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International Journal of Thermal Sciences 44 (2005) 1161–1168
International Journal of Thermal Sciences
www.elsevier.com/locate/ijts

Energy, exergy and cost analysis of a micro-cogeneration system based on an Ericsson engine

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Received 25 April 2005; accepted 7 September 2005

Available online 25 October 2005

Abstract

Hot air engines (Stirling and Ericsson engines) are well suited for micro-cogeneration applications because they are noiseless, and they require very low maintenance. Ericsson engines (i.e. Joule cycle reciprocating engines with external heat supply) are especially interesting because their design is less constrained than Stirling engines, leading to potentially cheaper and energetically better systems. We study the coupling of such an Ericsson engine with a system of natural gas combustion. In order to design this plant, we carry out classic energy, exergy and exergo-economic analyses. This study does not deal with a purely theoretical thermodynamic cycle. Instead, it is led with a special attempt to describe as accurately as possible what could be the design and the performance of a real engine. It allows us to balance energetic performance and heat exchanger sizes, to plot the exergy Grassmann diagram, and to evaluate the cost of the thermal and electric energy production. These simple analyses confirm the interest of such systems for micro-cogeneration purposes. The main result of this study is thus to draw the attention on Ericsson engines, unfortunately unfairly fallen into oblivion.

Keywords: Joule cycle; Ericsson engine; Micro-cogeneration; CHP; Exergy analysis

1. Introduction

In the low electrical power range (500 W ... 50 kW), combined heat and power (CHP), also called cogeneration, does not have the same development as for higher power. This lack of success, although the market for residential cogeneration could be strong, is mainly due to the absence of suitable systems for this power range: internal combustion engines generate noise and vibrations. The market seems more promising for systems based on external combustion. Especially a lot of developments are devoted to CHP systems with kinematic or free piston Stirling engines and some of these systems are already commercially available. We study a micro-cogeneration system based on an Ericsson engine coupled with a system of natural gas combustion. The objective of this plant is to produce sanitary and heating hot water and 11 kW of electric output. In order to design this system, we carry out energetic, exergetic and exergo-economic studies.

2. The ERICSSON engine: Principle and advantages

A tentative thermal machine classification has been proposed previously [1]. It allows to identify a special family of engines [2], with the following features: reciprocating engines, external heat supply, separate compression and expansion cylinders, regenerator or recuperator, monophasic gaseous working fluid. These engines are sometimes called hot air engines [3], even if the air used in the XIXth century engines has been replaced by high pressure hydrogen or helium in a lot of modern engines. Hot air engines have known commercial success during the XIXth century, but, since the beginning of the XXth century, they have been discarded and replaced by internal combustion engines or electric motors.

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1290-0729/$ – see front matter
doi:10.1016/j.ijthermalsci.2005.09.005

Nomenclature

c	fluid velocity	$\text{m}\cdot\text{s}^{-1}$
c_i	exergy cost of stream i	$€\cdot\text{J}^{-1}$
D	diameter	m
$\dot{E}x_i$	rate of exergy transfer of stream i	W
L_t	tube length	m
$\dot{m}$	mass flow rate	$\text{kg}\cdot\text{s}^{-1}$
N_t	number of pipes	
P	cost	€
p	pressure	Pa
$\dot{Q}$	thermal power	W
R	net efficiency (Figs. 4, 5)	
$\overline{R}$	universal gas constant	$\text{J}\cdot\text{mol}^{-1}\cdot\text{K}^{-1}$
r	ideal gas constant	$\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$
S	heat transfer area	m^2
T	temperature	K
$\dot{W}$	mechanical power	W
$\dot{Z}_i$	cost per unit time of component i	$€\cdot\text{s}^{-1}$

Greeks symbols

ε	heat exchanger effectiveness	
η	efficiency	
ξ	pressure loss coefficient	
ρ	density	$\text{kg}\cdot\text{m}^{-3}$
ψ	valve pressure loss factor	

Subscripts

amb	ambient combustion air
C	compression cylinder
CC	combustion chamber
chim	chimney
cr	working air between C and R
D	destroyed (exergy)
E	expansion cylinder
e	exhaust combustion gases, between CC and H
ep	exhaust combustion gases, between H and P
er	working air, between E and R
g	gaseous fuel
H	heater
h	working air, heater outlet
IC	inter-cooler
ind	indicated (work or efficiency)
in	in, inner
K	cooler
k	working air, inlet 1st stage of C
k1	working air, between $C1$ and IC
k2	working air, between IC and $C2$
mec	mechanical
net	net
out	out, outer
P	combustion air pre-heater
pa	pre-heated combustion air, between P and CC
R	regenerator or recuperator
rh	working air, between R and H
rk	working air, regenerator outlet
shell	shell
t	tube

