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The Effect of the Condenser Inlet Cooling Water Temperature on the Combined Cycle Power Plant Performance

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Abstract
This paper focuses on the influence of the condenser inlet cooling water temperature on thermoeconmic parameters of the power plant. The thermoeconomic parameters comprise the production costs per unit of electrical output and the annual cash flow of the power plant. Therefore, the thermoeconmic parameters are generated for two different range of condenser inlet cooling water (summer/winter case). A combined cycle gas turbine (CCGT) power plant (284 MW) with a typical design of commercial combined cycle power plant was demonstrated as a model. The results of the study revealed that if the condenser inlet cooling water temperature reduced to 10 ^{o}C (winter case), the annual cash flow is increased to 3.7 M$, and the production cost of the electricity is decreased to 0.147 cent-dollar per kilowatt-hour. However, if the condenser inlet cooling water temperature increased to 30 ^{o}C (summer case), the annual cash flow is decreased to 3.6 M$, and the production cost of the electricity is increased to 0.148 cent-dollar per kilowatt-hour.

Keywords: Cooling Water, Condenser Pressure, Steam Condenser, Thermoeconomic

1. Introduction

The condenser is a heat transfer device or unit used to condense a substance from its gaseous state to its liquid state, typically by cooling it. The main use of the condenser is to receive the exhaust steam from the steam turbine and condense it. The steam condenser generally condenses the steam to a pressure below atmospheric. This allows the turbine to generate additional work. The condenser also converts discharge steam back to feed water, which is returned to steam generators. In the condenser, the latent heat of condensation is conducted to the medium flowing through the cooling tubes. The thermal power plants are designed based on required conditions, however in the actual case the inlet conditions are not as per the designed conditions. Moreover, in practical situations, once the power plants are installed there are several constrains. This will possibly tends to decrease or increase the power output and the heat rate of the thermal power plants. Therefore, the designed power and the heat rate are difficult to achieve. Moreover, the designed features of the condenser such as (condenser heat transfer area) have a significant impact, but it is expansive to replace them when the plant is operating. Different approaches can be found in the literature regarding the performance of the steam condenser. (Alus: 2017) performed a thermoeconmic optimization of steam condenser for combined cycle power plant. The authors found that the thermoeconmic optimization procedure led to a significant improvement for the economic parameters, where the annual cash flow was increased by 1.1 M$, and the production cost of electricity was decreased by 0.043 cent-dollar per kilowatt-hour. (Haldkar: 2013) studied the parameters, which affect the performance of the condenser and the power plant. The authors found that the performance of the condenser was varied due to the cooling water inlet temperature, flow rate of cooling water and pressure in energy efficiency. (Mirjana: 2010) studied the influence of the cooling water temperature and flow rate on the condenser performance, and thus on the specific heat rate of the plant and its energy efficiency. The authors concluded that, increasing the condenser inlet cooling water, especially in the summer, led to increasing the condenser pressure and reducing the energy efficiency of the power plant. The earlier studies revealed that the condenser inlet cooling water has an influence on the condenser pressure and the energy efficiency of the thermal power plants. These aspects have prompted the authors of this research paper to focus on studying the effect of the condenser inlet cooling water on the thermoeconomic parameters.

2. Thermodynamic model

The thermodynamic model provides a modular structure for the triple-pressure HRSG combined cycle power plan (CCGT). Figure 1 shows a schematic diagram of a CCGT. The compute code to calculate the heat balance of triple-pressure HRSG of CCGT was developed using FORTRAN 90. The water-steam properties were derived from the standard IAPWS-IF97 (Wagner: 1998). The properties of the gas turbine exhaust gases, which are combustion products of the specified fuel, were calculated according to (Baehr: 1988). The assumptions and the studied parameters, which are selected for the thermodynamic analysis of the plant, are presented in the appendix according to (Alus: 2014).

