons (Figure 3.2) which include inputs, weighing functions, a summation, an activation and an output. These neurons are connected with other neurons in a network and are able to send and receive information from and to other neurons. The weights of the neuron inputs are determined by a training algorithm, in this way a NN controller has learning capacities [Harmon, 2005].

![Figure 3.2: neuron in a neural network](image)

The advantages and disadvantages of the above mentioned intelligence control types are summarised in Table 3.1. The main disadvantage of all these engineering intuitive strategies is the fact that they are not capable of capturing all of the dynamic behaviour of the hybrid drivetrain components in the control algorithm [Lin, 2001b]. Due to this the obtained control solution will never be the optimal one.

<table>
<thead>
<tr>
<th>Table 3.1: types of intelligent control</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Advantages</strong></td>
</tr>
<tr>
<td>Rule based control</td>
</tr>
<tr>
<td>Fuzzy logic control</td>
</tr>
<tr>
<td>Neural network control</td>
</tr>
</tbody>
</table>

**3.4.2 Advanced control**

Advanced control is introduced, since the proposed intelligence control types may not take full advantage of the hybrid drivetrains maximal performance. By using advanced control strategies for the control of the energy flow in a hybrid vehicle, the potential of taking the full advantage of small differences in the efficiency maps of the various drivetrain components comes available. In this way many more drivetrain phenomena can be captured in a more accurate and optimal
3.4. DRIVETRAIN ENERGY FLOW CONTROL

way. Advanced control can be applied in many different ways, the most important are mentioned below and their advantages and disadvantages are given.

Optimal control

The aim of optimal control is to minimise a certain (fuel) cost function $J_f$ over a time horizon by taking relevant constraints into account. This time horizon can either cover a fixed point in time, a time window or the length of a complete driving cycle. Optimisation for a fixed point in time is called static or local optimisation as described by Equation 3.6 while the dynamic optimum Equation 3.7 is calculated over a time horizon, called global optimisation.

$$J_f = \min_{x(t)} \dot{m}_f(x(t))$$ (3.6)

$$J_f = \min_{x(t)} \int^t_{t+\tau} \dot{m}_f(x(t)) \, dx$$ (3.7)

A static control optimum can be found by translating the battery energy into an equivalent amount of fossil fuel. This gives the opportunity to find the steady state optimal control input $x(t)$ for a fixed point in time. This simple point-wise optimisation can be extended to minimise fuel consumption and emissions simultaneously. Dynamic optimal control on the other hand uses a time horizon $\tau$ in stead of fixed point in time. This way of optimisation takes the dynamic behaviour of the drivetrain into account and will give more accurate results under transient conditions [Lin, 2001a]. Lots of algorithms known in literature can be used (whether or not in combination with other algorithms) to calculate the optimal control solution, a selection of these algorithms is explained later in this chapter.

Predictive control

In real world situations the complete driving cycle is never known a priori and thus a control solution can never be 100 [%] optimal in these online situations. Predictive control however can provide a near optimal solution $x(t)$ since it uses limited time horizon information coming from historical data, a vehicle model or a GPS navigation system (e-horizon). For each time step an optimisation algorithm is carried out over the predicted time horizon $\tau_p$ to minimise cost function $J_f$ (Equation 3.8). After this optimisation the first value $x(t)$ of the control sequence will actually be implemented [Koot, 2006].

$$J_f = \min_{x(t)} \int^t_{t+\tau_p} \dot{m}_f(x(t)) \, dx$$ (3.8)

Adaptive control

For the changing conditions where vehicle drivetrains deal with, an adaptive control strategy could be satisfying. In order to minimise cost function $J_f$, this kind of controller adjusts its characteristics when the system to be controlled by $x$ is subjected to changing driving or drivetrain conditions $u$, like a transient drive cycle or a depleting battery. It therefore consist of a control loop and an adjustment loop (Figure 3.3). Adjusting the controller can be done in a direct and indirect way; direct adjustment is done by using an explicit linearised system model in the loop
while indirect adjustment uses an estimate of the system which is used to update the control parameters [Harmon, 2005].

![Figure 3.3: general adaptive control block diagram](image)

The advantages and disadvantages of the above mentioned control types are summarised in Table 3.2.

<table>
<thead>
<tr>
<th>Types of advanced control</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Local optimal control</td>
<td>Simultaneous optimisation of different criteria possible</td>
<td>Only finds local optimal solution</td>
</tr>
<tr>
<td></td>
<td>Real-time implementable</td>
<td></td>
</tr>
<tr>
<td>Global optimal control</td>
<td>Accurate under transient conditions</td>
<td>Requires heavy computation</td>
</tr>
<tr>
<td></td>
<td>Finds global optimal solution</td>
<td></td>
</tr>
<tr>
<td>Predictive control</td>
<td>Near-optimal under transient conditions</td>
<td>Requires a detailed vehicle model captured in the controller</td>
</tr>
<tr>
<td>Adaptive control</td>
<td>Useful when characteristic conditions significantly change</td>
<td>Requires linear parameterisation of a non-linear vehicle model</td>
</tr>
</tbody>
</table>

3.4.3 Optimisation algorithms

Most of the mentioned control techniques require an optimisation algorithm when being implemented. For the search of maxima or minima in the design space several optimisation algorithms are in existence, the most interesting ones are explained below and their advantages and disadvantages are given.

**Linear Programming**

Invented in the second World War Linear Programming (LP) was developed for military planning purposes. Postwar it was widely used as a good general purpose code for all sorts of optimisation problems. The basic problem of LP is to minimise a linear objective function \( J \) (Equation 3.9) of continuous real variables, subjected to linear constraints (Equation 3.10, 3.11). Two methods can be derived to solve a LP problem. First, simplex methods introduced by Dantzig will try to find
3.4. **DRIVETRAIN ENERGY FLOW CONTROL**

basic solutions along the problem boundaries by reduction of the constraints to a square system. Barrier methods, by contrast, will search within the interior of the feasible region [Gregory, 2005].

$$J = \min_{x(t)} h^T x(t) + h_0$$

$$a \leq Ax(t) \leq b$$

$$Bx(t) = c$$

The system to be optimised is described by the vector $h$ and scalar $h_0$ in case of a linear system and additionally by matrix $H$ in case of a non-linear system, which will be discussed later. The constraint characteristics are given by matrix $A$ and $B$, while the constraints itself are given by vectors $a$, $b$, and $c$.

**(Sequential) Quadratic Programming**

The simplest way of solving an optimisation problem like the non-linear hybrid vehicle energy flow optimisation problem is by using Sequential Quadratic Programming (SQP) [Etman, 2006]. Basically this algorithm consists of Quadratic Programming (QP) sub-algorithms which minimise a quadratic approximation model $J$ (Equation 3.12) of the problem at each time step again subjected to linear constraints (Equation 3.13, 3.14). When using SQP this approximation is then used as a starting value for the next optimisation step [Koot, 2006]. SQP provides an efficient and robust way of optimisation but unfortunately does not take the fundamental dynamical behaviour of a hybrid drivetrain into account. This could result into frequent changing of hybrid operating modes which in practise leads to decreased drivability and even higher fuel consumption when taking starting behaviour of the internal combustion engine into account [Han, 2004].

$$J = \min_{x(t)} \frac{1}{2} x^T(t) H x(t) + h^T x(t) + h_0$$

$$a \leq Ax(t) \leq b$$

$$Bx(t) = c$$

**Dynamic Programming**

Since relevant information for the development of an EMS is hidden in the hybrid drivetrains dynamics the Dynamic Programming (DP) algorithm introduced by Bellman is interesting to use [Bellman, 1957]. This algorithm finds the global optimal control decision sequence by the knowledge that the most optimal decision made at one single moment is the most optimal decision made when considering the whole trajectory of decisions (Equation 3.15). These decisions are made on the basis of a (discredited) cost matrix which is defined by the use of an a priori known driving cycle and a proper vehicle model. Different linear and non-linear constraints (Equation 3.16, 3.17) can be taken into account when performing the DP routine. The accuracy of the final result depends on the resolution of the cost matrix and the step size of the control vector with the possible decisions. This however, significantly increases the computation time and can thus be undesirable.
\[ J = \min_{x(t)} \int_{t=0}^{t=\text{end}} L(x(t)) \, dx \]  

\[ a \leq A x(t) \leq b \]  

\[ B x(t) = c \]

Due to heavy computation time and the fact that the actual decisions made are non-causal, the results obtained by the use of the DP algorithm cannot be implemented in an online environment. Although, since the global optimal decision making sequence is found it can serve as a benchmark for other strategies or to gain a clear understanding of the potential performance of a hybrid drivetrain.

**Stochastic Dynamic Programming**

Stochastics are a good alternative to predict an uncertain factor like the future power demand of a driver when not considering a single a priori known driving cycle. The stochastic power demand is therefore determined from a set of representative driving cycles to be able to determine the chance what the future power demand, for a certain driving condition, will be when another demand has just occurred. A random Markov process can be used for this input when being implemented in the optimisation routine [Kim, 2005], [Lin, 2004]. The optimal powersplit as a function of the demanded power can be easily derived from the time-invariant infinite horizon optimisation over this stochastic random input denoted by \( Q \), which represents the expected value function (Equation 3.18). Again, like when performing a DP optimisation, several non-linear and linear constraints can be taken into account (Equation 3.19, 3.20).

\[ J = \min_{x(t)} Q(x(t)) \]  

\[ a \leq A x(t) \leq b \]  

\[ B x(t) = c \]

The advantages and disadvantages of the above mentioned optimisation algorithms are summarised in Table 3.3.
3.5 Combining methods

All different types of control and optimisation algorithms can be combined to find new approaches for the EMS development. Papers on EMS development found in literature describe the use of one or more control techniques in combination with one or more optimisation algorithms, in this way providing endless combinations for the control of energy flows in hybrid vehicles. Examples are the Rule Based Equivalent Consumption Minimisation Strategy (RB-ECMS) [Hofman, 2007b] which combines both the advantages of an Equivalent Consumption Minimisation Strategy (ECMS) and a RB controller. The ECMS strategy, which can be compared with a static optimal controller, has the advantage of using only one decision variable while relatively simple rules are used to predefine hybrid modes. Another example of a RB variation is the use of analytically obtained data from a DP optimal control strategy as a basis for a RB controller [Van Baalen, 2005]. In this way a DP trained RB controller is developed which has much better performance compared to a conventional RB controller. Improvement of a FL controller can be done by using the predictive information of a GPS signal and a road map in combination with the FL controller. FL rules are therefore extended with additional rules for the handling of a GPS signal in order to predict for example whether a downhill or uphill going road is ahead [Rajagopalan, 2002].

3.6 Analysis versus synthesis

Further research in this report describes the development of a tool for the analysis of parallel hybrid electric trucks and its applications. This analysis is done in order to gain understanding
and reasoning in the (theoretically) maximal fuel consumption reduction potential of a certain application. After analysing a certain vehicle application one might be interested in the implementation of the obtained results in the actual vehicle. The analysed results therefore need to be synthesised in the supervisory controller of the hybrid drivetrain.

3.7 Conclusions

Energy Management Strategies can be developed by using various techniques, the most interesting ones and the most used ones are discussed in this chapter. A set of these techniques will be chosen and used for further research.

In the first place a dynamic optimal control is desired for a proper analysis of the non-linear hybrid drivetrain behaviour and the transient cycles where these vehicle are subjected to. For the analysis of the potential of several hybrid electric truck applications the DP algorithm seems to suit the best. It provides the global numerical optimal solution over a predefined drive cycle which is desired since different applications can be compared in this way. For the same reason the DP solution can also be used as a benchmark for other EMS’s, which is not possible when using the other discussed algorithms. Computation time, mentioned as a disadvantage, can be shortened for relative analysis by reducing the step size of the design space. However, a high resolution design space is required when analysing in an absolute sense.

For the synthesis of the by DP obtained EMS in a vehicle model an intelligence controller will be used. A RB controller is chosen to fulfil this task. This is mainly done for the ease of its implementation and its accessibility when clear understanding of the EMS is desired. A RB strategy is also easy to compare with other EMS’s developed in other research.
Chapter 4

Vehicle Modelling

4.1 Introduction

This chapter describes the vehicle models which are built for the analysis of the longitudinal dynamic behaviour of the reference and hybrid vehicle. After the introduction of these vehicles the general power flow through their drivetrains will be discussed followed by the set of relevant parameters and drive cycles. The structure of the actual vehicle models will be described in Section 4.5.

4.1.1 Reference vehicle

The Base Line (BL) vehicle which is considered to be the reference vehicle in this study is the DAF FA CF65 4x2 distribution truck (Figure 4.1), developed and designed by DAF Trucks NV and assembled by Leyland Trucks Ltd. It forms the basis of many truck applications used for regional and city distribution purposes. The drivetrain of this vehicle consists of a conventional diesel Internal Combustion Engine (ICE), an automated clutch, a six speed automated manual transmission, and a final drive which is mounted in the rigid rear axle.

![Figure 4.1: DAF FA CF65 4x2 vehicle](image)

4.1.2 Hybrid vehicle

As explained in Chapter 2 the parallel hybrid electric drivetrain topology is the most general purpose and most promising concept when hybridising a distribution truck. The studied hybrid vehicle therefore has an AC permanent magnet motor/generator mounted between the clutch and the transmission. The energy accumulator is a Li-ion battery pack which is connected to the
motor/generator by means of an inverter which converts the DC current from the battery to AC current for the motor/generator and visa versa.

4.2 Power flow

The conventional reference vehicle has a power flow going one way, from the diesel fuel to the driven wheels. The hybrid vehicle however takes its advantages from the fact that power can flow from the wheels back to the drivetrain where it is stored in the battery for later use. The power flow through the diesel hybrid electric drivetrain is shown in Figure 4.2.

![Figure 4.2: power flow through a parallel hybrid electric drivetrain](image)

When putting $P_{mg} = 0$, the hybrid drivetrain from Figure 4.2 can be seen as a conventional drivetrain. In that case the fuel flow $\dot{m}_f$ is converted by the engine to mechanical power ($P_e = P_m$) which is used to propel the vehicle via the transmission and final drive (denoted as 'transmission') to overcome the vehicle load ($P_v$). This vehicle load can be positive as well as negative, the last one in case of decelerating. For the conventional drivetrain the negative power can be dissipated by the engine drag (when the clutch is closed) and by the mechanical brakes. When additional brake power is required the exhaust brake can be engaged by closing a valve in the exhaust pipe.

A part of the negative vehicle load from deceleration can also flow to the motor/generator when considering the hybrid drivetrain. In the summation point of Figure 4.2 the motor/generator power is therefore summed to the engine power in order to obtain the total mechanical power. The motor/generator power is chosen to be negative in Figure 4.2 for two reasons. The first reason is that within this report the hybrid drive unit is approached from the battery point of view. This means that an increasing, and therefore positive battery energy flow corresponds to the generator mode while a negative flow corresponds to the motor mode. The second reason has a relation to the optimisation of the hybrid vehicle control system and will be explained in Chapter 5. The mechanical power which is converted to $P_{bat,el(AC)}$ by the motor/generator, and to $P_{bat,el(DC)}$ by the inverter is stored in the battery as chemical energy $E_s$. Additional energy can be stored by converting $P_e$ to $E_s$ directly by injecting additional fuel. This energy can be used
later to propel the vehicle or to assist the engine when $P_{mg}$ is added to $P_e$. Each conversion in the described power flows has its own specific losses which finally will determine the most effective flow in every situation.

### 4.3 Component models

A vehicle model consists of several components which together determine the performance of the complete vehicle. Since not every mechanical detail of this vehicle can be translated in a mathematical model, a selection has to be made of the relevant components. These components are discussed in three sections; the general vehicle parameters (Section 4.3.1) which are the same for both the conventional and hybrid version of the CF65 vehicle, followed by the specific components of the conventional truck (Section 4.3.2) which cover the clutch and transmission unit. The hybrid drive unit and its components are discussed in Section 4.3.3. In Section 4.3.4 the parameters which are not related to the vehicle are given.

#### 4.3.1 General vehicle components

The basis of the considered truck is configured with the following components.

**Body and chassis**

The chassis is the backbone of a DAF truck where the axles, cabin, and cargo space are mounted on. The air drag coefficient $C_w$ and frontal area $A$ are determined by the body and cargo space shape. The Gross Vehicle Weight (GVW) is the total vehicle mass on the road including a full tank of fuel. Relevant dimensions are the wheelbase $wb$ and the height of the centre of gravity $h_{cg}$, which is a function of the GVW and the type of cargo. The type of cargo is an unknown variable, therefore $h_{cg}$ is assumed to be equal to 2 [m].

<table>
<thead>
<tr>
<th>Body and chassis: DAF FA CF65 4x2</th>
</tr>
</thead>
<tbody>
<tr>
<td>GGV (empty)</td>
</tr>
<tr>
<td>GGV (full)</td>
</tr>
<tr>
<td>$C_w$</td>
</tr>
<tr>
<td>$A$</td>
</tr>
<tr>
<td>$wb$</td>
</tr>
<tr>
<td>$h_{cg}$</td>
</tr>
</tbody>
</table>

**Internal combustion engine**

The vehicles primary power source is the Cummins ISBe 6 cylinder diesel ICE which has a maximum power of 138 [kW]. This engine is modelled by the use of (quasi-static) look-up tables for maximum performance and fuel consumption. The relevant performance characteristics and fuel consumption specifications of this Euro 3 specified engine are given in Table 4.2 and Figure 4.3. Where $T_{e,\text{max}}$ is the maximum engine torque and $T_{e,\text{aux}}$ is the auxiliary load torque for the steering-, water-, (brake) airpump, and alternator. The engine drag torque $T_{e,\text{drag}}$ is applied to
the ingoing transmission shaft when the throttle is closed and can act as a brake, this brake torque can be enlarged by the exhaust brake torque \( T_{e,\text{exh}} \). The exhaust brake is only used when driving down hill to prevent the mechanical brakes from overheating or when the maximum amount of available brake torque is required. This however will most likely not be the case for city distribution.

Figure 4.4 shows the fuel consumption rate \( \dot{m}_f \) of the engine as function of its speed and torque and is based on the obtained Brake Specific Fuel Consumption (BSFC) map. The fuel rate is calculated from the BSFC map by using Equation 4.1. The lowest BSFC values for each available engine speed are connected in the Optimal Operating Line (OOL) shown in Figure 4.3. The OOL is used as a basis for the shift strategy, which will be explained later.

\[
\dot{m}_f = \frac{BSCF \omega_e(t) T_e(t)}{3.6 \times 10^5}
\]  

(4.1)

where:

\[
\omega_e = \frac{n_e \pi}{30}
\]  

(4.2)

The data in Figure 4.3 and 4.4 is extrapolated in order to cover the complete speed and torque range of the ICE. The fuel map from Figure 4.4 is thereby constrained by the maximum engine speed and torque from Figure 4.3.

