The Impulse Drive research programme focusses on design methodologies for hybrid vehicles with significant reduction of fuel consumption (50% - 75%) and CO₂ emissions on a representative drive cycle. Powertrain hybridization implies adding a secondary power source or Energy Exchange System (EES) to a primary power source in order to increase the driving functions of a vehicle. The EES can enhance the fuel consumption, emissions, comfort, driving performance and safety. In this paper a bottom-up design process is discussed, which focusses on determining the generic design specifications of the EES independent of the applied component technology and topology. In this paper the influence of an EES on energy exchange with vehicle load and engine will be investigated separately. The simulation results show that the vehicle energy recovery efficiency is strongly determined by the motor/generator efficiencies, sizes and vehicle mass. The fuel economy is improved due to avoiding lower efficiency operation of the engine by increase of generator and motor size of the EES.

Topics: energy management strategy, optimization, fuel consumption, energy recovery, hybrid powertrain

1 INTRODUCTION

A hybrid vehicle uses at least two different power sources to propel the vehicle. One of them can generate power for energy storage in an accumulator, in order to increase the driving functions of the vehicle propulsion system. The driving functions can enhance the Fuel consumption, Emissions, Comfort, Driveability (performance) and Safety. A secondary power source which consists of a bi-directional energy accumulator with energy conversion components is further mentioned in this paper as an Energy Exchange System (EES). Examples of EESs are battery combined with an electromotor ([1], [2], [3]) or a flywheel combined with a continuously variable transmission. Examples are: (I) the fuel consumption and emissions can be reduced, because a smaller primary power source (engine or fuel cell) can used together with an EES which supplies high power to compensate the engine size. (II) The primary power source operation is kept in high-efficiency region by using the EES to manage the vehicle load, (III) more fuel consumption and emission reduction with the application of an EES can be achieved by restarting the primary power source after idle-stop, (IV) or by regenerate braking energy in the EES, which stores energy for later use, (V) The comfort and driveability can be increased by absorption of the engine and drive-line torques by the EES, (VI) an EES masks deficiencies of conventional drivetrains such as torque interruptions during shifts, (VII) an EES can provide high torque at low speed, which gives satisfying launch feel, (VIII) The safety can be enhanced by the application of advanced electric braking systems or torque traction systems (all-wheel-drive confidence). In this case, a safety function could also be combined with the fuel consumption function regarding brake energy recovery at all four wheels. The improvement of the described driving functions can be determined after a certain EES technology and topology is chosen ([1], [2], [3], [4]). However one of the major difficulties is how to determine the optimal topology fulfilling the required drive function. Honda Civic IMA and the Toyota Prius are clear examples of different topologies for the same driving functions. In this paper a bottom-up design process is proposed, in which the design specifications of the EES are determined in order to select the best topology regarding a certain driving function improvement objective.

1.1 Bottom-Up Design Process

The bottom-up design process which we propose can be divided in several design stages, see figure 1.

![Figure 1: Bottom-Up Design Process](image-url)
(I) The design specifications of an EES depend on the functional constraints of the vehicle, but also on the required drive function improvement. The various functions of the EES will lead to different specifications and therefore different EES technologies and topologies.

(II) Typically, the EES component specifications are maximum motor-generator power & efficiency and storage capacity & efficiency. The bottom-up design process is characterized by determining the generic design specifications of the EES independent of the component technology and topology by using an optimization tool for the energy management. In this phase the efficiencies are assumed to be constant and independent of engine torque, speed and state of the accumulator.

(III) The outcome of the simulations in phase II will be the optimal Energy Management Strategy (EMS) in accordance with the required constant system component efficiencies, sizes and power specifications. An approach called Dynamic Programming (DP) will be used in order to find the optimal control strategy for the EES, because with the DP technique it is possible to handle difficult constraint sets such as integer or discrete sets. Furthermore, DP leads to a globally optimal solution and is therefore useful as a benchmark for other optimization algorithms ([4]). The design specifications will be a good benchmark in order to choose system component technology from a Motor-Generator-Accumulator (MGA) library.

(IV) DP will be used to select alternative EES topologies with optimal energy management control with the chosen EES technologies. In this phase the power dependent efficiency of the system components and kinematical constraints of the hybrid drivetrain topology will be incorporated in these simulations.

(V) The control objectives or drive functions such as fuel consumption minimization or driveability are subjected to an integral constraint i.e. maintaining state-of-charge. This constraint requires for the DP solver to have foreknowledge of the drive cycle. So for real-time implementation other types of algorithms will be investigated (e.g. Rule-Based (RB) EMS). The RB EMS is mainly based on engineering intuition and analysis of efficiency specifications of system components, while the DP algorithm is based to compute the optimal control strategy. Additionally, DP can be used to construct improved rules ([5], [6], [7], [8]).

