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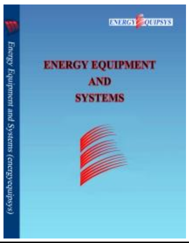

4E analysis and multi-objective optimization of gas turbine CCHP plant with variable ambient temperature

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ABSTRACT

In this paper a gas turbine power plant including air preheater (recuperator), heat recovery steam generator and air cooler system was modeled. Eight parameters were selected as the design variables. Fast and elitist non-dominated sorting genetic algorithm (NSGA-II) was applied (to maximize the exergy efficiency and to minimize the total cost rate) for the mentioned cogeneration system. The total cost rate is included the investment cost, operational cost and environmental impact penalty cost. The presented work included Energy, Exergy, Economy and Environmental (4E) analysis in which all system design parameters were optimally estimated. The optimization problem was developed for variable ambient temperature (VAT) during a year and their results were compared with constant ambient temperature (CAT) during a year. The results for a simple gas turbine showed that at the optimum point, the exergy efficiency reduced about 5.6 percent and total cost rate increased about 4.4 percent when the results for VAT was compared with CAT situation. When the system included a gas turbine with preheater, the total cost decreased and exergy efficiency increased for 39% and 30% respectively (in comparison with a simple gas turbine system). The above percentages were 39.5% and 29.8% respectively for variable ambient temperature. Furthermore when the system included a gas turbine with both preheater and inlet cooling, the total cost decreased and exergy efficiency increased 41% and 34% respectively (in comparison with a simple gas turbine system)..

Keywords: Gas Turbine, CFD, Optimization, Genetic Algorithm, Thermodynamic Analysis.

1. Introduction

Combined heat and power (CHP) system may be defined as a system which generates electricity and heat simultaneously using a single source of fuel. CHP systems play a significant role in efficient usage of energy in industrial and domestic applications. Furthermore, these systems have less harmful effects on the environment. Properly designed CHP systems may provide a thermal efficiency more than 80% [1]. Some works were performed by considering the exergy and economical aspects for a combined heat and power system [2-10]. Fumo et al. compared the various CCHP systems in emission point of view [11]. Jiang-Jiang et al. presented a mathematical modeling to investigate the seasonal atmospheric conditions on CCHP system which affect the thermal and electrical demand [12]. They employed three relative criteria including primary energy saving CO_2 emission

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reduction and annual total cost saving. Roque Diaz et al used a model based on the general theory of exergy cost and structural coefficients for a CCHP system operate with two heat engines [13]. Martinez-Lera and Ballester proposes simplified approaches to estimate the main parameters characterizing the thermal performance of the plant as well as to identify optimal designs for a given application under certain encouragement policies [14]. Mago and Hueffed considered the operational cost and carbon dioxide emission in CCHP system with driven with turbine under different operating strategies [15]. Sanaye et al. [16, 18] selected the number of prime movers in combined heat and power systems by considering the energy and economic analysis. Energy and environmental analysis and optimization of a micro-cogeneration system were carried out by Dorer and Weber [19]. It should be added that, one of the most interesting works was the CGAM problem developed originally by Frangopoulos et al [20-22].

In this paper after thermal modeling of a CCHP plant, the system was optimized by maximizing the exergy efficiency as well as by minimizing the total cost rate (sum of investment, environmental and operation costs) simultaneously.

Genetic algorithm optimization technique was applied to provide a set of Pareto multiple optimum solutions for the above multi-objective problem.

As a summary, the followings are the contribution of this paper into the subject:

- 4E (Energy-Exergy-Economic-Environmental) modeling and analysis of a CCHP system with compressor inlet air cooler and air preheater.
- Performing multi-objective optimization of the above CCHP plant with exergy efficiency and total cost rate as two objectives using NSGA-II.
- Considering the variable ambient temperature (VAT) during a year and comparing their results with constant ambient temperature (CAT) during a year.
- Proposing and comparison of a closed form equation for the total cost rate in terms of exergy efficiency at the optimal design point for CAT and VAT case studies.
- Analysis of the problem and comparison of results for four cases: a system with / without air preheater and a system with / without compressor inlet air cooling system in both CAT and VAT.

Nomenclature

A_{tot}	Total heat transfer area (m^2)
C_{inv}	Annual cost of investment ($/year)
C	Cost per unit of exergy ($/Mj^{-1})
c_p	Specific heat (kJ kg^{-1}K^{-1})
c_f	Cost of fuel per unit of energy ($/Mj^{-1})
COP	coefficient of performance $(-)$
E	Exergy (kJ)
e	Specific exergy (kJ kg^{-1})
GE	Excess free Gibbs energy (kJ)
h	Specific enthalpy (kJ/kg)
$\dot{E}_D$	Exergy Destruction (kJ)
G	Mass flux (kg/m^2s)
LHV	Lower Heating Value (kJ/kg)
$\dot{m}$	Average mass Flow rate in a year (Kg/s)
P	Pressure (bar)
$\dot{Q}$	Heat Transfer rate (kW)
R	Gas constant (kJ/kg.K)
S	Specific entropy (kJ kg^{-1}K^{-1})
T	Temperature(^{o}C)
TPZ	Adiabatic temperature in the primary zone of combustion chamber (K)
$\dot{W}$	Power (kW)
$\dot{Z}$	Capital cost rate ($/s)
Z_k	Purchase cost of the component ($)

Greek abbreviation

υ	Specific volume(m^3/kg)
ΔP	Pressure drop (Pa)
η	Efficiency
γ	Specific heat ratio
φ	Maintenance factor
ξ	Coefficient of Fuel Chemical exergy
ω	Humidity ratio

