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Optimum parametric performance characterization of an irreversible gas turbine Brayton cycle

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Abstract

A general mathematical model is developed to specify the performance of an irreversible gas turbine Brayton cycle incorporating two-stage compressor, two-stage gas turbine, intercooler, reheater, and regenerator with irreversibilities due to finite heat transfer rates and pressure drops. Ranges of operating parameters resulting in optimum performance (i.e., $\eta_I \geq 38 \geq \eta_{II} \geq 60\%$, ECOP ≥ 1.65, $x_{loss} \leq 0.150$ MJ/kg, BWR ≤ 0.525, $w_{net} \geq 0.300$ MJ/kg, and $q_{add} \leq 0.470$ MJ/kg) are determined and discussed using the Monte Carlo method. These operating ranges are minimum cycle temperature ranges between 302 and 315 K, maximum cycle temperature ranges between 1,320 and 1,360 K, maximum cycle pressure ranges between 1.449 and 2.830 MPa, and conductance of the heat exchanger ranges between 20.7 and 29.6 kW/K. Exclusive effect of each of the operating parameters on each of the performance parameters is mathematically given in a general formulation that is applicable regardless of the values of the rest of the operating parameters and under any condition of operation of the cycle.

Keywords: Intercooled reheat Brayton cycle; Regenerator; Turbine; Compressor; Operating parameters; Performance

Background

First gas turbines developed in the 1930's used to have representative simple cycle efficiencies of about 17% due to low compressor and turbine efficiencies and low turbine inlet temperatures for material stress and thermal limitations. Efforts to improve these efficiencies have specifically or concurrently concentrated in three areas: (1) modifying the working cycle, (2) increasing turbine inlet temperature, and (3) enhancing the performance of cycle components. Recently, developments in material science allow using turbine inlet temperatures up to 1,500°C (i.e., general electric uses a turbine inlet temperature of 1,425°C). Also, continuous modifications of Brayton cycle to include regeneration [1,2], isothermal heat addition [3-6], intercooled compression [7,8], reheat expansion [9,10], and combined modifications [11-14] have resulted in practically doubling the cycle efficiencies. This is because intercooling and reheating result in decreasing the average temperature at which heat is added. Finally, computer-aided design and simulation studies have enabled optimization of cycle components such as compressors and turbines.

The Brayton cycle, as a model of gas turbine power plants, has been optimized for entropy generation [15,16], reversible work [17,18], power [19-22], power density [23-25], internal irreversibilities of compressors and turbines [26,27], pressure drops in heaters, coolers, and regenerators [19,23,24,28], and external irreversibilities of coupling to external heat reservoirs or heat exchangers [20].

Most of the abovementioned literature studies have been carried out to improve the performance of real gas power plants through the optimization of design and operating parameters such as compressor and turbine inlet temperatures, pressure ratios of intercooling, reheat, and conductance of heat exchangers [12,29-33]. However, most of the previously published results found in the open literature are typically specific and valid only for the condition and parameter values taken into consideration in these studies. This means that according to the authors' knowledge, there is no general optimized work that has been done before. Therefore, and for the sake of generalized tackling of this issue, the main objective of the present study of an irreversible regenerative intercooled reheat gas turbine Brayton cycle is to identify the ranges of all design and operating parameters for optimized performance. The design and operating parameters include inlet temperatures to compressors and turbines and pressure ratios of

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intercooler and reheater. The performance parameters include the first and second law efficiencies, ecological coefficient of performance, back work ratio, exergy losses, network, and heat added.

Methods

Mathematical model

Consider a constant mass flow rate, $\dot{m}$, of air, as an ideal gas passing through the gas turbine cycle illustrated in Figures 1 and 2. The cycle can be characterized as follows:

a) Air is compressed from state 1 to state 4 by two non-isentropic low pressure (LP) and high pressure (HP) compressors with efficiencies, η_{c12} and η_{c34}, and a non-isobaric counter-flow intercooler with effectiveness, ε_{int}. The inlet temperature to the HP compressor is 5% higher than that of the LP compressor. The describing equations for these processes (e.g., [34,35]) are as follows:

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

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

$$\varepsilon_{\text{int}} = \frac{\dot{Q}_{23}}{\dot{Q}_{\text{int max}}} = \frac{(\text{UA})_{\text{int}}(\Delta T_{\text{LM}})_{\text{int}}}{\dot{Q}_{\text{int max}}} = \frac{\dot{Q}_{23}}{\min\left(\dot{C}_W, \dot{C}_{23}\right) \times (T_2 - T_{C2})} \quad (3)$$

Quantities $\dot{Q}_{23}$, $\dot{C}_W$, and $\dot{C}_{23}$ represent the rate of heat release and heat capacity rates for cooling fluid and air, respectively. The intercooler logarithmic mean temperature difference $(\Delta T_{\text{LM}})_{\text{int}}$ is defined as follows:

$$(\Delta T_{\text{LM}})_{\text{int}} = \frac{(T_2 - T_{C3}) - (T_3 - T_{C2})}{\ln((T_2 - T_{C3})/(T_2 - T_{C3}))} \quad (4)$$

b) Air is preheated from state 4 to state 5 in a regenerative counter-flow heat exchanger (that will be discussed later in the heat rejection process) and then heated up to a maximum temperature, T_6, by a counter-flow heat exchanger having a rate of heat

