

# THE POTENTIAL OF A SEPARATED ELECTRIC COMPOUND SPARK IGNITION ENGINE FOR HYBRID VEHICLE APPLICATION

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## Abstract

In-cylinder expansion of internal combustion engines based on Diesel or Otto cycles cannot be completely brought down to ambient pressure, causing a 20% theoretical energy loss. Several systems have been implemented to recover and use this energy such as turbocharging, turbo-mechanical and turbo-electrical compounding, or the implementation of Miller Cycles. In all these cases however, the amount of energy recovered is limited allowing the engine to reach an overall efficiency incremental improvement between 4% and 9%. Implementing an adequately designed expander-generator unit could efficiently recover the unexpanded exhaust gas energy and improve efficiency. In this work, the application of the expander-generator unit to a hybrid propulsion vehicle is considered, where the onboard energy storage receives power produced by an expander-generator, which could hence be employed for vehicle propulsion through an electric drivetrain. Starting from these considerations, a simple but effective modelling approach is used to evaluate the energetic potential of a spark-ignition engine electrically supercharged and equipped with an exhaust gas expander connected to an electric generator. The overall efficiency was compared to a reference turbocharged engine within a hybrid vehicle architecture. It was found that, if adequately recovered, the unexpanded gas energy could reduce engine fuel consumption and related pollutant emissions by 4% to 12%, depending on overall power output.

**Keywords:** hybrid vehicle; compound engine; spark-ignition engine; exhaust energy recovery

## Introduction

Regulations adopted worldwide to reduce the environmental impact of human activities force the development of market solutions capable of increasing fuel economy and respecting the environment. Among the various solutions, hybrid electric vehicles (HEVs), including plug-in varieties are a promising solution, showing significant fuel consumption reduction compared to traditional internal combustion engine vehicles (ICEVs), mainly in urban applications. HEVs have a relatively short history and their developmental stage is undoubtedly open to relevant improvements. Among interventions, the internal combustion engine is a key element with room for modifications and improvements [1]. One example is reducing the exhaust gas energy loss related to incomplete gas expansion inside the cylinder of engine operating according to the Diesel or Otto thermodynamic cycles. As an example, the dashed area 4-6-1-4 in Figure 1 represents the energy loss related to the unexpanded gas in an Otto Cycle.

Several systems have been studied and proposed to recover or use unexpanded gas energy in engines, the most common being represented by turbocharging. In this case, the turbine recovers only the energy required by the turbocompressor, thus considerably limiting the recovery amount. Several other systems can be traced in the scientific literature [2] [3]. Turbo-compounding, as an example, was widely adopted in the naval propulsion sector, employing a second turbine downstream the first to complete the exhaust gas expansion, thus adding power to the propeller shaft and increasing fuel economy. In the automotive sector, several versions of turbo-compounding have been proposed. In some cases, usually indicated as “electrical turbo-compound,” an electrical generator installed on the turbocharger shaft was controlled to recover the residual power produced by the turbine not employed by the turbocompressor [4][5][6][7]. Results generally show that overall engine efficiency cannot be increased more 6%. In other cases, an auxiliary turbo-generator was installed downstream of the first turbine [8][9], reaching

a maximum fuel economy improvement of 4%. A different version has also been proposed [10][11], with an auxiliary turbo-generator installed in parallel to the turbine of the turbocharger. Experiments with both fixed and variable geometry turbines have shown efficiency improvements up to 9%.

Another possible solution for greater exploitation of in-cylinder gas expansion to increase efficiency is to adopt over-expanded cycles, such as Atkinson and Miller Cycles [12]. According to the Atkinson Cycle (1-2-3-6-1 in Figure 1), gas expansion should be prolonged down to atmospheric pressure [13][14][15], thus completely recovering both the dashed areas of Figure 1. Full expansion could be practically obtained by adopting adequate intake valve phasing. The theoretical efficiency increment compared to Otto Cycle with the same compression ratio is around 19%. However, an extremely large and impractical in-cylinder volume would be necessary. The Atkinson Cycle engine displacement is approximately four times the Otto Cycle displacement, which strongly decreases the engine's power density (IMEP reduction in the order of -72%). In the Miller Cycle, the expansion stroke is prolonged while maintaining plausible in-cylinder volumes, thus reaching a final expansion pressure  $p_5$  substantially higher than atmospheric pressure [16]. This can be practically pursued by adopting high engine compression ratio (CR) and early (or late) intake valve closure (IVC) to limit the compression stroke and exploit the full expansion stroke [11].

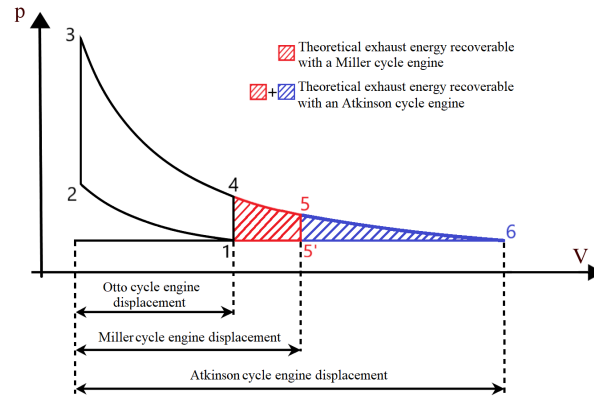


Figure 1 – Comparison between Otto, Miller, and Atkinson cycle

By adopting a CR of 14, an 8% theoretical efficiency increment can be obtained with respect to the Otto Cycle, to the detriment of power density, which is reduced by about 25%. A practical realization is represented by the Toyota Prius hybrid vehicle, which implements a Miller Cycle engine with a CR of 13, achieving a fuel economy improvement of 8.5% [17]. Although the implementation of the Miller Cycle can achieve good fuel consumption improvement, the resulting power density reduction still represents a crucial drawback that limits its advantage [18].

An alternative approach for recovery and utilization of the unexpanded gas energy is presented in this paper. A separated electric compound system is considered that involves an exhaust gas expander connected to an electric generator and a supercharger driven by an electric motor. The system proposed has the potential to increase the vehicle efficiency over a wide range of operating conditions.

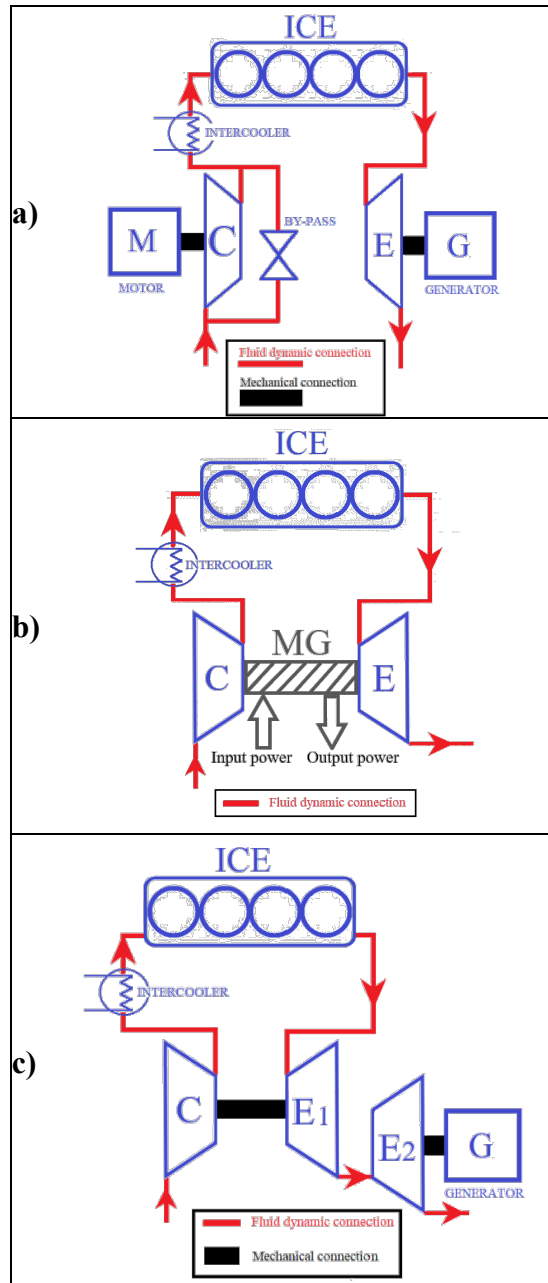


Figure 2 – Electric compound system [2]: a) Separated electric compound; b) High-pressure electric compound; c) Low-pressure electric compound (C=compressor; E=exhaust gas expander; M=electric motor; G=electric generator; ICE=internal combustion engine)

### Compound engine concept description

The powertrain architecture considered in this work is given as type (a) in Figure 2. Unlike other studied electric compound systems (types (b) and (c) in Figure 2), the two thermal machines (compressor *C* and expander *E*) operate independently in the system considered here. Each one is

connected to its own electric machine, and the exhaust gas expander operates at high pressure directly downstream of the thermal engine [2]. The compound system presented in this paper is hence composed of an electrically supercharged spark ignition (SI) engine whose exhaust gas flows through a properly designed exhaust gas expander (E) connected to an electrical generator (G). The system is specifically intended for a hybrid propulsion architecture application [22], where the onboard storage system may receive and store the energy produced by the expander-generator group. This energy can then be employed for vehicle propulsion. Moreover, in a hybrid propulsion system, the thermal engine is not involved throughout the wide and rapidly changing operating conditions of a traditional vehicle. Therefore, the exhaust gas expander could be used under quasi-steady conditions, and hence near-maximum expansion efficiency. For this reason, the compound engine (CE) considered in this paper is particularly suitable for applying to thermal-electric hybrid propulsion systems.

Figure 3 shows a possible hybrid propulsion layout that includes the compound engine concept. The net power produced by the expander-generator is summed to the power delivered by the thermal engine-generator (MG1) in the energy storage system, which in turn, supplies the second electric machine (MG2) and the motor-compressor employed for supercharging purposes. It is worth mentioning that in the system described, the expander-generator is always active in conjunction with the thermal engine, contributing to the energy balance of the whole vehicle. However, the motor-compressor unit is powered only when supercharging is required, i.e., only when higher engine loads are required.