<table>
<thead>
<tr>
<th>Table 4.2: internal combustion engine parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Engine: Cummins ISBe 6 cylinder diesel</strong></td>
</tr>
<tr>
<td>( P_{e,\text{max}} )</td>
</tr>
<tr>
<td>( T_{e,\text{max}} )</td>
</tr>
<tr>
<td>( n_{e,\text{idle}} )</td>
</tr>
<tr>
<td>( n_{e,\text{max}} )</td>
</tr>
<tr>
<td>( J_e )</td>
</tr>
<tr>
<td>( \dot{m}_{f,\text{idle}} )</td>
</tr>
</tbody>
</table>
Figure 4.3: primary power source (partial) torque characteristics and OOL
\(\text{---} = T_{e,\text{max}}; \cdot \cdot \cdot = T_{e,\text{drag}}; \text{---} = T_{e,\text{aux}}; \text{---} = T_{e,\text{exh}}; \| = OOL\)

Figure 4.4: fuel consumption map
CHAPTER 4. VEHICLE MODELLING

Axles and final drive

The wheels of the truck are connected to the chassis by means of a front and a rear axle. The maximal and minimal allowed axle loads determine the maximal allowed GVW of the vehicle. A minimal load of 20 [%] of the total GVW on the front axle is required by law for safe braking [WVV, 1994]. However, a value of 36 [%] is chosen since the rear axle can handle 11500 [kg] of mass of the total maximal GVW which equals 18000 [kg]. In this case, for static situations, 64 [%] of the total maximal GVW leans on the rear axle while 36 [%] leans on the front axle.

![Figure 4.5: final drive crown and pinion gears (example)](image)

The final drive reduction $i_{fd}$ is integrated in the rear axle (Figure 4.5), it has a reducing ratio of 4.10 [–] and is for this reason suitable for city distribution purposes. The final drive efficiency is assumed to be at a constant level, but has a different value when the engine is driving the vehicle or when the engine is being driven during deceleration. Both situations are depicted by Figure 4.6. The efficiency of the final drive $\eta_{fd}$ when the wheels are driven by the prop-shaft (left) will be higher in comparison to the situation where brake energy is being absorbed by the engine or in case of the hybrid vehicle by the motor/generator during deceleration (right). The tooth flanges of the interacting gears (B) are in that case less smooth compared to (A) due to their unfinished and thus more rough surface. It should be noted that the influence of this effect on the gear ratio efficiency is based on assumptions, further research is therefore desired. Table 4.3 gives an overview of all relevant axle parameters.

![Figure 4.6: situation for constant speed and acceleration (left) and deceleration (right)](image)
### 4.3. COMPONENT MODELS

#### Table 4.3: axle parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{n,\text{front, max}}/g$</td>
<td>7500</td>
<td>kg</td>
</tr>
<tr>
<td>$%F_{n,\text{front}}$</td>
<td>36</td>
<td>%</td>
</tr>
</tbody>
</table>

**Front axle: DAF 152N "I" section rigid beam**

**Rear axle: DAF SR1132 single reduction drive axle**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{n,\text{rear, max}}/g$</td>
<td>11500</td>
<td>kg</td>
</tr>
<tr>
<td>$%F_{n,\text{rear}}$</td>
<td>64</td>
<td>%</td>
</tr>
<tr>
<td>$i_{fd}$</td>
<td>4.10</td>
<td>-</td>
</tr>
<tr>
<td>$\eta_{fd+}$</td>
<td>96</td>
<td>%</td>
</tr>
<tr>
<td>$\eta_{fd-}$</td>
<td>90</td>
<td>%</td>
</tr>
</tbody>
</table>

**Wheels and tyres**

The FA CF65 distribution truck has double 275-70R22.5 tyres and steel wheels on the driven rear axle. The wheels and tyres on the front axle are not considered since they are not driven and only the longitudinal dynamics are analysed.

#### Table 4.4: wheel and tyre parameters (rear axle)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_0$</td>
<td>0.4783</td>
<td>m</td>
</tr>
<tr>
<td>$R_e$</td>
<td>0.4702-0.4628</td>
<td>m</td>
</tr>
<tr>
<td>$J_w$</td>
<td>3.36</td>
<td>kg.m$^2$</td>
</tr>
<tr>
<td>$f_r$</td>
<td>0.8</td>
<td>%</td>
</tr>
</tbody>
</table>

Wheels and tyres: double 275-70R22.5

The unloaded wheel radius $R_0$ can be calculated by using the tyre size data which is usually written on the tyre walls. The 275-70R22.5 tyre is 275 [mm] wide and has a wall height of 70 [%] of that width while the tyre is suitable for a wheel size of 22.5 inches; this results in a static wheel radius of 0.4783 [m]. An increase of the trucks payload causes significant reduction of the wheel roll radius. For this reason the effective tyre roll radius is used in the vehicle models. This tyre radius is calculated by using the relevant tyre parameters from the available 315-80R22.5 Magic Formula (MF) tyre model [Pacejka, 2002]. The effective tyre roll radius is plotted in Figure 4.7 as a function of the GVW, the rear axle load is therefore assumed to be 64 [%] of the vehicles GVW, which ranges from empty (5540 [kg] or 5.54t) till fully loaded (18000 [kg] or 18t). Appendix A shows more details concerning the MF tyre model and its formulas for the calculation of $R_e$.

The rotational inertia of the driven wheels is a required parameter for the calculation of the total drivetrain inertia. This inertia is calculated by assuming the wheels to consist of several cylinders and discs made of different materials, e.g. steel and rubber. One single wheel consisting of a steel wheel and a rubber tyre has a rotational inertia $J_w$ of 3.36 [kg.m$^2$] and a mass of 29 [kg]. Note that the rear axle has twin wheels on each side which brings the total wheel inertia at 13.44 [kg.m$^2$]. Appendix B shows more detailed information about the wheel inertia calculation.
The roll resistance coefficient is also obtained from the 315-80R22.5 MF tyre model [Pacejka, 2002] and equals 8 [kg/ton] (0.8 [%]). The tyre roll resistance coefficient \( f_r \) is assumed to be independent from the vehicle load.

![Figure 4.7: effective tyre roll radius of the 275-70R22.5 tyre](image)

**4.3.2 Conventional drive unit**

The reference truck is equipped with an Automated Manual Transmission (AMT) built by ZF and is coupled to the engine by means of an automated, push type dry clutch. The electro-hydraulic actuated ZF 6AS850 AMT (Figure 4.8) has 6 forward ratios and one reverse ratio.

![Figure 4.8: ZF 6AS850 AMT](image)

By lack of information, the efficiency of each reduction \( \eta_{gb} \) is assumed to be equal to 96 [%] and the inertias are assumed to be zero since the rotating inertias in the gearbox are rather small compared to the inertias of the engine \( (J_e) \), clutch \( (J_c) \), wheels \( (J_w) \), and the vehicle \( (J_v) \) itself.
4.3. COMPONENT MODELS

Changing the gear ratio of the AMT takes a certain amount of time $t_{shift}$ which equals $0.9 \ [s]$ is case of the ZF unit. The specifications of the conventional drive unit components are listed in Table 4.5.

| $i_{gb}$ (i1; i2; i3; i4; i5; i6; r) | 8.51; 4.66; 2.73; 1.78; 1.27; 1.00; 7.64 | [-] |
| $\eta_{gb}$ | 96 | [%] |
| $t_{shift}$ | 0.9 | [s] |
| $J_c$ | 0.812 | [kg.m$^2$] |

4.3.3 Hybrid drive unit

The hybrid drive unit (Figure 4.9), developed and build by the Eaton Corporation, which is used for the parallel hybrid electric truck consists of both mechanical and electrical components. The relevant parameters of these components will be discussed separately in this section.

Figure 4.9: Eaton hybrid drive unit

Transmission and clutch

The used transmission has its origins at Allison Corporation and is automated by Eaton by means of an X-Y actuator (on the left of Figure 4.9). This transmission serves as a basis for the hybrid drive unit. The clutch is assumed to be the same as used for the reference drive unit. The transmission efficiency $\eta_{gb}$ is again assumed to be 96 [%], while the ratios and the shift time show small differences compared to the (conventional) ZF transmission. The parameter overview can be found in Table 4.6.
CHAPTER 4. VEHICLE MODELLING

Table 4.6: transmission and clutch parameters

<table>
<thead>
<tr>
<th>Hybrid transmission: Allison/Eaton AMT</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$i_{gb}$ (i1; i2; i3; i4; i5; i6; r)</td>
<td>7.05; 4.13; 2.52; 1.59; 1.00; 0.78; 6.75</td>
</tr>
<tr>
<td>$\eta_{gb}$</td>
<td>96</td>
</tr>
<tr>
<td>$t_{shift}$</td>
<td>1.4</td>
</tr>
<tr>
<td>$J_c$</td>
<td>0.812</td>
</tr>
</tbody>
</table>

Motor/generator and power electronics

The motor/generator is mounted between the clutch and the transmission (on the right of Figure 4.9). This electric machine has a permanent magnet rotor and works with AC current which is provided by the power electronics. The efficiency of the motor/generator is determined by magnetic core losses (caused by hysteresis and eddy-currents), winding power losses (caused by heating), friction and drag losses (caused by bearing friction and rotor air drag), and stray-load losses which are all other load-dependent losses. The reader is referred to [Mohan, 2003] for a more detailed explanation of these losses.

Because of IP reasons a scaled efficiency map, based on a 58 [kW] PM machine [Menne, 1998], is used to represent the efficiency map of the motor/generator of the hybrid drive unit. This efficiency map incorporates the power electronics losses, which itself are assumed to be 95 [%]. The efficiency map is scaled to the known maximum torque, maximum speed, and base-line speed. Efficiencies for low speeds and torques are obtained by extrapolation towards zero since measurement data is usually not available in these regions of the efficiency map. The rotor inertia $J_{mg}$ is proportionally downscaled to the maximum power $P_{mg}$ of 44 [kW]. Figure 4.10 shows the scaled efficiency map with the maximum torque lines. It is assumed that the electric machine is able to operate at this maximum torque level without any time constraint.

The mechanical power of the motor/generator can now be calculated as a function of the electrical power $P_{mg,e}$ by Equation 4.3 (motor mode) and by Equation 4.4 (generator mode).

$$T_{mg}\omega_{mg} = P_{mg,e}\eta_{mg}(T_{mg}, \omega_{mg})$$ (4.3)

$$T_{mg}\omega_{mg} = \frac{P_{mg,e}}{\eta_{mg}(T_{mg}, \omega_{mg})}$$ (4.4)

where:

$$\omega_{mg} = \frac{n_{mg}\pi}{30}$$ (4.5)
4.3. COMPONENT MODELS

The characteristics of the motor/generator and power electronics are summarised in Table 4.7.

<table>
<thead>
<tr>
<th>Table 4.7: motor/generator and power electronics parameters</th>
<th>Eaton 44 [kW] PM AC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{mg,\text{max}}$</td>
<td>420 [Nm]</td>
</tr>
<tr>
<td>$T_{mg,\text{nom}}$</td>
<td>252 [Nm]</td>
</tr>
<tr>
<td>$P_{mg,\text{max}}$</td>
<td>44 [kW]</td>
</tr>
<tr>
<td>$P_{mg,\text{nom}}$</td>
<td>26 [kW]</td>
</tr>
<tr>
<td>$n_{mg,\text{max}}$</td>
<td>2600 [rpm]</td>
</tr>
<tr>
<td>$n_{mg,\text{base}}$</td>
<td>1000 [rpm]</td>
</tr>
<tr>
<td>$\eta_{mg}$</td>
<td>40-90 [%]</td>
</tr>
<tr>
<td>(including inverter)</td>
<td></td>
</tr>
<tr>
<td>$J_{mg}$</td>
<td>0.0178 [kg.m$^2$]</td>
</tr>
</tbody>
</table>

Battery

The Eaton Power Electronics Carrier (PEC) is a battery pack which consists of four Shinko-De-ki Li-ion modules (Figure 4.11). Each of these four modules consists of 48 Mn-type Li-ion cells which are rated at 3.6 [V], 3.6 [A] [Shinko-De-ki, 2000]. The Mn-type Li-ion cell uses manganese materials for the positive electrode instead of cobalt which is done for cost reasons. The PEC consists of two parallel strings of two in series connected modules resulting in a total voltage of 346 [V] at 7.2 [A] and a total nominal capacity of 2.5 [kWh].
The battery efficiency map is build by reconstructing the battery configuration from Figure 4.11 by using the Advisor Li-ion battery model [NREL, 2002]. This model contains information about the Saft 6 [A] Li-Ion cell which is assumed to have equal characteristics compared to the Shinkobe-Denki cell. The Saft cell characteristics are modified in such way that they represent two parallel Shinkobe-Denki cells, 48 of such cells now represents the PEC. Figure 4.12 depicts the equivalent battery circuit where the battery model is based on. The stored battery power $P_{\text{bat},s}$ can be described by Equation 4.6, while the electrical power $P_{\text{bat},e}$ which is send to and from the electric machine can be described by Equation 4.7 and 4.8 for discharging and charging respectively, these formulas include the battery losses. The Coulombic losses (irreversible losses and losses from the Peukert effect\(^2\)) are taken into account by using a constant efficiency rate of 90 [%].

\[ P_{\text{bat},s} = V_{oc}I_{\text{bat}} \]  (4.6)

\[ P_{\text{bat},e} = V_{oc}I_{\text{bat}} + I_{\text{bat}}^2R_i \]  (4.7)

\[ P_{\text{bat},e} = V_{oc}I_{\text{bat}} - I_{\text{bat}}^2R_i \]  (4.8)

The battery efficiency curve obtained from the Advisor model is plotted as a function of the stored chemical power and the electric charge/discharge power (Figure 4.13). The upper right quadrant represents the motor mode, the lower left quadrant the generator mode. The battery simulations

\(^2\)Effect that describes the battery capacity loss for high discharge rates
are carried out for three available temperatures (273 [K], 298 [K], 313 [K]) of which the results are depicted by the circles plotted in the vertical direction of Figure 4.13. The effect of these temperatures on the battery efficiency is assumed to be small and therefore neglected for further modelling. A quadratic fit through the simulated operating points provides a good estimation of the battery efficiency. This quadratic fit is characterised by the following coefficients ([2nd 1st order]) for both discharging and charging (Table 4.8).

![Figure 4.13: quadratic battery efficiency](image)

\( (\circ = \text{simulated operating points}; \quad \longrightarrow = \text{quadratic fit}; \quad \ldots = 100 \, \% \text{ efficiency}) \)

<table>
<thead>
<tr>
<th>Table 4.8: battery efficiency coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td>discharging:</td>
</tr>
<tr>
<td>[1.291e-3; 1.024]</td>
</tr>
<tr>
<td>charging:</td>
</tr>
<tr>
<td>[2.571e-3; 1.011]</td>
</tr>
</tbody>
</table>

The battery energy content is denoted as the State Of Energy (SOE) of the battery which is the relative amount of energy available in the battery. The total amount of energy in the battery, or 100 [\%] SOE equals the nominal battery capacity \( C_{\text{nom}} \). It is recommended to constrain the upper and lower SOE levels in order to prevent the battery from depletion or overload and to have the battery operating in its highest efficiency region. Since the battery efficiency is strongly dependent on the internal resistance of a single cell, it can be used to determine these constraints. Figure 4.14 therefore shows the internal cell resistance as function of the battery SOE. It can be concluded that the SOE range from 30 [\%] till 70 [\%] has the lowest internal resistance for both charging and discharging. This range is thus most suitable for bordering the energy level in the battery. The influence of the SOE on the battery efficiency can be neglected since the internal cell resistance \( R_i \) is small and constant in the proposed SOE range.
CHAPTER 4. VEHICLE MODELLING

Figure 4.14: internal cell resistance $R_i$ as function of battery SOE
($\circ = \text{available data}; - = \text{discharging}; - - - - = \text{charging}$)

For both motor mode and generator mode, the amount of electrical power involved can be calculated as a function of the stored chemical power $P_{bat,s}$ (Equation 4.9 and 4.10). This chemical power represents an amount of stored energy $Q$ which increases or decreases the SOE (Equation 4.12) and is constrained by a maximum charge and discharge power ($P_{ch,max}, P_{dis,max}$). The maximum charge power, which is relevant when brake energy is recuperated, seems to be (slightly) lower than the maximum (equivalent) charge power of the motor/generator.

$$-P_{mg,e} = P_{bat,c} = P_{bat,s}\eta_{bat,dis}(P_{bat,s})$$  \hspace{1cm} (4.9)

$$-P_{mg,e} = P_{bat,e} = \frac{P_{bat,s}}{\eta_{bat,chg}(P_{bat,s})}$$  \hspace{1cm} (4.10)

where:

$$P_{dis,max} \leq P_{bat,s} \leq P_{ch,max}$$  \hspace{1cm} (4.11)

$$SOE(t) = \frac{Q(t)}{Q_{nom}} \times 100[\%]$$  \hspace{1cm} (4.12)

where:

$$Q(t) = Q(0) + \int_{0}^{t} P_{bat,s}(t) \, dt$$  \hspace{1cm} (4.13)

$$30[\%] < SOE(t) < 70[\%]$$  \hspace{1cm} (4.14)
Table 4.9 gives the overview of the relevant battery parameters which are used in the simulation models.

<table>
<thead>
<tr>
<th>Battery: Eaton PEC</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{\text{max}}$</td>
<td>3.74 [kWh]</td>
</tr>
<tr>
<td>$C_{\text{nom}}$</td>
<td>2.5 [kWh]</td>
</tr>
<tr>
<td>$P_{\text{ch, max}}$</td>
<td>32 [kW]</td>
</tr>
<tr>
<td>$P_{\text{dis, max}}$</td>
<td>-50 [kW]</td>
</tr>
<tr>
<td>$SOE_{\text{max}}$</td>
<td>70 [%]</td>
</tr>
<tr>
<td>$SOE_{\text{min}}$</td>
<td>30 [%]</td>
</tr>
<tr>
<td>$\eta_{\text{bat, dis}}$</td>
<td>92.0 ($P_{\text{dis, max}}$) [%]</td>
</tr>
<tr>
<td>$\eta_{\text{bat, ch}}$</td>
<td>92.2 ($P_{\text{ch, max}}$) [%]</td>
</tr>
</tbody>
</table>

For further research it is recommended to consider the battery State Of Health (SOH) besides the battery SOE, since the battery performance characteristics deteriorate in time. The battery SOH gives an indication of the condition of the battery by considering charge acceptance, internal resistance, voltage, and self-discharge [Blinkhorne, 2005]. It is a subjective measurement which is derived from a variety of different (measurable) battery performance parameters. An alternative method of estimating the battery SOH can be obtained by using logged data from the battery usage history.

### 4.3.4 Non-vehicle related parameters

The environment where a vehicle is subjected to also has its influence on the behaviour of the drivetrain; the parameters in Table 4.10 [Leijendeckers, 2006] are relevant for the analysis of the vehicle's fuel consumption.

<table>
<thead>
<tr>
<th>Non-vehicle related parameters</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$g$</td>
<td>9.81 [m/s$^2$]</td>
</tr>
<tr>
<td>$\rho_{\text{air}}$</td>
<td>1.199 (T=293 [K]) [kg/m$^3$]</td>
</tr>
<tr>
<td>$\rho_{\text{fuel}}$</td>
<td>835 (T=293 [K]) [kg/m$^3$]</td>
</tr>
<tr>
<td>$H_{\text{low}}$</td>
<td>42.5 [MJ/kg]</td>
</tr>
</tbody>
</table>
4.4 Drive cycles

Three drive cycles with different characteristics are used for the drivetrain analysis. These cycles prescribe the desired vehicle speed and road grade as a function of time. All three cycles are interpolated quadratically and re-sampled at a frequency of 10 [Hz] for an equal comparison, an acceptable calculation time, and to guarantee smoothness. Weather influences are not taken into account and the road is assumed to be straight without any corners.

