Hybrid component specification optimization for a medium-duty hybrid electric truck

Abstract: This paper presents a modeling and simulation approach for determining the optimal degree-of-hybridization for the drive train (engine, electric machine size) and the energy storage system (battery, ultra capacitor) for a medium-duty truck. This approach includes the control strategy for optimal energy flow in the hybrid electric drive train. Using this approach, the influence of the gross-vehicle weight on optimal component sizing has been investigated. The results show that the degree-of-hybridization of known medium-duty hybrid electric trucks is close to the optimal degree-of-hybridization using the methods as described in this paper. Furthermore, it is found that the Li-ion battery (single storage) is from an energy - and power density specification as well as cost point of view the most preferable energy storage system. However, if the cost of ultra capacitor cells are significantly decreased (> 50%) then hybridization of a Li-ion battery with an ultra capacitor module in combination with a boost converter may become an attractive technology package in the future.

1 Introduction

Medium-duty trucks are used in different transport activities ranged from urban - and regional distribution to light-weight transport over long distances and special applications, such as used at the municipal cleaning department and the fire department. The diesel engine efficiencies used in these types of vehicles are al-
ready relatively high compared to petrol engines. Moreover, the potential of weight and air drag reduction is constrained by the payload carrying requirements. Nevertheless, advanced hybrid propulsion systems are very promising to achieve the future fuel economy and emission goals for trucks in this segment (Wu et al., 2004).

In Table 1 an overview of the component specifications of three different realized medium-duty hybrid electric trucks (parallel hybrid type) are listed. Notice that the hybrid trucks use three different electrical storage systems with different energy and power characteristics (Li-ion, Ni-MH and Ultra Capacitor (UC)).

Table 1  Reference medium-duty hybrid electric trucks

<table>
<thead>
<tr>
<th>Component</th>
<th>Unit</th>
<th>FedEx W700</th>
<th>Nissan Condor</th>
<th>Hino 4T Ranger</th>
<th>This paper</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>kW/Nm</td>
<td>125/569</td>
<td>152/500</td>
<td>132/461</td>
<td>125-205</td>
</tr>
<tr>
<td>Displacement</td>
<td>cil.</td>
<td>4/4.3</td>
<td>6/6.93</td>
<td>4/4.73</td>
<td>6/5.2-8/8.5†</td>
</tr>
<tr>
<td>Motor</td>
<td>kW/Nm</td>
<td>44/420</td>
<td>55/130</td>
<td>23/*</td>
<td>20/138-80/552†</td>
</tr>
<tr>
<td>Transmission</td>
<td>spd/type</td>
<td>6/AMT</td>
<td>6/AMT</td>
<td>*</td>
<td>6/AMT</td>
</tr>
<tr>
<td>Storage</td>
<td>V/Ah</td>
<td>340/7.2</td>
<td>346/60-kW</td>
<td>274/6.5</td>
<td>130-606/6</td>
</tr>
<tr>
<td>Type</td>
<td>Li-ion</td>
<td>UC</td>
<td>Ni-MH</td>
<td>Ni-MH</td>
<td></td>
</tr>
<tr>
<td>Mass</td>
<td>kg</td>
<td>7257</td>
<td>7756</td>
<td>3629</td>
<td>4000-12000</td>
</tr>
<tr>
<td>Cycle</td>
<td>-</td>
<td>FTP-75†</td>
<td>*</td>
<td>JP16 mode</td>
<td>FTP-75†</td>
</tr>
<tr>
<td>HFdt</td>
<td>%</td>
<td>26%</td>
<td>27%</td>
<td>15%</td>
<td>10%-40%</td>
</tr>
<tr>
<td>Fuel</td>
<td>km/l</td>
<td>+45%</td>
<td>+50%</td>
<td>+20%</td>
<td>Section 4</td>
</tr>
</tbody>
</table>

* = no data available, AMT = Automated Manual Transmission, † = modified cycle, ‡ = assuming linear scaling engine displacement or maximum torque with peak power

The control design of medium-duty hybrid electric or hydraulic trucks is extensively discussed in literature, see for instance (Lin et al., 2004), (Assanis et al., 2000), (Wu et al., 2004). However, not much work can be found specifically related to the overall design of a medium-duty hybrid electric truck to fuel economy and performance. The work presented in (Filipi et al., 2004) discusses the combined optimization of component design and power management of a hydraulic hybrid drive train for a 6x6 medium truck and comes close to the work presented in this paper. One of the main differences is that in (Filipi et al., 2004) Dynamic Programming (DP) is used for optimization of the pre-defined rule-based
Energy Management Strategy (EMS). The EMS optimization as presented in this paper is performed using a novel Rule-based EMS consisting of the combination of Rule-based - and Equivalence Consumption Minimization Strategies (Hofman et al., 2007). This speeds up the control design process and therefore the overall design process significantly. Therefore, in this paper we would like to focus on determining the optimal degree-of-hybridization and suitability of electrical storage technology (Li-ion, Ni-MH and UC) maximizing the fuel economy for a medium-duty hybrid electric truck as a design case study. At the optimal degree-of-hybridization is the vehicle performance (i.e. the acceleration time from 0-100 km/h (0-62 mph) and the maximum gradeability at 89 km/h (55 mph)) for the different energy storage systems determined. Maximum gradeability is defined as the maximum slope angle on which a vehicle is able to drive at the maximum combined output power (engine, electric machine). The iterative optimization design process is depicted in Figure 1. As an example application study estimated vehicle parameters and a given drive cycle (modified FTP-75, see, (Lin et al., 2003)) for the FedEx truck are used. After determining the optimal degree-of-hybridization the overall mass of an optimized dual-storage system is determined for a Li-ion and a Ni-MH battery in combination with an ultra capacitor pack. The Hybridization Factor (HF$_{dt}$) for a hybrid drive train is defined as (Lukic and Emadi, 2004),

$$HF_{dt} = \frac{P_{em,max}}{P_{em,max} + P_{e,max}} \cdot 100\%, \quad (1)$$

with $P_{em,max}$ and $P_{e,max}$ representing the maximum (continuous) electric machine power and engine power respectively.

2 Modeling Method and Assumptions

Due to the complexity of hybrid vehicle drive trains, the design of topologies, component technologies and the control strategy forms a considerable challenge for engineers (Butler et al., 1999), (Assanis et al., 1999), (Guzzella and Amstutz, 1999).
In order to alleviate the complex design problem at hand, in this paper:

- simplified power-based efficiency functions for each component are discussed, and

- in addition, a novel EMS algorithm (Hofman et al., 2007) based on the combination of Rule-Based - and Equivalent Consumption Minimization Strategies (Musardo et al., 2005), (Sciaretta et al., 2004) (RB-ECMS) is used, with which the fuel economy can be calculated very quickly and with sufficient accuracy.

In Figure 2 an overview of the modeling and simulation process as discussed in this paper is shown. The energy conversion components (engine, electric machine) are assumed to be operated at their maximum efficiency points. This assumption allows that the component efficiencies modeled as power-based functions can be described by a few characteristic parameters. The energy storage components for two different battery technologies (Li-ion, Ni-MH) and UC are also modeled using simplified power-based functions. The influence of component scaling on the characteristic parameters and the fuel economy will also be discussed.
Furthermore, the effects of (i) the gross-vehicle weight $m_v$ and (ii) hybridization of the energy storage system (battery, ultra capacitor) at the optimal hybridization factor as mentioned before for the drive train of a medium-duty truck is investigated. Typically, $m_v$ changes over time during picking up and delivering of goods. Since $m_v$ plays an important role on the fuel economy and driveability, the influence of different constant values for $m_v$ over a whole drive cycle has been investigated.

Finally, the second effect studies the influence of hybridization of the energy storage system on the overall energy storage size. Thereby, in this paper, we define an additional hybridization factor for the energy storage system, denoted as $HF_{es}$:

$$HF_{es} = \frac{P_{b,max}}{P_{b,max} + P_{uc,max}} \cdot 100\%,$$

with $P_{b,max}$ and $P_{uc,max}$ representing the maximum battery power and ultra capacitor power respectively. The size of a battery pack is usually constrained by power - and not by energy limitations, and vice-versa for an ultra capacitor pack. The advantage of using a dual-storage system is the reduction of the power demands to the battery and therefore the aging of batteries and should increase the lifetime significantly compared to the single-storage system. In addition, the cost would decrease and efficiency of the energy source would increase (Baisden and Emadi, 2004).

Since actual component data of the FedEx truck is not available, selected component data for the engine, electric machine and battery from ADVISOR (NREL, 2002) are used with specifications close to the actual components of the FedEx hybrid truck. The component characteristics (mass, maximum torque curve and static efficiency map) of the base engine and electric machine are linearly scaled as needed. The absolute mechanical (in-)output power during motoring and generating is assumed to be equal, whereas the electrical (in-)output power due to losses are differently. Furthermore, the maximum electric machine speed is assumed to be 2100 rpm equal to the maximum engine speed. The base component characteristics used for scaling and the transmission technology are shown in Table 2. The main characteristics of the storage components (Li-ion, Ni-MH and UC) are shown in
Table 3. The other vehicle simulation input parameters are listed in Table 4.

Auxiliary loads (e.g., air condition, cooling systems, heating of seats) can be in the order of 6-10 kW (Kessels et al., 2005) and play therefore an important role in the overall fuel economy. However, only the minimum necessary engine auxiliary loads (dynamo, waterpump, power steering, airpump for the brakes) have been taken into account by assuming an average constant auxiliary load torque of 10 Nm as a function of engine speed at the engine crack shaft, resulting in an average auxiliary power of 505 W for the Base Line (BL) truck.

