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► **To cite this version:**

Victor - Eduard Cenușă, Adrian Badea, Michel Feidt, Riad Benelmir. Exergetic Optimization of the Heat Recovery Steam Generators by Imposing the Total Heat Transfer Area. *International Journal of Thermodynamics*, 2004, 7 (3), pp.149 - 156. hal-02996629

HAL Id: hal-02996629

<https://hal.univ-lorraine.fr/hal-02996629v1>

Submitted on 9 Nov 2020

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Exergetic Optimization of the Heat Recovery Steam Generators by Imposing the Total Heat Transfer Area

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Abstract

The paper presents an original and fast method for the heat recovery steam generator (HRSG) exergetic optimization. The objective is maximizing the exergy transfer to the water / steam circuit. The proposed approach, different from the classical method that fixes the pinch point, is essentially thermodynamic but it considers also the economics by imposing the total heat transfer area of HRSG. The HRSG may have one or two steam pressures, without reheat. The input data from the gas turbine are: the mass flow rate, the temperature and the molar composition of flue gases. The results are the optimum pressures of the superheated steam. The numerical computations were realized in Delphi programming utility. The obtained results are in agreement with the recent literature.

Keywords: Heat recovery steam generator - HRSG, numerical model, exergetic optimization, imposed total heat transfer area

1. Introduction

HRSG is an interface between the gas cycle and the steam cycle in combined cycles. The classical approach to HRSG optimization is the "pinch-point" method, i.e. imposing the minimum temperature difference between the two agents (Burer et al., 2003). This hypothesis does not allow technical comparisons, in conditions of comparable costs for the HRSG, which means that thermodynamic optimum obtained could be far from the economical one. On the other hand, a good economical analysis cannot be done, without knowing many accurate data about the HRSG design. Engineering firms use commercial software for dimensioning HRSG's, e.g. Gate Cycle. Manufacturers use even more sophisticated programs. For a fast and inexpensive system study, without knowing too many data, it is possible to use appropriate simplified computation methods. Franco and Russo (2002) and Casarosa et al. (2003) use an alternative method to the "pinch-point" method, doing a thermo-economic optimization of the HRSG operating parameters. Valdés et al. (2003) carried out a thermo-economic optimization of

combined cycle gas turbine power plants using genetic algorithm. In order to eliminate the limits of the pinch-point method for optimizing the secondary pressure of a wet steam generator in a nuclear power plant, Alexe and Cenușă (2002) defined an 'equivalent logarithmic mean temperature difference'. It bases on the hypothesis that the global coefficients of heat transfer for economizer and for vaporizer (in the boiling region) are approximately same ($U_{ECO} \cong U_{EV}$).

The proposed solution in this paper is based mainly on the thermodynamic approach, but it considers also the economics by imposing the total heat transfer area of HRSG. This approach allows determination of an optimized technical solution that is closer to the economical optimum. We do not enter in details but we take into consideration the majority of the opposite tendencies that determine the existence of a technical restricted optimum. From the iterative computation routine, which takes into account the restrictions and maximizes the objective function, are obtained various equivalent HRSG's, with the same heat transfer area, not having the same pinch-point temperature difference.

2. The System Layout and the Corresponding $t [^{\circ}\text{C}] - h [\text{kJ/kg}]$ Diagram

The system layout for the HRSG 1p is presented in *Figure 1* (Darie et al. 2001) and that for the HRSG 2p in *Figure 2*. The two $t [^{\circ}\text{C}] - h [\text{kJ/kg}]$ diagrams (where ‘h’ represents the gas enthalpy) corresponding to the HRSG 1p and to the HRSG 2p are presented in *Figures 3 and 4* respectively together with the characteristic points of the installations

For the HRSG 2p it is considered that the high-pressure is maintained by a feed pump at the intake of the boiler, and the cold area of the high-pressure economizer (ECO_{HP_1}) is in parallel with the low-pressure economizer (ECO_{LP}). The other heat exchangers are in series (*Figures 2 and 4*). In the system layout of HRSG 2p (*Figure 2*), an intermediary pump, between

low-pressure drum and high-pressure drum is not necessary (Monteil 2003).

