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Combined cycle plant efficiency increase based on the optimization of the heat recovery steam generator operating parameters

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Abstract

At the present time combined cycle (CC) power plants allow to meet the growing energy demand with the least fuel consumption. Thus it is of great interest to define a strategy for the optimization of these systems, in order to get greater performances and efficiency from them.

The purpose of the most part of the world manufactures involved in this sector is to reach overall thermal efficiency of 60% in the short term period, above all by improving the gas turbine inlet temperature. In the present work, it is shown how it could be possible to reach the same performances by best fitting the existing technology. In particular we point our attention to the optimization of the heat recovery steam generator (HRSG), as a first step in the analysis of the whole plant, according to a hierarchical strategy. We handle this problem adopting both a thermodynamic and a thermoeconomic objective function instead that with the usual pinch point method. Thermodynamic optimization has the purpose to diminish energy losses, expressed on exergy basis, while the aim of the thermoeconomic optimization is the minimization of a cost function, sum of the cost of exergy inefficiencies and the cost of the HRSG. Proposed methods have been applied to some HRSG configurations, including some present commercial plants. The results of the application of the thermoeconomic optimization leads to a meaningful increase of the thermal efficiency of the plant that approaches the 60%, obtained with and increases of the heat surface and a decrease of the pinch-points. The economic evaluations are referred to US dollars ($) even if the costs are referred to European scenario, assuming 1 US $ = 1.1 euro. © 2002 Éditions scientifiques et médicales Elsevier SAS. All rights reserved.

Keywords: Heat Recovery Steam Generators (HRSG); Thermodynamic optimization; Exergy analysis; Thermoeconomic optimization; Combined cycle power plant; Parallel flow heat exchanger; Gas side effectiveness

1. Introduction

The liberalization of the electric power market, leads to a stronger competition among electrical utilities. The necessity to reduce pollution due to greenhouse gases, make combined cycle (CC) power plants one of the best choice to produce energy, because of their high efficiency and the use of low carbon content fuels. CC plants couple the Brayton thermal cycle (top cycle) with the Hirn-Rankine one (bottoming cycle). The basic idea is to use the hot exhaust gas, available at the end of the expansion stage in the Brayton cycle, to produce hot high-pressure steam in the bottoming cycle. The element, where the steam heating takes place, is the heat recovery steam generator (HRSG) (Fig. 1).

Nowadays advanced combined cycle plants, employing 2 or 3 pressure levels reheat steam cycle and having exhaust gas temperature (T_e) ranging from 700 to 920 K, reach a thermal efficiency up to 58% [1,2]. But all of the main world manufacturers claim to achieve a 60% efficiency in the short period. This aim is pursued mainly by increasing the turbine inlet temperature [3], despite the considerable technological efforts necessary to cool down the blades of the gas turbine. No appreciable improvement instead seems to be reached by adopting more than three pressure levels steam cycles, as stated in [4].

In order to obtain high thermal efficiencies of CC plants by the existing technologies, attention can be fixed on the HRSG and on the bottoming cycle, by optimizing its performances. This can be done by best choosing its operating parameters: mass flow rates, temperature profiles, operating pressure values, heat exchangers efficiencies, etc. The operating parameters are the strictly necessary information needed to actually design the HRSG, in full details.

The purpose of this work is to evaluate the possibility to reach 60% of overall thermal efficiency for combined cycle, or however to improve their present performances,

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Nomenclature

b	specific cost parameter of the HRSG	m^{-2}
b'	dimensionless cost of the HRSG	
CC	combined cycle plant	
c_p	specific heat	$J \cdot kg^{-1} \cdot K^{-1}$
c_{pa}	specific heat of gas at ambient temperature	$J \cdot kg^{-1} \cdot K^{-1}$
D	HRSG economic life	years
$\Delta K = K_{optimum} - K_{actual}$	additional cost of the optimized HRSG with respect to the real one	\$
E	generic exergy flow	W
E_Q	exergy associated with a heat flow	W
E_{Qx}	exergy flow associated with a heat flow at the temperature T_x	W
E_{res}	exergy still owned by steam at the entry of the condenser	W
g	number of connections between water flows of non contiguous elements	
G	heat rate $= c_p \cdot M$	$W \cdot K^{-1}$
G_{min}	the lowest value of the heat rates of the fluids streaming in a heat exchanger	$W \cdot K^{-1}$
G_{max}	the highest value of the heat rates of the fluids streaming in a heat exchanger	$W \cdot K^{-1}$
GT	gas turbine	
h	specific enthalpy	$J \cdot kg^{-1}$
H	number of plant working hours per year	$h \cdot year^{-1}$
HRSG	heat recovery steam generator	
I	irreversibility flow or exergy loss flow	W
I_{tur}	exergy losses in the steam turbines	W
K_{HRSG}	HRSG cost	\$
k_i	cost of the single HRSG section per surface unit	$\$ \cdot m^{-2}$
K_i	cost of the single HRSG section	\$
k_I	unit exergy loss cost	$\$ \cdot W^{-1} \cdot h^{-1}$
K_I	exergy loss cost	\$
K_{tot}	total cost	\$
LTG, LTG′	long term gain	\$
m	number of congruence conditions (e.g., the mixing of two water streams)	
M	mass flow	$kg \cdot s^{-1}$
n	number of heat exchange sections in the HRSG	
N	number of gas-side thermal units $= US/G_g$	
NE	number of equations describing a HRSG	
NIV	number of independent variables of the optimization process	
NV	total number of variables	
Ntu	general number of thermal units $= US/G_{min}$	
p	pressure	bar
PP	pinch-point	K
Q	generic heat flow	W
s	specific entropy	$J \cdot kg^{-1} \cdot K^{-1}$
S	surface	m^{-2}
ST	steam turbine	
T	temperature	K
T_a	ambient temperature	K
T_e	inlet temperature of the gas to the HRSG (i.e., exhaust gas temperature)	K
U	overall convective heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
v	number of evaporators	
x_2	number of sections with two water parallel flows	
x_3	number of sections with three water parallel flows	
y	number of water side mass conservations ($\Delta M_{wk} = 0$)	
W	mechanical power (MW)	
W_{tot}	power output of the plant (MW)	
z	number of gas side mass conservations ($\Delta M_{gk} = 0$)	

Greek symbols

ΔT	average temperature difference	K
ε	traditional effectiveness of a heat exchanger	
η	gas-side effectiveness of a heat exchanger	
η_{tur}	isoentropic efficiency of a steam turbine	
η_{CC1}	optimum efficiency of the combined cycle plant	
η_{CC0}	actual efficiency of the combined cycle plant	
ρ	ratio between the heat rates of water and gas $= G_w/G_g$	
τ	discount rate	

Subscripts

comm	in the commercial configuration
e	economizer
energy	of the energy
fuel	of the fuel
g	gas
HRSG	heat recovery steam generator
in	inlet
l	of the liquid
o	reference or environmental conditions
out	outlet
rh	reheater or reheated steam
sh	superheater or superheated steam
tur	steam turbine
v	evaporator
w	water
(′ and ″)	distinguish quantities concerning each flow in a parallel exchanger

Superscript

$*$	dimensionless variables

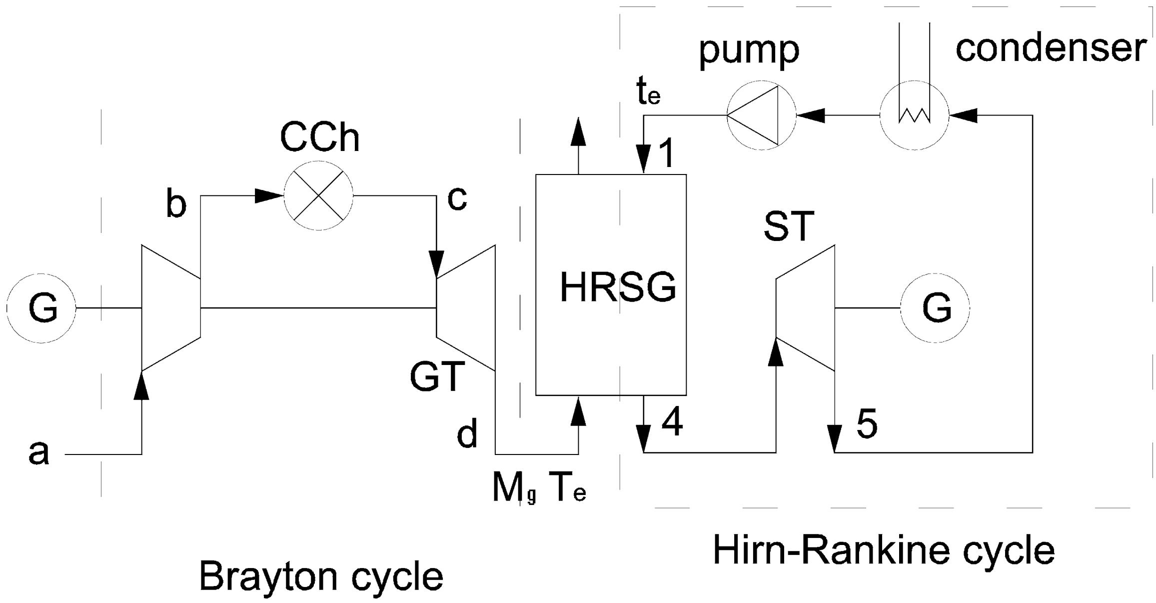

Fig. 1. Schematic representation of a combined cycle plant. (G: electrical generator; CCh: combustion chamber; GT: gas turbine; ST: steam turbine).

simply by the optimization of the operating parameters of the HRSG and consequently of the bottoming cycle. To accomplish this aim it has been proposed also the use of heat exchange sections with two or more parallel flows of water exchanging simultaneously with the same gas stream, and the removal of any constraint on the highest pressure of steam. In the parallel heat exchangers gas gives heat at the same time to two or more flows of water, carying out the same temperature increase. It is shown that the use of these exchangers is always advantageous. At the present time they are adopted only when the water streams have quite the same thermal properties (superheated and reheated steam for instance) [5], even if there is not any physical restriction to employ them also with streams of different nature (liquid and steam for instance). However in modern plants their presence is more and more spread.