Subscrips

ave	Average
a	Air
amb	Ambient
AP	Air preheater
AC	Air Compressor
CC	Combustion Chamber
Ch	Chemical
chil	chiller

D	Destruction
e	Exit Condition
ex	Exergy
env	Environment
GT	Gas Turbine
f	Fuel
g	Combustion gasses/ saturated vapor
hr	Hour
HE	heat exchanger
i	Interest rate
in	Inlet Condition
k	Component
L	Loss
nom	nominal capacity
out	Outlet
Ph	Physical
r_C	Compressor pressure ratio
tot	Total
T	Turbine
o	Reference ambient condition

2. Energy Analysis

The energy balance equations for various parts of the cogeneration system as shown in Fig. 1 are as follows:

2.1. Absorption chiller and air cooler heat exchanger

$$\dot{Q}_c = COP.\dot{Q}_h = COP.\dot{m}_{chil}.(h_{11} - h_{12}) \qquad (1)$$

$$T_1 = T_\circ + \frac{\omega h_g}{c_{p,a}} - \frac{\eta_{HE}.\dot{Q}_c}{\dot{m}_a.c_{p,a}} \qquad (2)$$

$$\frac{P_1}{P_\circ} = (1 - \Delta P_{HE}) \qquad (3)$$

where ω is humidity ratio

2.2. Air compressor

$$T_2 = T_1\left\{1 + \frac{1}{\eta_{AC}}\left[r_C^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right]\right\} \qquad (4)$$

$$\dot{W}_{AC} = \dot{m}_a\left(c_{p,a}(T_2 - T_1) + \omega(h_{g2} - h_{g1})\right) \qquad (5)$$

where h_g is the enthalpy of saturated vapor.

2.3. Air Preheater

$$\eta_{AP} = (T_3 - T_2)/(T_5 - T_2) \qquad (6)$$

$$\frac{P_3}{P_2} = (1 - \Delta P_{AP}) \qquad (7)$$

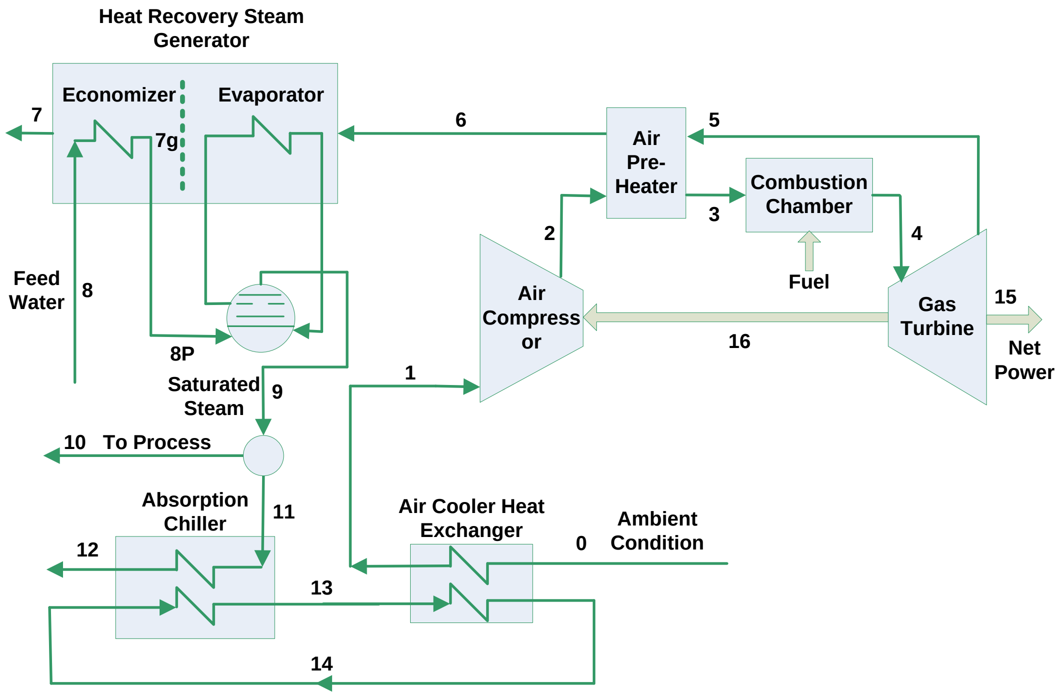

Fig. 1. Schematic diagram of a gas turbine with air preheater, heat recovery steam generator and inlet air cooling system

2.4. Combustion Chamber (CC)

$$\dot{m}_a h_3 + \dot{m}_f \, LHV \eta_{cc} = \dot{m}_g h_4 \tag{8}$$

$$\dot{m}_g = \dot{m}_f + \dot{m}_a \tag{9}$$

$$\frac{P_4}{P_3} = (1 - \Delta P_{cc}) \tag{10}$$

2.5. Turbine

Gas Turbine (GT) plays an important role in generating power for standalone cases, and interconnections [23]. The relations are as follows:

$$T_6 = T_5 \left\{ 1 - \eta_{GT} \left[1 - \left(\frac{p_4}{p_5} \right)^{\frac{1-\gamma_g}{\gamma_g}} \right] \right\} \tag{11}$$

$$\dot{W}_{GT} = \dot{m}_g . c_{p,g} (T_5 - T_6) \tag{12}$$

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} \tag{13}$$

2.6. Heat recovery steam generator

$$\dot{m}_s (h_9 - h_8) = \dot{m}_g (h_6 - h_7) \tag{14}$$

$$\dot{m}_s (h_9 - h_{8P}) = \dot{m}_g (h_6 - h_{7g}) \tag{15}$$

$$P_0 / P_6 = 1 - \Delta P_{HRSG} \tag{16}$$

These combinations of energy and mass balance equations were numerically solved and the temperature as well as enthalpy/exergy of each line of the plant was predicted based on the following basic assumptions:

- All the processes in our study were considered based on the steady state model.
- The principle of ideal- gas mixture was applied for the air and combustion products.
- The fuel injected to the combustion chamber was assumed to be natural gas.
- Heat loss from the combustion chamber was considered to be 3% of the fuel released energy. Moreover, all other components were considered adiabatic.

