The potential of thermoelectric generator in parallel hybrid vehicle applications

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Abstract

This paper reports on an investigation into the potential for a thermoelectric generator (TEG) to improve the fuel economy of a mild hybrid vehicle. A simulation model of a parallel hybrid vehicle equipped with a TEG in the exhaust system is presented. This model is made up by three sub-models: a parallel hybrid vehicle model, an exhaust model and a TEG model. The model is based on a quasi-static approach, which runs a fast and simple estimation of the fuel consumption and CO$_2$ emissions. The model is validated against both experimental and published data. Using this model, the annual fuel saving, CO$_2$ reduction and net present value (NPV) of the TEG’s life time fuel saving are all investigated. The model is also used as a flexible tool for analysis of the sensitivity of vehicle fuel consumption to the TEG design parameters. The analysis results give an effective basis for optimization of the TEG design.

Introduction

Currently 14% of global greenhouse gas emissions are from transportation [1]. In order to reduce greenhouse gas emissions, a number of CO$_2$ regulations for the road transport sector have been proposed around the world. In the EU, the law requires that by 2020, the fleet average to be achieved by all new cars is 95 grams of CO$_2$ per kilometer [2]. In the US, the Environment Protection Agency (EPA) and National Highway Traffic Safety Administration (NHTSA) raised the requirement for fuel economy of new passenger vehicles to 54.5 miles per gallon for the years 2017-2025 [3]. In Japan, the Top Runner program set 20.3 km per liter of fuel as the fuel efficiency target of passenger cars for 2020[4]. In the face of these internationally tightened requirements and regulations for passenger cars, the improvement of fuel economy and the development of alternative fuels has been a focus of research and design efforts.

Based on the typical energy flow path of an internal combustion engine (ICE), approximately one third of the energy is discharged by exhaust gas. Due to this great potential of waste heat recovery (WHR) in automotive, many efforts have been made in this field during the last few years, such as turbo-compounding [5], Rankine cycles [6], thermoelectric generators (TEG) [7], thermochemical recuperation (TCR) [8], and Stirling engines [9]. TEG has attracted substantial interest because of its advantage of silent operation, and compactness. Most of the current studies focus on the integration of TEG with conventional vehicles, using it to replace or relieve the alternator [7,10]. However, the use of a TEG in hybrids can be an especially desirable integration in the future where the number of hybrid vehicles is rising rapidly [11].

The main objectives of this paper:

- Using a simulation model of a TEG integrated in a hybrid vehicle to investigate the potential of fuel saving and CO$_2$ reduction.
- Identify the main TEG design parameters’ influence on fuel saving by means of a sensitivity analysis.

The body structure of this paper starts by highlighting advantages, limitations, and related research of the TEG integration with hybrid vehicle. The simulation model, which is made up by three sub-models, is displayed in the section of model structure. The following section presents the model validation. In the section showing simulation results, the annual fuel saving, CO$_2$ reduction and net present value (NPV) of the TEG’s life time fuel saving are calculated based on simulation results. In the following section, a sensitivity analysis is carried out for the whole vehicle. Finally, the last section presents the main conclusion.

Potential for TEG in Hybrid Vehicle

The use of TEG in hybrid vehicles has many advantages over using it in conventional ICE vehicles. According to Roland Berger’s report [11], the new vehicle sales market share of hybrid vehicles will increase significantly in the future, while the market share for the conventional ICE vehicles will shrink. Therefore, there is reason to believe that there will be a promising market for TEG integrated with hybrid vehicles. In a conventional ICE vehicle, the power generated by TEG can only be used when the electricity is needed in the vehicle. In hybrids, the electrical energy can be used directly for propulsion and the more energy recovered from the exhaust, the longer the motor can assist the engine. Vijayagopal et al [12] conducted a simulation analysis for the benefits of TEG varied with the type of vehicle: a conventional vehicle, a mild hybrid and a full hybrid. This study has shown that although the average power of TEG in conventional vehicle is higher than both two hybrid vehicles, the mild hybrid vehicle has the greatest fuel economy improvements because of its effective use of the recovered energy.