2 SIMULATION MODEL & METHOD

In this section the simulation model and method will be discussed in order to determine the design specifications of the EES (see figure 1, II). A generic energy conversion scheme for a hybrid vehicle is shown in figure 2. The energy sources (accumulators) are $E_f$, $E_v$, $E_a$ and $E_{el}$. The energy flow paths are depicted by the arrows. $E_f$ represents the primary or chemical energy source, which could be for example fossil fuel or hydrogen fuel. $E_v$ represents the vehicle load which accumulates and dissipates energy over a certain drive cycle. $E_a$ represents an energy accumulator, which is able to store energy, but also to supply energy (e.g. battery or flywheel). The electrical (accessory) loads are described by $E_{el}$. The energy conversion efficiencies, which are depicted by $\eta_1$ … $\eta_6$, are defined as an integral conversions efficiency between the energy sources.

2.1 Optimization criteria

The goal is to optimize the energy flow between the energy sources over a defined drive cycle in order to

- Minimize the fuel consumption and emissions.
- Maintain state-of-charge of the accumulator within a certain range.
- Accomplish any drive power demand.

The energy conversion efficiencies $\eta_2$, $\eta_3$ and energy accumulator $E_a$ are part of the EES (see figure 2). The Toyota Prius is used to further explain the defined energy conversion efficiencies by looking more detail to the applied topology and system components (in the described configuration the electrical loads are left out of consideration).

**Toyota Prius**

The topology and the system components of the Toyota Prius power drive train are shown in the following figure. The parallel-series hybrid power train of the Toyota Prius concept consists of a
combination of engine, motor-generator (M/G1) and planetary gear set and a combination of motor-generator (M/G2) and battery. The engine output power is split into two paths, and controlled by the generator. The first energy flow path, i.e. the mechanical path, from the engine to the carrier to the ring gear to the vehicle. The secondary energy flow path is from the engine to carrier and sun gear to the generator $G_1$ to the motor $M_1$ to the vehicle. The engine speed can be controlled to the desired value by changing the speed of the generator at a certain vehicle speed. This is the main difference between the Honda Civic IMA and the Toyota concept. The Toyota concept uses an electro-mechanical CVT instead of a mechanical CVT to control the speed of the engine. The total energy conversion efficiency is

$$\eta = \eta_{eng} \cdot \eta_{em, \text{CVT}}$$  \hspace{1cm} (1)$$

with $\eta_{em, \text{CVT}} = \eta_{m, \text{CVT}} (\eta_{plan}, \eta_{M/G1}, \eta_{M/G2})$. The generated electrical energy by the engine can also be used for charging the battery ($\eta_2$).

$$\eta_2 = \eta_{eng} \cdot \eta_{plan} \cdot \eta_{M/G1} \cdot \eta_{bat}.$$  \hspace{1cm} (2)$$

The battery energy can be used for power assisting the engine or fully electrical driving ($\eta_3$).

$$\eta_3 = \eta_{M/G2} \cdot \eta_{bat}.$$  \hspace{1cm} (3)$$

It can be seen that the total energy conversion efficiencies between primary energy source, $E_a$ and vehicle are determined by the component technology and topology.

### 3 ENERGETIC PROFITS WITH HYBRIDIZATION

The sensitivity to regenerate brake effectiveness regarding fuel consumption for different drivetrain topologies has been investigated in [9]. In this section the influence of the motor-generator size and efficiencies of the EES on energy recovery and fuel consumption reduction independent of topology and technology will be investigated separately.

#### 3.1 Energy Exchange with Vehicle Load

In order to estimate the influence of energy recovery during braking (BER) $\eta_{BER}$ is separately defined for generating $\eta_G$ and driving by the motor $\eta_M$. This is depicted by the figure 4. In order to determine the vehicle net energy dissipation $E_v$, the vehicle load is calculated. The total driving resistance is made up of the driving resistances as follows

$$F_v = \frac{1}{2} \rho \cdot v^2 \cdot A \cdot c_d + m_v \cdot (g \cdot f_r \cdot \cos \alpha + \ldots)$$

\[ \begin{align*}
\text{drag} \\
\text{red} \\
\text{inertia} \\
\text{gradient} \\
\text{inertia}
\end{align*} \]

with for $\alpha$ gradient angle, $v$ vehicle speed, $g$ gravity, $ho$ air density, $a$ acceleration, $R_w$ wheel radius and the rotational inertia coefficient $\lambda$,

$$\lambda = 1 + \frac{\sum J_{red}}{m_v \cdot R_w^2}.$$  \hspace{1cm} (5)$$

The moment of inertia of the rotating drive elements of engine, clutch, gearbox, drive shaft, etc. are reduced to the drive axle $J_{red}$ (see for other variables table 1). The required vehicle drive power $P_v$ is a product of the total vehicle resistance force $F_v$ and required vehicle speed $v_v$ (determined by the drive cycle). The total net energy dissipation $E_v$ by driving a vehicle over a drive cycle is the integral of the vehicle drive power,