The thermophysical properties of air and water such as specific heat were considered as temperature and/or pressure dependent.

3. Exergy analysis

Exergy can be divided into four distinct components. Physical and chemical exergy are computed here while two other components (kinetic and potential exergy) are assumed to be negligible as the elevation and speed changes are negligible [24-26]. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state [25]. 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 was obtained for each system component:

$$\dot{E}_Q + \sum_i \dot{m}_i e_i = \sum_e \dot{m}_e e_e + \dot{E}_W + \dot{E}_D \tag{17}$$

It should be noted that in Eq (17) subscripts e and i were related to the specific inlet and outlet flow exergy respectively for each system component and E_D, is the exergy destruction in that system component. Other terms in this equation are as follows [25]:

$$\dot{E}_Q = \left(1 - \frac{T_\circ}{T_i} \right) \dot{Q}_i \tag{18}$$

$$\dot{E}_W = \dot{W} \tag{19}$$

$\dot{E}_Q$ and $\dot{E}_W$ are the corresponding exergy of heat transfer and work which cross the boundaries of the control volume for each system component, T is the absolute temperature (K) and ($\circ$) refer to the ambient conditions respectively. In Eq.(17), term $\dot{E}$ is defined as follows:

$$\dot{E} = \dot{E}_{ph} + \dot{E}_{ch} \tag{20}$$

where

$$\dot{E} = \dot{m} e \, .$$

The physical exergy was computed from:

$$e_{ph} = (h - h_\circ) - T_\circ (S - S_\circ) \tag{21}$$

The gas mixture chemical exergy is defined as follows [25]:

$$ex_{mix}^{ch} = \left[\sum_{i=1}^{n} X_i ex^{ch_i} + RT_0 \sum_{i=1}^{n} X_i Ln X_i + G^E \right] \tag{22}$$

The last term in Eq. (22), G^E , is the excess free Gibbs energy which is negligible at low pressures for a gas mixture.

For approximate evaluation of the fuel

chemical exergy per unit mass the following ratio was applied [24]:

$$\xi = ex_f / LHV_f \tag{23}$$

The ratio of fuel chemical exergy to the fuel Lower Heating Value (LHV) for many gaseous fuels is usually close to 1, for example $\xi_{CH_4} = 1.06, \xi_{H_2} = 0.985$ [27].
For gaseous fuels with C_xH_y, the following experimental equation was used to estimate ξ [28]:

$$\xi = 1.033 + 0.0169\frac{y}{x} - \frac{0.0698}{x} \tag{24}$$

4. Genetic algorithm for multi-objective optimization

4.1. Multi-objective optimization

A multi-objective problem consists of optimizing (i.e. minimizing or maximizing) several objectives simultaneously, with a number of inequality or equality constraints. GAs are semi-stochastic methods, based on an analogy with Darwin's laws of natural selection [29]. The first multi-objective GA, called vector evaluated GA (or VEGA), was proposed by Schaffer [30]. an algorithm based on non-dominated sorting was proposed by Srinivas and Deb [31] and called non-dominated sorting genetic-algorithm (NSGA). This was later modified by Deb et al. [32] which eliminated higher computational complexity, lack of elitism and the need for specifying the sharing parameter. This algorithm is called NSGA-II which is coupled with the objective functions developed in this study for optimization.

4.2. Tournament selection

Each individual competes in exactly two tournaments with randomly selected individuals, a procedure which imitates survival of the fittest in nature.

4.3. Controlled elitism sort

To preserve diversity, the influence of elitism is controlled by choosing the number of individuals from each subpopulation, according to the geometric distribution [33],

$$S_q = S\frac{1-c}{1-c^w}c^{q-1}, \tag{25}$$

to form a parent search population, P_{t+1}(t denote the generation), of size S, where $0 < c < 1$. And w is the total number of ranked non-dominated.

4.4. Crowding distance

The crowding distance metric proposed by Deb [34] is utilized, where the crowding distance of an individual is the perimeter of the rectangle with its nearest neighbors at diagonally opposite corners. So, if individual $X^{(a)}$ and individual $X^{(b)}$ have same rank, each one has a larger crowding distance is better.

4.5. Crossover and mutation

Uniform crossover and random uniform mutation are employed to obtain the offspring population, Q_{t+1}. The integer-based uniform crossover operator takes two distinct parent individuals and interchanges each corresponding binary bits with a probability, $0 < p_c \le 1$. Following crossover, the mutation operator changes each of the binary bits with a mutation probability, $0 < p_m < 0.5$.