**Urban Traffic Eindhoven Cycle**

Distribution trucks regularly drive in urban environments, therefore the Urban Traffic Eindhoven Cycle (UTEC) with its low average speed and slow accelerations is chosen to serve as the city representative cycle (Figure 4.15). This cycle, provided by DAF Trucks NV, represents a physical route through the city centre of Eindhoven. The UTEC cycle also provides information on road gradability. Although these grades are very small they make a significant contribution to the total road load force when considering heavy vehicles.

![Figure 4.15: Urban Traffic Eindhoven Cycle](image1)

**TNO Dynamometer Cycle**

The UTEC cycle has opposite characteristics when compared to the TNO Dyno Cycle (TDC) (Figure 4.16). This cycle is much more aggressive, which means that accelerations and decelerations are at a much higher level. The prescribe speeds are also higher while they occur at a longer period in time, which implicates highway driving. The TDC cycle is developed by TNO Automotive by logging time and vehicle speed, by means of a GPS system, along a route in and around Eindhoven. The TDC cycle is obtained by compressing the logged cycle to a time scale which is more suitable for dynamometer tests. Compression is done without losing any of the cycle’s characteristics.

![Figure 4.16: TNO Dynamometer Cycle](image2)
FTP-75 Cycle

Consisting of both a transient phase and a highway phase, the Federal Test Procedure 75 cycle (FTP) [DieselNet, 2000] can be positioned between the TDC and UTEC cycle. The FTP cycle (Figure 4.17) is developed for the emission certification of light duty vehicle in the United States. For above reasons the FTP cycle is found to be a valuable contribution to the set of considered cycles.

![FTP-75 cycle graph](image)

Figure 4.17: FTP-75 cycle

The characteristics of the drive cycles are plotted in Figure 4.18 and 4.19. The typical characteristics of all discussed cycles are summarised in Table 4.11. The effect of different drive cycles on the hybrid truck will be analysed in Chapter 6.

<table>
<thead>
<tr>
<th>Cycle length</th>
<th>Urban Cycle</th>
<th>Traffic Cycle</th>
<th>TNO Dyno Cycle</th>
<th>FTP-75 Cycle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cycle duration</td>
<td>12.35</td>
<td>18.10</td>
<td>17.77</td>
<td></td>
</tr>
<tr>
<td>Standstill</td>
<td>1939</td>
<td>1574</td>
<td>1876</td>
<td></td>
</tr>
<tr>
<td>Average speed</td>
<td>8.7 (169)</td>
<td>11.4 (180)</td>
<td>18.1 (339)</td>
<td></td>
</tr>
<tr>
<td>Maximal speed</td>
<td>22.8</td>
<td>41.4</td>
<td>34.1</td>
<td></td>
</tr>
<tr>
<td>Maximal acceleration</td>
<td>+1.2</td>
<td>+1.6</td>
<td>+1.9</td>
<td></td>
</tr>
<tr>
<td>Maximal deceleration</td>
<td>-1.6</td>
<td>-2.5</td>
<td>-1.9</td>
<td></td>
</tr>
<tr>
<td>Maximal grade</td>
<td>+3.03</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Minimal grade</td>
<td>-2.98</td>
<td>0</td>
<td>0</td>
<td></td>
</tr>
</tbody>
</table>
Figure 4.18: cycle speed histogram
\( (-- = UTEC; \quad \cdots \cdots = TDC; \quad \cdots = FTP) \)

Figure 4.19: cycle acceleration histogram
\( (-- = UTEC; \quad \cdots \cdots = TDC; \quad \cdots = FTP) \)
4.5 Vehicle models and supervisory control

The first vehicle model to be build represents the reference FA CF65 truck. This forward facing (FW) vehicle model describes the longitudinal dynamic vehicle behaviour and includes a driver model. The reference model is validated by using DAF V40 simulation data and serves as a basis for the hybrid vehicle model. The vehicle simulation models are built using a MatLab m-script.

4.5.1 Reference vehicle model

The reference truck is modelled as a face forward or integrating model (Figure 4.20) which determines the actual vehicle speed by using a P controller as the driver representative.

\[ T_{load}(t) = \frac{1}{2}AC_{m}P_{atm}v(t)^2 + mgsin(\alpha(t))R + T_{e,aux}i_{gb}(t)i_{fd} + T_{pto}i_{gb}(t)i_{fd} \]  
\[ \omega_{v}(t) = \frac{T_{m}(t)\eta_{gb}\eta_{fd} + i_{gb}(t)i_{fd} - T_{load}(t)}{J_{eq}(t)} \]  
\[ \dot{x}_{act}(t) = \omega_{v}(t) \]  
\[ J_{eq}(t) = J_{v} + (J_{e} + J_{c})(i_{gb}(t)i_{fd})^2 + 4J_{w} \]  

Multiplied by the effective wheel radius \( R_e \), the integration of Equation 4.16 and 4.17 gives the actual vehicle speed \( v_{act} \) (Equation 4.19) which becomes the input variable of the driver model, and the actual driven distance \( x_{act} \) (Equation 4.20).

\[ v_{act}(t) = R_e \int_{t=0}^{t=end} \omega_{v}(t) \, dx \]
Finally, the engine speed is calculated by multiplying the angular wheel speed by the actual transmission ratio and the final drive ratio (Equation 4.21). The engine speed is constrained by the idle speed and the maximum engine speed.

\[ \omega_e(t) = \omega_v(t) i_{gb}(t)f_{fd} \]  

where:

\[ \omega_{e,\text{idle}} < \omega_e(t) < \omega_{e,\text{max}} \]  

### 4.5.2 Reference vehicle supervisory control

In order to gain drive torque to overcome the applied road load forces a driver model is introduced. Equation 4.23 represents the driver watching his speedometer \( v_{\text{act}} \) and compares it with the reference, desired vehicle speed \( v_{\text{ref}} \) imposed by the drive cycle. The difference is compensated by pressing the throttle paddle (under the condition that the clutch is closed), or by pushing the brakes both with an amplification factor \( p \). The resulting output of the driver model is the mechanical propulsion power at the ingoing transmission shaft \( P_m \).

\[ P_m(t) = p(v_{\text{ref}}(t) - v_{\text{act}}(t)) \]  

The shift strategy of the gearbox is designed in such way that it will try to keep the engine running in the most efficient operating area. The optimal engine speed \( \omega_{\text{opt}} \) is therefore determined by the (positive) demanded power \( P_m \) and the Optimal Operating Line (OOL), introduced in Section 4.3.1. For negative (brake) power, \( \omega_{\text{opt}} \) is chosen to be \( 1000 \ [\text{rpm}] \) \( (105 \ [\text{rad}]/\text{s}) \) in order to have the maximum engine torque available for accelerating at all times. The value of \( \omega_{\text{opt}} \) can also be chosen in such way that, in case of the hybrid vehicle, the motor/generator will run in its optimal area \( (1300 - 2150 \ [\text{rpm}], 136 - 225 \ [\text{rad}]/\text{s}] \) while brake energy is being recovered (this will be discussed later). The optimal gearbox ratio \( r_{\text{opt}} \) is now determined by Equation 4.24. This ratio is however continuous and should be discretised with respect to the (nearest) available gearbox ratios.

\[ r_{\text{opt}} = \frac{\omega_{\text{opt}}}{\omega_{v,f_{fd}}} \]  

Shifting from one ratio to another is subjected to a time constraint which imposes that a new ratio cannot be selected within the period \( t_{\text{shift\_lock}} \ (1.5 \ [\text{s}]) \) after the last ratio change. This is done to prevent the drivetrain from the undesired and unrealistic up/down shifting between two values at a high frequency. The comfort aspect which is related to \( t_{\text{shift\_lock}} \) is beyond the scope of this research. The chosen ratio is also constrained by the resulting engine speed which may not exceed the maximum engine speed \( \omega_{e,\text{max}} \). Finally the suggested ratio \( r \) is used. The actual ratio change takes a certain amount of time \( t_{\text{shift}} \), which is determined by the gearbox technology.

The amount of torque demanded by the driver \( T_m \) is used as an input for the vehicle model to overcome the road load forces and can be calculated directly from \( \omega_e \) and \( P_m \). \( T_m \) is constrained by the maximum engine torque or slip torque (Equation 4.25). The slip torque (at the ingoing
4.5. VEHICLE MODELS AND SUPERVISORY CONTROL

transmission shaft) is determined by the friction coefficient $\mu$ of the tyres and the mass transfer during accelerating or decelerating.

\[ T_{m,\text{min}}(t) < T_m(t) < T_{e,\text{max}}(\omega_e) \mid T_{m,\text{max}}(t) \]  

(4.25)

where:

\[ T_{m,\text{min}}(t) = -\frac{mg\mu R}{i_{gb}(t)i_{fd}}((1 - \%F_{n,\text{front}}) + \frac{\dot{\omega}_v(t)R\bar{h}_{cg}g}{wb} \eta_{gb}\eta_{fd}^-) \]  

(4.26)

\[ T_{m,\text{max}}(t) = -\frac{mg\mu R}{i_{gb}(t)i_{fd}}((1 - \%F_{n,\text{front}}) + \frac{\dot{\omega}_v(t)R\bar{h}_{cg}g}{wb} \eta_{gb}\eta_{fd}^+) \]  

(4.27)

The resulting engine torque $T_e$ (Equation 4.28) and brake torque $T_b$ (Equation 4.29) on the ingoing transmission shaft are now determined by the positive and negative drive torque respectively.

\[ T_e(t) = T_m(t) \mid T_m(t) > 0 \]  

(4.28)

\[ T_b(t) = T_m(t) \mid T_m(t) < 0 \]  

(4.29)

Realistic driver behaviour is obtained by tuning the $p$ controller. A too soft driver will result in not reaching the desired speed, while a too aggressive driver results in overshooting the reference vehicle speed or even instabilities. The driver has been tuned for the TDC cycle which represents the average use of the vehicle in question. A series of simulations is performed for a range of GVW and $p$ gain values. The GVW ranges from the mass of an empty vehicle (5540 [kg]) till a fully loaded vehicle (18000 [kg]).

The optimal $p$ gain as function of vehicle weight is determined by the criterion that the trackability $|\bar{\Delta}v|$ needs to be smaller than 1 [km/h] (0.2778 [m/s]). The trackability indicates how well the vehicle is able to follow the desired drive cycle speed. For high GVW the vehicle might not be capable of reaching the desired cycle at all, for these cases the optimal $p$ gain is the highest possible gain where instabilities do not occur.

The results of the tuning procedure can be found in Figure 4.21 and Figure 4.22 where $|\bar{\Delta}v|$ and the resulting fuel consumption $m_f$ are plotted as function of the GVW and $p$ gain value respectively. The circles mark the optimal value of $p$ depending on whether overshooting occurs or the desired trackability is reached (Figure 4.21). The small bend which can be noticed in each line indicates overshooting behaviour of the driver $p$ controller. As can be seen in Figure 4.22 the fuel consumption reached by the optimal $p$ gains is almost saturated.

The optimal $p$ gain is plotted as function the vehicle GVW in Figure 4.23, the obtained results from the tuning procedure are summarised in Table 4.12. Linear interpolation is used for the actual implementation in the vehicle model.
Figure 4.21: p gain tuning (trackability)
\((-\circ-) =\) optimal p gain; \(\circ =\) simulation data

Figure 4.22: p gain tuning (fuel consumption)
\((-\circ-) =\) optimal p gain; \(\circ =\) simulation data
4.5. VEHICLE MODELS AND SUPERVISORY CONTROL

![Figure 4.23: optimal p gain as function of GVW (○ = simulation data)](image)

Table 4.12: driver model parameters

<table>
<thead>
<tr>
<th>Driver model</th>
<th>Optimal p gain</th>
<th>GVW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>12e4; 14e4; 19e4; 25e4; 29e4; 30e4</td>
<td>5540; 7500; 10000; 12000; 15000; 18000 [kg]</td>
</tr>
</tbody>
</table>

4.5.3 Reference vehicle validation

The simulation results from the base line vehicle model need to be verified on trustworthiness. In order to do this manufacturer data obtained from the V40 simulation tool (Appendix C) are compared with the simulation results from the MatLab reference model. The V40 simulation tool is used by DAF Trucks NV to predict fuel consumption and performance of their vehicles.

The UTEC cycle is used to validate the reference vehicle on vehicle speed and fuel consumption. The simulation is done for a GVW of 12000 [kg] since this mass represents the average loaded vehicle. The relative deviation of the cumulative fuel consumption \( m_f \) as a function of time of the V40 data minus the results from the BL vehicle model are plotted in Figure 4.24. The dash-dotted lines represent the general accepted maximum deviation of +/- 1 [%]. The difference of the total amount of fuel used by the BL vehicle model and the amount of fuel given by the V40 data at the end of the UTEC cycle is found to be nil.
The same has been done for the vehicle speed which is not allowed to deviate more than ±1 km/h (+/− 0.2778 m/s), marked again with dash-dotted lines in Figure 4.25a. The average deviation equals -0.1017 m/s. The high peaks in Figure 4.25 can be ascribed to differences in the shift strategy; as a consequence the gear ratio change is not synchronous for the BL vehicle model and the V40 data. Figure 4.25b shows an enlarged view on the largest speed deviation together with the shift timing of the base line vehicle model and the V40 data.
4.5.4 Simulation results reference vehicle

Figure 4.26 shows the fuel consumption of the base line vehicle driving the TDC cycle for different vehicle masses ranging from empty (5540 [kg]) till full (18000 [kg]), a linear relation can be noticed.

![Figure 4.26: base line fuel consumption for TDC cycle](image)

*Figure 4.26: base line fuel consumption for TDC cycle*

*(\(\circ\) = simulation data)*
4.5.5 Hybrid vehicle model

The hybrid vehicle model (Figure 4.27) is built using the reference vehicle model as a basis. All vehicle components and equations remain the same except the conventional drive unit which is replaced by the hybrid drive unit. The driver model remains the same since the drivability of the hybrid vehicle is not allowed to differ from the reference vehicle. For the same reason the shift strategy is kept unchanged, the gearbox ratio is thus in all cases determined by the driver power demand $P_m$.

![Figure 4.27: face forward hybrid vehicle model structure](image)

4.5.6 Hybrid vehicle supervisory control

The task of the EMS is to determine the contribution of the electrical path to the total energy flow through the drivetrain. It therefore controls the engine, clutch, transmission, and the power electronics (inverter). Continuous control is performed by the EMS in the sense of a powersplit ratio between the electric motor and the engine (motor assist) or generator and brakes (brake blending). A series of discrete control variables can also be distinguished as being the hybrid modes. The development of the EMS itself is discussed in Chapter 5.

**Motor only mode**

The motor/generator can be used for pure electric driving in the Motor only (M) mode when the engine is shut off and the clutch is opened. In this situation the electric motor can move the vehicle over a limited distance at a limited speed. The M mode is described by Equation 4.30.

$$P_m = P_{mg}$$ (4.30)

**Brake Energy Recovery mode**

The greatest advantage of a hybrid drivetrain is the ability of recovering the normally wasted energy which is released while decelerating. In the Brake Energy Recovery (BER) mode, energy is stored in the battery and is used later when the primary power sources is running at a low efficiency level. It is therefore desired to recuperate as much energy as possible. For this reason
4.5. VEHICLE MODELS AND SUPERVISORY CONTROL

The rear axle, which is connected to the generator, provides all brake power required for decelerating the vehicle until the maximum generator power is reached. The remaining demanded brake power is supplied by the front brakes in the first place while the rear brakes provide additional brake power \( (P_b) \) when required (Equation 4.31). As a suggested safety feature, e.g. when the vehicle is slipping, the generator should shut off, while the mechanical brakes take care for a controlled deceleration of the vehicle by using the Anti-lock Brake System (ABS). Governmental legislations should however comply with this.

\[
P_m = P_{mg} + P_b \quad (4.31)
\]

where:

\[
P_b = 0 \quad | \quad P_m \geq P_{mg,min} \quad (4.32)
\]

Brake energy can be recovered by the hybrid drive unit in four different ways. These four BER strategies (Table 4.13) are distinguished by closing the clutch or opening the clutch while braking and shutting down the engine or not. In case of BER strategy 1 and 2, when the engine is stopped for some moments, the engine auxiliaries are temporarily not driven. This is not a problem for the alternator and the water pump of the cooling system. The pressure in the expansion tank of the pneumatic brake system however, needs to be kept at a sufficient level. Also the steering pump is required when the vehicle is driving, while the clutch is opened and the engine is shut off. It is recommended to examine the exact power demand for these auxiliaries in order to have them electrified to eliminate this problem.

Table 4.13 provides an overview of how the different BER strategies are implemented and what its consequences are. The highest fuel consumption reduction potential is expected from the first strategy, while the lowest is expected from the 3rd or 4th strategy since the engine is idling at standstill in these cases. The sensitivity of the different BER strategies on fuel consumption reduction is studied in Chapter 5.

### Table 4.13: brake energy recovery strategies

<table>
<thead>
<tr>
<th>BER strategy</th>
<th>Clutch actuation while BER</th>
<th>Engine while BER</th>
<th>Engine while standstill</th>
<th>Drag torque while BER</th>
<th>Auxiliary power while BER</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Open</td>
<td>Engine stop</td>
<td>Engine stop</td>
<td>No</td>
<td>If clutch closed</td>
</tr>
<tr>
<td>2</td>
<td>Closed</td>
<td>Fuel cut-off</td>
<td>Engine stop</td>
<td>Yes</td>
<td>If clutch closed</td>
</tr>
<tr>
<td>3</td>
<td>Closed</td>
<td>Fuel cut-off</td>
<td>Engine idle</td>
<td>Yes</td>
<td>Conventional</td>
</tr>
<tr>
<td>4</td>
<td>Open</td>
<td>Engine idle</td>
<td>Engine idle</td>
<td>No</td>
<td>Conventional</td>
</tr>
</tbody>
</table>

**Charge mode**

The Charge (CH) mode is introduced to provide additional energy to be stored in the battery and can be activated when the engine delivers more power than demanded by the driver (Equation 4.33). This should be done when the engine efficiency is sufficiently high.

\[
P_m = P_e + P_{mg} \quad | \quad P_{mg} < 0 \quad (4.33)
\]
CHAPTER 4. VEHICLE MODELLING

Motor Assist mode

The Motor Assist (MA) mode can fulfil two tasks. It can be used to assist the engine to increase its efficiency by lowering $P_e$ (Equation 4.34). Besides this it can also boost the vehicle in order to increase its drivability. This research, which focuses on fuel consumption reduction, only considers the assist mode.

$$P_m = P_e + P_{mg} \mid P_{mg} > 0$$  \hspace{1cm} (4.34)

Idle Stop

At standstill, as well as for some BER strategies while decelerating, the engine can be stopped to prevent it from idling. The Idle Stop (IS) mode can be activated at all times by simply opening the clutch and by cutting the fuel line. Because additional fuel is required for starting the engine it is recommended to consider a time constraint for the engine stop mode. This engine starting behaviour should be investigated in future research.