The main challenge and limitation for the integration of TEG with hybrid vehicle is the intermittent engine operation and lower total waste heat relative to a conventional vehicle. However, Kerstin et al [13] show that the total exhaust energy in hybrid vehicles is indeed less than in conventional vehicles, but because of high engine load, the exhaust temperature in hybrid operation is higher, which results in a high efficiency of TEG. Additionally, there are no conditions when either exhaust flow rate or temperature are low, and in general the number of operating points is fewer compared with engines in a conventional powertrain. Therefore, the design operating point is close to the maximum operating point and there is no need to bypass the exhaust flow.
**Model Structure**

As can be seen in Figure 1, the simulation model of hybrid vehicle equipped with TEG is made up by three sub-models: a parallel hybrid vehicle model, an exhaust model and a TEG model. The parallel hybrid vehicle model is used to calculate the engine and motor’s load and speed, fuel consumption and electrical energy consumption and generation based on the chosen driving cycle. The exhaust energy is computed at the exhaust model based on the engine speed and load. TEG model predicts the power output from recovering the exhaust energy and stores it in the battery. The energy balance of the whole system is controlled by the control system in the parallel hybrid vehicle model, using the equivalent consumption minimization strategy (ECMS) [14].

![Figure 1. Model structure](image)

The structure of the TEG system. The size of TEG constrained to 0.3m×0.24m×0.1m with TEMs on both sides of the hot side heat exchanger measuring 0.3m×0.24m×0.04m. 24 offset strip fins are used in hot side heat exchanger and they increase the total heat transfer area of each heat exchanger to around 0.7m². Based on reference [16], the average heat transfer coefficients for hot side and cold side heat exchanger are respectively 120 and 105 W/m²K.

![Figure 2. The structure of the TEG system](image)

The simulation is based on high temperature thermoelectrical material Skutterudites materials working at a $ZT_m$ value of 0.7[16]. Based on reference [17], the average thermal resistance of a Skutterudites TEM with the size of 16mm×13mm×2mm is 11K/W. Since the Skutterudite modules can be fabricated in different sizes and with different thermal resistances, instead of defining the number of TEMs, a fill factor $F$ is defined as

$$F = \frac{n_{TEM} A_{TEM}}{2 A_{HXR}}$$ (1)

where $n_{TEM}$ is number of TEMs in heat exchanger. $A_{TEM}$ is area of a TEM and $A_{HXR}$ is area of a heat exchanger surface.

**Table 1. Specification for Audi A6 E-Tron**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Curb weight ($m_w$)</td>
<td>km</td>
</tr>
<tr>
<td>Drag coefficient ($D_p$)</td>
<td>-</td>
</tr>
<tr>
<td>Vehicle frontal area ($A_f$)</td>
<td>m²</td>
</tr>
<tr>
<td>Tire radius ($R_t$)</td>
<td>m</td>
</tr>
<tr>
<td>Tire inertia ($I_t$)</td>
<td>kg·m²</td>
</tr>
<tr>
<td>Engine Displacement ($V_e$)</td>
<td>1</td>
</tr>
<tr>
<td>Engine Type</td>
<td>Gasoline</td>
</tr>
<tr>
<td>Engine inertia ($I_e$)</td>
<td>kg·m²</td>
</tr>
<tr>
<td>Engine gear ratio ($r_e$)</td>
<td>-</td>
</tr>
<tr>
<td>Motor gear ratio ($r_m$)</td>
<td>-</td>
</tr>
<tr>
<td>Engine power ($P_e$)</td>
<td>kW</td>
</tr>
<tr>
<td>Motor power ($P_m$)</td>
<td>kW</td>
</tr>
<tr>
<td>Auxiliary power ($P_{aux}$)</td>
<td>kW</td>
</tr>
</tbody>
</table>

