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Research Article

Combined effect of variable parameters on the performance of gas turbine cycles

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ABSTRACT

The gas turbine cycle is the key solution for the power generation system from the last decade. several types of research are going to enhance the overall system efficiency of the gas turbine cycle. In the present study, six sets have been investigated parametrically for energy and exergy. Thermodynamic assessments have been effectively achieved for the compressor pressure ratio from 4 to 14, turbine inlet temperature (*TIT*) from 1000K to 1500K, and ambient temperature 25^{0}C to 45^{0}C. Results show that for every 100 rises in ambient temperature the maximum decrease in peak output and peak thermal efficiency is 6.2% and 5% respectively at 1000K and 3.6% and 1.9% respectively at 1500K. The exergy loss by exhaust gases of set 4 and set 5 increased by 107% whereas set 6 is decreased by 85.5% as compared to set 1 under the conditions when the exergy loss by exhaust gases of set 1 is minimum at *TIT* 1000K. The total exergy destruction of set 3, set 4, set 5, and set 6 is increased by 102%, 32.6%, 133.3%, and 102% respectively as compared to set 1 under the conditions when the total exergy destruction of set1 is minimum at *TIT* 1000K. Regenerative Gas Turbine cycle is the best in terms of the ratio of net output to exergy destruction of the plant as well as initial, operational, and maintenance costs.

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INTRODUCTION

Worldwide, gas turbine combined cycles are used in different applications specially to promote the efficiency of power stations. The emerging need for enhancing power generation system efficiency is becoming more and more crucial [1–4]. Gas turbine cycle is a dynamic internal combustion engine which is widely used for high power applications such as power generators and in propulsion systems due to its high power density. O. N. Favorskii et al. [5] reviewed the global experience related to using combined-cycle and gas-turbine technologies in power engineering. They focused on the technical and production capacities of the Russian industry constructing power

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machinery and equipment concerning its ability of producing up-to-day and enhanced domestically produced new-generation gas-turbine and combined-cycle units. A classic gas turbine generally comprises three elements: compressor, combustion chamber and turbine. The working fluid used in gas turbines is free and does not require pre-treatment at all but air is required to precool to enhance the cycle performance [6]. Guangya et al. [7] proposed and investigate two humidified gas turbine cycles. Simulated results show that both cycle have better performance as compared to simple gas turbine cycle. Mohanad [8] parametrically investigate the thermodynamic performance of gas turbine power plant with the application of intercooling, reheating and regeneration using MATLAB software. Results shows that efficiency increases with increase of effectiveness of intercooler and regenerator, isentropic efficiency of compressor and turbine, and reheat temperature but decreases with increase of ambient temperature.

Performance enhancement of gas turbine cycle and combined cycle power plant by various approaches of inlet air cooling conducted by several researchers [5,9–12] and found that lower the inlet temperature of air to the compressor by indirect mean of exhaust gases of gas turbine significantly affect the cycle net power output and thermal efficiency. Agustín M. Delgado-Torres [13] studied the impact of the cycle configuration on the accuracy of the findings with the perfect gas model and they recommended that the procedure, rule or temperature at which the constant isobaric heat capacity is assessed should be constantly specified if a perfect gas based methodology is used. Khan [14] investigate the performance of air bottoming combined cycle and regenerative gas turbine cycle operated by the partial amount of exhaust gases from gas turbine. This study presents the unique technique to compare the performance of these two cycles and prove that for thermal efficiency and exhaust gases exergy loss by regenerative gas turbine cycle is much better as compared to air bottoming cycle but for net power output air bottoming cycle is better than regenerative gas turbine cycle. Hasan and Yasin [15] studied parametrically steam injected gas turbine (STIG) process to perform an adequate optimization for pertinent and specification parameter, it is found that all relevant parameters environmental, fuel price investment and performance are strictly related on each other. Yongyi Li et al. [16] considered the backpressure adjustable gas turbine combined cycle (BAGTCC) as a promising procedure to enhance design performance by adjusting turbine backpressure in order to enhance the performance under part-load conditions.

There is a significant amount of literature related to gas turbine performance where investigators changed some thermodynamic parameters and studied its impact on the performances of the gas turbine. These operating conditions include ambient temperature, humidity, turbine inlet temperature, specific fuel consumption, compression ratio air to fuel ratio, component efficiency, ambient pressure, rate of heat supplied. Gas turbine performance is highly influenced by ambient temperature [17–20] and the turbine inlet temperature (TIT) [21–24]. It is widely recognized that gas turbine power declines significantly with rising ambient temperature. In fact, the variation of both temperature and humidity of ambient air have a significant impact on the plant performance due to obviously suctioned compressors in gas turbine systems.

Seong Won Moon [25] evaluated the performance improvement of a gas turbine combined cycle using coolant intercooling and coolant pre-cooling methods and found that when applying the coolant intercooling method the performance was greater. When the degree of coolant cooling rises the performance difference between the methods augments leading to a constant increase of the TIT and pressure ratio. Panyam et al. [26] studied the influence of the inlet air cooling on the gas turbine cycle performance using vapor compression inlet air cooling.

Comparative study of simple gas turbine cycle with its all possible combination and with air bottoming cycle was investigated by Alklaibi et al. [27]. M. Fallah [28] carried out an advanced exergy analyses for a simple gas turbine, gas turbine with evaporative inlet air cooler, STIG and steam injection gas turbine with evaporative inlet air cooler. The latter was found to be the most beneficial for the designer. The optimization priority order for the system constituents varies when determined with unconventional exergy analysis compared to conservative exergy analysis. HUANG Di et al. [29] proposed a design of Humid Air turbine cycle based on multi-shaft gas turbine specially the aero-derivative gas turbine. It was found that converting the aero-derivative gas turbine into Humid Air Turbine cycle reduces the surge margin, essentially because of the lower turbine inlet temperature. Hou et al. [30] proposed novel cogeneration system including a gas turbine, a supercritical CO2 (S-CO2) recompression cycle, a steam power cycle and an organic Rankine cycle. Regenerative gas turbine cycle, intercool gas turbine cycle and reheat gas turbine cycle increase the overall performance of the plant in term of cycle thermal efficiency as well as cycle net-work output. In all such configuration, heat exchanger plays a vital role in the cycle performance. The performance of heat exchanger depends on number of factor like design criteria, working fluid etc. [31–33]. Khan and Tlili [34] investigated the importance of heat exchanger in regenerative topping gas turbine and air bottoming cycle connecting through bypass valve. Study proves that by proper use of heat exchanger and bypass valve work output of combined cycle increases from 13.5 to 45% and combined cycle efficiency increases from 15% to 31%.