Furthermore, in this paper, the fuel use for engine cranking for the Hybrid Electric Vehicle (HEV) has been neglected, because the engine can be started with the electric machine in a very short time period (typically < 500 ms). The required electrical energy for engine start (usually the engine is started when the vehicle is already driving) is therefore limited and very small, but the power is relative high. For simplicity, the required battery energy for start-stop is also left out of consideration.

For the remainder of this paper the outline will be as follows (see also Figure 2): in Section 3, the component models, the influence of component sizing on
the characteristic parameters, and the dual-storage system design model will be discussed. In Section 4, the simulation results are given. Finally, the conclusions are discussed in Section 5.

Table 2  Base engine, electric machine and transmission characteristics

<table>
<thead>
<tr>
<th>Description</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Engine</td>
<td>Detroit diesel series 50, 8-cylinder, 205-kW peak power (at 2100 rpm), 8.5-l CI, 1192-Nm peak torque (at 1200 rpm), 44% peak efficiency, mass 860 kg (FC_CI205*)</td>
</tr>
<tr>
<td>Base Electric Machine/Controller</td>
<td>PM brushless AC motor, 58-kW peak power (at 1250 rpm), 400-Nm peak torque, 92% peak efficiency, mass 70 kg (MC_PM58*)</td>
</tr>
<tr>
<td>Base Transmission</td>
<td>Eaton, type FSO-8406A, 6 speed, Automated Manual, maximum transmission input torque 1166 Nm, mass 359 kg</td>
</tr>
</tbody>
</table>

*model data file from ADVISOR

Table 3  Different energy (electrical) storage characteristics

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Storage type</th>
<th>Nominal voltage (V/cell)</th>
<th>Current capacity (Ah)</th>
<th>Storage mass (kg/cell)</th>
<th>No. cells/module</th>
<th>Model data file (ADVISOR)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Panasonic</td>
<td>Ni-MH</td>
<td>1.2</td>
<td>6</td>
<td>0.166</td>
<td>6</td>
<td>ESS_NIMH6</td>
</tr>
<tr>
<td>Saft</td>
<td>Li-ion</td>
<td>3.6</td>
<td>6</td>
<td>0.375</td>
<td>6</td>
<td>ESSLi7_temp</td>
</tr>
<tr>
<td>Maxwell</td>
<td>UC</td>
<td>2.7</td>
<td>-</td>
<td>0.460</td>
<td>1</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 4  Simulation model parameters

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total mass</td>
<td>$m_v$</td>
<td>7258(16000)</td>
<td>kg(lbs)</td>
</tr>
<tr>
<td>Frontal area</td>
<td>$A_f$</td>
<td>6.9</td>
<td>m²</td>
</tr>
<tr>
<td>Air drag coefficient</td>
<td>$c_d$</td>
<td>0.73</td>
<td>-</td>
</tr>
<tr>
<td>Rolling resistance</td>
<td>$c_r$</td>
<td>0.8%</td>
<td>-</td>
</tr>
<tr>
<td>Air density</td>
<td>$\rho$</td>
<td>1.2</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Gravity</td>
<td>$g$</td>
<td>9.81</td>
<td>m/s²</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>$r_w$</td>
<td>0.3970</td>
<td>m</td>
</tr>
<tr>
<td>Traction coefficient rear-wheels</td>
<td>$\mu_r$</td>
<td>0.9</td>
<td>-</td>
</tr>
<tr>
<td>Final drive ratio</td>
<td>$r_d$</td>
<td>0.30</td>
<td>-</td>
</tr>
<tr>
<td>Speed ratio set</td>
<td>$R$</td>
<td>{0.14; 0.24; 0.40; 0.63; 1.00; 1.28}</td>
<td>-</td>
</tr>
<tr>
<td>Regenerative brake fraction</td>
<td>$f_{rb}$</td>
<td>0.4</td>
<td>-</td>
</tr>
<tr>
<td>Transmission efficiency</td>
<td>$\eta_{AMT}$</td>
<td>0.96</td>
<td>-</td>
</tr>
<tr>
<td>Final drive efficiency</td>
<td>$\eta_d$</td>
<td>0.90 (generating), 0.96 (propulsion)</td>
<td>-</td>
</tr>
<tr>
<td>Height center of mass</td>
<td>$H$</td>
<td>1.3</td>
<td>m</td>
</tr>
<tr>
<td>Distance between the rear wheel and the center of mass</td>
<td>$L_r$</td>
<td>2.5</td>
<td>m</td>
</tr>
<tr>
<td>Length of wheel base</td>
<td>$L$</td>
<td>4.8</td>
<td>m</td>
</tr>
<tr>
<td>Inertia of rotating parts</td>
<td>$J_{\omega}$</td>
<td>20</td>
<td>kg m²</td>
</tr>
</tbody>
</table>
3 Modeling of Components

In Figure 3 the power flow in the hybrid drive train is shown. The drive train is a backwards facing or differentiating model, whereby the vehicle speed used as input is tracked exactly.

![Diagram of power flow in the hybrid drive train](image)

Figure 3: Power flow in the hybrid drive train (backwards facing control model)

Using the vehicle parameters as listed in Table 4 and the modified FTP-75 drive cycle (used for FedEx truck) the vehicle drive power demand as a function of time $P_v(t)$ can be calculated. The amount of recoverable brake energy plays an important role in the achievable fuel saving potential (Hofman et al., 2004). The braking energy ratio to the total traction energy is defined as:

$$\epsilon_{br} = 100\% \cdot \frac{\int_0^{t_f} P_v(t)P_v(t) < 0 \, dt}{\int_0^{t_f} P_v(t)P_v(t) > 0 \, dt} \tag{3}$$

for a drive cycle with time length $t_f$ and vehicle parameters (Table 4, 7.3 tons truck) the fraction $\epsilon_{br}$ reaches 48%. However, due to transmission losses (final drive, Automated Manual Transmission (AMT)) this ratio $\epsilon_{br}$ is reduced to 38.4%.

Evidently, the brake force distribution between the front and the rear wheels plays a key role in the amount of recuperated brake energy, since we assume that only the rear brakes are used. The brake force distribution ratio $f_{fb}(t)$ changes over time and is dependent on the amount of deceleration with $a_v(t) := -\min(0, a_v(t))$ (Fenton, 1998):

$$f_{fb}(t) = \frac{L_r}{L} + \frac{a_v(t)}{g} \frac{H}{L} \tag{4}$$

with the (vehicle) parameters $L_r$, $L$, $H$ and $g$ representing the distance between the rear wheels and the center of mass, the length of the wheel base, the height
of the center of mass and the gravity constant respectively. This ratio varies for a 7.3 tons truck between 53%-60% assuming that the adhesive capability between the road and the tires could be fully utilized (ideally braking). The regenerative brake fraction defined as the ratio between brake force between the rear wheel and the front wheels becomes with help of Equation (4):

$$f_{rb}(t) = 1 - f_{fb}(t)$$ (5)

Note that the values for $L_r$ and $H$ as listed in Table 4 are estimated values, because these values are usually not given by the manufacturer and are difficult to obtain.

Figure 4 shows the vehicle speed $v_v(t)$, the power demand $P_v(t)$ and regenerative brake fraction $f_{rb}(t)$ for a 7.3 tons truck. In this paper without loss of generality a constant regenerative brake fraction $f_{rb} := \min(f_{br}(t))$, based on the calculated $f_{rb}(t)$ (see Figure 4), of 40% is assumed, because the actual brake strategy is not known. In addition, for safety reasons mechanical or hydraulic back-up braking systems are still required and fully ‘brake-by-wire’ systems are yet not applicable. This causes that $\epsilon_{br}$ is significantly reduced from 38.4% to 15.4%. The remaining part of 60% of the total brake power is dissipated in the front wheel brake discs. The brake power in the rear wheels is regenerated up to the maximum generative power limitation of the battery/electric machine, larger brake powers are assumed to be dissipated in the wheel brake discs.

In the following sections the power-based component efficiency -, control design - and vehicle dynamics simulation model are discussed, which are used to calculate the fuel economy (l/100km), the acceleration time from 0-100 km/h (0-62 mph), and the gradeability in percentage at 89 km/h (55 mph).
3.1 Power-based Component Efficiency Models

In this paper, the energy conversion - and storage devices are modeled as power-based efficiency functions,

\[ P_{in} = P_{out} + P_{loss}(\cdot) = \phi(\cdot) \approx \sum_{j=0}^{j=n} c_j(\cdot) P_{out}^j, \quad \{j\} \subseteq \mathbb{N} \]  

with \( \phi(\cdot) \) defined as the inverse efficiency \( \eta^{-1}(\cdot) \) times the output power \( P_{out} \), which are approximated by polynomial fit functions (De Jager, 2003). For the engine and Electric Machine (EM) the losses \( P_{loss}(\cdot) \) are a function of the output power \( P_{out} \) and angular speed \( \omega \), or in case of a battery a function of the state-of-charge \( \xi \), the battery power \( P_s \) and the temperature \( T \) (Koot et al., 2005). In this section, the derivation of the characteristic parameters \( c_j(\cdot) \) describing the component efficiency for the engine, electric machine and battery technologies will be discussed.

