THERMOECONOMIC OPTIMIZATION OF TRIPLE PRESSURE HEAT RECOVERY STEAM GENERATOR OPERATING PARAMETERS FOR COMBINED CYCLE PLANTS

by

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The aim of this work is to develop a method for optimization of operating parameters of a triple pressure heat recovery steam generator. Two types of optimization: (a) thermodynamic and (b) thermoeconomic were performed. The purpose of the thermodynamic optimization is to maximize the efficiency of the plant. The selected objective for this purpose is minimization of the exergy destruction in the heat recovery steam generator. The purpose of the thermoeconomic optimization is to decrease the production cost of electricity. Here, the total annual cost of heat recovery steam generator, defined as a sum of annual values of the capital costs and the cost of the exergy destruction, is selected as the objective function. The optimal values of the most influencing variables are obtained by minimizing the objective function while satisfying a group of constraints. The optimization algorithm is developed and tested on a case of combined cycle gas turbine plant with complex configuration. Six operating parameters were subject of optimization: pressures and pinch point temperatures of every three (high, intermediate, and low pressure) steam stream. The influence of these variables on the objective function and production cost are investigated in detail. The differences between results of thermodynamic and the thermoeconomic optimization are discussed.

Key words: *exergy, heat recovery steam generator, thermoeconomic optimization*

Introduction

The heat recovery steam generator (HRSG) is used to recover wasted heat from exhausted gases leaving gas turbine in the combined cycle gas turbine plants; therefore making any change in its design directly affects the cycle efficiency, its power generation, the global cost, and other variables in the cycle. The design of the HRSG is affected by the steam cycle parameters such as: pinch point (*PP*), which is defined as temperature difference between the saturation temperature of water and the gas temperature of the gas leaving the evaporator, and steam drum pressures. Therefore the optimization is of the greatest relevance. A possible way to improve the efficiency of these systems is to minimize the exergetic losses in the HRSG by means of minimization of *PP*. On the other hand, *PP* reduction may be obtained by larger heat transfer surface areas, which increases the capital cost of HRSG. An optimum value for the pinch point may be found by making a trade-off between these effects.

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Different approaches can be found in the literature regarding the optimization of HRSG of combined cycle power plants. Alus and Petrović [1] performed an optimization of a triple pressure combined cycle gas turbine (CCGT). The objective of the optimization was to minimize the production cost of electricity in the CCGT power plant based on energetic and economic analysis. Mansouri *et al.* [2] and Ravi Kumar *et al*. [3] studied the effect of HRSG configurations on the performance of CCGT. Ahmadi and Dincer [4] introduced an objective function in terms of dollars per second, including the sum of the operating, maintenance, and capital investment costs. The optimum key variables were obtained by minimizing the objective function using a generic algorithm. They concluded that by increasing the fuel price, the optimized decision variables in the thermoeconomic design tend to those of the thermodynamic optimum design. Behbahani-nia *et al.* [5] presented an exergy based thermoeconomic method, which is applied to find optimal values of design parameters (the *PP* and the gas-side velocity) for a single pressure HRSG used in CCGT. Bracco and Siri [6] analyzed different objective functions for exergetic optimization of single level combined gas-steam power plants. Ghazi *et al.* [7] carried out an optimization study to find the best design parameters (high and low drum pressures, steam mass flow rates, pinch point temperature differences and the duct burner fuel consumption flow rate) of a dual pressure combined cycle power. Total cost per unit of produced steam exergy is defined as the objective function. Naemi *et al.* [8] presented a design method for dual pressure heat recovery steam generator using non-dimensional parameters. The thermodynamic and thermo- economic analyses are investigated to achieve the optimum steam cycle parameters of HRSG. Casarosa *et al.* [9] have performed a thermoeconomic optimization of the operating steam parameters of the HRSG, for combined cycle plants. This method is an alternative to the usual *PP* approach. It represents an attempt to find a compromise between economic and thermodynamic analysis, based on incorporating exergy-based production costs with economic evaluations. The analysis is based on the gas-side effectiveness of the sections of the HRSG instead of the usual *PP* method.

In this work, a gas turbine was selected at first. The exhaust gas parameters at the inlet of the HRSG (mass flow rate, temperature, and chemical composition) were defined and they were not subject of further consideration. Thermodynamic optimization for triple pressure HRSG applied here was based on the minimization of exergy destruction, while the thermoeconomic optimization is based on the minimization of the total annual cost of HRSG (sum of exergy destruction cost and annual investment cost). Furthermore, effect of pinch points (PP_{LP}, PP_{IP}, and PP_{HP}) and drum pressures (*LP*, *IP*, and *HP*) on the production cost of electricity, cycle efficiency and exergy destruction of HRSG are studied. Exergy destruction of each part of the bottoming cycle is computed. The optimal operating parameters of this HRSG are proposed. A comparison between an initial case and an optimization case was made in order to verify both the model and the methodology. One more comparison was made between two optimization cases: one with *PP* assumed to be the same for all evaporators, and another having different *PP* for every pressure level.

Thermodynamic analysis

Description of the used combined cycles

A triple pressure CCGT was selected for this research. As it is shown in the schematic diagram, fig. 1, the subsystem of a power plant include the gas turbine, HRSG, feed water tank, cooling system, and the high, intermediate and low pressure steam turbine. The gas turbine model used in this study was Siemens SGT5-PAC 4000 F. The assumptions and parameters selected for the thermodynamic analysis of the plant are tabulated in tab. 1.