The HRSG model analyzed in this paper: a) has not post-combustion; b) it has multiple through in the vaporizer, assisted with a circulation pump – see *Figures 1 and 3*, or natural through due to the density difference between the water and the water/vapor emulsion (Pănoiu 1982). The once-through circulation is not considered within this model.

3. The Numerical Model and Routines

The computation routines that optimize the HRSG from the exergetic point of view were prepared by the Delphi software. That insures an interactive structure and allows the graphical interpretation of the results.

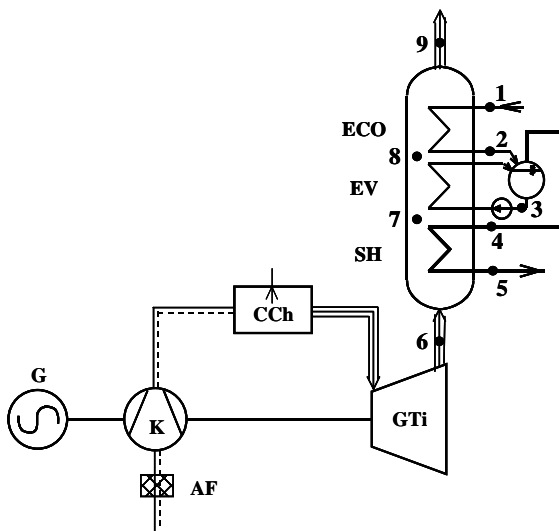


Figure 1. GT + HRSG 1p: the system layout

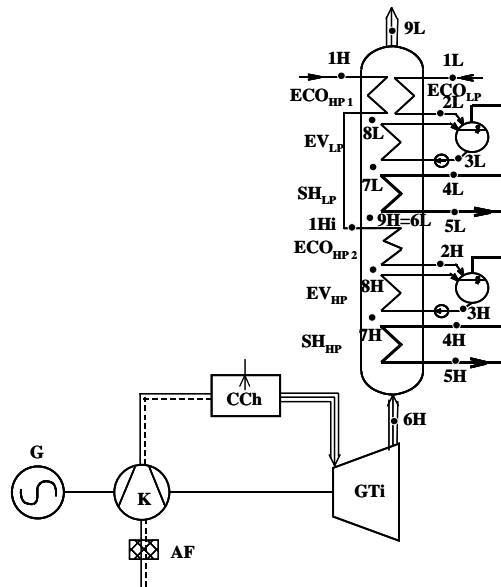


Figure 2. GT + HRSG 2p: the system layout

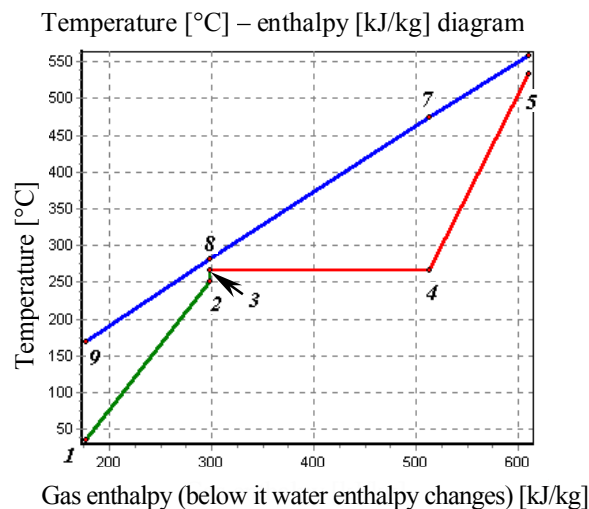


Figure 3. The $t [^{\circ}\text{C}] - h [\text{kJ/kg}]$ diagram for the HRSG 1p; the characteristic points