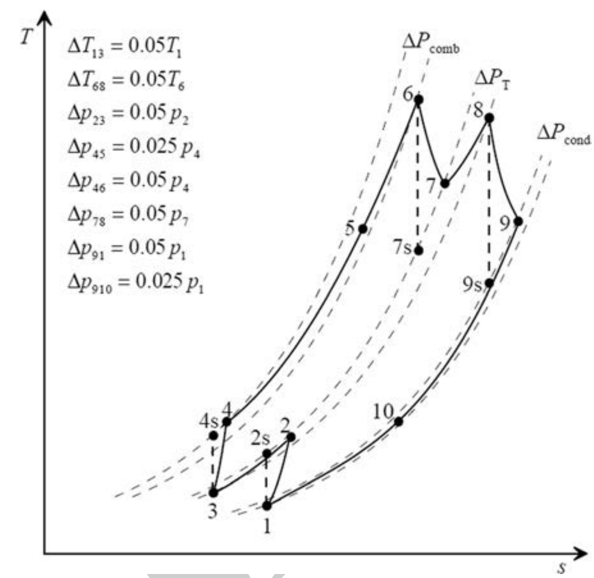

Figure 2 ***T-s*** **diagram of realistic Brayton cycle.**

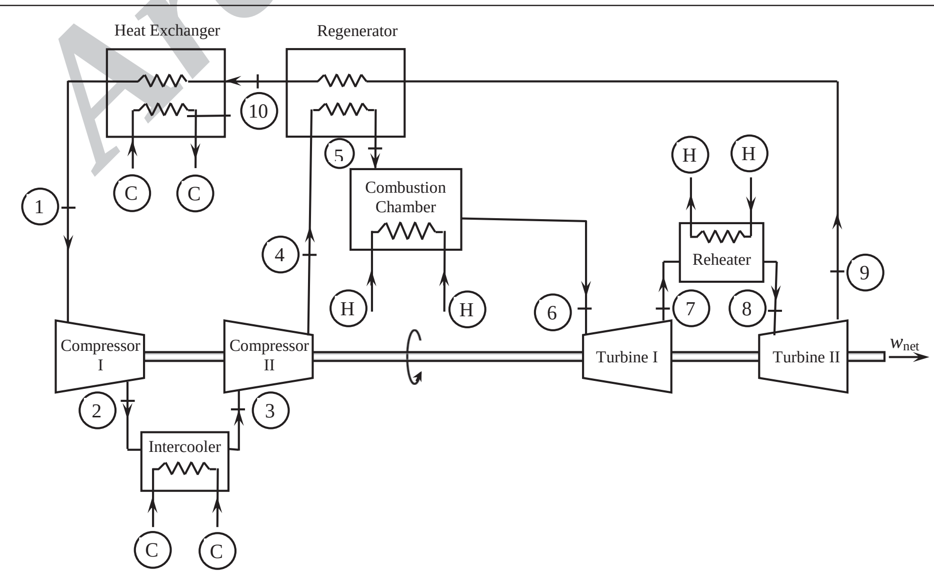

Figure 1 Schematic diagram of a realistic irreversible, regenerative, and reheat Brayton cycle.

addition $\dot{Q}_{56}$, an effectiveness ε_{add}, and a logarithmic mean temperature difference $(\Delta T_{\text{LM}})_{\text{add}}$ defined as follows:

$$\varepsilon_{\text{add}} = \frac{\dot{Q}_{56}}{\dot{Q}_{\text{add max}}} = \frac{(\text{UA})_{\text{add}}(\Delta T_{\text{LM}})_{\text{add}}}{\dot{Q}_{\text{add max}}} = \frac{\dot{Q}_{56}}{\min\left(\dot{C}_W, \dot{C}_{56}\right) \times (T_5 - T_{H5})} \tag{5}$$

$$(\Delta T_{\text{LM}})_{\text{add}} = \frac{(T_5 - T_{H6}) - (T_6 - T_{H5})}{\ln((T_5 - T_{H6})/(T_6 - T_{H5}))} \tag{6}$$

c) Air is expanded from state 6 to final state 9 by two non-isentropic LP and HP turbines with efficiencies η_{t67} and η_{t89} and one non-isobaric reheater having a rate of heat added, an effectiveness, and a logarithmic mean temperature difference as $\dot{Q}_{78}$, ε_{reh}, and $(\Delta T_{\text{LM}})_{\text{reh}}$. The inlet temperature to LP turbine is 5% lower than that of the HP turbine. The governing equations for these processes are as follows:

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

$$\eta_{t89} = \frac{w_{t89}}{w_{t89s}} = \frac{h_8 - h_9}{h_8 - h_{9s}} \tag{8}$$

$$\varepsilon_{\text{reh}} = \frac{\dot{Q}_{78}}{\dot{Q}_{\text{reh max}}} = \frac{(\text{UA})_{\text{reh}}(\Delta T_{\text{LM}})_{\text{reh}}}{\dot{Q}_{\text{reh max}}} = \frac{\dot{Q}_{78}}{\min\left(\dot{C}_H, \dot{C}_{78}\right) \times (T_{H7} - T_7)} \tag{9}$$

$$(\Delta T_{\text{LM}})_{\text{reh}} = \frac{(T_{H7} - T_8) - (T_{H8} - T_7)}{\ln((T_{H7} - T_8)/(T_{H8} - T_7))} \tag{10}$$

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

$$\varepsilon_{\text{reg}} = \frac{\dot{Q}_{45}}{\dot{Q}_{\text{reg max}}} = \frac{(\text{UA})_{\text{reg}}(\Delta T_{\text{LM}})_{\text{reg}}}{\dot{Q}_{\text{reg max}}} = \frac{\dot{Q}_{45}}{\min\left(\dot{C}_{45}, \dot{C}_{910}\right) \times (T_9 - T_4)} \tag{11}$$

$$(\Delta T_{\text{LM}})_{\text{reg}} = \frac{(T_9 - T_5) - (T_{10} - T_4)}{\ln((T_9 - T_5)/(T_{10} - T_4))} \tag{12}$$

$$\varepsilon_{\text{rej}} = \frac{\dot{Q}_{101}}{\dot{Q}_{\text{rej max}}} = \frac{(\text{UA})_{\text{rej}}(\Delta T_{\text{LM}})_{\text{rej}}}{\dot{Q}_{\text{bur max}}} = \frac{\dot{Q}_{101}}{\min\left(\dot{C}_W, \dot{C}_{101}\right) \times (T_{10} - T_{C10})} \tag{13}$$

$$(\Delta T_{\text{LM}})_{\text{rej}} = \frac{(T_{10} - T_{C1}) - (T_1 - T_{C10})}{\ln((T_{10} - T_{C1})/(T_1 - T_{C10}))} \tag{14}$$

e) The abovementioned heat exchangers (i.e., intercooler, regenerator, high temperature heat addition, reheater, and low temperature heat rejection) are of counter-flow types, and their effectiveness can be calculated (e.g., [35]) as follows:

$$\varepsilon_i = \frac{1 - \exp[-\text{NTU}(1 - C^*)]}{1 - C^* \exp[-\text{NTU}(1 - C^*)]}, \quad i = \text{int, reg, add, reh, rej} \tag{15}$$

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

Cycle performance parameters

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

$$\dot{Q}_{\text{add}} = \dot{m}[(h_6 - h_5) + (h_8 - h_7)] \tag{16}$$

$$\dot{Q}_{\text{rej}} = \dot{m}[(h_{10} - h_1) + (h_2 - h_3)] \tag{17}$$

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

The power produced by both LP and HP turbines ($\dot{W}_t$) is partially consumed by both LP and HP compressors ($\dot{W}_c$), and the remaining power is the net power ($\dot{W}_{\text{net}}$) as follows:

$$\dot{W}_{\text{t}} = \dot{m}[(h_6 - h_7) + (h_8 - h_9)] \tag{18}$$

$$\dot{W}_{\text{c}} = \dot{m}[(h_2 - h_1) + (h_4 - h_3)] \tag{19}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{t}} - \dot{W}_{\text{c}} \tag{20}$$

The back work ratio (BWR) and first and second law thermal efficiencies (η_{I}, η_{II}) of the cycle are as follows:

$$\text{BWR} = \frac{\dot{W}_{\text{c}}}{\dot{W}_{\text{t}}} \tag{21}$$

$$\eta_{\text{I}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{add}}} = 1 - \frac{\dot{Q}_{\text{rej}}}{\dot{Q}_{\text{add}}} \tag{22}$$

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{net}}}{\dot{W}_{\text{net,rev}}} = \frac{\dot{W}_{\text{net}}}{\dot{W}_{\text{net}} + \dot{X}_{\text{dest}}} \tag{23}$$

where $\dot{X}_{\text{dest}}$ is the rate of exergy destruction defined, with respect to the dead state temperature T_0 as follows:

$$\dot{X}_{\text{dest}} = T_0(\Delta\dot{S}_{12} + \Delta\dot{S}_{23-C2C3} + \Delta\dot{S}_{34} + \Delta\dot{S}_{45-910} + \Delta\dot{S}_{56-H5H6} + \Delta\dot{S}_{67} + \Delta\dot{S}_{78-H7H8} + \Delta\dot{S}_{89} + \Delta\dot{S}_{101-C10C1}) \tag{24}$$

where the above entropy changes are calculated according to [34], taking into consideration the temperature-dependent specific heats.