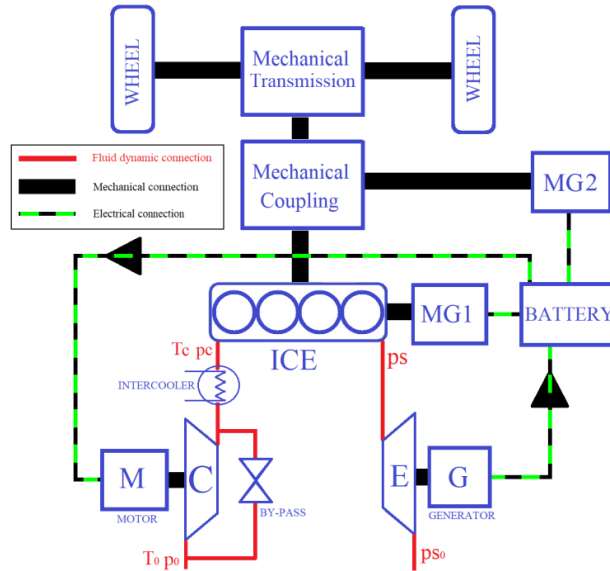


Figure 3 – Hybrid propulsion system endowed of the proposed compound engine (C=compressor; E=expander; ICE=internal combustion engine; M=electric motor; G=electric generator; MG1 and MG2=motor-generator units)

In this paper, the authors aim to evaluate the efficiency improvements obtainable by the proposed separated compound electric engine in comparison with a traditional turbocharged engine for hybrid vehicle applications. For this purpose, a performance comparison was conducted between the CE and a reference turbocharged engine through a simple modeling approach. Fundamentally, the approach consists of mass flow and power balance equations, accompanied by simplifying assumptions and relations. The theoretical approach was also supported by experimental data derived from relevant scientific literature or directly measured through experiments. Focusing on the hybrid vehicle application, the comparison considered only steady-state conditions and was carried out on an equal output power basis; all the propulsive units were sized for the same maximum output power of 73.5 kW (i.e., 100 HP).

The concept described here is novel, as there is no evidence of a study like the one presented here in the scientific literature. Although the separated electric CE has been cited in [23] among several other different systems that could be simulated, its advantages in terms of overall

efficiency were not evaluated, nor were its performances compared to a traditional internal combustion engine.

### **Baseline naturally aspirated engine**

To perform a fair comparison, the authors decided to evaluate the performance of both the electric CE and the comparative turbocharged engine starting from a common baseline naturally aspirated engine. As the first step, the steady-state performances of the baseline naturally aspirated engine were delineated. The authors employed the experimental data reported in [24], which were obtained on a gasoline VVT SI engine. Considering the application of the analyzed propulsion system to a plausible European Type C–Medium hybrid vehicle, the performances reported in [24] were adapted to a mid-level passenger car engine using a normalization procedure. For that purpose, the normalized mean piston speed  $u$  was employed:

$$u = \frac{u_m}{u_{m,\max}} \quad (0 \leq u \leq 1) \quad (1)$$

As a result, Figure 4 and Figure 5 show the resulting values of:

- 1) Brake mean effective pressure (BMEP)
- 2) Indicated mean effective pressure (IMEP)
- 3) Relative air-fuel ratio ( $\lambda$ )
- 4) Volumetric efficiency ( $\lambda_V$ )

as functions of the normalized mean piston speed and for the full load condition. Figure 6 presents the BMEP as a function of the manifold absolute pressure (MAP), which is the engine load control parameter for a fixed normalized mean piston speed.



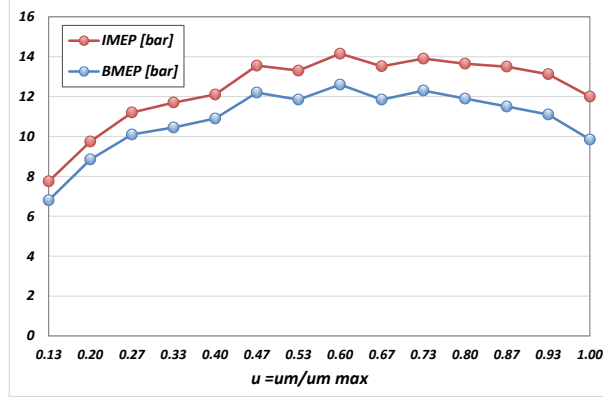


Figure 4 – Brake (BMEP) and indicated mean effective pressure (IMEP) as a function of the normalized mean piston speed at full load

Regarding the overall mechanical efficiency  $\eta_m$  is represented by Equation (2).

$$\eta_m = \frac{BMEP}{IMEP} = \frac{IMEP - FMEP}{IMEP} = 1 - \frac{FMEP}{IMEP} \quad (2)$$

The Chen-Flynn Model was followed in the calculation. The friction mean effective pressure (FMEP) was considered a function of the indicated mean effective pressure (IMEP), used here in place of the maximum in-cylinder pressure as the pressure-load related variable) and of the normalized mean piston speed  $u$ .

$$FMEP = A + B \cdot IMEP + C \cdot u + D \cdot u^2 \quad (3)$$

Therefore, the overall mechanical efficiency  $\eta_m$  is:

$$\eta_m = 1 - \frac{A + B \cdot IMEP + C \cdot u + D \cdot u^2}{IMEP} \quad (4)$$

The mechanical friction model A, B, C, and D parameters were determined using a least-squares regression performed employing the experimental BMEP and IMEP data of Figure 4 and Figure 6.

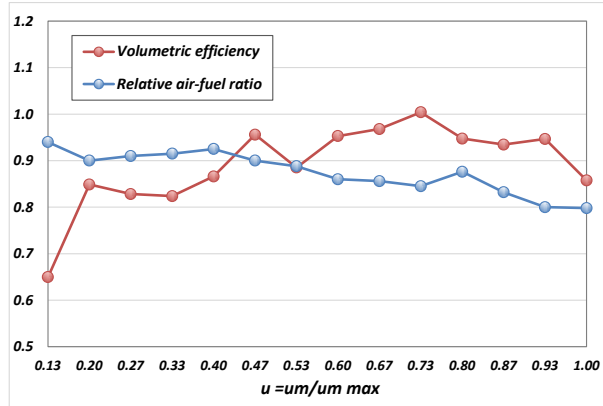


Figure 5 – Full load volumetric efficiency and relative air-fuel ratio as a function of the normalized mean piston speed

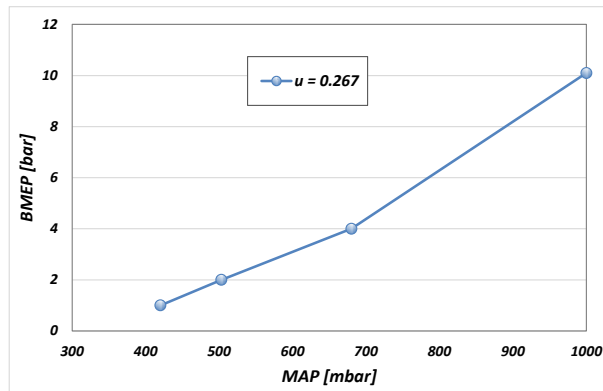


Figure 6 – Brake mean effective pressure as a function of manifold absolute pressure at  $u=0.267$

For the baseline naturally aspirated engine, the full load IMEP values reported in Figure 4 were entirely adopted while the BMEP value, for each speed and load, was obtained by the application of the FMEP model of the equation (3) using the parameters of Table 1:

$$BMEP = IMEP - FMEP \quad (5)$$

Concerning the air-fuel ratio, a stoichiometric mixture (i.e.,  $\lambda=1$ ) was assumed with MAP up to 0.9 bar, being the stoichiometric air-fuel ratio  $\alpha_{st}=14.7$  for gasoline; for the higher loads (i.e.,  $MAP>0.9$  bar), the usual air-fuel enrichment performed on gasoline SI engine to avoid knocking

phenomena was adopted, assuming a linear variation with MAP up to the full load values (already reported in Figure 5 for each normalized mean piston speed).

Table 1 – Parameters determined for the Chen & Flynn *FMEP* model

<b>A [bar]</b>	0.63043
<b>B [-]</b>	0.039807
<b>C [bar]</b>	-0.058036
<b>D [bar]</b>	1.05804

Taking into consideration the parameters which contribute to defining the *BMEP*:

$$BMEP = \frac{\delta_0 \cdot \lambda_v \cdot LHV}{\lambda \cdot \alpha_{st}} \cdot \eta_i \cdot \eta_m \quad (6)$$

and considering the manifold air density  $\delta_0$  at the ambient conditions  $p_0=1$  bar and  $T_0=288$  K ( $\Rightarrow \delta_0=1.209$  kg/m<sup>3</sup>) together with an average gasoline lower heating value *LHV* of 43 MJ/kg, it was possible to determine the engine indicated efficiency  $\eta_i$  for each load (i.e., *BMEP*) normalized mean piston speed, and the brake thermal efficiency  $\eta_b$ :

$$\eta_b = \eta_i \cdot \eta_m \quad (7)$$

as well as the brake specific fuel consumption *BSFC*:

$$BSFC_{[g/kWh]} = \frac{3600}{LHV_{[MJ/kg]} \cdot \eta_b} = \frac{\delta_{0[kg/m^3]} \cdot \lambda_v \cdot 36000}{BMEP_{[bar]} \cdot \lambda \cdot \alpha_{st}} \quad (8)$$

Given the full load *BMEP* curve shown in Figure 4, with a maximum value of 12.6 bar, the engine was sized considering the target output power of 73.5 kW. The required displacement  $V_A$  of the naturally aspirated baseline engine was 1352 cc. In line with current passenger car engines, a maximum mean piston speed of 17 m/s was adopted for each engine considered in this study:

this is equivalent to assuming that the different engines share the same technology level, hence the same mechanical performance.

Table 2 – Main characteristics of the gasoline SI baseline engine

<b>Engine</b>	4-stroke, naturally aspirated,
<b>Displacement</b>	1352 cc
<b>Number of</b>	4
<b>Bore</b>	72.9 mm
<b>Stroke</b>	80.9 mm
<b>Max mean piston</b>	17 m/s
<b>Compression</b>	11
<b>Injection system</b>	multi-point
<b>Valvetrain</b>	4 valves/cylinder,
<b>Max BMEP</b>	12.6 bar at 3780 rpm
<b>Max Power</b>	73.5 kW at 5880 rpm
<b>Min BSFC</b>	232.9 g/kWh

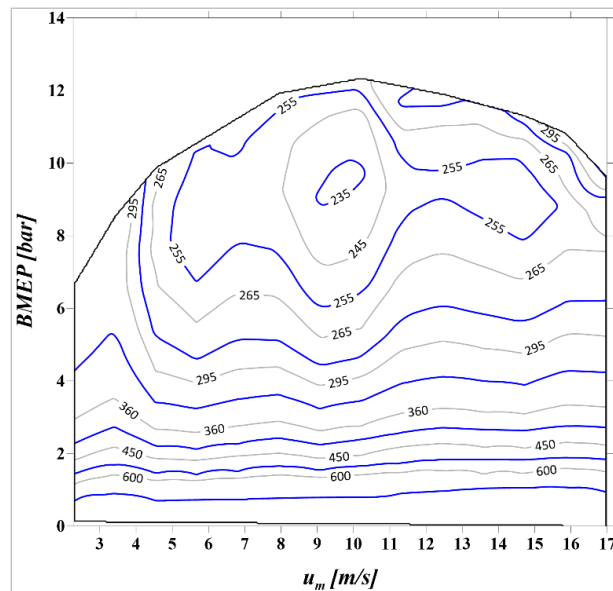


Figure 7 – Brake specific fuel consumption map of the baseline naturally aspirated engine

With values obtained for thermal efficiencies, the volumetric CR reported in [24] was assumed for the baseline engine. The value of 1.11 was assumed for the stroke-to-bore ratio for each engine model presented in this paper, as is the average and typical value adopted on current

engines employed in hybrid propulsion vehicles. For passenger car engines, the displacement of a single cylinder is typically approximately 0.5 L in a four cylinder layout. These dimensions were chosen for the baseline engine, whose main characteristics are presented in Table 2. The contour map of the BSFC obtained for the baseline SI engine is reported in Figure 7 as a function of mean piston speed  $u_m$  and BMEP.