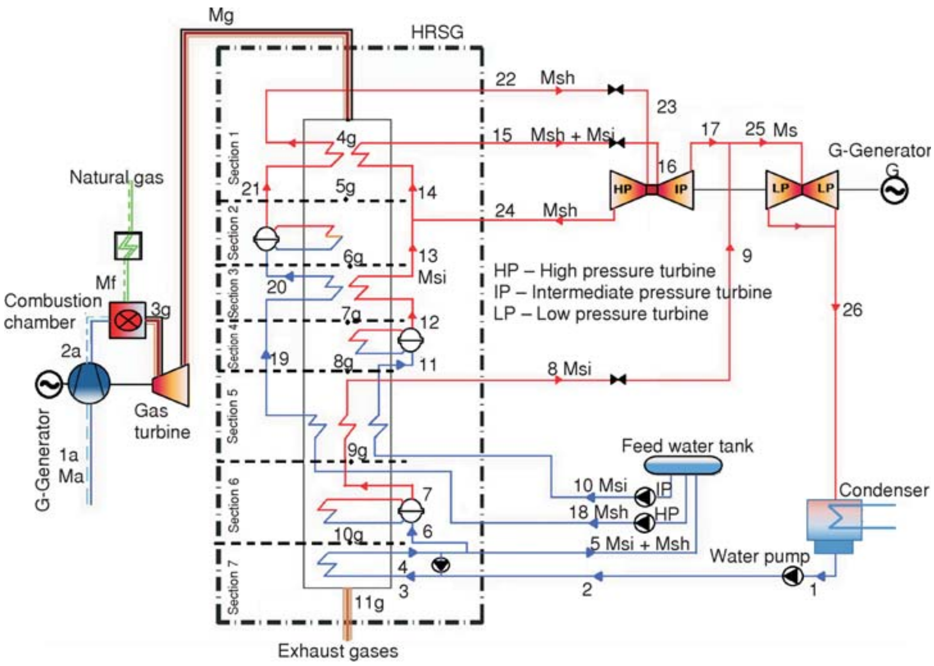

Figure 1. Schematic diagram of the triple pressure combined cycle power plant
(for color image see journal web-site)

Table 1. Gas turbine parameters and assumptions for component performances of the CCGT with the triple-pressure HRSG selected for the optimization

Parameter	Value
Ambient air pressure, [bar]	1.013
Ambient air temperature, [°C]	20
Electrical power at the generator output, [MW]	288
Exhaust gas mass flow, [kgs^{-1}]	688
Exhaust gas temperature at the gas turbine outlet, [°C]	512
The gas turbine efficiency, [%]	39.5
Lower heat value of the fuel, [kJkg^{-1}]	47141
Minimum stack temperature, [°C]	93
Assumption	
The isentropic efficiency of all three steam turbine parts	90%
The isentropic efficiencies of water pumps	82%
The mechanical efficiency	99.5%
The generator efficiency	98%
The heat recovery steam generator efficiency, [1]	99.3%
The pressure drops for water in the economizers, [1]	25%
The pressure drops for steam in the reheat and superheater tubes, [1]	8%
Minimum dryness fraction of steam at low steam turbine outlet, [1]	0.88
Low-pressure steam turbine outlet (condenser pressure) [bar]	0.055
The inlet cooling water temperature in condenser, [°C]	20
Feed water temperature at 3, [°C]	60
Price of natural gas (cf), [$kWh^{-1}]	0.0467
Selling price of electricity (S), [$kWh^{-1}]	0.114

Energy analysis

To model the triple pressure HRSG, the mass balance equation and the first law of thermodynamics are applied for each element of the HRSG. Distinctly the *PP* is not the same for each pressure level in our case. The temperature of the gas entering the *LP*, *IP*, and *HP* economizer and energy balance equations for various parts of the HRSG as in fig. 1 can be written as:

$$T_{6g} = T_{20} + PP_{\mathrm{HP}} \tag{1}$$

$$T_{8g} = T_{11} + PP_{\mathrm{IP}} \tag{2}$$

$$T_{10g} = T_{6} + PP_{\mathrm{LP}} \tag{3}$$

$$\dot{m}_g \eta_{HRSG} (h_{4g} - h_{6g}) = [(h_{22} - h_{20}) + (h_{15} - h_{24})]\dot{m}_{s,HP} + \dot{m}_{s,IP} (h_{15} - h_{13}) \tag{4}$$

$$\dot{m}_g \eta_{HRSG} (h_{6g} - h_{8g}) = (h_{20} - h_{19})\dot{m}_{s,HP} + \dot{m}_{s,IP} (h_{13} - h_{11}) \tag{5}$$

$$\dot{m}_g \eta_{HRSG} (h_{8g} - h_{10g}) = (h_{19} - h_{18})\dot{m}_{s,HP} + \dot{m}_{s,IP} (h_{11} - h_{10}) + \dot{m}_{s,LP} (h_8 - h_6) \tag{6}$$

$$\dot{m}_s = \dot{m}_{s,LP} + \dot{m}_{s,IP} + \dot{m}_{s,HP} \tag{7}$$

The thermodynamic properties of water-steam in all steam cycle points were calculated. The input and output, pressure, temperature, enthalpy, and entropy of each section of HRSG in fig. 1 are determined, in order to define the mass flow rate of steam generation in the HRSG, heat balance diagram, and performance, also to conduct thermodynamic analysis.