For the sake of ecological performance of the cycle, and its effect on environment, the following ecological coefficient of performance (ECOP), as was previously introduced by [36,37], is defined as the power output per unit loss rate of availability as follows:

$$\text{ECOP} = \frac{\dot{W}_{\text{net}}}{\dot{X}_{\text{loss}}} \tag{25}$$

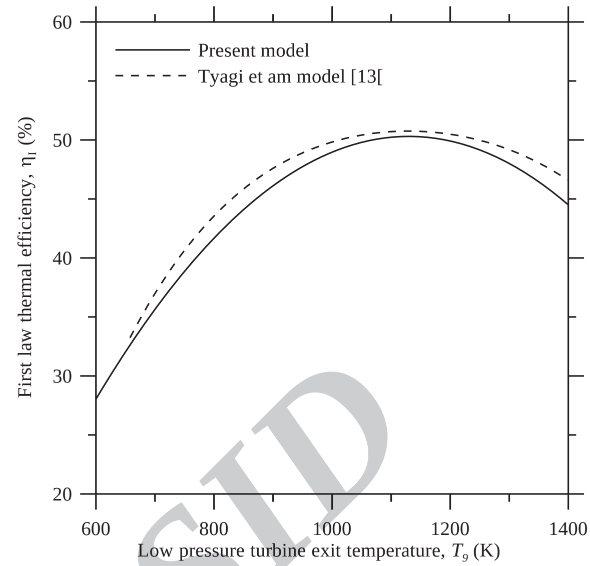

Figure 3 Variation of first law efficiency with low pressure turbine inlet temperature.

Solution procedure

The above set of equations represents complete thermodynamic modeling of the cycle, whose solution gives the cycle performance as dependent on its controlling parameters. Following conventionally reported methods of varying one or two of the controlling parameters at a time while keeping the rest of the constants will produce some specific performance results that will be valid only for those specific variation cases and cannot be of general practical applicability. Besides, these conventional solution methods can result in localized optimized performance values that are dependent on the specific values selected for the controlling parameters. To overcome these two issues (i.e., the generalization of the study and the global optimization), the present paper has adapted the Monte Carlo methodology (MCM) that concurrently searches the variation ranges of all controlling parameters at the same time to optimize the cycle performance over the whole domain of variations of all cycle controlling parameters.

Table 1 Surveyed ranges and accepted ranges of the cycle controlling parameters

Cycle controlling parameter	Surveyed range	Accepted range by MCM
T_1 entering LP compressor, (K)	300 to 450	300 to 448
P_1 entering to LP compressor, (kPa)	100 to 500	100 to 499
T_6 entering HP turbine, (K)	800 to 1500	973 to 1,483
LP compressor pressure ratios r_{p12}	1.2 to 5.4	1.281 to 5.393
HP compressor pressure ratios r_{p34}	1.2 to 5.4	1.359 to 5.393
HP turbine pressure ratio r_{p67}	1.2 to 5.4	1.353 to 5.397
η_{c12} of LP compressor	0.7 to 0.9	0.7024 to 0.8995
η_{c34} of LP compressor	0.7 to 0.9	0.7002 to 0.9000
η_{t67} of HP turbine	0.7 to 0.9	0.7000 to 0.8998
η_{t89} of LP turbine	0.7 to 0.9	0.7002 to 0.8994
ε_{int} of intercooler	0.7 to 0.95	0.7000 to 0.9500
ε_{reg} of regenerator	0.7 to 0.95	0.7012 to 0.9496
ε_{reh} of reheater	0.7 to 0.95	0.7010 to 0.9490
ε_{bur} of high temperature heat addition	0.7 to 0.95	0.7010 to 0.9496
ε_{rej} of low temperature heat rejection	0.7 to 0.95	0.7003 to 0.9495

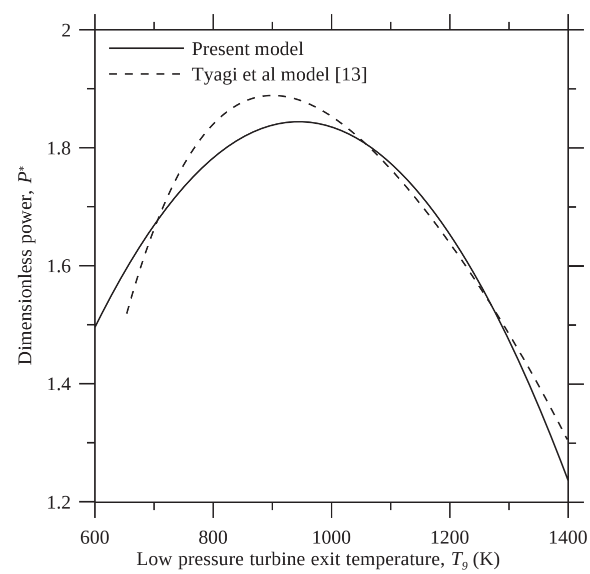

Figure 4 Variation of dimensionless power with low pressure turbine inlet temperature.

MCM optimization technique

The procedure of utilizing the MCM technique can be summarized as follows: (1) selection of the design and operating controlling parameters of the cycle, (2) selection of their practical variation ranges, (3) selection of the performance parameters sought to be optimized, (4) setting an acceptance-rejection criterion for the resulting performance values, (5) random selection of one complete set of values of all the controlling parameters within their variation ranges, (6) solution of the model equations (i.e., Equations 1, 2, 3, 4, 5, 6, 7 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25) for cycle performance to get a complete set of results based on the randomly selected set of controlling parameters, (7) applying the acceptance-rejection criterion to discard the unwanted performance values and to record the rest, and (8) repeating the above steps for another random selection of another complete set of values for the controlling parameters. The above eight steps are discussed as follows:

The design and operating parameters are as follows: inlet temperature and pressure to LP compressor T_1, P_1; maximum temperature T_6 entering HP turbine; pressure ratios r_{p12} and r_{p34} of LP and HP compressors; pressure ratio, r_{p67} of HP turbine; compressors and turbine