5. Objective functions, design parameters and constraints

In this study, exergy efficiency and total cost rate were considered as two objective functions. The total exergy efficiency of the system was estimated as:

$$\eta_{ex} = \frac{\dot{W}_{net} + \dot{E}_{10} - \dot{E}_8}{\dot{m}_f \times LHV \times \xi} \tag{26}$$

In which W_{net}, $\dot{m}_f$ and ξ are the gas turbine net output power, average mass flow rate of fuel injected into the combustion chamber in a year and chemical exergy ratio (Eq. 24).
The total cost rate includes investment cost rate ($\dot{Z}_{inv}$), operating (fuel) cost rate ($\dot{Z}_{fuel}$) and environmental related cost rate ($\dot{Z}_{env}$) as follows:

$$\dot{Z}_{total} = \dot{Z}_{inv} + \dot{Z}_{fuel} + \dot{Z}_{env} \tag{27}$$

where $\dot{Z}_{inv}$, $\dot{Z}_{fuel}$ and $\dot{Z}_{env}$ are:

$$\dot{Z}_{inv} = Z_k.a.\varphi/(N) \tag{28}$$

$$\dot{Z}_{fuel} = c_f \dot{m}_f LHV \tag{29}$$

$$\dot{Z}_{env} = C_{CO}\dot{m}_{CO} + C_{NO_x}\dot{m}_{NO_x} \quad (30)$$

In which Z_k is the purchase cost of k_{th} component in U.S dollar. The expression for investment cost rate of each component of the CCHP plant is listed in Appendix A [25, 35-37]. Also a is annual cost coefficient defined as:

$$a = \frac{i}{1-(1+i)^{-n}} \quad (31)$$

in which *i*, is the interest rate and *n* is the system life time in years.

In Eq. (28), *N* is the annual number of the operating hours of the unit, φ is the maintenance factor and c_f is the cost of fuel per unit of energy.

The amount of *CO* and *NOx* produced in the combustion chamber change mainly by the maximum flame temperature, to determine the pollutant emission in grams per kilogram of fuel, the following equations are proposed [37]:

$$\dot{m}_{NO_x} = \frac{0.15\times10^{16}\tau^{0.5}\exp(-71100/T_{pz})}{P_3^{0.05}\sqrt{(\Delta P_3/P_3)}} \quad (32)$$

$$\dot{m}_{CO} = \frac{0.179\times10^{9}\exp(7800/T_{pz})}{P_3^{2}\tau\sqrt{(\Delta P_3/P_3)}} \quad (33)$$

where τ is the residence time in the combustion zone (τ is assumed to be constant equal to 0.002 s); T_{pz} is the primary zone combustion temperature; P_3 is the combustor inlet pressure; $\Delta P_3/P_3$ is the non-dimensional pressure drop in the combustion chamber.

In this study air compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, gas turbine inlet temperature, nominal capacity of absorption chiller, steam mass flow rate passing through the generator of the chiller, recuperator effectiveness as well as inlet air cooling heat exchanger effectiveness were considered as eight design parameters.

Furthermore, the following usual inequality constraints are also satisfied:

$$T_{7p} > T_9 + \Delta T_{pinch,\min} \quad (34)$$

$$T_1 > 283.15K \quad (35)$$

$$T_7 > 400K \quad (36)$$

The constraint in Eq.(35) is introduced for anti icing in the inlet of air compressor. In addition the last constraint was applied for keeping the stack temperature above the dew point temperature and avoiding formation of sulfuric acid.

6. Case study

The cogeneration system optimum design parameters were obtained for a Sarcheshmeh cupper production plant located in south of Kerman city. the Sarcheshmeh has the average relative humidity 10% and variation of ambient temperature during a year for this case was depicted in Fig. 2. the average annual ambient temperature is about $19^{\circ}C$ for this case. The cupper plant required an integrated combined heat and power plant, which could provide $100MW$ of electric power and $50\,ton/hour$ saturated steam at 13 bar. System was optimized for depreciation time n=15 years and interest rate i=0.1.

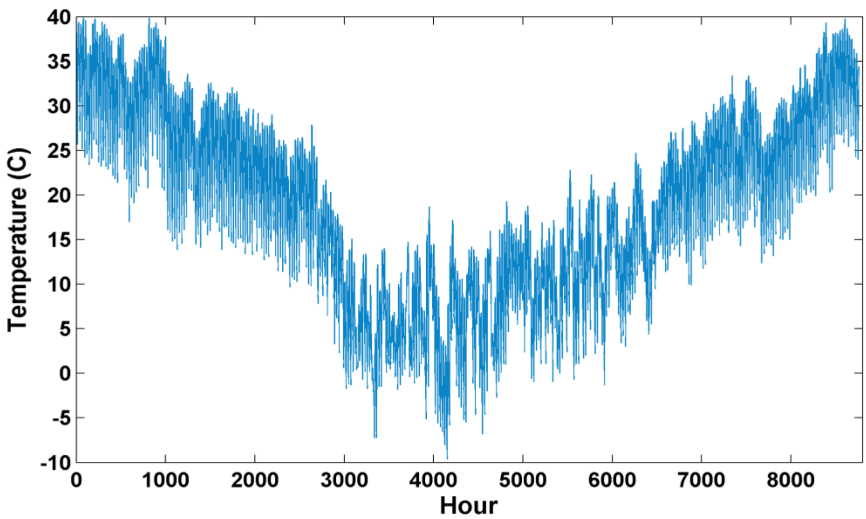

Fig. 2. Variation of ambient temperature during a year for the studied case

Furthermore the maintenance factor *φ=1.06* is considered in this case. C_{co} and C_{NO_x} are equal to 0.02086 $/kg and 6.853 $/kg respectively [38]. Four various cogeneration configurations (listed in Table. 1) with/without recuperator and inlet air cooling system were separately optimized. In addition the optimization problem was performed for two different scenarios i.e., with constant ambient temperature and variable ambient temperature during a year.

7. Discussion and results

7.1. Verification of modeling results

To verify the modeling results, the simulation output results were compared with the corresponding reported results given in literature. In order to ensure the validity of thermodynamic and economic modeling, as well as the optimization procedure, first a CHP unit with the same characteristics of the classic well-known CGAM problem [20] was modeled and optimized using genetic algorithm (Table. 2). As is shown in this table, our model was successful in reproducing the optimum values of CGAM problem with less then 0.1% difference that is acceptable for engineering problem.