4.6 Conclusions

Based on the available data two vehicle models have been built. One represents the parallel hybrid electric distribution truck while the other represents the base line CF FA65 4x2 vehicle. The purpose of the vehicle models is mainly to calculate fuel consumption, therefore only the longitudinal dynamics are taken account for. The base line vehicle has a calculated fuel consumption rate ranging from 16.4 till 36 [l/100 km] for the TDC cycle. The fuel consumption of the hybrid vehicle is studied in Chapter 5 and 6.

4.7 Recommendations

The shift strategy of both the base line and hybrid vehicle are kept the same since they are both based on the total demanded drive power. For future research it is recommended to optimise the shift strategy within the DP algorithm as well. Additional savings, however presumably small are expected when the shift strategy is optimised for the hybrid vehicle application.

The motor/generator and battery efficiency map are based on several assumptions because of the unavailability of the original data. For future research these efficiency maps should be revised in order to increase their accuracy. Also the extension of the battery efficiency map with a battery SOH computation will be valuable in order to investigate the ageing process and its effect on the battery performance. Regarding the engine characteristics it is desired to include the dynamic behaviour concerning fuel consumption in the engine model.
Chapter 5

Analysis and Synthesis of the EMS

5.1 Introduction

This chapter describes the Dynamic Programming (DP) optimisation algorithm and its use for the analysis of the hybrid trucks EMS. The DP algorithm is therefore integrated in a mathematical tool which provides the global optimal solution for the control of the energy flow through the parallel hybrid electric drivetrain for different truck applications. This optimal control solution will give an indication of the potential amount of fuel that can be saved by the hardware. Since these results can only be obtained theoretically, an online implementable control solution will be suggested.

5.2 Energy management analysis: Dynamic Programming

As concluded from Chapter 3, the DP algorithm is most suitable for the analysis of the hybrid drivetrain EMS. This section explains the DP principle and discusses the obtained results.

5.2.1 Dynamic Programming working principle

According to Bellman’s principle of optimality a DP routine [Bellman, 1957] finds the most (cost) efficient route from one end of a matrix to the other by keeping in mind that from any point on the optimal trajectory, the remaining trajectory is optimal for the corresponding trajectory initiated at that point. Equation 5.1 gives the mathematical representation of the DP problem. In order to minimise cost function $J_f$ (which represents the total amount of fuel used by the vehicle) the optimal control sequence of variable $x(t)$ (which represents the battery energy content) is found over a time horizon $0 < t < t_{end}$. The optimisation is carried out by taking several component constraints, discussed in Chapter 4, into account.

$$J_f = \min_{x(t)} \int_{t_0}^{t_{end}} \dot{m}_f(x(t)) \, dx$$  \hspace{1cm} (5.1)

The implementation of the optimisation problem is done as follows. Two matrices are built; the state space grid $S$ representing the (discretised) battery energy content over time and the cost matrix $C$ representing all possible instantaneous costs for each possible battery energy content along the pre-defined drive cycle. The values of the cost matrix $C$ are in terms of fuel consumption rate $\dot{m}_f$. While performing the DP routine a third matrix is build, the cost-to-go matrix $L$. 

73
This matrix contains all possible cumulative cost routes through the cost matrix and within the feasible domain of the state space grid from $t = 0$ till $t = t_{\text{end}}$. Finally one route will lead to the lowest cumulative cost at $t = t_{\text{end}}$ which is the theoretical lowest amount of fuel required for the analysed vehicle driving over the pre-defined drive cycle. The coordinates of this most optimal route represent the optimal control sequence of $x(t)$ and the corresponding optimal battery energy content $P_{s,\text{opt}}(t)$. The remaining of this section describes the mathematical background of the above mentioned computation schemes.

**State space grid**

The feasible domain of the optimisation problem is represented by the state space grid $S$. This grid contains the state variable which indicates the actual status of the hybrid drivetrain in terms of the discrete battery energy content which is visualised as the State Of Energy (SOE) of the battery. The SOE can be translated into the (mechanical) contribution of the motor/generator to the driver power demand $P_m$. The motor/generator operation within DP is considered from a battery point of view, which means that descending SOE (discharge) corresponds to motor mode while a rising SOE (charge) corresponds to the generator mode.

The dimensions of the feasible domain are determined by the battery specifications (Section 4.3.3) and the preferred charge mode. The charge mode can either be sustaining or depleting, depending on whether the battery is re-charged after driving a certain cycle. The shape of $S$ is explained in Figure 5.1 and 5.2 where the relevant battery parameters are depicted.

![Figure 5.1: state space grid; charge sustaining](image)

$(- = \text{feasible domain}; \bullet = \text{feasible node in } u; \ldots = \text{possible trajectory in } S)$

![Figure 5.2: state space grid; charge depleting](image)

$(- = \text{feasible domain}; \bullet = \text{feasible node in } u; \ldots = \text{possible trajectory in } S)$

The feasible domain is constrained by the maximum and minimum allowed SOE as well as by
the maximal charge and discharge power (indicated by the skewed borderlines in the graphs of Figure 5.1 and 5.2) which are determined by the storage technology. In this case, Li-ion, the maximum discharge rate is higher that the maximal charge rate.

The state variable, the vertical direction in $S$, is controlled by vector $u$ which contains the total amount of steps which can be chosen for the battery energy content. Control vector $u$ determines the grid density, and thereby the accuracy of the DP optimal solution. In case of the example above, $u$ contains 5 steps of 10 [%] SOE each. The step size of $u$ will be a trade-off between calculation time and control accuracy since a decreasing step size will increase the amount of control steps which need to be evaluated while performing the DP routine. In Figure 5.3 the (relative) Central Processing Unit (CPU) time for the optimisation process of the studied vehicle over the TDC cycle and the (relative) resulting fuel consumption $m_f$ is plotted as function of the step size $dE_s$. Step sizes smaller than 200 [J] result in a memory overload of the computer system and are therefore not feasible. The computation time for the smallest step size $t_{CPU,ref}$ is therefore the highest obtainable and equals 100 [%]. The CPU-time is obtained by using a HP xw4400 Workstation and includes the fore and after math of the actual DP routine. The most accurate result for the fuel consumption $m_{f,ref}$ is also set to 100 [%]. A step size $dE_s$ of 500 [J] seems to be a good compromise since it reduces the calculation time with 72 [%] while the fuel consumption only increases 0.084 [%] compared to the best obtainable result. The small irregularities in the fuel consumption trend are due to the fact that $u$ can only consist of (rounded) integer values corresponding to the step size $dE_s$.

![Figure 5.3: step size as function of relative calculation time and relative fuel consumption](PC: HP xw4400 Core2Duo 2.4 GHz (using single core), 4.0 Gb RAM)
Cost matrix

The cost matrix $C$ contains the momentary fuel cost $\dot{m}_f$ for all possible vehicle states $x$ as function of time. The basis of $C$ lies in the driver power demand $P_m(t)$ and the ingoing transmission shaft speed $\omega_m(t)$ for drive cycle $\omega_v(t)$. $P_m(t)$ and $\omega_m(t)$ are calculated by using a modified version of the base line vehicle model described in Chapter 4. The modifications include the implementation of BER strategies and the hybrid drive unit parameters. The input demand is down-sampled at a frequency of $1 \text{ [Hz]}$ or $dt = 1 \text{ [s]}$. At this frequency the most important longitudinal dynamic behaviour of the vehicle for the calculation of fuel consumption is captured, while the calculation time does not become excessively high.

The negative part of $P_m$ represents the brake power which can potentially be recovered. It is assumed that 100 [%] of this power is available in the BER mode which can be applied till vehicle standstill. The sensitivity on the BER percentage will be analysed. Also the effects of the different BER strategies (discussed in Section 4.5.6) are integrated in $C$. The front brakes are not applied unless the motor/generator is saturated.

In order to build $C$, a power demand matrix is constructed from vector $P_m$ with the size of the time horizon and the length of state vector $x$, an example can be found in Figure 5.4a where a sinusoidal power demand is suggested. Another matrix is constructed (Figure 5.4b) which contains the full range of operating points $P_{mg}$ of the motor/generator with a stepsize of $dP_s = dE_s dt$. Summing both matrices according to Equation 5.2 gives a third matrix containing the extended engine operating points as function of $x$ (Figure 5.4c). In this equation $P_{mg}$ is greater than zero in case of charging and smaller than zero when motoring.

$$P_e = P_m + P_{mg}$$  \hspace{1cm} (5.2)

where:

$$0 < P_e \leq P_{e,\text{max}}$$  \hspace{1cm} (5.3)

Cost matrix $C$ can now be calculated by using the engine fuel map (Figure 4.4) and by keeping in mind that infinitely high cost values have to be accounted for on the places where either the engine or motor/generator torque constraints are violated. Zero costs are calculated at vehicle standstill since engine stop is applied at these moments. When the engine stop mode at standstill is not activated it will be accounted for after the optimisation is performed.
5.2. ENERGY MANAGEMENT ANALYSIS: DYNAMIC PROGRAMMING

When executing the DP routine the cumulative costs for every control input for $C$ are saved in the cost-to-go matrix $L$. Meanwhile the most optimal control input route is saved in the node matrix $N$. While executing the DP routine the constraints of the state space matrix are not allowed to be violated.

The DP algorithm applied on $C$ evaluates the cumulative costs which are involved for each feasible control input $u(j)$ and each vehicle state $x(i)$ for every step in time $t(k + dt)$. The working principle of this algorithm can be described by the flow chart of Figure 5.5 where the three evaluation loops for variables $t$, $x$, and $u$ are shown. Since DP evaluates every single node in $C$, it does not constrain the engine shut off time and frequency, which means that the engine might be shut of for one time step ($t \in [s]$). The optimal stored battery power $P_{s,\text{opt}}(t)$ is obtained by retrieving the optimal control input $S$ from $N$. The resulting trajectory can be translated into the optimal mechanical contribution of the motor/generator $P_{mg,\text{opt}}(t)$.

---

**Figure 5.4: extended engine operating grid**

$\longrightarrow = \text{possible trajectory}$

**Cost-to-go matrix**

---

**Figure 5.5: DP working principle**
5.2.2 Results

The DP routine is used to calculate the theoretical optimal EMS and fuel consumption for a class 6 hybrid distribution truck. This means that in case of the CF65 hybrid truck, a GVW of 12000 [kg] is analysed. The TDC cycle is assumed to represent the average distribution truck usage and is therefore used in the first place. More information on driving cycles can be found in Section 4.4. Results for other vehicle GVW values and cycles are discussed in Chapter 6. The results for the DP optimal solution of the analysed vehicle are presented below by means of the vehicle’s average fuel consumption which is compared to the base line fuel consumption for three influential EMS variables; the BER strategy, the brake blending percentage, and the charge mode.

Brake energy recovery strategies

From Figure 5.6 it can be concluded that in case of 100 %BER at the rear wheels in combination with the charge sustaining mode and BER strategy 1 the hybrid truck in question has an average fuel consumption which is 12.1 [%] lower compared to the base line vehicle fuel consumption. For other BER strategies this reduction is significantly lower due to engine idle and the engine drag torque losses during regenerative braking. For the current vehicle and drive cycle combination the difference between the 3rd and 4th BER strategy is very small. It is concluded that in this case it does not matter whether the clutch is closed during BER (Table 4.13, BER strategy 3) or whether the engine drag losses are eliminated by opening the clutch during BER while the engine is idling (Table 4.13, BER strategy 4).

![Figure 5.6: DP optimal BER strategy sensitivity](image)

Brake blending

The brake blending percentage (%BER) at the rear wheels can be varied. For higher values of the BER percentage more kinetic energy is available for the BER mode. The fuel consumption reduction in case where 100 [%] brake power goes to the rear wheels is already pointed out and...
equals 12.1 \%], for decreasing %BER values the relative amount of profit reduces dramatically till 2.3 \% when no brake energy is recuperated. This trend can be seen in Figure 5.7 where the BER percentage is varied from 100 \% till 0 \%. The energy recovered during BER seems to be mainly responsible for the profit which is gained by the hybrid drive unit. In case when only engine stop is applied, 0.8 \% fuel is being saved due to the standstill periods. All simulations are carried out for BER strategy 1 and the sustaining charge mode is used.

![Figure 5.7: DP optimal brake blending sensitivity](image)

**Figure 5.7: DP optimal brake blending sensitivity**

### Charge mode

The first subdivision made for energy management strategies in Chapter 3 are the charge sustaining and charge depleting strategy. This section pays attention to charge depleting strategies to highlight its potential, possibly being a start for new research. A charge depleting mode can be valuable when a user has the opportunity to charge the vehicle’s battery. Starting the drive cycle with a ‘full’ battery (SOE = 70 \%) and allowing the battery to deplete towards a SOE of 30 \% is, not surprisingly, of significant influence on the amount of fuel left to be used by the engine. When decreasing the drive cycle length (in this example done by taking a percentage of the total drive cycle) this effect becomes even more visible. The resulting SOE trajectory of the battery for different drive cycle lengths is plotted in Figure 5.9. The percentage below each bar in Figure 5.8 represents the part of the TDC cycle over which the battery is depleted.

The results depicted in Figure 5.8 are strongly dependent on the course of the drive cycle. In general shorter drive cycles will results in higher fuel consumption reduction when a charge depleting strategy is applied. For some circumstances this might not be the case, for example when comparing the first 50 \% and 75 \% of the of the TDC cycle. The cause of this deviation in fuel consumption lies in the fact that a significant amount of energy can be recovered in BER mode between 50 and 75 \% of the TDC cycle, this can be seen in Figure 5.9.
Figure 5.8: DP optimal fuel consumption for charge depleting

Figure 5.9: DP optimal SOE trajectory for for charge depleting

\(-\) = SOE boundary; \(-\cdot\cdot\cdot\) = optimal sustaining SOE trajectory
5.2.3 Analysis of the obtained results

The optimal control solution calculated by the DP routine results in an only theoretically obtainable fuel consumption reduction for the vehicle and drive cycle in question. The fuel economy improvement is 12.1 [%] in the most beneficial situation, which is 100 %BER and engine shut down during braking and standstill. The amount of reduction is gained by the activation of the MA, M, and IS mode which subdivide the saved relative amount of fuel according to Figure 5.10. Apparently it seems that the Motor Assist (MA) mode is responsible for the majority of the profit followed by the Motor only (M) mode and the Idle Stop (IS) mode. This last mode has a very small contribution since the imposed drive cycle has a very low amount of stops, in practise a larger profit is expected due to e.g. other vehicles and traffic lights. Information regarding the total stop time can be obtained from the vehicle’s tachograph, an investigation over a truck fleet would be interesting. The cause of the MA mode to be most profitable can be explained through the engine fuel map characteristics (Figure 4.4) and more or less from the efficiency map of the motor/generator (Figure 4.10) and the battery (Figure 4.13). The engine fuel map characteristics are analysed in order to find the origin of the M and MA mode distribution.

![Figure 5.10: hybrid modes and fuel consumption reduction](12t CF FA65h 4x2 and TDC cycle)

The fuel consumption reduction of the parallel hybrid drivetrain is gained by applying the M and MA mode in order to improve the operating points of the combustion engine. The M mode is applied in order to fully replace the engine for low cycle power demands, while the MA mode lowers the engine power for high cycle demands. A range of motor/generator powers from 10 to 44 [kW] are subtracted from the cycle power demand and the resulting saved amount of fuel \( \Delta m_f \) is plotted in Figure 5.11. The MA mode is depicted by the dash-dotted lines for the various powers, while the M mode is depicted by a single solid line. Within the engine fuel map analysis it is assumed that an infinite amount of energy is available from the battery. The numbered areas correspond to the ones in Figure 5.12 and show the activity of the various hybrid modes as a function of the power which is demanded by the drive cycle. An explanation of these areas can be found in Section 5.3.

The blue line in Figure 5.11 shows the DP outcome which does take account for a neutral energy...
balance as well as for the motor/generator and battery efficiency. The DP outcome follows the
trends of the fuel map data, which emphasises the role of the engine in determining the optimal
EMS. It can be noticed that the highest amount of fuel can be saved in the higher power regions.
This explains the frequent activation of the MA mode by DP. The precondition of a neutral energy
balance prevents DP from applying more MA. The red line in Figure 5.11 represents the outcome
of the Rule Based (RB) EMS. This EMS is derived from the DP optimal control solution in Sec-
tion 5.3.2. It can be seen that the RB EMS does not have a specific preference area to apply the
MA mode. This is caused by the SOE neutraliser which is also explained in Section 5.3.2.

Trucks engines, in contradiction to passenger cars engines, are usually dimensioned on the ap-
lication where the truck is designed for. Therefore the applied drive cycle mostly requires full
throttle operation of the engine, whereas the overpowered passenger car engine runs on partial
load most of the time. This is also a reason that the greatest amount of fuel is saved in the higher
power regions of the trucks engine by applying the MA mode. Hybrid passenger cars gain their
profit mainly from applying the M mode.
5.2. ENERGY MANAGEMENT ANALYSIS: DYNAMIC PROGRAMMING

Figure 5.11: engine fuel map analysis

\[ \Delta \dot{m}_f \ [g/s] \]

\[ P \ [kW] \]

\[ -300 -250 -200 -150 -100 -50 0 50 100 150 \]

\[ -40 -20 0 20 40 60 80 100 120 140 \]

\[ P_{\text{split}} \ [kW] \]

\[ P_{\text{cycle}} \ [kW] \]

Figure 5.12: DP optimal power split

\[ + = P_{mg}; \circ = P_e \]
5.3 Energy management synthesis: Rule Based control

The DP result cannot be implemented in an online environment when the predefined drive cycle is not followed exactly. Therefore the obtained results are analysed accurately in this section in order to design a Rule Based (RB) controller which is online implementable. Using this controller in the vehicle model from Chapter 4 gives an indication of the practical potential fuel consumption reduction which can be achieved by the hybrid vehicle in a real world situation instead of the theoretical potentials. Even for different drive cycles the use of the RB controller should result in the best obtainable results. This will be analysed in Chapter 6. As a basis of the RB controller both the optimal power and torque split calculated by DP can be used. However, from previous research it is concluded that a torque split based RB controller is more robust than power split based control [Van Baalen, 2005]. A torque split is therefore used for this control application.

5.3.1 Torque split control

In Figure 5.13 the motor/generator torque $T_{mg}$ and engine torque $T_{e}$ are plotted as function of the demanded drive torque $T_{cycle}$, these results are generated using the DP procedure described in Section 5.2. Figure 5.13 gives valuable information required to build the RB controller. This synthesis is done for the 12t FA65 4x2 vehicle driving the TDC cycle. It is again assumed here that brake energy can be recovered down to standstill. From Figure 5.13 several operating areas for different hybrid modes can be distinguished, these modes are marked with numbers and form the basis of the RB controller.

![Figure 5.13: DP optimal torque split](image-url)
5.3. ENERGY MANAGEMENT SYNTHESIS: RULE BASED CONTROL

Brake Energy Recovery mode (area 1)

Starting with the 3rd quadrant (area 1, Figure 5.13) it can be noticed that the BER mode is fully exploited. It can also be seen that a larger generator (and battery) would be capable of regenerating more of the potentially available brake power. For this simulation only 43.6 [%] of the total available brake energy is recuperated by the motor/generator. However, the amount of recovered energy might be improved by having the generator operating in its sweetspot through shifting the transmission to lower ratios at higher shaft speeds.