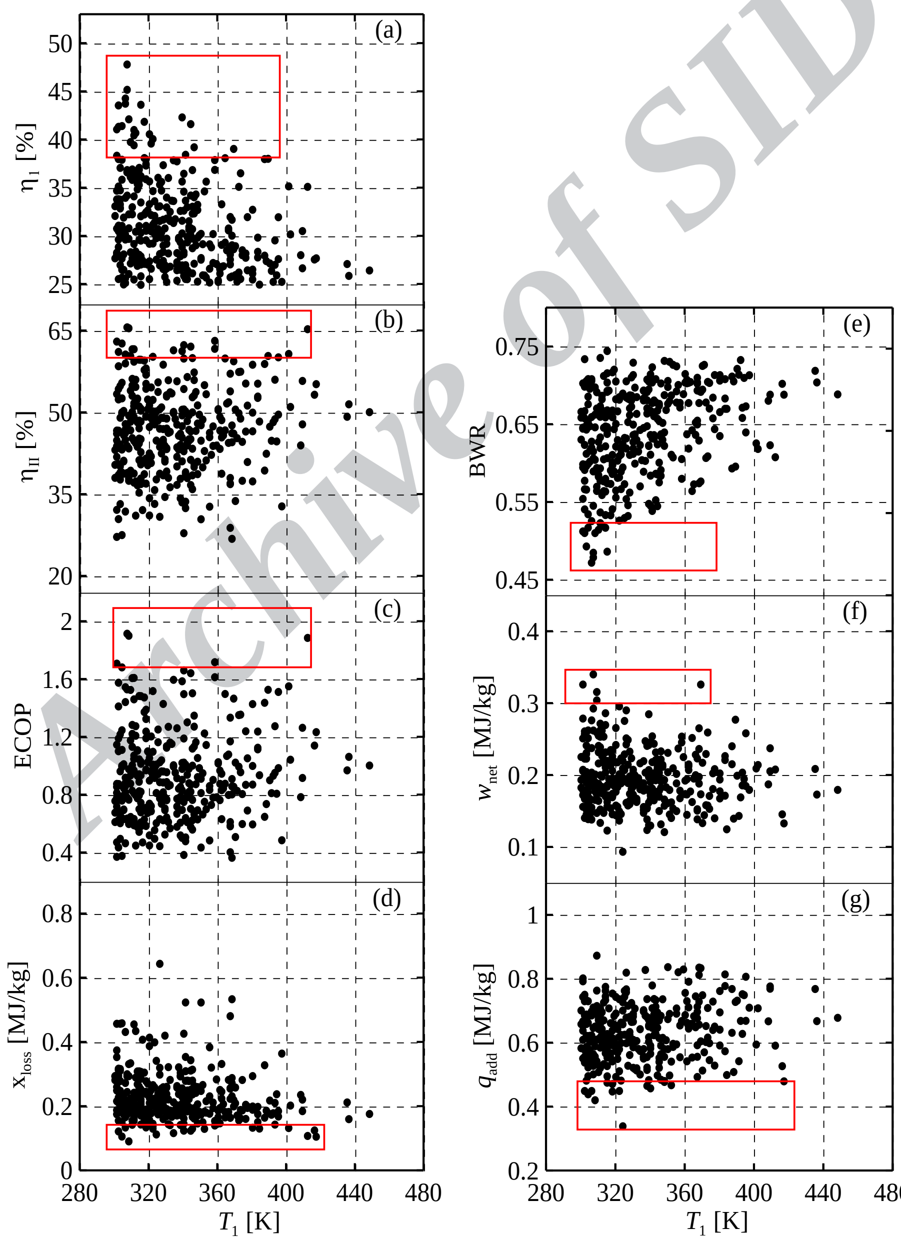

Figure 5 a-g Cycle performance parameters η_1, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} versus inlet air temperature T_1.

Table 2 Optimized performance parameters with their respected ranges of controlling 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]
Design and performance parameters							
T_1, K	301 to 389	301 to 412	301 to 412	302 to 417	301 to *315*	301 to 369	*302* to 352
T_6, K	1,220 to 1,480	1,200 to 1,480	1,220 to 1,480	1,000 to 1,420	1,220 to 1,480	*1,340* to 1,480	1,000 to *1,360*
P_4, kPa	750 to 7,570	750 to 7,570	864 to 4,490	750 to 4,490	864 to 4,030	*1,440* to 7,570	864 to *2,830*
UA, kW/K	13.6 to 37.0	16.8 to 37.0	*20.7* to 34.7	14.8 to 37	14.2 to 34.7	16.6 to *29.6*	13.8 to 34.7
Optimum ranges of performance parameters that are achieved by ranges of operating parameters shown above							
η_I	38 to 48	32 to 48	35 to 48	25 to 44	35 to 48	36 to 48	27 to 44
η_{II}	33 to 66	60 to 66	63 to 66	45 to 66	39 to 66	48 to 66	37 to 66
ECOP	1.54 to 1.92	1.56 to 1.92	1.69 to 1.92	1.01 to 1.91	0.79 to 1.92	1.53 to 1.92	0.98 to 1.91
x_{loss}, MJ/kg	0.093 to 0.525	0.093 to 0.199	0.093 to 0.191	0.093 to 0.150	0.093 to 0.356	0.177 to 0.337	0.093 to 0.248
BWR	0.473 to 0.600	0.473 to 0.640	0.479 to 0.608	0.487 to 0.708	0.473 to 0.523	0.479 to 0.577	0.511 to 0.690
w_{net}, MJ/kg	0.178 to 0.341	0.178 to 0.341	0.178 to 0.341	0.093 to 0.246	0.178 to 0.341	0.305 to 0.341	0.093 to 0.197
q_{add}, MJ/kg	0.422 to 0.835	0.422 to 0.808	0.422 to 0.794	0.340 to 0.674	0.422 to 0.803	0.711 to 0.874	0.340 to 0.469

efficiencies η_{c12}, η_{c34}, η_{t67}, and η_{t89}; and effectiveness of intercooler, regenerator, heat addition, reheater, and heat rejection ε_{int}, ε_{reg}, ε_{add}, ε_{reh}, and ε_{rej}, respectively. To reflect the commonly used realistic literature values, survey ranges of the controlling parameters are selected as shown in Table 1. The acceptable-rejection criteria used to disregard non-realistic performance values includes many conditional terms such as (and not limited to) follows: rejection of calculations based on violation of the second law of thermodynamics, exergy loss is negative, negative values of cycle efficiency, negative values of network, efficiencies higher than unity, unrealistic ratio of specific volumes of the two compressors, unrealistic ratio of the works of the two turbines,...etc. Based on random independent selections of values of the controlling parameters within their variation ranges, 5,000 complete calculation sets of cycle performance evaluation have been executed. Applying the acceptable-rejection criterion to these 5,000 sets of calculations has resulted on accepting only 345. The surveyed ranges of values of the controlling parameters given in the first column of Table 1 have been readjusted into acceptable ranges as shown in the second column in the same table. The results are discussed below.

