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OPTIMUM OPERATING PARAMETERS OF AN IRREVERSIBLE GAS TURBINE CYCLE

Maher M. Abou Al-Sood, Kassem. K. Matrawy,

Yousef M. Abdel-Rahim

Dept. of Mech. Eng., Faculty of Eng., Assiut Univ., Egypt, (71516)
Corresponding author: Email: m_aboualsood@hotmail.com,
Tel. +2-0122-636-8213, +2-088-241-1205, Fax +2-088-233-5572

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ABSTRACT

The present paper discusses the performance of an irreversible regenerative intercooler–reheat gas turbine cycle under wide ranges of design and operating parameters. The study aims to determine the optimum regions of these parameters that satisfy the most requirements of the gas turbine cycle. The operating parameters are the inlet air temperature to the first stage compressor, inlet temperature and pressure of air entering the high pressure turbine and the thermal size (the heat transfer coefficient–area product) of the whole heat exchangers used in the cycle. A mathematical simulation model is developed to calculate the performance parameters of the cycle under different operating conditions. The developed model achieves most of the requirements of the gas turbine cycle (maximum first and second law efficiencies, minimum back work ratio (BWR), maximum ecological coefficient of performance (ECOP), maximum work output, minimum exergy losses, and finally minimum heat added to the cycle). Achieving some or all of these requirements is based on selection of certain small region in the applicable range of operating parameters. The optimum region resulted from the given operating parameters are: 302 K – 315 K for the minimum temperature, 1340 K – 1360 K for the maximum temperature, 1440 kPa – 2830 kPa for the maximum pressure, and finally 20.7 kW/K – 29.6 kW/K for the heat exchangers' thermal size.

Keywords: Inter-cooled -reheat- gas turbine cycle; Regenerator; Turbine; Compressor; Power output; Operating parameters; Performance

Nomenclature

A	surface area, m^2
BWR	back work ratio
$\dot{C}$	heat capacity rate, W/K
C^*	heat capacity ratio $C^* = \min(\dot{C}_{cold}, \dot{C}_{hot}) / \max(\dot{C}_{cold}, \dot{C}_{hot})$, W/K
c_p	specific heat, J/kg.K
ECOP	ecological coefficient of performance
h	enthalpy, J/kg
$\dot{m}$	mass flow rate, kg/s
NTU	number of transfer units
p	pressure, Pa
$\dot{Q}$	heat rate, W
q	heat flux, MJ/kg
R	gas constant, J/kg.K
s	specific entropy, J/kg.K
T	temperature, K
U	overall heat transfer coefficient, $W/m^2.K$
UA	thermal size, W/K
v	specific volume, (m^3/kg)
$\dot{W}$	power, W
w	work, J/kg
$\dot{X}$	exergy rate, W
x	exergy, MJ/g

Greek Symbol

η	efficiency
ε	effectiveness

Subscripts

0	dead state
I	first law
II	second law
add	high temperature heat addition
bur	burner
c	cold fluid, compressor
dest	destroyed
cond	condenser (low temperature heat rejection)
int	intercooler
h	hot fluid
max	maximum
r	reduced
reg	regenerator

reht	reheater
rej	rejected
t	turbine
tot	total
w	water

1. Introduction

Gas turbines have experienced phenomenal growth and progress since their first successful development in the 1930's with their representative simple-cycle efficiencies of about 17 percent. The low efficiency was due to low compressor and turbine efficiencies and low turbine inlet temperatures due to material limitations despite their versatility and ability to burn a variety of fuels. Efforts to improve cycle efficiency concentrated on three areas: (1) increasing turbine inlet temperature; (2) increasing efficiencies of cycle components; and (3) modifying the basic cycle. Recently, developments in material science allow using inlet temperatures up to1500°C (i.e. General Electric uses a turbine inlet temperature of 1425°C). Also, computer aided design and simulations have enabled designers to reach optimum performance of cycle components such as compressor and turbine. Finally, continuous modifications of Brayton cycle to include regeneration [1-3], isothermal heat addition [4-7], intercooled compression [7, 8], reheat expansion [9], and combination of cycle units [10-14] have resulted in practically doubling cycle efficiencies. The cycle back work ratio (BWR = ratio of compressor work to turbine work) has improved as a result of intercooling and reheating, especially when accompanied by regeneration. This is because intercooling decreases the average temperature at which heat is added, and reheating increases the average temperature at which heat is rejected.

Moreover, the Brayton cycle, as a model of gas turbine power plants cycles, undergoes an optimization of entropy generation [15, 16], reversible work [17, 18], power [18-22] and power density [23-25]. Some of the proposed models account just for internal irreversibilities of the compressor and turbine [26, 27], pressure drops in the heater, the cooler, and the regenerator [19, 23, 24], external irreversibilities of coupling to external heat reservoirs or heat exchangers [20].

Most of the above mentioned literature have been carried out to improve the performance of real complex gas power plants at optimum parameters such as turbine outlet temperature, intercooling, reheat, and cycle pressure ratios [12, 28]. However, those optimization results are specific to the gas turbines under consideration. Therefore, the main objective of the present study is to identify, for irreversible regenerative intercooler–reheat gas turbine cycle, the ranges of all operating parameters including compressor and turbine inlet temperatures, and intercooling, reheat, and cycle pressure ratios, that give optimum operating parameters for the first and second laws efficiencies, ecological coefficient of performance (ECOP), and back work ratio (BWR).

2. Theoretical Model

The theoretical model, depicted in Figures 1 and 2 and partially based on [12], has a constant mass flow rate working medium, $\dot{m}$ of air as an ideal gas with variable specific heats. The cycle, as characterized below, has the overall heat transfer coefficient (U), and the heat transfer surface area (A) in each process.

a. The air is compressed from its initial state 1 to state 4 by two non-isentropic compressors with efficiencies, η_{c12} and η_{c34}, and a non-isobaric intercooler (i.e.: counterflow heat exchanger) with effectiveness, ε_{int}. Inlet temperature to the second compressor is slightly higher than that of the first compressor. The describing equations for these processes as in Ref. [29] are:

$$\eta_{c12} = \frac{w_{c12s}}{w_{c12}} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

$$\eta_{c34} = \frac{w_{c34s}}{w_{c34}} = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{2}$$

$$\frac{p_{2s}}{p_1} = \frac{p_{r2}}{p_{r1}}, \quad \frac{v_{2s}}{v_1} = \frac{v_{r2}}{v_{r1}} \tag{3}$$

$$\frac{p_{4s}}{p_3} = \frac{p_{r4}}{p_{r3}}, \quad \frac{v_{4s}}{v_3} = \frac{v_{r4}}{v_{r3}} \tag{4}$$

$$\varepsilon_{\text{int}} = \frac{\dot{Q}_{23}}{\dot{Q}_{\text{int max}}} = \frac{(UA)_{\text{int}} (\Delta T_{LM})_{\text{int}}}{\dot{Q}_{\text{int max}}} = \frac{\dot{Q}_{23}}{\min\left(\dot{C}_W, \dot{C}_{23}\right) \times \left(T_2 - T_{C2}\right)} \tag{5}$$