Since gas turbine operates with constant combustion, specific fuel consumption which is influenced by many factors is therefore considered an engineering challenge. Christina Salpingidou et al. [35] studied the performance of a helicopter engine in order to underline the significance of taking into consideration the demanded coolant mass which influences the thermal efficiency and the specific fuel consumption. Mohamed Mostafa et al. [36] analyzed numerically the performance of simple and regenerative gas turbine cycles at dry and wet compression conditions. It was concluded that if the recuperator joined with the wet compression and the overspray is used, the specific fuel consumption is decreased notably and the output power augments leading to an enhancement of the thermal efficiency.

From the above discussion, it is observed that researchers have done a lot of efforts to improve the performance of the gas turbine or/and combined cycle power plant. The major factors that affect the performance of gas turbine or/and combined cycle power plants are cycle combinations, selection of pressure ratio, turbine inlet temperature, compressor inlet temperature, etc. under different imposed conditions. It is well-known that there are several combinations in the gas turbine cycle which, enhances the cycle performance with respect to the simple gas turbine cycle. But in the literature, it is not found which gives the comparative energetic and exergetic performance of these basic different cycle combinations with a simple gas turbine cycle. Therefore, the present study covers the energetic and exergetic performance of six basic practical combinations of gas turbine cycle with a simple gas turbine cycle under the same imposed conditions and constraints.

DESCRIPTION OF GAS TURBINE CYCLES

This study covers six set of practical combinations of gas turbine cycles starting from the simple gas turbine cycle (set 1) to Intercool reheat regenerative cycle (set 6). The pressure ratio of the compressor or turbine (r_p) and turbine inlet temperature (*TIT*) in each cycle vary from 4 to 12 and 1000K to 1500K respectively. Another variable in all these cycles is the ambient temperature or air compressor inlet temperature which varies from 25^{0}C to 45^{0}C. The key parameter of the power plant cycle under study is tabulated in table 1.

Table 1. Key parameters of all sets under study [8,14,24]

Assumed parameters	
Parameter	**Values**
Isentropic efficiency of turbine	85%
Isentropic efficiency of Compressor	80%
Combustion Chamber Efficiency	100%
The mass flow rate of air	1g/s
The effectiveness of Heat Exchangers	0.75
Specific Heat of gases	1.14 kJ/kg-K
Specific Heat of air	1.005 kJ/kg-K
Specific Heat Ratio for air	1.4
Specific Heat Ratio for gases	1.33
Lower Calorific value of fuel	42000
Composition of fuel	Natural gas
Inlet pressure of air to the air compressor	1.013 bar

In simple gas turbine cycle and reheat gas turbine cycle, the compressed air from the compressor directly goes to the combustion chamber whereas in regenerative gas turbine cycle and intercool reheat regenerative gas turbine cycle, the compressed air from the single compressor and high pressure compressor respectively goes to the combustion chamber via heat exchanger where temperature of compressed air raises by gaining heat from the exhaust gases of the turbine but in intercool gas turbine cycle, the compressed air from the High pressure Compressor (HPC) directly goes to the combustion chamber. The combustible product from the combustion chamber at high pressure and high temperature directly enters the single gas turbine in simple gas turbine cycle, regenerative gas turbine cycle and intercool gas turbine cycle whereas in reheat gas turbine cycle and reheat regenerative gas turbine cycle, the combustible product from the combustion chamber first enters the High Pressure Turbine (HPT) and then enters the Low Pressure Turbine (LPT) via re-heater where the temperature of exhaust gases of HPT again raises to the temperature of combustible product from the combustion chamber in ideal case without pressure loss in the re-heater. The general energy and exergy analysis of all cycles are given in section 3. The net-work output of simple is the difference of turbine work and compressor work, this means that there three possibilities to increase the net output of the cycle. First one is decrease the compressor work without disturbing the turbine work and possible configuration for this is intercool cycle (set 3). The second one is increase the turbine work without disturbing the compressor work and the possible configuration for this is reheat cycle (set 4). And the third possibility of increasing the net out of the simple cycle is adopting the intercool-reheat cycle (set 5) in which turbine work increases and compressor work decreases simultaneously. The initial cost as well as maintenance cost of the plant of set 3, set 4 and set 5 is much high as compared to set 1. The thermal efficiency of the simple gas turbine cycle (set 1) is the ratio of net output of the cycle to the heat supplied in the combustion chamber. This means that all configurations used to increase the net output of the cycle, automatically increases the thermal efficiency of the cycle in the same proportion of net output of the cycle. The other possibility which is purely used to increase the thermal efficiency of the simple gas turbine cycle is decreasing the heat supplied by the combustion chamber without disturbing the compressor and turbine work. The regenerative gas turbine cycle (set 2) is the possible configuration for the action. The last but not the least possibility option for increasing the thermal efficiency of the cycle is simultaneous increases the net output of the cycle and decreases the heat supplied to the combustion chamber which is achieved by using the last configuration (set 6). This is practically very complex cycle and need lot of initial cost as well as proper and continuous expertise monitoring.

Energy and Exergy Analysis of Gas Turbine Cycles

Power required to run the compressor

$$(P_c) = \dot{m}_a(h_e - h_i) \tag{1}$$

Exit temperature of air from the air compressor

$$(T)_e = (T)_i.\left\{1+\frac{r_p^{\alpha}-1}{\eta_c}\right\} \tag{2}$$

where $\alpha = (\gamma - 1)/\gamma$

Rate of heat supplied by the combustion chamber

$$(Q) = \dot{m}_g.(h_g)_e - \dot{m}_a.(h_a)_i \tag{3}$$

Power delivered by the turbine

$$(P_t) = \dot{m}_g.[(h_g)_e - (h_g)_i] \tag{4}$$

Exit temperature of gases from the gas turbine

$$(T)_e = (T)_i.\left\{1 - \eta_t\left(1-r_p^{-\beta}\right)\right\} \tag{5}$$

where $\beta = (\gamma' - 1)/\gamma$

Net power output

$$(P_{net}) = P_t - P_c \tag{6}$$

Thermal Efficiency

$$(\eta_{th}) = \frac{P_{net}}{Q} \tag{7}$$

Mass flow rate of fuel in the combustion chamber

$$\dot{m}_f = \dot{m}_a.\left\{\frac{(h_g)_e - (h_a)_i}{\eta_{comb.}.(LCV)-(h_g)_e}\right\} \tag{8}$$

Specific fuel consumption

$$(SFC) = 3600.\frac{\dot{m}_f}{P_{net}} \tag{9}$$

Effectiveness of the heat exchanger

$$(\varepsilon) = \frac{\dot{m}_a.\left[(h_a)_e - (h_a)_i\right]}{\dot{m}_g.(h_g)_e - \dot{m}_a.(h_a)_i} \tag{10}$$