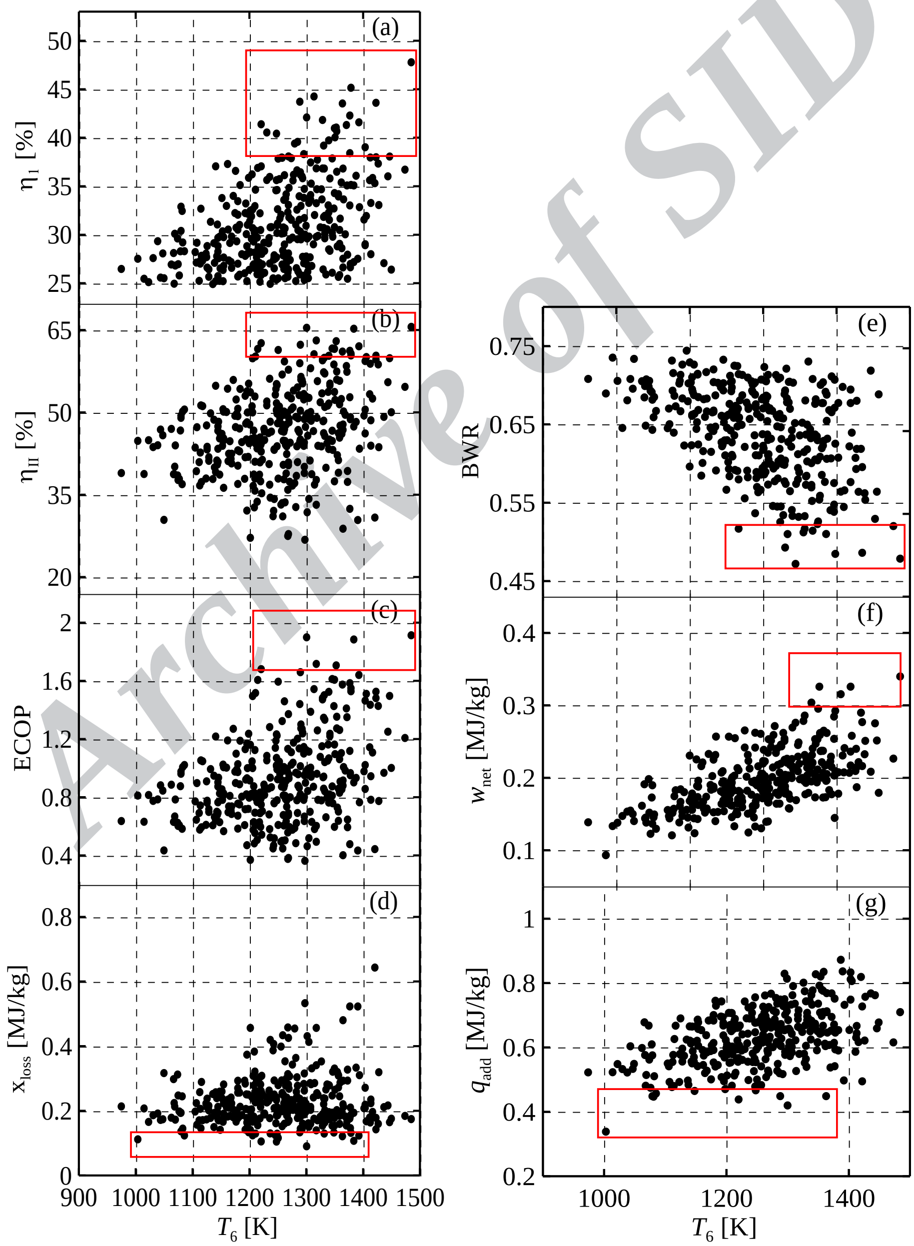

Figure 6 a-g Cycle performance parameters η_1, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} versus maximum cycle temperature T_6.

Results and discussions

Validation of the present model

The operating parameters of the present model have been modified to agree with those employed in the theoretical model of Tyagi et al. [13]. Variations of first law efficiency and dimensionless power output with the low pressure turbine exit temperature for the present model and its comparison of Tyagi et al. [13] are illustrated in Figures 3 and 4, respectively. Comparisons show slight deviations that could be attributed to the pressure drop employed in the present model and neglected in Tyagi et al. model.

Sensitivity analysis

The dependency of the performance parameters on the controlling parameters are displayed below as dependents, η_I, η_{II}, BWR, ECOP, x_{loss}, w_{net}, and q_{add}, and independents, T_1, T_6, P_4, and conductance of the whole cycle (i.e., summation of heat transfer coefficient-area product for all heat transfer units) UA. The shown figures display the 345 accepted results plotted as scattered points to relate the performance parameters to the controlling parameters. Each point on any of these figures represents a complete set of accepted cycle calculation, with controlling parameter values that lie within their variation ranges. Optimal