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Table 2. Specification for TEG

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEG install position</td>
<td>after catalytic convertor</td>
</tr>
<tr>
<td>TEG weight $M_{TEG}$</td>
<td>kg</td>
</tr>
<tr>
<td>Size of a heat exchanger</td>
<td>m³</td>
</tr>
<tr>
<td>Heat transfer area $A_{TEG}$</td>
<td>m²</td>
</tr>
<tr>
<td>Heat transfer coefficient of hot side heat</td>
<td>W/m² K</td>
</tr>
<tr>
<td>Heat transfer coefficient of cold side heat</td>
<td>W/m² K</td>
</tr>
<tr>
<td>Figure of merit ZT</td>
<td>-</td>
</tr>
<tr>
<td>Size of a TEM</td>
<td>m³</td>
</tr>
<tr>
<td>Thermal resistance of a TEM $R_{TEM}$</td>
<td>K/W</td>
</tr>
<tr>
<td>Fill factor of TEMs in heat exchanger $F$</td>
<td>-</td>
</tr>
<tr>
<td>Coolant mass flow rate $\dot{m}_\text{cool}$</td>
<td>kg/s</td>
</tr>
<tr>
<td>Coolant temperature $T_{\text{cool}}$</td>
<td>K</td>
</tr>
</tbody>
</table>

To tune and validate the model, test data is collected from engine laboratory and published papers. The data include dynamic exhaust data of a BMW 530i six-cylinder gasoline in NEDC cycle [10] and TEG test data taken from a CAT C6.6 ACERT diesel engine.

**Vehicle Model**

The vehicle model used here is a parallel hybrid vehicle model using the ECMS control strategy [14]. The diagram of the parallel hybrid powertrain considered is shown in Figure 3.

![Diagram of the parallel hybrid vehicle](image)

**Figure 3. Diagram of the parallel hybrid vehicle**

Based on the quasistatic approach, the wheel speed $\omega_w$ and torque $T_w$ of each instant are calculated so that they meet the driving cycle’s demand. At the power split device (PSD), the basic relationship of the torque balance is

$$T_w(t) = T_p(t) + T_{\text{e}}(t)$$  \hspace{1cm} (2)

In the fuel path, based on the gear ratio of the engine $r_e$, the engine torque $T_{ic}(t)$ and speed $\omega_{ic}(t)$ can be calculated as follow:

$$T_{ic}(t) = \frac{T_p(t)}{r_e}$$  \hspace{1cm} (3)

$$\omega_{ic}(t) = \omega_w(t)r_e$$  \hspace{1cm} (4)

The fuel consumption of the engine $\dot{m}_f(t)$ is calculated as a tabulated function of engine speed $\omega_{ic}$ and torque $T_{ic}$

$$\dot{m}_f(t) = f_{ic}(T_{ic}(t), \omega_{ic}(t))$$  \hspace{1cm} (5)

In the electrical path, the motor torque $T_{em}(t)$ and speed $\omega_{em}(t)$ can also be calculated according to the gear ratio of the gearbox $r_m$

$$T_{em}(t) = \frac{T_{epath}(t)}{r_m}$$  \hspace{1cm} (6)

$$\omega_{em}(t) = \omega_w(t)r_m$$  \hspace{1cm} (7)

The output power of the battery $P_b(t)$ includes two parts: the output power of the motor $P_{em}$ and the TEG $P_{TEG}$.

$$P_b(t) = P_{em}(T_{em}(t), \omega_{em}(t)) + P_{TEG}(T_{ic}(t), \omega_{ic}(t))$$  \hspace{1cm} (8)

The ECMS regulates the torque distribution between the thermal and electrical paths with the torque split factor $u(t)$, which is defined as

$$u(t) = \frac{T_{epath}(t)}{T_w(t)}$$  \hspace{1cm} (9)

When $u(t) = 0$, it means that all the torque needed at the wheels is provided by the fuel path. When $u(t) = 1$, it means all the torque needed at the wheel is provided by the electrical path or all the braking energy at the wheel is regenerated along the electrical path.