7.2. Optimization results

Design parameters (decision variables) and the range of their variations are listed in Table 3. The genetic algorithm optimization was performed for 400 generations, using a search population size of M = 100 individuals, crossover probability of p_c = 0.85, gene mutation probability of p_m = 0.035 and controlled elitism value c = 0.55 in each case.

The results clearly reveal the conflict between two objectives, the exergy efficiency and the total cost rate. Any change in design parameters listed in Table 3 that increases the efficiency, leads to an increase in the total cost rate and vice versa. This shows the need for multi-objective optimization techniques in optimal design of this equipment.

7.2.1. Variable ambient temperature (VAT) during a year

The optimization procedure was done for variable ambient temperature (Fig. 2) and results are depicted in Fig. 3. As an example the results for Pareto-optimal curve for the system with both air preheater and inlet air cooling (case 1) with variable ambient temperature during a year (Fig. 2) are shown in Fig. 4. It is shown in Fig. 4 that the maximum exergy efficiency exists at design point A (0.4615), while the total cost is the biggest at this point. On the other hand the minimum total cost occurs at design point E (3016*$/hour*), with a smallest exergy efficiency value (0.4380) at that point. Actually the design point A shows a plant specification with the highest exergy efficiency but the design point E shows a plant design specifications with the lowest total cost rate. Design points in the range of C-D show a plant design specifications with the moderate values of both objective functions.

The selection of a single optimum point from existing points on the Pareto front needs a process of decision-making. In fact, this process is mostly carried out based on engineering experiences and importance of

Table.1. Four various configurations studied in this paper

Case 1	A CCHP system with both air preheater and inlet air cooling
Case 2	A CCHP system with air preheater and without inlet air cooling
Case 3	A CCHP system without air preheater and with inlet air cooling
Case 4	A CCHP system without air preheater and without inlet air cooling

Table.2. The comparison of modeling output and the corresponding results reported in reference [20]

Variables	Optimum design values reported by[20]	Optimum design values in the present paper
r_C	8.5970	8.5160
η_{AC}	0.8465	0.8466
η_T	0.8787	0.8786
$T_3(K)$	913.14	913.31
$T_4(K)$	1491.97	1492.50

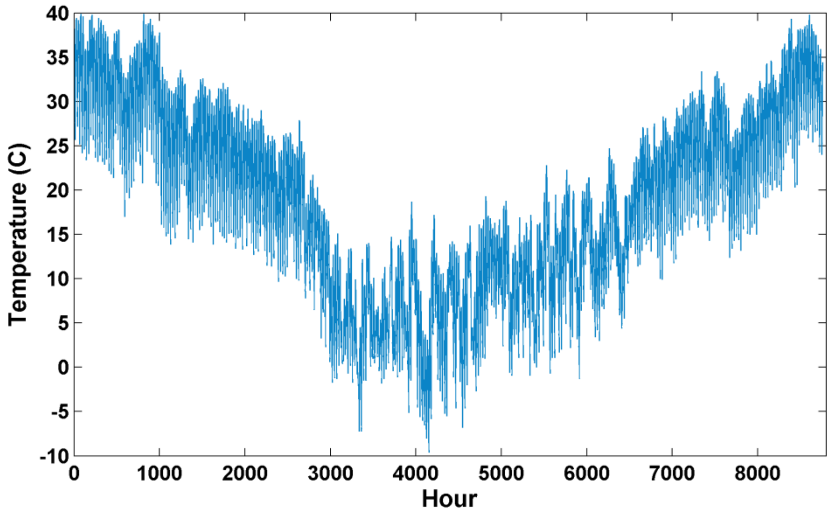

Fig. 2. Variation of ambient temperature during a year for the studied case

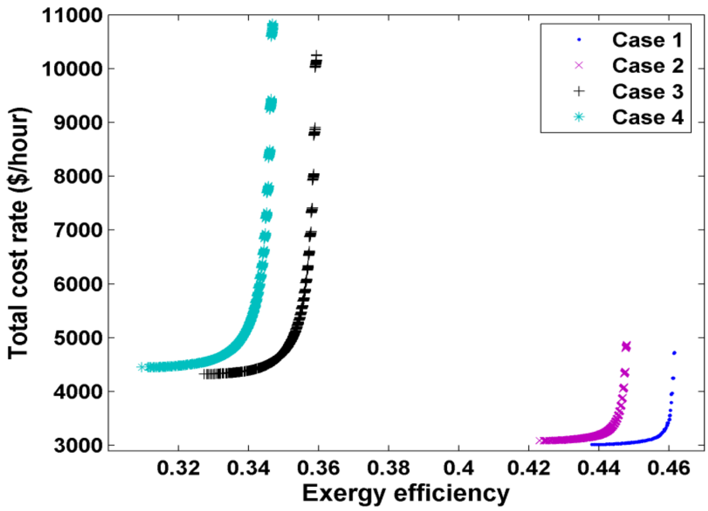

Fig. 3. Pareto optimal fronts for plants at VAT situation for case studies described in Table 1