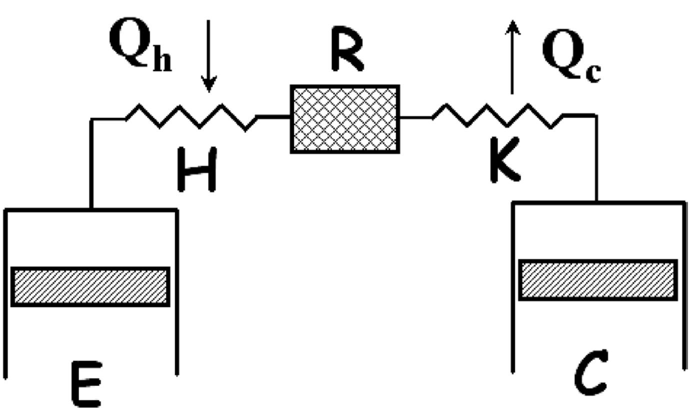

Fig. 1. Typical Stirling engine.

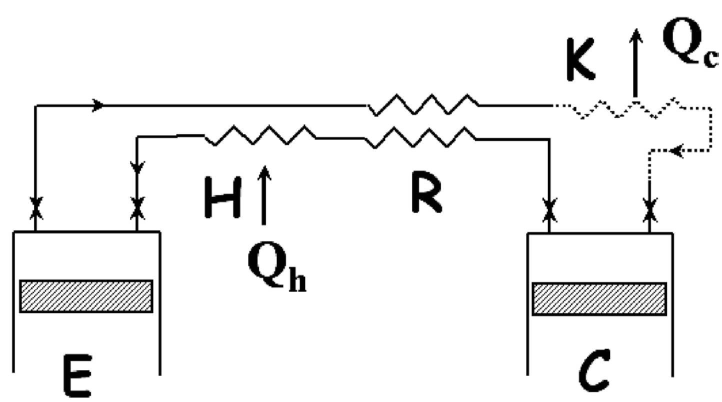

Fig. 2. Typical Ericsson engine.

The family of hot air engines is divided in two subgroups: the Stirling engines, invented in 1816, have no valves (Fig. 1) whereas Ericsson engines, invented in 1833 (Fig. 2) have valves in order to isolate the cylinders.

Since the work of the Philips company, around the second world war, the attention has been drawn on Stirling engines and lots of research and developments have been carried out. However, up to now, very little interest is dedicated to Ericsson engines.

On the opposite of internal combustion engine, the operation of hot air engine is not noisy and requires very low maintenance. The Ericsson configuration, with valves, shows several advantages compared to the Stirling configuration [4]. Amongst them, it is worth to note that the Ericsson engine heat exchangers are not dead volumes, whereas the Stirling engine heat exchangers designer has to face a difficult compromise between as large heat transfer areas as possible, but as small heat exchanger volumes as possible.

3. Theoretical reference cycle

It is usual to refer to some theoretical cycles to study thermal machines. This theoretical cycle is assumed to represent the operation of an ideal machine based on the same principle and configuration as the studied machine. This approach is often useful. Theoretical so-called Stirling cycle and Ericsson cycle are presented in most thermodynamic handbooks [5,6]. It could be thought that these cycles are suitable reference cycles for Stirling engines and Ericsson engines, respectively. After Organ [7], we emphasized that the theoretical Stirling cycle (two isothermal processes, and two isochoric processes) is not appropriate to study Stirling engines [8]. Unfortunately it happens to be the same for Ericsson engines. The theoretical Ericsson cycle is made up of two isothermal processes and two isobaric processes. Why is it not suitable to describe an ideal Ericsson engine? Fig. 2 gives the answer: due to the lack of heat exchange surface in the cylinder, the heat transfers between the working fluid and the hot and cold sources take place in external heat exchangers adjacent to the cylinders instead of through the cylinders wall [9]. So, the cycle made up of two isentropic and two isobaric processes, usually called the Brayton or Joule cycle, seems more suitable to describe the Ericsson engine. This theoretical cycle is often used to describe the gas turbine engine. Actually, the Ericsson engine is a special gas turbine engine where the turbocompressor is replaced by a reciprocating compressor and the turbine is replaced by a reciprocating piston/cylinder expander. The engine initially designed by Joule was a simplified version of the engine built by Ericsson twenty years earlier, as the Joule engine had no recuperator.

4. Configuration of the studied system

Fig. 3 shows the studied system. On the top of the figure, we can see a sketch of the combustion system with a combustion chamber *CC* and the air pre-heater *P*. The burned gases transfer thermal energy to the reciprocating engine through a heat exchanger *H*, called "heater". This engine is located on the lower part of the figure.