The ECMS finds control variable $u(t)$ by minimizing the cost function $J(t, u)$, which is defined as

$$J(t, u) = \Delta E_f(t, u) + s(t)\Delta E_e(t, u)$$  \hspace{1cm} (10)

$s(t)$ is the equivalence factor that is calculated online as a function of the current system status and some control parameters. The detail of calculation of $s(t)$ is presented in [14]. $\Delta E_f(t, u)$ and $\Delta E_e(t, u)$ are respectively the fuel energy and electrical energy used in interval $\Delta t$. Here $\Delta E_e(t, u)$ not only includes the energy consumed and generated by the electrical motor, but also includes the energy generated by the TEG.

$$\Delta E_f(t, u) = \int_{t}^{t+\Delta t} H_{\text{LHV}} \dot{m}_f(t, u) dt$$  \hspace{1cm} (11)

$$\Delta E_e(t, u) = \int_{t}^{t+\Delta t} P_{em}(t, u) dt + \int_{t}^{t+\Delta t} P_{TEG}(t, u) dt$$  \hspace{1cm} (12)

$H_{\text{LHV}}$ is the lower heating value of the fuel.

**Exhaust Model**

The function of the exhaust model is to calculate the exhaust flow rate and temperature. The exhaust flow rate $\dot{m}_{exh}$ can be estimated based on the fuel consumption $\dot{m}_f(t)$ and air-fuel ratio $\lambda$, which can be expressed as

$$\dot{m}_{exh}(t) = (1 + \lambda)\dot{m}_f(t)$$  \hspace{1cm} (13)

where air-fuel ratio $\lambda = 14.7$.  

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The power output of TEG is more sensitive to the exhaust temperature than the flow rate [18]. To capture the dynamic of exhaust temperature, the calculation of the exhaust temperature is based on the mean value engine model (MVEM) developed by Eriksson [19]. The core of the MVEM for exhaust temperature is to model the engine out temperature first and then model the temperature drop along the exhaust pipe.

A linear model is used for the engine out temperature $T_{e_{\text{out}}}$:

$$T_{e_{\text{out}}} = T_{c_{\text{yl},0}} + \dot{m}_{\text{ex}} e K \quad (14)$$

where $T_{c_{\text{yl},0}}$ and $K$ are tuning constants.

The temperature drop in the exhaust pipe $T_{e_{\text{ex}}}$ is expressed as:

$$T_{e_{\text{ex}}} = T_w + (T_{e_{\text{in}}} - T_w) e \frac{h_{\text{pipe}}}{\dot{m}_{\text{ex}} c_{ph}} \quad (15)$$

$c_{ph}$ is the specific heat at constant pressure of the exhaust gas. $A_{\text{pipe}}$ is the pipe’s surface area. $h$ is the heat transfer coefficient. Here $T_{e_{\text{in}}}$ can be the engine out temperature $T_{e_{\text{out}}}$ but also can be the gas out temperature from any pipe. $T_w$ is the pipe wall temperature, which is determined by the following equation:

$$\frac{dT_w}{dt} - m_w c_w = \dot{Q}_l - \dot{Q}_e \quad (16)$$

$c_w$ is the heat capacity of the pipe wall material, and $m_w$ is the pipe wall mass. $\dot{Q}_l$ and $\dot{Q}_e$ are respectively the heat transfer from the interior and exterior. The detail of calculation for $\dot{Q}_l$, $\dot{Q}_e$ and $h$ are presented in [19].