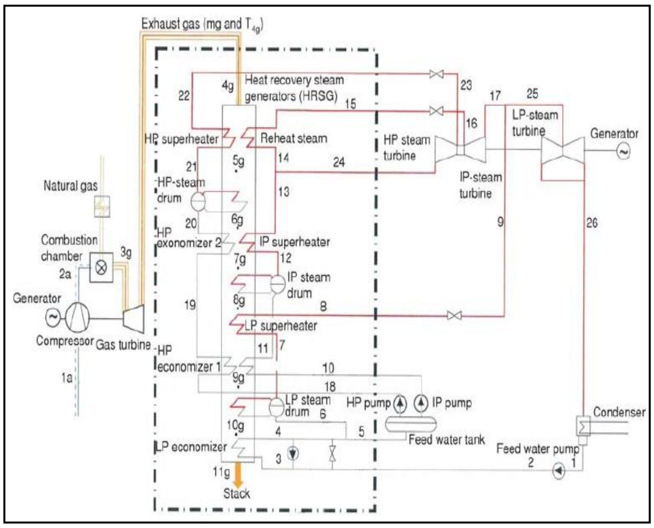

Fig.1: Thermodynamic plant model- heat balance diagram

2.1 Element of surface condenser

In power plants, the most commonly used condensers are the steam surface condensers. Consequently, steam surface condenser with open cycle cooling water supply system in this study is assumed. Figure 2 presents the heat transfer diagram of the steam surface condenser according to the heat balance diagram in Figure 1. The steam temperature (t_s) is the saturation temperature at the condenser pressure. The difference between the steam temperature (t_s) and the water inlet temperature (tw_1) is defined as the initial temperature difference (ITD). The difference between the steam temperature (t_s) and the water outlet temperature (tw_2) is known as the terminal temperature difference (TTD). The circulating water inlet temperature should be sufficiently lower than the steam saturation temperature to result reasonable values of *TTD*. The difference between the water outlet temperature (tw_2) and the water inlet temperature (tw_1) in the condenser is defined as temperature rise (TR) (Alus: 2017).

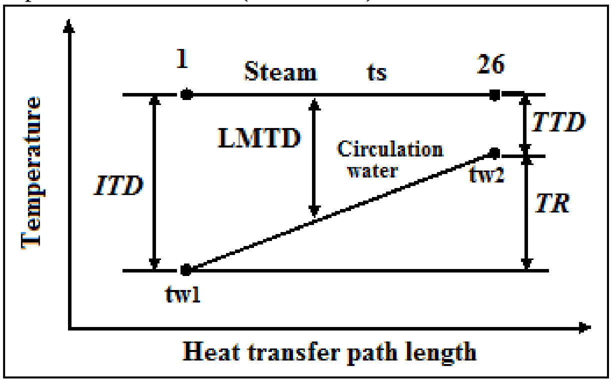

Fig. 2: Heat transfer diagram of condenser

2.2 Saturation temperature of condenser calculation

The saturation temperature (t_s) was determined by equation 1 according to (Zhao: 2013). The saturation pressure corresponding to the saturation temperature (t_s) is condenser pressure $(p_{cond.})$. The saturation temperature (t_s) can be calculated under different inlet cooling water (sea water source) and through iterative calculation.

$$t_s = t_{w1} + TR + TTD \qquad (1)$$

Where TR is assumed to be 10 ºC according to (Li: 1985). TTD is assumed to be 2 ºC according to (Kapooria: 2008).

2.3 Condenser heat transfer area calculations

The heat transfer rate between the cooling water and the steam vapour is the key parameter of the thermal analysis of the steam surface condenser (Li: 1985).

The heat from the exit steam is:

$$\dot{Q}_{ST} = \dot{m}_{ST}\left(h_{26} - h_1\right) \qquad (2)$$

The heat which the cooling water acquire:

$$\dot{Q}_w = \dot{m}_w \, cp_w (tw_2 - tw_1) \qquad (3)$$

Where $\dot{m}_w$ represents the cooling water flow rate in (kg/s), and cp_w is the specific heat of the water in (kJ/kg.K).

The heat transferring in condenser is:

$$\dot{Q}_{Cond.} = A_{Cond.} \cdot U_{Cond.} \cdot LMTD_{Cond.} \qquad (4)$$

Where, $\dot{Q}_{Cond.}$ is the heat transferred and $U_{Cond.}$ is the heat transfer coefficient for the condenser in (W/m²? K).