$$E_v = \int_0^{t_{end}} P_v \cdot dt = E_v^* - E_a$$  \hspace{1cm} (6)$$

which is the difference between the required total drive energy

$$E_v^* = \int_0^{t_{end}} P_v \cdot dt \text{ for } P_v(t) \geq 0$$  \hspace{1cm} (7)$$

and net energy recovered during braking, stored in an accumulator and re-used for driving the vehicle,

$$E_a = \eta_M \cdot \eta_G \cdot \int_0^{t_{end}} P_v \cdot dt \text{ for } P_v(t) < 0$$  \hspace{1cm} (8)$$

with the assumption that $\eta_G$ and $\eta_M$ are constant and independent of operation point of generator and motor. In principle for a conventional vehicle no energy recovery during braking is possible i.e. $E_{BER} = 0$. However, when the clutch is engaged during braking, the kinetic energy of the vehicle can be used to compensate engine friction losses thereby reducing the fuel consumption. The net energy dissipation reduction or energy recovery efficiency $\eta_{ber}$ for a vehicle after substitution of eq. (8) in eq. (6) is defined as

$$\eta_{ber} = \frac{\eta^2 \cdot E_{BER}}{E_v^*} = \eta^2 \cdot \eta_{ber,max}$$  \hspace{1cm} (9)$$

with the assumption that $\eta_3 = \eta_G = \eta_M$. For a certain vehicle mass can be concluded that to obtain more than 50% of $\eta_{ber,max}$ the generator efficiency $\eta_G$ needs to be at least 70%. It is assumed that the generator and motor are sufficiently large to convert and transmit all brake energy from vehicle through all wheel-road interfaces to the accumulator and vice-versa.
3.2 Vehicle parameters In order to investigate which vehicle parameters, as listed in the table 1, have significant influence on $E_t$, each parameter is varied +/-20% to a reference vehicle. The reference vehicle is assumed to be a mid-size passenger car of 1375 kg. The power train parameters of the reference vehicle used are listed in table 1. The mechanical transmission efficiency between engine and vehicle is assumed to be 100 %. For calculating the vehicle drive power the NEDC has been used. The results are shown in figure 5. The vehicle parameter sensitivity analysis shows that the motor-generator efficiency $\eta_{G}$ and the vehicle mass $m_v$ have significant influence on the vehicle net energy dissipation. The maximum amount of BER also depends on the generator size. Therefore, the energy recovery efficiency $\eta_{er}$ as function of the generator size for different vehicle classes has been calculated and is shown in figure 6. In these calculations the rolling resistance, air drag resistance and frontal area are assumed to be constant and equal to the values of the reference vehicle. The generator efficiency is assumed to be 100%. It can be seen that the maximum energy recovery efficiency $\eta_{er,max}$ is in the order of 22%-31%. The energy recovery efficiency $\eta_{er}$ increases with increasing the vehicle mass. Moreover, the minimum required generator size to obtain approximately $\geq 95\%$ of $\eta_{er,max}$ is 9 kW for a mini-compact passenger car and 25 kW for a luxurious passenger car. Although the vehicle mass of the luxurious car is almost twice of the mass of the mini-compact car, the required generator size is almost three times the size of the mini-compact car. This caused by the relatively higher specific power dissipation (Wh/kg) due to air drag of the mini-compact car.

3.3 Energy Exchange with Engine In this section the influence of the generator size and motor/generator efficiency on the fuel consumption reduction will be investigated. The engine supplies energy directly to the vehicle (path 2) and indirectly to the vehicle via the EES (path 1). This is shown in figure 7. In this chapter, all negative vehicle drive powers are assumed to be dissipated i.e. $E_{BER} = 0$. The fuel consumption is determined by the engine operation strategy. The generator size $P_{Gi,max}$ determines the minimum engine operation power $P_{ice,min}$ or $P_{ice,max} = P_{ice,min}$. In order to investigate the influence of the generator size the following strategy is chosen,

$$P_v = \begin{cases} P_{ice}, & P_v > P_{Gi,max} \\ P_M \cdot \eta_M, & P_v \leq P_{Gi,max} \end{cases}$$

(10)

All required vehicle drive powers $P_v$ above $P_{Gi,max}$ is supplied by the engine ($P_{ice}$) and powers below $P_{P2,max}$

![Figure 5: Vehicle parameter influence on total net energy dissipation](image)

![Figure 6: Energy recovery efficiency $\eta_{BER}$ in % as function of Generator Size in kW](image)