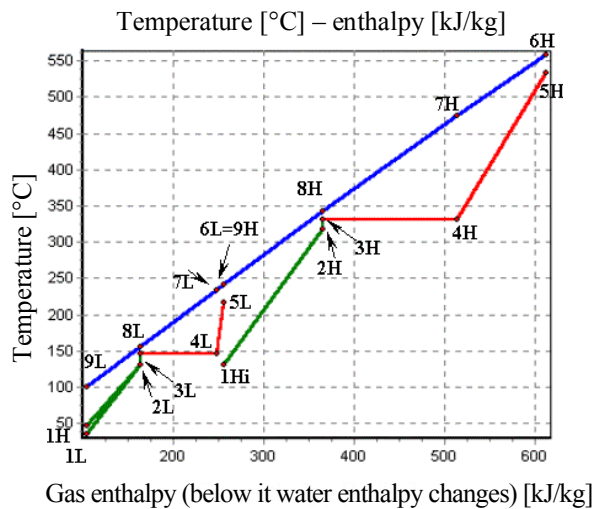


Figure 4. The $t [^{\circ}\text{C}] - h [\text{kJ/kg}]$ diagram for the HRSG 2p; the characteristic points

3.1 Hypotheses

The main input data for the program determine the thermal and exergetic potential of the flue gases at the inlet of HRSG. These data are: molar composition, the gas turbine exhaust mass flow rate and the gas turbine exit temperature: t_6 , t_{6H} . They can be obtained from measurements on the GT or from catalog data and computation (Cenușă et al. 2004). The gas enthalpy and entropy depend on the temperature, the pressure and the molar composition of the gases: N_2 , O_2 , H_2O , CO_2 (Dupleac 1997). Due to the large excess air, the combustion is supposed being complete (without CO). The thermodynamic properties of water / steam are determined with routines found in literature (Alexe et al. 2001).

Pressure drops are given in percentage of the absolute pressure: In the super heater it is 8 % of the main steam pressure, and in the economizer 10 % of the drum pressure.

Because of the small water drops in the wet steam, it is considered that at the drum outlet the steam quality is 99 %. In order to avoid water boiling in the economizer the economizer water outlet temperature is lower than the drum temperature (t_{sat}). This temperature difference, currently named in literature "approach - point" (Subrahmanyam et al. 1995) is a parameter given at keyboard.

The thermal resistance of the conductive heat transfer in the wall of the tubes and that of the internal convective heat transfer can be neglected in comparison to the thermal resistance, of the external heat transfer.

Bontemps et al. (1998) consider that the variation of the external heat transfer coefficient with the temperature is practically negligible (in particular for ECO and EV). Generally the gas flow is turbulent, the pressure drop is small, this heat transfer coefficient is proportional to $T^{0.12}$ (Kelvin). We will assume that for the case of a HRSG with one or more (2 or 3) pressure levels, the global thermal resistance at the ECO [$(m^2 \cdot K)/kW$] is practically equal to the overall thermal resistance of the EV (Dumont and Heyen 2001). It results the equality between the heat transfer coefficients $kW/[m^2K]$ for ECO and EV, no matter the pressure levels.

Regarding the internal convective thermal resistance on the superheating side, this is not negligible, which means that the global thermal resistance of SH is higher than the global thermal resistances of ECO and EV. The overall heat transfer coefficient of SH is smaller than the global heat exchange coefficient of ECO and EV. The mathematical model considers this by using the ratio $f_{SH} = U_{ECO}/U_{SH} > 1$.

In the HRSG 2p case the heat transfer coefficients for the low-pressure side l and the high-pressure side are assumed to be equal. Also the pinch-point temperature differences (Paren and Parietti 1991) and approach-points for the low and high-pressure stages are considered being the same. The HRSG superheated steam temperature (t_5 or t_{5H}), results imposing the temperature difference: $t_6 - t_5$ or $t_{6H} - t_{5H}$. In the HRSG 2p case, the low-pressure steam temperature (t_{5L}) can be given directly (which is not the usual case) or can be computed by the program, depending on the temperature difference: $t_{6L} - t_{5L}$. We recommend the same temperature difference: $t_{6L} - t_{5L} = t_{6H} - t_{5H}$.