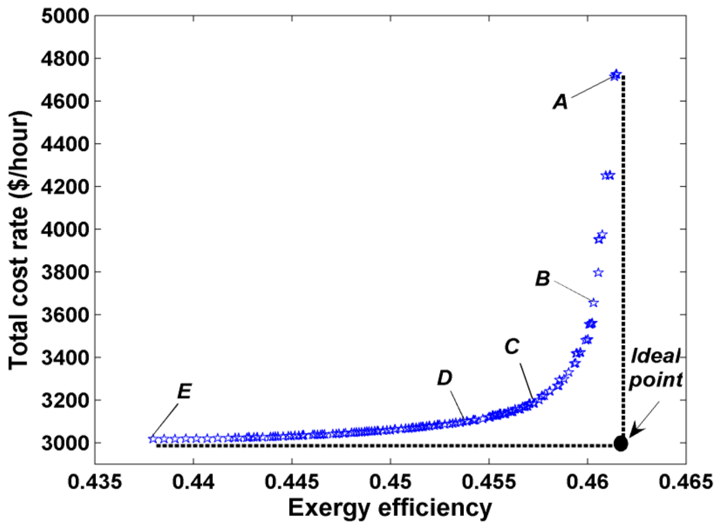

Fig. 4. Pareto optimal front and equilibrium point for the plant with both air preheater and inlet air cooling system (case 1) for VAT situation

Table.3. The design parameters and their range of variation for the optimization procedure

Design variables	From	To
Air compressor isentropic efficiency	0.7	0.9
Turbine isentropic efficiency	0.7	0.9
Air compressor pressure ratio	4	16
Turbine inlet temperature (K)	1000	1550
Chiller cooling capacity (MW)	0.5	7
Chiller steam mass flow rate (tons/hour)	0	10
Inlet air cooling heat exchanger effectiveness	0.5	0.9
Recuperator effectiveness	0.5	0.9

each objective for decision makers. The process of final decision-making in Fig. 4 is usually performed with the aid of a hypothetical point named as equilibrium point, that both objectives have their optimal values independent of the other objectives [39]. It is clear that it is impossible to have both objectives at their optimum point, simultaneously. The equilibrium point is not a solution located on the Pareto frontier. In this paper, LINMAP method was used to find the final optimum solution in Pareto front [39].

In the LINMAP method, each objective is nondimensionalized using the following relation:

$$F_{ij}^{n} = \frac{F_{ij}}{\sqrt[2]{\sum_{i=1}^{m}(F_{ij})^{2}}} \tag{37}$$

where i is the index for each point on the Pareto frontier, j is the index for each objective in the objectives space and m denotes the number of points in the Pareto front. After nondimensionalization of all objectives, the distance of each solution on the Pareto frontier from the ideal point obtained. The closest point of Pareto frontier to the equilibrium point (design point C) might be considered as a desirable final solution with the 45.61 percent efficiency and 3210 \$/hour as total annual cost rate. The optimum values of design parameters and objective functions for the equilibrium point at the Pareto optimal front are listed in Table 4 for four case studies introduced in Table 1. The results show that the highest value of the exergy efficiency and the lowest value of the total cost rate were obtained for case (1). The exergy efficiency and the total cost rate were improved about 34 and 41 percent respectively with changing the system from case (4) to the case (1).

7.2.2. Constant ambient temperature (CAT) during a year

With considering constant average ambient temperature (equal to $19^{\circ}C$) during a year, the Pareto optimal fronts are obtained and shown in Fig. 5 for each case studies described in Table. 1. It was observed that the Pareto front in CAT situation shifted to the right and down in comparison with the VAT situation which means that optimization for CAT situation provided bigger exergy efficiency and lower total cost. Furthermore the optimum values of design parameters and objective functions for the above mentioned selected specific point (equilibrium point) at the Pareto optimal front are listed in Table 5 for four case studies introduced in Table 1. The results show that the highest value of the exergy efficiency and the lowest value of the total cost rate were obtained for case (1). The exergy efficiency and the total cost rate were improved about 34 and 39 percent respectively with changing the system from case (4) to the case (1).

7.2.3. Comparison of results for CAT and VAT situations

- Closed form relations between exergy efficiency and total cost

To provide a useful tool for the optimal design of the CCHP plant, the following equations for exergy efficiency versus the total cost rate (in case 1) were derived for the Pareto curves of VAT and CAT situations in Figs. 3 and 5.

$$\dot{Z}_{tot}(\$/hour) = \frac{2990\eta_{ex}^{2} - 2762\eta_{ex} + 638}{\eta_{ex}^{2} - 0.9236\eta_{ex} + 0.2133} \tag{38}$$

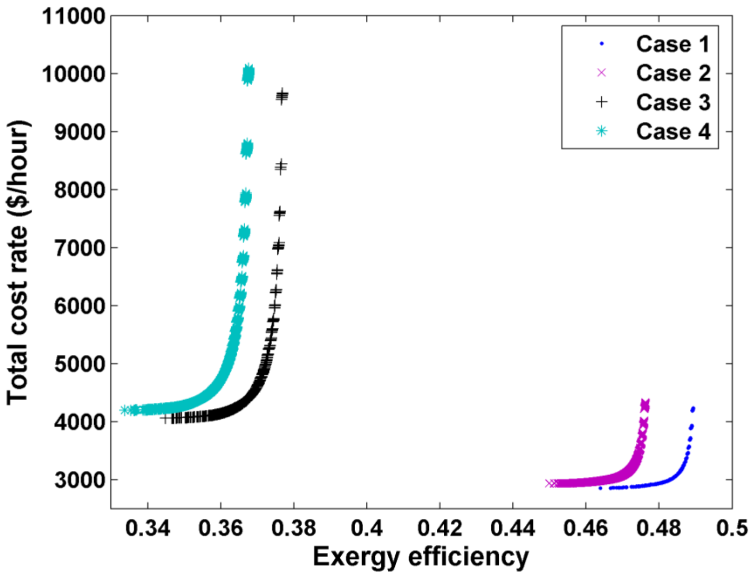

Fig. 5. Pareto optimal fronts for plants at CAT situation with constant average ambient temperature equal 19oC during a year for case studies described in Table 1

Table.5. The optimum values of design parameters and objective functions for four case studies introduced in Table1 at selected equilibrium points for CAT situation