The optimization of the HRSG represents only a first step in the optimum design of the whole plant, according to a hierarchical strategy that implies the following levels:

- the optimization of the gas turbine cycle;
- the optimization of the operating parameters of the HRSG;
- the detailed optimization of the single heat exchanger sections in the HRSG.

In the present work we suppose to fix a particular gas turbine, and we use its outlets (i.e., the flow mass M_g and the temperature T_e of the available flue gas) to find the optimum operative parameters of the HRSG.

The criteria chosen to carry out the optimization of the HRSG are two: a purely thermodynamic one and a thermoeconomic one, completely described in [6,7]. The former is useful to set up the proposed method and to stress some fundamental results, even if it is meaningless from a realistic point of view (e.g., it produces infinite exchange surfaces). The latter instead is able to supply actually useful results, both combining thermodynamic and economic aspects.

This method can also be considered an alternative solution for the usual method of the HRSG design based on the choice of the pinch-point (PP). The pinch-point, defined as the difference between the temperature of gas at the entry of the evaporator (side economizer) and the saturation temperature, is the main parameter that influences the dimension of the HRSG, and consequently its thermal performances and its equipment cost. The higher the value of the pinch-point, the worse the efficiency of the HRSG, but, on the other hand, the less its costs are. Usually the values of pinch-point are chosen according to the experience of the manufacturer, and range typically between 8 and 20 K [1,8]. With the present method, instead, the values of the pinch-point are the consequence of an optimization procedure strongly depending on the cost structure assumed, and does not rise from a more or less arbitrary choice.

Developed methods have been applied to a certain number of HRSG configurations, some of which present in existing modern plants. In particular the optimization method is applied in the present work on two commercialized CC plants, which represent two different strategies in order to get high performances. They are respectively produced by ABB–Alstom (2 pressure levels steam cycle, $T_e = 920$ K, overall efficiency of 57.6%) and by Siemens (3 levels pressure steam cycle, $T_e = 853$ K, overall efficiency of 57.1%). The former plant employs a higher T_e in order to reach a higher superheated steam temperature and thus a higher steam cycle efficiency [5]; the latter instead addresses its attention to the heat recovery process, attempting to increase its effectiveness.

2. Optimization criteria

The first step to satisfy in every optimization problem is to define the objective function. By minimizing (or maximizing) it, it is possible to determine the ideal configuration of the examined system. In this paper, two different optimization criteria are taken into account, so two different objective functions are needed. The first one is a measure of the thermal efficiency of the HRSG; the second one represents a total cost of the HRSG, sum of the cost of exergy losses and the cost of its sections.

2.1. Exergy concept and thermodynamic objective function

The efficiency of the HRSG can be expressed through the exergy losses that take place in the heat exchange [6]. The use of the exergy concept is necessary because by means of exergy it is possible to take into account both quantitative and qualitative aspects of thermal energy. Let's remember that the exergy is defined as the maximum quantity of work obtainable from a certain form of energy (thus with a reversible transformation) having the environmental parameters as reference [9].

According to this definition, the exergy flow E_{Q1} associated to a heat flow Q, available at the temperature T_1, with respect to the environment at temperature T_0, is:

$$E_{Q1} = Q\left(1 - \frac{T_0}{T_1}\right) \tag{1}$$

The higher T_1 is, with respect to T_0, the higher the mechanical power we can obtain from Q. This shows the necessity to define a measure of the performances of the HRSG based on the exergy concept: actually it is not equivalent to recover the same quantity of energy at different temperatures.

If a thermal energy flow Q, from a heat source at a constant temperature T_1, is transferred to a body at a constant temperature T_2, with $T_1 > T_2$, according to the first Law of Thermodynamics the process can happen without losses. The second Law of Thermodynamics states instead that the process is irreversible, and during it there is a degradation of energy. The exergy losses taking places during the process can be calculated as the drop of exergy of the system between the beginning and the end of the process. Thus, in this case, the exergy losses I is given by:

$$\begin{aligned} I &= E_{Q1} - E_{Q2} = Q\left(1 - \frac{T_0}{T_1} - 1 + \frac{T_0}{T_2}\right) \\ &= Q\left[\frac{T_0(T_1 - T_2)}{T_1 T_2}\right] \end{aligned} \tag{2}$$

Eq. (2) shows that the exergy loss I raises as the difference $T_1 - T_2$ between the two bodies (or streams) increases, and that this loss can be reduced but not completely avoided if the heat flow must happen. Eq. (2) is still meaningful even if the temperatures of the bodies are not constant: in this case it in necessary to use opportunely defined average temperature values. However it can be affirmed that, for a given thermal power transferred, the exergy losses are directly proportional to the temperature difference between the high and the low exchange temperature.

In a typical HRSG configuration, increasing the number of pressure levels in the steam cycle, the mean temperature difference between the hot fluid (gas) and the cold one (steam) can be reduced, thus decreasing also the inefficiencies of the process. Nowadays up to 3 pressure levels are used in advanced plants, further increase in the number of levels looking disadvantageous if compared to the complexity of the system.

Eq. (2) also explains the direct influence of the value of the pinch-point on the performances of the HRSG: the higher this value, the higher the temperature difference between the two streams, thus reducing the thermal exergetic efficiency. The irreversibility in the HRSG can be found by writing a certain number of equations like (2), each for every heat exchange section. More simply the irreversibility can also be calculated through an exergy flow balance extended to the whole HRSG, considered as an open system. In this case we have:

$$E_{g_{\mathrm{in}}} + \sum_{\mathrm{in}} E_{w_{\mathrm{in}}} = E_{g_{\mathrm{out}}} + \sum_{\mathrm{out}} E_{w_{\mathrm{out}}} + E_Q + I \tag{3}$$

In the practice, rearranging Eq. (3), where $E_{g_{\mathrm{out}}}$, E_Q and $\sum_{\mathrm{in}} E_{w_{\mathrm{in}}}$ are negligible, we obtain:

$$I = E_{g_{\mathrm{in}}} - \sum_{\mathrm{out}} E_{w_{\mathrm{out}}} \tag{4}$$

that represents the objective function that must be minimized for the thermodynamic optimization of the HRSG. The aforementioned terms are zero because the inlet temperature of water is equal to the environmental one, the exhaust gas leave the HRSG and is not more used, so the residual exergy of the gas can be considered null too. The exergy losses E_Q related to the heat exchange with the environment are considered negligible. Expression (4) is completely general, and can always be applied.

In the rest of this paper only the physical component of the exergy will be considered, assuming that the other components (kinetic, potential, chemical, etc ...) are negligible. A further hypothesis done, for simplicity, is to disregard losses due to pressure drop along the HRSG, which are negligible with respect to those related to heat rates. Thanks to the use of a performance measure based on exergy concept, however also pressure drop losses can be considered without many complications. But it is necessary to point out that a correct definition of the exergy losses due to the pressure drop requires a detailed description of the HRSG sections, not available at this step of the optimization procedure [7]. Eq. (4), defining the exergy loss, can be made in dimensionless terms, for convenience:

$$I^* = \frac{I}{M_{\mathrm{g}} c_{\mathrm{pa}} T_{\mathrm{a}}} = \frac{E_{g_{\mathrm{in}}} - \sum_{\mathrm{out}} E_{w_{\mathrm{out}}}}{M_{\mathrm{g}} c_{\mathrm{pa}} T_{\mathrm{a}}} \tag{5}$$

The dimensionless exergy losses I^* so defined, even if not meaningful on a practical point of view, permits a better control of the optimization procedure, because it will be the objective function and it assumes values in a well defined range, between 0.01 and 0.2.

2.2. Thermoeconomic objective function

As previously mentioned, Eq. (2) shows that the irreversibility flow reduces as the mean temperature difference $\Delta T = T_1 - T_2$ likewise decreases. In the hypothesis of mainly convective heat exchange (as in the HRSG with good approximation occurs), the heat flow through a surface S with global heat exchange coefficient U is:

$$Q = US\Delta T \tag{6}$$

It easy to understand that the same heat flow Q, for the same overall heat transfer coefficient U, can be hold, though reducing ΔT, only increasing the dimension of surfaces S, and then the cost of the HRSG. Thus thermoeconomic optimization gives the HRSG configuration which is the best compromise between thermodynamic efficiency and economic advantage. So the objective function is a total cost function, given by the sum of the costs of the HRSG and of exergy losses [7]:

$$K_{\text{tot}} = K_{\text{I}} + K_{\text{HRSG}} \tag{7}$$

2.2.1. Cost of exergy losses

Established the period of time to evaluate investment convenience, for instance the economic life of the plant, exergy loss cost K_{I} can be expressed:

$$K_{\text{I}} = k_{\text{I}} HDI \tag{8}$$

The choice of the term k_{I} is very sensitive: it represents the weight of thermodynamic efficiency (and thus of energy saving) with respect to the required economic investment. Exergy losses can be computed, as discussed in detail in [6, 7] as the selling price of the desired plant output, i.e., the electrical energy, about three times the cost of fuel per unit of energy.