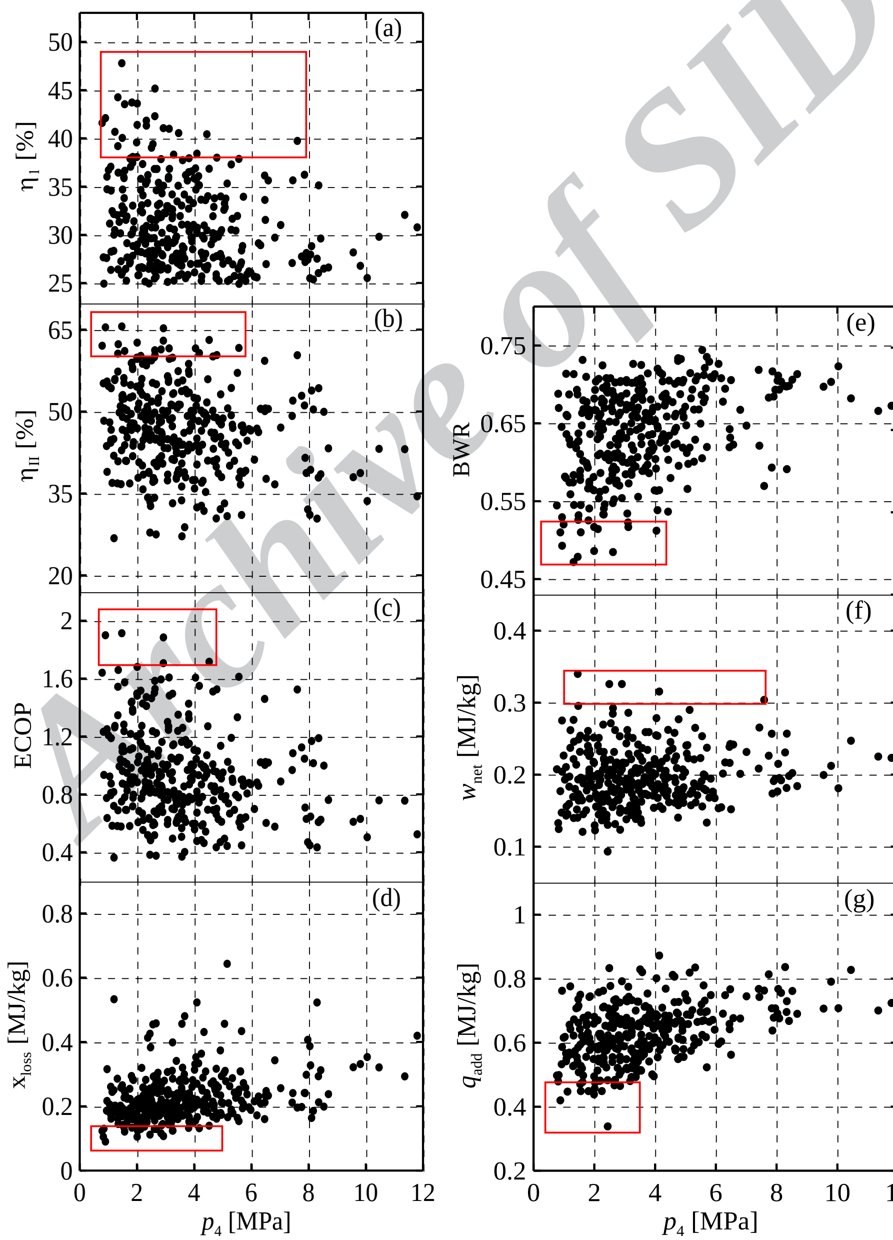

Figure 7 a-g Cycle performance parameters η_1, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} versus maximum cycle pressure P_4.

performance values and the required operating parameter ranges are discussed in the following sections.

Sensitivity of cycle performance to lowest cycle temperature T_1

Figure 5a,b,c,d,e,f,g shows the dependency of cycle performance on T_1 at values of other controlling parameters that lie within their variation ranges in Table 1. Values of η_I in Figure 5a is very sensitive to T_1 where it exhibits a steep decrease with T_1, where its optimum values >38% that lie in the T_1 range of about 301 to 389 K, regardless of the values of all other controlling parameters. This signifies that, outside this T_1 range, no modifications of other design or operating parameters can enhance the values of η_I beyond 38%. As expected, the lower the value of T_1, the higher is the value of η_I, with its optimum value decrease from about 48% to about 38% within this 301 to 389 K range. Figure 5b,c,d,e,f,g shows that the abovementioned range of T_1 results in optimum η_{II} in the range 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

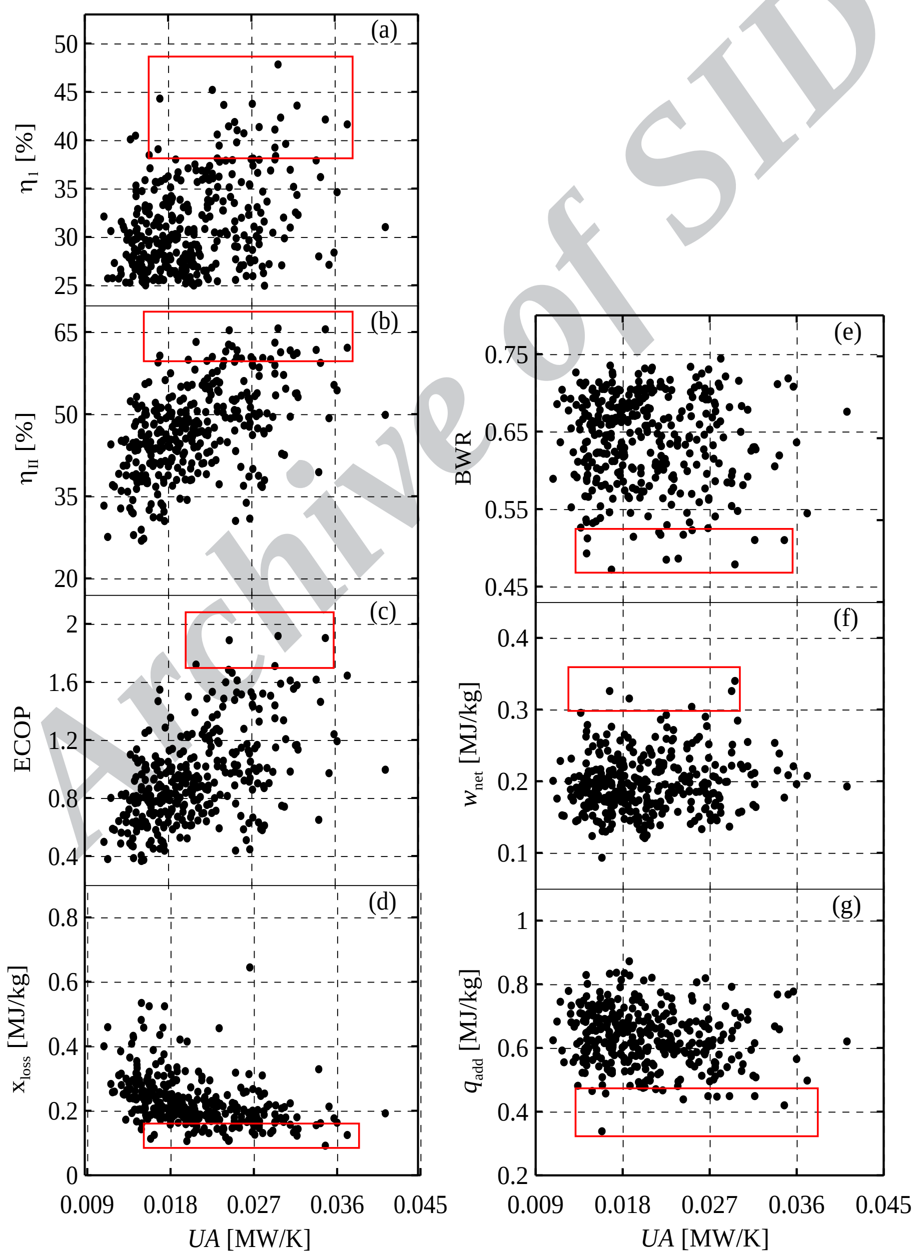

Figure 8 a-g Cycle performance parameters η_I, η_{II}, ECOP, x_{loss}, BWR, w_{net} and q_{add} versus total heat transfer coefficient-area product.

Table 3 Simultaneously optimum operating design parameters to achieve optimum performance parameters of an irreversible gas turbine Brayton cycle

Design parameters	Optimum range
Compressor inlet air temperature, T_1, K	302 to 315
Maximum cycle temperature, T_6, K	1,340 to 1,360
Maximum cycle pressure P_4, kPa	1,440 to 2,830
Heat exchanger conductance, UA, kW/K	20.7 to 29.6

0.341 MJ/kg, and optimum q_{add} within 0.422 to 0.835 MJ/kg. In these figures, respectively, ranges of T_1 are 301 to 412 K for $\eta_{\mathrm{II}} \geq 60\%$ and ECOP ≥ 1.65, 302 to 417 K for $x_{\mathrm{loss}} \leq 0.150$ MJ/kg, 301 to 315 K for BWR ≤ 0.525, 301 to 369 K for $w_{\mathrm{net}} \geq 0.300$ MJ/kg, and 302 to 352 K for $q_{\mathrm{add}} \leq 0.47$ MJ/kg. Effects of these ranges on other performance parameters are listed in Table 2. Although the performance values of these parameters suffer some deteriorations outside the abovementioned ranges of T_1, yet, and except for BWR, their sensitivity towards T_1 is not too critical. Variations of BWR show steep losses with the values of T_1.