Quantities, $\dot{Q}_{23}$, $\dot{C}_W$ and $\dot{C}_{23}$ are rates of heat release, heat capacity rate for cooling water and heat capacity rate for air (defined as $\dot{m}$ times average air specific heat over temperature range) respectively. The intercooler logarithmic mean temperature difference $(\Delta T_{LM})_{\text{int}}$ is defined as:

$$(\Delta T_{LM})_{\text{int}} = \frac{\left(T_2 - T_{C3}\right) - \left(T_3 - T_{C2}\right)}{\ln\left(\left(T_2 - T_{C3}\right) / \left(T_2 - T_{C3}\right)\right)} \tag{6}$$

b. After the second compressor, the air is pre-heated from state 4 to state 5 in a regenerative counterflow heat exchanger that will be discussed later in heat rejection process. The exit air from regenerator (state 5) is heated up to a maximum temperature, T_6, by counter flow heat exchanger having rate of heat addition, effectiveness and logarithmic mean temperature difference of $\dot{Q}_{56}$, ε_{add} and $(\Delta T_{LM})_{add}$ defined as:

$$\varepsilon_{add} = \frac{\dot{Q}_{56}}{\dot{Q}_{add\,\text{max}}} = \frac{(UA)_{add} (\Delta T_{LM})_{add}}{\dot{Q}_{add\,\text{max}}} = \frac{\dot{Q}_{56}}{\min\left(\dot{C}_W, \dot{C}_{56}\right) \times \left(T_5 - T_{H5}\right)} \tag{7}$$

$$(\Delta T_{LM})_{add} = \frac{\left(T_5 - T_{H6}\right) - \left(T_6 - T_{H5}\right)}{\ln\left(\left(T_5 - T_{H6}\right) / \left(T_6 - T_{H5}\right)\right)} \tag{8}$$

c. Similarly to the compression process, air is expanded from state 6 to final state 9 by two non-isentropic turbines with efficiencies, η_{t67} and η_{t89}, and one non-isobaric reheater, effectiveness and logarithmic mean temperature difference, $\dot{Q}_{78}$, ε_{reh} and $(\Delta T_{LM})_{reh}$. Inlet temperature to the second turbine is slightly lower than that of the first turbine. The governing equations for these processes are:

$$\eta_{t67} = \frac{w_{t67}}{w_{t67s}} = \frac{h_6 - h_7}{h_6 - h_{7s}} \tag{9}$$

$$\eta_{t89} = \frac{w_{t89}}{w_{t89s}} = \frac{h_8 - h_9}{h_8 - h_{9s}} \tag{10}$$

$$\frac{p_{7s}}{p_6} = \frac{p_{r7}}{p_{r6}}, \ \frac{v_{7s}}{v_6} = \frac{v_{r7}}{v_{r6}} \tag{11}$$

$$\frac{p_{9s}}{p_8} = \frac{p_{r9}}{p_{r8}}, \ \frac{v_{9s}}{v_8} = \frac{v_{r9}}{v_{r8}} \tag{12}$$

$$\varepsilon_{reh} = \frac{\dot{Q}_{78}}{\dot{Q}_{reh\max}} = \frac{(UA)_{reh}(\Delta T_{LM})_{reh}}{\dot{Q}_{reh\max}} = \frac{\dot{Q}_{78}}{\min(\dot{C}_H, \dot{C}_{78}) \times (T_{H7} - T_7)} \tag{13}$$

$$(\Delta T_{LM})_{reh} = \frac{(T_{H7} - T_8) - (T_{H8} - T_7)}{\ln((T_{H7} - T_8)/(T_{H8} - T_7))} \tag{14}$$

d. In the heat rejection process between the exit of the low pressure turbine to the inlet of the first compressor (process 9-1), air is firstly cooled in the regenerator (with $\dot{Q}_{45}$, ε_{reg} and $(\Delta T_{LM})_{reg}$ as the rate of heat added, effectiveness and logarithmic mean temperature difference) and finally cooled to state 1 in a counterflow heat exchanger (i.e. low temperature heat rejection with its respective parameters as $\dot{Q}_{101}$, ε_{rej} and $(\Delta T_{LM})_{rej}$). The governing equations are:

$$\varepsilon_{reg} = \frac{\dot{Q}_{45}}{\dot{Q}_{reg.\max}} = \frac{(UA)_{reg}(\Delta T_{LM})_{reg}}{\dot{Q}_{reg.\max}} = \frac{\dot{Q}_{45}}{\min(\dot{C}_{45}, \dot{C}_{910}) \times (T_9 - T_4)} \tag{15}$$

$$(\Delta T_{LM})_{reg} = \frac{(T_9 - T_5) - (T_{10} - T_4)}{\ln((T_9 - T_5)/(T_{10} - T_4))} \tag{16}$$

$$\varepsilon_{rej} = \frac{\dot{Q}_{101}}{\dot{Q}_{rej_{\max}}} = \frac{(UA)_{rej}(\Delta T_{LM})_{rej}}{\dot{Q}_{bur\max}} = \frac{\dot{Q}_{101}}{\min(\dot{C}_W, \dot{C}_{101}) \times (T_{10} - T_{C10})} \tag{17}$$

$$(\Delta T_{LM})_{rej} = \frac{(T_{10} - T_{C1}) - (T_1 - T_{C10})}{\ln((T_{10} - T_{C1})/(T_1 - T_{C10}))} \tag{18}$$

e. The above-mentioned heat exchangers (i.e. intercooler, regenerator, high temperature heat addition, reheater, and low temperature heat rejection) are of counterflow types and their effectiveness can be alternatively calculated as in [30] as:

$$\varepsilon_i = \frac{1 - \exp[-\mathrm{NTU}(1 - C^*)]}{1 - C^* \exp[-\mathrm{NTU}(1 - C^*)]}, \quad \text{i = int, reg, add, reh, rej,} \tag{19}$$

where: C^* is the specific heat ratio ($C^* = \min(\dot{C}_{cold}, \dot{C}_{hot}) / \max(\dot{C}_{cold}, \dot{C}_{hot})$) and NTU is the number of transfer unit ($\mathrm{NTU} = UA / \min(\dot{C}_{cold}, \dot{C}_{hot})$).