The exergy destruction in the compressor can be written

$$(E_D)_C = m_a.T_{ref}.(s_{ae} - s_{ai}) \tag{11}$$

The exergy destruction in combustion chamber is given by

$$\begin{aligned}(E_D)_{comb.} = T_{ref.}.\Bigg[&\left(m_g.C_{pg}.\ln\left(\frac{T_{ge}}{T_{ref}}\right) - m_g.R_g.\ln\left(\frac{P_{ge}}{P_{ref}}\right)\right)\\ &-\left(m_a.C_{pa}.\ln\left(\frac{T_{ai}}{T_{ref}}\right) - m_a.R_a.\ln\left(\frac{P_{ai}}{P_{ref}}\right)\right)\\ &+\Delta s_{ref}\Bigg]\end{aligned} \tag{12}$$

The exergy destruction in turbine is given by

$$(E_D)_t = m_g.T_{ref}.(s_{ae} - s_{ai}) \tag{13}$$

The exergy destruction in HRSG is given by

$$(E_D)_{HRSG} = T_{ref}[m_s.(s_{se} - s_{si}) + m_g.(s_{ge} - s_{gi})] \tag{14}$$

Exergy loss via exhaust gases

$$(E)_{Exh.} = \int_{(T_g)_e}^{T_{ref}}\left(1-\frac{T_{ref}}{T}\right).dQ \tag{15}$$

The Carnot efficiency may be defined as

$$\eta_{Carnot} = \left(1-\frac{T_{ref.}}{TIT}\right) \tag{16}$$

Exergy destruction of plant

$$(E_D)_{Plant} = \sum_{i=1}^{n}(E_D)_n \tag{17}$$

Where "n" is total number of components in the plant.

RESULTS AND DISCUSSION

The net output, thermal efficiency, specific fuel consumption (SFC) and exergy loss by exhaust gases of Simple gas turbine cycle (Set-1), Regenerative gas turbine cycle (Set-2), Intercool gas turbine cycle (Set-3), Re-heat gas turbine cycle (Set-4), Intercool re-heat gas turbine cycle (Set-5) and Intercool Re-heat Regenerative gas turbine cycle (Set-6) are investigated parametrically by commercial Engineering Equation Solver (EES) software based on the assumptions mention in table 1 with variation of pressure ratio, ambient temperature and turbine inlet temperature.

The schematic and T-s diagram of a simple gas turbine cycle is shown in fig.1. the Iso-efficiency curves and

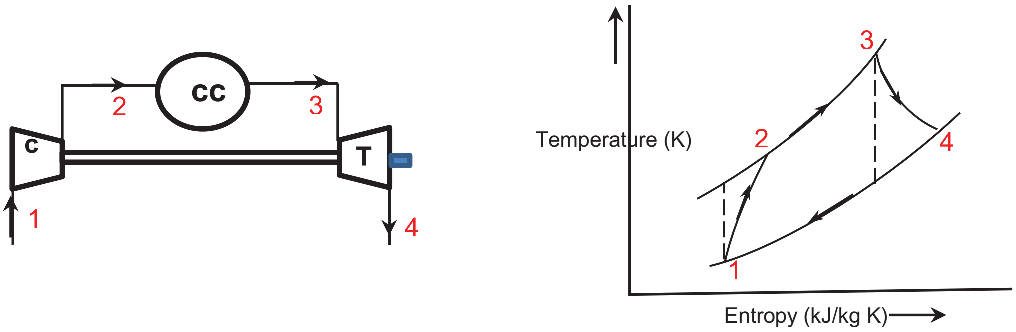

Figure 1. Schematic diagram and T-s digram of Set 1 (Simple Gas Turbine cycle).

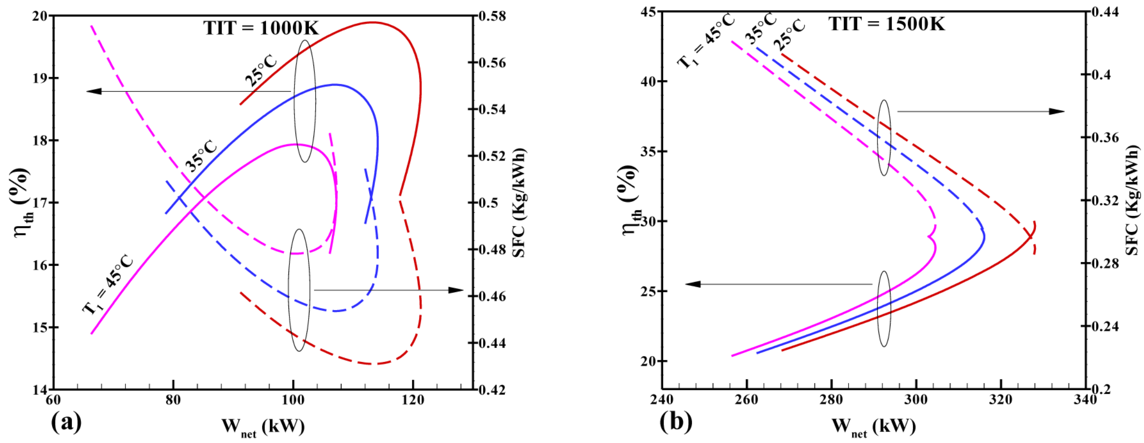

Figure 2. Iso-efficiency and Iso-SFC curves of set 1 at (a) TIT 1000K & (b) 1500K.

Iso-SFC curves of this cycle are shown in fig.2(a) and 2(b) at TIT 1000K and 1500K respectively. It is found that with the increase of ambient temperature the net output and thermal efficiency of the cycle decrease because the compressor work increases with an increase of ambient temperature which results in a decrease in net output as well as thermal efficiency of the cycle. Due to a decrease in net output and thermal efficiency of the cycle, the SFC of the cycle increases according to equation (9). Also an increase of turbine inlet temperature, the net output, and thermal efficiency of the cycle increases which results in a decrease in SFC. The variation of net output and SFC w.r.t pressure ratio and thermal efficiency Vs net output is plotted at TIT 1000K and 1500K in fig. 3(a) and 3(b) respectively. It is noted from these figures that at r_p=5.5 and r_p=11, the net output is maximum when TIT 1000K and 1500K respectively whereas the SFC is minimum at r_p=8 and r_p=12 when TIT 1000K and 1500K respectively for T_1=25^{0}C. Also with every 10^{0}C rise of ambient temperature, the peak value of net output approximately decreased by 6%, the peak value of thermal efficiency decreased by 5% and the minimum value of SFC approximately increased by 5%, at TIT 1000K, whereas, the peak value of net output decreases by 3.6%, the peak value of thermal efficiency decreases by 1.8% and the value of minimum SFC increases by approximately 1.75% at TIT 1500K.

The regenerative gas turbine cycle is shown in fig.4 with Iso-efficiency curves and Iso-SFC curves of at TIT 1000K and 1500K in fig.5(a) and 5(b) respectively. From these figures, it is seen that the thermal efficiency and net output of the cycle decreases and SFC increase with an increase in ambient temperature and vice versa with the increase of turbine inlet temperature. Figure 6(a) and 6(b) show the variation of net output and SFC w.r.t pressure ratio at TIT 1000K and 1500K respectively. From these figures that at r_p=5.5 and r_p=11, the net output is maximum when TIT 1000K and 1500K respectively whereas the SFC is minimum at r_p=8 and r_p=12 when TIT 1000K and 1500K respectively for T_1=25^{0}C.