Motor only mode (area 2)

For lower torque demands the engine is not used while the electric motor is solely propelling the vehicle (area 2, Figure 5.13). This Motor only (M) mode determines the all electric driving limit.

Engine only mode (area 3)

In area 3 of Figure 5.13 the engine operates on its own, which means that for this area propelling the vehicle in the conventional way is the most efficient.

Motor Assist mode (area 4)

The Motor Assist (MA) mode is active over the entire area 4. The distribution of the MA modes over area 4 will be discussed further in Section 5.3.2.
5.3.2 Control rules

The control rules being implemented in the hybrid vehicle (model) can be split into two pairs. In the BER and M mode the motor/generator works alone while in the MA and CH mode the engine has a contribution to the process. Besides these control rules, the different BER strategies introduced in Section 4.5.6 are also integrated in the vehicle model. Figure 5.14 shows the DP optimal power split for the first 500 seconds of the TDC cycle. This plot gives an impression of the desired behaviour of the EMS. The upper graph shows the required engine power when being assisted by the M mode (2nd graph) and MA mode (3rd graph). The 4th graph shows the activation of the BER mode where brake energy is being recuperated.

BER and M mode control

The BER and M mode can be translated into a single rule in the RB controller which implicates that for drive torques below \( T_M = 190 [Nm] \) (Figure 5.13, area 1 and 2) the motor/generator is active while the engine, depending on the BER strategy is shut off, idling, or applying its drag torque in case the clutch is closed. Equation 5.4 depicts the situation for the M mode and Equation 5.6 shows the equality for the BER mode.

\[
T_m = T_{mg} \tag{5.4}
\]

where:

\[
0 < T_m \leq T_M \tag{5.5}
\]

\[
-T_m = -T_{mg} \tag{5.6}
\]
where:

\[-T_{mg,\text{max}} \leq T_m < 0\]  \hspace{1cm} (5.7)

**SOE neutraliser**

The optimal control strategy obtained by DP can either be charge depleting or charge sustaining depending on the application of the vehicle. In case of the vehicle in question a charge sustaining EMS is desired because the applied duty cycles are rather long. According to the DP optimal sustaining control split, the battery however receives more energy in BER mode than it uses in M mode since more profit can be gained by activating the MA mode (Section 5.2). A SOE neutral solution can be obtained by activating the MA mode and the CH mode, this solution is visualised in Figure 5.15 (left). In order to obtain this balance in an online environment a SOE neutraliser is introduced. It usually ensures a balance between the M and MA mode on the one hand and the CH and BER mode on the other. For example when not enough brake energy is available for recuperation (Figure 5.15, right) the CH mode will cover the difference. Figure 5.17 shows the situation for the analysed vehicle where the CH mode is not present. The SOE neutraliser also enables the RB EMS to be compatible with drive cycles other than the one where the optimisation is done for.

The SOE neutraliser is based on a simple controller (Equation 5.8) which brings the battery energy content \(E_s(t)\) to a desired level \(E_{s0}\).
\[ P_{mg} = K_s(E_s(t) - E_{d0}) \] (5.8)

In case of \( E_s(t) > E_{d0} \) the MA mode will be stimulated in area 3 \((P_{mg} > 0)\) while \( E_s(t) < E_{d0} \) results in activating the CH mode \((P_{mg} < 0)\). It is however important to have battery capacity available for the free energy obtained in the BER mode. For robustness we introduce separate values of \( K_s \) for the MA and CH mode. A charge rate parameter \( \xi_{CH} \) determines the ratio between the two values of \( K_s \) (Equation 5.9). The \( K_{s,MA} \) gain will be applied when \( E_s(t) > E_{d0} \), the \( K_{s,CH} \) gain in case when \( E_s(t) < E_{d0} \). Since the value of \( E_{d0} \) for a charge depleting EMS equals an empty battery the CH mode will be dis-activated. The value of \( \xi_{CH} \) equals zero for the analysed vehicle and cycle since the CH mode is not being activated.

\[ \xi_{CH} = \frac{K_{s,CH}}{K_{s,MA}} \] (5.9)

The SOE neutraliser equations now become as follows for the MA mode (Equation 5.10) and the CH mode (Equation 5.13). The variables \( T_{MA,low}, T_{MA,high}, T_{CH,low}, \) and \( T_{CH,high} \) represent the borders of the areas from Figure 5.13.

\[ P_{mg} = K_s(E_s(t) - E_{d0}) \] (5.10)

where:

\[ E_s(t) > E_{d0} \] (5.11)

\[ T_{MA,low} < T_m < T_{MA,high} \] (5.12)

\[ P_{mg} = \xi_{CH}K_s(E_s(t) - E_{d0}) \] (5.13)

where:

\[ E_s(t) < E_{d0} \] (5.14)

\[ T_{CH,low} < T_m < T_{CH,high} \] (5.15)

It is important to choose a value of the SOE neutraliser gain such that the target fuel consumption calculated by DP is approached and that a neutral battery SOE is obtained. Lower gains \((K_s \rightarrow 0)\) result in a drift of the SOE trajectory and a lack of battery capacity for the BER mode, while a too nervous operation of the MA mode for high values of \( K_s \) is undesirable as well. The nervous behaviour of the MA mode for high \( K_s \) values \((K_s = 0.1 - 100)\) is depicted in Figure 5.18 where the MA power of the motor/generator is plotted as a function of a time window within the TDC cycle. As a benchmark, the DP optimal solution for the MA mode operation is also shown in this figure. It can be seen that the DP solution does not take account for drivability issues.

Figure 5.19 shows for a mid-range of \( K_s \) values the resulting fuel consumption relative to the DP optimal fuel consumption. A gain value of \( K_s = 0.04 \) results in the desired and almost neutral battery SOE (Figure 5.20) while the target fuel consumption is almost reached.
5.3. ENERGY MANAGEMENT SYNTHESIS: RULE BASED CONTROL

Figure 5.18: Nervous and drifting MA operation ($\xi_{CH} = 0$)

Figure 5.19: $K_s$ tuning ($\xi_{CH} = 0$)

($\circ = simulation data$)
**DP versus RB**

In Figure 5.20 the largest deviation in fuel consumption is found around $t = 1000 \ [s]$ where the DP SOE is the lowest. This is the consequence of the fact that the drive cycle is known a priori by the DP algorithm, the DP solution ‘knows’ that a regenerative braking opportunity is ahead and allows the battery to deplete a little more. DP provides a both global and local optimal solution resulting in a neutral SOE. The RB controller finds almost the same local optimal solution from a set of rules trained by DP, as can be seen in Figure 5.20. The global optimisation is done using the SOE neutraliser based on instantaneously available information and does not result in a neutral SOE. The robustness of the RB controller on different truck applications is studied in Chapter 6.

![Figure 5.20: DP versus RB results; $K_s = 0.04$](image)

Figure 5.20: DP versus RB results; $K_s = 0.04$

(- - - - = DP; — = RB; · · · = neutral SOE)

The RB control parameters which are used in the equations discussed above are summarised in Table 5.1.

<table>
<thead>
<tr>
<th><strong>Table 5.1: RB control parameters</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>RB control parameters</strong></td>
</tr>
<tr>
<td>$T_M$</td>
</tr>
<tr>
<td>$T_{MA,low}$</td>
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<tr>
<td>$T_{MA,high}$</td>
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<tr>
<td>$T_{CH,low}$</td>
</tr>
<tr>
<td>$T_{CH,high}$</td>
</tr>
<tr>
<td>$K_s$</td>
</tr>
<tr>
<td>$\xi_{CH}$</td>
</tr>
</tbody>
</table>
5.3.3 Results

The synthesised RB controller is used to perform a simulation over the same drive cycle with the same vehicle as done for the DP optimisation. The RB EMS should perform in approximately the same way as the DP optimal EMS does. Besides this, for the RB EMS it is also important that it is compatible with different cycles and different vehicle masses. The sensitivity of the RB EMS for those parameters is analysed in Chapter 6.

![RB torque split](image)

Figure 5.21: RB torque split

\[ (+ = T_{mg}; \circ = T_e) \]

Figure 5.21 gives the resulting RB torque split which shows, apart from the amount of operating points, approximately the same result for area 1, 2, 3, and 4 as the DP optimal split from Figure 5.13. It should be remarked that the RB solution results in a fuel consumption which is 0.9 [%] higher than the cumulative cost calculated by DP. This deviation is compensated by the battery SOE for the RB EMS which ends at 54.8 [%] while the DP EMS has its predefined SOE ending at exactly 50 [%] (Figure 5.20). The difference of 4.8 [%] SOE equals 432 [kJ] of energy additionally present in the battery. When considering an average engine efficiency of 35 [%] and the fuel lower heating value from Section 4.3.4, it can be calculated that the equivalent amount of fuel equals 29 [g]. This corresponds approximately to the results depicted by Figure 5.20.

5.4 Conclusions

From this chapter can be concluded that the amount of fuel which can be saved by a 12t FA65 4x2 distribution truck driving the TNO Dynamometer cycle equals 12.1 [%]. In order to reach this reduction it is necessary to apply BER strategy 1, i.e. opening the clutch while energy is being recuperated and stopping the engine during BER and standstill. The fuel consumption reduction can be increased by another 4 [%] by applying a charge depleting EMS. This is however not
desired since in practise duty cycles are rather long.

5.5 Recommendations and discussion

During the analysis and synthesis of the energy management lots of findings have been encountered. Divided in component, model, control, and drive cycle related issues this section discusses the most relevant experiences.

Component related

The DP outcome relies on the efficiency data of the various drivetrain components, it is therefore desired to use highly detailed maps in order to obtain the clearest possible view on the objective solution. Especially the engine fuel consumption map, which is mainly decisive in determining optimal EMS, needs to be implemented at the highest possible level of accuracy. Also the efficiency maps for the electric components and gear ratios of transmission and final drive require further analysis. At a lower priority the inertias of various rotating parts should be determined more accurately.

Related to the sizing of the hybrid drive unit components it should be mentioned that the battery’s maximally receivable power is smaller than the electric machine can generate. This results in a reduced amount of recoverable energy in BER mode while for the analysed vehicle a larger motor/generator is already desired in the first place. The optimal component sizing is studied in [Hofman, 2007a] for further reading.

Model related

When the DP results are used as a control input for the vehicle model from Chapter 4 in order to validate the DP solution, an uncertainty is noticed in the resulting response. The DP optimal solution is obtained from down-sampled 1 [Hz] signals resulting in a 1 [Hz] control solution, while the actual vehicle model runs at 10 [Hz]. This phenomenon is caused by the reconstruction of a 10 [Hz] optimal control input from the 1 [Hz] output signal. To validate the accuracy of the 1 [Hz] DP solution compared to the vehicle model by looking at the relative cumulative fuel consumption deviation from both solutions, see Figure 5.22. For this result the 10 [Hz] vehicle model outcome is down-sampled to 1 [Hz]. It can be concluded that the accurateness of the DP solution compared to the actual vehicle model is within the allowed regions of +/- 1 [%] deviation. The average fuel consumption for the DP solution equals 23.4836 [l/100 km], for the vehicle model it equals 23.6991 [l/100 km].
Control related

Hybrid drivetrains take their advantages from regenerating energy which is released during decelerating. To gain the highest possible fuel consumption reduction it is thus important to be able to regenerate as much energy as possible. The EMS of the hybrid drivetrain should therefore only activate the conventional brake system when the generator is saturated, thus 100 %BER at the driven rear axle is desired. Also opening the clutch while BER is useful since it takes away the parasitic braking losses caused by the engine friction torque. In case when the BER mode is not activated during the analysed cycle, corresponding with 0 [%] BER, a very small amount of fuel is being saved by the hybrid drivetrain. This fuel is saved during the activation of the MA mode for high cycle demands which is fed by the CH mode. The CH mode is activated during low cycle demands.

The maximum performance from the motor/generator is reached when it can operate in its sweetspot. Shifting to a lower gear during deceleration should be done at shaft speeds which correspond to this optimal area. This is also applicable when the generator is operating in motoring mode.

From the charge mode sensitivity analysis it is denoted that the optimal control split changes for decreasing cycle length. The EMS should be made adaptive to this effect in order to have it performing in an optimal way for varying vehicle conditions.

From the engine point of view it might be worthwhile to add a warm-up phase after a cold start. During this period the engine is not allowed to be stopped by the EMS in order to pre-heat the catalytic materials of the exhaust gas treatment system and the engine itself in order to operate optimally.
Drive cycle related

The goal of optimising the EMS is to minimise the vehicles fuel consumption. This is mainly due to regeneration of normally wasted brake energy and by stopping the engine while it is not being used. The vehicle does not take full advantage from the engine stop functionality when driving the TDC cycle. In real world situations the vehicle will face many more standstill periods, e.g. during charging and discharging cargo, at traffic lights, in traffic jams compared to the TDC cycle which has a relative standstill time of only 11.4 [%].
Chapter 6

Analysis and Synthesis of Hybrid Truck Applications

Introduction

In several aspects distribution trucks differ extremely from passenger cars. This chapter therefore discusses and analyses three important aspects where (hybrid) distribution trucks are subjected to. At first, the EMS sensitivity for changing vehicle mass of such vehicle is studied. Secondly the EMS sensitivity on the enforced drive cycle will be analysed. Three different drive cycles are therefore chosen with low, medium, and high average demands; these cycles are discussed earlier in Chapter 4. Finally the potentials of synergies with the hybrid drive unit are investigated and an electronic Power Take-Off (ePTO) for the high auxiliary power demand of a refrigeration system is proposed. For all simulations done in this chapter, unless indicated otherwise, the used BER strategy = 1, 100 [%] BER is assumed and the charge sustaining mode is applied.

6.1 GVW sensitivity

A passenger car is primarily configured to move people and a small amount of luggage from one place to another. The vehicle mass and therefore its fuel consumption do not significantly change during its lifetime. Heavy duty vehicles on the other hand, especially distribution trucks, have to deal with highly fluctuating vehicle loadings when starting a day with a fully loaded truck and ending it with an empty one. These fluctuations in GVW do have a significant impact on the fuel consumption, mainly while accelerating due to the increase of vehicle inertia. The GVW of the vehicle analysed in this report (FA65 4x2) has a maximum value of 18t (18000 [kg]) when fully loaded while the empty vehicle has a total on the road mass of 5540 [kg] [Leyland, 2002]. For a range of vehicle masses (5540 [kg], 7.500 [kg], 10.000 [kg], 12.000 [kg], 15.000 [kg], 18.000 [kg]) the sensitivity of the DP optimal fuel consumption is analysed (Figure 6.1). Like for the BL fuel consumption a linear relation between fuel consumption and GVW can be noticed.
The amount of fuel saved by adding the hybrid drive unit increases slightly for increasing GVW ($2.5 [l/100 km] \cdot 3.4 [l/100 km]$). This can be explained by the fact that during BER more energy can be recovered by heavier vehicles. Also the application of the MA mode is more beneficial when high (engine) power is demanded, which is concluded from Figure 5.11. Simulations have proven that for low GVW the contribution of the M mode to the total amount of saved fuel becomes much larger. Figure D.5 in Appendix D shows the distribution of the MA, M, and IS mode for the three different drive cycles (Section 4.4) and a range of vehicle weights.

As a rule of thumb it can be claimed that in case of the TDC cycle an amount of fuel can be saved which is described by Equation 6.1 where the GVW is expressed in tonnes.

$$\Delta m_f = 0.07GVW + 2.1$$  \hspace{1cm} (6.1)

The reader is referred to Appendix D for more simulation results concerning the DP EMS sensitivity for varying GVW.
6.2 Drive cycle sensitivity

For three imposed drive cycles, each with a different character, and the range of GVW values the DP optimisation is performed. The DP EMS sensitivity on these variables is visualised in Figure 6.2. All figures show this reduction in percentages of the base line fuel consumption, except when indicated otherwise. For other simulation results regarding absolute fuel savings the reader is referred to Appendix D. It can be seen that for both changing GVW and drive cycle there is a significant spread in fuel consumption reduction; from less than 10 [%] up to more than 24 [%] can be saved. Recalling that the absolute fuel consumption for the BL vehicle is highest in case of the UTEC cycle and the lowest in case of the TDC cycle, it can be concluded that the latter cycle is least suitable for hybrid distribution applications when intending to save fuel.

The amount of fuel that can be saved holds a direct relation with the amount of energy that is recovered in BER mode. In Figure 6.3 the percentage of the available brake energy which is actually recuperated is plotted for the DP optimal EMS. It can be concluded that for most situations lots of free energy is left unused. Only in case of the UTEC cycle in combination with low GVW all energy can be recuperated. The cause of this lies in the (non) optimal sizing of the hybrid drive unit components; it seems that the vehicle in question is optimised for a GVW below 10t and urban drive cycles.
The practical optimal fuel consumption reduction which can be obtained by simulations using the forward vehicle model and the RB EMS (Figure 6.4) is probably more interesting since it indicates the hybrid drive unit potential in a real world environment. The RB controller parameters are derived from the DP optimal torque split obtained from the power demand of a 12t vehicle driving the TDC cycle (Chapter 5). In general it seems that for the whole GVW range and the various drive cycles the reduction trend corresponds to Figure 6.4; between 9 [%] and 32 [%] fuel can potentially be saved in a practical manner. The absolute amount of fuel which is saved for each application lies between 375 [g] for empty vehicles and 550 [g] for fully loaded vehicles in case of the UTEC cycle. More information concerning these results can be found in Appendix D.

Figure 6.3: DP optimal recovered energy
(simulation data: □ = UTEC; ○ = FTP; △ = TDC)
6.2. DRIVE CYCLE SENSITIVITY

Figure 6.4: RB optimal fuel consumption reduction (BL-RB)/BL
(simulation data: □ = UTEC; ⋄ = FTP; △ = TDC)

By comparing the DP and RB results in Figure 6.5 the robustness of the RB EMS can be analysed for situations where the vehicle is subjected to varying GVW and different cycles.

Figure 6.5: RB robustness for fuel consumption reduction (RB-DP)/RB
(simulation data: □ = UTEC; ⋄ = FTP; △ = TDC)

The RB EMS performs within a margin of 2 [%] equal to the DP optimal EMS for most situations.
While this deviation is less than 1 [%] for the situation where the RB EMS was originally designed for. For vehicle masses below 9t in combination with the UTEC cycle it seems that the RB EMS performs far better than the DP optimal EMS. When taking a look at the SOE trajectory for these cases it seems that the SOE balance is distorted as a result of applying the M mode while during BER not enough energy can be recuperated. The DP optimisation aims at a SOE neutral solution which can only be obtained by activating the CH mode. Figure 6.6 shows the optimal torque split for an empty vehicle (GVW = 5540 [kg]) driving the UTEC cycle, the charge mode can be distinguished in area 2’.

![Figure 6.6: DP optimal torque split with CH mode](image)

The RB EMS can be extended by an additional rule which activates the CH mode within area 2’ under the condition that the battery is (almost) empty. However, with the knowledge that in practice a distribution truck is not driving in an urban environment all day long, a BER opportunity will announce itself soon.