The logarithmic mean temperature difference ($LMTD_{Cond.}$) for the condenser is defined as:

$$LMTD_{Cond.} = \frac{TR}{\ln \dfrac{1}{1 - \dfrac{TR}{ITD}}} \qquad (5)$$

The condenser heat transfer area ($A_{Cond.}$) is defined as:

$$A_{Cond.} = \frac{\dot{Q}_{Cond}}{U_{Cond.} \cdot LMTD_{Cond.}} = \frac{\dot{m}_{ST}\,(h_C - h_1)}{U_{Cond.} \cdot LMTD_{Cond.}} = \frac{\dot{m}_w\, cp_w (tw_2 - tw_1)}{U_{Cond.} \cdot LMTD_{Cond.}} \quad (6)$$

2.4 Thermodynamic parameters

The thermodynamic parameters of this study are the power output and the overall cycle efficiency. The power output and thermal efficiency for a combined cycle plant are defined by the following equations (Li: 1985):

$$W_{CCGT} = W_{GT} + \sum W_{ST}\,(\dot{m}_{ST}, h_{cond}, \eta_{is,ST}) \quad (7)$$

The thermal efficiency in a combined cycle plant is defined by the following relation:

$$\eta_{CCGT} = \frac{W_{CCGT}}{Q_{add}} = \frac{W_{ST}\,(\dot{m}_{ST}, h_{cond}, \eta_{ST})}{Q_{add}} \quad (7)$$

Where $\dot{m}_f$ is mass flow rate of fuel in (kg/s), H_l Lower heat value of the fuel (kJ/kg).

3. Thermoeconomic model

To perform the thermoeconmic analysis, some economic parameters should be assumed, and relations between the thermodynamic parameters and capital cost have to be established and thermoeconomic parameters defined. The economic assumptions, which are selected for the thermoeconomic analysis of the plant, are presented in the appendix.

3.1 Functions of component capital costs

The basic problem in the analysis of the economic effectiveness of investments in energy systems is the determination of capital costs. In this study, the cost functions for the major components of the combined cycle power plants were taken from literature: cost of gas turbine C_{GT} (Roosen: 2003), cost of HRSG C_{HRSG} (Behbahani: 2010), cost of steam turbine C_{ST}, cost of pump C_{Pump}, cost of generator $C_{Gen.}$ (Silveira: 2003) and Cost of condenser $C_{Cond.}$ (Attala: 2001).

The total capital costs (investment costs) of a combined cycle power plants are given by:

$$C_{CCGT} = R \cdot \sum_i C = R \cdot \left(C_{GT} + C_{HRSG} + C_{ST} + C_{Cond.} + C_{Pump} + C_{Gen.}\right) \quad (8)$$

Where R is the factor which covers the addition cost such as: transport and assembly costs, supervising, accessories, engineering and project management, commissioning and other related costs (Alus: 2014).

3.2 Thermoeconomic parameters

The cost functions for the major components and the economic assumptions of the combined cycle power plants were taken according to (Alus: 2014). The annual cash flow (B) and the production cost of electricity (C_{kWh}) have been chosen as main parameters of the thermoeconomic.

The annual cash flow (B) is the difference between the annual total income (I_{tot}) and the total cost per year (C_{tot}) as pointed out by (Valdes: 2003 and Manuel: 2004):

$$B = I_{tot} - C_{tot} \quad (9)$$

The production cost of electricity was defined as pointed out by (Valdes: 2003 and Manuel: 2004):

$$C_{\mathrm{kWh}} = \frac{C_{\mathrm{tot}}}{W_{CCGT} \,.\, h} \quad (10)$$

Where C_{tot} the total cost per year and the product $(W_{CCGT} \,.\, h)$ is the total annual energy production.

The annual total income can be calculated using the following equation:

$$I_{tot} = S \cdot W_{CCGT} \cdot h \quad (11)$$

where S is the selling price per unit of electricity.

The total cost per year includes the total annual fuel cost, the amortization cost and the operating and maintenance cost, as shown in the following equation:

$$C_{tot} = C_{TF} + C_a + C_{o\&m} \quad (12)$$

The total annual fuel cost (C_{TF}) is defined as follows:

$$C_{TF} = c_f \cdot \left(\frac{W_{CCGT}}{\eta_{CCGT}}\right) \cdot h \quad (13)$$

The amortization cost (C_a) summarized as pointed out by (Manuel: 2004):

$$C_a = \frac{C_{CCGT}}{N} \quad (14)$$

The operating and maintenance cost $(C_{o\&m})$ is assumed to be 10 % of the total plant cost as pointed out by (Manuel: 2004):

$$C_{o\&m} = 0.10 \cdot (C_{tot}) \quad (15)$$

Results & Discussion

The condenser inlet cooling water ($t_{w1.}$) was in the range of 5 to 35 ºC by 1 ºC adding incrementally in each step. The thermodynamic and the thermoeconomic parameters were calculated for each step.