### Reference turbocharged engine

As already mentioned, the advantages of implementing the separated electric compound SI engine were evaluated by comparing it with a reference traditional turbocharged engine. In this section, the authors describe the calculations carried out to determine the size and performance of the turbocharged reference engine, which is schematically represented in Figure 8. As can be observed, a waste-gate valve was considered for the turbine by-pass, and an intercooler between the engine and compressor was assumed to cool down the air charge. The BMEP of the turbocharged engine was evaluated based on the specific performance and parameters of the baseline naturally aspirated engine, delineated in the previous section, and assuming a maximum allowed boost pressure of 1.5 bar (absolute).

The specific performance of the turbocharged engine can be evaluated starting from the air mass flow to the engine  $G_C$ , which depends on the mean piston speed  $u_m$  and the manifold pressure  $MAP$ :

$$G_C = \frac{V_T \cdot n}{60 \cdot \varepsilon} \cdot \delta_c \cdot \lambda_{vc} \frac{\alpha}{\alpha + 1/\delta}, \quad (9)$$

where  $\delta_C$  is the air density in the manifold,  $\lambda_{V,C}$  is the turbocharged engine volumetric efficiency,  $n$  is the engine speed,  $V_T$  is the turbocharged engine displacement, and  $\delta'$  is the relative fuel density, i.e., the ratio between fuel and air density:

$$\delta' = \delta_F / \delta_A \quad (10)$$

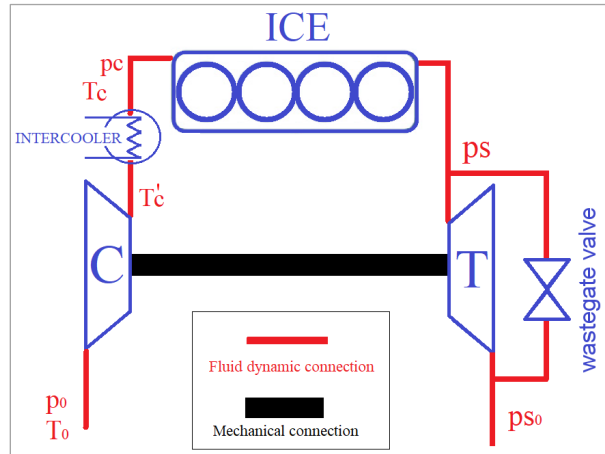


Figure 8 – Schematic representation of the turbocharged baseline engine (C=compressor; T=turbine; ICE=internal combustion engine)

The air density depends on the manifold absolute pressure  $MAP$  and the intercooler outlet temperature  $T_C$ :

$$\delta_C = \frac{MAP}{R' \cdot T_C} \quad (11)$$

In the calculation, gasoline was assumed to enter the cylinder with 40% of the mass already evaporated; this allowed evaluating the average fuel density  $\delta_F$  as:

$$\delta_F = \delta_{F,V} \cdot 0.4 + \delta_{F,L} \cdot 0.6 \quad (12)$$

where  $\delta_{F,L}$  and  $\delta_{F,V}$  are the fuel densities in the liquid and vapor phase, respectively. The latter was determined utilizing the perfect gas law adopting the molecular mass of 100 g/mol for the gasoline [27].

It is worth noting that the manifold absolute pressure (MAP) may differ from the compressor outlet pressure  $p_c$  due to the necessary gas throttling at part load. Assuming an intercooler efficiency  $R_{INT}$  of 0.7, the gas temperature at the intercooler outlet was evaluated as:

$$T_C' = T_0 \cdot \left( 1 + \frac{\beta_C^{\frac{k_c-1}{k_c}} - 1}{\eta_C} \right) \quad (13)$$

$$T_C = T_C' - R_{INT}(T_C' - T_0) \quad (14)$$

where the compressor outlet temperature  $T_C'$  was calculated in the equation (13) as a function of the compression ratio  $\beta_C = p_C/p_0$ , of the air isentropic coefficient  $k_c$  and compressor adiabatic efficiency  $\eta_C$  (whose evaluation is described later on). Due to the higher inlet temperatures caused by gas compression, the authors applied the reduction of the engine volumetric compression ratio  $\rho$  typically adopted in turbocharged or supercharged engines to avoid knocking. The reduction applied was based on literature data and considerations and the strategies adopted in modern SI engines. With 11 as the CR of the naturally aspirated engine, and assuming a maximum manifold pressure of 1.5 bar, the CR of the turbocharged engine ( $\rho'$ ) was plausibly assumed to be 10, which is in line with current turbocharged VVT engines.

The volumetric efficiency  $\lambda_{V,C}$  of the turbocharged engine was evaluated starting from the volumetric efficiency of the naturally aspirated engine  $\lambda_{V0}$  at the same mean piston speed  $u_m$ , and adjusted using two different corrections. The first one accounted for the pressure difference

variation between the inlet and exhaust. The second one was related to the inlet temperature increase (which is well known to cause a volumetric efficiency increase) due to compression.

The first correction is given as:

$$\frac{\lambda_v'(n)}{\lambda_v(n)} = 1 + \frac{MAP - p_s}{k \cdot MAP \cdot (\rho - 1)} \quad (15)$$

which is 1 if the pressure difference between inlet and exhaust is null. Considering  $p_{s0}$  as the reference exhaust backpressure of the baseline naturally aspirated engine, and taking into account the inlet to exhaust pressure difference variation due to the variation of both  $MAP$  and  $p_s$ , the authors considered, for each mean piston speed, the following correction factor:

$$\frac{\lambda_v(u_m)}{\lambda_{v0}(u_m)} = \frac{1 + \frac{MAP - p_s}{k \cdot MAP \cdot (\rho - 1)}}{1 + \frac{p_0 - p_{s0}}{k \cdot p_0 \cdot (\rho - 1)}} \quad (16)$$

where  $\lambda_{v0}$  is the baseline naturally aspirated engine volumetric efficiency, corresponding to the manifold pressure  $p_0$  and the exhaust pressure  $p_{s0}$  (here assumed  $=1.06 p_0$ ). As for the second correction on the volumetric efficiency, i.e., due to the increased inlet temperature, the authors followed the widely adopted relation:

$$\frac{\lambda_v(u_m)}{\lambda_{v0}(u_m)} = \sqrt{\frac{T_C}{T_0}} \quad (17)$$

As a final result, the volumetric efficiency  $\lambda_{v,C}$  of the turbocharged engine was evaluated as:

$$\lambda_{v,C}(u_m) = \lambda_{v0}(u_m) \cdot \sqrt{\frac{T_C}{T_0}} \cdot \left[ \frac{1 + \frac{MAP - p_s}{k \cdot MAP \cdot (\rho' - 1)}}{1 + \frac{p_0 - p_{s0}}{k \cdot p_0 \cdot (\rho - 1)}} \right] \quad (18)$$



The power balance between turbocompressor and turbine can be resumed as:

$$P_{comp} = P_{turb} \quad (19)$$

where  $P_{comp}$  is the power required by the turbocompressor:

$$P_{comp} = G_C \cdot cp_c \cdot \frac{T_0}{\eta_c} \cdot \left( \beta_c^{\frac{k_c-1}{k_c}} - 1 \right) \quad (20)$$

and  $P_{turb}$  is the power delivered by the turbine:

$$P_{turb} = G_T \cdot cp_s \cdot \eta_t \cdot T_s \cdot \left( 1 - \beta_s^{\frac{1-k_s}{k_s}} \right) \quad (21)$$

where  $G_C$  and  $G_T$  are the gas mass flow in the compressor and the turbine, respectively,  $T_0$  and  $T_s$  are the gas temperatures at compressor and turbine inlet,  $\eta_t$  is the turbine efficiency (whose evaluation is described further on),  $cp_c$  and  $cp_s$  are the specific heats at a constant pressure of fresh air and exhaust gas respectively, and  $\beta_s = p_s/p_{s0}$  is the pressure ratio across the turbine, with  $p_s$  being the gas pressure in the exhaust manifold of the engine. A relation exists between the turbine mass flow  $G_T$  and the compressor mass flow  $G_C$ :

$$G_T = \Omega \cdot G_C \frac{\alpha + 1}{\alpha} \quad (22)$$

where the ratio  $[(\alpha+1)/\alpha]$  accounts for the fuel mass flow, while  $\Omega$  represents the fraction of exhaust gas flowing in the turbine, with the rest by-passed by the waste-gate valve if the boosting pressure  $p_c$  tends to exceed the maximum allowed value (usually is  $0.4 \leq \Omega \leq 1$ ). The turbocharging compression ratio  $\beta_C$  can be evaluated through the power balance of the equation (19):

$$\beta_C = \left[ 1 + \Omega \cdot \frac{\alpha + 1}{\alpha} \cdot \frac{cp_s}{cp_C} \cdot \frac{T_s}{T_0} \cdot \eta_r \cdot \eta_C \cdot \left( 1 - \frac{1}{\beta_S^{k_s}} \right) \right]^{\frac{k_c}{k_c - 1}} \quad (23)$$

In the calculation performed, the parameter  $\Omega$  was adaptively reduced (which corresponds to increasing the waste-gate opening) with the aim to prohibit the compression ratio  $\beta_C$  exceeding the maximum allowed value of 1.5. The temperature of the exhaust gas at pressure  $p_s$  was evaluated employing a simple yet effective commonly used correlation:

$$T_s = T_{1,C} \cdot \frac{p_s}{MAP} \cdot \frac{(k_s - 1)}{k_s} + \frac{T_{4,C}}{T_{1,C}} \cdot \frac{1}{k_s} \quad (24)$$

where  $k_s$  is the isentropic coefficient of the exhaust gas,  $T_{1,C}$  is the inlet temperature at intake valve closure (IVC), here assumed equal to the gas temperature at the intercooler outlet  $T_C$ , while  $T_{4,C}$  is the gas temperature inside the cylinder when the exhaust valves open (EVO). The ratio  $T_{4,C}/T_{1,C}$  plays an important role in determining the exhaust gas temperature from Equation (24). It depends on the physical characteristics of the particular engine and usually varies with changing engine speed and load. For gasoline fuelled SI engines, it ranges between 3.5 and 4.5. Given the simple approach followed by the authors in this paper, in the calculation performed, the temperature ratio  $T_{4,C}/T_{1,C}$  was supposed to remain constant, apart from engine speed and load variation. However, with the aim to ascertain the importance of its role, the calculations were repeated for three different values of  $T_{4,C}/T_{1,C}$ , namely 3.5, 4.0, and 4.5.