Engine and Electric Machine Efficiency: For the engine and the electric machine, the static power losses \( P_{loss}(\cdot) \) at zero output torque are dependent on the speed.
ωᵢ. At zero output torque no measurement data is available. Since the static losses of an engine play an important role in calculating the fuel economy improvement, the static losses are estimated by linear extrapolating the fuel mass flow curves to zero output torque for the different given speeds. The characteristic parameters for the engine and the electric machine are determined by assuming that these components are operated at their maximum efficiency points. Note that not all optimal operation points can be reached during the engine-only - and the electric-only driving modes (propulsion at low speeds and regenerative braking by electric machine) due to speed ratio limitation of the AMT. For example if only the engine mode is utilized over a whole drive cycle whereby optimal gear ratios (close to maximum engine efficiency points) are selected, then the total fuel use is approximately increased by 4% (205-kW engine, vehicle parameters of Table 4). To explain the relative small fuel consumption increase is referred to Figure 5. In this figure the normalized component input power with the maximum component power for the electric machine (58 kW) and engine (205 kW) as a function of the normalized output power with the total available output power (electric machine + engine) for different electric machine and engine angular speeds are shown. The calculated engine input powers with the AMT and the required input powers for the engine operated at the Optimal Operation Line (OOL) are also shown. The OOL connects the set Ω of optimal operation points,

$$\Omega = \left\{ (\omega_i^o, T_i^o) \middle| P_i = T_i \omega_i \land \omega_i,_{min} \leq \omega_i \leq \omega_i,_{max} \land T_i,_{min}(\omega_i) \leq T_i \leq T_i,_{max}(\omega_i) \right\},$$

fulfilling the condition of minimum input power,

$$\left(\omega_i^o, T_i^o\right) = \arg \min_{(\omega_i, T_i) \in \Omega} \phi_i(\omega_i, P_i), \text{ with } i \in \{e, em\}.$$  

(8)

The lines for the engine for different speeds show that for a given output power the influence of choosing a different operation point by changing the engine speed on the input power is relative small. If a quadratic function is fitted through the operation points, it can be seen that the difference between the OOL and the fit
function is relative small. The influence of the AMT is smaller for the electric machine due to its higher efficiency compared to the engine.

Furthermore, coupling of the electric machine and the engine to the same transmission input shaft requires a trade-off in selecting the optimal speed ratio which maximizes the combined engine and electric machine/battery efficiency during a hybrid driving mode (charging or motor-assisting during driving). The effect of this on the total fuel use has been left out of consideration in this paper. The reader is referred to (Lee et al., 2000), in which more static optimization to design the shift logic for a hybrid vehicle can be found, whereby also shift quality and driving comfort aspects have been taken into account. The clutch losses and engine auxiliaries have been taken into account, but are assumed to be supplied by the engine operated at the OOL. If the input power values at the OOL as a function of the output power values for the engine and the electric machine are plotted, then it appears (see Figure 5) that the fuel power $P_f$ (engine input power) is well approximated by a quadratic function with the engine power $P_e$ at the crank shaft.
\[ P_f = \phi_e(\omega^e, P_e) \approx c_0 + c_1 P_e + c_2 P_e^2, \quad \{c_2, c_1, c_0\} \subseteq \mathbb{R}_0^+. \]  
(9)

The battery output power \( P_b \) (electric machine input power) is well approximated by a linear function with the electric machine power \( P_{em} \),

\[ P_b = \phi_{em}(\omega^e_{em}, P_{em}) \approx \max\left( c^-_0 P_{em}, c^+_0 P_{em} \right) + \max\left( c^-_1 P_{em}, c^+_1 P_{em} \right), \]

\[ (c^-_1 < 0, c^+_1 > -1 \text{ and } \{c^-_0, c^+_0\} \subseteq \mathbb{R}_0^+). \]

The characteristic parameters are approximately constant \( c_j(\cdot) \approx c_j \), because the influence of changing the outlet power \( P_i \) given the optimal angular speed \( \omega^o_i \) on the characteristic parameters is nihil.

**Battery Efficiency:** The battery static losses are dependent on the output power \( P_s(t) \), the state-of-charge \( \xi(t) \) and the temperature \( T(t) \),

\[ P_b(t) = P_s(t) + P_{loss}(\xi(t), P_s(t), T(t)). \]

\[ (11) \]

The charging (coulomb) losses due to irreversible parasitic reactions in the battery has been taken into account by using an estimate of the average coulomb efficiency \( \eta_c \) (see data files ESS_NIMH6, ESSLI7_temp of ADVISOR). Self-discharge, or parasitic current is not separately considered, but these losses are assumed to be modeled by \( \eta_c \). The state-of-charge \( \xi \) is calculated as follows,

\[ \xi(t) = \frac{Q(t)}{Q_0} \cdot 100\%, \text{ with } Q(t) = \eta_c \int_0^t I(\tau) \, d\tau. \]

\[ (12) \]

The (dis-)charge currents are assumed to be low enough, due to the relative high battery pack voltage, so that the charge capacity \( Q_0 \) change (Peukert effects) is nihil. Furthermore, the Li-ion and Ni-MH battery pack are both assumed to be operated within a state-of-charge window of \( \xi_{min} = 40\% \leq \xi(t) \leq \xi_{max} = 80\% \).

Within this defined window of operation the open circuit voltage changes a little from 1.27 V/cell to 1.30 V/cell and 3.52 V/cell to 3.75 V/cell for the Ni-MH and the Li-ion battery respectively. Therefore, for simplicity, the state-of-charge and
the state-of-energy of the battery are assumed to be approximately similar and only 40% of the rated energy capacity is effectively available:

$$\triangle E_{s,max} \approx 0.40 \cdot Q_0 \cdot U_{b,nom} \cdot n_b \cdot n_p,$$

(13)

with the nominal battery voltage per module, the number of battery modules and the number of parallel strings of in series connected battery modules represented by the parameters $U_{b,nom}$, $n_b$ and $n_p$ respectively. Assuming the variation in state-of-charge $\xi(t)$ (due to the high $Q_0$), and temperature small ($T(t) = 25^\circ C$) the losses become only dependent on the (in-) output power and are approximated as quadratic with the stored power:

$$P_{loss}(P_s(t)) = \max \left( c_2^+ \frac{P_s(t)}{P_s(t)}, c_2^- \frac{P_s(t)}{P_s(t)} \right) P_s^2(t), \quad \{c_2^+, c_2^-\} \subseteq \mathbb{R}_0^+.$$

(14)

Therefore, the battery losses are assumed to be different during charging and discharging, and only increase with the stored or retrieved power. In order to increase the efficiency at relative high battery powers, in this paper, a battery topology consisting of of two parallel strings of in series connected battery modules were chosen ($n_p = 2$), such that the battery pack’s internal resistance is reduced.

**Ultra Capacitor Efficiency:** In contrary to batteries, the state-of-charge of the ultra capacitor strongly depends on the voltage across the capacitor, denoted as $U_{uc}(t)$ (Barrade and Rufer, 2004). The discharge voltage ratio $\sigma$ represents the ratio between the minimum and the maximum allowed capacitor voltage $\sigma = U_{uc,min}/U_{uc,max}$ and is used to calculate the state-of-charge,

$$\xi(t) = \frac{Q(t)}{Q_0} \cdot 100\% = \frac{C_{uc} \left( U_{uc}(t) - U_{uc,min} \right)}{C_{uc} \left( U_{uc,max} - U_{uc,min} \right)} \approx \frac{1}{1 - \sigma} \left( \frac{U_{uc}(t)}{U_{uc,max}} - \sigma \right),$$

(15)

with the capacitance $C_{uc}$ assumed to be approximately constant. However, it is found, that the minimum and maximum relative static losses, e.g., at 44 kW, vary a little between 1.1% and 6.4% for $\xi = 0$ and $\xi = 1$ respectively. Therefore, in this paper, for simplicity, the fit coefficients of Equation (14) found for the mean static power losses have been used.
3.2 Sizing of Conversion and Storage Components

The engine and electric machine static-efficiency maps are scaled linearly with maximum output power. Some results of the found fit coefficients are listed in Table 5. Looking at the results in Table 5 the internal efficiency of the electric machine during charging is approximately equal to discharging and corresponds with the slope of the linear curves, i.e., \( c_1^+ \approx 1/c_1^- = 91\% \). The slope of the various engine curves is approximately constant at 2.1 corresponding to a typical combustion efficiency (without static losses) of approximately 47\%. It can be seen, that a smaller engine or electric machine size results in smaller static losses, i.e. a smaller \( c_0 \).

### Table 5 Parameters for different engine and electric machine sizes

<table>
<thead>
<tr>
<th>Component</th>
<th>Parameter value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>125 150 170 205</td>
<td>kW</td>
</tr>
<tr>
<td>( c_2 )</td>
<td>1.10 0.92 0.87 0.72</td>
<td>( 10^{-6} \cdot \text{W}^{-1} )</td>
</tr>
<tr>
<td>( c_1 )</td>
<td>2.10 2.10 2.09 2.09</td>
<td>-</td>
</tr>
<tr>
<td>( c_0 )</td>
<td>7.84 9.41 11.29 13.61</td>
<td>kW</td>
</tr>
<tr>
<td>Electric machine</td>
<td>22 44 60 80</td>
<td>kW</td>
</tr>
<tr>
<td>( c_1^+ / c_1^- )</td>
<td>-0.92/-1.08</td>
<td>-</td>
</tr>
<tr>
<td>( c_0^+ / c_0^- )</td>
<td>-102/-301</td>
<td>W</td>
</tr>
</tbody>
</table>

The maximum absolute battery pack power is sized to meet the maximum output power specifications of the electric machine. However, due to losses of the electric machine the net (in-)output power as described by Equation (15) is reduced. Thereby, a minimum number of 40 and 13 battery modules for the Ni-MH and the Li-ion battery (as listed in Table 6) are needed to meet the minimum voltage requirement \( U_{pe,min} \) and maximum current \( I_{pe,max} \) allowed by the motor controller/electric machine, which are estimated to be 260 V and 170 A respectively. These values are kept constant and are based on the battery specification of the Eaton hybrid electric drive train with the 44-kW electric machine (Table 1).