There are several possibilities for the choice of the working fluid and for the layout of the engine. For instance, hydrogen or helium can be used as working fluid. This naturally leads to high pressure, compact, engines, requiring a quite high level of technology. This is the usual choice for most Stirling micro-cogeneration systems developed up to now. However, for micro-cogeneration systems, it is not sure that the low size of the system is a determining factor, especially if reduced size is obtained from a high additional cost. We prefer a low-pressure system with air as the working fluid. In this case, we can use an open cycle. The engine is composed of five working spaces that we can easily locate on Fig. 3: the afore mentioned heater *H*, the recuperator *R*, the inter-cooler *IC*, which are heat exchangers, and the expander space *E* and the compression spaces *C*1 and *C*2. The engine operating principle is rather simple. Air is allowed at atmospheric temperature and pressure and then, it is compressed. The compression is realized into two stages with water inter-cooling. Then, air goes through the recuperator where it recovers energy before going through the heater. There, it recovers the energy of the combustion gases before being expanded in the expansion cylinder. On the other hand, the expansion is realized in one stage in two parallel expanders without reheating. In the compression and expansion spaces, we have double-acting pistons.

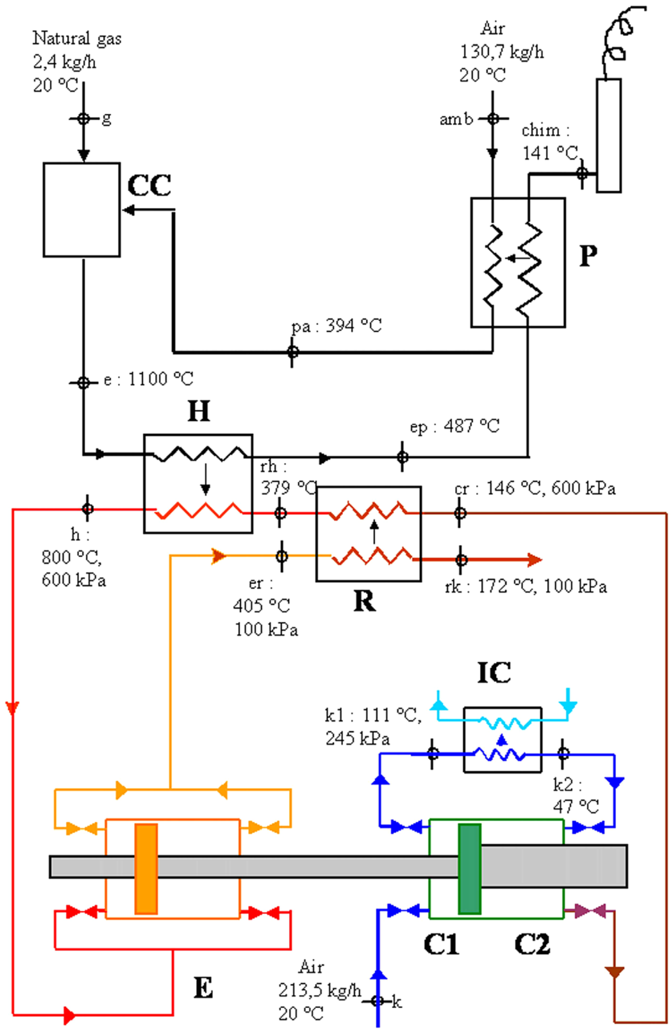

Fig. 3. Energy system main features.

5. Energy analysis

5.1. Modeling

5.1.1. Inlet data

A model is developed in order to obtain a suitable design for the cylinders and the heat exchangers. At this point of the study, the model used is not a dynamic model, but a 0-D global model whose variables are time-independent [10]. In order to study the energetic system under consideration, several quantities are assumed as inlet data. These are:

- The fuel mass flow rate; we consider natural gas with $\dot{m}_g = 0.67 \times 10^{-3}$ kg·s^{-1}.
- The combustion gas temperature at the outlet of the combustion chamber, and the inlet of the heater; we assume $T_e = 1100\,^\circ$C, in accordance with steel whose cost and machining is affordable.
- The compressed air temperature T_h in the engine is also limited for technological reasons of material thermal and mechanical resistance; we assume $T_h = 800\,^\circ$C; the energetic performance of the system obviously deeply depends on this temperature so that a detailed technical-economic study would be necessary to assess this value.
- The minimum working fluid pressure is the atmospheric pressure since the cycle is open; the maximum air pressure is set to $p_{cr} = 6 \times 10^5$ Pa; it is well known that the cycle efficiency slowly decreases as the pressure ratio increases if the recuperator effectiveness is equal to 1 [5]; if the recuperator effectiveness is lower than 1, there is an optimum pressure that maximizes the power plant thermal efficiency [6]; for higher pressure values the efficiency decrease is quite slow; however we aim to maximize the global efficiency, not the thermal efficiency. The global efficiency also depends on the mechanical efficiency, which in turn depends on the pressure levels in the engine. Considering this, our simulation results show that, from a pressure $p_{cr} = 6 \times 10^5$ Pa, the power plant performance hardly changes with the pressure ratio.
- The heat exchangers are described by means of their effectiveness. Quite high values for gas/gas heat exchangers are investigated, i.e., $\varepsilon_P = \ldots 0.8 \ldots$, $\varepsilon_H = \ldots 0.85 \ldots$, $\varepsilon_R = \ldots 0.9 \ldots$ and $\varepsilon_{IC} = \ldots 0.7 \ldots$ respectively for the combustion air pre-heater, the heater, the recuperator and the compressor inter-cooler; this high effectiveness of the exchangers is needed in order to obtain a high efficiency of the micro-cogeneration plant.