The isentropic coefficient  $k_s$  in the equation (24) was computed as:

$$k_s = \frac{c_{p,s}(T_s)}{c_{v,s}(T_s)} \quad c_{v,s}(T_s) = c_{p,s}(T_s) - R_s \quad (25)$$

where  $c_{p,s}$  and  $c_{v,s}$  are the burned gas specific heats at constant pressure and constant volume, respectively, both evaluated at the exhaust gas temperature  $T_s$ , while  $R_s'$  represents the perfect gas law constant. Both  $c_{p,s}$  and  $R_s'$  were calculated as weighted averages based on the burned gas composition, i.e.:

$$c_{p,s} = c_{p,CO_2}(T_s) \cdot x_{CO_2} + c_{p,H_2O}(T_s) \cdot x_{H_2O} + c_{p,N_2}(T_s) \cdot x_{N_2} + c_{p,CO}(T_s) \cdot x_{CO} \quad (26)$$

$$R'_s = R'_{CO_2} \cdot x_{CO_2} + R'_{H_2O} \cdot x_{H_2O} + R'_{N_2} \cdot x_{N_2} + R'_{CO} \cdot x_{CO} \quad (27)$$

where  $x$  is the mass fraction of the generic chemical species. For that purpose, to remain as close as possible to gasoline use, the combustion of a surrogate hydrocarbon with H/C=1.87 was considered for the calculation of each mass concentration [27]. The last terms of both equations (26) and (27) refer to the presence of carbon monoxide, which was taken into consideration only when rich air-fuel mixtures were supposed to be employed. For each chemical species, specific heat at constant pressure was computed as a function of the exhaust gas temperature  $T_s$  using the Shomate equations and coefficients available on the NIST Chemistry WebBook [28].

The system of equations describing the turbocharger power and mass flow balances is completed by the characteristic turbine curve, which correlates the mass flow to the pressure drop and takes into account the turbine swallowing capacity. Following a simplifying approach, a single curve was adopted to describe the mass flow parameter (MFP) of the turbine in place of several curves at different rotation speeds. A suitable mathematical expression was found to faithfully reproduce the typical trend of the MFP as a function of the pressure ratio  $\beta_s$ :

$$MFP = \frac{a \cdot b + c \cdot \beta_s^d}{b + \beta_s^d} \quad (28)$$

where the model parameters  $a$ ,  $b$ ,  $c$ , and  $d$  were established employing data fitting performed on the characteristics of a real commercial product (IHI RHF3). Since the turbine mass flow  $G_T$  depends on the MFP and the pressure and temperature conditions at the turbine inlet:

$$G_T = MFP \frac{P_s}{\sqrt{T_s}} \quad (29)$$

the MFP values obtained by the equation (28) were reduced or amplified, thus sizing the turbine to the engine displacement and the parameters resulting from calculations. For example, in Figure 9, the turbine MFP is represented as a function of the pressure ratio  $\beta_s$ , by a solid curve, while an amplified and a reduced MFP are reported as dashed curves.

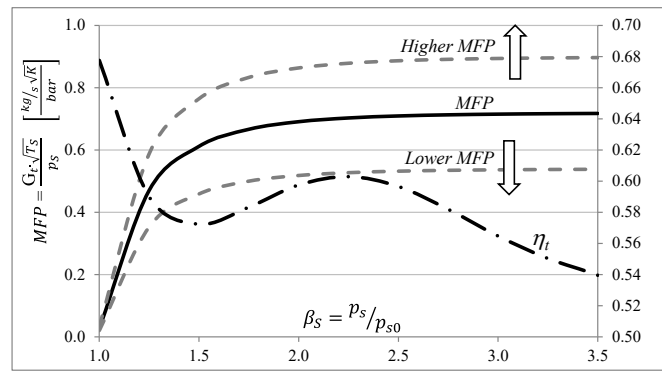


Figure 9 – Performance characteristics of the exhaust gas turbine (efficiency and mass flow parameter as a function of pressure ratio)

A least square regression, performed on the data available for the already mentioned commercial turbine, allowed determination of a polynomial curve expressing the efficiency as a function of the pressure ratio  $\beta_s$  with a maximum error of 9%:

$$\eta_T = -\frac{292.1}{10^4} \beta_s^5 + \frac{3933}{10^4} \beta_s^4 - 2.049 \beta_s^3 + 5.113 \beta_s^2 - 6.072 \beta_s + 3.321 \quad (30)$$

The resulting efficiency ( $\eta_t$ ) curve is also reported in Figure 9 as a dash-dot curve. It is worth highlighting that the efficiency represented in Figure 9 already accounts for the bearing frictional

losses, as indicated by the producer of the commercial turbine. Also, in the case of the turbocompressor, the authors performed an iterative sizing procedure based on the performance map of a commercial unit (IHI-RHF3). To fit a turbocompressor to the engine (whose displacement varies during the whole calculation process), the minimum and maximum values on both axes of the performance map (Figure 10) were altered with a double task: 1) maintain the operative turbocharging points (expressed by the two coordinates mass flow  $G_C$  and compression ratio  $\beta_C$ ) within the limits of the contour map, and 2) exploit the best compressor efficiency.

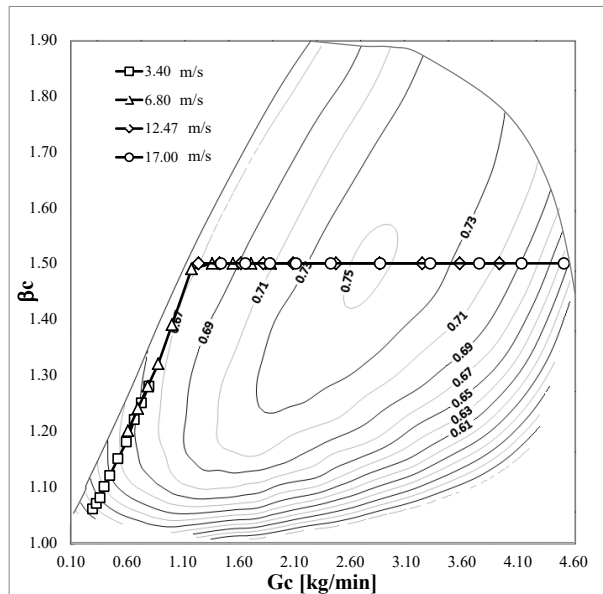


Figure 10 – Turbocharger performance map with reported operative points obtained for every engine load and four different mean piston speeds

As a result, Figure 10 shows the turbocharging points required by the engine at four different mean piston speeds. The compressor efficiency  $\eta_C$  was deduced from the points' position on the previously digitized contour map. This iterative adaptation of both turbocharger elements represents the selection process usually followed to adequately fit the right turbocharger to the particular engine. Solving the system of equations recursively from (13) to (30) for each mean piston speed  $u_m$  (from 2.27 to 17 m/s) and manifold pressure MAP (from 0.6 bar up to the maximum allowed value of 1.5 bar) allows calculating the turbocharger performance.

Once the solution for the turbocharger is obtained, the thermodynamic condition of the gas at the engine inlet and outlet can be calculated, which in turn allows evaluating the engine performance. To this purpose, the authors made the simplifying assumption that two similar engines, with the same technology level, running at the same mean piston speed and in full load condition, are characterized by the same gross indicated efficiency, even with different manifold absolute pressures. On account of this assumption, the gross indicated efficiency of the turbocharged engine  $\eta_{ig,c}$  was evaluated based on the gross indicated efficiency of the naturally aspirated engine  $\eta_{ig}$  for the same normalized MAP value and same mean piston speed, and finally corrected to account for the different engine compression ratio. Defining the normalized MAP as:

$$\phi = \frac{MAP}{MAP_{\max}} \quad (31)$$

the simplifying assumption gives:

$$\eta_{ig,c}(\phi, u_m) = \eta_{ig}(\phi, u_m) \cdot f_{CR} \quad (32)$$

where  $f_{CR}$  is the correction factor used to account for the different CR between turbocharged and naturally aspirated engines, calculated through the basic theory of the ideal Otto cycle:

$$f_{CR} = \frac{\eta_{Otto,c}}{\eta_{Otto}} = \frac{1 - \frac{1}{\rho^{k-1}}}{1 - \frac{1}{\rho'^{k-1}}} \quad (33)$$

The gross indicated mean effective pressure of the turbocharged engine was hence evaluated as:

$$IMEP_{g,c} = \frac{\delta_c \cdot \lambda_{v,c} \cdot LHV}{\alpha + 1/\delta'} \cdot \eta_{ig,c} \quad (34)$$

and the resulting net indicated mean effective pressure as:

$$IMEP_C = IMEP_{g,c} + PMEP_C \quad (35)$$

where the pumping mean effective pressure  $PMEP_C$  was simply evaluated as:

$$PMEP_C = MAP - ps \quad (36)$$

The friction mean effective pressure was computed employing the same equation (3) used for the naturally aspirated engine; the turbocharged engine brake mean effective pressure  $BMEP_C$  could be hence calculated together with the related brake specific fuel consumption  $BSFC_C$ :

$$BMEP_C = IMEP_C + FMEP_C \quad (37)$$

$$BSFC_C = \frac{\delta_c \cdot \lambda_{\nu,c}}{BMEP_C \cdot \left(\alpha + \frac{1}{\delta'}\right)} \quad (38)$$

The maximum  $BMEP_C$  calculated for the turbocharged engine was used to determine the displacement  $V_T$  necessary to deliver the required output power of 73.5 kW. As mentioned before, the entire calculation was repeated considering the three different values of the temperature ratio  $T_{4,C}/T_{1,C}$ . The main characteristics and performance of the resulting reference turbocharged engine are summarized in Table 3, while Figure 11 reports the brake specific fuel consumption contour map obtained for the reference turbocharged engine with  $T_{4,C}/T_{1,C}=4$ .