2.2.2. Cost of the HRSG sections

While the cost of exergy losses can be defined quite simply, due to the availability of the required data, the definition of the HRSG cost structure is more difficult. HRSG total cost can be set as the sum of the costs of its single sections, so that:

$$K_{\text{HRSG}} = \sum_{i=1}^{n} K_i \tag{9}$$

Then every section cost can be considered proportional to the exchange surface, through a coefficient $k_i(p,T)$ that represents the surface unit cost and that depends on the working pressure and temperature, as these two parameters affect material and thickness choices:

$$K_i = k_i(p,T)S^x \tag{10}$$

x is a suitable exponent that has been set equal to 1, being this hypothesis conservative with respect to the common suggestion found in literature, that states to put $x = 0.8$. Observing realistic cost data from [10], it has been decided to simplify expression (10), considering 4 different types of exchange sections (economizers, evaporators, superheaters and reheaters) and k_i constant for every sections.

$$K_{\text{HRSG}} = \sum_{\text{e}} k_{\text{e}} S_{\text{e}} + \sum_{\text{v}} k_{\text{v}} S_{\text{v}} + \sum_{\text{sh}} k_{\text{sh}} S_{\text{sh}} + \sum_{\text{rh}} k_{\text{rh}} S_{\text{rh}} \tag{11}$$

Eq. (11), far from being the most realistic cost structure for the HRSG, represents a reasonable and simple model that allows doing suitable calculations. Nothing prevents from defining other cost structures, thanks to the separation of thermodynamic and economic analysis done.

2.2.3. Total cost function

By substituting Eqs. (11) and (8) in Eq. (7), and rearranging all the terms in order to give dimensions to the expression and to put in evidence the cost of the evaporators, we have:

$$K_{\text{tot}}^* = I^* + b\left(\sum_{\text{e}} k_{\text{e}}^* S_{\text{e}} + \sum_{\text{v}} S_{\text{v}} + \sum_{\text{sh}} k_{\text{sh}}^* S_{\text{sh}} + \sum_{\text{rh}} k_{\text{rh}}^* S_{\text{rh}}\right) \tag{12}$$

where:

$$k_{\text{e}}^* = \frac{k_{\text{e}}}{k_{\text{v}}} \qquad k_{\text{rh}}^* = \frac{k_{\text{rh}}}{k_{\text{v}}} \qquad k_{\text{sh}}^* = \frac{k_{\text{sh}}}{k_{\text{v}}} \tag{13a}$$

and

$$b = \frac{k_{\text{v}}}{k_{\text{I}} c_{\text{pa}} T_{\text{a}} \text{HDM}_{\text{g}}} \tag{13b}$$

where b represents the ratio between the cost of the reference exchange section (the evaporator) and the cost of exergy loss. It is interesting to observe that b, depends on the gas mass flow M_{g}, and decreases as the plant size gets greater. So we can expect that it will be more convenient improving the HRSG exergetic efficiency particularly on big size power plants.

3. Two useful tools for the HRSG analysis

The analysis of the HRSG is performed using also these two original concepts:

- the so-called gas-side effectiveness of a heat exchanger;
- the definition of the HRSG sections with two or more parallel streams of water.

3.1. The gas-side effectiveness of a heat exchanger

The gas-side effectiveness (see Fig. 2 for the symbols used) is defined as:

$$\eta = \frac{T_{g_{\text{in}}} - T_{g_{\text{out}}}}{T_{g_{\text{in}}} - T_{w_{\text{in}}}} \tag{14}$$

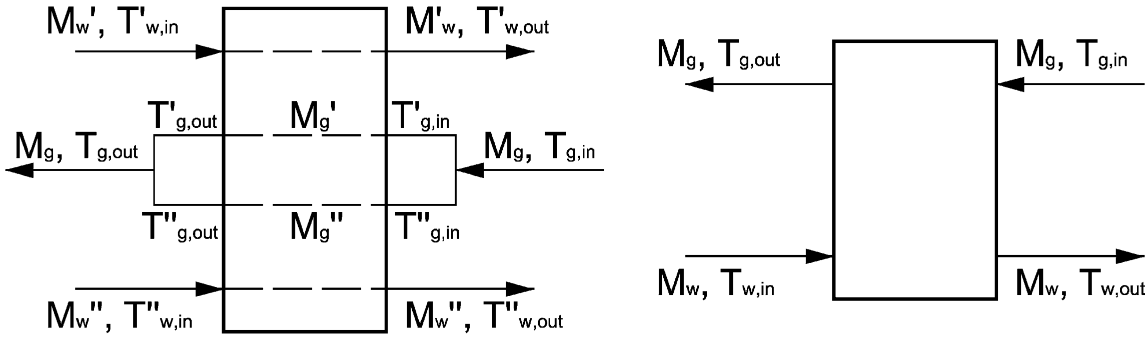

Fig. 2. Schematic representation of a simple flow heat exchanger (on the right) and of a parallel heat exchanger with two flows (on the left).

This definition is meaningful regardless of which of the two streams, of gas or of water, has the lowest heat rate, and allows to study the trend of the temperature profiles in the HRSG following the behaviour of a fluid (the gas) that does not change physical state.

The value of the traditional effectiveness ε of a heat exchanger, as it is defined in [11,12] can always be derived from the value of the gas-side effectiveness η through these simple expression:

$$\eta = \rho\varepsilon \quad \text{if } \rho < 1 \tag{15}$$

$$\eta = \varepsilon \quad \text{if } \rho \geqslant 1 \tag{16}$$

where:

$$\rho = \frac{G_\mathrm{w}}{G_\mathrm{g}} \tag{17}$$

and:

$$G_\mathrm{w} = M_\mathrm{w} c_\mathrm{pw}, \tag{18a}$$

$$G_\mathrm{g} = M_\mathrm{g} c_\mathrm{pg} \tag{18b}$$

Since $\rho \to \infty$ for the evaporators, their gas side effectiveness is always equal to their traditional effectiveness. Eqs. (15) and (16) are obtained from the definition of the traditional efficiency just by considering the thermal balance of the heat exchanger in the hypothesis that the heat rates are constant. For each type of heat exchanger, its traditional efficiency can be expressed as a function of two parameters: the number of transfer unit Ntu and the ratio between the heat rates of the two fluids $G = G_\mathrm{min}/G_\mathrm{max}$:

$$\varepsilon = f(\mathrm{Ntu}, G) \tag{19}$$

with:

$$\mathrm{Ntu} = \frac{US}{G_\mathrm{min}} \tag{20}$$

Eq. (19) allows to design the surface of a heat exchanger, and thus to actually design it, once fixed up its performance (through the coefficient ε and heat rates ratio G) and once known its overall thermal exchange coefficient U.

It can be shown that also for the gas-side effectiveness an expression similar to (19) can be written:

$$\eta = f(N, \rho) \tag{21}$$

where N is the number of gas-side thermal units:

$$N = \frac{US}{G_\mathrm{g}} \tag{22}$$

For instance, in the case of a counterflow heat exchanger, it is:

$$N = \frac{\rho}{\rho - 1} \ln\left(\frac{1-\eta}{1-\eta/\rho}\right) \tag{23}$$

Performing the actual optimization calculation, it has been assumed:

- to consider counterflow heat exchangers;
- to consider different mean values for the overall thermal exchange coefficients in the four characteristics HRSG sections.

With no further complications the proposed method can be applied also to other types of heat exchangers (i.e., crossflow ones).

3.2. *Heat exchangers with two or more parallel streams of water*

Parallel flow heat exchangers are exchangers in which gas gives heat to two or more water flows at the same time. Their schematic representation is given in Fig. 2. It is obvious that exergy losses due to the mix of the two streams of gas at the outlet of the exchanger can be reduced if their temperatures are equal (so that $T'_{g\mathrm{out}} = T''_{g\mathrm{out}} = T_{g\mathrm{out}}$); instead exergy losses due to the difference of pressure cannot be avoided.

This type of exchangers is very useful when two or more flows of water are required to accomplish the same temperature rise. In this case it is simple to demonstrate that exergy losses due to thermal exchange can be sensibly reduced, thanks to the lower mean temperature difference between gas and water, with respect to the configuration with two simple-flow exchangers in series (Fig. 3).

It is evident however that the presence of two or more separated streams of water is justified only if they are at different pressures. It is simple to demonstrate that the condition minimizing the exergy losses occurs when the two flows M_k' and M_k'' mix at the same temperature and the conditions

$$T'_{w_{\rm in}} = T''_{w_{\rm in}} = T_{w_{\rm in}} \quad \text{and} \quad T'_{w_{\rm out}} = T''_{w_{\rm out}} = T_{w_{\rm out}} \tag{24}$$

are simultaneously verified. So the section can be represented by the simplified scheme of Fig. 2 and it is possible to extend to the section with two water streams the same concept of the gas side effectiveness defined by Eq. (14). Then it can be simply verified that an univocal heat rates ratio can be defined, so that:

$$\rho = \frac{G'_{\rm w}}{G'_{\rm g}} = \frac{G''_{\rm w}}{G''_{\rm g}} \tag{25}$$

Remembering Eqs. (14) and (21), to define in an univocal way also the gas-side effectiveness for this type of exchangers it is necessary that a sole number of transfer unit exists, that is:

$$N = \frac{US'}{G'_{\rm g}} = \frac{US''}{G''_{\rm g}} \tag{26}$$

Given U' and U'' , fixed up M' and M'' (so that they satisfy the thermal balances), previous equation can always be verified with just an opportune choice of exchange surfaces S' and S''.

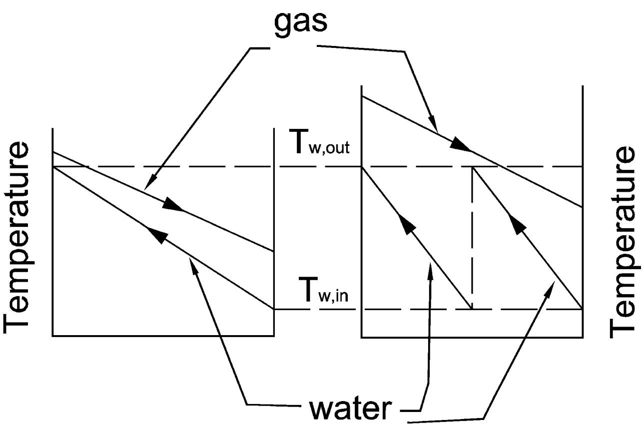

Fig. 3. Qualitative temperature profiles of two water streams accomplishing the same temperature rise: in a parallel heat exchanger (on the left) and in two simple flow exchangers in series (on the right).