Exergy analysis

Energy analysis based on the first law of thermodynamics cannot determine the quality of used energy, nor does it locate points of exergy destruction. In order to determine and quantify exergy destruction due to irreversibility, the appropriate tool is analysis by the second law of thermodynamics. Exergy can be divided into four components: physical, chemical, kinetic and potential. In this study, latter two are assumed to be negligible as the elevation changes and velocities are small [4]. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion process. Applying the first and the second laws of thermodynamics, the following exergy balance is obtained [10, 11]. Applying the first and the second law of thermodynamics, the following exergy balance is obtained:

$$\dot{E}x_Q + \sum_i \dot{m}_i ex_i = \sum_e \dot{m}_e ex_e + \dot{E}x_W + \dot{E}x_D \tag{8}$$

where ex_i and ex_e denotes the specific exergy of control volume inlet and outlet flow and $\dot{E}x_{\mathrm{D}}$ is the exergy destruction. Other terms in this equation are:

$$\dot{E}x_{\mathrm{Q}} = \left(1 - \frac{T_{\mathrm{o}}}{T_i}\right) Q_i \tag{9}$$

$$\dot{E}x_{\mathrm{W}} = \dot{W} \tag{10}$$

where $\dot{E}x_{\mathrm{Q}}$ and $\dot{E}x_{\mathrm{W}}$ are the corresponding exergy of heat transfer and work which cross the boundaries of the control volume. T is the absolute temperature and index o refers to the ambient conditions, respectively. It must be noted that, in this study, the exergy destructions caused by the heat losses from the components to the environment, are neglected in the first term of eq. (8), $Q_i = 0$ due to assumed ideal insulation. Therefore, the $\dot{E}x_{\mathrm{D}}$ will be:

$$\dot{E}x_{\mathrm{D}} = \sum_i \dot{E}x_i - \sum_e \dot{E}x_e - \dot{W} \tag{11}$$

where $\dot{E}x = \dot{m}ex$, term $\dot{E}x$ is defined as:

$$\dot{E}x = \dot{E}x_{\mathrm{PH}} + \dot{E}x_{\mathrm{CH}} \tag{12}$$

The specific physical exergy can be expressed as:

$$e_{\mathrm{PH}} = (h - h_{\mathrm{o}}) - T_{\mathrm{o}}(s - s_{\mathrm{o}}) \tag{13}$$

Specific chemical exergy of a substance can be obtained from standard chemical exergy tables [10, 11] relative to specification of the environment. For mixtures containing gases other than those presented in the reference tables, chemical exergy per unit mole can be evaluated using eq. (14). In operational calculations in this paper, the chemical exergy has been converted to exergy unit per mass:

$$e_{\mathrm{CH}} = \sum x_n (e_{\mathrm{CH}})_n \overline{\mathrm{R}} T_{\mathrm{o}} \sum x_n \ln x_n \tag{14}$$

where x_n is the mole fraction of the n_{th} gas in the mixture and $\overline{\mathrm{R}}$ is the universal gas constant. In the exergy analyses, another significant matter which must be noted is the reference conditions in tab. 1.

Thermal efficiency, exergy efficiency and exergy destruction

Thermal efficiency of the whole plant can be expressed as in eq. (15):

$$\eta_{\mathrm{CCGT}} = \frac{W_{\mathrm{GT}} + W_{\mathrm{ST}}}{\dot{m}_{\mathrm{f}} LHV} \tag{15}$$

where $\dot{m}_{\mathrm{f}}$ and LHV are fuel mass flow rate and lower heat value of the fuel, respectively.

The exergy efficiency defined as ratio of exergy output to exergy input; there are two equivalent expressions, given by eq. (16) [12].

$$\varepsilon = \frac{\Sigma \dot{E}x_{\mathrm{out}}}{\Sigma \dot{E}x_{\mathrm{in}}} \qquad \varepsilon = 1 - \frac{\dot{E}_{\mathrm{loss}}}{\Sigma \dot{E}x_{\mathrm{in}}} \tag{16}$$

$$\dot{E}_{\mathrm{loss}} = \dot{E}x_{\mathrm{D}} + \dot{E}_{\mathrm{west}} \tag{17}$$

where $\dot{E}_{\mathrm{loss}}$ is the exergy losses and $\dot{E}_{\mathrm{west}}$ – the unused exergy (exergy associated with exhaust gases).

The exergy efficiency $\varepsilon_{\mathrm{tot}}$ for the bottoming cycle CCGT in this work is given by:

$$\varepsilon_{\mathrm{tot}} = 1 - \frac{\dot{E}x_{\mathrm{D,tot}} + \dot{E}_{11g}}{\dot{E}_{4g}} \tag{18}$$

The exergy destruction rates $\dot{E}x_{\mathrm{D,tot}}$, according eq. (11), for the bottoming cycle CCGT and for each component of the bottoming cycle power plant (fig. 1) are written in tab. 2.