Instead of analysing the EMS sensitivity for drive cycles ($v(t)$) it is recommended to study its sensitivity for routes ($x(t)$) as well. A route will give a better representation of the real world driving situations since it can take the effects of for example traffic lights, traffic jams, and other vehicle’s speed into account. The course of the vehicle speed can be chosen as a variable which can be optimised in order to recuperate as much energy in BER mode as possible. An anticipating driving method would be required in order to implement such strategy in a real world situation.

An example of a modified drive cycle can be found in Figure 6.7. The solid lines represent a worst case scenario where during braking (at $t = 16$ [s]) zero energy is recovered due to an infinitely high deceleration. The optimal amount of brake energy can be recovered when the deceleration time is maximised. This can be done by exploiting the present stop time at the end of the cycle.
and by braking at an earlier moment \((t = 10 \text{ [s]})\). It is however important that the same distance is travelled within the given time window. The optimised drive cycle is represented by the dashdotted curves in Figure 6.7. In the example the drive cycle is optimised for the standstill periods only. Even more brake energy can be recovered in practise when anticipating on other vehicles. For example when the driver preserves a larger distance between the truck and the vehicle driving in front of him a longer and less firm deceleration can be obtained when slowing down resulting in more recovered energy. This however has huge consequences for the course of the drive cycle and therefore requires further research.

The DP EMS analysis is performed for the studied vehicle driving over a modified version of the TDC cycle (Figure 6.8 shows the first 500 \([\text{s}]\)). The modifications are made by using the method described in Figure 6.7 in order to highlight the potentials of anticipated driving methods. According to the EMS analysis the fuel consumption reduction has increased to 20.9 \([\%]\) instead of 12.1 \([\%]\) for the unmodified TDC cycle. It can be concluded that brake energy recovery plays the key role for the EMS of hybrid electric distribution trucks.
6.3 Hybrid synergies: electrified power take-off

For various truck applications lots of electric devices are used. For conventional vehicles the required power is mostly generated by means of a Power Take-Off device (PTO). This PTO draws off torque from the vehicles drivetrain to drive an additional generator (Figure 6.9), typically between 8 and 10 [kW] in power rating.

![Figure 6.9: conventional drivetrain topology with PTO](image)

Because generators prefer running at a constant speed in order to generate an (almost) constant 50 [Hz] current, the Constant Speed-Power Take Off (CS-PTO) is introduced [De Cloe, 2003]. The constant speed is realised by using a Continuously Variable Transmission (CVT) (Figure 6.10).

![Figure 6.10: conventional drivetrain topology with CS-PTO](image)

In the case of a hybrid electric truck, an electrochemical accumulator is used to supply and to temporarily store electric energy in order to improve the drivetrain performance. This electric energy could also be used in a synergetic way as a power supply for the trucks peripheral equipment. Besides saving the costs of a PTO device and its generator, this synergy has the potential to further decrease the vehicles total energy consumption. The potentials of the electrified PTO (ePTO, Figure 6.11) are analysed and a proposal for vehicle integration is given.
6.3. HYBRID SYNERGIES: ELECTRIFIED POWER TAKE-OFF

6.3.1 Refrigeration system

An example of an (electric) power consuming device is a refrigeration system which keeps the cargo at a desired temperature by reconditioning the surrounding air. The amount of power required to keep the cargo space at a constant temperature can be calculated by determining the total heat conduction through the cargo space walls. The following assumptions are made in order to obtain a representative constant thermal loss:

- The influence of the cargo heat capacity is neglected
- A constant energy flow through the cargo space walls is assumed
- All six cargo space walls deal with the same thermal losses
- The cargo space size equals 6 x 2.5 x 3 [m] (L x W x H) and the wall thickness \( d \) equals 0.15 [m]
- The doors are kept closed at all times

By assuming the refrigeration system efficiency to be 100 [%] the following equation (Equation 6.2) can be used directly to obtain the amount of electric power required for continuous operation which equals 6480 [W]. In this equation \( A \) represent the total heat conducting surface of the cargo area (81 [m²]). The temperature difference between in and outside the cargo space \( \Delta T \) equals 30 [K] [ATP, 1974] for class A refrigeration unit and the thermal conductivity \( k \) of the walls equals 0.4 [W/mK] [ATP, 1974].

\[
P_{e,ref} = \frac{kA\Delta T}{d} \tag{6.2}
\]

A cargo temperature deviation of +/- 2 [K] [De Cloe, 2003] is allowed; when the lower bound temperature is reached the refrigeration system can be shut off. This could have consequences on the systems operating efficiency but is not taken into account in this analysis.
6.3.2 Vehicle integration of the ePTO

The integration of the new ePTO system in the vehicle can be done in several ways. On different locations in the electrical path of the hybrid drivetrain the connection to the ePTO system can be created. The configuration depicted in Figure 6.11 is found to be most suitable for distribution truck applications. The highest efficiency can be obtained when the required power is primarily drawn from the motor/generator while it is driven by the engine or the vehicle in BER mode which will be the case most of the time. When the motor/generator is used for the M or MA mode, the required power is drawn from the battery. At vehicle standstill it might be preferred to draw electric power from the battery (charge depleting) for reasons of noise pollution e.g. when the vehicle is parked in crowded areas. For now the engine will propel the motor/generator at standstill.

The conversion efficiencies of the cyclo converter (AC/AC) and inverter (DC/AC) are assumed to be 95 [%] for the whole operating range. For this configuration the cyclo converter is the only required additional component. Alternatively it is recommended to study the possibilities of directly using the power from the motor/generator while being supported by the DC/AC converter from the hybrid drive unit.

6.3.3 Modelling the CS-PTO

The performance of the ePTO is best compared with the performance of the CS-PTO because both systems are able to supply a constant electrical power flow. The conventional PTO has a varying power supply to the refrigeration unit depending on the engine speed (Figure 6.12). The CS-PTO is therefore modelled as an additional load at the crankshaft which is determined by the required power divided by the current shaft speed (Equation 6.3). The conversion efficiency of the CS-PTO, $\eta_{CS-PTO}$, lies between 40 and 90 [%] but is assumed to be 65 [%].

$$T_{CS-PTO} = \frac{P_{e,refr}}{\omega e \eta_{CS-PTO}}$$  \hspace{1cm} (6.3)

Figure 6.12: indicative plot for CS-PTO and conventional PTO power supply

\[— = CS-PTO; \quad - \cdot - \cdot - = conventional PTO\]
6.3.4 Modelling the ePTO

Integration of the ePTO in the DP algorithm can be done by modifying control vector $u$ such that an additional battery load is applied corresponding to the above discussed ePTO application. For values of $u$ smaller or equal to zero (discharging during M and MA mode and at vehicle standstill or engine only mode) the additional battery load $P_{s,refr,dis}$ is determined by the conversion efficiencies of the inverter and cyclo converter of the system according to Equation 6.4. For $u$ greater than zero a penalty is applied for the amount of energy which is not received by the battery during the BER or CH mode, this penalty is determined by Equation 6.5. Note that the (momentary) battery efficiency during charging or discharging in Equation 6.4 and 6.5 is depending on the momentary electric power flow determined by the state vector of the battery.

$$P_{s,refr,dis} = \frac{P_{e,refr}}{\eta_{DC/AC}\eta_{AC/AC}\eta_{bat,dis}} \quad (6.4)$$

$$P_{s,refr,chg} = \frac{P_{e,refr}\eta_{AC/AC}\eta_{bat,chg}}{\eta_{AC/DC}} \quad (6.5)$$

The battery load vector for the ePTO $P_{s,refr} = [P_{s,refr,dis} P_{s,refr,chg}]$ is built and used to re-formulate $u$. The new control vector $u$ is calculated by using Equation 6.6. The refrigeration power penalty is rounded since vector $u$ can only contain integer values.

$$u = u - \text{round} \left( \frac{P_{s,refr}}{dE_s} \right) \quad (6.6)$$

6.3.5 Simulation results

Simulations have been performed for the studied 12t FA65 4x2 truck driving the TDC cycle. Like in other simulations done in this research the applied BER strategy is 1 and 100 [%] BER at the rear axle is applied. The charge mode is again sustaining, although a depleting mode might be valuable for the application. The bars in Figure 6.13 show the average fuel consumption for the Base Line (BL) and DP optimal hybrid vehicle including and excluding a cargo refrigeration system. The analysed hybrid vehicle is equipped with the ePTO while the base line vehicle is equipped with the CS-PTO, both systems are capable of supplying a constant amount of electric power to the refrigeration unit. According to the calculated fuel consumptions it seemed worthwhile to integrate a power take-off in a synergetic way in the hybrid drive train. The additional required amount of fuel to operate the refrigeration unit is significantly less for the hybrid vehicle than for the base line vehicle. The difference can be explained firstly by the free energy used by the ePTO during BER without the intervention of the battery. Secondly, the hybrid drive unit provides power for the ePTO directly by activating the CH mode in optimal operating areas of the engine. It can also be seen that the hybrid vehicle with ePTO has approximately the same fuel consumption as the base line vehicle without PTO.
CHAPTER 6. ANALYSIS AND SYNTHESIS OF HYBRID TRUCK APPLICATIONS

Figure 6.13: optimal ePTO versus base line CS-PTO

It should be noticed that the integration of an ePTO in the hybrid drivetrain will not affect the drivability (in terms of the trackability: 0.26 [m/s] vs. 0.31 [m/s]) in contradiction to the application of a CS-PTO as a result of engine performance constraints. In case of the ePTO the battery performs as a buffer which preserves the drivability of the vehicle.

The DP optimal torque split including the ePTO (Figure 6.15) has been changed significantly compared to the torque split which is calculated by DP without the ePTO (Figure 6.16). As can be seen in area 4 the MA mode has almost disappeared and the Charge (CH) mode has appeared over almost the whole range of torque demands especially for lower demands (area 2 and 3) where the engine efficiency seemed highest. The benefit of the M mode is reduced to zero and therefore not present anymore. A smaller amount of energy is send to the battery in BER mode, instead a part is going to the refrigeration unit directly. As a result the torque split of Figure 6.15 shows a reduced amount of torque in area 1. Partially in area 4 as well as during standstill the ePTO draws energy from the battery. An indicative sketch of the energy balance is shown in Figure 6.14.

Figure 6.14: sustaining balance for the analysed vehicle with ePTO
6.3. HYBRID SYNERGIES: ELECTRIFIED POWER TAKE-OFF

![Figure 6.15: DP optimal torque split for ePTO integration](image)

Figure 6.15: DP optimal torque split for ePTO integration

\((+ = T_{mg}; \circ = T_e)\)

![Figure 6.16: DP optimal torque split without ePTO](image)

Figure 6.16: DP optimal torque split without ePTO

\((+ = T_{mg}; \circ = T_e)\)

The cost effectiveness of the ePTO as proposed is mainly determined by the additional cost of the required inverter. It is desired to make an outline of all costs involved for utilising a hybrid truck with ePTO in order to compare the base line vehicle with the CS-PTO.
6.4 Conclusions

From simulations it can be concluded that one single RB EMS is robust and (sub-)optimal for a wide range of vehicle loadings, both for varying vehicle mass and drive cycles. The fuel consumption reduction however strongly depends on these variables and can not always be guaranteed to be at a certain level. The main cause of this lies in the sizing of the hybrid drive unit components. A synergy of the ePTO and the hybrid drive unit results in even higher reductions when compared to a base line vehicle with CS-PTO. Electric power from BER and CH mode is used by the peripheral equipment directly without the intervention of the battery.

6.5 Recommendations

The proposed TDC cycle does not seem to be suitable when intending to save fuel, another bench mark drive cycle with more acceleration and deceleration and lower speeds should therefore be introduced. Besides such city cycle it is interesting to study the (DP) EMS sensitivity for a route instead of a cycle. In this way the EMS can be made adaptive to external influences, like vehicles on the road and other traffic obstructions by leaving the vehicle speed as a variable to be optimised. This enables the system to recover more brake energy and thereby increases the vehicles fuel consumption reduction further.

Concerning the vehicle, it is strongly recommended to use a larger generator capacity (battery and motor/generator unit) for vehicles with a 7.5t+ GVW. Components sizing optimisation for the motor/generator and battery [Hofman, 2007a] for the most occurring vehicle road load should be performed.

Further research is required within the field of the ePTO since in practise a refrigeration unit does not have a constant power demand; both the CS-PTO and ePTO can be optimised for this. After the optimisation the RB EMS should be adapted in such way that an optimal operation of the ePTO is guaranteed at all times.
Chapter 7

Marketing the Hybrid Electric Distribution Truck

7.1 SWOT analysis

A company that is about to grow by launching new products develops ambitious targets and a plan. During this process one may never lose sight of reality. A feasible plan should therefore be based on decent analysis of strengths and weaknesses of the new technology and the opportunities and threats of the external influences. All information which is gained from this analysis can be outlined in a SWOT-analysis [Verhage, 2000], which stands for Strengths, Weaknesses, Opportunities, and Threats. By performing this SWOT analysis a series of points of interest will be discussed to come to a conclusion concerning the value of hybrid electric road cargo distribution. The SWOT-analysis is performed by examining (mostly) internet forums and websites concerning (hybrid) trucks. Each point of interest that could be accommodated in the SWOT matrix (Table 7.1) is discussed in this chapter.
Table 7.1: SWOT analysis

<table>
<thead>
<tr>
<th>Internal analysis</th>
<th>External analysis</th>
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<tbody>
<tr>
<td><strong>Strengths</strong></td>
<td><strong>Opportunities</strong></td>
</tr>
<tr>
<td>1. Operational savings (fuel, maintenance, extended engine lifetime)</td>
<td>1. Extension of product range</td>
</tr>
<tr>
<td>2. Productivity gains (better drivability, idle time reduction)</td>
<td>2. Improved market position</td>
</tr>
<tr>
<td>3. Wider range of applications can be served (ePTO, stealth mode driving)</td>
<td>3. Image booster</td>
</tr>
<tr>
<td>4. Meets (future) emission standards</td>
<td>4. Technological leadership</td>
</tr>
<tr>
<td>5. Reputation of DAF Trucks NV (growing market share)</td>
<td>5. Follow up market trends</td>
</tr>
<tr>
<td>6. Close cooperation with research institutes (TU/e and TNO)</td>
<td>6. Enter new markets (e.g. army applications)</td>
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<tr>
<td></td>
<td>7. Meet customers demands (fuel economy, reduced maintenance)</td>
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<table>
<thead>
<tr>
<th>Weaknesses</th>
<th>Threads</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Reliability not (yet) at same level as OEM components</td>
<td>1. Success of technology depending on accumulator development</td>
</tr>
<tr>
<td>2. High additional cost of the technology</td>
<td>2. Vehicle should meet customer demands</td>
</tr>
<tr>
<td>3. Operational savings strongly dependent on application</td>
<td>3. Concurrency of other manufacturers developing the same sort of technology</td>
</tr>
<tr>
<td>4. Supplier base still underdeveloped</td>
<td>4. European technological inferiority compared to US and Japan</td>
</tr>
<tr>
<td>5. Further training of DAF employees required</td>
<td>5. Dependency on incentives and subsidise</td>
</tr>
<tr>
<td>6. Only one component size available</td>
<td></td>
</tr>
</tbody>
</table>
7.2 Internal analysis

By performing an internal analysis the, by the company controllable variables are studied. The company is hereby criticised on financial capability, production technology, know-how, product range, capacity, and reputation. Since commercial and financial information are only available in limited detail, the internal analysis mainly focusses on the technical strengths and weaknesses of the hybrid electric distribution truck.

7.2.1 Strengths

1. Operational savings

Compared to conventional distribution trucks, hybrid electric distribution trucks have various advantages which can be marked as the strengths of the product. First of all the operational savings, which play the major role in developing hybrid trucks since the market demands cheaper transportation. The typical costs of road transportation are depicted in Figure 7.1 [Rijsenbrij, 2005] to illustrate this advantage. As can be seen the fuel cost are highest representing 37 [%] of the total cost. The studied hybrid electric truck has the ability to reduce the fuel consumption by 10 to 25 [%] compared to the base line vehicle (Chapter 5), the relative fuel cost for transportation can thus be reduced by 3 to 7 [%] when keeping all other costs at the same level. It should be noted that the higher the fuel price goes, the higher the cost reduction will be, resulting in a shorter payback time.

When considering a charge depleting EMS which enables the truck owner to use (renewable) electricity from the grid during a drive cycle possibly even more cost (and emission) reductions can be obtained. An indication of the cost effectiveness will be given by a ‘tank-to-crank’ analysis which will result in the cost of mechanical power which is available at the vehicles ingoing transmission shaft in [€/kWh]. Within this analysis the diesel fuel price is assumed to be 1.0066 [€/l] which is the average diesel fuel price of January 2007 in The Netherlands [TLN, 2007]. The electricity costs are rated at 0.204 [€/kWh] which is the price for industrial use (15000 [kWh/year] on average) at the 1st of January 2007 in The Netherlands [Delta, 2007]. A graph (Figure 7.2) is obtained by using the engine data from Chapter 4 which shows the absolute cost per [kWh] mechanical engine power at the crankshaft. The (lowest possible) cost of the mechanical power from
the motor/generator using electricity from the grid is also shown in Figure 7.2. For charging the vehicles battery, a conversion efficiency is taken into account of 95 [%] for the charging device, 92.2 [%] for (highest) charge rate, and 92.0 [%] for the (most efficient) discharge rate of the battery (Table 4.9). The motor/generator efficiency is 90 [%] at maximum. Surprisingly diesel power is cheaper than electrical power for most power demands, as can be seen from Figure 7.2. Short and low demanding drive cycles will thus be cost effective for charge depleting strategies.

![Graph of mechanical power cost for diesel propulsion](image)

**Figure 7.2:** mechanical power cost for diesel propulsion

(−·−·− = Eaton Hybrid Drive Unit; — = Cummins ISBe 6 cylinder diesel engine)

Other operational saving are on the field of maintenance (less brake wear) and an extended engine life time (less engine usage). However both aspects require additional research and more practical experience.

2. **Product improvements**

Faster launches from standstill due to the onboard motor/generator are marked as a drivability improvement. Also idle time reduction is marked as a product improvement since the engine can be shut of more frequently.

3. **Extended applications**

Another advantage of the hybrid electric drivetrain is the fact that, compared to conventional trucks, it can serve a wider range of vehicle applications by providing electrical power by means of an ePTO. Pure electric ‘stealth mode’ driving, which is advantageous for the provisioning of city centers and manoeuvring in emission-free (exhaust and noise) indoor logistic centres, is also one of the additional functionalities.
7.2. INTERNAL ANALYSIS

4. Emission standards

The hybrid electric distribution truck technology can be employed to meet the latest global, European, national, or municipal emission standards.

5. Reputation

DAF Trucks NV has a strong reputation on the truck market when considering the quality, reliability, and durability of its products. This is deflected by a growing market share in Western Europe (Figure 7.3, [De Ruijter, 2007]).

![Figure 7.3: market share development in Western Europe (>15t GVW)](image)

6. Cooperation

DAF Trucks NV is situated in the innovative automotive region of Eindhoven where it has the ability to cooperate with knowledge institutes like the TU/e and TNO, and innovation centres like Drivetrain Innovations (DTI).