In order to study the energetic system, the following assumptions are made:

- The combustion gases, the combustion air and the working fluid air are assumed to obey the ideal gas law; the specific heat of these gases is a polynomial function of temperature [11]; the combustion gases composition is computed from the composition of natural gas and the computed excess air factor assuming a complete combustion.
- The pressure variations are realized in reciprocating piston/cylinder machines; so the compression and the expansion are assumed to be isentropic in the cylinders [12]; however the actual pressure ratio in the cylinders is different from the p_{cr}/p_k due to pressure losses through the valves.
- Pressure losses through the inlet and exhaust valves of the cylinders are modeled by

$$\Delta p = \xi \rho \frac{c^2}{2} \tag{1}$$

The working air is assumed to obey the ideal gas law. This yields:

$$\frac{\Delta p}{p}\frac{T}{T_k} = \frac{\xi}{r T_k}\frac{c^2}{2} = \psi \tag{2}$$

We consider that the valves around the cylinders can be designed so that the pressure loss factor ψ is the same for each valve and we assume $\psi = 0.03$; it should be noted that the performance of the system deeply depends on this value.
- A counter-flow shell and tube design is considered for all heat exchangers, the combustion gases (heater H, preheater P) and the low pressure air (recuperator R) flowing out of the tubes; thermophysical properties of the fluids are computed at the mean temperature between the inlet and the outlet of the heat exchanger. The dynamic viscosity and thermal conductivity of the combustion gases are assumed to be the same as those of air. They are computed as a function of the mean temperature from a Sutherland relationship [7]. Heat transfer coefficients inside and outside the tubes are computed by the Gnielinski correlation [13] for clean pipes. In order to design the heat exchangers, we fix the maximal velocity of the fluid in the tubes and in the shell and we fix the tubes diameter for each heat exchanger.
- Mechanical losses are quite difficult to evaluate as the models developed for internal combustion engines are not appropriate. We consider a separate mechanical efficiency for the expansion and the compression cylinders. The net shaft power is then obtain from:

$$|\dot{W}_{net}| = \eta_{mec,E}|\dot{W}_E| - \frac{|\dot{W}_C|}{\eta_{mec,C}} \tag{3}$$

In this relation we assume that both compression and expansion mechanical efficiencies are equal to 0.9. It is worth noting that the performance of the power plant deeply depends on this value.
- The cylinder volumes are computed from the working fluid mass flow rate and from the engine rotational speed. For technological reasons, and for mechanical efficiency, we choose a low speed, i.e. 1000 rpm. This leads to high cylinder capacities. The number of cylinders is determined by limiting the mean linear piston velocity to 8 m·s^{-1}.
- The net power delivered by the engine is the shaft power. The electromechanical efficiency of the electric generator and the power consumption of the auxiliary pumps, ventilators, … are not taken into account. They are masked by the uncertainty on the mechanical efficiency.

5.1.2. Calculated parameters

These fixed values allow to calculate the values of the other parameters. The computation procedure is the following. First we choose an initial value for the pre-heated air temperature T_{pa}. This allows to compute the air/fuel ratio in order to obtain $T_e = 1100\,^\circ$C. It is then possible to calculate all temperatures and mass flow rates in the system and to obtain the pre-heater effectiveness. This computed effectiveness is compared to the assumed pre-heater effectiveness. The pre-heated air temperature value is corrected and the whole procedure is repeated until the difference between the assumed and the computed pre-heater effectiveness is negligible. Then, we calculate the mechanical characteristics of the engine. As soon as we obtain all these parameters, we can calculate the characteristics of each heat exchanger [14].