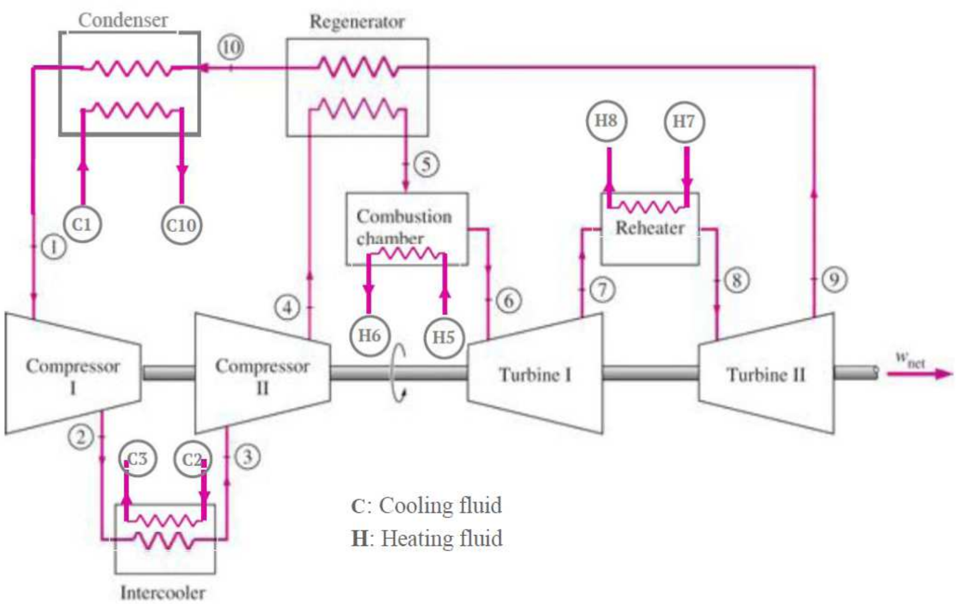

Fig. 1. Schematic diagram of a realistic irreversible-regenerative and reheat Brayton cycle.

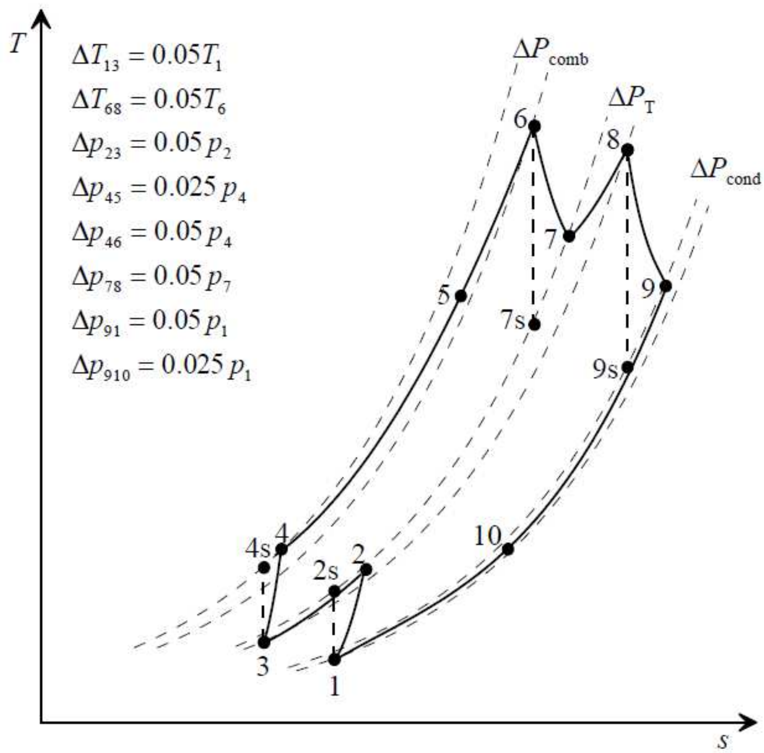

Fig. 2. *T*-*s* diagram of realistic Brayton cycle

3. Cycle Performance Parameters

The heat added to the system along processes 5- 6 and 7-8, and heat rejected from system through processes 10-1 and 2-3 are given, in terms of enthalpy as:

$$\dot{Q}_{add} = \dot{m}\left[(h_6 - h_5) + (h_8 - h_7)\right] \tag{20}$$

$$\dot{Q}_{rej} = \dot{m}\left[(h_{10} - h_1) + (h_2 - h_3)\right] \tag{21}$$

where $h_6 > h_8$ because $T_6 > T_8$ (assuming that $\Delta T_{86} = T_6 - T_8 = 0.05T_6$) and also $h_3 > h_1$ because $T_3 > T_1$ (assuming that $\Delta T_{13} = T_3 - T_1 = 0.05T_1$).

The power produced from the turbines ($\dot{W}_t$), runs the compressors ($\dot{W}_c$). Thus, the remaining power $\dot{W}_{net}$ and the back work ratio (BWR) becomes as:

$$\dot{W}_t = \dot{m}\left[(h_6 - h_7) + (h_8 - h_9)\right] \tag{22}$$

$$\dot{W}_c = \dot{m}\left[(h_2 - h_1) + (h_4 - h_3)\right] \tag{23}$$

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \tag{24}$$

$$\mathrm{BWR} = \frac{\dot{W}_c}{\dot{W}_t} \tag{25}$$

The first law thermal efficiency (η_{I}) and the second law efficiency (η_{II}), defined as the ratio of actual net work to the reversible net work are given by:

$$\eta_{\mathrm{I}} = \frac{\dot{W}_{net}}{\dot{Q}_{add}} = 1 - \frac{\dot{Q}_{rej}}{\dot{Q}_{add}} \tag{26}$$