Figure 4 shows the Effect of condenser inlet cooling water ($t_{w1.}$) variations on the condenser pressure ($p_{Cond.}$), which is explained by Eq. 1. It can be seen that the inlet cooling water temperature rise, then the condenser pressure increases. As the inlet cooling water temperature increases continually, the condenser pressure increases quickly.

Figure 5 shows the combined cycle gross power and thermal efficiency as a function of condenser inlet cooling water. The result shows that the combined cycle gross power and thermal efficiency decreases with increase in t_{w1}. Decreasing t_{w1} will result to higher power output for the same mass flow rate and fuel input into gas turbine unit, resulting in higher work output of the steam turbine.

Figure 6 shows the effect of the condenser inlet cooling water (t_{w1}) on the annual cash flow (B), which is defined by Eq. 10. As can be seen, annual cash flow (B) increases with increase in t_{w1}. The result shows that the optimal value for t_{w1}, at which the maximum B is not observed.

Figure 7 shows the effect of t_{w1} on C_{kWh}, which is defined by Eq. 11 for CCGTs with a triple-pressure HRSG. As seen from that, C_{kWh} decreases with an increase in t_{w1}. The result shows that the optimal value for t_{w1}, at which the minimum C_{kWh} is not observed.

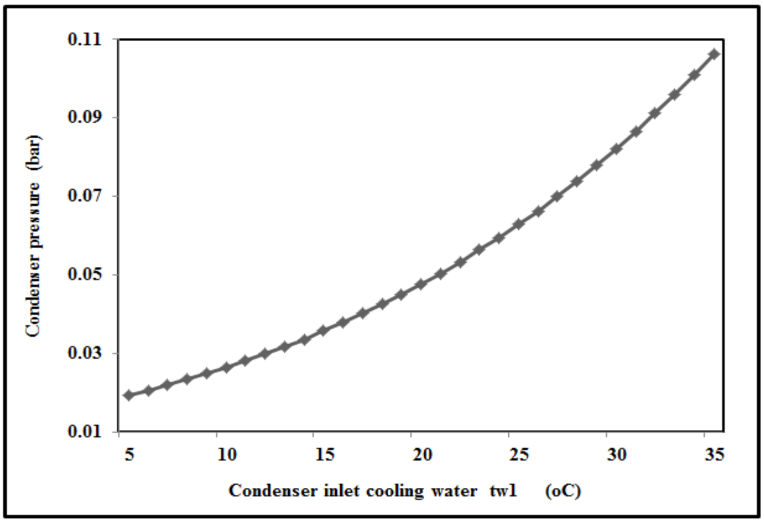

Fig. 3: Effect of condenser inlet cooling water variations on the condenser pressure

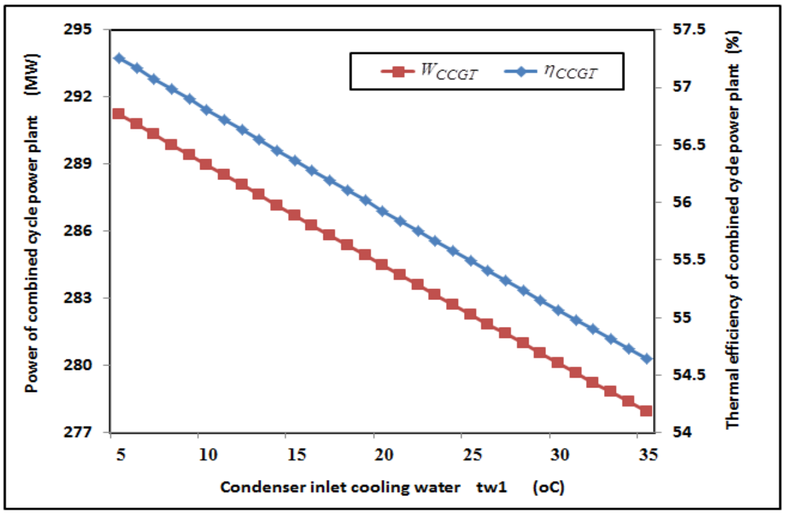

Fig. 4: Effect of condenser inlet cooling water variations on the combined cycle gross power and thermal efficiency