The main model hypothesis (constant total heat transfer area of the HRSG) results indirectly from the previous hypotheses, by imposing the $(U_{ECO}) \cdot (S_{HRSG})$ factor (which should be chosen in order to obtain usual values for the pinch point), and the ratio U_{ECO}/U_{SH} . That allows taking into consideration the economics, because the initial cost of the HRSG is approximately proportional to the heat exchange surface. Whenever the total heat transfer area is given, that kind of a model can offer an optimum partition to different surfaces.

3.2 The mathematical model

The target of optimization is finding the maximum of the total exergy transfer to water/steam. For Combined Cycles with condensing steam turbine, we recommend using the condensing steam temperature as the reference temperature for the exergy transfer. We optimize the HRSG 1p pressure and respectively the HRSG 2p pressures of steam production.

To determine the temperature distribution in the HRSG, pinch-point is needed, however it is unknown in this present model being unknown. In the papers that use the „pinch-point” method, this temperature difference is an imposed parameter. In our approach, the „pinch-point” is computed. The manner to do this is iterative. By using the middle interval numerical method (Feidt 1996), we choose the „pinch-point” for which the difference between $S_{HRSG} \cdot U_{ECO}$ imposed and the calculated value is extremely small ($< 10^{-5}$).

Having the temperature distribution in the HRSG it is possible to compute the mean logarithmic temperatures differences for each heat exchange area. The thermal energy flow transferred through each area will be determined using the general relation (1). This is an approximation assuming near counter flow conditions:

$$\dot{Q}_j = U_j S_j \Delta t_{m \ln, j} \quad (1)$$

where: „j” refers to the heat exchange area, \dot{Q} represents the thermal energy flow transmitted from the gas to the water/steam, U – global heat transfer coefficient, S – the heat exchange area, $\Delta t_{mln,j}$ – the mean logarithmic temperatures difference that corresponds to every surface „j”.

The general computation relation for each heat exchange area, as well as the computation relation of the total heat exchange surface of HRSG are obtained by using relation (1):

$$S_{HRSG} = \sum_j S_j = \sum_j \frac{\dot{Q}_j}{U_j \cdot \Delta t_{mln,j}} \quad (2)$$

Taking into account the considered computation hypotheses, the equation (2) is multiplied by U_{ECO} and becomes:

$$\begin{aligned} S_{HRSG} \cdot U_{ECO} &= U_{ECO} \cdot \sum_j \frac{\dot{Q}_j}{U_j \cdot \Delta t_{mln,j}} \\ &= \sum_{j1} \frac{\dot{Q}_{j1}}{\Delta t_{mln,j1}} + f_{SH} \cdot \sum_{j2} \frac{\dot{Q}_{j2}}{\Delta t_{mln,j2}} \end{aligned} \quad (3)$$

where „j1” represents the heat exchange areas of economizers and vaporizers and „j2” represents the heat exchange areas of super heaters, f_{SH} is the ratio of the overall heat transfer coefficients U_{ECO} / U_{SH} .

3.3 The computation logic

The program for the HRSG 1p optimizes the steam pressure at the HRSG outlet (p_5 optimum) insuring the maximum total exergy transfer to the water/steam (Figure 5). The optimum is given by the fact that the exergetic efficiency increases as

p_5 grows, while the thermal energy flow decreases.

It is generally accepted that introducing supplementary levels of steam production pressures, the mean temperature difference between the two fluids diminishes and the exergetic efficiency is increases. For the same reasons, p_{SH} and p_{SL} being the independent parameters, it is possible to obtain a maximum exergy transfer to the water/steam (see Figure 6).

The search method for the maximum exergy transfer remains the same for both configurations: HRSG 1p and HRSG 2p. In the first case the search for the optimized pressure is done in one step and in the second case in two steps.