The above conditions can be respected only when none of the water flows is changing phase. In general in an HRSG there can be parallel flows exchangers with as many water flows as the number of pressure levels.

In existing HRSGs, parallel heat exchangers are realized only with water flows of similar thermodynamic characteristics (i.e., with two liquid flows or with two steam flows). However there is no reason to avoid their realization with water flows of different characteristics (i.e., a liquid flow and a superheated steam) and not to take advantage of their useful presence in reducing exergy losses, whenever it is possible.

4. Analysis of HRSG

In order to carry out an optimization process it is first of all necessary to model the system and to establish the set of optimization variables. Each configuration of HRSG can be modelled using a certain number of elementary units, arranged in a proper way. These elementary components are:

- heat exchangers (single phase, simple flow and parallel flow, and evaporators);
- connections among the different elements (Fig. 4).

The thermodynamic properties of steam can be influenced also by the expansion phase in the turbines: for instance the conditions of steam at the beginning of reheating are coincident to those at the end of the expansion in the high-pressure turbine. Thus in this model we take into account also the presence of steam turbines. Their behaviour is described just by the isoentropic efficiency, taken $\eta_{\rm tur} = \text{constant} = 0.9$, apart from the pressure drop realized.

This modelization has been applied to some configurations of HRSG, ranging from those very simple (one level of pressure, no reheat), to some more complex, at present operating also in the most modern existing plants (two or three pressure levels with reheat). In details, the HRSG configurations examined are quoted in Table 1.

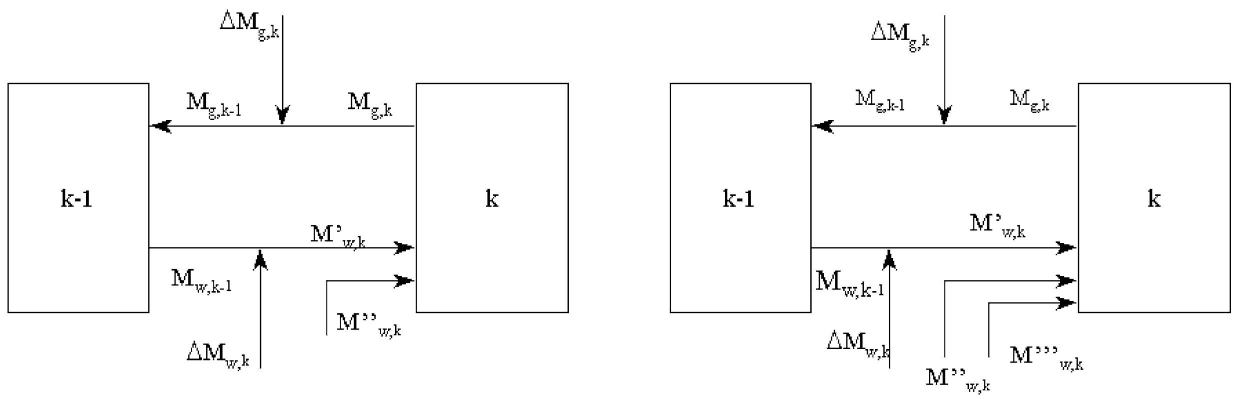

Fig. 4. Two examples of possible connections between two consecutive exchangers.

In Figs. 5 and 6 the diagrams concerning the plants 2PRH and 3PRH, obtained using the simple components above mentioned, are shown. The corresponding cycles are represented in the corresponding thermodynamic cycles of Figs. 7 and 8. These two configurations are employed also by two of the greatest manufacturers of combined cycle plants in the world: respectively ABB and Siemens. The actual plant diagrams are quite different from those of Figs. 5–8, mainly for the presence of parallel flow heat exchangers, that in the real case are used only in the sections with superheated and reheated steam. However it is of great interest to compare the results obtained with the considered optimization procedure with the actual performances of the real plants, to see if there can be improvements.

Table 1
Configurations of HRSG examined

Number of pressure levels	Main thermal cycle properties	Cycle name and abbreviation
1	Superheated steam	Hirn – H
	Reheat and reheater inlet temperature higher than saturation temperature	Hirn with reheat 1 – HR1
	Reheat and reheater inlet temperature lower than saturation temperature	Hirn with reheat 2 – HR2
2	High pressure superheated, low pressure saturated, expansion in just one steam turbine	Hirn–Rankine coupled – HRK1
	High pressure superheated, low pressure saturated, expansion in two different steam turbine	Hirn-Rankine uncoupled – HRK2
	High pressure reheated, low pressure superheated	2 pressure reheat – 2PRH
3	High pressure reheated, intermediate and low pressure superheated	3 pressure reheat – 3PRH

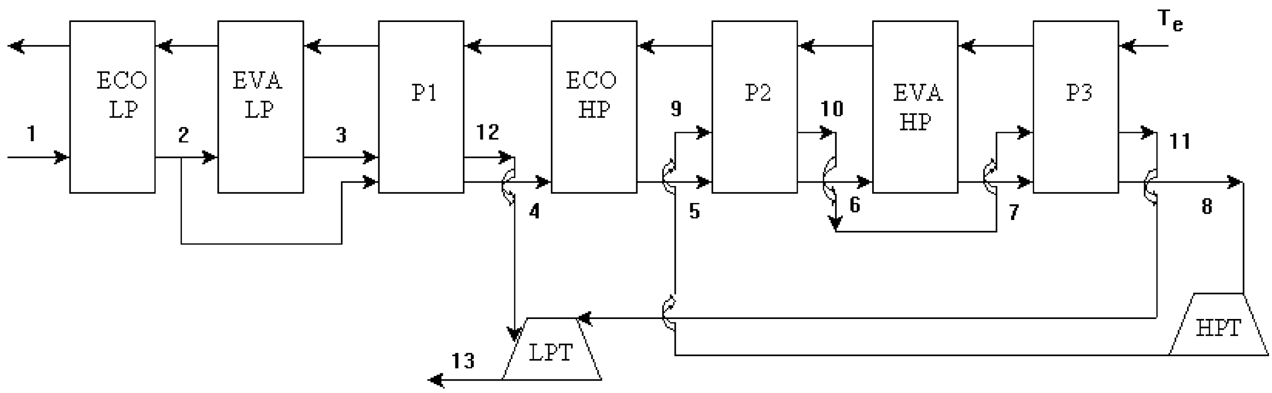

Fig. 5. Plant diagram of the two pressure levels with reheat steam cycle (2PRH).

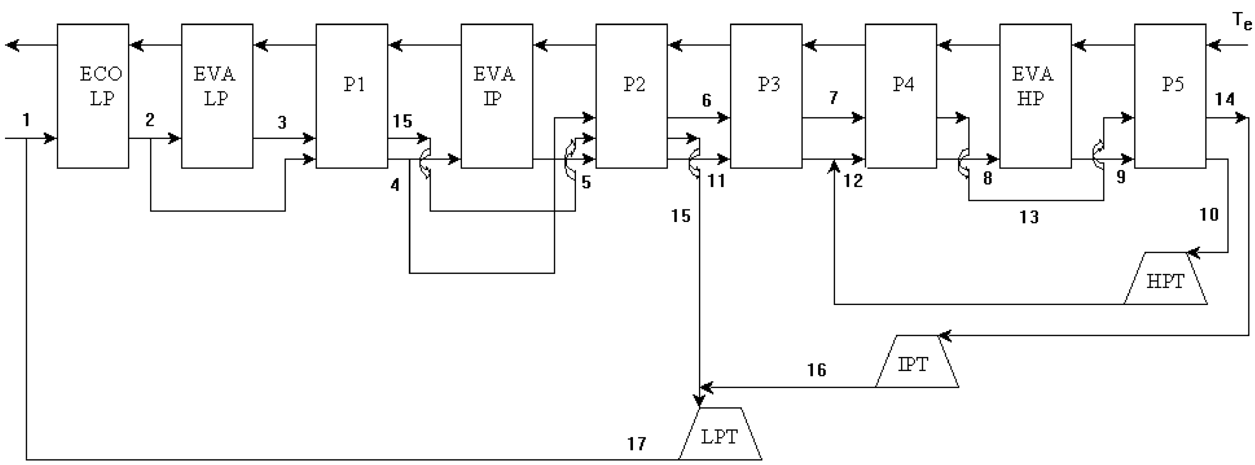

Fig. 6. Plant diagram of the three pressure levels with reheat steam cycle (3PRH).

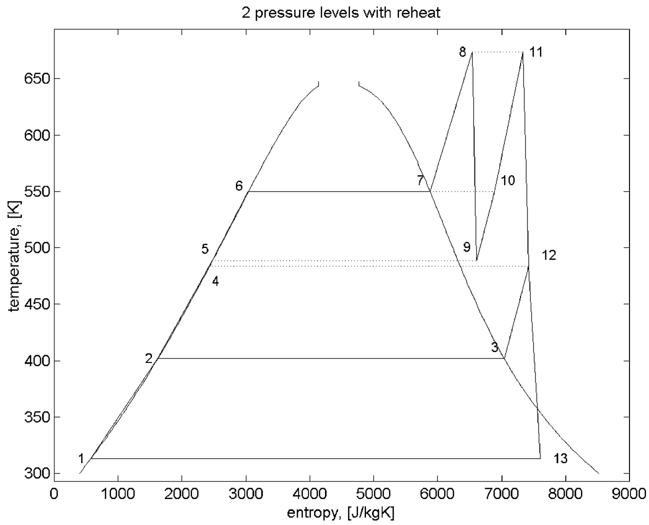

Fig. 7. 2PRH steam cycle on the T–S plane.