For both battery technologies each module consists of 6 cells.

Since the ultra capacitor pack is not limited by power -, but by energy constraints, the number of required caps was iteratively optimized by performing different simulation runs, until the available energy content of the capacitor pack is sufficiently.
The maximum allowable systems bus voltage is assumed to be 600 V. Due to this limitation sizing of the Ni-MH battery pack or the ultra capacitor pack larger than 60 kW by selecting more than 78 modules or 222 caps respectively is not possible. Table 6 shows, that overall sized Li-ion battery pack has a much higher energy - (Wh/kg) and power density (kW/kg) specification compared to the Ni-MH battery pack. The maximum losses at 44 kW during (dis-)charging correspond approximately with 9.8% and 12.3%, 13.6% and 16.3%, and 2.7% and 2.3% of the storage power for the Li-ion, Ni-MH and UC respectively. Note that for other component sizes, the characteristic parameters are determined by interpolating between the values as given in Table 5 and 6.

**Table 6** Parameters for different sized energy storage technologies

<table>
<thead>
<tr>
<th>Type</th>
<th>EM Size (kW)</th>
<th>22</th>
<th>44</th>
<th>60</th>
<th>80</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ni-MH</td>
<td>Nominal voltage</td>
<td>310</td>
<td>450</td>
<td>580</td>
<td>-</td>
<td>V</td>
</tr>
<tr>
<td></td>
<td>No. modules†</td>
<td>40·2</td>
<td>58·2</td>
<td>78·2</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Mass</td>
<td>40</td>
<td>58</td>
<td>79</td>
<td>-</td>
<td>kg</td>
</tr>
<tr>
<td></td>
<td>$c^+/c^-$</td>
<td>4.95/4.78</td>
<td>3.71/3.10</td>
<td>2.76/2.31</td>
<td>-</td>
<td>$10^{-6}\cdot W^{-1}$</td>
</tr>
<tr>
<td></td>
<td>Power density</td>
<td>0.55</td>
<td>0.76</td>
<td>0.76</td>
<td>-</td>
<td>kW/kg</td>
</tr>
<tr>
<td></td>
<td>Energy density</td>
<td>46.5</td>
<td>46.6</td>
<td>46.6</td>
<td>-</td>
<td>Wh/kg</td>
</tr>
<tr>
<td>Li-ion</td>
<td>Nominal voltage</td>
<td>283</td>
<td>293</td>
<td>316</td>
<td>338</td>
<td>V</td>
</tr>
<tr>
<td></td>
<td>No. modules†</td>
<td>13·2</td>
<td>14·2</td>
<td>15·2</td>
<td>16·2</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Mass</td>
<td>30</td>
<td>32</td>
<td>34</td>
<td>36</td>
<td>kg</td>
</tr>
<tr>
<td></td>
<td>$c^+/c^-$</td>
<td>2.64/2.31</td>
<td>2.80/2.24</td>
<td>2.43/1.81</td>
<td>2.28/1.63</td>
<td>$10^{-6}\cdot W^{-1}$</td>
</tr>
<tr>
<td></td>
<td>Power density</td>
<td>0.73</td>
<td>1.38</td>
<td>1.76</td>
<td>2.22</td>
<td>kW/kg</td>
</tr>
<tr>
<td></td>
<td>Energy density</td>
<td>56.6</td>
<td>55.1</td>
<td>55.8</td>
<td>56.3</td>
<td>Wh/kg</td>
</tr>
<tr>
<td>UC</td>
<td>Nominal voltage</td>
<td>150</td>
<td>180</td>
<td>200</td>
<td>-</td>
<td>V</td>
</tr>
<tr>
<td></td>
<td>Voltage swing, $\sigma$</td>
<td>0.58</td>
<td>0.45</td>
<td>0.43</td>
<td>-</td>
<td>V/V</td>
</tr>
<tr>
<td></td>
<td>No. caps</td>
<td>167</td>
<td>215</td>
<td>222</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Mass</td>
<td>78</td>
<td>101</td>
<td>104</td>
<td>-</td>
<td>kg</td>
</tr>
<tr>
<td></td>
<td>$c^+/c^-$</td>
<td>0.52/0.59</td>
<td>0.53/0.61</td>
<td>0.52/0.63</td>
<td>-</td>
<td>$10^{-6}\cdot W^{-1}$</td>
</tr>
<tr>
<td></td>
<td>Power density</td>
<td>1.9</td>
<td>1.8</td>
<td>1.8</td>
<td>-</td>
<td>kW/kg</td>
</tr>
<tr>
<td></td>
<td>Energy density</td>
<td>3.7</td>
<td>4.5</td>
<td>4.5</td>
<td>-</td>
<td>Wh/kg</td>
</tr>
</tbody>
</table>

† two parallel strings of in series connected battery modules

### 3.3 Control Design Model - RB-ECMS

The control strategy used is based on the combination of Rule-Based - and Equivalent Consumption Minimization Strategies (RB-ECMS), whereby the constraints which determine when to switch between the different driving modes are optimized (Hofman *et al.*, 2007). The RB-ECMS determines if it is beneficial to
propel the vehicle only by the electric machine (Motor only: M mode), only by the
engine (Engine only: E mode), or to assist the engine with the electric machine in
motoring mode (Motor-Assist: MA mode), or to charge using the electric machine
in generative mode (CHarging: CH mode) during driving. Recuperation of brake
energy (Brake Energy Recovery: BER mode) is very beneficial and is always per-
formed. During the BER, M mode and vehicle standstill the engine is off and has
no drag or idle losses.

**Optimization Problem:** The optimization problem is finding the optimal control
power flow $P_s(t)$ given a certain power demand at the wheels $P_v(t)$, while the
cumulative fuel consumption denoted as the variable $\Phi_f$ over a certain drive cycle
with time length $t_f$ is minimized subjected to several constraints, i.e.,

$$
\Phi_f = \min_{P_s(t)} \int_0^{t_f} \dot{m}_f(E_s(t), P_s(t), t) \, dt,
$$

subject to $\vec{h} = 0, \vec{g} \leq 0$.  

(16)

where $\dot{m}_f(t)$ is the fuel mass flow in g/s. The state is equal to the stored energy
$E_s(t)$ in the reversal energy buffer in J, and the control input is equal to the power
flow $P_s(t)$ in W. The energy level in the battery (or ultra capacitor) is a simple
integration of the power and is calculated as follows,

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

(17)

The main constraints on the battery (or ultra capacitor) are energy balance conser-
vation of $E_s(t)$ over the drive cycle, constraints on the power $P_s(t)$, and the energy
$E_s(t)$:

$$
\begin{align*}
 h_1 &:= E_s(t_f) - E_s(0) = 0, \\
g_{1,2} &:= P_{s,min} \leq P_s(t) \leq P_{s,max}, \\
g_{3,4} &:= E_{s,min} \leq E_s(t) \leq E_{s,max}.
\end{align*}
$$

(18)

The optimal control power flow from and to the secondary source during the M and
the BER mode respectively is assumed to be,

$$P_{s,I}(t) = \max \left\{ \frac{P_v(t)}{\eta_b(P_v(t)) \eta_{em}(P_v(t)) \eta_{AMT}} \right\},$$

whereby the power set-point is limited between the following constraints,

$$P_{s,min} \leq P^0_M \leq 0 \leq P^0_{s,I}(t) \leq P_{s,max}.$$  \(\text{(20)}\)

The minus sign in Equation (19) indicates discharging during propulsion and charging during braking. The battery - \(\eta_b\) and electric machine efficiency \(\eta_{em}\) are assumed to be implicit time dependent by the required (in-)output power demand \(P_v(t)\) during the BER and M mode respectively. Whereas the AMT - \(\eta_{AMT}\), final drive - , and differential efficiency \(\eta_{fd}\) are assumed to be constant, but dependent on direction of power flow \(P_s(t)\). Powers larger than the maximum charging power \(P_{s,max}\) are assumed to be dissipated by the wheel-brake discs. During the motor only mode (M) the vehicle is propelled up to the maximum discharging motoring power \(P^0_M\). If only the M and/or the BER mode are utilized, then the energy difference \(\Delta E_{s,I}(t_f)\) at the end of the drive cycle becomes,

$$\Delta E_{s,I}(t_f) = \int_0^{t_f} P^0_{s,I}(t) \, dt, \quad \Delta E_{s,I}(t_f) \in \mathbb{R}$$