The model structure of this exhaust model is presented in Figure 4. Since here the TEG is installed downstream of the catalytic converter, the exhaust temperature model can be divided into four sub-models (Appendix A):

- Engine out temperature $T_{e_{\text{out}}}$
- Temperature drop in the exhaust manifold $T_{e_{\text{ex}}_{\text{man}}}$
- Temperature before the catalytic converter $T_{e_{\text{ex}}_{\text{bc}}}$
- Temperature after the catalytic converter $T_{e_{\text{ex}}_{\text{ac}}}$

In the three temperature drop sub-models, they have different pipe dimensions $A_{\text{pipe}}$, heat transfer coefficient $h$ and pipe wall mass $m_w$. Since the exhaust manifold collects the exhaust gases from multiple cylinders into one pipe; the exhaust flow rate in sub-model of temperature drop in the exhaust manifold also need to be divided by the number of the cylinders $n$.

**TEG Model**

As can be seen in Figure 5, a complete TEG system consists of thermoelectric modules (TEMs), heat exchangers, a heat source, a heat sink and connecting wires. In this TEG model, the exhaust gas of an ICE serves as the heat source and the engine coolant serves as a heat sink.

Here the TEG is a 0D black box model. The inputs are exhaust temperature $T_{e_{\text{ex}}}$ and flow rate $\dot{m}_{\text{ex}}$; coolant temperature $T_{c_{\text{ol}}}$ and flow rate $\dot{m}_{\text{col}}$. The output is maximum electrical power output $P_{E_{\text{TEG}}}$, achieved by matching the external load. The purpose of the TEG model is to reduce the complexity of a potentially detailed TEG model and instead to gain a vehicle perspective. Thus, the model only captures the influence of main design parameters, such as the size of TEG $A_{\text{TEG}}$, the heat transfer coefficients of heat exchangers $U_h$ and $U_c$ and the thermo-electrical material $Z_{\text{TEG}}$. The influence of backpressure is not considered by this 0D TEG model.

Since the TEG is symmetrical with respect to its height, only half of the domain is simulated. In Figure 6, the exhaust flow absorbed by TEG is defined as $\dot{Q}_{h}$, which can be calculated as

$$\dot{Q}_{h} = c_{ph} \dot{m}_{\text{ex}} (T_{e_{\text{ex}}_{\text{bc}}} - T_{e_{\text{ex}}_{\text{out}}}) \quad (17)$$

$T_{e_{\text{ex}}_{\text{out}}}$ is the gas-out temperature of TEG; the expression of Esarte et al [20] is used:

$$T_{e_{\text{ex}}_{\text{out}}} = T_{e_{\text{ex}}} - (T_{e_{\text{ex}}} - T_{c_{\text{ol}}}) \frac{1 - e^{-NTU(1+C_R)}}{1 + C_R} \quad (18)$$

Where

$$NTU = \frac{1}{R_{E_{\text{TEG}}} \dot{m}_{\text{ex}} c_{ph}} \quad (19)$$

$$C_R = \frac{\dot{m}_{\text{ex}} c_{ph}}{\dot{m}_{\text{col}} c_{pc}} \quad (20)$$

$\dot{m}_{\text{col}}$ and $c_{pc}$ are respectively the coolant mass flow rate and specific heat for coolant. $R_{E_{\text{TEG}}}$ is the total thermal resistance of the TEG. It can be seen that the heat transfer across the TEG $\dot{Q}_{h}$ depends on the thermo-physical properties of the TEG and the hot and cold fluids. The coolant-out temperature can be expressed as
The total thermal resistance of a TEG $R_{TEG}$ is made up of the thermal resistance of respectively hot side and cold side heat exchanger $R_h, R_c$ and the total thermal resistance of the TEMs $R_{TEMs}$.

$$R_{TEG} = R_h + R_{TEMs} + R_c$$  \hspace{1cm} (22)

The thermal resistance of the heat exchangers can be calculated as follow:

$$R_h = \frac{1}{\frac{1}{U_h A_{TEG}} + \frac{1}{A_{TEG}}}$$  \hspace{1cm} (23)

$$R_c = \frac{1}{\frac{1}{U_c A_{TEG}} + \frac{1}{A_{TEG}}}$$  \hspace{1cm} (24)

$U_h$ and $U_c$ are respectively the heat transfer coefficient of hot side and cold side heat exchangers. $A_{TEG}$ represents the heat transfer area of the heat exchanger.