For every 10^{0}C rise of ambient temperature, the peak value of net output and thermal efficiency decrease by 6% and 5% respectively and the minimum value of SFC increase by approximately 4% at TIT 1000K, whereas the peak value of net output and thermal efficiency decrease by 3.6% and 1.9% respectively and the minimum value of SFC increase by approximately 2% at TIT 1500K. As compared

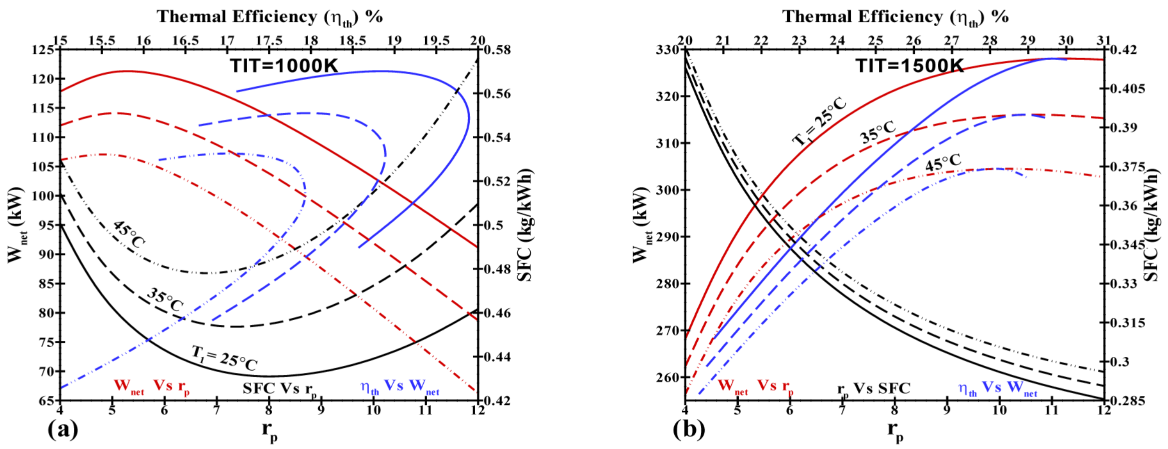

Figure 3. Variation of thermal efficiency, net output & SFC of set 1 w.r.t pressure ratio.

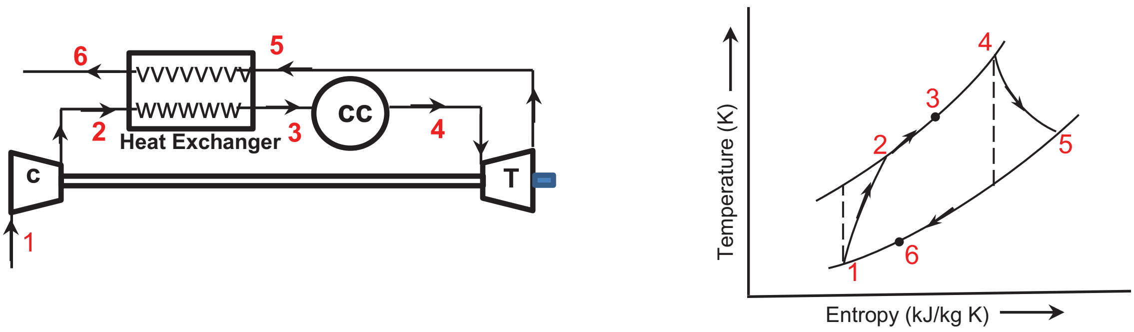

Figure 4. Schematic diagram and T-s digram of set 2 (Regenerative Gas Turbine cycle).

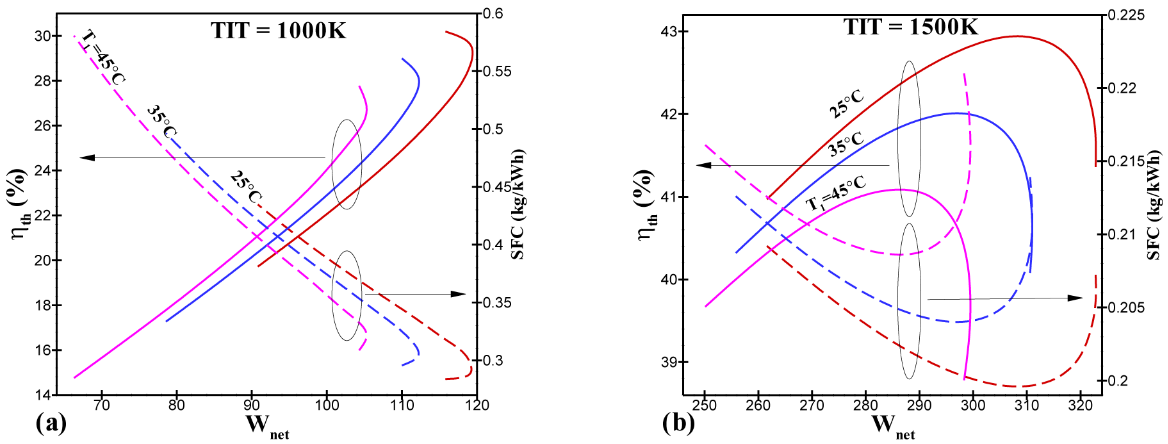

Figure 5. Iso-Efficiency and Iso-SFC curves of set 2 at (a) TIT 1000K & (b) 1500K.

as simple gas turbine cycle, the peak value of net output, the peak value of thermal efficiency and minimum value of SFC of regenerative gas turbine cycle is decreased by 1.6%, increased by 54.8 to 51.7%, and decreased by 55 to 52% respectively at TIT 1000K whereas at TIT 1500K, the peak value of net output, the peak value of thermal efficiency and minimum value of SFC of regenerative gas turbine cycle is decreased by 1.6%, increased by 41.8 to 107%, and decreased by 41.8 to 43% respectively.