Table 2. Equations for the exergy destruction in each component for the system in power plant

Item	Exergy destruction equation	
HRSG	$\dot{E}x_{\mathrm{D,HRSG}} = \dot{E}_{4g} + \dot{E}_{24} + \dot{E}_{18} + \dot{E}_{10} + \dot{E}_{2} - \dot{E}_{22} - \dot{E}_{15} - \dot{E}_{8} - \dot{E}_{5} - \dot{E}_{11g}$	(19)
Steam turbine	$\dot{E}x_{\mathrm{D,ST}} = \dot{E}_{16} + \dot{E}_{23} + \dot{E}_{9} - \dot{E}_{26} - \dot{E}_{24} - \dot{W}_{\mathrm{ST}}$	(20)
Feed water pump	$\dot{E}x_{\mathrm{D,FP}} = \dot{E}_{5} + P_{\mathrm{pump,HP}} + P_{\mathrm{pump,IP}} - \dot{E}_{10} - \dot{E}_{18}$	(21)
Condenser	$\dot{E}x_{\mathrm{D,Con}} = \dot{E}_{26} - \dot{E}_{1}$	(22)
Bottoming cycle CCGT	$\dot{E}x_{\mathrm{D,\,tot}} = \dot{E}x_{\mathrm{D,ST}} + \dot{E}x_{\mathrm{D,HRSG}} + \dot{E}x_{\mathrm{D,FP}} + \dot{E}x_{\mathrm{D,con}}$	(23)

Calculation of the heat transfer area

The heat transfer area of HRSG is computed by method as:

$$Q_i = U_i A_i \, LMTD_i \tag{24}$$

where U and $LMTD$ refer to the global heat transfer coefficient and logarithm means temperature difference, respectively. The model is assumed to be counter flow heat exchanger. The values of overall heat transfer coefficient of the economizer, evaporator, superheater, and reheat sections of the HRSG are 42.6, 34.7, 50, and 50 [$\mathrm{Wm^{-2}K^{-1}}$], respectively [9]. The LMTD was calculated by:

$$LMTD_i = \frac{\Delta T_{1i} - \Delta T_{2i}}{\ln\left(\dfrac{\Delta T_{1i}}{\Delta T_{2i}}\right)} \tag{25}$$

where ΔT_1 represents the temperature difference between gas and steam at the inlet of the heater and ΔT_2 represents the temperature difference between gas and steam at the exit of the heater. The HRSG area A which is necessary to ensure the heat transfer at a given PP was calculated by:

$$A_{\mathrm{HRSG}} = \sum_{ec} A_{\mathrm{ec}} + \sum_{v} A_{\mathrm{v}} + \sum_{su} A_{\mathrm{su}} + \sum_{re} A_{\mathrm{re}} \tag{26}$$

Economic analysis

In this methodology it is necessary to estimate the annual cost associated with owning and operating each plant component and the annual cost associated with exergy destruction.

Total capital cost and annualized cost

Several methods have been suggested to express the purchase cost of equipment in eqs. (27) and (28). However, we have used the cost functions for components of the CCGT were taken from literature: cost of heat recovery steam generator $C_{\mathrm{I,HRSG}}$ [5], cost of steam turbine $C_{\mathrm{I,ST}}$ [13], cost of pump $C_{\mathrm{I,pump}}$ [13], cost of generator $C_{\mathrm{I,gen}}$ [13], cost of gas turbine $C_{\mathrm{I,GT}}$ [14], and cost of condenser $C_{\mathrm{I,con}}$ [14],. The total capital costs (investment costs) of a combined cycle gas turbine $C_{\mathrm{I,CCGT}}$ and the cost of HRSG $C_{\mathrm{I,HRSG}}$ are given by:

$$C_{\mathrm{I,CCGT}} = R(C_{\mathrm{I,GT}} + C_{\mathrm{I,HRSG}} + C_{\mathrm{I,ST}} + C_{\mathrm{I,con}} + C_{\mathrm{I,\,pump}} + C_{\mathrm{I,gen}}) \tag{27}$$

where R is a correction factor assumed to be 3.0 [1] and

$$C_{\mathrm{I,HRSG}} = 2.31(k_{\mathrm{ec}}A_{\mathrm{ec}} + k_{\mathrm{v}}A_{\mathrm{v}} + k_{\mathrm{su}}A_{\mathrm{su}} + k_{\mathrm{re}}A_{\mathrm{re}}) \tag{28}$$

where k_{ec}, k_{v}, k_{su}, and k_{re}, are the unit price of surface area of the economizer, evaporator, super heater, and reheat sections of the HRSG, respectively [9].

The annualized investment costs $C_{\rm Ia,CCGT}$ and $C_{\rm Ia,HRSG}$ are calculated by:

$$C_{\rm Ia,CCGT} = \frac{C_{\rm I,CCGT}}{N} \quad \text{and} \quad C_{\rm Ia,HRSG} = \frac{C_{\rm I,HRSG}}{N} \tag{29}$$

where N is economic life of the plant.

Cost of exergy destruction

The exergy destruction cost in HRSG $C_{\rm D,HRSG}$ can be expressed by eq. (30). [9, 15]

$$C_{\rm D,HRSG} = cf\, H \dot{E}_{\rm D,HRSG} \tag{30}$$

where cf is the price of the fuel (natural gas) as given in tab. 1, H – the number of operating hours of the plant per year, and $\dot{E}x_{\rm D,HRSG}$ – the exergy destruction in HRSG.

Optimization

The objective functions, operating parameters and constraints, as well as the overall optimization are considered in this section. The first step of system optimization is to specify an appropriate objective function which can be either thermodynamic or a thermoeconomic one. In this study, the initial case parameters for the case study are listed in tab. 3.