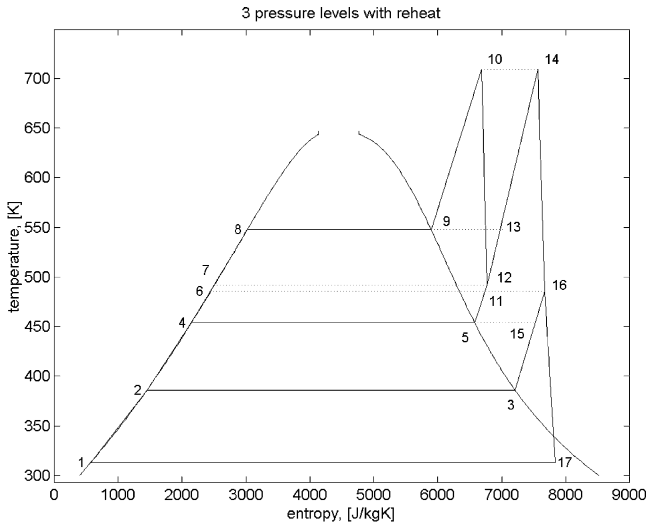

Fig. 8. 3PRH steam cycle on the T–S plane.

Each elementary unit, that forms the HRSG, is physically described by a certain number of equations and variables. Only on the basis of the HRSG configuration, thus it is possible to foresee the total number of free parameters (NIV), as difference of the number of equations (NE) from the number of variables (NV):

$$\mathrm{NIV} = \mathrm{NV} - \mathrm{NE} \tag{27}$$

The NIV free parameters are the control levers, operating on which it is possible to optimize the system. In general the equations that describe an elementary unit are: heat flow balances, mass flow balances and the definitions of gas-side effectiveness.

The variables instead are: gas and water temperatures, gas and water mass flows, gas-side effectiveness. Moreover there are also some further conditions that must be respected to correctly describe the HRSG. Remembering that the gas and water inlet temperatures and the mass gas flow are given data of the problem, the number of independent variables of the optimum design problem is:

$$\mathrm{NIV} = 3n - 1 + x_2 + 2x_3 - (v + g + z + y + m) \tag{28}$$

with the variables defined in the nomenclature. For instance, for the 2PRH and 3PRH HRSG we have, respectively:

$$n = 7, \quad v = 2, \quad x_2 = 3, \quad y = 5, \quad z = 6$$
$$g = 3, \quad m = 1 \quad \text{so that NIV} = 6$$

and:

$$n = 9, \quad v = 3, \quad x_2 = 4, \quad x_3 = 1, \quad y = 7$$
$$z = 8, \quad g = 5, \quad m = 2 \quad \text{so that NIV} = 7$$

Notice that HRSGs up to three pressure levels are considered, according to the actual production.

4.1. Mathematical solution of the optimization problem

From a mathematical point of view the problem stated is:

$$\text{Find} \quad \min\big[F(\boldsymbol{X}, \boldsymbol{Y})\big] \tag{29}$$

with the constrains:

$$\boldsymbol{Y} - \boldsymbol{f}(\boldsymbol{X}) = \boldsymbol{0} \tag{30}$$

$$\boldsymbol{g}_1(\boldsymbol{X}) \leqslant \boldsymbol{0} \tag{31}$$

$$\boldsymbol{g}_2(\boldsymbol{Y}) \leqslant \boldsymbol{0} \tag{32}$$

where $\boldsymbol{X}$ is a vector containing the NIV independent parameters, and $\boldsymbol{Y}$ a vector containing the NV–NIV dependent variables. F is the objective function, given by (5) or by (12).

Eqs. (31) represent the set of bilateral non-linear constrains binding $\boldsymbol{Y}$ to $\boldsymbol{X}$ (e.g., the full set of equations describing the HRSG), while (31) and (32) are conditions that stand for the physical sense of all the variables (e.g., temperature and mass flows greater than zero). In actual calculations, steam saturation temperatures, superheated steam temperatures and evaporators effectiveness were chosen as independent variables.

This problem of non-linear constrained minimization has been solved, after a linearization, by using the simplex method [13,14]. The main advantages in using the simplex method are:

- it does not require the evaluation of the derivatives of the function F;
- it does not require that F be continuous;
- it does not require to know the dominion of existence of F.

On the other hand it is not very effective in terms of calculus quickness, above all as the number of the variables increases, but it works quite good in the present case. Fig. 9 shows the general algorithm diagram used to solve the optimum design problem.

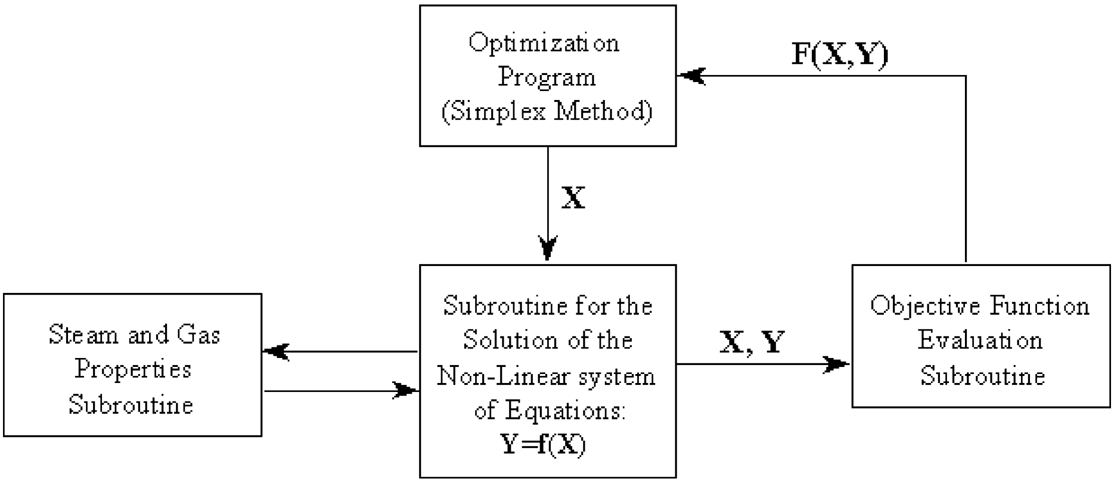

Fig. 9. Algorithm diagram used to solve the minimization problem considered.

Table 2
Results of thermodynamic optimization with $T_e = 773$ K

		H	HR1	HR2	HRK1	HRK2	2PRH	3PRH
	I^*	0.1749	0.1712	0.1389	0.09605	0.08428	0.05806	0.04892
HP	p (bar)	46.5	21.9	220	125	162	217	220
	T_{sh} (K)	600	773	762	743	773	747	773
	m (kg·s^{-1})	56.3	43.7	39.9	43.9	41.4	40.4	37.6
IP	p (bar)	–	–	–	–	–	–	32.5
	T_{sh} (K)	–	–	–	–	–	–	512
	m (kg·s^{-1})	–	–	–	–	–	–	3.78
LP	p (bar)	–	–	–	14.2	6.44	2.78	4.21
	T_{sh} (K)	–	–	–	sat	sat	410	512
	m (kg·s^{-1})	–	–	–	13.9	19.3	16.1	9.50
	W (MW)	52.0	53.1	58.3	58.3	61.1	65.0	66.7

5. Analysis of the optimization results

Calculations have been performed with the value of some parameters fixed as in Section 2.2.3. In particular, as it regards the flow mass M_g of gas, the chosen value is equal to that used in the ABB plant. Sensitivity analysis has shown that M_g has only a scale effect on the extensive variables, while it does not affect the intensive ones (i.e., temperatures and pressures). The results obtained can thus be considered general respect to the mass flow of the gas M_g. Other parameters that have been fixed are: the isoentropic efficiency of the steam turbines ($\eta_{tur} = 0.9$), and the minimum steam quality allowed at the end of expansion in the condensation turbines ($x_{min} = 0.8$).

5.1. Results of the thermodynamic analysis

Tables 2 and 3 summarize the main operative parameters of the steam cycle for all the HRSG configurations considered and for two values of the gas entry temperature ($T_e = 773$ and $T_e = 823$ K) setting:

$M_g = 386.7$ kg·s^{-1} (reference mass flow of gas)

$T_a = 293$ K (reference ambient temperature)

$c_{pa} = 1.0645$ kJ·kg^{-1}·K^{-1} (gas specific heat at ambient temperature for a standard gas composition considered by volume of 75% N_2, 4% CO_2, 11% O_2, 10% H_2O)

$T_{lin} = 313$ K (inlet temperature of the liquid to the HRSG)

Observing these data, two elements arise among the others:

- exergy losses I^* decrease as the complexity of the steam cycles increases;
- increasing the gas entry temperature T_e the steam high pressure gets greater for all the cycles, and reaches the highest allowed value, the critical one (about 220.4 bar).