5.2. Modeling results

5.2.1. Characteristics for the chosen configuration

The heat exchangers of the chosen configuration are characterized by the following effectiveness: $\varepsilon_P = 0.8$, $\varepsilon_H = 0.85$, $\varepsilon_R = 0.9$ and $\varepsilon_{IC} = 0.7$ (see Section 5.2.2). For this configuration, the thermodynamic characteristics of the cycle are given in Fig. 3. The energetic performance is reported in Tables 1–3. The energy input $\dot{Q}_{\mathrm{g}}$ is computed from the higher calorific value of the gaseous fuel. The net efficiency takes into account the net shaft power only:

$$\eta_{\mathrm{net}} = \frac{\dot{W}_{\mathrm{net}}}{\dot{Q}_{\mathrm{g}}} \tag{4}$$

The global efficiency (first principle) also takes into account the heat of the working air at the exhaust of the engine (state rk, Fig. 3). Indeed the clean, dry, hot air at the exit of the engine can be directly used for heating purposes. The heat evacuated from the inter-cooler by the cooling water is also included in the net efficiency. The whole heat at the chimney (state *chim*) is assumed to be lost.

$$\eta_{\mathrm{global}} = \frac{\dot{W}_{\mathrm{net}} + \dot{Q}_{\mathrm{rk}} + \dot{Q}_{IC}}{\dot{Q}_{\mathrm{g}}} \tag{5}$$

Table 1
Thermal powers

	$\dot{Q}$ [kW]
fuel (higher calorific value) (g)	36.2
heater (H)	27.8
engine exhaust (rk)	9.2
inter-cooler (IC)	3.8
chimney (chim)	8.4

Table 2
Mechanical powers

	$\dot{W}$ [kW]
indicated	14.7
net	10.8

Table 3
Efficiencies

	η
indicated	0.529
mechanical	0.736
net (higher calorific value)	0.299
global first principle	0.657

The dimensional characteristics are given in Tables 4 and 5. Even if the technological choices lead to a large engine and the shell-and-tubes heat exchanger configuration is not the best suitable for compactness, it can be seen that the total size of the micro-cogeneration power plant is acceptable, since the whole power plant can be installed on a floor area of less than 1 m^2.

5.2.2. Effectiveness variations

With the view of finding the best design, we run the model with different heater and recuperator effectiveness while the combustion air pre-heater effectiveness is kept constant. Fig. 4 plots the iso-net efficiency and the iso-total heat exchanger transfer area as a function of the effectiveness of the heater and the recuperator. As expected, the net efficiency increases with the effectiveness. We obtain a net efficiency varying from 0.26 to 0.31. The total heat exchanger transfer area varies from 18 to 47 m^2.

The increase in total heat transfer area is more sensitive to an increase in the recuperator rather than the heater effectiveness. If the recuperator effectiveness increases, the total heat exchanger transfer area can increase a lot, without necessary increasing the net efficiency. However, it is possible to move

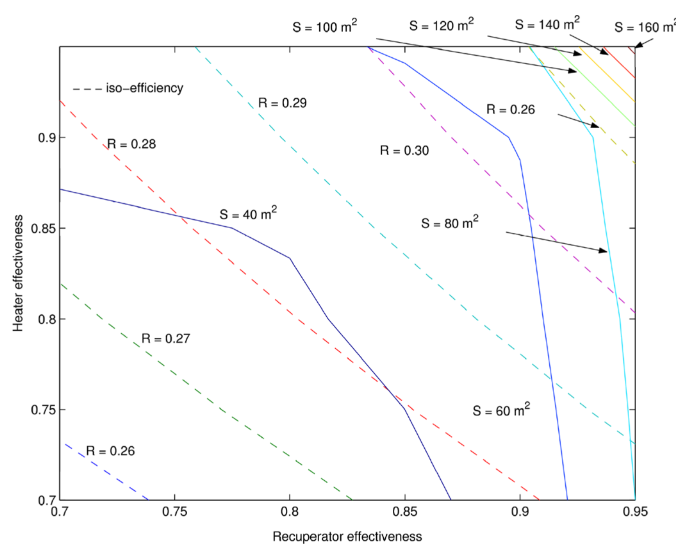

Fig. 4. Iso-net efficiency and iso-total heat exchanger transfer area as a function of the heater and the recuperator effectiveness.

Table 4
Cylinders

	C_1 [mm]	C_2 [mm]	E [mm]
bore	136	91	207
stroke	207	207	207

Table 5
Heat exchangers dimensional characteristics

	D_{shell} [mm] in/out	D_t [mm] in/out	N_t	L_t [m]
pre-heater P	90/100	10/12	33	3.9
heater H	90/100	7/10	54	3.4
recuperator R	100/110	4/6	126	5.9

along an iso-net efficiency curve and to find the corresponding minimum total heat exchanger transfer area. This point corresponds to heater effectiveness higher than the recuperator effectiveness. Finally, Fig. 4 shows that our design assumptions allow to reach a good compromise between a good global efficiency and an acceptable total heat exchanger transfer area.