Sensitivity of cycle performance to maximum cycle temperature T_6

Compared to the almost unified range of T_1 discussed above that produce optimum values of all the performance parameters, Figure 6a,b,c,d,e,f,g shows that T_6 has drastically changed ranges depending on which performance parameter is to be optimized. Same optimum values of η_{I}, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} mentioned previously and listed in Table 2 require T_6 to be in the ranges

Table 4 Coefficient of least-square fitting of the data of each performance parameter with operating parameter

		Polynomial coefficients $y = \sum_{i=0}^{4} a_i x^i$					
		a_0	a_1	a_2	R^2	Range of y	Range of x
Effect of T_1 (K)	η_1 (%)	38.70743	0.14107	-3.71733×10^{-5}	0.977	47.88% to 26.5%	307 to 448 K
	η_{II} (%)	−10.80760	0.49830	-8.11052×10^{-5}	0.99	56.76% to 50.21%	307 to 448 K
	ECOP	0.10328	0.01473	-2.85902×10^{-5}	0.98	1.92 to 1.01	307 to 448 K
	x_{loss} (kJ/kg)	38.7074	0.14107	−0.0003	0.97	47.88 to 26.5 kJ/kg	307.448 K
	BWR	−3.98656	0.02608	−0.0000373	0.99	0.5234 to 0.5758	311 to 345 K
	w_{net} (kJ/kg)	4,098.474	22.73832	0.033916	0.95	285.2 to 326.5 kJ/kg	301 to 3,689 K
	q_{in} (kJ/kg)	−768.6663	6.51454	−0.008414	0.85	440.1 to 500.3 kJ/kg	304 to 417 K
Effect of T_6 (K)	η_1 (%)	19.9733	0.11057	5.18151×10^{-6}	0.88	40.64% to 47.88%	1,219 to 1,483 K
	η_{II} (%)	−135.8492	0.286828	-1.01925×10^{-4}	0.91	61.74% to 65.76%	1,213 to 1,483 K
	ECOP	−2.06871	4.66995×10^{-3}	-1.33033×10^{-6}	0.9	1.92 to 1.61	1,213 to 1,483 K
	x_{loss} (kJ/kg)	2.17010×10^{3}	−3.31852	1.33394×10^{-3}	0.92	107.7 to 177.4 kJ/kg	1,021 to 1,483 K
	BWR	4.54487	-6.027229×10^{-3}	2.23556×10^{-6}	0.91	0.485 to 0.521	1,219 to 1,472 K
	w_{net} (kJ/kg)	-1.35172×10^{3}	2.1247	-6.63125×10^{-4}	0.97	304.5 to 340.6 kJ/kg	1,338 to 1,483 K
	q_{in} (kJ/kg)	593.0112	−0.69715	4.44143×10^{-4}	0.93	340 to 499.2 kJ/kg	1,002 to 1,421 K
Effect of P_4 (MPa)	η_1 (%)	39.4442	3.78362×10^{-3}	-5.15619×10^{-7}	0.95	45.25% to 35.22%	0.75 to 8.32 MPa
	η_{II} (%)	-4.93274×10^{-8}	-4.9076×10^{-4}	66.37495	0.95	65.76% to 60.49%	0.86 to 7.57 MPa
	ECOP	1.88093	5.12392×10^{-5}	-1.81698×10^{-8}	0.99	1.92 to 1.62	0.86 to 5.53 MPa
	x_{loss} (kJ/kg)	-4.04347×10^{-7}	0.015887	79.08315	0.99	163.2 to 93.3 kJ/kg	0.85 to 6.43 MPa
	BWR	8.93573×10^{-9}	-3.58163×10^{-5}	0.51752	0.96	0.537 to 0.479	0.923 to 4.41 MPa
	w_{net} (kJ/kg)	9.38396×10^{-7}	−0.0142	358.3297	0.99	340.5 to 304.5 kJ/kg	1.44 to 7.57 MPa
	q_{in} (kJ/kg)	2.47949×10^{-6}	0.01457	405.67103	0.99	502.5 to 421.5 kJ/kg	0.86 to 3.94 MPa
Effect UA (kW/K)	**η1 (%)**	16.2101	2.39670	−0.04650	0.86	47.88% to 40.14%	13.61 to 37.04 kW/K
	ηII (%)	32.70171	2.35327	−0.04146	0.9	65.76% to 60.78%	16.79 to 37.4 kW/K
	ECOP	−0.03198	0.13002	-2.14243×10^{-3}	0.98	1.92 to 1.55	16.79 to 34.67 kW/K
	x_{loss} (kJ/kg)	0.10875	−5.16757	168.692	0.99	126.4 to 107.7 kJ/kg	15.8 to 37.04 kW/K
	BWR	4.35305×10^{-4}	−0.0201	0.68964	0.98	0.545 to 0.473	14.22 to 37.04 kW/K
	w_{net} (kJ/kg)	−0.23225	12.10661	178.35128	0.73	340.6 to 30.1 kJ/kg	13.6 to 29.56 kW/K
	q_{in} (kJ/kg)	0.33363	−16.06035	6.33.8598	0.94	399.2 to 440.1 kJ/kg	13.31 to 26.78 kW/K

1,220 to 1,480 K, 1,200 to 1,480 K, 1,220 to 1,480 K, 1,000 to 1,420 K, 1,220 to 1,480 K, 1,340 to 1,460 K, and 1,000 to 1,380 K, respectively. These values are generally expected since the higher is the T_6, the better η_I, η_{II}, ECOP, and w_{net}. Optimum values of the other two performance parameters, i.e., x_{loss} and q_{add}, necessitate that T_6 must be low in the range 1,000 to 1,380 K. In regards to sensitivity, and except for x_{loss} and q_{add} which are less sensitive to T_6, all other performance parameters exhibit great sensitivity to T_6, where their values greatly deteriorate outside the abovementioned optimum ranges of T_6. The wide ranges of T_6 mentioned above for optimum performance are in favor of the practical application of the cycle, which signifies that the cycle can accommodate any minor deterioration of the cycle components that are dependent on this high temperature. It is worthy to mention here that the material selection of the cycle components that are exposed to this high cycle temperature will put further restrictions and some adjustments to make these ranges practically appropriate.

Sensitivity of cycle performance to maximum cycle pressure P_4

The effects of maximum pressure P_4 on optimum performance are shown in Figure 7a,b,c,d,e,f,g. Optimum values of η_I, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} require P_4 to be in the ranges 0.75 to 7.57 MPa, 0.75 to 7.57 MPa, 0.864 to 4.49 MPa, 0.75 to 4.49 MPa, 0.864 to 4.03 MPa, 1.44 to 7.57 MPa, and 0.864 to 2.830 MPa, respectively. In contrast to T_6, the lower the P_4, the better the cycle is. Optimum exergy loss and heat added to the cycle necessitate that P_4 must be low (i.e., in the range of 0.864 to 2.830 MPa, Figure 7d,g) to result in less losses and less amount of heat added. Although all performance parameters show different degrees of sensitivity to the value of P_4, where they show some deterioration outside the abovementioned optimum ranges of the pressure, yet w_{net} has the least sensitivity. Although pressure values up to 12 MPa have been used in the MCM, the maximum value that results in optimum value of any of the performance parameters never exceeds 7.57 MPa, which is greatly in favor of practical applications of the cycle. Again, material selections of components that are exposed to this high pressure may have some limitations imposed by their stress requirement and pumping losses.