The intercool gas turbine cycle is shown in fig.7 with Iso-efficiency curves and Iso-SFC curves of at TIT 1000K and 1500K in fig.8(a) and 8(b) respectively. From these figures, it is seen that the thermal efficiency and net output of the cycle decrease and SFC increases with an increases

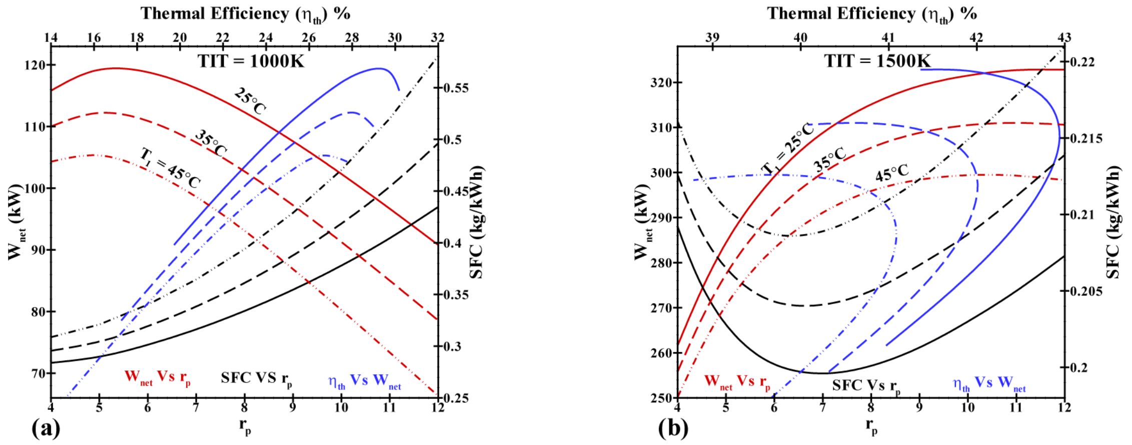

Figure 6. Variation of thermal efficiency, net output & SFC of set 2 cycle w.r.t pressure ratio and ambient temperature at (a) TIT 1000K & (b) 1500K.

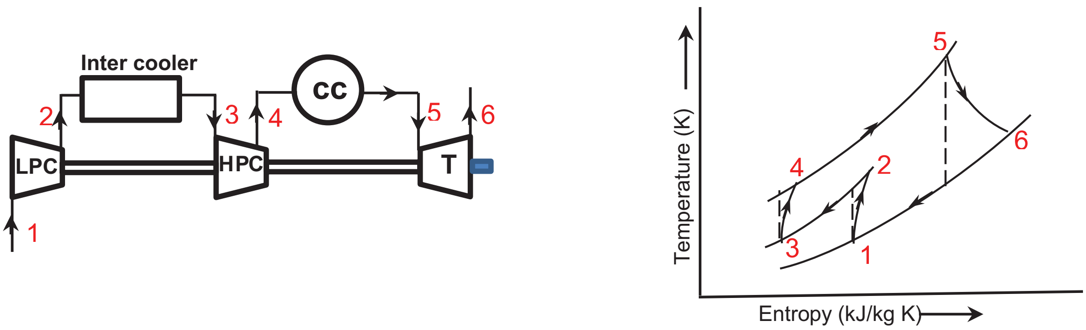

Figure 7. Schematic diagram and T-s digram of set 3 (Intercool gas turbine cycle).

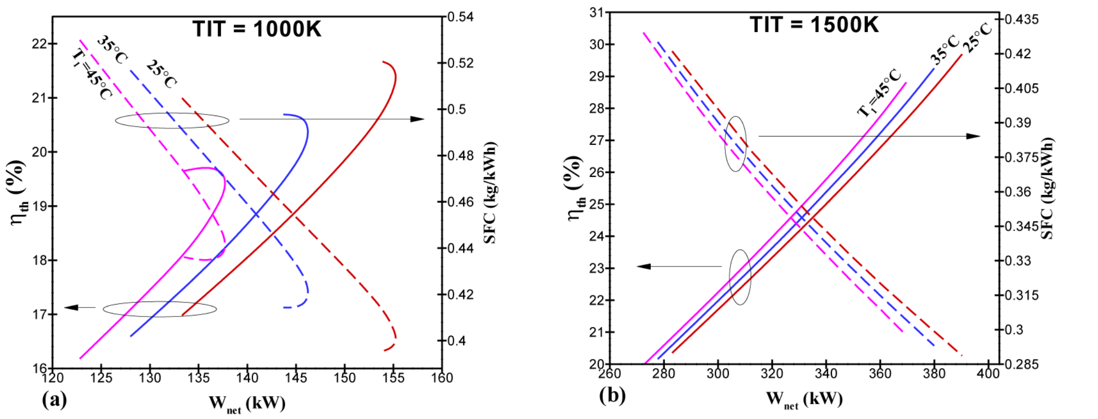

Figure 8. Iso-efficiency and Iso-SFC curves of set 3 at (a) *TIT* 1000K & (b) 1500K.

in ambient temperature and vice versa with the increase of turbine inlet temperature. Figure 9(a) and 9(b) shows the variation of net output and SFC w.r.t pressure ratio and thermal efficiency Vs net output at TIT 1000K and 1500K respectively. From these figures that at r_p=10 and r_p=12, the net output is maximum at TIT 1000K and 1500K respectively whereas the SFC is minimum at r_p=12 and TIT 1000K and 1500K when ambient temperature 25^{0}C.

For every 10^{0}C rise of ambient temperature, the peak value of net output, the peak value of thermal efficiency and the minimum value of SFC decreases by 5.8%, 4.75% and 2.5 % respectively at TIT 1000K. At TIT 1500, the peak value of net output, the peak value of thermal efficiency and the minimum value of SFC decreased by 2.64%, 1.5% and 1.5% respectively. As compared as simple gas turbine cycle, the peak value of net output, the peak value of thermal

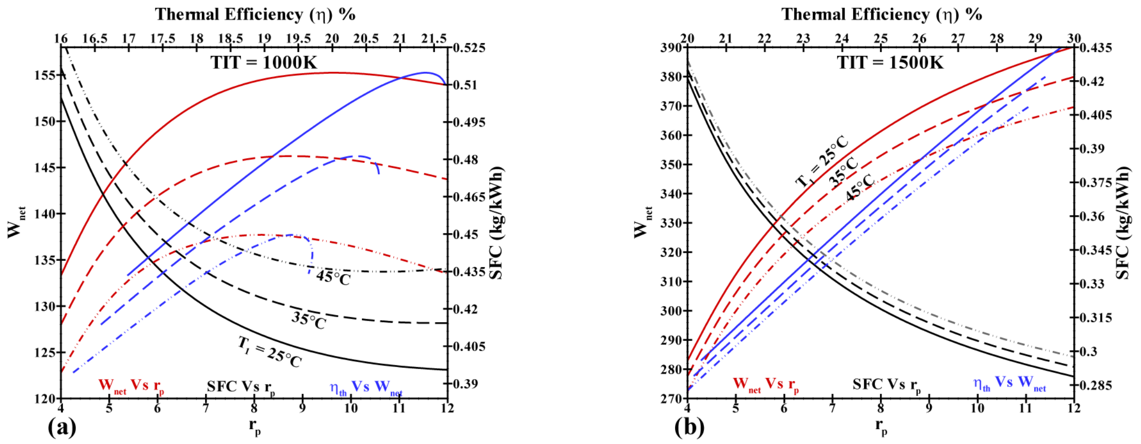

Figure 9. Variation of thermal efficiency, net output & SFC of set 3 w.r.t pressure ratio and ambient temperature at (a) *TIT* 1000K & (b) 1500K.