7.2.2 Weaknesses

1. Reliability

Several weaknesses of the current hybrid drivetrain system can be identified. First of all DAF Trucks NV requires its products to be highly reliable. This reliability is at the current development stadium not (yet) at the same level as the Original Equipment Manufacturer (OEM) components. Especially the battery technology needs a lot of study and improvements.

2. Cost

The additional hybrid drivetrain components are built at a low volume and are therefore expensive (around 100 k€ per unit, Figure 7.4). Increasing the production volume will however result in an expected decrease of the selling cost [Parish, 2006] according to Figure 7.4. Unfortunately, at
the moment the supplier base is underdeveloped and therefore an increasing production volume cannot be expected yet.

![Figure 7.4: expected incremental cost development for increasing production volumes](image)

3. **Savings**

The potentials of the hybrid drivetrain seem to be strongly dependent on application where it is subjected to. The drive cycle and the utilisation are of great influence on for example the fuel consumption reduction (Chapter 6).

4. **Education**

As with all new technology, a new product like the hybrid electric truck requires further training of employees within the company. Employees need to be trained to work safely and responsibly with the high voltage components of the drivetrain.

5. **Application**

DAF Truck NV provides a wide range of products, mostly customised, for all sorts of applications. At the moment the hybrid drivetrain unfortunately only comes in one size and is thus optimised for one single application which is in contrast to other DAF products undesirable.
7.3 External analysis

The external factors are mapped by identifying market trends which do have a direct influence on selling or making the product. It is determined whether these trends are a threat or whether they are an opportunity to the company, irrelevant issues are left out. Opportunities and threats usually originate from social, economical, juridic, and technological fields. A clearly formulated mission; the (marketing) feasibility of hybrid electric distribution gives a handle which is used to scan the outer world for its influences.

7.3.1 Opportunities


Several opportunities can be perceived, most of them can be ascribed to market demands.

The extension of the existing product range by the introduction of a hybrid electric truck (range) will result in a new impetus for market share. This will give the opportunity for an improved market position [Van Amburg, 2006]. Besides this the sale of environmental friendly high tech vehicles will be an image booster for the company and enables it to obtain technological leadership within the truck market.

5. **Follow up market trends**

Within the current truck market several trends can be observed [Van Amburg, 2006] which can be followed up by the introduction of hybrid electric trucks, these trends are depicted below.

1. Productivity/performance complaints from cleaner engines; increasing vehicle performance and cleaner (diesel) engines are conflicting demands. By using the hybrid technology these demands will become less conflicting or even result in a win-win situation.

2. Incentives are proposed by governments in order to stimulate the use of environmental friendly vehicles. Hybrid trucks are such environmental friendly vehicles.

3. Especially for the US market and probably in the near future for other markets idle management will be required by law. Idle management legislation will enforce truck drivers to reduce the amount of (low output) engine idling time. Hybrid electric vehicle drivetrains are able to comply with this trend by the ability to quickly start the engine using the motor/generator and to provide electrical power for peripheral equipment without having the engine running.

4. An increasing electrical power need in heavy vehicles and their equipment is expected. The onboard generator of the hybrid electric vehicle has the ability to supply this need.

6. **New markets**

Besides the observed market trends the new technology could be used to strike new markets. An example is the army which can use the vehicle for stealth mode operations as well as to provide electric power during military operations. The reduced fuel consumption of such vehicles is an additional advantage since the fuel price at the battle field is significantly higher compared to civilised areas.
CHAPTER 7. MARKETING THE HYBRID ELECTRIC DISTRIBUTION TRUCK

7. Customer demands

From the customers point of view [Calstart, 2007] it should be remarked that most users identified fuel economy and increasing maintenance intervals as key benefits where they would want improvements over their current trucks. Reducing brake replacement costs was also identified as valuable.

7.3.2 Threads

1. Technological dependency

The success of hybrid electric drivetrain technology is (partly) depending on energy accumulator technology development. Higher power and energy density is desired to reduce weight and increase the accumulator’s performance. This accumulator, a Li-ion based battery in case of the studied vehicle, plays a major role in the performance improvement gained by hybrid electric drivetrains. Battery technology is however improving rapidly and breakthroughs are expected soon from companies like Saft, GAIA, A123 Systems, and NanoSafe.

2. Customer demands

Research done by the Hybrid Truck Users Forum [Calstart, 2007] amongst fleet owners and truck drivers indicated that new truck (drivetrain) technology should be able to meet following requirements:

1. Up-front costs are important in the logistical market and a payback of the incremental cost increase is desired within 3 years.
2. Equal reliability is demanded and can be translated to maintain costs and mean time to failure.
3. Payload is a key issue for (distribution) trucks; losing payload capacity is not acceptable.
4. The new truck should operate similarly or better - from the driver standpoint - compared to the conventional truck.
5. The vehicles drivability should be similar or better compared to the conventional vehicle.
6. The vehicle should be able to tow a trailer in all possible (hybrid) driving modes.

At this moment, where the vehicle is in the prototype stadium, the hybrid truck cannot fulfil all of these demands. Especially the first 3 items require further investigation.

3. Concurrency

Hybrid (electric) drivetrains are innovative but not a truly new technology. Several other manufacturers are developing the same sort of technology and therefore form a potential threat to the future market share of the DAF hybrid electric distribution truck. Current concurrency threats are summed below [Parish, 2006].
7.3. **EXTERNAL ANALYSIS**

- Volvo announces hybrid heavy trucks planned for 2009 production
- GM, DCX announced to be partners on hybrid technology development
- FedEx added 75 hybrid delivery vans to its fleet and plans to add 75 more, UPS commits to 50 hybrid vehicles and Purolator orders 115 more Azure hybrid delivery vans
- The city of New York ordered 500 extra Daimler Chrysler (Orion/BAE) hybrid transit buses
- The International/Eaton collaboration reported 40-60 [%] fuel consumption reduction for a hybrid utility truck
- Dana and PermoDrive became partner on commercial hydraulic hybrid vehicles and target on refuse trucks
- Eaton unveils its hydraulic hybrid shuttle bus
- Azure introduces parcel delivery and shuttle bus platforms; early production was started in late 2006
- Peterbilt is testing hydraulic hybrid refuse trucks (63000 pound GVW); pre-production was planned for 2006
- International announced to be ready for the first commercial production of Class 5-7 hybrid electric trucks in late 2006

4. **Foreign markets**

Another market threat is the fact that compared to the American and Japanese hybrid truck developments, the European hybrid truck developing companies deal with a technological inferiority as can be concluded from Chapter 2. Especially quantitatively (numbers of vehicles, variety of developed hybrid technologies) considerable arrears can be noticed.

5. **Financing**

The chance of success of the new drivetrain technology might be depending on incentives or subsidises. This is of course not desired, but as long as several expensive components are needed this financial support is required.
7.4 Market confrontation

Strategic marketing decisions are made based on both the internal and external analysis. A tool which can be used to determine the direction of the strategic actions is the confrontation matrix (Table 7.2) [Verhage, 2000]. This table shows, for each combination of external opportunities/threats and internal strengths/weaknesses the preferred action to be taken. The strategic marketing decisions which can be made by using the confrontation matrix are out of the scope of this research. For now this tool is used to give recommendations regarding the hybrid technology by discussing the outcome of each cell.

Table 7.2: confrontation matrix

<table>
<thead>
<tr>
<th>Opportunities</th>
<th>Strengths</th>
<th>Weaknesses</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Exploit</td>
<td>Improve</td>
</tr>
<tr>
<td></td>
<td>Grow</td>
<td>Restructure to strengths</td>
</tr>
<tr>
<td>Threats</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Defend</td>
<td>Avoid or withdraw</td>
</tr>
<tr>
<td></td>
<td>Compete</td>
<td>Collaborate</td>
</tr>
</tbody>
</table>

7.4.1 Strengths to defend

By using the detected strengths the organisation can defend itself against the present threats. By highlighting the unique strengths one can even restructure a threat into an opportunity. The threats that should be defended are the dependency on accumulator development (T1) and the European technological inferiority (T4) by starting research in cooperation with institutes like TNO and TU/e (S6). This is a long term development program and therefore not feasible for short term introduction of the technology on the market, it is however important to keep improving which might result in the opportunity of gaining technological leadership (O4).

The strong reputation of DAF Trucks (S5) can be employed to defend the organisation from concurrency (T3). This concurrency threat can even be restructured to an opportunity when a niche market is entered (O6) with specific and unique technology, hereby possibly gaining technological leadership (O4).

The customer (T2) can be satisfied by tuning such that operational savings (S1), productivity gains (S2), and a wide range of applications (S3) are offered.

The threatening dependency on incentives and subsidise (T5) can be restructured into an image boost (O3) (finally resulting in an improved market position (O2)) by meeting future emission standards (S4) far in advance. In this way a tendency towards governmental stimulation of the use of hybrid electric trucks can be set.

7.4.2 Weaknesses to improve/restructure to strengths

It is desired to improve the weaknesses of the hybrid technology or better, restructure them to strengths by making use of the external opportunities. The weaknesses which need to be improved are the reliability (W1) and the high cost (W2) which can be done by entering new markets
like the defence industry. For both the army and DAF Trucks it will be advantageous when DAF enters the defence market with their new technology.

The low operational savings for some applications (W3) and the fact that (at the moment) only one component size is available (W6) can be restructured to a strength when focusing on one single niche application like city distribution. When the technology becomes successful other markets can be entered. The underdeveloped supplier base (W4) only can be improved by making investments in the involved suppliers or by starting an own production facility. Finally, employees who get in touch with the new technology need to be educated (W5) with the additional benefit that technological leadership (O4) comes a little closer.

### 7.4.3 Opportunities to exploit

The present opportunities should be exploited by the found strengths as much as possible, they will possibly result in a growth of the company. The possible combinations are summed below, the several items are marked with a letter (Strength, Weakness, Opportunity, Threat) and the item number.

The opportunity of extending the product range (O1) and entering new markets (O6) can be exploited by the fact that the hybrid technology serves a wider range of applications (S3) compared to the conventional technology. Nearly all strengths (S1,2,3,4) can result in an improved market position (O2) and an image boost (O3) by following up market trends (O5) and by meeting customer demands (O7). Technological leadership (O4) can be obtained by the close cooperation with research institutes like TNO and the TU/e. The reputation which DAF Trucks already has will help exploiting these opportunities effectively.

### 7.4.4 Threats to avoid/withdraw

Combinations of weaknesses and threats are undesired. An organisation with certain weaknesses can, if justified, avoid or withdraw external threats. When considering the new technology like hybrid electric distribution trucks it should be remarked that other manufacturers deal with the same sorts of weaknesses (W1,2,3,6) and threats (T1,2,4,5), cooperation can be interesting in this case. For instance collaborating with battery manufacturers can result in several opportunities. The threat of concurrency (T3) can be avoided by entering other markets with new applications for example if company A develops hybrid refuse truck and company B develops service vehicles than DAF Trucks should be entering the specific market of city distribution specifically.
7.5 Conclusions

From this chapter it can be concluded that there certainly is a chance of success for the use of hybrid electric vehicles for distribution purposes. When analysing the SWOT content by using the confrontation matrix it can be concluded that its feasibility will be improved when:

- New markets are entered, especially niche markets are of interest
- The hybrid truck technology aims at a single application for a start
- Close cooperation is accomplished with research institutes
- Joining forces with other companies and suppliers developing (parts of) the hybrid technology and production facilities
- Encouraging the need for incentives and subsidise by involving authorities

Further (strategic) research is required for a successful introduction of hybrid electric distribution vehicles on the current or future truck market.
Chapter 8

Conclusions and Recommendations

8.1 Conclusions

Based on the research described in this thesis the following can be concluded:

1. The parallel hybrid electric drivetrain topology as described in Chapter 2 is the most suitable hybrid drivetrain concept for the purpose of road cargo distribution when intending to save fuel. Optimal energy management and smart hybridisation play an important role in reaching the imposed targets.

2. The hybrid vehicle’s Energy Management Strategy (EMS) is mainly decisive for the optimal energy flow through the hybrid drivetrain and thus mainly decisive for the performance of the total vehicle. The EMS needs to be optimised which can be done in various ways. The Dynamic Programming (DP) algorithm provides the global optimal EMS for the control of the hybrid vehicle, and is therefore chosen for the EMS analysis. The DP solution is not online implementable since it requires a priori information regarding the drive cycle. The DP results are synthesised in a Rule Based (RB) which is actually implementable. The RB controller is chosen since it provides clear understanding of the EMS and an easy way of implementation. The robustness of the RB controller is within a margin of 2 [%] compared to the optimal fuel consumption reduction calculated by DP.

3. The analysis and synthesis tool for parallel hybrid electric truck energy management uses the DP algorithm for the EMS analysis on the one hand and a RB controller in which the obtained data is synthesised on the other. When considering the studied 12t vehicle (DAF CF FA65 4x2) driving the TNO Dynamometer Cycle (TDC), the by DP calculated fuel consumption reduction of the hybrid vehicle equals 12.1 [%] compared to the base line vehicle. This theoretically obtainable reduction is for the most beneficial situation where engine is stopped during braking and standby, and the clutch is opened during the Brake Energy Recovery (BER) mode.

4. In the BER mode 100 [%] of the brake energy is assumed to be available (100 %BER). Significantly less fuel is saved for lower %BER values. Changing the BER strategy (e.g. opening or closing the clutch while BER) reduces the amount of saved fuel with approximately 2 [%] in the worst case situation. Changing the charge mode from a sustaining to a depleting strategy increases the fuel consumption reduction significantly. The depleting strategy is however not
suitable for the application of road cargo distribution since rather long drive cycles are involved.

5. The Motor Assist (MA) mode, where the motor/generator assists the engine is responsible for the majority of the gained profit followed by the Motor (M) mode and the Idle Stop (IS) mode in case of the studied 12t vehicle. The low amount of fuel saved in the IS mode is due to the short standstill periods of the TDC cycle. The engine fuel map plays the key role in determining the optimal EMS. According to the fuel map analysis, most fuel can be saved for high power regions. A truck's engine is usually operating in these regions since a truck is dimensioned on its application, in contrast to passenger cars which are mostly overpowered. This explains the low contribution of the M mode to the total amount of saved fuel for trucks, the opposite is observed for passenger cars in other studies.

6. For the studied vehicle the fuel consumption reduction of the RB EMS is only 0.9 [%] lower compared to the DP optimal EMS. The difference is due to the only local optimisation performed by the RB EMS whereas the DP EMS optimises both locally and globally. In case of the RB EMS the global optimisation is done by using a State Of Energy (SOE) neutraliser algorithm which keeps the battery energy content at a desired level.

7. Changing the vehicle’s GVW or the imposed drive cycle has great influence on the obtained fuel consumption reduction which ranges between 10 to 24 [%] in an optimal situation. A relationship between the amount of recovered brake energy in BER mode and the saved amount of fuel can be noticed. It is concluded that the fuel consumption reduction is strongly depending on the sizing of the hybrid drivetrain components. From simulations it is also proven that the MA mode is not necessarily responsible for the majority of the gained profit. For low GVW the M mode has the largest (relative) contribution to the fuel consumption reduction.

8. The robustness of the earlier discussed RB EMS is within a margin of 2 [%] compared to the DP optimal solution for most applications. Charging the battery by using the engine is desired for low GVW and urban drive cycles like the UTEC cycle according to DP. When applying the RB EMS the battery will deplete, resulting in a higher fuel consumption reduction compared to the charge sustaining situation calculated by DP.

9. Higher fuel consumption reductions can be obtained when modifying the drive cycle such that more brake energy can be recuperated. It is therefore required to increase the deceleration time (or reducing the brake power), however with preservation of the traveled distance within the designated time window. Fuel consumption reduction increases till 20.9 [%] for the analysed 12t vehicle driving a modified version of the TDC cycle. It is concluded that the recovery of brake energy in BER mode is the most important factor when reducing fuel consumption.

10. The onboard electric power capacity can be used in a synergetic way by using an electrified Power Take-Off (ePTO) system. This synergy results in a fuel consumption reduction of 14.1 [%] for the studied 12t vehicle, compared to same vehicle equipped with a Constant Speed Power Take-Off (CS-PTO) for powering the same cargo refrigerator unit. The additional cost for vehicle integration of the ePTO is determined by the cost of the required inverter.

11. The chance of a successful market introduction of hybrid electric distribution trucks is rather high. Close cooperation and joining forces with research institutes and involved suppliers is how-
ever desired. Also entering new markets and encouraging the need for incentives and subsidise will contribute substantially.
8.2 Recommendations

As with all research projects, new research questions will arise when others just have been answered. This section therefore gives the recommendations which result from the research described in this report. The first part gives the recommendations concerning future research, whereas the second part lists some recommendations regarding technological improvements of the studied hybrid electric drivetrain. The items regarding the future research have been split in vehicle modelling, drivetrain analysis, and conceptual related issues.

8.2.1 Future research

Vehicle modelling

1. An accurate analysis of the hybrid vehicle’s EMS requires accurate models of the hybrid drivetrain and the vehicle. The used battery and motor/generator efficiency map as well as some gear ratio efficiencies are determined by making several assumptions. It is therefore desired to obtain more accurate component data by measurements on the drivetrain itself. The addition of a battery SOH calculation in the battery model is also valuable. The accurateness of the engine fuel map is important since it plays the key role in determining an optimal EMS. Further research is required to obtain the dynamic behaviour of the engine in terms of fuel consumption.

2. A small amount of fuel might be saved additionally by optimising the shift strategy for the hybrid vehicle using DP. Shift strategy optimisation can be done for the engine when it is propelling the vehicle and for the motor/generator during deceleration while brake energy is recovered.

3. It is assumed that the final drive of the vehicle has different efficiencies when changing the direction of the powerflow through it. Measurements should point out the exact differences between a positive and negative power flow.

Drivetrain analysis

1. The amount of recovered brake energy holds a direct relation with the saved amount of fuel. It is thus desired to recuperate as much energy as possible. This can be done by increasing the deceleration time within standstill periods when braking the vehicle, the imposed drive cycle can be modified for this purpose. To maximise the deceleration time during the whole trip it is desired to use a route instead of a drive cycle in order to leave the vehicle speed as a variable. It is recommended to study the EMS sensitivity on a route by taking account for e.g. traffic lights, other vehicles, and traffic jams.

2. Optimal component sizing contributes to a maximum amount of brake energy recovery. The studied vehicle seemed to be optimal for GVW up to 10t and urban like drive cycles like the UTEC cycle. The size of the motor/generator as well as the charge acceptance of the energy accumulator should be optimal for the specific truck application. Further analysis of the drivetrain for different component sizes is desired.

3. Engine auxiliaries (like various pumps and the alternator) are not driven when opening the clutch and shutting off the engine during BER in case of the proposed BER strategy 1. Although it is desired to apply this BER strategy in order to increase the amount of BER, it is not desired to
8.2. RECOMMENDATIONS

lose brake pressure when the airpump is deactivated. The behaviour of crucial auxiliaries needs to be studied in order to have them electrified properly. Electrification of auxiliaries can also be an opportunity to save even more fuel when they are integrated in the hybrid drivetrain EMS.