![Figure 7: Energy flow scheme](image)
is only supplied by the motor of the EES (see figure 8). The energy required for vehicle driving by the motor is

\[ \int_0^{t_m} P_M \cdot dt = \int_0^{t_c} P_{\text{ice, min}} \cdot \eta_G \cdot dt \]  \hspace{1cm} (11)

with

\[ \int_0^{t_m} P_m \cdot dt = \frac{1}{2} \cdot \frac{P_{\text{ice, min}}}{\eta_M} \cdot t_m \]  \hspace{1cm} (12)

for the required motor drive power as function of time as shown in figure 9. Thereby, the charging time of the accumulator by the engine \( t_c \) should be

\[ t_c = \frac{1}{2} \cdot t_m \cdot \frac{1}{\eta_M \cdot \eta_G} \]  \hspace{1cm} (13)

after substitution of eq. (12) in eq. (11). It can be seen that the charging time \( t_c \) increases when the motor/generator efficiency decreases. For optimal engine operation the engine should be operated at the Optimal Operating Line (OOL). The OOL for minimum fuel consumption can be determined from the Brake Specific Fuel Consumption (BSFC) engine map and iso-power curves. The optimal engine operation points are defined as the points where the engine power curve crosses with the OOL. The engine is a 1.6 l 4-cylinder petrol engine with a maximum power rating of 75 kW at 540 rad/s and is assumed to be operated at the OOL (see figure 9). The additional fuel consumption due to the powertrain inertia losses (crankshaft engine, flywheel etc.) caused by engine speed changes have been left out of consideration. Furthermore, for the time that the vehicle is not moving \( t_v \) (= 480 s for the NEDC), the fuel consumption is assumed to be zero for both hybrid and conventional vehicle (the idle fuel consumption would be approximately 110 g). The motor efficiency is assumed to be equal to the generator efficiency. The cumulated operation times over the drive cycle \( t_{\text{cycle}} = 1181 \) s of the engine \( t_e + t_c \) and motor \( t_m \) are shown for different motor/generator efficiencies in figure 10. It can be seen that \( t_c > t_m \) for \( \eta_G < 85\% \) and small generator sizes \( P_{G,\text{max}} < 2.2 \) kW. For all \( P_{G,\text{max}} > P_{\text{t, max}} \) the required vehicle drive power \( P_v \) is only supplied by the motor i.e. \( t_e = 0 \). Then all engine power is used for charging via path 1. For different generator sizes the fuel consumption has been calculated for the reference vehicle and is shown in figure 11. If \( P_{G,\text{max}} = 0 \) all \( P_v \) is directly supplied by the engine operated on the OOL, represented by energy flow path 2 of figure 7. It can be seen that the total engine fuel consumption, which is the sum of the fuel consumption by energy supplied via path 1 and via path 2, decreases when the generator size increases. The engine power demands are increased and so the engine is operated at a higher efficiency level. The absolute minimum fuel consumption is achieved for a fully series type of hybrid with engine operation at its ‘sweet spot’ (34 kW), only if the efficiencies of the motor and generator are 100%. The total engine fuel consumption is shown for different generator sizes and motor/generator efficiencies in figure 12. The total fuel consumption increases due to higher energy losses of the motor and generator of the EES, if the motor/generator efficiency decreases. It can be seen that for \( \eta_G < 90\% \) the fuel consumption with ‘sweet spot’ engine power operation is higher than for the conventional
vehicle operated at the OOL, because the conventional vehicle is assumed to have no other energy conversion losses.

4 CONCLUSIONS

The bottom-up design process for determining the design specifications of the EES in order to select the best topology regarding a certain drive function improvement objective has been discussed. The design process focuses on determining the generic design specifications of the EES independent of the applied component technology and topology. The energetic profits with hybridization by the application of an EES to the vehicle load and engine have been investigated separately. Additionally, the sensitivity of the vehicle parameters on the vehicle load has been investigated. The simulation results show that the energy recovery efficiency \( \eta_{er} \) is strongly determined by the motor/generator efficiencies, sizes and vehicle mass. The maximum of \( \eta_{er} \) and minimum required generator size varies between 22% - 31% and 13 kW - 34 kW for vehicle masses between 800 kg and 1650 kg for \( \eta_G = 100\% \) (NEDC). The required \( \eta_G \) needs to be at least 70% to obtain more than 50% of \( \eta_{er,max} \). If the engine exchanges energy with the EES, the fuel consumption decreases significantly (20% at ‘sweet spot’) compared to a conventional vehicle operated with engine at the OOL. By increase of generator size the engine power operation range is constrained and at higher efficiency level. However, for \( \eta_G < 90\% \) the fuel consumption is higher than for the conventional vehicle, because the conventional powertrain has assumed no other energy conversion losses. In future research the influence of engine downsizing with an EES on the fuel consumption will be investigated.

5 REFERENCES


Acknowledgment This study is part of “Impulse Drive” which is a research project at the Technische Universität 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 Schemes 2000/2001.