Table 3. Initial case parameters

No.	Parameter	Simbol	Value
1	The pinch point temperature difference for *HP*, *IP* and *LP* [°C]	*PP*	13
2	Live steam pressure *HP* [bar]	p_{22}	104
3	Pressure of reheat steam (*IP* steam turbine) [bar]	p_{15}	36
4	Pressure of the inlet *LP* steam turbine [bar]	p_9	5
5	Live steam temperature at the inlet of the *HP* steam turbine [°C]	T_{22}	545
6	Temperature of the superheated steam [°C]	T_8	235
7	Low-pressure steam turbine outlet [bar]	p_{26}	0.055
8	Feed water temperature [°C]	T_3	60
9	Temperature of the superheated steam at 13 [°C]	T_{13}	325

Thermodynamic optimization

The thermodynamic objective function in this section was defined as the minimize exergy destruction ($Ex_{\rm D}$). Minimization of exergy destruction ensures that the HRSG will operate efficiently. The exergy destruction was chosen as the objective function, which can successfully measure both the quality and quantity of energy flow in the plant. Although this approach dose not considers costs, it proposes some rough design of the HRSG parameters.

Thermoeconomic optimization

Definition of objective functions

An objective function is here defined as the sum of two parts; the annual capital cost $C_{\rm Ia+D,HRSG}$ which stands for the capital investment and maintenance expenses, and the corresponding cost of exergy destruction of the HRSG. Therefore, the objective function *OF* represent the total annual cost rate of the HRSG in terms of dollars per year and it is defined as:

$$OF = C_{\mathrm{Ia+D,HRSG}} = C_{\mathrm{Ia,HRSG}} + C_{\mathrm{D,HRSG}} \tag{31}$$

The objective function is to be minimized so that the values of optimal operating parameters would be obtained. The decision variables (operating parameters) considered in this study are: pinch points (PP_{LP}, PP_{IP}, PP_{HP}) and drum pressures (LP, IP, and HP).

Production cost of electricity

The production cost of electricity C_{kWh} is the total annual cost of C_{tot} the CCGT plant divided by mean annual energy output $W_{\mathrm{CCGT}}H$ [16]:

$$C_{\mathrm{kWh}} = \frac{C_{\mathrm{tot}}}{W_{\mathrm{CCGT}}H} \tag{32}$$

The total annual cost C_{tot} includes the total fuel cost C_{Tf}, the annual investment cost $C_{\mathrm{Ia,CCGT}}$ and the operating and maintenance cost C_{om} which is assumed to be the 10% of the total plant cost:

$$C_{\mathrm{tot}} = C_{\mathrm{Tf}} + C_{\mathrm{Ia,CCGT}} + C_{\mathrm{om}} \tag{33}$$

$$C_{\mathrm{Tf}} = cf\,\dot{m}_{\mathrm{f}}\,LHV \tag{34}$$

where cf, $\dot{m}_{\mathrm{f}}$, and LHV are fuel cost, fuel mass flow rate, and lower heat value of the fuel, respectively.

Results and discussions

As mentioned, the simple iterative optimization has been performed to find the maximum exergetic efficiency of the bottoming cycle system and the minimum total annual cost rate of the HRSG.

The optimization procedure

In this study a simple procedure for optimizing six steam cycle parameters was developed. The procedure was described as below.

First step: optimum pinch points (PP_{LP}, PP_{IP}, PP_{HP})

(1) Seeking the optimum PP_{LP}. In order to find the optimum PP_{LP} the initial values for steam cycle parameters PP_{IP}, PP_{HP}, LP, IP, and HP were taken from the initial case. The value for PP_{LP} varies in the range 3-30 °C in steps of 0.5 °C. Both thermodynamic parameters and thermoeconomic parameters are calculated. The optimal value for PP_{LP} is determined based on the minimized OF and constrain limitations. However, the considered PP_{LP} in current study has to be higher than optimum one to maintain the exhaust gas temperature above the dew point of potentially corrosive acidic vapors.
(2) In order to define its optimal value PP_{IP}, it was varied over using the previously determined PP_{LP}, while other steam cycle parameters were kept constant from the initial case.
(3) A procedure similar to the one mentioned in item 2 is applied to find the optimum PP_{HP}.
(4) The procedure is then repeated as in 1, 2, and 3 for new refinements of pinch points, until the values converge and give the optimum PP_{LP}, PP_{IP}, and PP_{HP} for this step in several iterations.

Second step: optimum (LP, IP, and HP)

(1) The determined optimum values of PP_{LP}, PP_{IP}, and PP_{HP} were considered constant during this step.
(2) In order to find optimal values for LP, IP, and HP, the value for one parameter had been varied, while the other two parameters were kept constant. The procedure is then repeated for

the other two steam drum pressures. In this proposed method, the *HP*, *IP*, and *LP* were varied in following ranges: *HP* = 120-200 bar, *IP* = 34-60 bar, and *LP* = 1-10 bar.

Third step

The newly determined *LP*, *IP*, and *HP* were kept unchanged and the first step was repeated to find new optimum *PP* value. Then the second step was repeated to find the new optimum drum pressure values. After that, both first and second steps were repeated in an iterative manner until the values of the steam cycle parameters converge.

Result of the thermodynamic optimization

Pinch point PP_{LP}, PP_{IP}, and PP_{HP}

In this model, the influences of pinch points on the exergy efficiency and exergy destruction were investigated as shown in fig. 2. As the *PP* increases, the exergy efficiency decreases while the exergy destruction increases. This indicates that large amounts of heat are dissipated to atmosphere without being recovered by the HRSG. The irreversibility of the steam turbine is decreased, because of low mass flow rate from steam generation in HRSG. From the previous discussion it is clear that the maximum efficiency, minimum exergy destruction and maximum steam turbine gross power will be reached at a null value for *PP* and with an infinite heat transfer surface HRSG area.