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{net}}{\dot{W}_{net,rev}} = \frac{\dot{W}_{net}}{\dot{W}_{net} + \dot{X}_{\mathrm{dest}}} \tag{27}$$

where $\dot{X}_{\mathrm{dest}}$ is the rate of exergy destruction defined, in terms of the dead state temperature T_o as:

$$\begin{aligned}\dot{X}_{\mathrm{dest}} &= T_o \dot{S}_{gen} = T_o\left(\Delta\dot{S}_{tot}\right) = T_o(\Delta\dot{S}_{12} + \Delta\dot{S}_{23-C2C3} + \Delta\dot{S}_{34} + \Delta\dot{S}_{45-910} + \Delta\dot{S}_{56-H5H6} + \\ &\qquad \Delta\dot{S}_{67} + \Delta\dot{S}_{78-H7H8} + \Delta\dot{S}_{89} + \Delta\dot{S}_{101-C10C1}) \\ &= T_o\left(\dot{C}_{pC2C3}\ln\frac{T_{C3}}{T_{C2}} + \dot{C}_{pH5H6}\ln\frac{T_{H6}}{T_{H5}} + \dot{C}_{pH7H8}\ln\frac{T_{H8}}{T_{H7}} + \dot{C}_{pC10C1}\ln\frac{T_{C1}}{T_{C10}}\right)\end{aligned} \tag{28}$$

Entropy change for any process (e.g. process a-b) follows the expression in [29] as:

$$\Delta s_{ab} = \begin{cases} \left(s_b^o - s_a^o\right) - R\ln\dfrac{p_b}{p_a} & \text{For variable specific heat} \\ c_p\ln\dfrac{T_b}{T_a} - R\ln\dfrac{p_b}{p_a} & \text{For constant specific heat} \end{cases} \tag{29}$$

Final form of Eq. (28), after using Eq. (29), is

$$\dot{X}_{\mathrm{loss}} = T_o\left(\dot{C}_{pC2C3}\ln\frac{T_{C3}}{T_{C2}} + \dot{C}_{pH5H6}\ln\frac{T_{H6}}{T_{H5}} + \dot{C}_{pH7H8}\ln\frac{T_{H8}}{T_{H7}} + \dot{C}_{pC10C1}\ln\frac{T_{C1}}{T_{C10}}\right) \tag{30}$$

Recently [31], an ecological performance of the cycle, called ecological coefficient of performance (ECOP), is introduced to incorporate the cycle effect on environment. ECOP is defined as the power output per unit loss rate of availability as

$$\mathrm{ECOP} = \frac{\dot{W}_{net}}{\dot{X}_{\mathrm{loss}}} \tag{31}$$

4. Solution Procedure

The model presented by Eqs. (1-31) is used to calculate the performance parameters of the gas-turbine cycle. The operating and performance parameters are: inlet temperature and pressure to the first stage compressor T_1, p_1; maximum temperature entering the first stage turbine T_6; first and second stages compressor pressure ratios ,r_{p12}, r_{p34}; first stage turbine pressure ratio, r_{p67}; compressors and turbines efficiencies η_{c12}, η_{c34}, η_{t67}, η_{t89}; effectiveness of intercooler, regenerator, heat addition, reheater, and heat rejection ε_{int}, ε_{reg}, ε_{add}, ε_{reh}, ε_{rej}. Values of all these parameters are given in Table 1 as their commonly used values in the realistic literatures. Based on random independent selections of the values of these parameters within their variation ranges, 5001 complete calculations sets of cycle performance evaluation are recorded. Some rejection criteria are used to disregard any calculated cycle with nonrealistic performance values (e.g.: violation of the second law of thermodynamics, exergy loss is negative, negative efficiency values, negative work net, efficiencies higher than unity, … etc). Only 345 cycles were accepted and used in the present study. The accepted ranges of values of the controlling parameters are given in Table 1.

5. Results and Discussion

The effect of the operating parameters on the cycle performance is illustrated in Figs. 1-6. The performance parameters are the: first and second law efficiencies, back work ratio (BWR), the ecological coefficient of performance (ECOP), exergy losses, work net and heat added to the cycle. The main operating parameters are the inlet temperatures and pressures of air entering the compressor and gas turbine. All efficiencies and effectivnesses needed for the study and all pressure and temperature drops through heat transfer and fluid flow lines are indicated in Fig. 1 as well as the heat transfer coefficient-area products for all heat exchangers used in the cycle. Each of the scattered points in these figures represents a complete cycle with controlling parameter values within their variation ranges. Optimal operating parameter ranges are discussed in the next sections.

5.1. Optimum range of T_1

Figures 3a-g represent the dependency of cycle performance parameters on inlet air temperature T_1 at different values of the other cycle controlling parameters covering their accepted variation ranges given in Table 1. Values of η_{I} , shown in Fig. 3a, has its optimum values at T_1 in the range 301–389 K, irrespective of all other values of the cycle controlling parameters. This signifies that, as long as T_1 is within this range, η_{I} value will be optimum of about 38% to 48%. The above-mentioned range of T_1 results in optimum η_{II} within 33% to 66% , optimum ECOP within 1.56 to 1.92, optimum x_{loss} within 0.093 to 0.525 MJ/kg, optimum BWR within 0.473 to 0.6, optimum w_{net} within 0.178 to 0.341 MJ/kg and optimum q_{add} within 0.422 to 0.835 MJ/kg.

Figures 3b-g show the ranges of T_1 resulting in optimum values for the other cycle performance parameters (i.e. η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add}) regardless of the values of the other controlling parameters. In these figures respectively, ranges of T_1 are 301 to 412 K for $\eta_{II} \geq 60\%$ and ECOP$\geq$1.65, 302 to 417 K for $x_{loss} \leq 0.150$ MJ/kg, 301 to 315K for BWR ≤ 0.525, 301 to 369 K for $w_{net} \geq 0.300$ MJ/kg, and 302 to 352K for $q_{add} \leq 0.47$ MJ/kg. Effects of these ranges on other performance parameters are listed in Table 2.