4. From simulations it is proven that the synergy between an electrified PTO and the hybrid drivetrain is advantageous. Further analysis is however required in order to optimise the ePTO for different loadings. The power consumption of e.g. a cargo refrigeration unit as a function of time needs to be analysed for this purpose. The RB control strategy needs to be extended to anticipate for the power consumption of the ePTO.

5. When short duty cycles are involved for a certain truck application it is desired to apply a charge depleting EMS. In this way renewable energy can be used to propel the vehicle and additional fuel can be saved. This has an influence on the proposed RB control strategy, which should be made adaptive for this effect. In practise it is also desired to extend the RB EMS with an engine warm-up phase for cold starts. In this period the engine is not allowed to be shut off in order to pre-heat the catalytic materials of the exhaust gas treatment system and the engine itself to operate optimally.

6. The obtained results from the analysis and synthesis tool are based on theoretical calculations and several assumptions. The proposed RB EMS needs to be verified on the AES Heavy Duty Chassis Dynamometer (Appendix E). For reasons of timing this experimental analysis is not performed within the graduation period.

Hybrid concepts

1. Anticipated driving behaviour can increase the amount of recovered brake energy. Research for adaptive driving behaviour is needed and the development of some sort of drivers-aid is desired to help the driver recuperating as much brake energy as possible. For this purpose one can think of using radar or GPS sensing technologies or possibly present sensors from e.g. Adaptive Cruise Control (ACC) or navigation systems.

2. Smart electrification of engine auxiliaries is useful but requires hardware modifications. An example of such modification is when the motor/generator is used to drive the engine auxiliaries directly. A design for such modification requires further research.

3. EMS optimisation can be done in several ways as discussed in Chapter 3. A promising method is predictive control and the use of GPS data, called e-horizoning. This might be an opportunity for future research.

4. Combinations of the current Li-ion battery with storage systems like supercapacitors are worthwhile to consider in order to improve cost and lifetime as well as the performance of the energy accumulator.

5. The parallel hybrid drivetrain can be employed in order to improve the vehicles comfort and drivability. The closing of the clutch and the engine shut down can be smoothen by activating the motor/generator. The motor/generator can also be used to boost the vehicle performance during overtaking other traffic.
8.2.2 Technological improvements

1. One of the major weaknesses of a hybrid electric drivetrain is the (Li-ion) battery pack. Lifetime and cost improvements are required when the vehicle is put on the market.

2. Finishing the non-driving tooth flanges of the final drive is required in order to increase the efficiency of this gear ratio during BER. The efficiency should be at the same level of 96 [%] as other gear ratio’s to maximise the amount of BER.

3. Decreasing the final drive ratio brings the engine down to lower speeds which saves another portion of fuel. The performance deficits can be compensated by using the motor/generator which enables the application of powershift technology.

4. It is proposed to apply 100 %BER on the rear axle, which means that the front brakes are not used during deceleration until saturation of the motor/generator. Since legislations require trucks to use both front and rear brakes for braking it should be inquired whether it is allowed to (limitedly) use the rear brakes only.

5. Extending the applications of the parallel hybrid electric drivetrain is desired for marketing reasons. Stealth mode driving and ePTO applications are examples which require further research and development.
Dankwoord

Dit werk is tot stand gekomen met behulp van velen. Het coöperatieve karakter van het project heeft daar natuurlijk zijn bijdrage aan geleverd. Mensen van DAF Trucks NV, Leyland Trucks Ltd, TNO en natuurlijk de TU/e zijn betrokken geweest bij het aanleveren van gegevens, het beantwoorden van vragen en het leveren van terugkoppeling op het gerealiseerde werk. In het bijzonder zou ik graag de volgende mensen willen bedanken.

Allereerst Theo de Ruijter en het DAF projectteam voor hun bijdrage in de totstandkoming van dit onderzoek en de prettige samenwerking met DAF. Hopelijk kunnen we dit in de toekomst voortzetten. Naast DAF is ook dank verschuldigd aan Leyland Trucks onder leiding van Tony Ellis voor het beschikbaar stellen van alle noodzakelijke voertuiggegevens en de gastvrijheid tijdens het bezoek aan de Leyland fabrieken. Via dit schrijven wil ik ook Salem Mourad van TNO bedanken voor zijn bijdrage aan onder andere de Universiteitsquiz, vorig jaar. Dimitri de Rooij, tevens van TNO, wil ik bedanken voor het meedenken in het proces van het onderzoek aan hybride aandrijvingen.

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Tot slot wil ik Thijs van Keulen heel veel succes toewensen met de voortzetting van dit mooie project!
# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>road gradability</td>
<td>[rad]</td>
</tr>
<tr>
<td>$\dot{m}_f$</td>
<td>fuel mass flow</td>
<td>[g/s]</td>
</tr>
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<td>cyclo converter efficiency</td>
<td>[-]</td>
</tr>
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</tr>
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<td>$\eta_{CS-PTO}$</td>
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<td>$\eta_{DC/AC}$</td>
<td>inverter efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_{fd+}$</td>
<td>final drive efficiency while accelerating</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_{fd-}$</td>
<td>final drive efficiency while decelerating</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_{gb}$</td>
<td>gearbox efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_{mg}$</td>
<td>motor/generator efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\mu$</td>
<td>tyre friction coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>$\omega_e$</td>
<td>angular engine speed</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>$\omega_v$</td>
<td>angular vehicle (wheel) speed</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>$\omega_{e,idle}$</td>
<td>angular engine idle speed</td>
<td>[rad/s]</td>
</tr>
<tr>
<td>$\omega_m$</td>
<td>ingoing transmission shaft speed</td>
<td>[rad/s]</td>
</tr>
<tr>
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<td>[g]</td>
</tr>
<tr>
<td>$\rho_{air}$</td>
<td>air density</td>
<td>[kg/m$^3$]</td>
</tr>
<tr>
<td>$\rho_{fuel}$</td>
<td>fuel density</td>
<td>[kg/m$^3$]</td>
</tr>
<tr>
<td>$\tau$</td>
<td>time window</td>
<td>[s]</td>
</tr>
<tr>
<td>$\xi_{CH}$</td>
<td>charge rate</td>
<td>[-]</td>
</tr>
<tr>
<td>$A$</td>
<td>frontal area</td>
<td>[m$^2$]</td>
</tr>
</tbody>
</table>
NOMENCLATURE

\begin{itemize}
\item \(A\) surface area \([m^2]\)
\item \(a, b, c, h\) vector \([-\]
\item \(A, B, H, L, Q\) matrix \([-\]
\item \(BS FC\) Brake Specific Fuel Consumption \([g/kW h]\)
\item \(C\) DP cost matrix \([g]\)
\item \(C_w\) air drag coefficient \([-\]
\item \(C_{max}\) maximal battery capacity \([Wh]\)
\item \(C_{nom}\) nominal battery capacity \([Wh]\)
\item \(d\) thickness \([m]\)
\item \(dE_s\) energy grid step size \([J]\)
\item \(dP_s\) power grid step size \([s]\)
\item \(dt\) time step \([s]\)
\item \(E_s\) stored energy (state variable) \([J]\)
\item \(F_n\) normal force \([N]\)
\item \(f_r\) roll resistance coefficient \([kg/ton]\)
\item \(g\) gravitational acceleration \([m/s^2]\)
\item \(GVW\) Gross Vehicle Weight \([kg]\)
\item \(H\) height \([m]\)
\item \(h_0\) scalar \([-\]
\item \(H_{low}\) lower heating value \([MJ/kg]\)
\item \(i_{fd}\) final drive ratio \([-\]
\item \(i_{gb}\) gearbox ratio \([-\]
\item \(J\) cost function \([-\]
\item \(J_c\) clutch inertia \([kg.m^2]\)
\item \(J_e\) engine inertia \([kg.m^2]\)
\item \(J_f\) fuel cost function \([-\]
\item \(J_v\) vehicle inertia \([kg.m^2]\)
\item \(J_w\) wheel inertia \([kg.m^2]\)
\item \(J_{eq}\) total equivalent inertia \([kg.m^2]\)
\end{itemize}
NOMENCLATURE

\( J_{mg} \) motor/generator inertia [kg\( \cdot m^2 \)]
\( k \) thermal conductivity [W/mK]
\( K_s \) SOE gain [\( - \)]
\( K_{s,CH} \) CH SOE gain [\( - \)]
\( K_{s,MA} \) MA SOE gain [\( - \)]
\( L \) DP cost-to-go matrix [g]
\( L \) length [m]
\( m \) mass [kg]
\( m_f \) fuel mass [g]
\( N \) DP node matrix [\( - \)]
\( n_e \) engine speed [rpm]
\( n_{mg} \) motor/generator speed [rpm]
\( p \) proportional gain [\( - \)]
\( P_b \) brake power [W]
\( P_e \) engine power [W]
\( P_m \) power at ingoing transmission shaft [W]
\( P_s \) control power (control variable) [W]
\( P_v \) vehicle power demand [W]
\( P_{ch,max} \) maximal battery charge power [W]
\( P_{cycle} \) drive cycle power [kW]
\( P_{dis,max} \) maximal battery discharge power [W]
\( P_{e,refr} \) electrical refrigeration power [W]
\( P_{mg,\text{opt}} \) optimal motor/generator power [W]
\( P_{mg} \) motor/generator power [W]
\( P_{s,\text{opt}} \) optimal battery power [W]
\( P_{s,\text{refr,chg}} \) stored refrigeration power penalty [W]
\( P_{s,\text{refr,dis}} \) stored refrigeration power in battery mode [W]
\( P_{s,refr} \) stored refrigeration power [W]
\( R_0 \) unloaded wheel radius [m]
NOMENCLATURE

\( R_e \) effective wheel radius \([m]\)
\( R_i \) internal electrical resistance \([\Omega]\)
\( S \) DP state space grid \([-]\)
\( SOE \) state of energy \([\%]\)
\( SOE_{\text{max}} \) maximal battery state of energy \([\%]\)
\( SOE_{\text{min}} \) minimal battery state of energy \([\%]\)
\( T \) temperature \([K]\)
\( t \) time \([s]\)
\( T_b \) brake torque \([Nm]\)
\( T_e \) engine torque \([Nm]\)
\( T_m \) torque at ingoing transmission shaft \([Nm]\)
\( T_{\text{CH,high}} \) higher CH mode torque \([Nm]\)
\( T_{\text{CH,low}} \) lower CH mode torque \([Nm]\)
\( t_{\text{CPU}} \) calculation time \([s]\)
\( T_{\text{CS-PTO}} \) CS-PTO torque \([Nm]\)
\( T_{\text{cycle}} \) drive cycle torque \([Nm]\)
\( T_{e,\text{aux}} \) engine auxiliary torque \([Nm]\)
\( T_{e,\text{drag}} \) engine drag torque \([Nm]\)
\( T_{e,\text{exh}} \) engine exhaust brake torque \([Nm]\)
\( T_{\text{load}} \) load torque \([Nm]\)
\( T_{\text{MA,high}} \) higher MA mode torque \([Nm]\)
\( T_{\text{MA,low}} \) lower MA mode torque \([Nm]\)
\( T_{\text{mg}} \) motor/generator torque \([Nm]\)
\( T_{\text{pto}} \) power take-off torque \([Nm]\)
\( t_{\text{shift}} \) shift duration \([s]\)
\( u \) DP control vector \([-]\)
\( v_{\text{act}} \) actual vehicle speed \([m/s]\)
\( W \) width \([m]\)
\( wb \) wheelbase \([m]\)
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<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>$x$</td>
<td>control variable</td>
<td></td>
</tr>
<tr>
<td>$x_{act}$</td>
<td>actual displacement</td>
<td>$[m]$</td>
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Bibliography


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Appendix A

Effective tyre roll radius

unloaded radius for 275-70R22.5 tyre:

\[ R_0 = 0.47825 \text{ m} \]  
unloaded tyre radius

obtained from MF tyre parameter file for the 315-80R22.5 tyre:

\[ C_z = 1e+006 \]  
 tyre vertical stiffness \([\text{N/m}]\)

\[ B = 2.2 \]  
 low load stiffness \([\text{N/m}]\)

\[ F = -0.046 \]  
 high load stiffness \([\text{N/m}]\)

\[ D = 1.0414 \times R_0 \]  
 peak value of e.r.r. \([\text{m}]\)

\[ F_{z0} = 30000 \]  
 nominal wheel load \([\text{N}]\)

FA CF65 truck parameters:

\[ g = 9.81 \]  
 gravitational coefficient \([\text{m/s}^2]\)

\[ n = 4 \]  
 number of wheels on rear axle [-]

\[ \text{max}_\text{mass} = 18000 \]  
 max vehicle mass (CF65) \([\text{kg}]\)

\[ \text{max}_\text{axle}_\text{load} = 11500 \]  
 max rear axle load \([\text{kg}]\)

\[ \text{GVW} = 4000:1000:18000 \]  
 gross vehicle weight \([\text{kg}]\)

\[ \text{load}_\text{front} = 0.36 \]  
 load on front axle \((\geq 20\% \text{ by law})\)

\[ F_{\text{axle}\_\text{rear}} = \min(\max((\text{GVW} - \text{max}_\text{mass} \times \text{load}_\text{front}) \times g, \text{GVW} \times g \times (1 - \text{load}_\text{front})), \text{max}_\text{axle}_\text{load} \times g) \]  
 rear axle load \([\text{N}]\)

\[ F_{\text{axle}_\text{front}} = \text{GVW} \times g - F_{\text{axle}\_\text{rear}} \]  
 front axle load \([\text{N}]\)

\[ F_z = F_{\text{axle}_\text{rear}} / n \]  
 single rear wheel load \([\text{N}]\)

Effective tyre roll radius for the 275-70R22.5 tyre:

\[ \rho = F_z / C_z \]  
 radial tyre deflection \([\text{m}]\)

\[ \rho_{\text{Fz0}} = F_{z0} / C_z \]  
 nominal tyre deflection \([\text{m}]\)

\[ \rho_d = \rho / \rho_{\text{Fz0}} \]  
 dimensionless radial tyre deflection [-]

\[ R_e = R_0 - (\rho_{\text{Fz0}} \times (D \times \arctan(B \times \rho_d)) + F \times \rho_d) \]  
 effective tyre roll radius \([\text{m}]\)
Appendix B

Wheel inertia calculation

material properties
rho tyre = 1300 [kg/m^3]
rho steel = 7950 [kg/m^3]

tyre walls (twice)
r1 = 22.5*25.4/2 = 286 [mm] = 0.286 [m]
r2 = 0.7*275+22.5*25.4/2 = 478 [mm] = 0.478 [m]
d = 5 [mm] = 0.005 [m] (approximated)
V= 0.00492 [m^3]
m = rho*V = 5.99 [kg]
J = m*(r1^2+r2^2)/2 = 0.93 [kg.m^2]

tyre tread
r = 0.7*275+22.5*25.4/2 = 478 [mm] = 0.478 [m]
d = 5 [mm] = 0.005 [m] (approximated)
b = 0.275 [m]
m = rho*V = 5.34 [kg]
J = m*r^2 = 1.22 [kg.m^2]

rim centre
r = 22.5*25.4/2 = 286 [mm] = 0.286 [m]
d = 3 [mm] = 0.003 [m] (approximated)
m = rho*V = 6.13 [kg]
J = m*r^2/2 = 0.25 [kg.m^2]

rim bed
r = 22.5*25.4/2 = 286 [mm] = 0.286 [m]
d = 3 mm = 0.003 [m] (approximated)
b = 0.275 [m]
m = rho*V = 11.72 [kg]
J = m*r^2 = 0.96 [kg.m^2]

wheel assembly
m = 29.18 [kg]
Jw = 3.36 [kg.m^2]
Appendix C

DAF V40 simulation data

Vehicle: FA65_136
Engine: CE 136 C0
Clutch: Default VMS clutch
Transmiss.: ZF 6S850 8.51-1.00
Rear axle: SR 1132 4.10
Driv. wheel: 275/70-R22.5
Retarder: Not present
Cool.syst.: Not present
Mass: 12.00 ton
Roll.res.: 8.00 kg/ton
P.T.O.: 0.00 kW
Front. Area: 8.000 m2
CDrag Eff.: 0.726
CD modelkey: -
Nr.of axles: 2
Topspeed: 121.7 km/h

<table>
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<tr>
<th>Route</th>
<th>Length: 12.346 km</th>
<th>Perc. dist. 50 %</th>
<th>Strive speed 40.0 km/h</th>
<th>Veh. mass 12.00 ton</th>
<th>CDrag 0.726</th>
<th>Wind dist. -</th>
<th>Fuel cons. V40Fuel01.csv</th>
<th>Sim. data A03PART01.csv</th>
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Table C.1: V40 simulation data
Appendix D

Simulation results

Figure D.1: fuel consumption for various drive cycles
(○ = BL; × = RB; △ = DP)
Figure D.2: theoretical optimal fuel consumption reduction (BL-DP)

(simulation data: □ = UTEC; ◆ = FTP; △ = TDC)
Figure D.3: practical optimal fuel consumption reduction (BL-RB)

(simulation data: □ = UTEC; ○ = FTP; △ = TDC)
APPENDIX D. SIMULATION RESULTS

Figure D.4: RB EMS robustness (RB-DP)
(simulation data: □ = UTEC; o = FTP; △ = TDC)
Figure D.5: distribution of MA, M, and IS mode (BL-DP)/BL.
Appendix E

AES Heavy Duty Chassis Dynamometer

The Heavy Duty Chassis Dynamometer of the Automotive Engineering Science department (AES) at the TU/e is developed and built in order to fulfill a broad spectrum of automotive test applications, among others heavy duty vehicles. The history of the original chassis dynamometer which served as a basis for the newly developed system dates back to the sixties. At that time it was equipped with a mechanical gauge which measured the reaction force applied by the vehicle to be tested (Figure E.3). Nowadays the system includes a user interface including data logging functionality and a drivers aid for road load testing as well as a safety system with integrated peripheral equipment operation.

![Figure E.1: the chassis dynamometer in the year 2007](image)

The AES Heavy Duty Chassis Dynamometer is fairly unique in its sort due to the large rolls ($R = 1 \ [\text{m}]$) and the ability to add energy to the analysed drivetrain which is of interest when testing hybrid vehicles. The energy which is released by the vehicle when accelerating or or driving at a constant speed is recovered and sent back to the electrical grid by the frequency convertor. Automated testing and exhaust gas analysers can be add for future extension. The current chassis dynamometer layout is depicted in Figure E.2 while the photographs of Figure E.1 which shows the final result and an impression of the wide range of applications.
Depending on the application, testing can be done at circumferential speeds up to 225 \([km/h]\). The specification sheet of Table E.1 shows the systems limitations for each application.

<table>
<thead>
<tr>
<th>application</th>
<th>(v_{\text{max}}) ([km/h])</th>
<th>(a_{\text{max}}) ([m/s^2])</th>
<th>(m_{\text{max}}) ([kg])</th>
<th>(F_{\text{vert,max}}) ([kN])</th>
<th>(F_{\text{lat,max}}) ([kN])</th>
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<th>(P_{\text{max,cont}}) ([kW])</th>
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<td>7500</td>
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<td>-</td>
<td>20</td>
<td>45</td>
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Table E.1: AES Heavy Duty Chassis Dynamometer data sheet
Figure E.3: the chassis dynamometer in the year 1961