In both cases, an initial maximal value $p [0]$ is chosen (180 bar for multiple through in the boiling area). For this $p [0]$ we determine the exergy transfer $Ex [0]$ that is the first reference value. Because the optimal main steam pressure is lower that $p [0]$ (see Figures 5, 6 and 7), we continue the computation taking lower main steam pressures, with an initial decreasing pressure step Δp (generally around 1 bar):

$$p [m] = p [m-1] - \Delta p \quad (4)$$

The first optimum is found when:

$$Ex(p [m]) - Ex(p [m-1]) < 0 \quad (5)$$

At this moment, we change the searching direction with a smaller step: $\Delta p_{new} = - \Delta p_{old} / 10$. The iteration continues, until the condition (6) is satisfied.

$$| p [m] - p [m-1] | < \delta \quad (6)$$

here $\delta = 5 \cdot 10^{-5}$ bar.

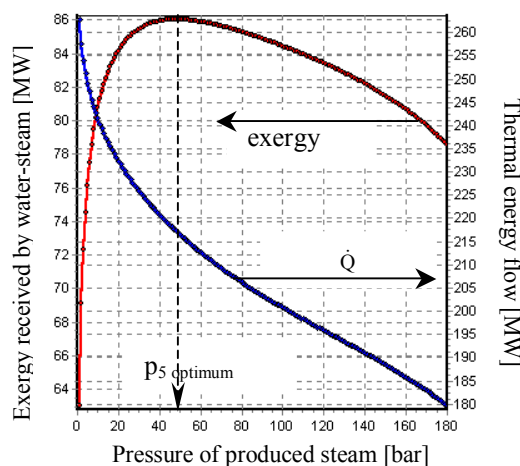


Figure 5a. The case when the GT upstream of HRSG is „heavy duty” type: PG9231(EC)

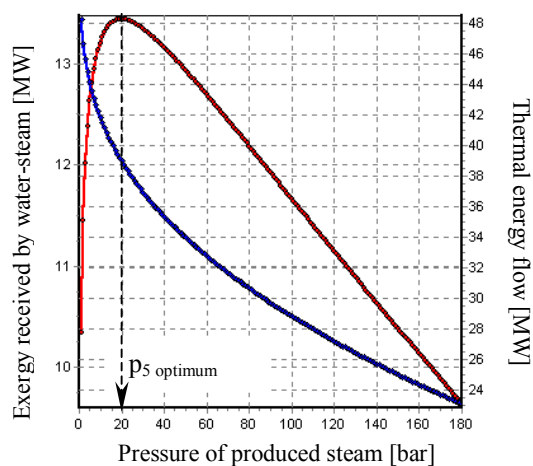


Figure 5b. The case when the GT upstream of HRSG is „aeroderivative” type: LM6000 Sprint

Figure 5. HRSG 1p optimization. The dependence of total exergy transfer (the left vertical axis) and total thermal energy transfer (the right vertical axis) on the steam pressure.

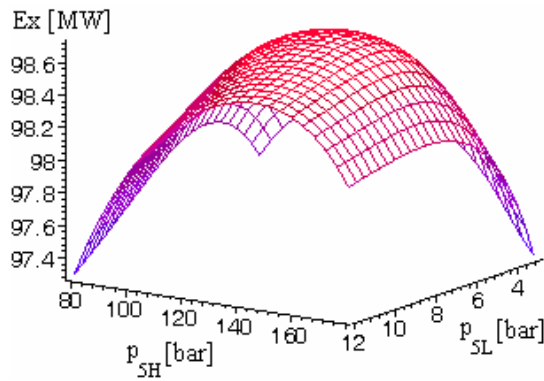


Figure 6a. The case when the GT upstream of the HRSG is „heavy duty” type: PG9231(EC)

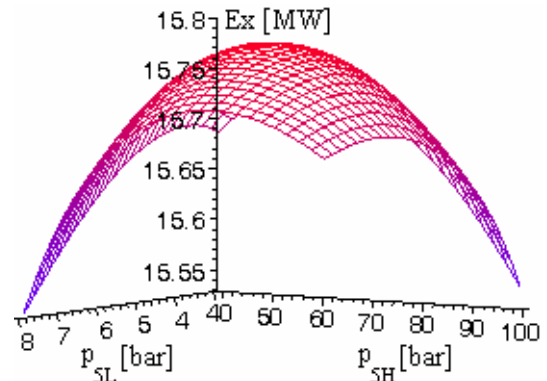


Figure 6b. The case when the GT upstream of the HRSG is „aeroderivative” type: LM6000 Sprint

Figure 6. HRSG 2p optimization: The dependence of the total exergy transfer to water/steam on the steam pressures: p_{5H} and p_{5L} .