Table 3 - Main characteristics of the turbocharged engine

<b>Engine</b>	<b>4-stroke, spark ignition</b>		
<b>Injection</b>	Gasoline multi-point injection		
<b>Valvetrain</b>	4 valves/cylinder, VVT		
<b>Compression</b>	10		
<b>Max boost</b>	1.5 bar		
$T_{4,C}/T_{1,C}$	3.5	4.0	4.5
<b>Displacement</b>	1058 cc	1006 cc	968 cc
<b>Number of</b>	3	3	3

<b>Bore</b>	74.0 mm	72.7 mm	71.8 mm
<b>Stroke</b>	82.1 mm	80.7 mm	79.7
<b>Max BMEP</b>	17.62 bar at 3728	17.88 bar at 3790	18.09 bar at 3840
<b>Min BSFC</b>	232.0 g/kWh	230.2 g/kWh	228.9 g/kWh

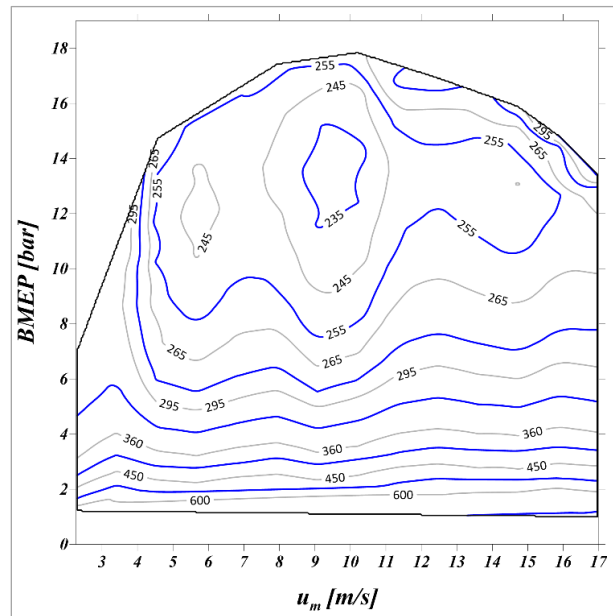


Figure 11 – Brake specific fuel consumption [g/kWh] of the turbocharged engine ( $T_{4,C}/T_C=4$ )

### Separated electric compound spark-ignition engine

This section presents the procedure followed to evaluate the performance of the proposed CE and the resulting comparison with the traditional reference turbocharged engine carried out in a hybrid propulsion architecture. Figure 12 provides a schematic representation of the compound system, composed of a SI engine, an electric driven supercharger, and an expander-generator group to complete the exhaust gas expansion. Unlike the traditional turbocharged engine, the compressor is not connected to the expander but is driven by the electric motor, which, in turn, is powered by the same energy storage system (e.g., the batteries of the hybrid vehicle) which receives the power produced by the expander-generator. Furthermore, the management system is assumed to control the rotation speed of the motor-compressor to increase the air pressure only



when needed (i.e., when MAP values higher than 1 bar are required) and reduce its power absorption for the part-load operation (i.e., when  $MAP < 1$  bar). As illustrated in Figure 12, a by-pass valve lets the air flow to the engine in this condition.

The expander-generator group is considered permanently active, thus continuously recovering the maximum possible power from the exhaust gas. Regarding the compressor, a turbocompressor similar to the one considered for the turbocharged engine was employed. Hence, its efficiency was evaluated following the same calculation (and sizing) procedure described in the previous section. However, other opportunities may be considered, such as using a Roots-type or a screw compressor, which are commonly employed for engine supercharging. Regarding the exhaust gas expander represented in Figure 12, it must be pointed out that such a machine is not currently available on the market. This machine is substantially different from turbines commonly used for turbocharging purposes. These turbines, in effect, consist of a single radial stage designed to deliver sufficient power to drive the turbocompressor. They usually function under wide variations of speed and mass flow, and as a result, are not optimized for steady-state operation.

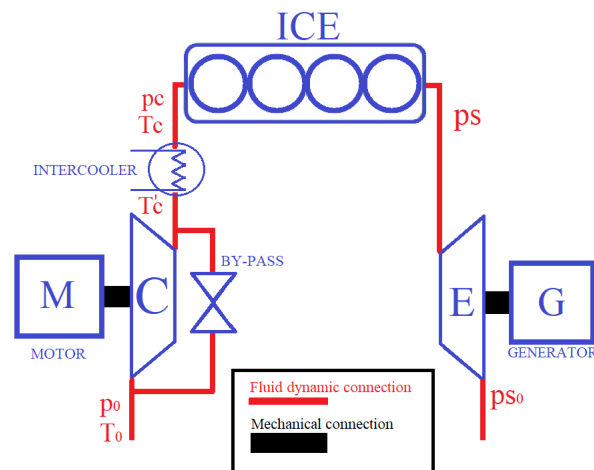


Figure 12 – Schematic representation of the separated electric compound engine (C=compressor; E=expander; ICE=internal combustion engine; M=electric motor; G=electric generator)

In contrast to a conventional powertrain, the exhaust gas expander to be employed in the CE considered here is involved in hybrid propulsion application and should function under almost steady-state conditions. Moreover, the generator's torque/current control would let the expander run at its best efficiency speed ratio, independently from the power produced. As a result, the exhaust gas expander considered in this work should be composed of two or more stages, optimized for power production, and deliver power up to 16 kW (as shown further on) with efficiency higher than common turbocharging turbines. The only products already available on the market, or studied up to now, [19][20][21], consist of radial turbines derived from turbocharging, connected to electric generators, and characterized by limited power (6 kW), being designed only to supply the vehicle electric accessories. According to the previous considerations, the authors reasoned that, apart from the power produced, the exhaust gas expander should work with almost unchanged speed ratio and, hence, with constant efficiency  $\eta_E$ . With the aim to ascertain the effect of the expander efficiency on the overall energetic performance of the CE, two different efficiency levels were considered, 0.70 and 0.75. As already explained, the two assumed efficiency values are significantly higher than in a common turbocharging turbine under the assumption that the exhaust gas expander should be a machine conceived and optimized for steady-state power production.

The performance of the CE concept was evaluated at the same mean piston speed  $u_m$  (from 2.27 to 17 m/s) and  $MAP$  values (from 0.6 bar to the maximum allowed 1.5 bar) as the turbocharged engine. The inlet air density  $\delta_C$  was evaluated using equations (11), (13) and (14). The gross indicated mean effective pressure of the supercharged engine is given as:

$$IMEP_{g,C} = \frac{\delta_C \cdot \lambda_{v,C} \cdot LHV}{\alpha + 1/\delta'} \cdot \eta_{ig,C} \quad (39)$$

The  $IMEP_{g,C}$  was then evaluated after determining both indicated gross efficiency  $\eta_{ig,c}$  and volumetric efficiency  $\lambda_{v,c}$ .

It can be observed that the use of the exhaust gas expander produces an increase in the exhaust gas back pressure  $p_s$ . The effect is stronger than in a common turbocharged engine for at least two reasons. First of all, the expander-generator group is always active, even at partial load operation, when the engine manifold absolute pressure is lower than 1 bar. Second, unlike the turbocharged system, where part of the exhaust mass flow by-passes the turbine through the waste-gate valve, in the system proposed, the task of the expander is to exploit the maximum available power, working with the whole exhaust mass flow and with sufficiently high-pressure ratio  $\beta_s = p_s/p_{s0}$ .

It is evident that changing the exhaust backpressure may have repercussions on both the volumetric efficiency and the indicated efficiency of the engine. Increasing the exhaust pressure causes a small increase in the amount of in-cylinder residual gas, resulting in a reduction of the entrapped fresh charge and, hence, reduced engine volumetric efficiency. Moreover, with the flame propagation speed and combustion efficiency strongly influenced by fresh charge dilution with residual gas, an exhaust pressure increase could easily compromise engine indicated efficiency. Based on these considerations, the effect produced by the exhaust pressure increase was carefully considered. Regarding the volumetric efficiency, Equation (18) was used to account for the pressure difference effect between the intake and the exhaust. Concerning the second effect, i.e., the indicated efficiency worsening due to the exhaust pressure increase, no useful reference could be found in the scientific literature.

To compensate for lack of available literature on the effect of exhaust back pressure, the authors executed a dedicated series of experimental tests on a SI engine test bench, with the aim to

correlate the in-cylinder residual gas fraction (RGF) increment to the indicated efficiency deterioration. For that purpose, a throttle valve was installed in the exhaust duct of a four-cylinder multi-point 1.2 L SI engine and used to modulate the exhaust backpressure. In-cylinder pressure was measured using an AVL GU13X piezoelectric pressure sensor flush-mounted in the combustion chamber and sampled with the resolution of 1 CAD, together with air and fuel mass flows, manifold absolute pressure, and engine torque. A more detailed description of the engine test bed employed is reported in [25] and [26].

Table 4 summarizes the operating conditions of the experiment expressly performed to determine the relationship between the exhaust pressure increase and the indicated efficiency deterioration. For each tested engine speed, a 0.1 bar increment of exhaust pressure was imposed until heavy combustion instability was found up to a maximum of 2 bar. The tests were performed employing CNG as fuel in place of gasoline. It was assumed that the particular fuel employed had a negligible influence on the relation between indicated efficiency variation and exhaust pressure variation.

Table 4 – Operating conditions of the experimental test

<b>Engine speed</b>	1500-2500-3500
<b>Engine load</b>	Full
<b>MAP</b>	1.00
<b>Exhaust pressure <math>p_s</math></b>	1.0 to 2.0 in steps of
<b>Fuel</b>	CNG
<b>Spark advance</b>	Optimal
<b>Air-fuel ratio</b>	Stoichiometric

As shown in Figure 13, the results of the tests performed confirmed that an exhaust pressure increase (with constant MAP) reduces the gross indicated thermal efficiency of the engine due to the increased residual gas fraction (i.e., the ratio between the residual gas mass and the total in-cylinder mass). The experimental measurements revealed that the variation of the gross indicated

efficiency can be expressed as a function of the residual gas fraction increment using the relation:

$$\frac{\eta_{i,g,C}}{\eta_{i,g,C,0}} = \frac{1}{1 + b \cdot x^\gamma} \quad (40)$$

$$b = 0.1384 \cdot u_m^{-2.635} \quad \gamma = -0.2556 \cdot u_m + 6.0391$$

where  $\eta_{i,g,C,0}$  is the reference gross indicated efficiency of the engine (i.e., without throttling the exhaust duct),  $x = RGF / RGF_0$  is the ratio between the RGF measured with and without exhaust throttling (whose evaluation is described in **Appendix A**), while the parameters  $b$  and  $\gamma$  are correlated to the mean piston speed  $u_m$ , which, as shown in Figure 13, has a strong influence on the efficiency variation.

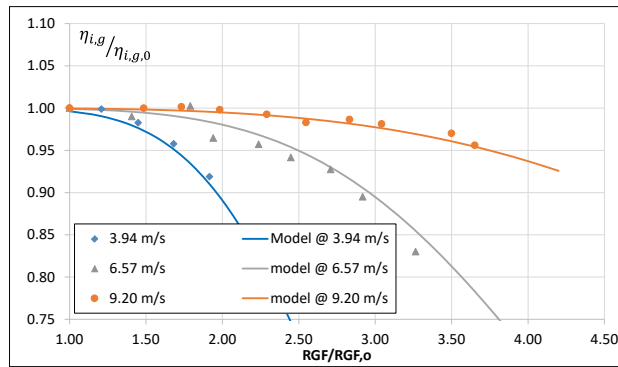


Figure 13 – Gross IMEP variation as a function of the RGF increment

The performance of the supercharged engine was evaluated compared to the baseline engine according to the mentioned assumption that the same technology level produces equal specific performances unless modification or parameters variations occur. The gross reference indicated that the efficiency  $\eta_{i,g,C,0}$  of the engine (i.e., without throttling the exhaust duct) was evaluated according to the same assumption made for the turbocharged reference engine, i.e., using equations (31)(32) and (33), where the engine CR was considered reduced to 10 also in this case:

$$\eta_{i,g,C,0}(\phi, u_m) = \eta_{ig}(\phi, u_m) \cdot f_{CR} \quad (41)$$

The gross indicated efficiency  $\eta_{i,g,c}$  of the supercharged engine was hence calculated by means of equations (40) and (41) for each  $MAP$  and mean piston speed. Hence, the net indicated mean effective pressure  $IMEP_C$  could be computed as:

$$IMEP_C = IMEP_{g,c} + PMEP_C \quad (42)$$

where the pumping mean effective pressure  $PMEP_C$  was obtained by equation (36). Equation (3) and equation (37) were then employed to evaluate the  $FMEP_C$  and the engine  $BMEP_C$ , respectively.