All the TEMs are connected thermally in parallel, thus the total thermal resistance of the TEMs $R_{TEMs}$ is calculated as

$$R_{TEMs} = \frac{R_{TEM}}{\eta_{TEM}}$$  \hspace{1cm} (25)

Substituting Equation (1) into Equation (25), the total thermal resistance of the TEMs can be expressed as:

$$R_{TEMs} = \frac{R_{TEM}A_{TEG}}{2FA_{HXR}}$$  \hspace{1cm} (26)

For both simplicity and generality, the power output is calculated using respectively the energy absorbed by the TEG $Q_h$ and TEMs’ efficiency $\eta_{TEMs}$

$$P_{TEG} = Q_h \eta_{TEMs}$$  \hspace{1cm} (27)

The idealized $\eta_{TEMs}$ can be written as [16]:

$$\eta_{TEMs} = \frac{m Z T_1}{Z T_1 + m Z T_1 + (m + 1)^2} \frac{T_1 - T_2}{T_1}$$  \hspace{1cm} (28)

where $T_1$ and $T_2$ are respectively the hot and cold side temperature of the TEMs. $T_m = (T_1 + T_2)/2$. $m = \frac{r_{int}}{r_1}$, which is the ratio of load electrical resistance $r_1$ and internal electrical resistance $r_{int}$.

A number of papers [16,21,22] have recently reported the maximum power output of TEG is achieved when $m$ is slightly greater than 1. When the thermal impedance matching is satisfied, $m$ can be expressed as:

$$m = \sqrt{Z T_1 + 1}$$  \hspace{1cm} (29)

Substituting Equation (29) into Equation (28), the efficiency of TEMs at maximum power output $\eta_{TEMs,mp}$ can be expressed as

$$\eta_{TEMs,mp} = \left[ \frac{\sqrt{Z T_1 + 1} - 1}{\sqrt{Z T_1 + 1 + \frac{r_{int}}{r_1}}} \right] \frac{T_1 - T_2}{T_1}$$  \hspace{1cm} (30)

The hot side and cold side temperature of TEMs can be expressed as follows [16]:

$$T_1 = T_h - \frac{R_h}{R_h + R_c + R_{TEMs}}(T_h - T_c)$$  \hspace{1cm} (31)

$$T_2 = T_c + \frac{R_c}{R_h + R_c + R_{TEMs}}(T_h - T_c)$$  \hspace{1cm} (32)

where $T_h$ and $T_c$ are respectively the average temperature of the hot side and cold side heat exchanger.

$$T_h = \frac{T_{exh} + T_{exh, out}}{2}$$  \hspace{1cm} (33)

$$T_c = \frac{T_{col} + T_{col, out}}{2}$$  \hspace{1cm} (34)

**Model Validation**

**Validation of the Exhaust Model**

The published dynamic exhaust data of BMW’s 3L-gasoline engine in the NEDC [10] are used to tune and validate the exhaust model. When using BMW’s published data, some assumptions are made. The mass flow rate data from BMW’s six-cylinder engine to Audi A6’s four-cylinder engine is simply extrapolated using a multiplication factor of 0.8. The multiplication factor for exhaust gas temperature is 0.92. The comparison of the extrapolated test data to the simulation results for the NEDC driving cycle is shown in Figure 7.
The simulation result of flow rate corresponds well with the test results, with a mean absolute error around 5%. For the exhaust temperature, errors at the beginning of the cycle can be explained by the unmodeled dynamics of heating and cooling effects and sensor dynamics. The variations in the temperature profile have a minor influence on the power output of the TEG \[10\]. Therefore, this exhaust model for this gasoline engine is validated and is used to provide inputs for the TEG model.