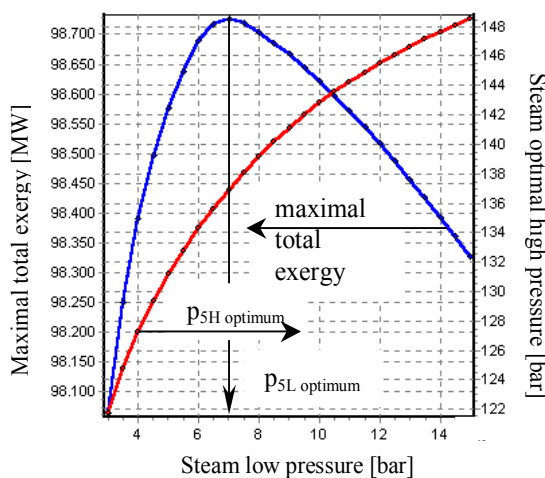


Figure 7a The case when the GT upstream of the HRSG is „heavy duty” type: PG9231(EC)

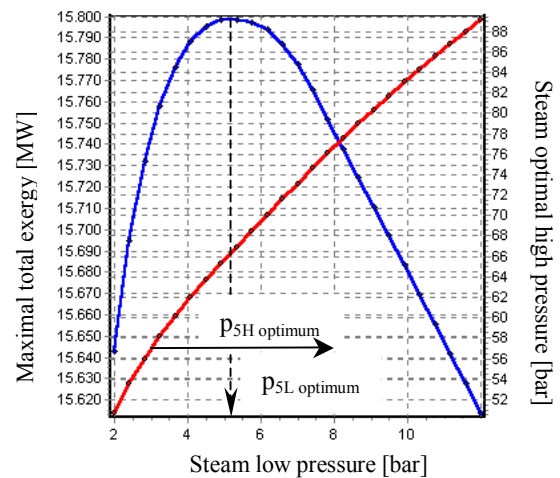


Figure 7b The case when the GT upstream of the HRSG is „aeroderivative” type: LM6000 Sprint

Figure 7. HRSG 2p optimization: The dependence of total exergy transfer to water/steam (left) with p_{5L} , when p_{5H} is the optimal one; $p_{5H_optimum}$ on the right scale.

The model for HRSG 2p optimizing allows determining:

- One of the steam pressure, $p_{5H_optimum}$ or $p_{5L_optimum}$ imposing the other one.
- The optimum pair of parameters, $p_{5H_optimum}$ and $p_{5L_optimum}$, (Figures 6 and 7). In this case we search the maximum of the maximal values ($EX_{HRSG_max-max}$) for the total exergy (EX_{HRSG}) received by the water/steam. The monotone ascending curve that appears at the right side of Figure 7 shows the variation of the optimum high pressure ($p_{5H_optimum}$) depending on the low pressure (p_{5L}).

The method allows computing the thermal energy flow and the exergy received by water/steam through each area, respectively the participation of the heat exchange areas in the total HRSG area and the investment participations in the total investment for each

heat exchanger of the HRSG (Cenușă 2003).

3.4 Case studies

The main data that practically impose the optimal pressure(s) is the flue gas temperature at the GT exit and the HRSG intake. This is different for “heavy-duty” and “aeroderivative” GT (Cenușă et al. 2003a). We chose two representative GT:

- a heavy-duty one: PG9231(EC), with 508 kg/s gas flow and $t_6 = t_{6H} = 558$ °C
- an aeroderivative one: LM6000 Sprint, with 127 kg/s gas flow and $t_6 = t_{6H} = 450$ °C

The Figure 5 shows directly the optimal pressures ($p_{5_optimum}$) for the HRSG 1p, and indirectly the ratio between the maximal exergy flow and the corresponding thermal one:

- For the heavy-duty GT: $p_5 = 48$ bar and $Ex/\dot{Q} = 39.6$ %;

- For the aeroderivative one: $p_5 = 20$ bar and $Ex/\dot{Q} = 35.5\%$

The optimal pressures and the ratios between the maximal exergy flow and the corresponding thermal one are bigger for the heavy duty GT than for the aeroderivative GT case. This difference comes from the higher inlet temperature in the HRSG for the aeroderivative GT than for the heavy duty GT. Generalizing, the conclusion is that increasing the inlet temperature in the HRSG, the optimal pressure and the exergetic efficiency are increasing too.