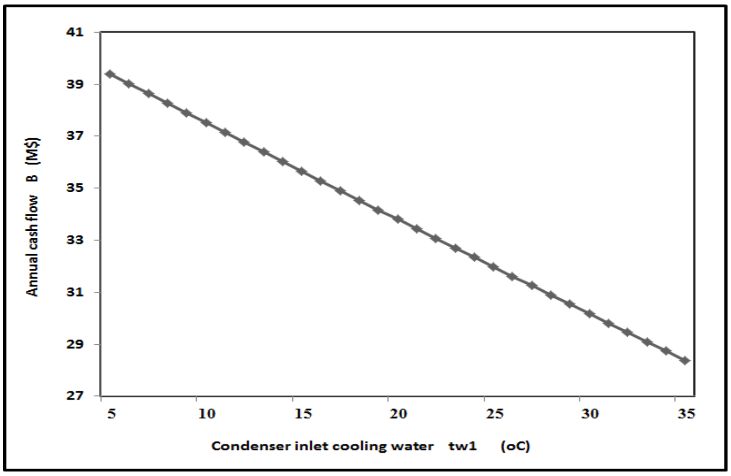

Fig. 5: Effect of condenser pressure variations on the annual cash flow

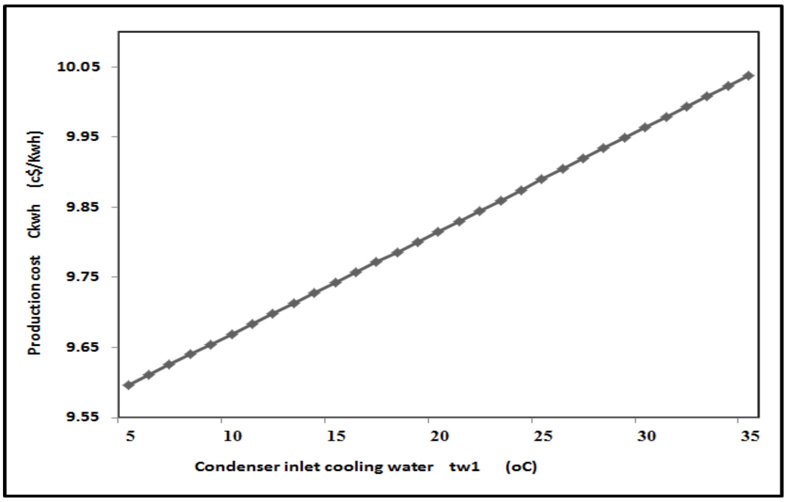

Fig. 6: Effect of condenser pressure variations on the production cost of electricity

Table 2 shows a comparison of the results between the designed case and the winter/summer case of triple-pressure HRSG. The results show that the financial parameters are significantly better than the designed case in winter case and worse than designed case in summer. In the winter, the efficiency of the selected combined cycle configuration could be increased in by about 0.88 % and the power output by more than 4 MW. On the other hand, the B was increased by 3.7 Million-dollar. In addition, by selecting the t_{w1}, the C_{kWh} was decreased by 0.147 cent-dollar per kilowatt-hour.

Table 2: Comparison between the designed case and winter/summer case of triple-pressure HRSG CCGTs

Parameter	Designed case	Studied case	
		Winter case	Summer case
Condenser inlet cooling water (t_{w1})	20 °C	10 °C	30 °C
Condenser pressure ($P_{Cond.}$)	0.0467 bar	0.0265 bar	0.0821 bar
Combined cycle-efficiency (η_{CCGT})	55.93 %	56.81 %	55.07 %
Combined cycle-gross power (W_{CCGT})	284.5 MW	288.96 MW	280.12 MW
Production cost (C_{kWh})	9.816 c$/kWh	9.669 c$/kWh	9.964 c$/kWh
Annual cash flow (B)	33.808 M$	37.513 M$	30.177 M$

Conclusions

In this research paper, the influence of the condenser inlet cooling water temperature on the thermoeconmic parameters of the power plant was investigated. The results indicated that, increasing the condenser inlet cooling water in the summer led to an increase of the condenser pressure and also decrease of the thermal efficiency of the CCGT.
Comparing the designed case with the winter/ summer case (studied case), the annual cash flow in the winter was increased to 3.7 M$, and the production cost of the electricity was decreased to 0.147 cent-dollar per kilowatt-hour. However, in the summer, the annual cash flow was decreased to 3.6 M$, and the production cost of the electricity was increased to 0.148 cent-dollar per kilowatt-hour. Authors' work still continues to investigate the optimal value of the condenser inlet cooling water temperature in order to achieve maximum annual cash flow and/or to achieve minimum production costs per unit of electrical output.