Table 1: Ranges of operating and design parameters of an irreversible gas turbine cycle

Cycle controlling parameter	Surveyed range	Accepted range
T_1 entering 1st compressor, (K)	300-450	300-448
p_1 entering to 1st compressor, (kPa)	100-500	100-499
T_6 entering 1st turbine, (K)	800-1500	973-1483
1st compressor pressure ratios r_{p12}	1.2-5.4	1.281-5.393
2nd compressor pressure ratios r_{p34}	1.2-5.4	1.359-5.393
1st turbine pressure ratio r_{p67}	1.2-5.4	1.353-5.397
η_{c12} of 1st compressor	0.7-0.9	0.7024-0.8995
η_{c34} of 2nd compressor	0.7-0.9	0.7002-0.9000
η_{t67} of 1st turbine	0.7-0.9	0.7000-0.8998
η_{t89} of 2nd turbine	0.7-0.9	0.7002-0.8994
ε_{int} of intercooler	0.7-0.95	0.7000-0.9500
ε_{reg} of regenerator	0.7-0.95	0.7012-0.9496
ε_{reh} of reheater	0.7-0.95	0.7010-0.9490
ε_{bur} of high temperature heat addition	0.7-0.95	0.7010-0.9496
ε_{cond} of low temperature heat rejection	0.7-0.95	0.7003-0.9495

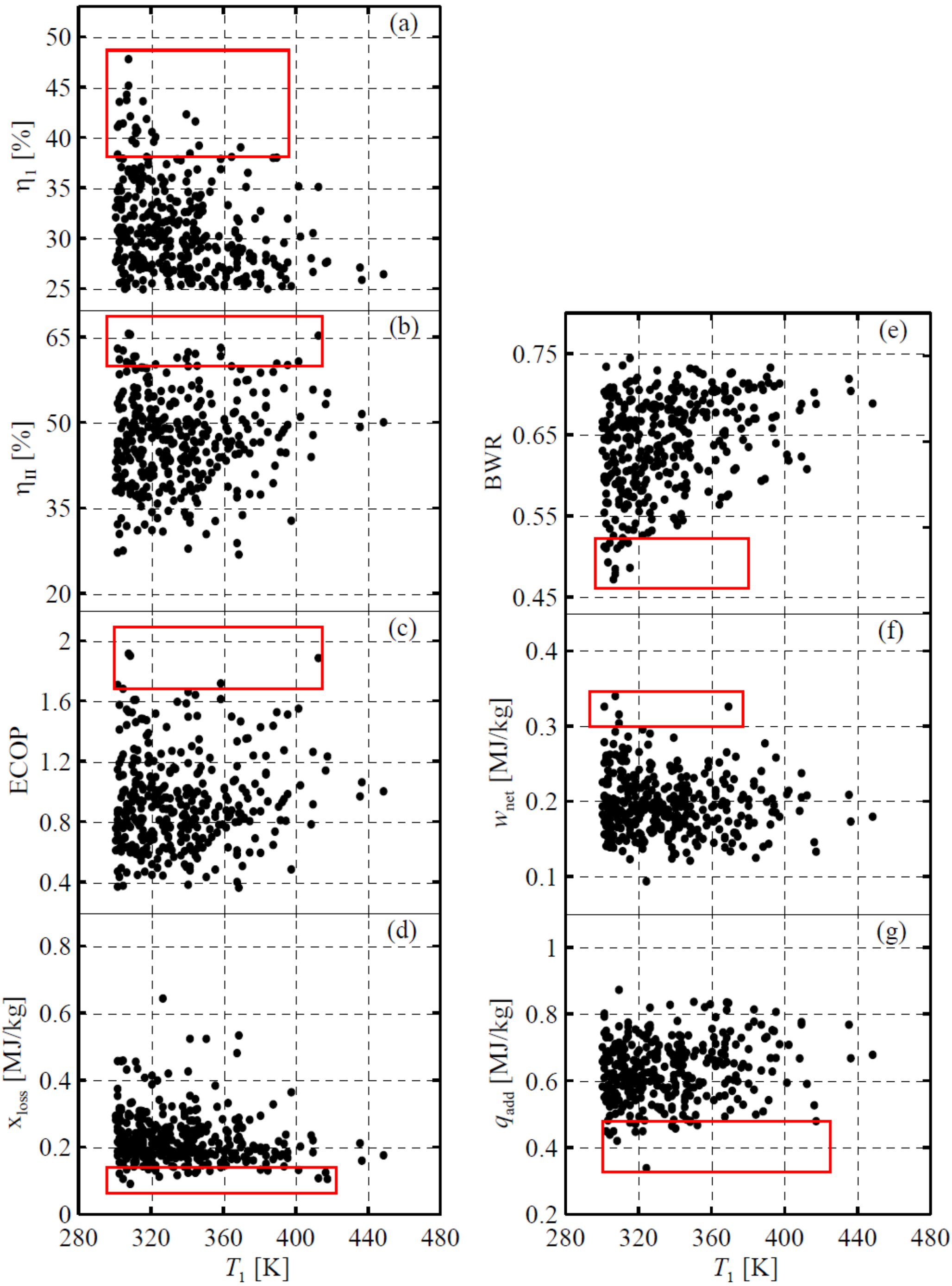

Fig. 3. Cycle performance parameters η_1, η_{II}, ECOP, x_{loss}, BWR, w_{net}, q_{add} versus inlet air temperature T_1.

Table 2: Ranges of operating, design, and performance parameters of an irreversible gas turbine Brayton cycle

Design and Performance parameters	$\eta_I \geq 38\%$	$\eta_{II} \geq 60\%$	ECOP ≥ 1.65	$x_{loss} \leq 0.150$ [MJ/kg]	BWR ≤ 0.525	$w_{net} \geq 0.300$ [MJ/kg]	$q_{add} \leq 0.470$ [MJ/kg]
T_1 ,K	301-389	301-412	301-412	302-417	301-**315**	301-369	**302**-352
T_6 ,K	1220-1480	1200-1480	1220-1480	1000-1420	1220-1480	**1340**-1480	1000-**1360**
P_4 ,kPa	750-7570	750-7570	864-4490	750-4490	864-4030	**1440**-7570	864-**2830**
UA, k W/K	13.6-37.0	16.8-37.0	**20.7**-34.7	14.8-37	14.2-34.7	16.6-**29.6**	13.8-34.7
η_I	38-48	32-48	35-48	25-44	35-48	36-48	27-44
η_{II}	33-66	60-66	63-66	45-66	39-66	48-66	37-66
ECOP	1.54-1.92	1.56-1.92	1.69-1.92	1.01-1.91	0.79-1.92	1.53-1.92	0.98-1.91
x_{loss}, MJ/kg	0.093-0.525	0.093-0.199	0.093-0.191	0.093-0.150	0.093-0.356	0.177-0.337	0.093-0.248
BWR	0.473-0.600	0.473-0.640	0.479-0.608	0.487-0.708	0.473-0.523	0.479-0.577	0.511-0.690
w_{net}, MJ/kg	0.178-0.341	0.178-0.341	0.178-0.341	0.093-0.246	0.178-0.341	0.305-0.341	0.093-0.197
q_{add}, MJ/kg	0.422-0.835	0.422-0.808	0.422-0.794	0.340-0.674	0.422-0.803	0.711-0.874	0.340-0.469