Case studies defined in Table 1	Case 1	Case 2	Case 3	Case 4
Exergy efficiency	0.4852	0.47169	0.3719	0.3624
Total cost rate ($/hour)	3104	3149.6	4899.3	5073.1
Investment cost ($)	4.26×10^7	3.99×10^7	1.16×10^8	1.15×10^8
Air compressor isentropic efficiency	0.88824	0.88745	0.88588	0.88588
Turbine isentropic efficiency	0.9	0.9	0.9	0.9
Air compressor pressure ratio	6.0625	5.8489	15.998	15.999
Turbine inlet temperature (K)	1413.4	1397.2	1225	1226.9
Chiller cooling capacity (MW)	5.5613	–	5.3357	–
Chiller steam mass flow rate (tons/hour)	8.1865	–	7.9314	–
Inlet air cooling heat exchanger effectiveness	0.80372	–	0.89922	–
Recuperator effectiveness	0.9	0.9	-	–
Fuel mass flow rate (kg/s)	4.3692	4.4944	5.7002	5.8501
Cycle thermal efficiency	0.4578	0.4450	0.3509	0.3419

$$\dot{Z}_{tot}(\$/hour)=\frac{-5159\eta_{ex}^2-30410\eta_{ex}+16180}{\eta_{ex}^3-7.904\eta_{ex}^2-5.881\eta_{ex}+4.667} \quad (39)$$

Equations (38) and (39) are valid in the range of exergy efficiency $0.438<\eta_{ex}<0.461$ and $0.464<\eta_{ex}<0.489$ respectively. The interesting point in the above equations is that with considering a numerical value for the exergy efficiency in the mentioned range, the minimum total cost for that optimal point along with other optimal design parameters would be obtained.

Based on the explained procedure in 7.2.1, for selecting an optimum design point (equilibrium point) on Pareto front, the selected equilibrium points in Figs. 3 and 5 are listed in Tables. 4 and 5 for VAT and CAT scenarios. The results showed that at the optimum point, the exergy efficiency reduced about 5.6 percent and the total cost rate increased about 4.4 percent when the VAT system was selected. Subsequently 9.64% decrease in compressor pressure ratio, 0.2% increase in TIT, 17.6% increase in chiller capacity and 12% increase in inlet air cooler effectiveness were observed for the VAT situation in comparison to the CAT one. Moreover the value of investment cost for the VAT situation is generally lower than the CAT situation.

- Air preheater or inlet air cooling system

It is note worthy that the effect of preheater existing in the system, increases the exergy efficiency and decreases the total cost rate, more than that for the inlet air cooling system (absorption chiller and heat exchanger) in both CAT and VAT situations. It was also observed that the optimum compressor pressure ratio was smaller and the turbine inlet temperature was bigger in case with air preheater in comparison with when no air preheater existed.

- The effects of ambient relative humidity and temperature

The system optimization results and Pareto fronts at optimum design points for plant with both air preheater and inlet air cooling (case 1) and various values of ambient relative humidity and temperature are shown in Figs. 6 and 7 respectively. The results show that both exergy efficiency and total cost rate decrease about 3.5 percent by increase of each 30 percent in relatively humidity. It was observed from Fig. 7 that at the same total annual cost, the exergy efficiency decreases with increase of ambient temperature while at the higher ambient temperatures, a lower values for exergy efficiency were obtained. Actually all the optimum results obtained in higher ambient temperature is dominated by the optimum results obtained in lower ambient temperature.

7.3. Determination of an equivalent average ambient temperature (EAAT)

It is very useful to find a constant equivalent ambient temperature (CAT situation) during a year that provides a Pareto front similar to the variable ambient temperature (VAT situation). For this purpose the root mean square difference parameter which specifies the closest of Pareto fronts in CAT (which is named EAAT) and VAT situations was defined as follows:

$$RMSE=\left[\sqrt{\frac{1}{N\times\eta_{ex,\max}}\sum_{i=1}^{N}(\eta_{ex,VAT}-\eta_{ex,CAT,EAAT})^2}\right]+\left[\sqrt{\frac{1}{N\times\dot{Z}_{total,\max}}\sum_{i=1}^{N}(\dot{Z}_{total,VAT}-\dot{Z}_{total,CAT,EAAT})^2}\right] \quad (40)$$

where N is the number of points on Pareto front with $\eta_{ex.VAT}$ and $\dot{Z}_{total/VAT}$ for VAT situation and the corresponding points of $\eta_{ex.CAT}$ and $\dot{Z}_{total/CAT}$ for CAT situation. An optimization procedure was performed in CAT situation to minimize RMSE in which the ambient temperature was a single decision variable. The equivalent average ambient temperature (EAAT) was obtained equal $28.63^{\circ}C$ when RMSE was minimized (0.0312). This temperature was about $10^{\circ}C$ higher than the constant average yearly ambient temperature during a year.