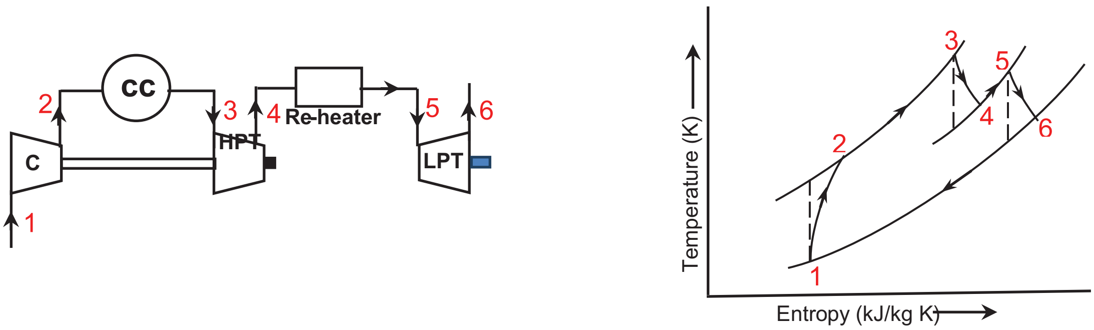

Figure10. Schematic diagram and T-s digram of set 4 (Re-heat gas turbine cycle).

efficiency and minimum value of SFC of intercool gas turbine cycle is increased by 28%, increased by 9.8 to 8.6%, and decreased by 8.2 to 9% respectively at TIT 1000K whereas at TIT 1500K, the peak value of net output, the peak value of thermal efficiency and minimum value of SFC is increased by 15.9 to 21.3%, increased by -0.5 to 43%, and increased by 1% respectively.

The reheat gas turbine cycle is shown in fig.10 with Iso-efficiency curves and Iso-SFC curves of at TIT 1000K and 1500K in fig.11(a) and 11b(b) respectively. From these figures, it has been seen that the thermal efficiency and the net output of the cycle decreases, and SFC increase with an increase in ambient temperature and vice versa with the increase of turbine inlet temperature. Figure 12(a) and 12(b) show the variation of net output and SFC w.r.t pressure ratio and thermal efficiency Vs net output respectively at TIT 1000K and 1500K. From these figures it is observed that the net output is maximum at r_p=9 when TIT 1000K and r_p=12 when TIT 1500K respectively whereas the SFC is minimum at r_p=12 for both TIT 1000K and TIT 1500K when ambient temperature 25^{0}C. For every 10^{0}C rise of ambient temperature, the peak value of net output and peak value of thermal efficiency decreased by 6.2%, and 4.7% respectively. The minimum value of SFC increased by approximately 4.9% at TIT 1000K whereas the peak value of net output and peak value of thermal efficiency decreased by 3%, and 1.39% to 1.13% respectively. The minimum value of SFC increased by approximately 1.45% at TIT 1000K**.**

As compared as simple gas turbine cycle, the peak value of net output, the peak value of thermal efficiency and minimum value of SFC of reheated gas turbine cycle is increased by 34%, increased by 5.1 to 6%, and decreased by 5 to 6% respectively at TIT 1000K whereas at TIT 1500K, the peak value of net output, the peak value of thermal efficiency and minimum value is increased by 33.5 to 31.6%, decreased by 3.6 to 4.4%, and increased by 3.8 to 4.6% respectively.

The intercool Reheated gas turbine cycle is shown in fig 13 with Iso-efficiency curves and Iso-SFC curves of at TIT 1000K and 1500K in fig.14(a) and 14(b) respectively. From these figures it has been seen that the thermal efficiency and the net output of the cycle decreases and SFC increases with increases of ambient temperature and increases with increase of turbine inlet temperature. Figure 15(a) and 15(b) shows the variation of net output and SFC w.r.t pressure

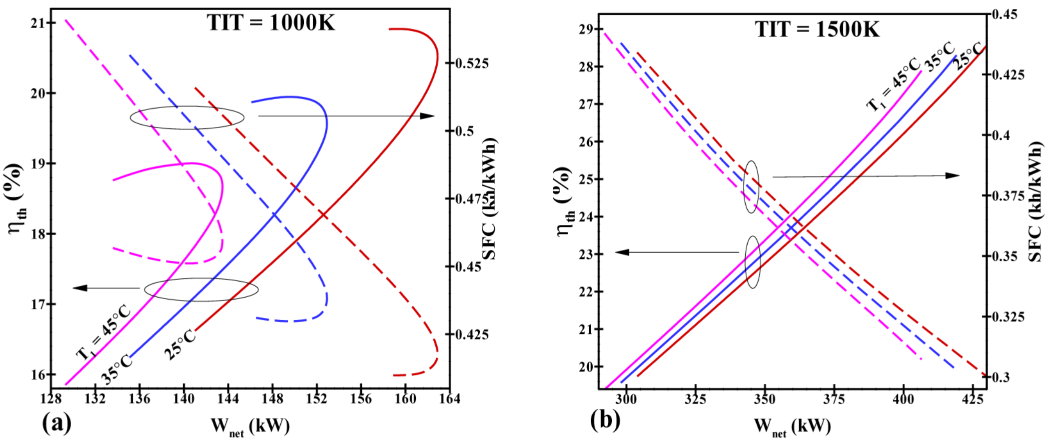

Figure 11. Iso-efficiency and Iso-SFC curves of set 4 at (a) TIT 1000K & (b) 1500K.

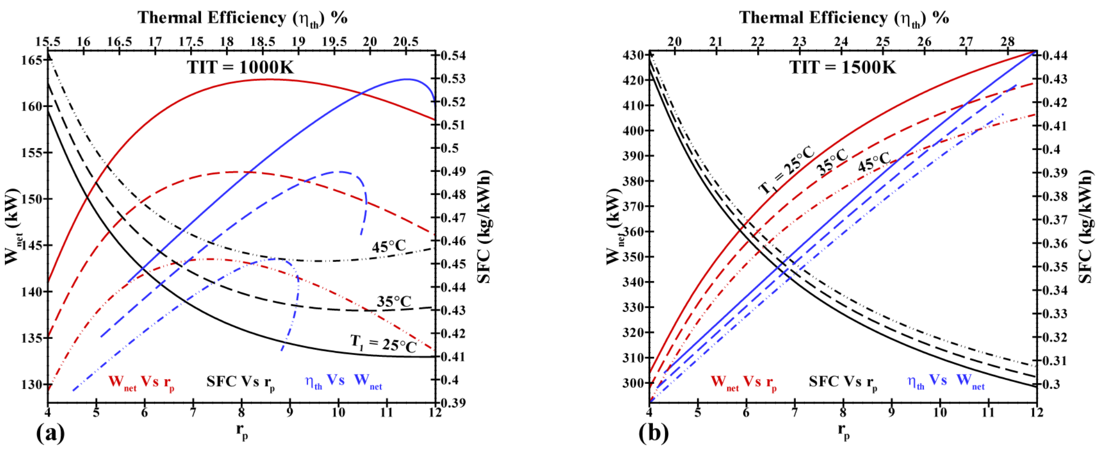

Figure 12. Variation of thermal efficiency, net output & SFC of set 4 w.r.t pressure ratio and ambient temperature at (a) TIT 1000K & (b) 1500K.