Appendix: The assumptions and the parameters which are selected for the thermodynamic and the thermoeconmic analysis (Alus: 2017 and Alus: 2014).

Parameter	Value
1. Gas Turbine Cycle (Alstom GT24/1994)	
Electrical power at the generator output [MW]	187.7
Exhaust gas mass flow [kg/s]	445
Exhaust gas temperature at the gas turbine outlet [°C]	612
2. Steam Turbine Cycle	
The pinch point temperature difference for HP, IP and LP [°C]	10
The minimum temperature difference between the gas turbine exhaust gases and live/reheat steam [°C]	25
Live steam pressure (HP) [bar]	104
Live steam temperature at the inlet of the HP steam turbine [°C]	545
Pressure of reheat steam (IP steam turbine) [bar]	36
Pressure of the inlet LP steam turbine [bar]	5
Temperature of the superheated steam at 8 [°C]	235
Temperature of the superheated steam at 13 [°C]	320
Feed water temperature at 3 [°C]	60

Condenser inlet cooling water [°C]	20
The overall heat transfer coefficient for the condenser [$Wm^{-2} K^{-1}$]	2500
3. Combined Cycle Power Plant	
Net total electrical power (MW)	284.5
Net plant efficiency (%)	55.93
4. Economic Assumption	
The average life of the combined cycle power plant [years]	20
Operating hours [$h.year^{-1}$]	7500
The unit selling price of generated electricity [$/kWh]	0.114
Price of natural gas [$/kWh]	0.0467
The installed costs of the economizer, evaporator, superheater and reheat sections of the HRSG are 45.7, 34.8 and 96.2$/m², respectively	

References

1. Alus M. *et. al.*, 2017, Thermoeconomic Optimization of Steam Condenser for Combined Cycle Power Plant, International Journal of Science and Engineering Applications, 6(3): 70-75.
2. Haldkar V. *et. al.*, 2013, Parametric Analysis of Surface Condenser for thermal Power Plant. International Journal of Thermal Technologies, 3(4): 155-159.
3. Lakovic' M. S., Mitrovic' D. D., 2010, Impact of the Cold End Operating Conditions on Energy Efficiency of the Steam Power Plants, Thermal Science, 14(1): S53-S66.
4. Wagner W., Kruse A., Properties of Water and Steam, IAPWS-IF97, *Springer*, Berlin, 1998.
5. Behr H.D., Diederichsen C., 1988, Equations for Calculation of Enthalpy and Entropy of the Components of Air and Combustion Gases (in German), *BWK*, 40(1-2): 30–33.
6. Alus M., New Method for Thermoeconomic Optimization of Parameters of Triple-Pressure Combined Cycle Power Plants, Ph.D. thesis, University of Belgrade, Serbia, 2014.
7. Qingsen Z. *et. al*., 2013, The Condenser Performance Test and Thermal Performance Analysis of Variable Conditions in TQNPC, International Journal of Energy and Power Engineering, 5: 566-569.
8. Li W. and Priddy P., Power Plant System Design, John Wiley & Sons, Canada, 1985.
9. Kapooria R. K. *et. al*., 2008,Technological Investigation and Efficiency Analysis of Steam Heat Exchange Condenser: Conceptual Design of a Hybird Steam Condenser, Journal of Energy in Southern Africa, 19(3): 35-45.
10. Roosen P. *et al*, 2003, Pareto optimization of a combined cycle power system as a decision support tool for trading off investment vs. operating costs, International Journal of Thermal Sciences, 42: 553-560.
11. Behbahani-nia A. *et al*, 2010, Thermoeconomic optimization of the pinch point and gas-side velocity in heat recovery steam generators, Journal of Power and Energy, 224: 761-771.
12. Silveira J., Tuna C., 2003, Thermoeconomic analysis method for optimization of combined heat and power systems, Part I, Progress in Energy and combustion Science, 29: 479-485.
13. Attala L. *et. al*., 2001, Thermoeconomic optimization method as design tool in gas–steam combined plant realization, Energy Conversion and Management, 42: 2163-2172.
14. Valdes M. *et. al.*, 2003, Thermoeconomic optimization of combined cycle gas turbine power plants using genetic algorithms. Applied Thermal Engineering, 23: 2169-2182.
15. Manuel V. *et. al.*, 2004, Influence of the heat recovery steam generator design parameters on the thermoeconmic performance of combined cycle gas turbine power plants, International Journal Energy Research, 28: 1243-1254.