The *Figure 7* shows the pair of pressures ($p_{5H_optimum} / p_{5L_optimum}$) that offers the maximal exergy flow for the HRSG 2p and the ratios between this flow and the thermal one:

- For the heavy-duty GT: $(p_{5H} / p_{5L}) = (137/7)$ and $Ex/\dot{Q} = 39.7\%$;
- For the aeroderivative one: $(p_{5H} / p_{5L}) = (66/5)$ and $Ex/\dot{Q} = 35\%$.

And here the optimal pressures are bigger for the heavy duty GT than for the aeroderivative GT case. The ratios between the exergetic and the thermal flows are almost the same like for HRSG 1p case, but the both flows are bigger.

Comparing the optimal pressures for the both cases, we remark that:

- the $p_{5H_optimum}$ for HRSG 2p is bigger than the double of $p_{5_optimum}$ for HRSG 1p;
- the $p_{5L_optimum}$ for HRSG 2p is lower than a quarter from $p_{5_optimum}$ for HRSG 1p.

About the maximal exergy flows transmitted to the water/steam, at the optimal pressures in the HRSG 1p and HRSG 2p case, for the same GT, we remark that:

- For heavy duty GT, the exergy flow increases from 86 MW, for the HRSG 1p, to 98.7 MW, for the HRSG 2p. The

percentage increasing is 14.8 %, while S_{HRSG} increases with 57 %;

- In the case with aeroderivative GT, the exergy flow increases from 13.45 [MW], for the HRSG 1p, to 15.8 MW, for the HRSG 2p. The percentage increasing is 17.7 %, while S_{HRSG} increases by 50 %.

We mention that the increasing of the S_{HRSG} was necessary in order to have almost the same values of the pinch points for all the HRSG.

4. Model Validation

The validation was done by means of the data of the heavy duty GT10C – with the nominal electric power 29 100 kW, the heat rate 10 000 kJ/kWh, the flue gases flow 91.1 kg/s and the gas outlet temperature 518°C. For this GT we had, from the manufacturer – (ALSTOM Power 2001), values of the produced steam flow for different steam pressures and temperature at the HRSG 1p outlet (*Figure 8a*).

Using the model described in Cenușă et al. (2004) and the manufacturer data, we obtained the molar composition of the flue gases: 75.72 % N_2+Ar , 13.83 % O_2 , 3.17 % CO_2 , 7.28 % H_2O . These data were used in the HRSG 1p model, together with the following supplementary ones: $S_{HRSG} * U_{ECO} = 570$ kW/°C (S_{HRSG} [m²], U_{ECO} [kW/m²/°C]), $FSH = U_{ECO}/U_{SH} = 1.3$ [-]. According to Dumont and Heyen (2001), the water temperature at HRSG inlet (t_1) = 35°C and the „approach point” = 8°C. According to Subrahmanyam et al. (1995). For $S_{HRSG} * k_{ECO} = 570$ kW/°C, the computed values of the pinch point are between 8.7 and 13.5°C, being in the interval from 8 to 20°C, resulting from the manufacturer experience (Franco and Russo 2002).

Figure 8b presents the computed data with the HRSG 1p model. We note the good coincidence of the values in *Figures 8a and 8b*.