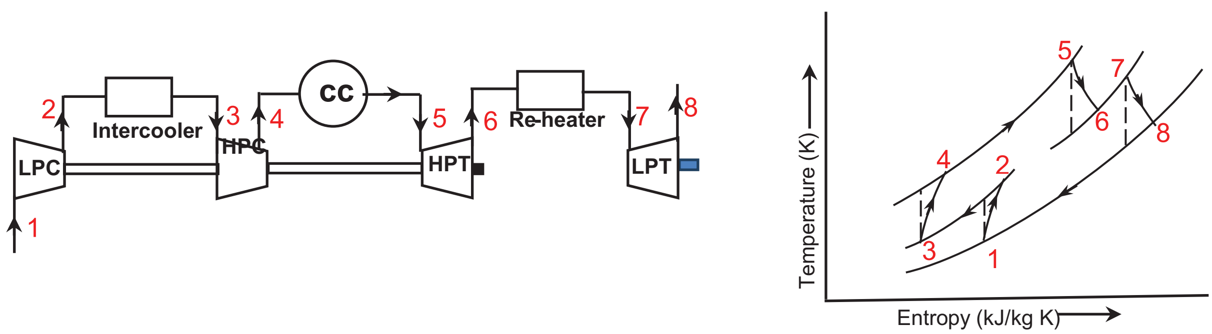

Figure 13. Schematic diagram and T-s digram of set 5 (Intercool Re-heat gas turbine cycle).

ratio and thermal efficiency Vs net output at TIT 1000K and 1500K respectively. From these figures that the peak value of output, peak value of thermal efficiency and minimum value of SFC is found at r_p=12 for both TIT 1000K and 1500K when ambient temperature T_1=25^{0}C.

For every 10^{0}C rise of ambient temperature, the peak value of net output decreased by 4.5% to 4.8%, the peak value of thermal efficiency decreased by 3% to 3.2% and minimum value of SFC increased by approximately 3.2% at TIT 1000K and the peak value of net output decreased by 2%, the peak value of thermal efficiency decreased by 1.1% and minimum value of SFC increased by approximately 1.2% at TIT 1500K**.** As compared as simple gas turbine cycle, the peak value of net output, the peak value of

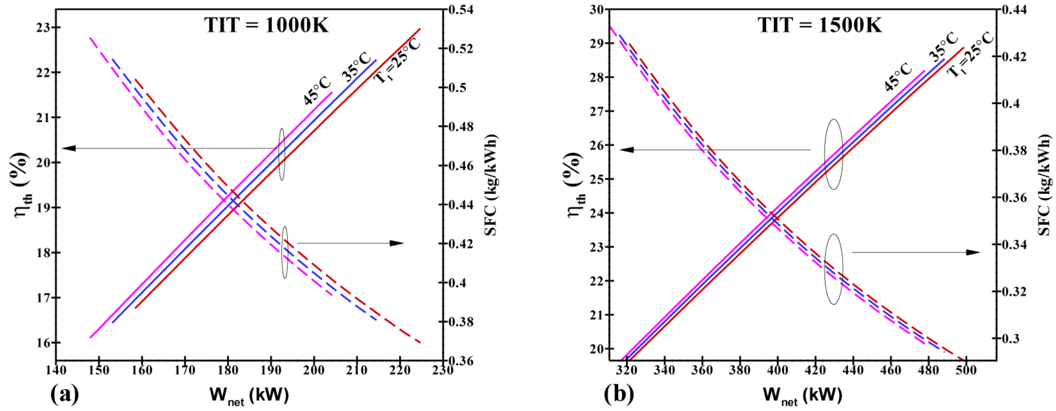

Figure 14. Iso-efficiency and Iso-SFC curves of set 5 at (a) TIT 1000K & (b) 1500K.

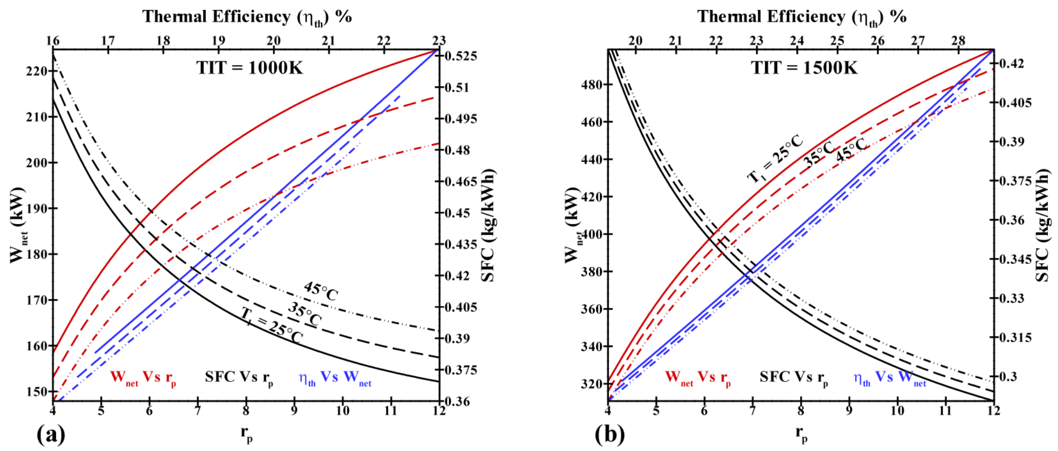

Figure 15. Variation of thermal efficiency, net output & SFC of set 5 w.r.t pressure ratio and ambient temperature at (a) TIT 1000K & (b) 1500K.

thermal efficiency and minimum value of SFC of intercool reheated gas turbine cycle is increased by 85.5 to 90.8%, increased by 15.5 to 20.2%, and decreased by 14.3 to 17.7% respectively at TIT 1000K whereas at TIT 1500K, the peak value of net output, the peak value of thermal efficiency and minimum value is increased by 52 to 57%, decreased by 2.6 to 3.8%, and increased by 0.5 to 1.7% respectively.