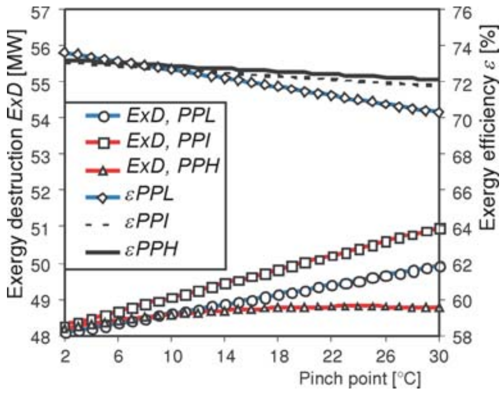

Figure 2. Effect of *PP* variations on the exergy efficiency and exergy destruction

Drum pressures (LP, IP, and HP)

Exergy efficiency and exergy destruction *vs*. drum pressures are presented in figs. 3, 4, and 5. Figure 3 shows that the exergy efficiency increases with variation of *LP* to a maximum of 4 bar and then it starts to decrease, while the exergy destruction reduces. Figure 4 shows that the

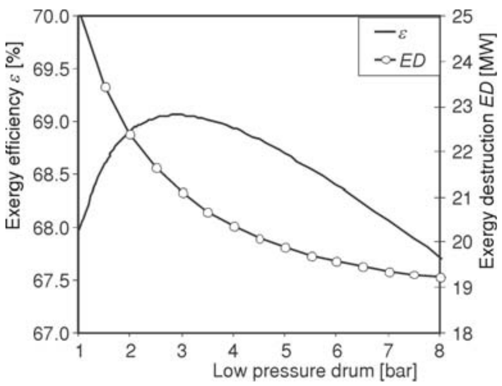

Figure 3. Effect of *LP* variations on the exergy efficiency and exergy destruction

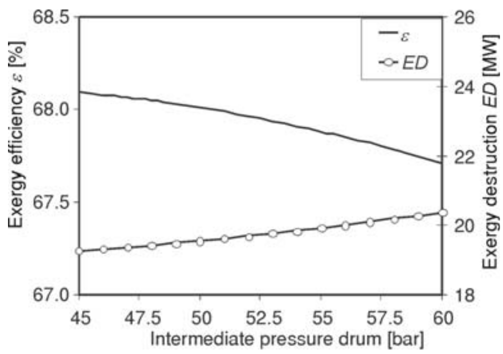

Figure 4. Effect of *IP* variations on the exergy efficiency and exergy destruction

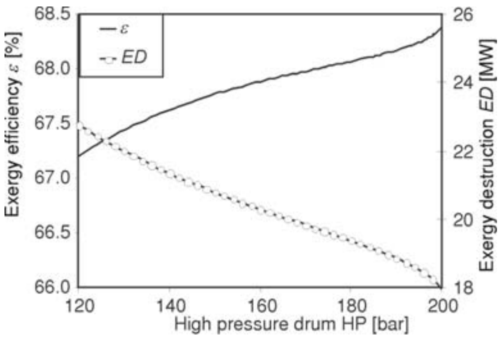

Figure 5. Effect of *HP* variations on the exergy efficiency and exergy destruction

exergy efficiency increases and the exergy destruction reduce with the reduction of *IP*. Figure 5 shows that the exergy efficiency increases while exergy destruction reduces by increasing the *HP*. It is easy to notice from the previous explanation that the *HP* must be high to attain a good exergetic utilization of the wasted heat by generating a high quality steam. This means that there is no upper limit value for the *HP*. Hence the *IP* steam pressure must be low and the lower limit of *IP* is the temperature difference between T_{9g} and T_{18} which should be positive.

Result of thermoeconomic optimization

In order to conduct a correct thermoeconomic optimization of the considered plant, fig. 1, the influences of *PP* and *LP*, *IP* and *HP* on the $C_{D,HRSG}$ and $C_{Ia,HRSG}$ were investigated as shown in figs. 6, 7, 8, and 9. There is an opposing response of $C_{D,HRSG}$ and $C_{Ia,HRSG}$ as a result of varying the *PP*, *LP*, *IP*, and *HP*. That means there is only one optimal point for each parameter which gives the minimum HRSG investment cost and minimum exergy destruction cost.

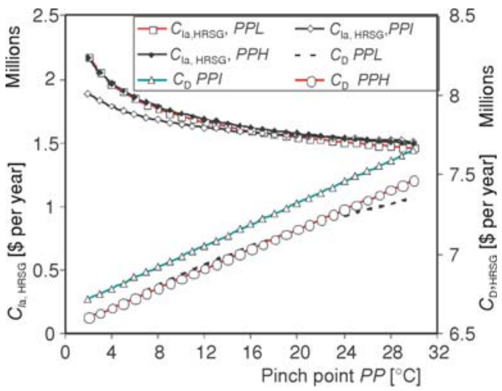

Figure 6. Effect of *PP* variations on the investment cost and exergy destruction cost

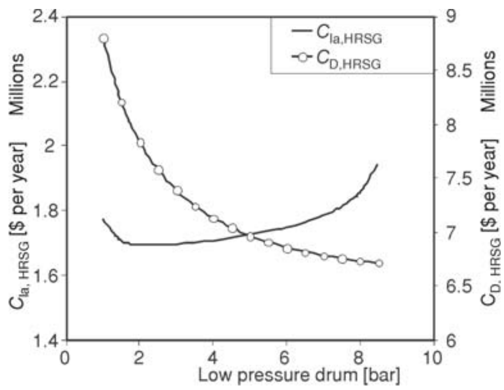

Figure 7. Effect of *LP* variations on the investment cost and exergy destruction cost

Figure 6 shows that the exergy destruction cost increases, while the investment cost decreases by increasing the PP_{LP}, PP_{IP}, and PP_{HP}. Figures 7 and 9 show that the investment cost increases, while the exergy destruction decreases by increasing the *LP* and *HP*. On the other hand, the increase of *IP* produces opposite effects, as shown in fig. 8.