5.2. Optimum range of T_6

Compared to the almost uniform effect of T_1 discussed above that produces optimum performance, the effect of maximum temperature T_6 on optimum performance parameters is shown in Figs. 4a–g. Optimum values of performance parameters η_I, η_{II} , ECOP, $\dot{X}_{\text{loss}}$, BWR , w_{net} and q_{add} as mentioned previously in section 5.1 and listed in Table 2 requires T_6 to be in the ranges 1220 – 1480 K, 1200 – 1480 K, 1220 – 1480 K, 1000 – 1420 K, 1220 – 1480 K, 1340 – 1460 K and 1000 – 1380 K respectively. These values are generally expected, since the higher the maximum temperature the better the cycle performance (i.e. η_I, η_{II} , ECOP, and w_{net}) will be acceptable. The other two performance parameters, i.e. loss of exergy and the heat added to the cycle, necessitates that T_6 must be low to result in less exergy losses and less amount of heat added. This is manifested by the corresponding range of T_6 of 1000 – 1380K for these two performance parameters (Figs. 4d and 4g). The wide ranges of values of T_6 mentioned above are in favor of the practical application of the cycle that will accommodate any minor deterioration of the higher heating supply.

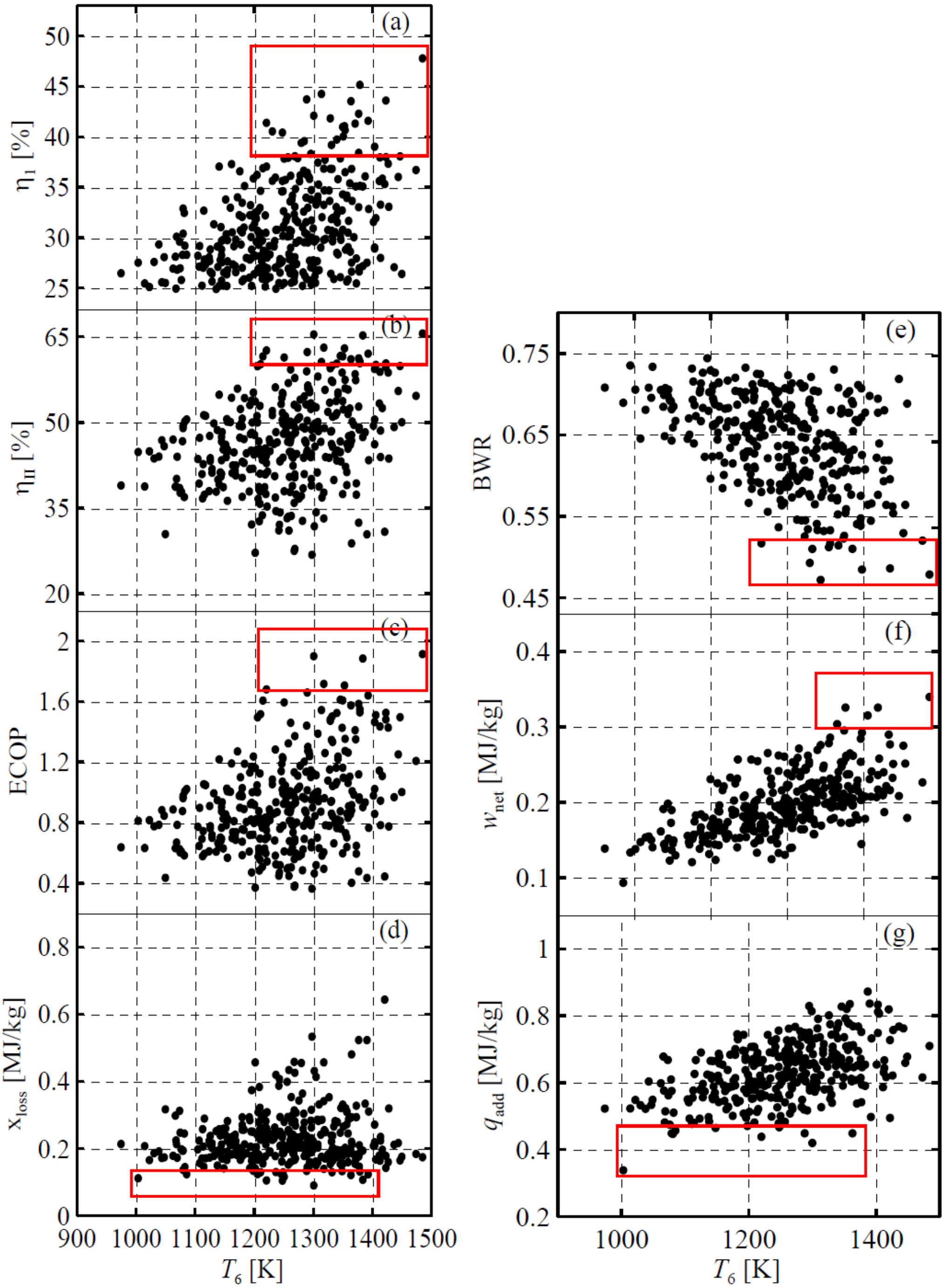

Fig. 4. Cycle performance parameters η_I, η_II, ECOP, x_{loss}, BWR, w_{net}, q_{add} versus maximum cycle temperature T_6

5.3. Optimum range of p_4

Another important design parameter for the gas turbine is the maximum pressure p_4. Its effects on optimum performance parameters are shown in Figs. 5a–g. Optimum values of performance parameters η_I, η_{II} , ECOP, $\dot{X}_{\mathrm{loss}}$, BWR , w_{net} and q_{add} require p_4 to be in the ranges 750 – 7570 kPa, 750 – 7570 kPa, 864 – 4490 kPa, 750 – 4490 kPa, 864 – 4030 kPa, 1440 - 7570 kPa and 864 – 2830 kPa respectively. Similar to T_6, the higher the maximum pressure the better the cycle performance. Optimum loss of exergy and heat added to the cycle necessitate that p_4 must be low to result in less losses and less amount of heat added. The corresponding range of p_4 of 864 – 2830 kPa is required to achieve optimality for these two performance parameters (Figs. 5d and 5g).