The overall specific output of the compound system  $BMEP_{TOT}$  is composed by the engine  $BMEP_C$ , with the addition of the specific output of the expander-generator group (here called recovery mean equivalent pressure  $RMEP$ ), and the reduction due to the specific power required by the motor-compressor (here called compressor mean equivalent pressure  $CMEP$ ):

$$BMEP_{tot} = BMEP_C + RMEP - CMEP \quad (43)$$

The recovery mean effective pressure is clearly related to the power recovered by the expander  $P_{exp}$ :

$$RMEP = \frac{60 \cdot \varepsilon \cdot P_{exp}}{V_C \cdot n} \quad (44)$$

where  $\varepsilon$  is the number of revolutions per cycle (2 for a 4 stroke engine), and  $V_C$  is the supercharged engine displacement. Similarly, the compressor mean effective pressure is related to the power required by the compressor  $P_{comp}$ :

$$CMEP = \frac{60 \cdot \varepsilon \cdot P_{comp}}{V_C \cdot n \cdot \eta_{EM}} \quad (45)$$

where  $\eta_{EM}$  is the electric motor efficiency, considered here since the motor-compressor represents an ancillary device that burdens the engine's energy balance. According to equation (44), the generator's efficiency was not considered in the *RMEP* calculation coherently with the evaluation of the power produced by the engine, which was not reduced by the efficiency of the electrical machine connected in the generic hybrid propulsion system. The power required by the compressor has the same formulation given in equation (20), while, similarly to the power delivered by the turbine of equation (21), the power produced by the expander  $P_{exp}$  can be expressed as:

$$P_{exp} = G_C \cdot \frac{\alpha + 1}{\alpha} \cdot c_{p_s} \cdot \eta_E \cdot T_S \cdot \left( 1 - \beta_S^{\frac{1-k_s}{k_s}} \right) \quad (46)$$

where  $T_S$  is the exhaust gas temperature at the expander inlet (evaluated using equation (24)),  $\alpha$  is the air-fuel ratio,  $G_C$  is the air mass flow to the engine,  $\eta_E$  is the expander efficiency,  $\beta_S = p_s/p_{s0}$  is the pressure ratio across the expander,  $k_s$  and  $c_{p_s}$  are the isentropic coefficients and the constant pressure specific heat of the exhaust gas, both evaluated at the temperature  $T_S$ , as already described through equations (25) (26) and (27). Given the air mass flow to the engine:

$$G_C = \frac{V_T \cdot n}{60 \cdot \varepsilon} \cdot \delta_C \cdot \lambda_{vC} \frac{\alpha}{\alpha + 1/\delta} \quad (47)$$

*RMEP* and *CMEP* become:

$$RMEP = \delta_C \cdot \lambda_{v,C} \cdot \frac{\alpha + 1}{\alpha} \cdot c_{p,s} \cdot T_s \cdot \eta_E \cdot \left( 1 - \beta_S^{\frac{1-k_s}{k_s}} \right) \quad (48)$$

$$CMEP = \frac{\delta_C \cdot \lambda_{v,C} \cdot c_{p_c} \cdot T_0}{\eta_{EM} \cdot \eta_C} \cdot \left( \beta_c^{\frac{k_c-1}{k_c}} - 1 \right) \quad (49)$$

For each mean piston speed  $u_m$  and manifold absolute pressure  $MAP$ , the overall brake thermal efficiency  $\eta_{bTOT}$  of the proposed compound system is:

$$\begin{aligned}\eta_{bTOT} &= \frac{BMEP_{TOT} \cdot \left(\alpha + \frac{1}{\delta'}\right)}{\delta_c \cdot \lambda_{v,c} \cdot LHV} = \\ &= \frac{(BMEP + RMEP - CMEP) \cdot \left(\alpha + \frac{1}{\delta'}\right)}{\delta_c \cdot \lambda_{v,c} \cdot LHV}\end{aligned}\quad (50)$$

It is worth noting that, for each required output power, or, which is the same, for each  $BMEP_{TOT}$ , the overall efficiency of the considered compound system depends on the exhaust pressure; on the one hand, boosting the exhaust pressure produces higher  $RMEP$ , on the other hand, it causes  $PMEP$  increments (equation (36)) and indicates efficiency reduction (equation (40)) due to the increased amount of residual gas mass, and, as a final result, a decrease of engine  $BMEP$ . Therefore, for each power level (or  $BMEP_{TOT}$ ), a compromise exists between the advantages and disadvantages produced by the exhaust pressure increase. The optimal exhaust pressure value was determined to be the value corresponding to the maximum overall brake thermal efficiency  $\eta_{bTOT}$  for each engine load ( $MAP$ ) and speed ( $u_m$ ). A genetic algorithm was employed for the optimization process since  $\eta_{bTOT}$  is not a linear or polynomial function of the exhaust pressure. This procedure was repeated for each considered temperature ratio  $T_{4,C}/T_{1,C}$  (3.5, 4.0, and 4.5) and for each expander efficiency  $\eta_E$  (0.70 and 0.75). For example, Figure 14 and Figure 15 report the contour maps of the optimal exhaust pressure levels determined for each overall load and mean piston speed for the two extreme cases  $T_{4,C}/T_{1,C} = 3.5 - \eta_E = 0.70$  and  $T_{4,C}/T_{1,C} = 4.5 - \eta_E = 0.75$ .



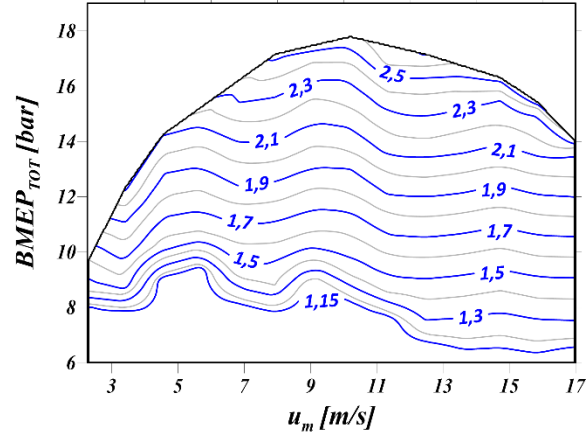


Figure 14 – Optimal exhaust pressure levels as function of load and speed ( $\eta_E=0.7$ ,  $T_{4,c}/T_{1,c}=3.5$ )  
 It can be noted that, for the higher power level, the optimal exhaust pressure was estimated to range between 2.6 and 3.3 bar: this requirement should be adequately considered in the optimal design of the exhaust gas expander. It was also observed that the optimal exhaust pressure increased when considering higher temperature ratio  $T_{4,c}/T_{1,c}$  or higher expander efficiency  $\eta_E$ .

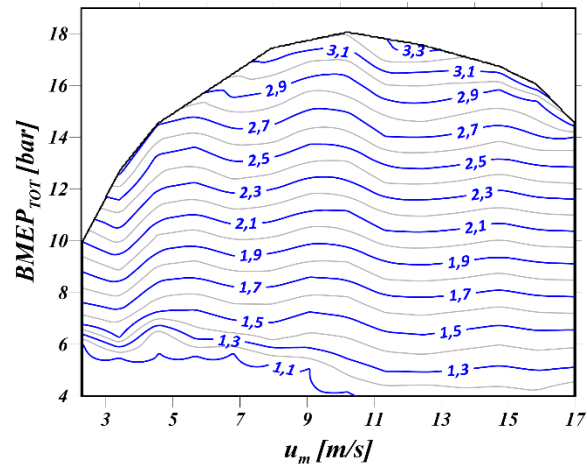


Figure 15 – Optimal exhaust pressure levels as function of load and speed ( $\eta_E=0.75$ ,  $T_{4,c}/T_{1,c}=4.5$ )

The overall brake specific fuel consumption of the compound engine  $BSFC_{TOT}$  was evaluated from the overall brake mean effective pressure:

$$BSFC_{TOT} = \frac{\delta_c \cdot \lambda_{v,c}}{BMEP_{TOT} \cdot \left(\alpha + \frac{1}{\delta'}\right)} \quad (51)$$

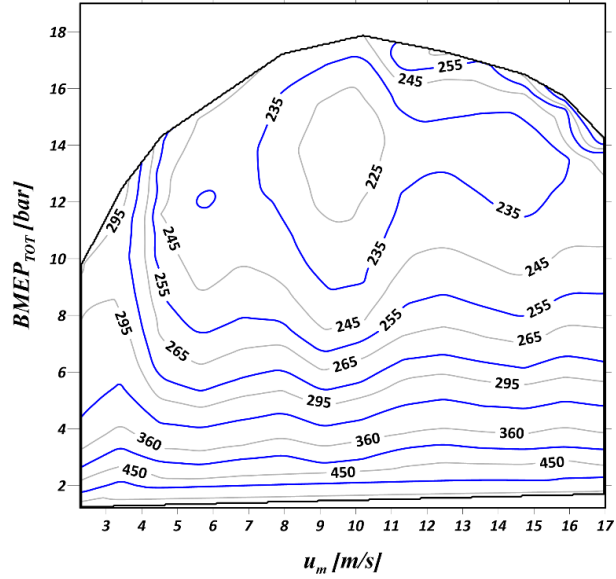


Figure 16 – Brake specific fuel consumption [g/kWh] of the CE, as a function of overall load and mean piston speed ( $T_{4,C}/T_{1,C}=4$  and  $\eta_E=0.7$ )

Figure 16 represents the contour map of the specific fuel consumption obtained for the case  $T_{4,C}/T_{1,C}=4$  and  $\eta_E=0.7$ , which can be compared to the map traced for the reference turbocharged engine with the same temperature ratio  $T_{4,C}/T_{1,C}$  (shown in Figure 11). It can be observed that the proposed compound system exhibits better fuel economy even though the two propulsive solutions obtained similar BMEP. Once the optimal exhaust pressure levels were determined, the maximum value of the  $BMEP_{TOT}$  allowed determining the engine displacement  $V_C$  necessary to develop the target power of 73.5 kW. Table 5 summarizes the main characteristics of the CE together with some performance parameters for each value adopted for the temperature ratio  $T_{4,C}/T_{1,C}$ , and considering the expander efficiency of 0.70.

Table 5 – Main characteristic of the CE obtained with  $\eta_E=0.70$

<b>Engine</b>	<b>4-stroke, spark ignition</b>		
<b>Injection system</b>	Gasoline multi-point injection		
<b>Valvetrain</b>	4 valves/cylinder, VVT		
<b>Compression</b>	10		
<b>Max boost</b>	1.5 bar		
<b>Expander</b>	0.7		
$T_{4,C}/T_{1,C}$	3.5	4.0	4.5

<b>Displacement</b>	950 cc	923 cc	905 cc
<b>Number of</b>	3	3	3
<b>Bore</b>	71.3 mm	70.7 mm	70.2 mm
<b>Stroke</b>	79.2 mm	78.4 mm	77.9 mm
<b>Max BMEP<sub>TOT</sub></b>	17.83 bar at 3864	17.90 bar at 3900	18.00 bar at 3926
<b>Min BSFC<sub>TOT</sub></b>	220.2 g/kWh	216.6 g/kWh	212.7 g/kWh
<b>Variation of min BSFC</b>	-5.1%	-5.9%	-7.1%
<b>Max RMEP</b>	2.90 bar at 4722	3.52 bar at 4768	4.16 bar at 4799
<b>Max RMEP/BMEP<sub>TOT</sub></b>	19%	23.9%	29.2%

In comparison, Table 6 summarizes the results obtained considering an expander efficiency of 0.75.