In order to fulfill the equality constraint \(h_1\) of Equation (18) this energy has to be counterbalanced with the energy difference \(\Delta E_{s,II}(t_f)\) at the end of the cycle during the MA and the CH mode as is shown in Figure 6, hence

$$-\Delta E_{s,I}(t_f) = \Delta E_{s,II}(t_f).$$

The optimal power flow during the CH and the MA mode is calculated using the equivalent fuel mass flow \(\dot{m}_{f,eq}\). The equivalent fuel mass flow uses an electric-energy-to-fuel-conversion-weight-factor, or equivalence (weight) factor \(\lambda_0\). The \(\lambda_0\) is used to assign future fuel savings and costs to the actual use of electric power \(P_s\). Moreover, a well determined \(\lambda_0\) assures that discrepancy between the buffer
The optimal power flow during the CH/MA mode is calculated by minimization of the equivalent fuel mass flow $\dot{m}_{f,\text{eq}}$ at the current time $t$, that is

$$ P_{s,\text{II}}(t) = \arg\min_{P_{s,\text{II}}(t)} (\dot{m}_{f,\text{eq}}(t)|\lambda_0) = \arg\min_{P_{s,\text{II}}(t)} (\dot{m}_f(t) - \lambda_0 P_{s,\text{II}}(t)), \quad (23) $$

whereby the power set-point is limited between the following constraints,

$$ P_{s,\text{min}} \leq P_{s,\text{II}}(t) \leq P_{s,\text{max}}. \quad (24) $$

Then $\Delta E_{s,\text{II}}(t_f)$ is discharged (charged) at vehicle power demands where the fuel savings (costs), i.e., $\Delta \dot{m}_f$ are maximum (minimum). Summarized, the optimal power set-point for the secondary power source as discussed in the previous two sections during the BER/M and the CH/MA mode becomes respectively:

$$ P_{s}(t) = \begin{cases} 
  P_{s,I}(t) \text{ (cf. Equation (19)), if } & \frac{-P_{s}(t)}{m_t(P_s(t)) \eta_{\text{em}}(P_v(t)) \eta_{\text{AMT}} \eta_{f_d}} \geq P_{s,\text{II}}(t) \\
  P_{s,\text{II}}(t) \text{ (cf. Equation (23)), elsewhere.} & 
\end{cases} \quad (25) $$

Finally, the EMS optimization scheme is shown in Figure 7. Starting with arbitrary values for $P_M$ (limited by power constraints) and $\lambda$, the values for $P_M$ and $\lambda$ are iteratively (loops 2 and 1 in Figure 7 respectively) updated until the cumulative fuel consumption $\Phi_f$ is minimized and the energy balance conservation over the drive cycle is fulfilled.
3.4 Vehicle Dynamics Model

In this section the equations are derived in order to calculate the acceleration time \( t_a \) and the maximum gradeability \( \theta_{\text{max}} \). The schematic layout of the hybrid drive train structure and the torques acting on the driven rear wheels are shown in Figure 8.

*Figure 8*  Vehicle dynamics model

*Acceleration Time 0-100 km/h (0-62 mph)*: The dynamic torque balance at the
propulsion shaft of the vehicle wheels gives for the vehicular acceleration (see Figure 8): 

\[ \alpha_v(t) = \dot{\omega}_v(t) \cdot r_w = \frac{1}{J_v} \cdot (T_v(t) - T_{rl}(t)) \cdot r_w, \]  

(26)

with the total vehicle wheel torque \( T_v(t) \) consisting of the sum of the engine - \( T_e(t) \) and electric machine torque \( T_{em}(t) \) and an additional inertia torque term due to engine speed change during shifting,

\[ T_v(t) = \left( \frac{T_e(t) + T_{em}(t)}{r_{AMT}(t) \cdot r_d} - \frac{J_e + J_{em}}{r_{AMT}(t) \cdot r_d} \cdot \dot{\omega}_e(t) \right) \eta_{AMT} \eta_{fd}. \]  

(27)

Since the vehicle wheel speed is a function of the engine speed and the gear ratio of the AMT

\[ \omega_v(t) = \omega_e(t) \cdot r_{AMT}(t) \cdot r_d, \quad r_{AMT}(t) \in \mathbb{R}, \]  

(28)

the time derivative of the engine speed as is used in Equation (27) can be written as

\[ \dot{\omega}_e(t) = \frac{\dot{\omega}_v(t)}{r_{AMT}(t) \cdot r_d} - \omega_e(t) \cdot \frac{\dot{r}_{AMT}(t)}{r_{AMT}(t)}. \]  

(29)

The road load torque \( T_{rl}(t) \) due to air drag - , roll - and road slope \( \theta(t) \) resistance holds,

\[ T_{rl}(t) = \frac{1}{2} \rho_c d A_f \omega_v(t)^2 r_w^3 + c_r m_v \ g \cos(\theta(t)) \ r_w + m_v \ g \sin(\theta(t)) \ r_w. \]  

(30)

The engine, clutch, and starter flywheel - and electric machine inertia are represented by the variables \( J_e \) and \( J_{em} \) respectively. The vehicle inertia \( J_v \) consists of the vehicle mass \( m_v \), the inertia of the rotating parts \( J_w \) including the wheels, and the final drive:

\[ J_v = J_w + m_v \ r_w^2 \]  

(31)

Initially, the vehicle acceleration is calculated under the assumption that no wheel slip occurs. Then, using this acceleration value the torque required to initiate slip is calculated:

\[ T_{slip}(t) = \mu_v \ m_v \ g \left( \frac{L_r}{L} + \frac{a_v(t) \ H}{L} \right) \ r_w \]  

(32)
for rear wheel driven vehicles with the traction coefficient $\mu_r$. The wheel base length and height of the center of gravity are represented by the parameters $L$ and $H$. Whereby $H$ is assumed to increase linearly with increase of $m_v$. The wheel torque $T_v(t)$ during acceleration is compared with the wheel slip torque $T_{\text{slip}}(t)$. If $T_v(t) > T_{\text{slip}}(t)$ then slip occurs and the tractive torque becomes equal to the slip torque, i.e., $T_v(t) = T_{\text{slip}}(t)$. Using Equation (32) substituted into Equation (26) the vehicle acceleration under the slip condition is calculated. The gear ratio change rate $\dot{r}_{\text{AMT}}(t)$ is assumed to be sufficiently limited, such that a positive vehicle acceleration is guaranteed. A gear change time delay of 0.9 s at zero engine torque to the shifting sequence is assumed. During an upshift the net acceleration falls below zero during this shift period, due to road load forces that are acting on the vehicle. The vehicle speed is used to trigger upshifts. Therefore, in order to prevent downshifts during shift periods, upshifts are forced once the delay period is over. During acceleration the engine and the electric machine are assumed to be operated (as much as possible) at their wide-open throttle - and maximum torque curve respectively. Figure 9 shows an example result of the engine speed and vehicle speed over time during maximum acceleration. Note that $t_a$ is limited by the minimum state-of-charge of the battery or the ultra capacitor ($\xi(t_a) \geq \xi_{\text{min}}$).

Gradeability at 89 km/h (55 mph): The feasible road slopes $\theta_0$ at a desired vehicle speed $v_v$, which is a function of the speed ratio $r_{\text{AMT}}$, are elements of the set of the stationary vehicle wheel torques $T_v$ which are in balance with the road load torques $T_{rl}$. Whereby $T_v$ is a function of the speed ratio $r_{\text{AMT}}$ and the vehicle speed $v_v$ and $T_{rl}$ is a function of the road slope $\theta$ and the vehicle speed $v_v$: 

\[
\theta_0(r_{\text{AMT}}) \subseteq \left\{ T_v(r_{\text{AMT}}, v_v) - T_{rl}(\theta, v_v) = 0 \mid v_v = 89 \text{ km/h} \land r_{\text{AMT}} \in \mathcal{R} \land 0^\circ \leq \theta \leq 45^\circ \right\}, \tag{33}
\]

The target is to find the optimal gear ratio $r_{\text{AMT}}^\theta$ which maximizes the feasible road slopes:

\[
r_{\text{AMT}}^\theta = \arg \max_{r_{\text{AMT}} \in \mathcal{R}} \left( \theta_0(r_{\text{AMT}}) \right) \land \theta_{\text{max}} = \theta_0(r_{\text{AMT}}^\theta), \tag{34}
\]
An example result of the engine speed and the vehicle speed over time whereby no wheel slip occurs. Figure 10 shows an example result of the wheel torque as a function of the vehicle speed for different speed ratios.

Figure 9  An example result of the engine speed and the vehicle speed over time

Gradability at 89 km/h

Figure 10  An example result of the wheel torque as a function of the vehicle speed for different speed ratios
3.5 Dual-Energy Storage Design Model

In Figure 11 the power flow in a dual-energy storage system is schematically shown. In this paper the optimal dual-storage size is determined by minimizing the overall energy storage mass, denoted with the variable $\Phi_{es}$, which is determined by the total number of battery modules $n_b$ and ultra capacitor cells $n_{uc}$:

$$\Phi_{es} = K_b \cdot n_b(f_c) + K_{uc} \cdot n_{uc}(f_c).$$

(35)

The parameters $K_b$ and $K_{uc}$ represent the conversion factors (kg/cell) from number of cells to storage mass (see Table 3). In order to determine the total number of required cells, power separation of the original optimized power signal $P_{os}(t)$ as in Equation (25) is performed by using a digital filter $H_f$ (Butterworth low-pass digital filter), given that the desired storage (battery) and peak power (ultra capacitor) characteristics are achieved. The design variables $(n_b, n_{uc})$ are a function of the cutoff frequency $f_c$. Moreover, at each frequency $f$, the frequency content of the battery usage $H_b(f)$ and capacitor usage $H_{uc}(f)$ is weighted:

$$H_b(f) + H_{uc}(f) = 1,$$

(36)

which is directly analogous to the Bode frequency criteria. The reader is referred to (Ozatay et al., 2004) and (Schroeck and Messner, 1999) where frequency-domain control structures are discussed to achieve frequency-based separation of the battery/ultra capacitor usage. Note that $P_{os}(t)$ used for the design of the dual-storage...
system is based on the EMS calculated with a single-storage system with a different storage efficiency. This is done, because influence of this effect on the overall desired size is assumed to be nihil.