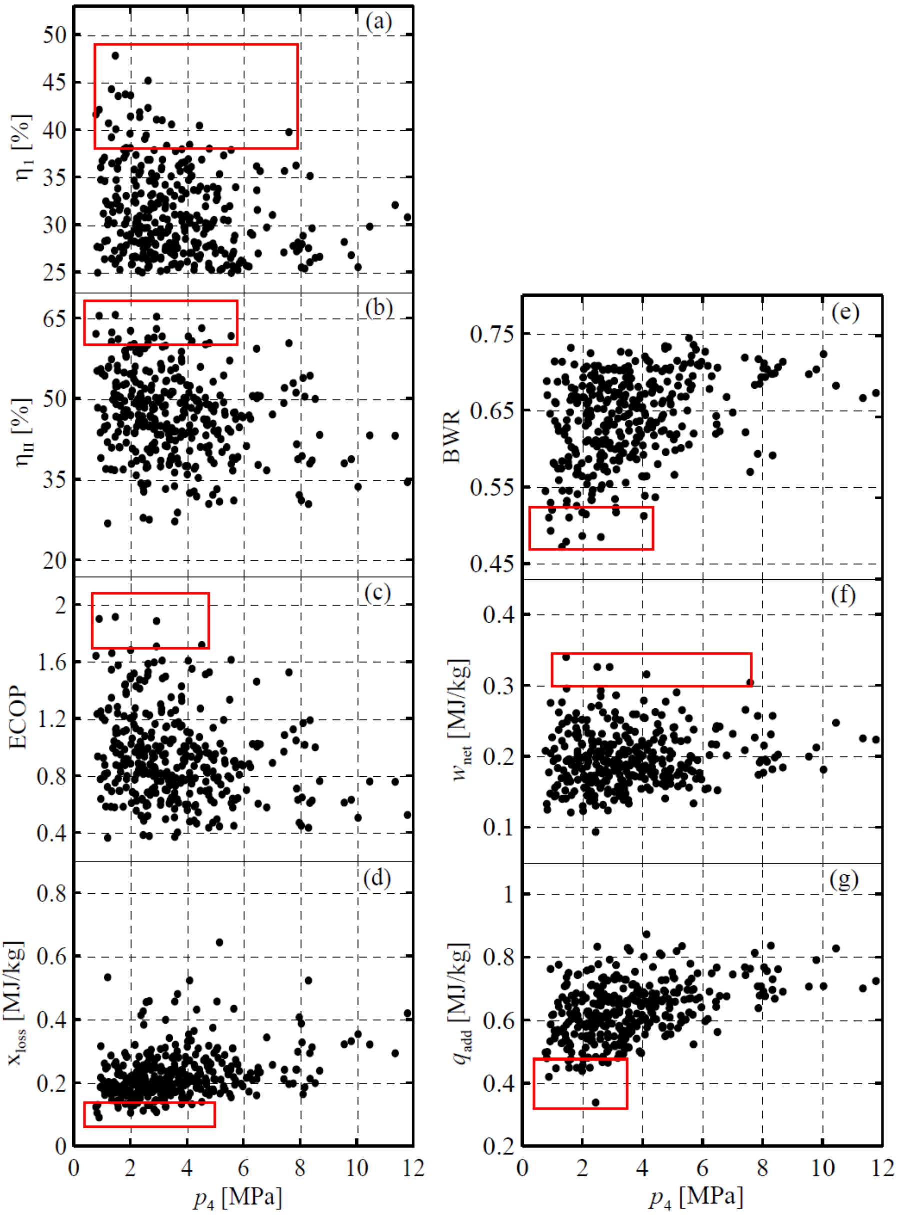

Fig. 5. Cycle performance parameters η_1, η_{II}, ECOP, x_{loss}, BWR, w_{net}, q_{add} versus maximum cycle pressure p_4.

5.4. Optimum range of UA

The heat exchanger thermal size is defined as the product of overall heat transfer coefficient and surface area of the heat exchanger ($UA = \dot{Q}_{add} / \Delta T_m$) [30]. The thermal size of a heat exchanger may become an object of optimization study based on a first law of thermodynamics and cost analysis. Therefore, thermal sizing of a heat exchanger is considered as important parameter among operating/design parameters of a heat exchanger and should be highlighted. The selection of an optimum range for *UA* of heat exchangers is illustrated in Figs. 6a-d. Optimum values of performance parameters η_I, η_{II} , ECOP, $\dot{X}_{\text{loss}}$, BWR , w_{net} and q_{add} require *UA* to be in the ranges 13.6-37 kW/K, 16.8-37 kW/K, 20.7-34.7 kW/K, 14.8-37 kW/K, 14.2-34.7 kW/K, 16.6-29.6 kW/K, and 13.6-34.7 kW/K , respectively.

5.5. Ranges of operating design parameters for optimum performance parameters simultaneously

Table 3 shows the ranges of the operating design parameters that give the optimum performance parameters (maximum η_I, η_{II}, ECOP, w_{net}, and minimum x_{loss}, BWR, and q_{add}) simultaneously for the irreversible gas turbine Brayton cycle used in this study. Range for each parameter is: air inlet temperature T_1 is 302-315 K; maximum cycle temperature T_6 is 1340-1360 K; maximum cycle pressure is 1440-2840 kPa; and heat exchangers thermal size is 20.7-29.6 kW/K. Surprisingly, ranges of the compressor inlet air temperature T_1 and maximum cycle temperature T_6 are almost constant and around 310 K and 1350 K, respectively.

Table 3: Optimum operating parameters to achieve optimum performance parameters simultaneously of an irreversible gas turbine Brayton cycle

Design parameters	Optimum Range
Compressor inlet air temperature, T_1 ,K	302-315
Maximum cycle temperature, T_6 ,K	1340-1360
Maximum cycle pressure P_4 ,kPa	1440-2830
Heat exchanger thermal size, *UA*, kW/K	20.7-29.6