Table 6 – Main characteristic of the CE obtained with  $\eta_E=0.75$

<b>Engine</b>	<b>4-stroke, spark ignition</b>		
<b>Injection system</b>	Gasoline multi-point injection		
<b>Valvetrain</b>	4 valves/cylinder, VVT		
<b>Compression</b>	10		
<b>Max boost</b>	1.5 bar		
<b>Expander</b>	0.75		
$T_{4,c}/T_{1,c}$	3.5	4.0	4.5
<b>Displacement</b>	938 cc	912 cc	896 cc
<b>Number of</b>	3	3	3
<b>Bore</b>	71.0 mm	70.4 mm	70.0 mm
<b>Stroke</b>	78.9 mm	78.1 mm	77.7 mm
<b>Max BMEP<sub>TOT</sub></b>	17.85 bar at 3880	18.00 bar at 3917	18.10 bar at 3939
<b>Min BSFC<sub>TOT</sub></b>	218.3 g/kWh	214.3 g/kWh	210.0 g/kWh
<b>Variation of min BSFC</b>	-5.9 %	-6.9 %	-8.3 %
<b>Max RMEP</b>	3.27 bar at 4743	3.94 bar at 4788	4.62 bar at 4815
<b>Max RMEP/BMEP<sub>TOT</sub></b>	21.9 %	27.4 %	33.3 %

The compound system considered here revealed approximately the same specific output power ( $BMEP_{TOT}$ ) as the reference turbocharged engine (reported in Table 3). Regarding fuel economy, apart from the expander efficiency and the temperature ratio, the minimum fuel consumption

obtainable by the electric CE is always lower than the consumption of the turbocharged reference engine. Reductions between 5.1% and 8.3% have been found. Table 5 and Table 6 also report the maximum value assumed by the *RMEP*, which ranged from 2.90 to 4.62 bar (corresponding to the power delivered between 10.8 and 16.6 kW), and revealed a higher sensitivity to the temperature ratio (with increments up to 43%) than to the expander efficiency (increments of about 12%). The same tables also show that the expander contributes to the overall output power, with a maximum share of power contribution from 19% to 33%, depending on the particular temperature ratio or expander efficiency. This means that, if adequately recovered, the unexpanded gas energy may constitute a relevant part of the whole propulsion energy and may contribute to lowering both the vehicle fuel consumption and related emissions.

Although the initial results are promising, a comparison based on minimum fuel consumption or maximum specific power is not exhaustive. Considering the application to hybrid propulsion vehicle, a comparison between the proposed CE and the traditional turbocharged engine was carried out on an equal output power basis. Figure 17 and Figure 18 show, for the two extreme cases ( $T_{4,C}/T_{1,C}=3.5$ ,  $\eta_E=0.70$  and  $T_{4,C}/T_{1,C}=4.5$ ,  $\eta_E=0.75$ , respectively) the efficiency improvement obtainable by the proposed CE for the reference turbocharged engine, as a function of the (overall) output power and mean piston speed. The first observation is that the energetic advantage of the expander-generator increases with the overall output power. This observation can be easily explained considering that the power contribution of the expander increases with the exhaust gas mass flow and the in-cylinder pressure levels. In further detail, the efficiency improvement remains within 5% for power level within 10 kW, reaching a maximum of about 10% in the first case (Figure 17) and 12% in the second (Figure 18) at the maximum output power.

This is an encouraging result, especially because the effect of the increased exhaust pressure on the gross indicated efficiency was not considered for the turbocharged engine.

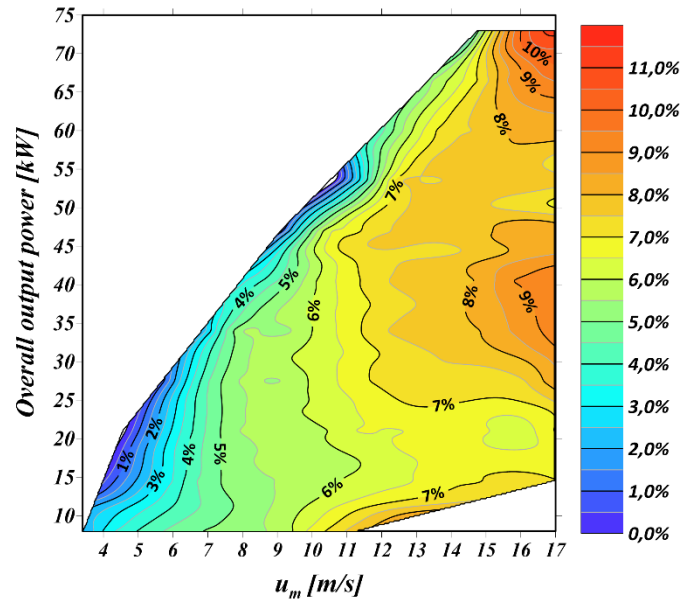


Figure 17 – Efficiency improvement of the CE compared to the reference turbocharged engine as a function of power output and mean piston speed ( $T_{4,C}/T_{1,C}=3.5$  and  $\eta_E=0.70$ )

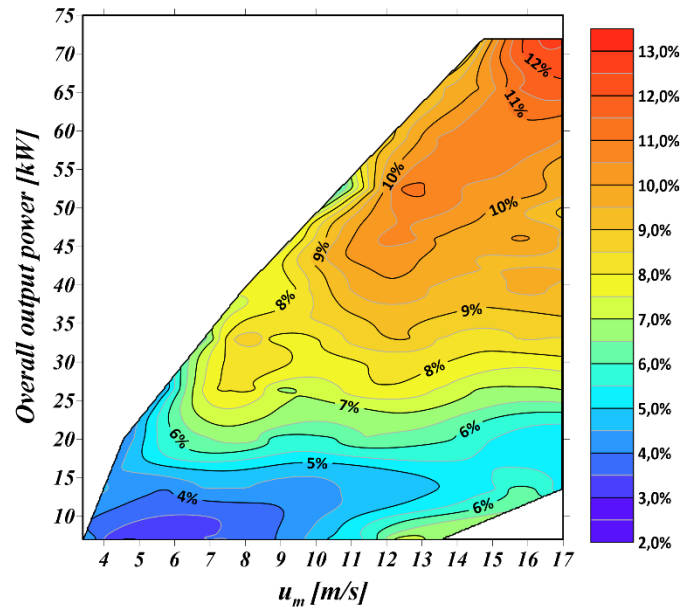


Figure 18 – Efficiency improvement of the CE compared to the reference turbocharged engine as a function of power output and mean piston speed ( $T_{4,C}/T_{1,C}=4.5$  and  $\eta_E=0.75$ )

Moreover, better improvements were obtained with the higher expander efficiency and temperature ratio. Hence, it can be concluded that a proper study and optimization of the compound performance of both engine and expander could allow even better results. To ascertain the effect on an after-treatment device like a catalyst, whose inlet gas temperature should safely remain above 400°C, the exhaust gas temperature downstream the expander was also evaluated. As expected, the worst case is represented by  $T_4/T_1=3.5$  and expander efficiency  $\eta_E=0.75$ , which gave expander outlet temperatures between 808 and 918K, depending on the engine load and speed of rotation. These temperatures are, however, sufficiently higher to allow a stable and efficient conversion in the catalyst. Hence, it may be concluded that the expander-generator implementation would not cause any critical situation for exhaust after-treatment devices like three-way catalysts.

Focusing on the hybrid propulsion, however, it is worth pointing out that in such vehicles, the thermal machine is usually employed on its best efficiency curve, i.e., the curve connecting the operative conditions which ensure, for each power request, the maximum efficiency. Therefore, a further, and fairer, comparison could be based on the best efficiency curves of both CE and turbocharged engine. To this purpose, for each output power, the authors determined the best efficiency operative conditions (load and speed) on each of the two propulsive solutions. The results of this evaluation are reported in Figure 19, which refers to the case  $T_{4,C}/T_{1,C}=3.5$  and  $\eta_E=0.70$ , and Figure 20, obtained considering  $T_{4,C}/T_{1,C}=4.5$  and  $\eta_E=0.75$ . Both diagrams report the best efficiency curves of the two propulsive solutions, together with the efficiency increments obtainable by the proposed CE system. It should be noted that the benefit introduced by the exhaust energy recovery strictly depends on the output power level (as already revealed by the contour maps in Figure 17 and Figure 18), and may reach values as high as 8.3% in the first case,

and 12.8% in the second. Considering the entire power range, the average efficiency increased by 4.1% compared to the turbocharged engine in Figure 19, and 7.7% in Figure 20.

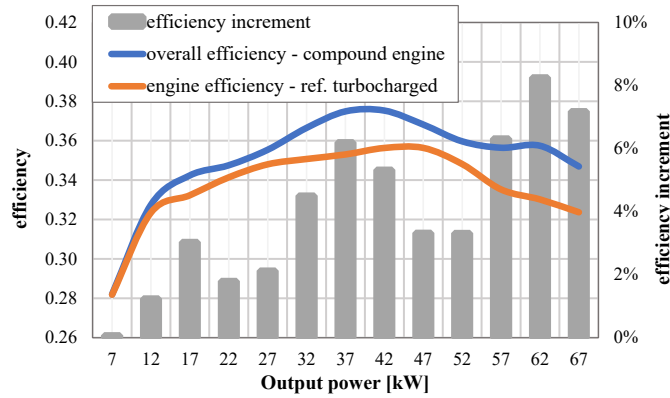


Figure 19 – Comparison between the best efficiencies obtained for different power output ( $T_{4,C}/T_{1,C}=3.5$  and  $\eta_E=0.70$ )

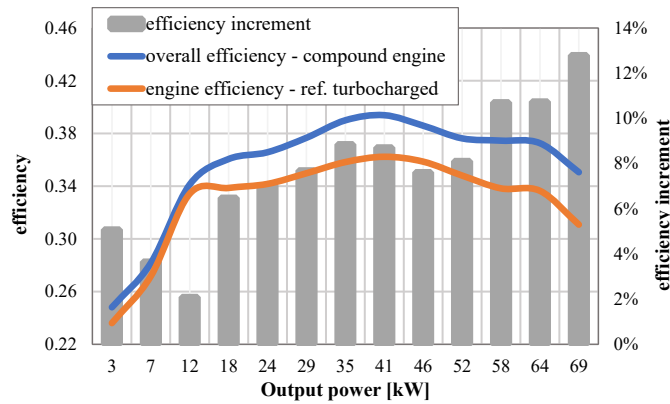


Figure 20 – Comparison between the best efficiencies obtained for different power output ( $T_{4,C}/T_{1,C}=4.5$  and  $\eta_E=0.75$ )

According to the results obtained by this last comparison, the compound system composed of an electrically supercharged SI engine equipped with an exhaust expander-generator has excellent potential to improve fuel economy with corresponding emissions reduction. Hence, it can be concluded that the CE system merits further and deeper investigation, focusing on the optimization of the compound system constituted by the engine and the expander generator.