In this section, results of optimization are presented. Figure 10 shows the behavior of *OF* with variation of *PP*. The minimum *OF* is achieved for the following values of *PP* at different pressure levels: PP_{LP} = 7.5 °C, PP_{IP} = 4 °C, PP_{HP} = 8 °C. When the value of *PP* is kept constant for all three pressure levels, the optimum has been achieved for *PP* = 7.5 °C. However, comparing the plotted curves of these two cases (pinch point variable PP_{LP}, PP_{IP}, PP_{HP}, and pinch point constant *PP*), it can be observed that the change of obtained gradient of *OF* in the second case is significantly greater than in the first case as shown in fig. 10.

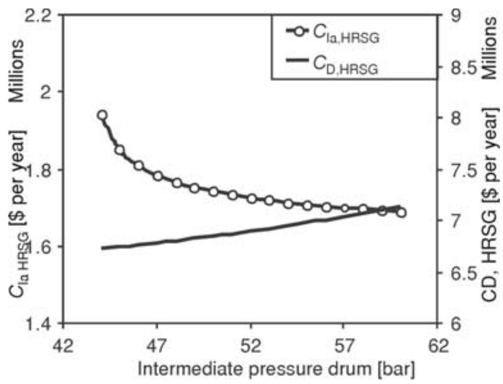

Figure 8. Effect of *IP* variations on the investment cost and exergy destruction cost

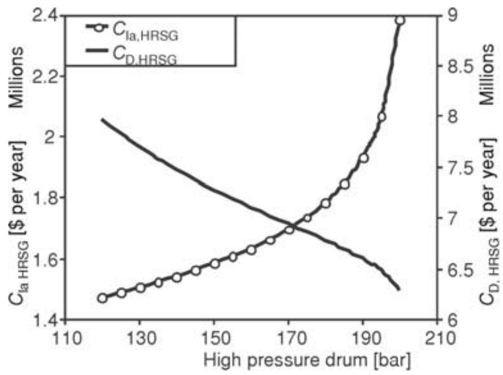

Figure 9. Effect of *HP* variations on the investment cost and exergy destruction cost

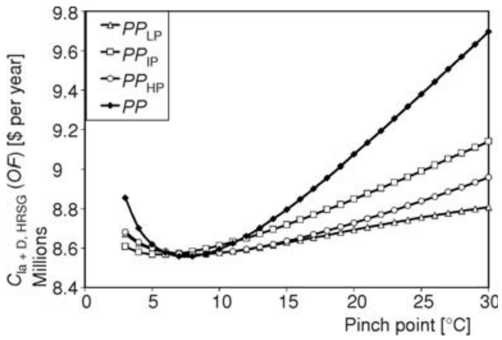

Figure 10. Effect of *PP* variations on the objective function

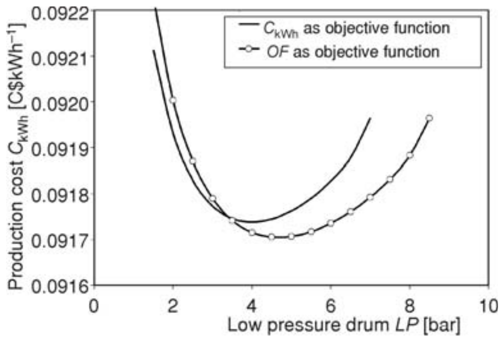

Figure 11. Effect of *LP* variations on the production cost for different objective function

This provides a wider range for the increment of one of *PP* pressure levels while maintaining the other two within the optimum values, which results in relatively lower amount of total annual cost *OF* increase compared to the second procedure. This can be applied especially with the high pressure level PP_{HP}, because the investment cost of the heater for the high pressure level is four times greater than for the intermediate pressure level and 2.4 times greater than for the low pressure level. The comparative results between the base optimum case (the total annual cost rate *OF* of the HRSG as objective function) and the second optimum case (where is the production cost C_{kWh} is used as the objective function) are presented in figs. 11, 12, and 13. From fig. 11, it could be observed that the value of minimal C_{kWh} for the base case is lower than its minimal value in the second case. In the case with *IP*, it can be seen that the minimal C_{kWh} approximately equal for both cases, as shown in fig. 12.