\[ P_{s,b}(t) := H_f(f_c) \cdot P_s^o(t) \]  

\[ E_{s,uc}(t) := \int_0^t (1 - H_f(f_c)) \cdot P_s^o(t) \, dt \]  

Generally, an ultra capacitor is constrained by energy - and not by power limitations, and a battery is constrained by power - and not by energy limitations:

\[ E_{s,uc,min}(n_{uc}) \leq E_{s,uc}(t) \leq E_{s,uc,max}(n_{uc}) \]  

\[ P_{s,b,min}(n_b) \leq P_{s,b}(t) \leq P_{s,b,max}(n_b) \]  

The optimal number of battery modules and ultra capacitor cells, denoted as \( n_b^o \) and \( n_{uc}^o \) respectively, given \( P_s^o(t) \) are calculated from the following minimization,

\[ (n_b^o, n_{uc}^o) = \arg \min_{(n_b, n_{uc}) \in \mathcal{N}} \left( \Phi_{es}(n_b, n_{uc}) \mid P_s^o(t) \right), \]  

where the set \( \mathcal{N} \) covers the feasible solutions that satisfy the constraints in Equation (37)-Equation (40):

\[ \mathcal{N} = \left\{ (n_b, n_{uc}) \mid P_{s,b}(t) - H_f(f_c) \cdot P_s^o(t) = 0 \wedge E_{s,uc}(t) - \int_0^t (1 - H_f(f_c)) \cdot P_s^o(t) \, dt = 0 \wedge P_{s,b,min}(n_b) \leq P_{s,b}(t) \leq P_{s,b,max}(n_b) \wedge E_{s,uc,min}(n_{uc}) \leq E_{s,uc}(t) \leq E_{s,uc,max}(n_{uc}) \right\}. \]  

Next the battery and ultra capacitor design models are derived for determining the available battery power and the capacitor’s storage energy.

**Battery Design Model:** In Figure 12 the equivalent circuit of a battery is shown. The
battery's open-circuit voltage, the series internal resistance causing a voltage drop, and the terminal voltage are represented by the variable $U_{oc}$, $R_b$ and $U_b$ respectively. Kirchhoff’s law for the equivalent circuits yields the following equation,

$$U_b = U_{oc} - R_b I,$$

with the current defined as $I = P_b / V_b$. The battery pack consists of two parallel strings of in series connected battery modules reducing the overall internal resistance. For (dis)-charging the maximum current is limited by the maximum allowed motor controller current $I_{pe,max} = 170 \text{ A}$,

$$|I| \leq I_{pe,max}.$$  \hspace{1cm} (44)

Although, the open circuit voltage depends $U_{oc}$ on the state-of-charge, an average value for $U_{oc}$ is used for determining the number of battery modules fulfilling the maximum power requirements. The maximum (dis-)charging output power as a function the voltage and the number of battery modules $n_b$ is calculated as,

$$P_{b,i}(n_b) = -2 \frac{U_{b,i}^2}{R_b} + 2 \frac{U_{oc}}{n_b} U_{b,i} n_b, \quad i \in \{min, max\},$$  \hspace{1cm} (45)

with the maximum (dis-)charge input power as a function of number of battery modules,

$$P_{s,b,i}(n_b) = P_{b,i}(n_b) - \left( \frac{R_b}{n_b U_{b,i}^2} \right) P_{b,i}^2(n_b).$$  \hspace{1cm} (46)

The battery voltage corresponding with the minimum discharging output power by setting the derivative of $P_{b,min}$ with respect to $U_b$ to zero becomes,

$$\frac{\partial P_{s,b,min}}{\partial U_{b,min}} = 0 \Rightarrow U_{b,min} = U_{oc}/2.$$  \hspace{1cm} (47)
However, the minimum discharging output power $P_{b,\text{min}}$ is limited by three parameters, which are all related to the available battery voltage at the terminal: the minimum input voltage of the motor controller $U_{pe,\text{min}} = 260$ V, the minimum battery’s voltage $U_{b,\text{min}}$, and the open circuit voltage divided by two $U_{oc}/2$ as calculated with Equation (47),

$$U_{b,\text{min}} := \max \left( U_{pe,\text{min}}/n_b, U_{b,\text{min}}, U_{oc}/2 \right).$$

Finally, the maximum charging input power $P_{b,\text{max}}$ and output power $P_{s,b,\text{max}}$ as a function of $n_b$ is, besides the current limitation (cf. Equation (44)), mainly limited by the maximum battery’s voltage $U_{b,\text{max}}$.

Ultra Capacitor Design Model: The maximum available usable storage energy, denoted as $\Delta E_{s,uc,a}$, that a capacitor can provide is defined by the equation:

$$\Delta E_{s,uc,a}(n_{uc}) = \frac{1}{2} C_{uc} U_{uc,\text{max}}^2 n_{uc} \left(1 - \sigma^2\right),$$

with the discharge voltage ratio limited to the battery minimum voltage to nominal voltage ratio,

$$\sigma \leq \frac{U_{b,\text{min}}}{U_{b,\text{nom}}}.$$ (50)

If an independent power processor interfaces the capacitor to the terminal voltage, then its voltage swing $\sigma$ is only limited by the minimum input voltage of its power converter, which is typically 0.33 of the operating voltage (Miller, 2004). Knowing the needed maximum usable energy for a given $f_c$,

$$\Delta E_{s,uc,n}(f_c) = \max \left( E_{s,uc}(t) \right) - \min \left( E_{s,uc}(t) \right),$$

the number of ultra capacitor cells $n_{uc}$, which are coupled in series, is easy to identify after substitution of Equation (51) into Equation (49),

$$\Leftrightarrow n_{uc} = \frac{2 \Delta E_{s,uc,n}(f_c)}{C_{uc} U_{s,uc,\text{max}}^2 (1 - \sigma^2)}.$$ (52)
4 Simulation Results

4.1 Fuel Economy, Acceleration time and Gradeability Results

The base line FedEx truck is equipped with a Cummins 175HP (131-kW) 6BT5.9 and an Allison AT542 NFE 4-speed automatic transmission (Werts and Steffen, 2001). Therefore, the 205-kW base engine is down scaled to produce 131 kW, which is used in the simulations for the Base Line (BL) vehicle. In Figure 13 the relative fuel savings (in Table 7 the reference values are listed) of the Hybrid Electric Vehicle (HEV) as a function of the hybridization factor $HF_{dt,125kW}$ for different vehicle masses are shown (kept constant: Li-ion battery and 125-kW engine). The following conclusions can be drawn for a constant engine size:

1. If the vehicle mass increases, then the optimal $HF_{dt}$ (or in this case the optimal electric machine size) increases due to increase of regenerative brake energy potential.

2. With a larger electric machine size (i.e. a higher $HF_{dt}$) effectively more brake energy can be recuperated. However, the (static) losses of the electric machine increase progressively thereby reducing again the fuel saving potential. The curves show a certain maximum fuel saving potential.

The idle fuel mass flow strongly influences the relative fuel saving (idle stop during BER, M mode and stand still) and is linearly scaled with engine size. However, the optimal $HF_{dt,125kW}$ values for a truck mass of 7.3 ton and 4 ton are 23.3% and 15.0% respectively. The optimal $HF_{dt}$ values are close to the hybridization factors of the FedEx ($HF_{dt,125kW} = 26\%, m_v = 7.3$ ton), Nissan Condorr ($HF_{dt,152kW} = 27\%, m_v = 7.8$ ton) and Hino 4T Ranger truck ($HF_{dt,132kW} = 15\%, m_v = 3.6$ ton).

In Figure 14 the optimal $HF_{dt,125kW}$ for a 7.3 tons truck as a function of the vehicle mass is shown. In the same figure the hybridization factor values are shown for the case that the minimum braking power determines the electric machine size:
Figure 13 Fuel saving as a function of HF_{dt,125kW} for different vehicle masses (kept constant: Li-ion and 125-kW engine)

\[ HF_{dt,125kW}(m_v) = \left( \frac{P_{em,min}}{P_{em,min} + 125 \text{ kW}} \right) m_v \wedge P_{em,min} = \min \left( P_e(t) \eta_{AMT} \eta_{fd}^+ \right) \] (53)

Figure 14 shows that the discrepancy between the hybridization factor HF_{dt,125kW} determined by the maximum generative power and determined by the maximum fuel saving increases with vehicle mass. However, the maximum discrepancy between the fuel economy for the different vehicle masses for both cases is found to be negligible small. Nevertheless, in order to reduce the require power specifications for the electric machine the following conclusion can be made:

3. For large vehicle masses (see, Figure 14 for \( m_v \geq 6 \text{ ton} \)) the optimal degree of hybridization should be determined based on maximizing the fuel savings over a whole drive cycle and can not be determined based on the maximum braking power.
Based on recuperation of max. braking power
Based on maximum fuel saving

Figure 14 Optimal HF\textsubscript{dt,125kW} as a function of the vehicle mass \(m_v\) (kept constant: Li-ion and 125-kW engine)

Table 7 Fuel economy results for different vehicle masses (kept constant: Li-ion and 125-kW engine)

<table>
<thead>
<tr>
<th>Mass, (m_v)</th>
<th>Fuel economy</th>
<th>(10^3) kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>BL (131-kW engine)</td>
<td>11.9 14.5 16.0 17.0 19.5 21.9</td>
<td>l/100km</td>
</tr>
<tr>
<td>HEV (125-kW engine)</td>
<td>9.3 11.4 12.7 13.5 15.5 17.4</td>
<td>l/100km</td>
</tr>
<tr>
<td>Optimal HF\textsubscript{dt,125kW}</td>
<td>15.0 21.9 23.3 26.0 29.4 30.6</td>
<td>%</td>
</tr>
<tr>
<td>(P_{em,max} / P_{M})</td>
<td>22/14 35/18 38/22 44/24 55/29 70/34</td>
<td>kW/kW</td>
</tr>
<tr>
<td>Fuel saving = (100% \cdot (1 - 2)/1)</td>
<td>21.8 21.1 20.9 20.9 20.6 20.6</td>
<td>%</td>
</tr>
</tbody>
</table>

In Figure 15 the relative fuel savings (in Table 8 and 9 the reference values are listed) of the HEV as a function of HF\textsubscript{dt,P\_max} for different engine sizes and two different vehicle masses are shown (kept constant: Li-ion). The following conclusions can be drawn for a constant vehicle mass or load:

4. The relative fuel saving as a function of HF\textsubscript{dt} for a smaller engine size is smaller than for a larger engine size due to increase of the engine static losses with increase of the engine size.