The first result is quite expected, as it has been already told concerning to the number of pressure levels. Moreover, comparing data from the HR, HR1 and HR2 cycles, it can be stressed also that the configuration with two parallel type heat exchangers is more efficient than the others, proving the benefit of the use of parallel flow heat exchanger. Comparing instead data of the HRK1 and HRK2 cycles, it can be noticed

Table 3
Results of thermodynamic optimization with $T_e = 823$ K

		H	HR1	HR2	HRK1	HRK2	2PRH	3PRH
	I^*	0.9069	0.1023	0.08866	0.07214	0.03064	0.04321	0.03025
HP	p (bar)	220	220	220	212	220	215	220
	T_{sh} (K)	823	822	817	794	823	808	823
	m (kg·s^{-1})	58.7	57.4	46.6	58.2	59.0	45.9	45.6
IP	p (bar)	–	–	–	–	–	–	16.8
	T_{sh} (K)	–	–	–	–	–	–	477
	m (kg·s^{-1})	–	–	–	–	–	–	1.19
LP	p (bar)	–	–	–	22.4	1.96	1.33	1.06
	T_{sh} (K)	–	–	–	sat	sat	462	477
	m (kg·s^{-1})	–	–	–	5.27	9.01	12.35	9.05
	W (MW)	75.0	73.8	76.1	73.5	79.2	79.6	81.3

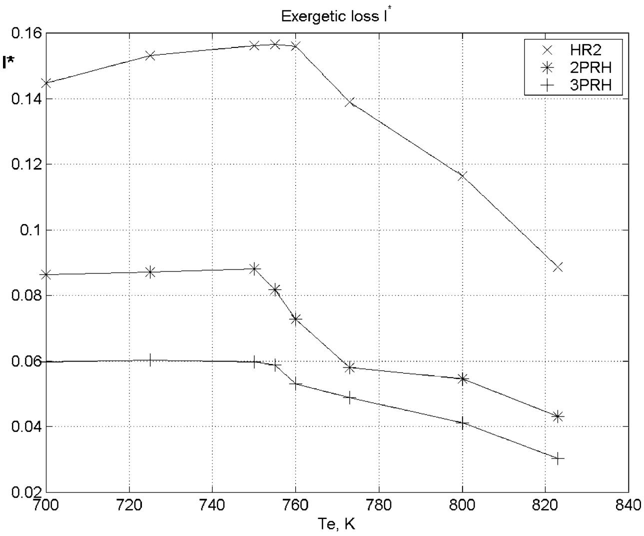

Fig. 10. Exergetic loss I^* vs. gas entry temperature T_e diagram for the HR2, 2PRH and 3PRH cycle.

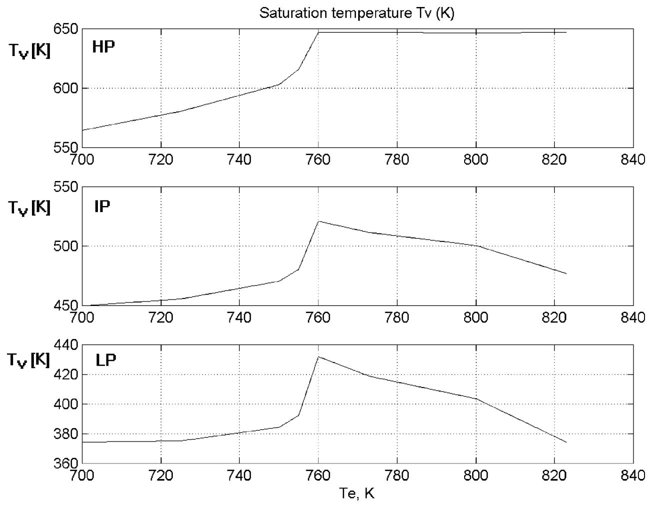

Fig. 11. Saturation temperatures T_v vs. gas entry temperature T_e diagram for the 3PRH cycle.

that losses bound to the second configuration are always lower than the others. This can be explained because the HRK1 cycle has one more constraint with respect to the HRK2 cycle (the condition of mixing of steam from high pressure expansion and steam saturated at low pressure). This constraint restricts the set of possible solutions, and the minimum found is just an allowable minimum and not the absolute one. Fig. 10 provides the diagram of exergy losses vs. the gas inlet temperature, respectively for the HR2, 2PRH and 3PRH cycles. It shows once more that the more advanced configurations permits low exergy loss, while in Fig. 11 we can see the behaviour of the saturation temperature with respect to T_e for the particular case of 3PRH HRSG configuration. The most meaningful fact, that can be observed by this diagram, is that there is a value of the HRSG inlet gas temperature (or at least a small interval) at which the high pressure steam turns near the critical conditions (saturation temperature is almost equal to 647 K, upper limit allowed). The transition towards the critical condition happens in a sudden way, while at the same time the exergetic loss begins to reduce, after having reached a maximum. This is true not only for the three cycles HR2, 2PRH and 3PRH, as Fig. 11 refers, but also for all the other cycles considered. Table 4 quotes the gas entry temperatures that bring the steam cycles taken in account near the critical conditions.

This behaviour can be easily explained. By the critical conditions for water, when the latent heat of vaporization is very small, the heat exchange stage at constant temperature nearly lacks, and the temperature profiles of water and gas get very close thus reducing losses.

It must be stressed that the tendency to go to the critical conditions is common to all the cycles when the gas entry temperature exceeds certain values. At least from a thermodynamic point of view, this can mean two matters. The first is that if it is not accepted the possibility to realize supercritical cycles, it is not then useful to arise the gas entry temperature too much, thus avoiding all the problems concerning with the cooling down of the blades of the gas turbine. The second matter is that perhaps water is not the most suitable fluid for all the gas temperatures,

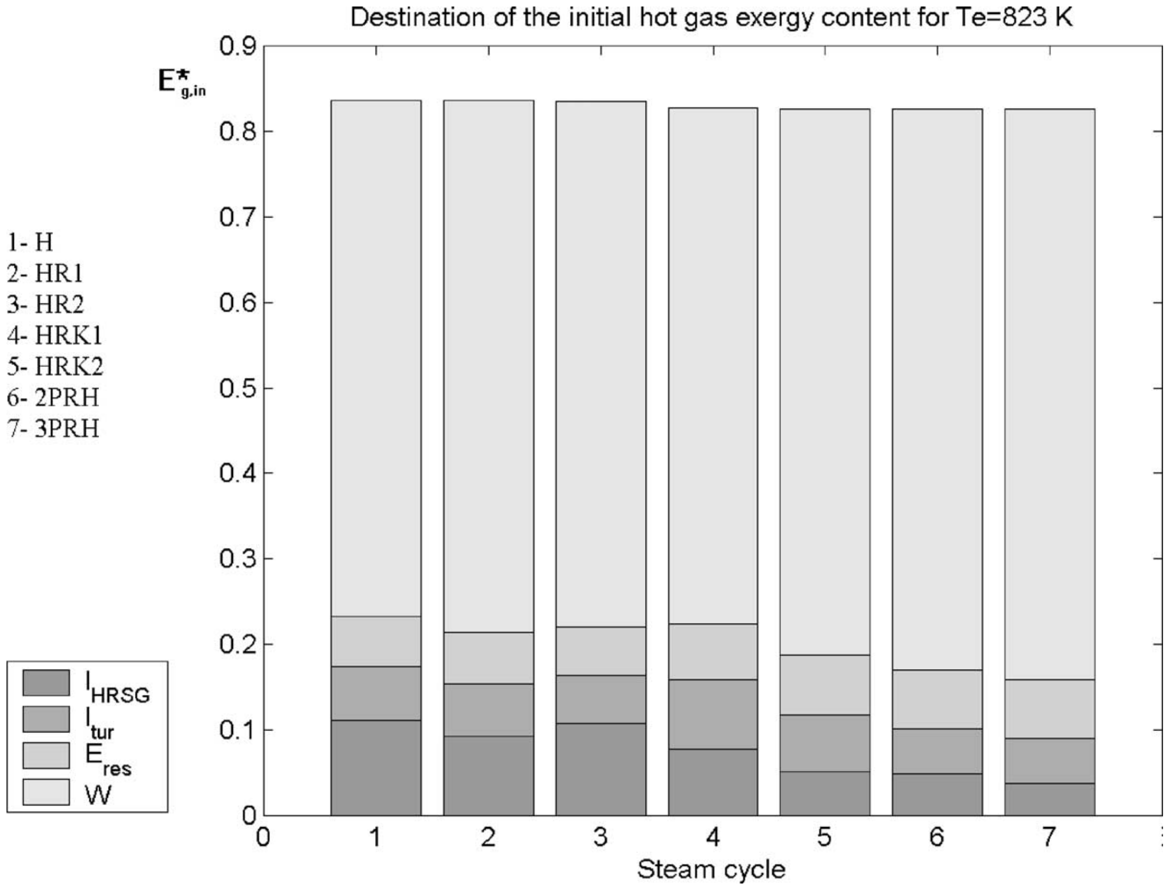

Fig. 12. Destination of the initial hot gas exergy content for $T_e = 823$ K.

Table 4
Inlet gas temperatures T_e that bring the various steam cycles towards critical conditions

HRSG configuration	T_e (K)
H	815–823
HR1	780–790
HR2	765–773
HRK1	830
HRK2	823
2PRH	720–725
2PRH	755–760

but only for those under a certain threshold. Obviously these considerations can be modified by the economical analysis, but they are justified from the thermodynamic one.

A further specification must be done about the apparent inconsistency between the value of the exergetic loss and the value of power output W in some cases (inconsistency is well clear for instance in the Tables 2 and 3, comparing data concerning HR2 and HRK1). However, we must remember that the exergetic loss in the HRSG is just a part of the overall losses that take place in the whole plant, and that affect the power output. The main other components of the overall losses are due to the irreversibility of the expansion in the steam turbines (I_{tur}), and to the residual exergy content of steam at the entry of the condenser (E_{res}), exergy that is fully wasted. Fig. 12 is a graphical representation for $T_e = 823$ K of the exergy balance of the transformation of the initial exergy quantity available in hot gases, taking in account all losses reasons above mentioned. In this figure the total height of each column is equal to the initial exergy content of gas (which depends only on T_e). The little differences in the height of the columns are due just to the calculus approximations.