Research and development should also be carried out on the expander-generator unit itself, as performance strictly influences the energetic advantages of the proposed compound system.

## **Conclusion**

In this paper, the authors evaluated the energetic advantages of a separated electric CE, i.e., a propulsive unit composed of an electrically supercharged SI engine equipped with an exhaust gas expander connected to an electric generator, to transform the unexpanded exhaust gas energy, typical of conventional thermodynamic engine cycles, into electrical energy. The system proposed is specifically intended for hybrid vehicles, where the energy produced by the expander-generator can be stored in the storage system and hence profitably employed for vehicle propulsion.

Several operating conditions were assessed to evaluate the effective applicability of the proposed CE in a vehicle scenario, and the resulting overall efficiency was compared to the efficiency of a reference turbocharged engine. The comparison was carried out considering a hybrid vehicle application, i.e., on an equal output power basis and in steady-state operative conditions. To remain as close as possible to the real engine efficiency, actual engine performance data and values retrieved from literature or experimental tests were adopted in the simple and effective calculations performed.

Moreover, to make the result obtained reliable and valid on a wide range of engines, three different values were considered for the temperature ratio  $T_4/T_1$  (namely 3.5, 4.0, and 4.5), which is the ratio between the in-cylinder gas temperature at the end of the expansion stroke and the beginning of the compression stroke. This temperature ratio depends on the particular engine configuration and specification. It strongly influences the temperature of the exhaust gas entering the expander or the turbine, thus playing an important role in determining the energetic



advantage of the CE proposed concerning the reference turbocharged engine. Also, the exhaust gas expander plays a fundamental role and may substantially affect the entire system's efficiency. For this reason, the effect of a variation of its average efficiency was also taken into account by considering two different efficiency values (namely  $0.70$  and  $0.75$ ).

When considering the exhaust gas expander, the effect of exhaust backpressure variation on engine efficiency, pumping cycle, and volumetric efficiency was carefully considered. Hence, the optimal exhaust pressure of the proposed CE was established for each operative condition, maximizing its overall efficiency. As expected, the advantage connected to the exhaust gas energy recovery increased with the overall output power, which has been explained considering that the power contribution of the expander increases with the exhaust gas mass flow and with the in-cylinder pressure levels. As a final result, considering the whole operative condition map, the efficiency improvement obtained by the CE reached a maximum value between 10% and 12%, depending on the particular temperature ratio or expander efficiency considered. Furthermore, focusing on the hybrid vehicle application, the authors restricted the efficiency comparison to the best efficiency curves of each propulsive unit: in this case, the maximum advantage of the CE varied between 8.3% and 12.8%.

The evaluation performed also established that the contribution of the expander-generator to the overall power produced by the CE could be considerable, reaching a share of 33% and a maximum delivered power of 16.6 kW. This means that, if adequately recovered, the unexpanded gas energy may constitute a relevant part of the whole propulsion energy, and may contribute to lower both vehicle fuel consumption and related emissions.

Regarding the expander operating conditions, the optimal exhaust gas pressure was estimated to range between 2.6 bar and 3.3 bar. Moreover, better improvements were obtained for the higher

speed and load of the engine, meaning that the exhaust gas temperature  $T_S$  at the expander inlet could reach 800-900°C. These results should be properly considered for an optimized expander design, which could be a multistage radial-axial turbine. Furthermore, as expected, better improvements were obtained with higher expander efficiency and temperature ratio. It can be concluded that a proper study and optimization of the compound performance of both engine and expander could achieve even more significant results than those presented here.

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### Symbols/Abbreviations

$a, b, c$	Parameters of the turbine MFP model
$BMEP$	Brake mean effective pressure
$BMEP_{max}$	Maximum BMEP
$BMEP_{TOT}$	Overall BMEP of the compound engine
$BSFC$	Brake specific fuel consumption
$BSFC_{TOT}$	Overall BSFC of the compound engine
$CE$	Compound engine
$CMEP$	Compressor mean equivalent pressure
$c_{p,c}$	Specific heat at a constant pressure of the air
$c_{p,s}$	Specific heat at a constant pressure of burned gas
$c_{p,u}$	Specific heat at a constant pressure of unburned gas
$CR$	Volumetric compression ratio
$EVO$	Exhaust valve open
$f_{CR}$	Correction factor related to compression ratio
$FMEP$	Friction mean effective pressure
$G_C$	Air mass flow (turbo/supercharged engine)
$G_O$	Air mass flow (naturally aspirated engine)
$G_T$	Turbine gas mass flow
$HEV$	Hybrid electric vehicle

<i>ICEV</i>	Internal combustion engine vehicle
<i>IMEP</i>	Indicated mean effective pressure
<i>IMEP<sub>g</sub></i>	Gross indicated mean effective pressure
<i>ISFC</i>	Indicated specific fuel consumption
<i>IVC</i>	Inlet valve closure
<i>k<sub>s</sub></i>	Isentropic coefficient of the exhaust gas
<i>LHV</i>	Lower heating value of the fuel
<i>LIVC</i>	Late Intake Valve Closure
<i>m<sub>0</sub></i>	Fresh charge mass
<i>MAP</i>	Manifold absolute pressure
<i>MAP<sub>max</sub></i>	Maximum MAP
<i>m<sub>s</sub></i>	Residual gas mass
<i>p<sub>c</sub></i>	Boosting pressure
<i>P<sub>comp</sub></i>	Power required by the compressor
<i>P<sub>exp</sub></i>	Power produced by the expander
<i>PMEP</i>	Pumping mean effective pressure
<i>p<sub>s</sub></i>	Engine exhaust pressure
<i>p<sub>so</sub></i>	Exhaust pipe pressure
<i>RGF</i>	Residual gas fraction
<i>R<sub>INT</sub></i>	Intercooler efficiency
<i>RMEP</i>	Recovery mean equivalent pressure
<i>R<sub>s</sub>'</i>	Specific gas constant of exhaust gas or burned gas
<i>SI</i>	Spark ignition
<i>T</i>	Temperature
<i>T<sub>0</sub></i>	Air temperature in the intake manifold
<i>T<sub>1</sub></i>	In-cylinder gas temperature at IVC
<i>T<sub>4</sub></i>	In-cylinder gas temperature at EVO
<i>T<sub>C</sub></i>	Air temperature at the intercooler outlet
<i>T'<sub>C</sub></i>	Air temperature at the compressor outlet
<i>T<sub>R</sub></i>	Residual gas temperature
<i>T<sub>S</sub></i>	Exhaust gas temperature
<i>u</i>	Normalized mean piston speed = $u_m / u_{m,max}$



$u_m$	Mean piston speed
$u_{m,max}$	Maximum mean piston speed
$V_A$	Naturally aspirated engine displacement
$V_C$	Engine displacement in the compound unit
$V_T$	Turbocharged engine displacement
$\alpha$	Air-fuel ratio
$\alpha_{ST}$	Stoichiometric air-fuel ratio
$\beta_C$	Compressor pressure ratio
$\beta_S$	Expander and turbine pressure ratio
$\gamma, \sigma$	Parameters of the model for the gross indicated efficiency variation as a function of RGF
$\delta_0$	Air density in the intake manifold
$\delta_C$	Density of the compressed air in the intake manifold
$\varepsilon$	Number of revolutions per cycle
$\eta_b$	Brake thermal efficiency of the engine
$\eta_{b,max}$	Maximum brake thermal efficiency of the engine
$\eta_{b,TOT}$	Overall brake thermal efficiency of the compound engine
$\eta_C$	Compressor efficiency
$\eta_E$	Expander efficiency
$\eta_{EM}$	Electric motor efficiency
$\eta_i$	Indicated thermal efficiency of the engine
$\eta_{i,g}$	Gross indicated thermal efficiency of the engine
$\eta_{i,max}$	Maximum indicated thermal efficiency of the engine
$\eta_m$	Overall mechanical efficiency of the engine

$\eta_T$	Turbine efficiency
$\lambda$	Relative air-fuel ratio
$\lambda_V$	Volumetric efficiency of the engine
$\lambda_{V,C}$	Volumetric efficiency of supercharged / turbocharged engine
$\lambda_{V,max}$	Maximum volumetric efficiency of the engine
$\rho$	Engine compression ratio (naturally asp.)
$\rho'$	Engine compression ratio (turbo/supercharged)
$\phi$	Relative MAP = $MAP/MAP_{max}$
$\psi$	Normalized load variable = $BMEP/BMEP_{max}$

## Subscripts

$0$	Reference condition
$c$	Compression/compressed
$g$	Gross
$s$	Exhaust gas

## Appendix A

### Residual Gas Fraction (RGF) evaluation

When engine inlet valves close (IVC), the mass entrapped in the cylinder results in the sum of the residual gas from the previous cycle ( $m_s$ ) and of the fresh charge ( $m_o$ ); the residual gas fraction (RGF), which represents the ratio between the residual gas mass and the total in-cylinder mass, is hence:

$$RGF = \frac{m_s}{m_{TOT}} = \frac{m_s}{m_o + m_s} \quad (52)$$

The fresh charge mass entrapped in the cylinder depends on the engine volumetric efficiency  $\lambda_V$ :

$$m_0 = \lambda_v \cdot \frac{MAP}{R'_0 \cdot T_0} \cdot V \quad (53)$$

where  $T_0$  and  $MAP$  are the temperature and pressure in the intake manifold, respectively, and  $V$  is the engine displacement. Assuming the residual gas mass as the amount of in-cylinder exhaust gas at the ideal end of the exhaust stroke (i.e., at top dead center):

$$m_S = \frac{p_S}{R'_S \cdot T_R} \cdot \left( \frac{V}{\rho - 1} \right) \quad (54)$$

where  $T_R$  and  $p_S$  represent the temperature and pressure of the in-cylinder residual gas,  $\rho$  is the engine compression ratio and hence  $V/(\rho-1)$  the in-cylinder volume at top dead center. The residual gas temperature  $T_R$ , in line with the simple approach followed in this paper, can be evaluated neglecting the heat transfer with in-cylinder wall during the exhaust stroke, thus assuming an isentropic transformation:

$$T_R = T_4 \cdot \left( \frac{p_4}{p_S} \right)^{\frac{1-k_S}{k_S}} \quad (55)$$

where  $p_4$  and  $T_4$  are the in-cylinder gas pressure and temperature when the exhaust valves open (EVO). As mentioned, experimental findings confirmed by data reported in the scientific literature show that for a spark-ignition engine, the ratio  $T_4/ T_1$  ranges from 3.5 to 4.5. The isentropic coefficient  $k_S$  should be evaluated as a function of the exhaust gas composition and temperature, as described above (equation (25)).

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