Due to the different temperature levels, a unit of transfer area in the heater is obviously more expensive than in the recuperator and in the air pre-heater. However, from our model, the complete design of all three heat exchangers is known, that is, the heat transfer area, the number of tubes, the shell diameter and the length of each heat exchanger. From that, and from tubes manufacturers cost data for suitable heater, recuperator and pre-heater tubes, we can determine the cost of the raw material for each heat exchanger. On Fig. 5, the global raw material cost of the heater and the recuperator has been plotted as a function of the effectiveness of these two heat exchangers. From this figure, it is possible to determine the heat exchangers transfer area distribution that will minimize the raw material cost for a given net efficiency.

The cost curves in Fig. 5 are less vertical than the area curves in Fig. 4, meaning that the cost sensitivity to the recuperator effectiveness is less pronounced than in the case of the total area. So, as expected by the higher cost assign to the heater tubes, the design that will minimize the heat transfer area cost for a given net efficiency is obtained for a lower heater effectiveness but a higher recuperator effectiveness than in Fig. 4.

We note that the cost of the raw material for the two heat exchangers varies from 5000 € to 12 000 €. For the chosen effectiveness values of $\varepsilon_H = 0.85$ and $\varepsilon_R = 0.9$, we observe that we have the heat transfer area distribution that minimize the raw material cost, which is about 7000 €, for a net efficiency of nearly 30%. A small efficiency increase would lead to a dramatic heat exchangers cost increase. Note that the ranges of effectiveness considered in this sensitivity analysis are narrow enough to have no significant influence on the sizes and the costs of the cylinders.

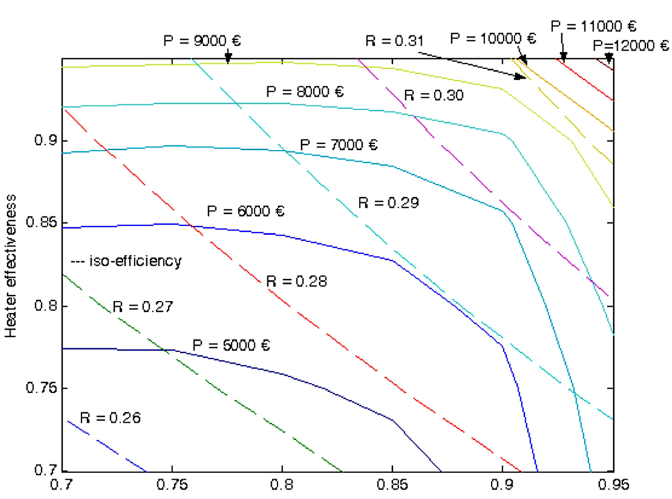

Fig. 5. Iso-net efficiency and iso-raw material cost for the heater and the recuperator as a function of the heater and the recuperator effectiveness.

We note that it is very important to assess each heat exchanger effectiveness appropriately to obtain a good net efficiency and to avoid huge transfer surfaces and prohibitive costs. From our analysis, it is possible to find the best compromise between the net efficiency, the heat transfer area and the cost of the heat exchangers. Of course, this sensitivity analysis could be extended to include the air pre-heater effectiveness, but this would go beyond the scope of this study.

6. Exergy analysis

6.1. Calculation of exergy transfer rates and exergy efficiencies

The exergy concept was developed in order to propose a method of analysis taking the first two principles of thermodynamics into account [5,6,15–18].

The values of exergy transfer rates are calculated from the exergy definition, and from the results of the energy analysis. The results are reported in Fig. 6.

For each component of the system, we can determine the exergetic efficiency and the exergy destruction in this component. The definition of the component exergetic efficiency depends on the function of the component. For example, if we consider the heater, its function is to heat air in the engine. Therefore, its exergetic efficiency can be defined as the ratio of the difference of exergy transfer rates between the cold stream outlet and inlet of the heater, on the difference of exergy transfer rates between the hot stream inlet and outlet of the heater:

$$\eta_H = \frac{\dot{E}x_{\mathrm{h}} - \dot{E}x_{\mathrm{rh}}}{\dot{E}x_{\mathrm{e}} - \dot{E}x_{\mathrm{ep}}} \tag{6}$$

The heat rejected by the inter-cooler *IC* and by the exhaust air stream rk is assumed to be useful. If the exergy destructions