Sensitivity of cycle performance to heat exchanger's conductance UA

The heat exchanger's conductance, defined as the product of overall heat transfer coefficient and surface area of the heat exchanger $\left(UA = \dot{Q}_{add}/\Delta T_m\right)$, is considered an important operating/design parameter that is to be optimized based on the first law of thermodynamics and cost analysis. The selection of an optimum range for UA of heat exchangers is illustrated in Figure 8a,b,c,d. Optimum values of η_I, η_{II}, ECOP, x_{loss}, BWR, w_{net}, and q_{add} require UA to be in the ranges 13.6 to 37 kW/K, 16.8 to 37 kW/K, 20.7 to 34.7 kW/K, 14.8 to 37 kW/K, 14.2 to 34.7 kW/K, 16.6 to 29.6 kW/K, and 13.6 to 34.7 kW/K, respectively. All optimum cycle performance parameters require almost the same wide range of UA which is considered in favor of the cycle practical use. Although among the performance parameters, only ECOP and w_{net} show higher sensitivity with UA, where their values deteriorate very much outside their respective optimum ranges of UA, yet the non-sensitivity of the other performance parameters with UA is considered another positive point from a practical point of view.

Unified operating ranges for simultaneous optimum performance

Table 3 shows the unified ranges of the operating parameters that give simultaneous optimum performance (maximum η_I, η_{II}, ECOP, w_{net}, x_{loss}, BWR, q_{add}) for the cycle. Inspection of the ranges discussed in the above sections leads to the conclusion that there are some unified ranges of the operating parameters that simultaneously optimize all the performance parameters. These ranges are as follows: T_1 (302 to 315 K), T_6 (1,340 to 1,360 K), P_4 (1.440 to 2.830 MPa), and UA (20.7 to 29.6 kW/K). Although the unified ranges for both T_1 and T_6 are very narrow, which might represent some restrictions, the good design of the components of the cycle can cope with these narrow ranges.

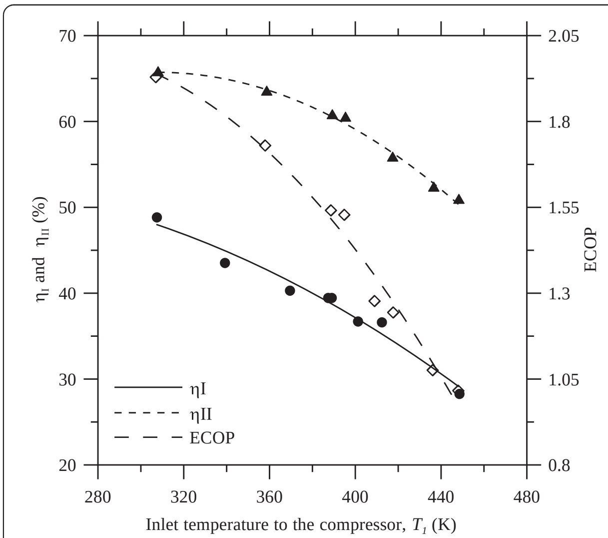

Figure 9 Optimum MCM results of η_1, η_{II}, and ECOP and their fitted equations w.r.t. T_1. With ranges of other operating parameter values as in Table 1.

Generalized optimal performance equations

From the MCM results and their representative figures discussed above, least-square fitting of the data of each performance parameter with each operating parameter, that only lie on the optimum envelop (i.e., maximum or minimum), gives the following equations together with their regression coefficients R^2 and the respective ranges of its application. These equations are displayed in Table 4. Figure 9 exemplifies one set of the fitted equations (i.e., optimal η_I, η_{II}, and ECOP versus T_1). Effects of T_1, T_6, P_4, and UA on each performance parameter is shown in Table 4.

The set of equations displayed in Table 4 can form a good basis for designing an optimal cycle, where the effect of each of the operating parameters on each of the performance parameters has been exclusively demonstrated in this mathematical form along with the applicable ranges of these two parameters regardless of the values of the other parameters. It is worthy to mention that the above equations are the result of a survey that concurrently covers all the practical ranges of the operating parameters, which can be easily understood to be the global optimal representation of the performance of the cycle. Also, the results discussed above are generally applicable to the cycle and are not restricted to some specific values of operating parameters or conditions of operation.

Conclusions

The present study has developed a general mathematical model to specify the performance as dependent on design and operating parameters of an irreversible gas turbine Brayton cycle incorporating two-stage compressor, two-stage gas turbine, intercooler, reheater, and regenerator with irreversibilities due to finite heat transfer rates and pressure drops. Ranges of operating parameters resulting in optimum performance (i.e., $\eta_I \geq 38\%$, $\eta_{II} \geq 60\%$, ECOP $\geq$ 1.65, $x_{loss} \leq 0.150$ MJ/kg, BWR $\leq$ 0.525, $w_{net} \geq 0.300$ MJ/kg, and $q_{add} \leq 0.470$ MJ/kg) are determined and discussed using the Monte Carlo method. These operating ranges are as follows: minimum cycle temperature ranges between 302 and 315 K, maximum cycle temperature ranges between 1,320 and 1360 K, maximum cycle pressure ranges between 1.449 and 2.830 MPa, and conductance of the heat exchanger ranges between 20.7 and 29.6 kW/K. The exclusive effect of each of the operating parameters on each of the performance parameters is mathematically given in a general sense that is applicable regardless of the values of the rest of the operating parameters and under any condition of operation of the cycle.

Notations

a surface area
BWR back work ratio
$\dot{C}$ heat capacity rate
C^* heat capacity ratio
ECOP ecological coefficient of performance
h enthalpy
$\dot{m}$ mass flow rate
NTU number of transfer units
P pressure
$\dot{Q}$ heat rate
q heat flux
R gas constant
s specific entropy
T temperature
UA conductance
$\dot{W}$ power
w work
x exergy
ε effectiveness
η efficiency

Subscripts

0 dead state
add high temperature heat addition
bur burner
c cold fluid, compressor
dest destruction
I first law
II second law
int intercooler
max maximum
reg regenerator
reh reheater
rej rejected
t turbine

Competing interests

The authors declare that they have no competing interests.

Authors' contributions

MMAAS conceived the concept and procedures of the present work, developed the model, carried out the analysis of the results, and wrote the manuscript. KKM checked the equations and analysis and reviewed the manuscript. YMAR developed the model, carried out the computations, and reviewed the manuscript. All authors read and approved the final manuscript.

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