5. The maximum relative fuel saving at a certain engine size for a smaller vehicle mass is higher than at the same engine size (\textit{i.e.} equal static losses) for a larger
vehicle mass. The maximum relative fuel saving is decreased due to increase of the average vehicle power demands with vehicle mass.

In addition to conclusion 5 it can be concluded that:

6. The absolute maximum fuel saving at a certain engine size for a smaller vehicle mass is lower than at the same engine for a larger vehicle mass. This can be seen by comparing the results of Table 8 with Table 9.

![Figure 15](image-url)

**Figure 15** Fuel saving as a function of $HF_{dt,P_{e,\text{max}}}$ for different engine sizes and two vehicle masses (kept constant: Li-ion)

### Table 8 Fuel economy results for different engine sizes (kept constant: Li-ion and $m_v = 7.3$ ton)

<table>
<thead>
<tr>
<th>Engine size, $P_{e,\text{max}}$</th>
<th>125</th>
<th>150</th>
<th>170</th>
<th>205</th>
<th>kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>BL</td>
<td>16.0</td>
<td>16.5</td>
<td>16.8</td>
<td>17.4</td>
<td>1/100km</td>
</tr>
<tr>
<td>HEV</td>
<td>12.7</td>
<td>12.9</td>
<td>13.1</td>
<td>13.3</td>
<td>1/100km</td>
</tr>
<tr>
<td>Optimal $HF_{dt,P_{e,\text{max}}}$</td>
<td>23.3</td>
<td>20.2</td>
<td>18.3</td>
<td>15.6</td>
<td>%</td>
</tr>
<tr>
<td>$P_{e,m,\text{max}} / - P_{O}$</td>
<td>38/22</td>
<td>38/22</td>
<td>38/22</td>
<td>38/22</td>
<td>kW/kW</td>
</tr>
<tr>
<td>Fuel saving = 100% · (1 - 2)/1</td>
<td>20.9</td>
<td>21.6</td>
<td>21.9</td>
<td>23.4</td>
<td>%</td>
</tr>
</tbody>
</table>

In Table 10 the vehicle improvements regarding the fuel economy and drivability for different storage systems are shown (kept constant: $HF_{dt,125kW} = 23.3\%$, $m_v = 7.3$ ton and 125-kW engine). Table 10 shows that the fuel economy and the vehicle performances are improved compared to the BL vehicle. Note that there is only
Table 9  Fuel economy results for different engine sizes (kept constant: Li-ion and \( m_v = 12 \) ton)

<table>
<thead>
<tr>
<th>Engine size, ( P_{e,max} )</th>
<th>Fuel economy</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>150</td>
</tr>
<tr>
<td>BL</td>
<td>21.7</td>
</tr>
<tr>
<td>HEV</td>
<td>17.4</td>
</tr>
</tbody>
</table>

Optimal HF \( \frac{P_{e,max}}{P_{o,M}} \):

<table>
<thead>
<tr>
<th>( P_{e,max} )</th>
<th>( P_{o,M} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>70/34</td>
<td>70/34</td>
</tr>
<tr>
<td>70/35</td>
<td>70/35</td>
</tr>
</tbody>
</table>

Fuel saving = 100\% \cdot (1 - 2) / 1

<table>
<thead>
<tr>
<th>( P_{e,max} )</th>
<th>19.8</th>
<th>20.6</th>
<th>20.7</th>
<th>21.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>%</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A small difference in fuel economy for the HEV equipped with a Ni-MH battery compared to the HEV with a Li-ion battery and UC. For the HEV equipped with a Li-ion battery pack or an UC pack, the fuel economy values are approximately similar.

Table 10  Fuel economy and drivability improvements for Ni-MH, Li-ion and UC (kept constant: HF\( d_t \) = 23.3\%, \( m_v = 7.3 \) ton and 125-kW engine)

<table>
<thead>
<tr>
<th>Engine size (kW)</th>
<th>BL</th>
<th>HEV</th>
<th>Change (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>131</td>
<td>-</td>
<td>125</td>
<td>-</td>
</tr>
<tr>
<td>Electric machine size (kW)</td>
<td>-</td>
<td>38</td>
<td>-</td>
</tr>
<tr>
<td>Fuel economy (l/100km)</td>
<td>16.1</td>
<td>12.7/12.9†</td>
<td>+21%/+20%</td>
</tr>
<tr>
<td>Gradeability (( \theta_{max} )) at 89 km/h (%)</td>
<td>4.5</td>
<td>6.7</td>
<td>+49%</td>
</tr>
<tr>
<td>Acceleration time (( t_a )) 0-100 km/h (s)</td>
<td>49.0</td>
<td>36</td>
<td>+27%</td>
</tr>
<tr>
<td>Maximum vehicle speed (( v_{max} )) (km/h)</td>
<td>116</td>
<td>129</td>
<td>+11%</td>
</tr>
</tbody>
</table>

\(-/+ = (de-)increased, \^ = Li-ion, UC, \^ = Ni-MH\)

The time to sustain the maximum achievable vehicle speed \( v_{max} \), after an acceleration period, denoted as \( t_{v,max} \), is limited by the minimum state-of-charge or available energy storage (see also Equation (13)) of the battery and ultra capacitor pack. During the acceleration test the initial state-of-charge for both the battery and ultra capacitor packs are 80\% and 100\% assumed respectively. It is assumed that the battery and ultra capacitor can be discharged to \( \xi_{min} = 40\% \) and \( \xi_{min} = 0\% \) respectively. The results for the different storage systems until the minimum state-of-energy constraint \( \xi (t_a + t_{v,max}) = \xi_{min} \) is reached are listed in Table 11.

Although, the type of electrical storage system has in this case a relative small influence on the overall vehicle's performance (total power is the same), \( t_{v,max} \) is limited and significantly different for each type of storage system, due to the differ-
Table 11  Time to sustain the maximum speed after acceleration $t_{v,\text{max}}$ for different single-storage technologies (kept constant: $HF_{\text{at,125kW}} = 23.3\%$, $m_v = 7.3$ ton and 125-kW engine)

<table>
<thead>
<tr>
<th>Storage type</th>
<th>No. modules$^\dagger$/No. caps</th>
<th>$t_{v,\text{max}}$(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>UC</td>
<td>202</td>
<td>8.3</td>
</tr>
<tr>
<td>Li-ion</td>
<td>13 · 2</td>
<td>12.2</td>
</tr>
<tr>
<td>Ni-MH</td>
<td>57 · 2</td>
<td>21.7</td>
</tr>
</tbody>
</table>

$^\dagger$ two parallel strings of in series connected battery modules

...ent energy contents. Summarized, the Li-ion and UC have the highest efficiency and therefore the highest fuel economy. In addition, the Ni-MH battery has the largest energy content and therefore the longest duration time $t_{v,\text{max}}$ until the minimum state-of-charge $\xi_{\text{min}}$ is reached.

4.2 Dual-energy Storage Systems

The UC cell (see also Table 3) has an equivalent series resistance value of 0.40 mΩ and a capacitance $C_{uc}$ of 2600 F (Maxwell, 2007). In Table 12 the average system voltage and the storage mass of the optimized (single-)dual-energy buffers are listed. The battery and ultra capacitor are assumed to be direct parallel coupled. The required battery pack size and output power specifications for both battery technologies is significantly reduced. However, the mass of the dual-storage system (without boost converter) due to the relative high specified minimum motor controller voltage value of 260 V, is higher than the single-storage system mass.

A boost converter can solve this problem by increasing the storage output voltage and thereby reducing the number of required battery/capacitor cells (Muta et al., 2004). For example the Toyota Prius contains a motor which utilizes voltages of approximately 500 V. Without a boost converter 417 Ni-MH battery cells would be needed instead of the applied 168 cells. The converter boosts the voltage from 202 V up to 500 V. Assume that a boost converter is connected between the electric machine and the direct parallel coupled battery and ultra capacitor modules, and that the voltages are boosted from approximately 130 V up to 260 V. The results are also shown in Table 12. Only the total mass of the dual-storage system
consisting of a Ni-MH/UC with boost converter is significantly reduced with 23% compared the initial system mass. Although, adding an UC to a battery reduces the battery power demands and therefore the battery wear significantly, a boost converter is a critical element in reducing the dual-storage system mass. In general, especially for battery technologies with a relative low power density (W/kg). In this case the Ni-MH battery has a lower power density than the Li-ion battery.