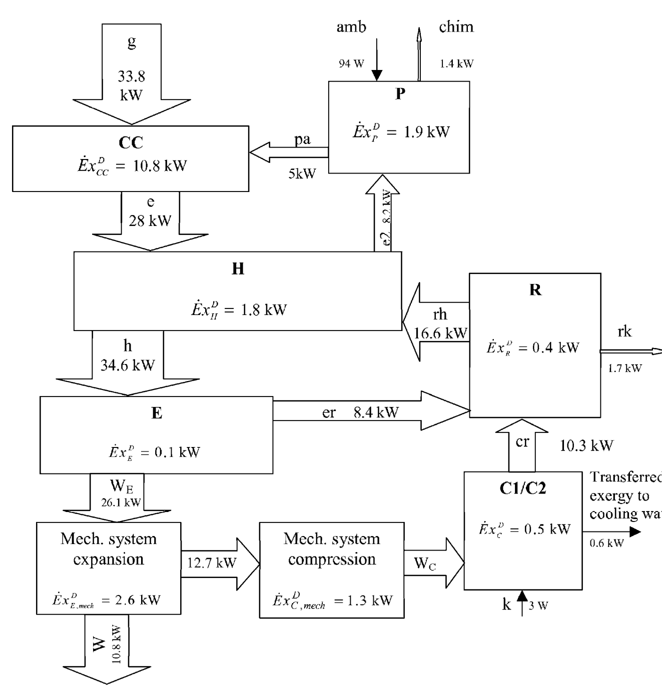

Fig. 6. Rates of exergy transfer and exergy destruction.

related to the transfer of these thermal energies are not taken into account, the exergetic efficiency of the global system can be defined by the relation:

$$\eta = \frac{\dot{W} + \dot{E}x_{\mathrm{rk}} + \dot{E}x_{\mathrm{k1}} - \dot{E}x_{\mathrm{k2}}}{\dot{E}x_{\mathrm{g}} + \dot{E}x_{\mathrm{amb}} + \dot{E}x_{\mathrm{k}}} \tag{7}$$

Assuming no exergy losses in the components of the system under consideration, the rate of exergy destruction is obtained by the difference of the exergy transfer rates of the streams flowing in the component considered and the sum of exergy transfer rates of the streams flowing out of the component considered plus the time rate of energy transfer by mechanical work. For the heater, we obtain:

$$\dot{E}x_H^D = \dot{E}x_{\mathrm{e}} + \dot{E}x_{\mathrm{rh}} - \dot{E}x_{\mathrm{h}} - \dot{E}x_{\mathrm{ep}} \tag{8}$$

It is useful to introduce the exergy destruction number, defined as the ratio of exergy destroyed by one component to the total exergy consumed by the system [19].

Table 6 presents the exergy efficiencies, exergy destruction rates and exergy destruction numbers for each component and for the global system. Moreover, in order to visualize the rates of exergy transfer and exergy destruction, we draw the exergy Grassmann diagram (Fig. 6).

We notice that the combustion chamber is the main source of exergy destruction. Indeed, the rate of exergy destruction in the combustion chamber amounts to 10.8 kW, that is the same value as the useful mechanical power produced by the engine. The combustion chamber is thus characterized by the highest exergy destruction number, i.e. 32.1% (Table 6).

The mechanical system of expansion, the air pre-heater and the heater are also responsible for high rates of exergy destruction, with exergy destruction numbers respectively 7.7%, 5.5%, 5.3%. Finally, Table 6 confirms that the component exergetic efficiency may help to compare similar components for one specific purpose, but is not useful for process optimization. For instance, the mechanical systems of expansion and compression have the same exergetic efficiencies, while the exergy destruction number on the expansion side is more than twice the one on the compression side.

7. Cost analysis of the system

The cost analysis allows us to establish a cost balance for each element k of the system, in order to determine the monetary value of the kilowatt of produced power and the exergy costs of the different streams i. Assuming as previously that all components are adiabatic (no heat transfer to the environment or to other components, that is no exergy losses), this equation will be as follows [16]:

$$\dot{Z}_{\mathrm{k}} + \left(\sum_i c_i\,\dot{E}x_i\right)_{\mathrm{in},k} = \left(\sum_i c_i\,\dot{E}x_i\right)_{\mathrm{out},k} + c_W\,\dot{W} \tag{9}$$

In Eq. (9), $\dot{Z}_{\mathrm{k}}$ denotes the non-exergy related cost rate associated with component k. It results from the cost rates associated to capital investment, operating and maintenance. For each component, $\dot{Z}_{\mathrm{k}}$ would be known from a preliminary economic evaluation. The plant under consideration is made up of ten components, namely the air pre-heater P, the combustion chamber CC, the compression spaces $C1$ and $C2$, the inter-cooler IC, the recuperator R, the heater H, and the mechanical compression and expansion devices, so that Eq. (9) leads to a system of ten equations.