**Validation of the TEG Model**

The same apparatus as described in references \[19, 23\] is used for the TEG model validation. It can be seen in Figure 8 that a TEG with a plate-fin heat exchanger which contains four European Thermodynamics modules (GM250-127-28-10) is mounted in the EGR path of a CAT C6.6 ACERT diesel engine. All four TEMs are connected electrically in series. The cold side temperature of the TEG was maintained using chilled water from a laboratory recirculation chiller.

In order to control the hot side temperature of the modelled TEMs to within the limit of the module specifications, the authors elected to use a 30% torque NRTC (Non-Road Transient Cycle) on the C6.6 engine. The chiller is set as a fixed temperature 20\(^\circ\)C and its mass flow rate 0.1kg/s. The load electric resistance is set close to the modules’ resistances. The comparison of the test data with the simulation results is presented in Figure 9. The simulation results correspond well with the measurement results. Since the \(ZT_m\) in the TEG model is set as constant in the model while in reality it is temperature-dependent, this results in an overestimation of power at low temperature and an underestimation at high temperature. However, the errors can be further diminished when accumulating the overall power output of TEG in a driving cycle. Thus, this 0D black box TEG model can provide a reasonable accuracy.

**Validation of the ECMS**

Control parameters of the ECMS are selected based on the data listed in Table 1 and Table 2. The performance of the ECMS for this TEG hybrid vehicle in the NEDC cycle is validated throughout the range of torque distribution and battery state-of-charge(SOC), which are shown in Figure 10 and Figure 11. The ECMS distributes the torque between the engine and motor and maintains the balance of SOC with no SOC variations at the end of the NEDC cycle.
Simulation Result

In this section of the paper, the simulation results are presented. The power output of the TEG in NEDC is shown in Figure 12. During the urban section of the cycle the TEG power output averages 80W and increases to 200W over the extra-urban cycle. The peak power output reaches 450W in the extra-urban cycle.

The fuel economy and CO₂ reduction due to the TEG in hybrid vehicle operation is compared with hybrid without TEG and the results shown in Table 3. In order to make them comparable, the ECMS constrains the initial and terminal SOC for both hybrid vehicles as 0.7 and no SOC variations at the end of the cycle. The fuel saving due to the TEG is 0.15L/100km (3.4%) and the CO₂ reduction is 3.4g/km (3.3%).

<table>
<thead>
<tr>
<th>Table 3. Fuel economy and CO₂ reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hybrid vehicle</td>
</tr>
<tr>
<td>Fuel consumption (L/100km)</td>
</tr>
<tr>
<td>CO₂ emission (g/km)</td>
</tr>
</tbody>
</table>

As can be seen in Table 4, for the customers, the average fuel saving is 27.8 € while the NPV of the fuel saving over the vehicle lifetime is 253 €. For the original equipment manufacturer (OEM), the annual CO₂ reduction is 39.4kg, which helps to reduce the CO₂ emission target for the vehicle fleet. The economic benefit potential of TEG integrated with hybrid is large for both customers and OEMs.

Sensitivity Analysis

A sensitivity analysis is essential to identify which TEG design parameters have the greatest impact on the fuel saving. The core information of this sensitivity analysis is that the input variables are perturbed slightly, and the corresponding change in the outputs is reported as a percentage change in the outputs [24]. In a previous study [18], the sensitivity analysis of TEG is only on a standalone model. Here it is analysed from a vehicle system point of view.

The process of sensitivity analysis is conducted as follows. The size of the heat exchanger and the size of TEM and its thermal resistance are fixed to form a baseline. The remaining parameters are selected as TEG. The TEG parameters given in Table 2 and fuel consumption of working at these parameters are used as the base values. One TEG variable is increased 20% around its base value, while all other variables are fixed at their respective base values. Then the fuel consumption is computed and recorded as the percentage changing above or below the base values C(%), which can be expressed as follow:

\[
C(\%) = \frac{\dot{m}_{f,e} - \dot{m}_{f,b}}{\dot{m}_{f,b}} \times 100\% \quad (37)
\]
Where \( \dot{n}_{f,c} \) is fuel consumption of 20% increase in a variable and \( \dot{n}_{f,b} \) is the baseline fuel consumption.