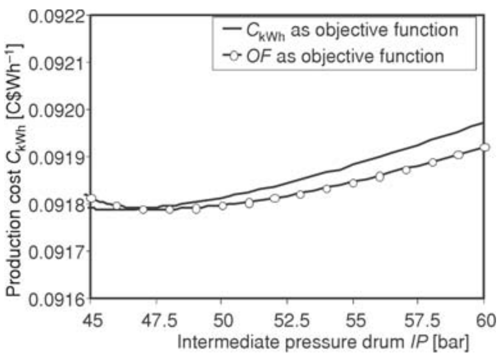

Figure 12. Effect of *IP* variations on the production cost for different objective function

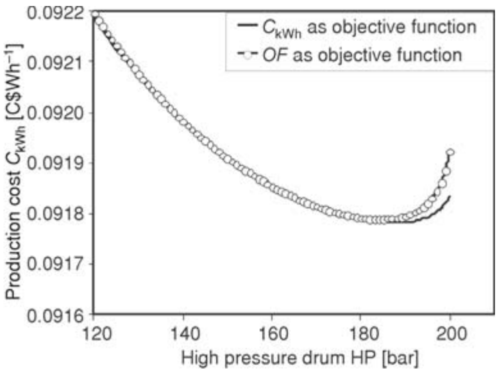

Figure 13. Effect of *HP* variations on the production cost for different objective function

The effect of variation in the HP with C_{kWh} is shown in fig. 13. The value of the minimal C_{kWh} in the base case is higher than its value in the second case. On the other hand, due to the pressure limitation in the *HP* (its value must be less than critical one; the limit was defined according to the professional experience at the level of 180 bar), the C_{kWh} was identical for both cases below this pressure limit.

Table 4 shows a comparison of the results for the initial case and the optimized case. The results show that the economic parameters of the optimized case are significantly better than in the initial case. Thermoeconomic optimization intends to achieve a trade-off between enhancing the efficiency and minimum C_{kWh}. In our case, applying the developed method, the efficiency and electrical output of the selected combined cycle were increased. On the other hand, the production costs C_{kWh} of electricity, exergy destruction cost and total annual cost of HRSG were decreased.

Table 4. Comparison between the initial case and optimized case

Parameter		Initial case	Optimized case	Change
Pinch point	PP_{LP}	13	7.5	
	PP_{IP}		4	
	PP_{HP}		8	
High drum pressure [bar]		104	180	
Intermediate drum pressure [bar]		36	47	
Low drum pressure [bar]		5	7	
Combined cycle thermal efficiency [%]		57.461	58.416	+1.66%
Combined cycle gross power [MW]		418.95	425.92	+1.67%
Exergy destruction cost [$ per year]		18,169,128	16,975,175	–1,193,953
HRSG investment cost [$ per year]		1,123,164	1,784,763	+661,599
HRSG Total annual cost [$ per year]		9,777,179	8,568,394	–1,208,785
Production cost of electricity [c$/kWh]		9.2768	9.1792	–1.0%

Conclusions

In the present study, thermodynamic and thermoeconomic modeling of a triple pressure CCGT were conducted. An optimization method for the operating parameter (PP_{LP}, PP_{IP}, PP_{HP} and *LP*, *IP*, *HP*) of the HRSG was developed. The aim of these optimizations was to improve the performances of power plants and to enhance the economics of the plants. The conclusion based on thermodynamic optimization established that the zero pinch points is optimum one, as expected. Also, the optimal value for steam pressures could not be found from the thermodynamic optimization. Thermoeconomic optimization through the proposed procedure successfully leads to optimal operating parameters with the aim to minimize C_{kWh} and to minimize the total annual cost rate of the HRSG (investment cost and exergy destruction cost). In addition, two comparisons were also undertaken: first one between an initial case and an optimization case and second between C_{kWh} resulting by two objective functions $C_{D,HRSG}$ and C_{kWh}. The conclusion of these comparisons is that the optimization case with $C_{D,HRSG}$ as objective function, gives the best results in the sense of production cost of electricity and total annual cost of HRSG. In the considered case, the production cost of electricity was decreased by 1%, the annual total costs were decreased by about 12.3%, thermal efficiency was increased by 1.66% and the power production increase by 1.67% compared with initial case. As a final conclusion, the considered optimization method with proposed objective function could achieve the desired goal.

Nomenclature

A – heat transfer area, [m^2]
C_D – exergy destruction cost, [$ per year]
C_I – investment cost, [$]
C_{Ia} – annualized investment cost, [$ per year]
C_{kWh} – production cost of electricity, [$ per kWh]
C_{Tf} – total annual cost of the fuel, [$ per year]
cf – price of the fuel (natural gas), [$ per kWh]
$\dot{E}x$ – exergy rate, [kJ]
e – specific exergy, [kJkg^{-1}]
H – number of operating hours of the plant per year, [hour]
HP – high pressure, [bar]
h – specific enthalpy, [kJkg^{-1}]
IP – intermediate pressure, [bar]
LHV – lower heat value of the fuel, [kJkg^{-1}]
$LMTD$ – log mean temperature difference, [°C]
LP – low pressure, [bar]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
N – economic life of the plant, [year]
OF – total annual cost rate of the HRSG [$ per year]
PP – pinch point temperature difference, [°C]
Q – heat transferred, [kW]
S – selling price of kWh, [$ per kWh]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [K or °C]
U – overall heat transfer coefficient, [kWm^{-2}K^{-1}]
W – power, [MW]

Greek symbols

ε – exergy efficiency
η – thermal efficiency

Subscripts

CH – chemical
CCGT – combined cycle gas turbine
con – condenser
e – outlet
ec – economizer
FB – feed water pump
GT – gas turbine
g – gas
gen – generator
HP – high pressure
HRSG – heat recovery steam generator
i – inlet
IP – intermediate pressure
LP – low pressure
out – outlet from the gas turbine
PH – physical
pump – pump
re – reheater
ST – steam turbine
s – steam
su – superheater
tot – total
v – evaporator
o – dead state condition

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