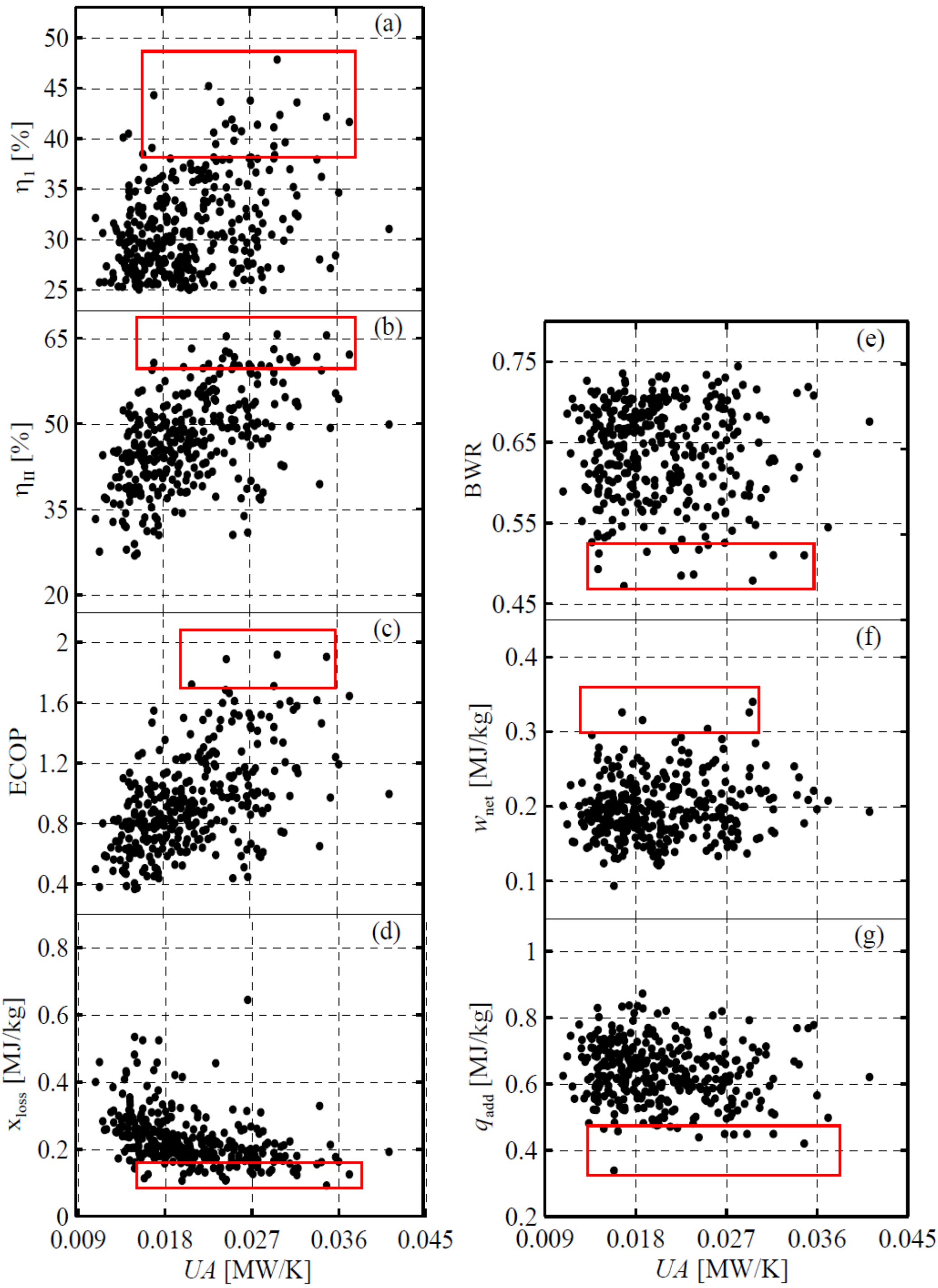

Fig. 6. Cycle performance parameters η_I, η_{II}, ECOP, x_{loss}, BWR, w_{net}, q_{add} versus total heat tansfer coefficient-area product.

6. Conclusions

Gas turbine cycles had received an extensive research focus due to their realistic representation of industrial and electrical power generation units. In this study, a simulation mathematical model was developed to specify the optimum performance of an irreversible gas turbine unit with different design and operating parameters. The cycle used in this study incorporated: two-stage compressor, two-stage gas turbine, intercooler, combustion chamber, re-heater, regenerator, and heat exchanger (condenser). The study included wide applicable ranges of the design and operating parameters and the irreversibilities due to finite rate of heat transfer, pressure drop in different lines of the cycle. Present results highlight design and operating ranges that can satisfy the optimum value of each performance parameter of the cycle separately and all performance parameters simultaneously. The optimization results showed that the minimum temperature ranges between 302 K and 315 K while the maximum temperature ranges between 1320 K–1360 K taking the meteorological impacts into consideration. The maximum pressure in the gas turbine cycle can be selected in the range of 1449 kPa to 2830 kPa in order to satisfy optimality of all performance parameters. Finally, the optimum thermal size of the heat exchangers ranges between 20.7 to 29.6 kW/K.

7. Acknowledgments

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8. References

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متغيرات التشغيل المثلى لدورة التوربين الغازي اللاإنعكاسية

تناقش الورقة الحالية أداء دورة التوربين الغازي الاإنعكاسية الاسترجاعية ذات تبريد بينى وإعادة تسخين تحت مدى واسع لمتغيرات الأداء و التصميم. تهدف الدراسة إلى الحصول على المدى المناسب لهذه المتغيرات و التي تحقق معظم متطلبات دورة التوربين الغازي. شملت المتغيرات المدروسة: درجة حرارة دخول الهواء لمرحلة الانضغاط الأولى – درجة حرارة وضغط الهواء الداخل للمرحلة الأولى في التوربين بالإضافة إلى الحجم الحراري (حاصل ضرب المساحة في معامل الانتقال الحراري) لكل المبادلات الحرارية المستخدمة في الدورة.

تم تصميم نموذج محاكاة رياضي لحساب متغيرات الأداء للدورة عند ظروف التصميم المختلفة لتحقيق معظم متطلبات الدورة مثل أقصى كفاءة للقانون الأول و الثاني للديناميكا الحرارية – اقل نسبة شغل خلفي – أقصى معامل أداء بيئى – أقصى قدرة بالإضافة إلى اقل مفاقيد للاكسرجى. ولتحقيق بعض أو كل هذه المتطلبات تم اختيار مدى للدراسة على متغيرات التصميم المختلفة.

أظهرت النتائج أن المنطقة المثلى لتحقيق أداء أمثل لدورة التوربين الغازي: 302 – 315 كلفن لأقل درجة حرارة في الدورة – 1340 –1360 كلفن لأقصى درجة حرارة و 1440 – 2830 كيلو باسكال لأقصى ضغط في الدورة. أخيرا كان الحجم الحراري المثالي للمبادلات الحرارية في الدورة هو 20.7 – 29.6 كيلووات لكل كلفن.