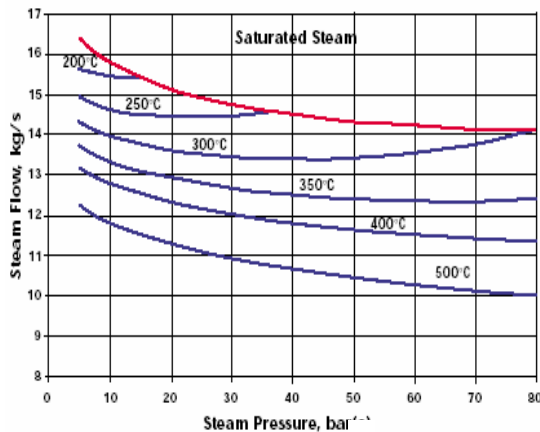


Figure 8a Manufacturer diagram

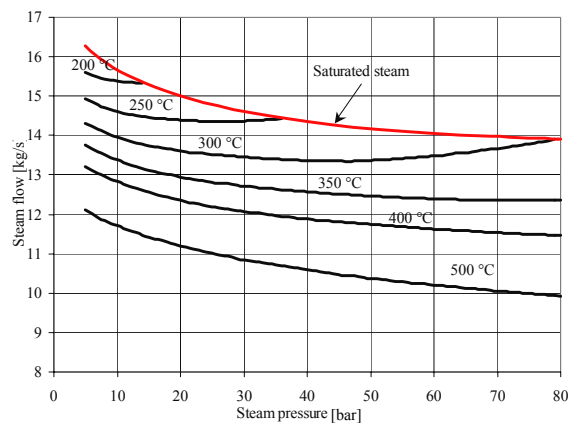


Figure 8b Diagram resulted from the HRSG 1p program

Figure 8. HRSG 1p: The variation of the steam flow with the steam pressure and temperature.

5. Conclusions

The paper presents:

- an original method for the exergetic optimization of the HRSG (used in combined steam and gas cycle), with one or two pressure(s) levels of steam, without reheat, by maximizing the exergy transfer to the water / steam circuit,
- the mathematical model associated
- the obtained results of the analysis for 2 cases.

The main hypothesis, that the total heat transfer area of the HRSG is constant, was introduced indirectly, making the assumption that the factor $U_{ECO} * S_{HRSG}$ is constant.

The case analyses were done for two gas turbine types, upstream the HRSG: PG9231(EC) is a heavy-duty type and LM6000 Sprint is an aeroderivative one. The exergetic optimum is highlighted for HRSG 1p and HRSG 2p by graphical way. In the case of HRSG 2p the exergy transfer is maximized for the high pressure and also simultaneously for both pressures of steam: low and high.

The obtained results agree with the results given in the recent literature. Also, the mathematical model was validated by comparing (graphically) the results of the HRSG 1p model with the results provided by a manufacturer (ALSTOM).

The mathematical model presented in this paper, could be a starting point for the optimization of the combined steam and gas cycles without steam reheat. It can be developed for other HRSG configuration, including the three pressure levels of steam production case.

Nomenclature

AF	air filter
CCh	combustion chamber
ECO	economizer
EV	vaporizer
Ex	exergy (function)
f_{SH}	coefficient ($= U_{ECO}/U_{SH}$)
G	electrical generator
GTi	gas turbine itself
GT	gas turbine system
HRSG	heat recovery steam generator
h	enthalpy [kJ/kg]
K	compressor
m	variable [-]
p m]	pressure array [bar]
\dot{Q}	thermal energy transfer from the gas to the water/steam [kW]
S	heat exchange area [m ²]

SH	super heater
t	temperature [°C]
T	temperature [K]
U	global (overall) heat transfer coefficient [kWm ² /°C]
δ	absolute computed error [bar]
Δp	pressure step [bar]
Δt_{mln}	mean logarithmic temperatures difference
1	one pressure level for the steam production
2p	two pressure levels for the steam production
1..6, 1L..6L, 1H..6H, 1Hi	characteristic points

Subscripts

ECO	economizer
EV	vaporizer
GT	gas turbine
HP	high pressure
HRSG	heat recovery steam generator
J	heat exchange areas
j1	heat exchange areas of economizers and vaporizers
j2	heat exchange areas of super heaters
LP	low pressure
sat	saturation
SH	super heater
max-max	the maximum of the maximal values

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