This process is repeated for every TEG variable. Each time, the ECMS algorithm ensures there are no SOC variations. The sensitivity analysis results are presented in both Table 5 and Figure 13.

<table>
<thead>
<tr>
<th>Input Parameters</th>
<th>C (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEG’s weight ( M_{\text{TEG}} )</td>
<td>+0.0776</td>
</tr>
<tr>
<td>Heat transfer coefficient of hot side heat exchanger ( U_h )</td>
<td>-0.0588</td>
</tr>
<tr>
<td>Heat transfer coefficient of cold side heat exchanger ( U_c )</td>
<td>-0.0752</td>
</tr>
<tr>
<td>Heat transfer area ( A )</td>
<td>-0.2669</td>
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<td>Figure of merit ( Z )</td>
<td>-0.3245</td>
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<td>Fill factor of TEMs in heat exchanger ( F )</td>
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</tr>
<tr>
<td>Coolant mass flow rate ( \dot{m}_{\text{col}} )</td>
<td>-0.0092</td>
</tr>
<tr>
<td>Coolant temperature ( T_{\text{col}} )</td>
<td>-0.0095</td>
</tr>
</tbody>
</table>

It can be seen that the fuel consumption is most sensitive to the thermolectric material properties (\( Z\text{T}_{\text{m}} \) value). For the heat transfer coefficient \( (U_h \) and \( U_c) \), heat transfer area \( (A_{\text{TEG}}) \) and added weight of TEG, they both influence the fuel saving at a moderate level. On the other hand, fill factor of TEMs in heat exchanger \((F)\), coolant temperature \((T_{\text{col}})\) and its flow rate \( \dot{m}_{\text{col}} \) only have a minor effect on the fuel saving. The negligible effect of the fill factor can be explained by the fact that the \( F \) is already close to its optimal value in current TEG systems. The sensitivity analysis results can be used as guidance for the further fabrication and optimization of the TEG.

**Conclusion**

A model of the TEG integrated with a parallel hybrid vehicle has been developed to investigate the potential of the TEG to reduce CO\(_2\) emissions. The model has been validated against both experimental data and published data. The simulation results show that for the NEDC cycle, a 3.4 % fuel saving with 80-200W energy recovered from the TEG. The annual CO\(_2\) reduction is 39.4 kg. The NPV of the TEG’s life time fuel saving is 253 €. Thus, if the cost of the TEG to the OEM is less than 200 € then the installation of a TEG represents a significant gain for the customers.

A sensitivity analysis reveals the priority order of the parameters to achieve maximum fuel saving and offer guidance to the further fabrication and optimization of the TEG. When designing a TEG, the priority is to choose the material with the maximum \( Z\text{T}_{\text{m}} \) available in the desired temperature range. Then, there should be an optimisation of the design of the TEG structure with both large heat transfer area and heat transfer coefficient but still lightweight. After that, an optimisation of fill factor of TEMs in heat exchange is also necessary. Last in the priority order is maintaining the coolant temperature at low value and high flow rate.

**References**


Abbreviations

DISI    Direct Injection Spark Ignition
EGR    Exhaust Gas Recirculation
ECMS    Equivalent Consumption Minimization Strategy
ICE    Internal Combustion Engine
OEM    Original Equipment Manufacturer
NEDC    New European Driving Cycle
NRTC    Non-Road Transient Cycle
NPV    Net Present Value
QSS    Quasi Static Simulation
TCR    Thermochemical Recuperation
TEM    Thermoelectric Module
TEG    Thermoelectric Generator
WHR    Waste Heat Recovery

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Appendix A

Figure A. The overview of the TEG hybrid model

Figure B. The overview of the Exhaust model

Figure C. The overview of the TEG model