5.2. Results of the thermoeconomic analysis

The thermoeconomic optimization can be carried out considering the general HRSG cost expressed by Eq. (12) and searching for the set of parameters that minimize it. In order to have a simpler evaluation it is possible to consider the annual cost of the HRSG and to express the surfaces in term of the number of transfer units N of the single sections. In this way it is possible to write

$$K^* = I^* + b'\left(\sum_{e} k_e^* \frac{c_{pe}}{c_{pa}} \frac{U_v}{U_e} N_e + \sum_{v} \frac{c_{pv}}{c_{pa}} N_v + \sum_{sh} k_{sh}^* \frac{c_{psh}}{c_{pa}} \frac{U_v}{U_{sh}} N_{sh} + \sum_{rh} k_{rh}^* \frac{c_{prh}}{c_{pa}} \frac{U_v}{U_{rh}} N_{rh}\right) \tag{33}$$

Table 5
Cost structure for the HRSG sections

	Economiser	Evaporator	Superheater	Reheater
Specific cost [$\$\cdot m^{-2}$]	$k_e = 45.7$	$k_v = 34.9$	$k_{sh} = 96.2$	$k_{rh} = 56.2$
Dimensionless specific cost	$k_e^* = 1.31$	$k_v^* = 1$	$k_{sh}^* = 2.74$	$k_{rh}^* = 1.61$
Heat transfer coeff. [$W\cdot m^{-2}\cdot K^{-1}$]	$U_e = 42.6$	$U_{rh} = 43.7$	$U_{sh} = 50.0$	$U_{rh} = 50.0$

Table 6
Results of thermoeconomic optimization with $T_e = 773$ K

		H	HR1	HR2	HRK1	HRK2	2PRH	3PRH
	K_{tot}^*	0.1843	0.1812	0.1723	0.1109	0.1018	0.08680	0.07593
	I^*	0.1764	0.1728	0.1464	0.09887	0.8813	0.06639	0.06907
HP	p (bar)	46.6	20.5	220	121	153	214	178
	T_{sh} (K)	601	768	752	742	764	744	767
	m (kg·s^{-1})	56.1	43.7	40.2	43.4	41.7	39.3	32.9
	PP (K)	0.7	0.6	3.7	2.2	1.3	3.1	2.1
IP	p (bar)	–	–	–	–	–	–	22.1
	T_{sh} (K)	–	–	–	–	–	–	491
	m (kg·s^{-1})	–	–	–	–	–	–	10.2
	PP (K)	–	–	–	–	–	–	2.9
LP	p (bar)	–	–	–	13.4	6.3	3.2	2.4
	T_{sh} (K)	–	–	–	sat	sat	434	491
	m (kg·s^{-1})	–	–	–	14.3	18.9	16.0	9.22
	PP (K)	–	–	–	0.9	1.3	2.4	1.2
	W (MW)	51.9	52.9	57.5	57.9	60.7	64.1	65.2

where

$$b' = \frac{k_v}{k_I t_a H D U_v} \tag{34}$$

and the ratio between the specific heat of the gas, $c_{p,i}$, and the reference specific heat, c_{pa}, varies in the range between 1 and 1.12. For what concerns all the economic evaluations, all the used data are reported in US \$, but the cost structure is related to an European scenario, assuming 1 US \$ = 1.1 euro. Ultimately, in the next calculations, k_I has been set equal to 6.8×10^{-5} $\$\cdot W^{-1}\cdot h^{-1}$, which corresponds about to the selling price of electrical energy in Italy. For what concern the HRSG section costs, various structures have been tested. In the rest of the paper, only as example for the application of the thermoeconomic optimization procedure, a feasible structure, obtained from [10] and reported in Table 5 has been assumed. In addition Table 5 provides also the values of the overall heat transfer coefficient. From Table 5 it can be observed that the cost of superheater section is sensibly higher than that of the reheater, because it is considered that the former operates at higher pressure. Moreover, in addition with the data already defined in Section 5.1, it has been assumed:

$H = 8000$ h·year^{-1} (working hours per year)

$D = 10$ years economic lifetime of the plant (depreciating plant period)

In the same way as in the thermodynamic analysis, Tables 6 and 7 report the main characteristics for all the cycles considered.

At a first glance it can be immediately observed the appearance of the pinch-points (PP), that are the most evident proof that the heat exchange surfaces (in particular of the evaporators), and thus also the costs, become finite. The values of the pinch-point obtained are quite smaller than those usually employed by all the manufacturers, that commonly range between 5 and 15 K. Two reasons can explain this fact, reasons that can be both true at the same time:

- the commonly used values of the pinch-points are not optimized, and thus there is a certain margin of reduction to improve the thermodynamic efficiencies with an economic convenience;
- the cost hypotheses done do not match completely the real cases, or they are too favourable to the energy saving.

The second reason is confirmed also by the fact that, as it can be noticed by the tables, the exergy losses costs are predominant with respect to the total costs (I^* and K_{tot}^* are directly comparable because they are homogeneous quantities). Increasing the weight of the cost of the HRSG respect to that of the exergy losses, it has be found that actually the values of the pinch-point increase too.

This shows that there is a strong link between the choice of the pinch-points and the economic analysis, so that they should not be chosen only on the basis of a custom or of the past experience of the manufacturer, but after a

Table 7
Results of thermoeconomic optimization with $T_e = 823$ K

		H	HR1	HR2	HRK1	HRK2	2PRH	3PRH
	K^*_{tot}	0.1433	0.1058	0.1355	0.09599	0.07894	0.07323	0.06254
	I^*	0.1117	0.09250	0.1075	0.07802	0.05120	0.04837	0.03716
HP	p (bar)	219	220	220	212	217	216	220
	T_{sh} (K)	821	760	801	794	820	800	808
	m (kg·s^{-1})	57.1	58.8	46.7	56.4	56.0	46.0	45.0
	PP (K)	0.3	0.6	7.1	5.5	1.8	4.7	9.0
IP	p (bar)	–	–	–	–	–	–	22.1
	T_{sh} (K)	–	–	–	–	–	–	491
	m (kg·s^{-1})	–	–	–	–	–	–	2.89
	PP (K)	–	–	–	–	–	–	2.3
LP	p (bar)	–	–	–	22.4	3.0	1.8	1.6
	T_{sh} (K)	–	–	–	sat	sat	486	487
	m (kg·s^{-1})	–	–	–	7.28	11.6	11.3	8.83
	PP (K)	–	–	–	1.0	1.9	2.1	1.9
	W (MW)	72.8	75.0	74.1	72.8	77.0	79.2	80.5

Table 8
Results of thermoeconomic optimization with $T_e = 823$ K and cost of exergy losses reduced of a factor 1.5 with respect to the case of Table 7

		H	HR1	HR2	HRK1	HRK2	2PRH	3PRH
	K^*_{tot}	0.1566	0.1168	0.1469	0.1043	0.0914	0.0833	0.0741
	I^*	0.1171	0.0925	0.1127	0.0803	0.0689	0.0516	0.0424
HP	p (bar)	218	220	220	212	213	216	220
	T_{sh} (K)	821	754	798	794	817	795	805
	m (kg·s^{-1})	56.7	58.9	46.7	55.8	54.8	46.0	44.4
	PP (K)	0.7	0.7	10.0	7.4	2.6	7.1	12.6
IP	p (bar)	–	–	–	–	–	–	21.3
	T_{sh} (K)	–	–	–	–	–	–	489
	m (kg·s^{-1})	–	–	–	–	–	–	3.6
	PP (K)	–	–	–	–	–	–	2.6
LP	p (bar)	–	–	–	22.4	3.7	2.1	1.7
	T_{sh} (K)	–	–	–	sat	sat	494	489
	m (kg·s^{-1})	–	–	–	7.9	12.5	11.2	8.7
	PP (K)	–	–	–	1.5	3.5	3.1	2.6
	W (MW)	72.2	75.0	73.5	72.5	76.0	78.7	79.9

process of optimization. The values found, far from being really the optimum, show that the proposed method works and gives suitable results. Thanks to the separation of the economic and thermal analysis, moreover, it is possible to consider different economic backgrounds, so that the method is completely general.

Another remarkable fact, that can be deduced from data of Tables 6 and 7, is that the saturation temperatures are not meaningfully different from those corresponding in the thermodynamic case. In particular it is not changed the behaviour to go towards the critical conditions as the gas temperature arises. At a first glance this result could surprise, because it was expected to find saturation pressures more similar to those actually employed in real plants. But at a further consideration it is clear that this is due only to the cost structure assumed, according to which the cost of heat exchangers per unit of surface are constant average values. Thus from an economic point of view it is indifferent to have steam at 160 bar of pressure, considered in general as a technological limit, rather than at 220 bar, so that the choice of the pressure values is left only to the thermodynamic optimum. It has been verified that, using alternative cost functions to take into account in a more detailed way also the influence of the pressure, it is possible to obtain some different results. A complete review of the results obtained for different cost structures can be found in [15]. It must be stressed, however, that any economic hypothesis can shift towards a higher point the temperature of gas at which the steam cycle becomes critic, but it cannot completely cancel this trend. An analogous result can be obtained if the cost of the exergy losses k_I, defined in Section 5.2, is reduced of a factor of 1.5 so that $k_I = 45$ \$·(MWh)$^{-1}$. Table 8 provides

Table 9
Comparison between data from ABB and Siemens plants and the results from optimization calculus

		2PRH ($T_e = 920$ K, $M_g = 386.7$ kg·s^{-1})			3PRH ($T_e = 852$ K, $M_g = 653.1$ kg·s^{-1})		
		ABB	tmd	tme	Siemens	tmd	tme
	K^*_{tot}	–	0	0.07344	–	0	0.07712
	I^*	–	0.03737	0.4248	–	0.03412	0.05882
HP	p (bar)	164	220	220	117	217	216
	T_{sh} (K)	836	829	820	813	852	835
	m (kg·s^{-1})	58.9	67.3	68.0	65.7	79.8	76.4
	PP (K)	–	4.6	5.3	–	0	14.6
IP	p (bar)	–	–	–	15.2	17.4	28.2
	m (kg·s^{-1})	–	–	–	22.1	3.96	8.22
	PP (K)	–	–	–	–	0	8.5
LP	p (bar)	6.9	1.6	2.2	3.7	1.2	2.2
	T_{sh} (K)	593	sat.	403	615	503	504
	m (kg·s^{-1})	11.4	4.53	4.10	8.92	14.3	16.3
	PP (K)	–	0	0.3	–	0	4.2
RH	p (bar)	38.2	69.7	73.2	14.6	17.4	28.2
	$T_{rh,in}$ (K)	–	647	647	573	502	542
	$T_{rh,out}$ (K)	835	829	820	813	852	835
	PP (K)	58.9	67.3	68.0	82.9	83.8	84.6
	W (MW)	97.0	106.4	105.9	131	150	146
	W_{CC} (MW)	280.0	289.4	288.9	380	399	395
	η_{CC}	57.6	59.5	59.4	57.1	59.9	59.5

the data of a thermoeconomic optimization, that are not sensibly different for those of Table 7 even if the pinch point are increasing.