8. Conclusion

A CCHP plant was optimally designed using multi objective optimization technique with air preheater and inlet air cooling system. The design parameters (decision variables) were air compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, gas turbine inlet temperature, nominal capacity of absorption chiller, steam mass flow rate passing through the generator of the chiller, recuperator effectiveness as well as inlet air cooling heat exchanger effectiveness. In this optimization problem, the exergy efficiency and total cost rate were considered as two objective functions. The optimization problem was developed for four plants, a gas turbine with/without air preheater and inlet air cooling system. In addition the two constant (CAT) and variable (VAT) ambient temperature situations were investigated. The results showed that at the optimum design point, the exergy efficiency reduced about 5.6 percent and the total cost rate increased about 4.4 percent when the VAT system was selected. Subsequently 9.64% decrease in compressor pressure ratio, 0.2% increase in TIT, 17.6% increase in chiller capacity and 12% increase in inlet air cooler effectiveness were observed for the VAT situation in comparison to the CAT one. The results provided the specifications of a gas turbine system with just an air preheater with about 39 percent decrease in total cost and about 30 percent increase in exergy efficiency in comparison with a simple gas turbine system. The above results for a gas turbine with both air preheater and inlet air cooling system revealed about 41 percent

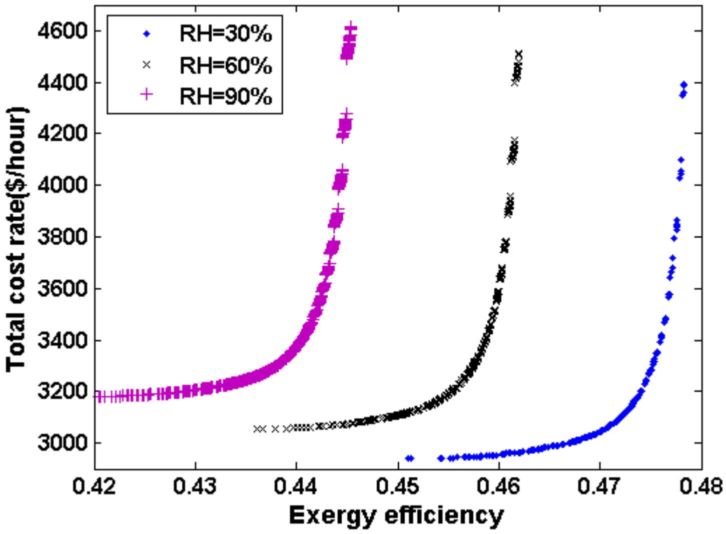

Fig. 6. Pareto optimal front for the plant with both air preheater and inlet air cooling system (case 1) at CAT situation and constant average ambient temperature equal $19^{\circ}C$ and various values of ambient relative humidity

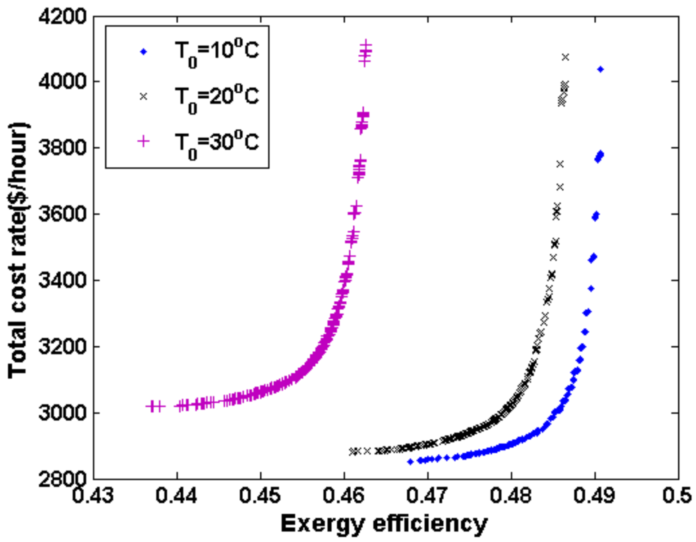

Fig. 7. Pareto optimal front for the plant with both air preheater and inlet air cooling system (case 1) at CAT situation and various values of constant average ambient temperature and relative humidity of 10 % for CAT situation

decrease in total cost and 34 percent increase in exergy efficiency. It was obtained that the optimum air compressor pressure ratio was lower in a cycle with both air preheater and inlet air cooling than that for a simple gas turbine cycle. On the other hand the optimum value of turbine inlet temperature was higher in a cycle with both air preheater and inlet air cooling than that for a simple gas turbine cycle.

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Appendix A:
Purchase equipment cost functions [24, 34- 36]

Components	Investment cost
Air compressor	$Z_{AC}=\left(\frac{a_{11}\dot{m}_a}{a_{12}-\eta_{AC}}\right)\left(\frac{p_2}{p_1}\right)\ln\left(\frac{p_2}{p_1}\right)$
Combustion chamber	$Z_{CC}=\left(\frac{a_{21}\dot{m}_a}{a_{22}-\frac{p_4}{p_3}}\right)\left[1+EXP\left(a_{23}T_{TIT}-a_{24}\right)\right]$
Gas turbine	$Z_{GT}=\left(\frac{a_{31}\dot{m}_g}{a_{32}-\eta_T}\right)\ln\left(\frac{p_4}{p_5}\right)\left(1+EXP\left(a_{33}T_3-a_{34}\right)\right)$
Air preheater	$Z_{AP}=a_{41}\times A^{0.6}$
HRSG	$Z_{HRSG}=a_{51}\left[\left(\frac{\dot{Q}_{ECO}}{(\Delta T_{lm})_{ECO}}\right)^{0.8}+\left(\frac{\dot{Q}_{EV}}{(\Delta T_{lm})_{EV}}\right)^{0.8}+\left(\frac{\dot{Q}_{SH}}{(\Delta T_{lm})_{SH}}\right)^{0.8}\right]+a_{52}\dot{m}_s+a_{53}\dot{m}_g^{1.2}$
Absorption chiller and inlet air cooling	
a) Absorption chiller	$Z_{chil}=\begin{cases}-81.552\ln(\dot{Q}_{nom})+778 & \dot{Q}_{nom}>1000\ (kW)\\ -35.4\ln(\dot{Q}_{nom})+431 & \dot{Q}_{nom}<1000\ (kW)\end{cases}$
b) Inlet air cooler	$Z_{HE}=a_{41}\times A^{0.6}$