**Table 12** Optimal dual-storage size (kg) (kept constant: $HF_{dt,125\text{kw}} = 23.3\%$, $m_v = 7.3$ ton and 125-kW engine)

<table>
<thead>
<tr>
<th>Storage type</th>
<th>Nominal voltage (V)</th>
<th>Voltage swing, $\sigma$ (-)</th>
<th>Battery mass (kg)</th>
<th>Capacitor mass (kg)</th>
<th>Total mass (kg)</th>
<th>Relative mass (%)</th>
<th>$HF_{es}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>UC†</td>
<td>545</td>
<td>0.48</td>
<td>-</td>
<td>93</td>
<td>93</td>
<td>100</td>
<td>0</td>
</tr>
<tr>
<td>UC†</td>
<td>446</td>
<td>0.29</td>
<td>-</td>
<td>76</td>
<td>76</td>
<td>82</td>
<td>0</td>
</tr>
<tr>
<td>Li-ion</td>
<td>291</td>
<td>-</td>
<td>32</td>
<td>93</td>
<td>93</td>
<td>100</td>
<td>100.0</td>
</tr>
<tr>
<td>Li-ion/UC†</td>
<td>261</td>
<td>0.65</td>
<td>27</td>
<td>44</td>
<td>71</td>
<td>222</td>
<td>6.2</td>
</tr>
<tr>
<td>Li-ion/UC†</td>
<td>132</td>
<td>0.51</td>
<td>14</td>
<td>22</td>
<td>36</td>
<td>113</td>
<td>9.5</td>
</tr>
<tr>
<td>Ni-MH</td>
<td>412</td>
<td>-</td>
<td>53</td>
<td>-</td>
<td>53</td>
<td>100</td>
<td>100.0</td>
</tr>
<tr>
<td>Ni-MH/UC</td>
<td>263</td>
<td>0.67</td>
<td>34</td>
<td>44</td>
<td>78</td>
<td>147</td>
<td>6.4</td>
</tr>
<tr>
<td>Ni-MH/UC†</td>
<td>138</td>
<td>0.67</td>
<td>18</td>
<td>23</td>
<td>41</td>
<td>77</td>
<td>12.0</td>
</tr>
</tbody>
</table>

† with boost converter

In Figure 16, *e.g.*, the power - and energy distribution between the battery (Ni-MH) and UC with boost converter as a function of time are shown respectively. The found cut-off frequency is $f_c = 5.83$ mHz corresponding to a battery time constant of approximately 172 s.

### 4.3 Cost Price of Dual-Storage Systems

The cost price of the batteries and an UC in $/unit is shown in Table 13. In the same table the life expectancy is also given, which has been estimated by assuming that 1500 cycles correspond to approximately 5 years (Griffith, 2002). It can be seen in Table 13 that an UC has approximately three times the life expectancy of a battery (Li-ion, Ni-MH). In Table 14 the cost prices of the (single-) dual-storage systems are listed assuming an average cost price for the Li-ion and Ni-MH battery of 275 $$/kWh. Table 14 shows that only the total system cost for a dual-storage system consisting of a Ni-MH battery/UC with boost converter is reduced compared to the initial single-storage system. Note that the cost price for the boost converter
Figure 16  Power and energy distribution over time between battery (Ni-MH) and ultra capacitor pack with boost converter

is not considered. Moreover, what has not been taken in to account for the cost analysis is the ability to increase the life expectancy of the battery with the use of the capacitor. The life expectancy for the battery for the single and the dual storage system is kept constant, whereas in reality the life expectancy for the battery is potentially improved (Baisden and Emadi, 2004). The relative high cost price for the UC module is caused by the relative high required energy buffer size and the relative high minimum controller input voltage specification. Although, even with a boost converter (increasing the voltage swing from 0.48 to 0.29, see, Table 12) the cost price is still relatively high compared to battery systems. Hybridization of the dual-storage system with a Li-ion battery has no cost benefits. This is explained further with Figure 17. In this figure the system cost price (single, dual)
as a function of the average battery cost price in $/kWh for the Li-ion and Ni-MH battery is shown. The break-even cost price for the Ni-MH battery is 269 $/kWh, whereas the break-even cost price for the Li-ion battery is much higher (> 300 $/kWh). It can be seen, that if the life expectancy of the battery for example would be increased from 3.3 years to 5 years, then the break-even cost price for a Ni-MH battery would decrease by 15% from 269 $/kWh to 230 $/kWh. Additionally, the figure shows that the difference in cost of dual-storage system with a Li-ion battery is close to a dual-storage system with a Ni-MH battery. Overall, it can be concluded, that the Li-ion battery is from a energy -, power density specification and a cost price point of view the most preferable energy storage system.

Furthermore, if the cost price of the UC decreases by 38% from 0.01 $/Farad to approximately 0.0062 $/Farad, then the cost price of a dual-storage system with a Li-ion and UC becomes equal to the initial single-storage battery system. This is shown in Figure 18. Note that the UC cost price reduction target for 2006 was 50% resulting in a decrease from 0.01 $/Farad to 0.005 $/Farad (Maxwell, 2007). Obviously, energy storage hybridization of a Li-ion with an UC module may become in the future from a cost price point of view also an attractive option. However, this development strongly depends on the Li-ion battery specification developments and still requires drastic reduction of UC cost price.

Table 13  Cost-price and cycle life of storage systems

<table>
<thead>
<tr>
<th>Storage type</th>
<th>Cost price</th>
<th>Cycle life</th>
<th>Life expectancy</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>Li-ion</td>
<td>300 $/kWh</td>
<td>1000+</td>
<td>3.3+ years</td>
<td>Saft</td>
</tr>
<tr>
<td>Ni-MH</td>
<td>250 $/kWh</td>
<td>750 – 1200+</td>
<td>2.5 – 4+ years</td>
<td>Saft</td>
</tr>
<tr>
<td>UC</td>
<td>0.01 $/Farad</td>
<td>$10^5 – 10^6</td>
<td>10 years</td>
<td>Maxwell</td>
</tr>
</tbody>
</table>

5 Conclusion

A modeling and simulation approach in characterizing the component technologies for a medium-duty hybrid electric truck has been discussed. The optimal
Table 14 System cost prices based on 10 years life expectance (kept constant: HF_{125kW} = 23.3\%, m_v = 7.3\ ton and 125-kW engine)

<table>
<thead>
<tr>
<th>Storage type</th>
<th>Cost price (10 years)</th>
<th>Total cost price (excl. boost converter)</th>
</tr>
</thead>
<tbody>
<tr>
<td>UC</td>
<td>$5252</td>
<td>$5252</td>
</tr>
<tr>
<td>UC†</td>
<td>$4290</td>
<td>$4290</td>
</tr>
<tr>
<td>Li-ion</td>
<td>$1441</td>
<td>$1441</td>
</tr>
<tr>
<td>Li-ion/UC</td>
<td>$1293/$2516</td>
<td>$3809</td>
</tr>
<tr>
<td>Li-ion/UC†</td>
<td>$654/$1273</td>
<td>$1928</td>
</tr>
<tr>
<td>Ni-MH</td>
<td>$2039</td>
<td>$2039</td>
</tr>
<tr>
<td>Ni-MH/UC</td>
<td>$1304/$2536</td>
<td>$3840</td>
</tr>
<tr>
<td>Ni-MH/UC†</td>
<td>$682/$1327</td>
<td>$2009</td>
</tr>
</tbody>
</table>

† with boost converter; battery cost price of 275 $/kWh and life expectance of 3.3 years assumed for Li-ion and Ni-MH battery cell

Figure 17 System cost price as a function of the battery cost price in $/kWh (SS = Single-storage; DS = Dual-storage)

degree-of-hybridization for the drive train and energy storage system (dual-storage system) has been determined. Furthermore, the control strategy of a hybrid electric drive train is designed. The influence of the gross-vehicle weight on optimal component sizing has been investigated. The results show that the degree-of-hybridization of developed medium-duty hybrid electric trucks, whereby the fuel economy is mea-
Figure 18  Cost price benefits in $ by comparing a single-storage battery system with a dual-storage system as a function of the UC cost price in $/Farad and the battery cost price in $/kWh.

sured on different duty drive cycles, is close to the determined optimal degree-of-hybridization using the methods as described in this paper.

Only the total mass of the dual-storage system consisting of a Ni-MH/UC with boost converter is significantly reduced with 23% compared the initial system mass. Although, adding an UC to a battery reduces the battery power demands and therefore the battery wear significantly, a boost converter is a critical element in reducing the dual-storage system mass. In general, especially for battery technologies with a relative low power density (W/kg). In this case the Ni-MH battery has a lower power density than the Li-ion battery. Finally, it can be concluded that the Li-ion battery is from an energy-, power density specification as well as cost point of view the most preferable energy storage system. However, if the cost of ultra capacitor cells are significantly decreased (> 50%) then hybridization of a Li-ion battery with an ultra capacitor module in combination with a boost converter may become an
attractive technology package in the future.

Acknowledgements

This study is part of “Impulse Drive” which is a research project at the Technische Universiteit Eindhoven in The Netherlands within the section Control Systems Technology of the Department of Mechanical Engineering. The project is financially supported by the NWO Technology Foundation within the Innovational Research Incentives Scheme 2000/2001.

References and Notes


Butler, K.L., M. Ehsani and P. Kamath (1999). A matlab-based modeling and
simulation package for electric and hybrid electric vehicle design. *IEEE-Trans. on Vehicular Technology* **48**, 1770–1778.


Muta, K., M. Yamazaki and J. Tokieda (2004). Development of a new genera-