5.3. Comparison between 2PRH and 3PRH cycles in their real applications

As it has been previously mentioned, the optimization programs were run on the 2PRH and 3PRH HRSG configurations also using the same boundary conditions that are characteristic of two existing plants that realize these cycles, respectively produced by ABB and Siemens. It must be stressed however that the HRSG layouts here adopted are quite different from those really existing, above all for the presence of the parallel type exchanger sections anywhere it is possible. Moreover in the ABB plant there is also a supplementary heat introduction in the HRSG, from the cooling down of the gas turbine blades, that it is not considered in our scheme.

Table 9 shows the available data from the two real plants and the results obtained. The main difference we can found both for the 2PRH and 3PRH cycles is about the value of the highest pressure, as it was already expected. It can be supposed that all the other differences that arise from the comparison (above all the pressures and the flow rates in the IP and LP sections) are a consequence of this primary characteristic, that influences the properties of the thermal cycles deeply. It is quite surprising that the optimum 2PRH cycles in the thermodynamic optimization requires a non-null HP pinch-point, but this is probably due to the high value of the gas inlet temperature ($T_e = 920$ K) and to the limit critical temperature of water (647 K), that cannot be exceeded. The thermoeconomic optimization seems to promise an increase of the power output of about the 9% for the 2PRH cycle and of 11% for the 3PRH one, with respect to their real applications, thus bringing the overall efficiencies of the plants from 57.6% to 59.4% in the former case, and from 57.1% to 59.5% in the latter one. This shows the possibility to reach efficiencies close to 60%, as sought by all the manufacturers, by only best fitting the existing technology.

In Table 10 are shown the values of the surfaces and of the gas-side effectiveness for all the heat exchangers of the 2PRH and 3PRH HRSGs configurations. It can be observed that the optimization calculus leads to negligible surfaces of some of the heat exchangers, so that their presence could be avoided in a real manufacturing.

The size obtained for the heat exchangers is quite realistic, as the corresponding HRSG cost. Under the hypothesis done, the optimized 2PRH HRSG costs 20.4 million of dollars, while the cost of the optimized 3PRH HRSG is 20.3 million dollars. Even if in our analysis the thermoeconomic optimization gives the solution with the minimum HRSG cost, it could be observed that the convenience could be strongly dependent on the cost structure adopted and to have considered the cost of the exergy losses equal to the selling cost of energy.

Table 10
Surfaces and gas-side effectiveness of the sections of the thermoeconomically optimized 2PRH and 3PRH HRSGs under the conditions of ABB and Siemens plants

		LP				IP-RH						HP			
2PRH	eco	eva	sh p1	–	–	–	–	–	rh p2	rh p3	eco p1	eco	eco p2	eva	sh p3
S 10^3 [m^2]	115	41.4	0	–	–	–	–	–	0	10.1	3.44	212	0	18.1	27.2
η	0.727	0.987	0.185	–	–	–	–	–	0	0.886	0.185	0.921	0	0.833	0.886
3PRH	eco	eva	sh p1	sh p2	eco p1	eva	sh p2	sh p3	rh p4	rh p5	eco p2	eco p3	eco p4	eva	sh p5
S 10^3 [m^2]	54.6	41.5	2.88	0	38.0	20.3	0	1.04	3.68	23.4	0.43	18.0	111	20.6	56.6
η	0.575	0.923	0.527	0	0.527	0.707	0	0.408	0.913	0.767	0	0.408	0.913	0.699	0.767

Table 11
Values of the parameters to evaluate convenience expression (35)

Plant	$K_{optimum}$ M$	K_{comm} M$	ΔK M$	W_{tot} MW	H hour·year^{-1}	k_{energy} $·(MWh)$^{-1}$	η_{CC1}	η_{CC0}	τ	LTG M$
ABB	20.4	7.9	12.5	280	8000	68	0.594	0.576	8%	9.7
Siemens	20.3	10.2	10.1	380	8000	68	0.595	0.571	8%	30.4

To roughly evaluate, the economic convenience of the optimized solutions with respect to the ABB and Siemens' with the optimized HRSG with respect to the selling price of energy, it can be defined a function called long term gain (LTG) of the plant. Assuming an economic life of the plant equal to 20 years it is:

$$\mathrm{LTG} = -\Delta K + \left[(1+\tau)^{20}\right] \cdot \frac{W_{\mathrm{tot}}}{\eta_{\mathrm{CC0}}} \cdot H \cdot k_{\mathrm{energy}} \times (\eta_{\mathrm{CC1}} - \eta_{\mathrm{CC0}}) \tag{35}$$

which results positive if the proposed investment is favourable, negative in the opposite case.

With the values of the above mentioned parameters summarized in Table 11, according to the expression (35), we could cover the initially greater required investment in both cases, and gain in 20 years 9.7 M$ with respect to the ABB plant and 30.4 M$ with respect to the Siemens one. So it seems really convenient, also on a pure economic criterion, to strongly reduce pinch points ad to use high pressure level close to the critical one, even if it determines increases in the HRSG cost of more than 100%.

Most manufacturers and users observe that the economic convenience of a plant must be reported to the fuel cost. This function considers the future cash flows at their present value, referring to the cost of fuel and to its better qualification; in this case the function that define the long term gain of the plant is:

$$\mathrm{LTG}' = -\Delta K + \left[(1+\tau)^{20}\right] \cdot W_{\mathrm{tot}} \cdot H \cdot k_{\mathrm{fuel}} \times \left(\frac{1}{\eta_{\mathrm{CC0}}} - \frac{1}{\eta_{\mathrm{CC1}}}\right) \tag{36}$$

that obviously determines lower values of the long term gain, because k_{fuel} is about three times lower than the selling price of energy.

6. Conclusions

A method to optimize the operative parameters of a HRSG has been proposed, in order to attempt to improve the overall efficiency of combined cycle plants. Actually two different objective functions have been considered: one given by the exergetic losses due to the heat transfer between fluids; the other represented by a cost, sum of the cost of the HRSG and the cost of the exergy losses.

The minimization of the former objective function allows to find the best configuration of the HRSG, though if it is not actually realizable, because of the infinite exchange surfaces obtained. The minimization of the latter objective function allows instead to find the optimum available compromise between thermodynamic efficiency and economic profit.

The operative parameters of a HRSG (such as temperatures, pressures, mass flow rates, exchanger efficiencies) are those information needed to completely define its behaviour, and to actually design it, in full details. Naturally only some of these parameters are independent from the others, and can be modified in order to minimize the objective functions. The dependent parameters can be found from them by solving the mathematical model of the system.

Applying the proposed method to a certain number of HRSG and steam cycle configuration, it has been stressed that:

- This method can be used instead of that based on the aprioristic choice of the pinch-points for the design of a HRSG, as their values come from an exact optimisation process.
- Thermodynamic analysis provides zero pinch-points (i.e., infinite evaporators' surfaces), as expected.

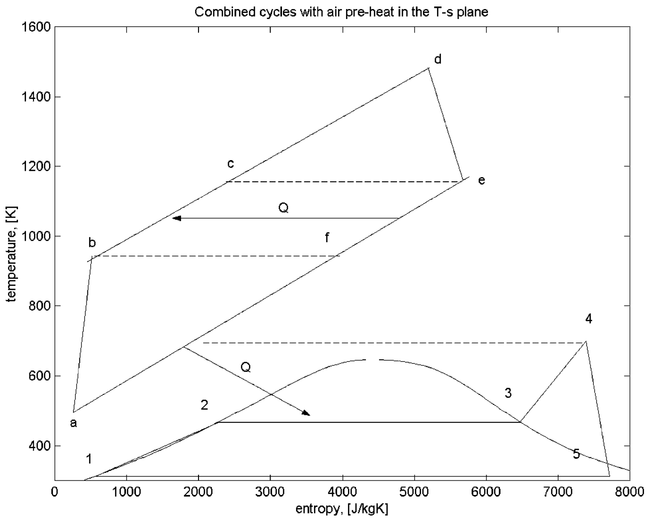

Fig. 13. Exhaust gas at very high temperature can be used, before going to the HRSG, to warm fresh air after the compression, thus improving the efficiency of the gas cycle.

- Beyond certain values of the gas entry temperature (different among the cycles) the higher pressure of steam tends to the critical condition.
- The pinch-points rise when we introduce economic aspects, proving that their choice is the consequence of a compromise between thermodynamic efficiency and investment costs.
- The values of the pinch-points depend on the cost hypothesis done; thanks to the approach adopted it is however possible to consider different cost functions, so that the method is completely general.
- Economic aspects (i.e., cost of exchanger surfaces dependent on running pressure) can shift the value of the gas entry temperature at which the steam cycle becomes critical, but they cannot delete this behaviour, as it is due to thermodynamic aspects.
- With this method it seems possible to reach overall combined cycle efficiencies near to 60% on existing plants, just by optimizing the heat recover and the steam cycle, and without modifying the gas turbine characteristics.

The thermoeconomic optimization method carried out in the paper provides also useful feedback for the optimization of the gas turbine group and consequently of the whole combined cycle. For instance, if the available gases at the exit from the gas turbine are at very high temperatures, it could be advantageous to use them to warm fresh air before the combustion, thus reducing the quantity of fuel needed and improving the gas cycle efficiency. Combining a regenerative gas cycle with the HRSG optimization, it is possible to aim to obtain combined cycle plant efficiencies higher than 60% with the existing gas turbine technology (Fig. 13).

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