To solve this system, the cost of each component should have been assessed first. For the heat exchangers, we know the raw material cost (shell, tubes, ...), from pipe suppliers list prices, and we multiply this cost by an arbitrary factor to take all costs into account, including heat exchanger manufacturing cost... In our case, we multiply the raw material cost by (3). On the other hand, for the other components, we use an empirical relation such as:

$$P(X) = P(X)_{\mathrm{known}}\left(\frac{X}{X_{\mathrm{known}}}\right)^{\alpha} \tag{10}$$

where $P(X)$ is the searched cost of the device of characteristic value X; the couple (X_{known}, $P(X)_{\mathrm{known}}$) and the value of the exponent α are evaluated from data taken from [16]. The results are reported in Table 6. Note that the cost of all electric devices (generator, control, ...) has been combined with the cost of the expander, and the cost of the inter-cooler IC, not mentioned in Table 6, is 1.1 k€. The total cost of the system is 37.2 k€. It corresponds to less than 3500 € per kilowatt of electricity, which seems to be a promising result. However, it should be emphasized that very large uncertainties lie in the cost of the components.

As soon as the cost P_{k} of component k is determined, it is converted into cost per unit time, $\dot{Z}_{\mathrm{k}}$, assuming a 10 years useful life for each component with 8000 operating hours by year.

The linear system of ten equations (Eq. (9)) can now be solved for the ten unknown costs per unit exergy of stream c_i or work c_W.

Table 6
Cost, exergy efficiencies, exergy destructions and exergy destruction numbers

Component	Cost [k€]	Exergetic efficiency [%]	Rate of exergy destruction [W]	Exergy destruction number [%]
combustion chamber *CC*	1.2	71.2	10845	32.1
air pre-heater *P*	2.8	72.6	1869	5.5
heater *H*	12.8	90.9	1801	5.3
recuperator *R*	8.3	93.8	420	1.2
compressor C_1/C_2	2.3	95.2/90.1	544	1.6
mechanical system of compression	0.1	90.0	1270	3.8
expander *E*	7.2	99.6	107	0.3
mechanical system of expansion	1.4	90.0	2613	7.7
global system	37.2	38.5	19469	57.6

Table 7
Cost of the different exergy streams

Exergy stream	Cost [€·kWh^{-1}]
c_{g}	0.03
c_{pa}	0.09
c_{e}	0.05
c_{k1}	0.16
c_{k2}	0.20
c_{cr}	0.17
c_{rh}	0.17
c_{h}	0.11
c_{Wc}	0.15
c_{We}	0.12
c_{W}	0.13

Table 7 shows the results. The first line indicates the cost of the gaseous fuel, used as input data. The last line gives the cost of one kilowatt-hour of electric energy production. It is worth noting that this last result alone can be obtained in a much easier way, without the need of exergy analysis. Indeed, if we sum up the ten equations (Eq. (9)), all the exergy transfer rates in the system vanish, and we obtain a single equation with the single unknown c_W. However, the method used here gives a deeper insight into the system. For instance, it is interesting to point out that the cost of the pre-heated air is much higher than the cost of the gaseous fuel. The pre-heated air being obtained from energy from the fuel and by means of a combustion chamber and a pre-heater, the cost by unit exergy of the pre-heated air should obviously be higher than the cost per unit exergy of the fuel used to pre-heat it. The method developed here allows to quantify this cost and to conclude that the unit exergy recuperated through the air pre-heater is three times more expensive than the unit exergy contained in the fuel.

Each kilowatt-hour of electric energy production is accompanied by a (free) production of 1.2 kWh of recoverable thermal energy for heating or sanitary hot water purposes. If we compare the value of 0.13 €·kWh^{-1} with the electricity domestic prices (≈0.10 €·kWh^{-1} of electric energy) and the natural gas domestic price (≈0.03 €·kWh^{-1} of thermal energy) in France, we notice that these values are close and therefore, the studied system could be profitable. However, the cost of the various components of this micro-cogeneration system should be known more exactly to refine these results.

8. Conclusion

The results of the energy study, the exergy study and the cost study of a micro-cogeneration system are presented. The plant under consideration is based on an Ericsson engine, that is a Joule cycle reciprocating engine. The energy study allows to design and to size the plant and to analyze the influence of the heat exchangers areas distribution on the efficiency and the raw material cost of the heat exchangers. We note that it is very important to assess each heat exchanger effectiveness appropriately to obtain a good efficiency and to avoid huge transfer surfaces and prohibitive costs. From our analysis, it is possible to find a suitable compromise between efficiency, heat transfer area and cost of the heat exchangers. The exergy study allows to draw the Grassmann diagram and to observe the exergy streams and the exergy destructions in the system. Finally, the cost study allows to calculate the cost of the heat and of the electricity produced and to conclude to the possible profitability of the system by comparison with the domestic prices given by the French energy suppliers.

It should be emphasized that the assumptions used in this work are quite severe and conservative. For instance, the assumed mechanical efficiency is low, the valve pressure losses are quite high, the amount of heat rejected at the chimney is important. Nevertheless, the modeled system proves to be interesting. The main issue of this work is to put a new light on an old, but cheap, simple, and efficient technology and to draw the attention on Ericsson engines, unfairly not yet brought out of oblivion.

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