The intercool reheated regenerative gas turbine cycle is shown in fig.16 with Iso-efficiency curves and Iso-SFC curves of at TIT 1000K and 1500K in in fig.17(a) and 17(b) respectively. From these figures it has been seen that the thermal efficiency and the net output of the cycle decreases and SFC increases with increases of ambient temperature and vice versa with the increase of turbine inlet temperature. Figure 18(a) and 18(b) show the variation of net output and SFC w.r.t pressure ratio and thermal efficiency Vs net output at TIT 1000K and 1500K respectively. From these figures that the max output, maximum efficiency and minimum SFC is found at r_p=12 for both TIT 1000K and 1500K when ambient temperature 25^{0}C. For every 10^{0}C rise of ambient temperature, the peak value of net output decreased by 4.5% to 4.8%, the peak value of thermal efficiency decreased by 3.5% and minimum value of SFC increased by approximately 3.6% at TIT 1000K and the peak value of net output decreased by 2%, peak value of thermal efficiency decreased by 1.7% and minimum value of SFC increased by approximately 1.7% at TIT 1500K**.**

As compared as simple gas turbine cycle, the peak value of net output, the peak value of thermal efficiency and minimum value of SFC of intercool reheated regenerative gas turbine cycle is increased by 82.1 to 87.3%, increased by 77.4 to 83.3%, and decreased by 43.6 to 45.5% respectively at TIT 1000K whereas at TIT 1500K, the peak value of net output, the peak value of thermal efficiency and minimum value is increased by 48.2 to 53%, increased by 60%, and decreased by 37.5% respectively.

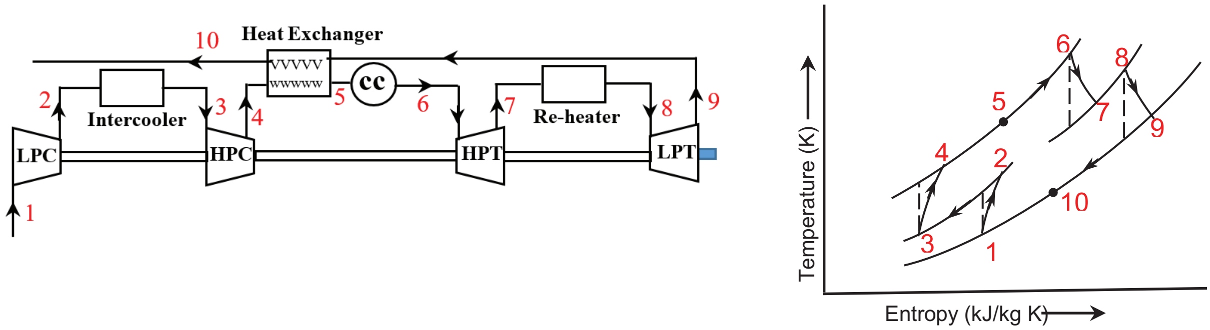

Figure 16. Schematic diagram and T-s digram of set 6 (Intercool Re-heat regenerative gas turbine cycle).

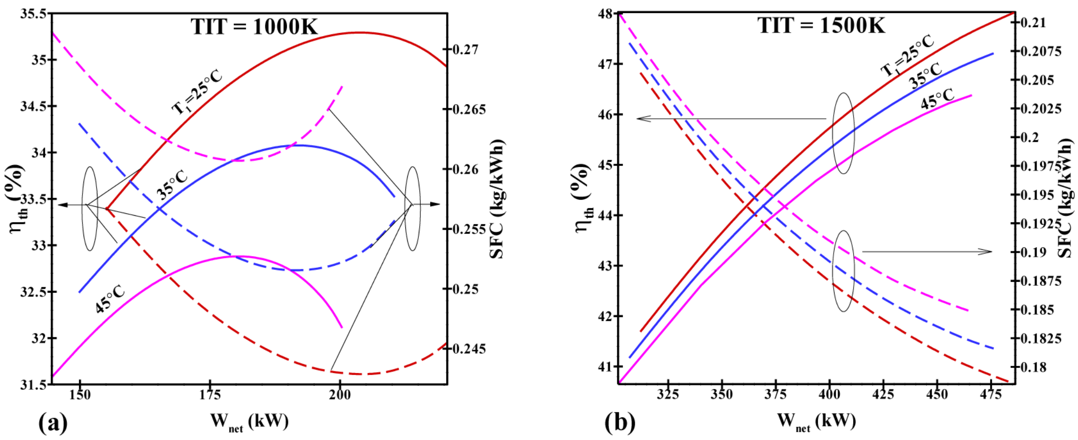

Figure 17. Iso-efficiency and Iso-SFC curves of set 6 at (a) *TIT* 1000K & (b) 1500K.

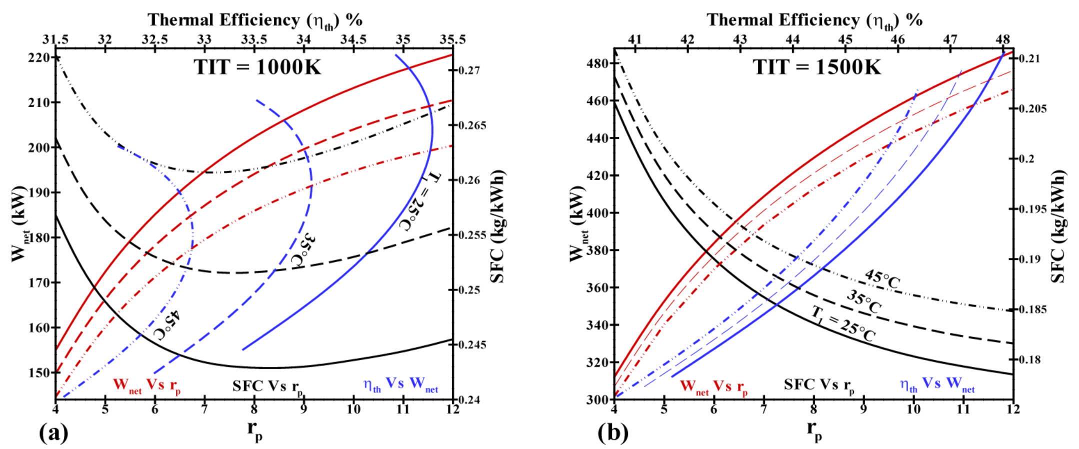

Figure 18. Variation of thermal efficiency, net output & SFC of set 6 w.r.t pressure ratio and ambient temperature at (a) *TIT* 1000K & (b) 1500K.

The exergy loss by exhaust gases Vs pressure ratio for set1 to set 6 is plotted for different values of ambient temperature is shown is fig. 19 (a) and 19 (b) for TIT 1000K and 1500K respectively. It is noted that at the exergy loss of exhaust gases in set 2 and set 6 increases with increase of pressure ratio, ambient temperature and decreases with increase of TIT whereas in set 1, set 3, set 4 and set 5 the exergy loss of exhaust gases decreases with increase of pressure ratio, ambient temperature and increases with increase of TIT. Exergy destruction of the plant is the summation of exergy destruction of different components of that plant. The exergy destruction of the plant Vs pressure ratio for

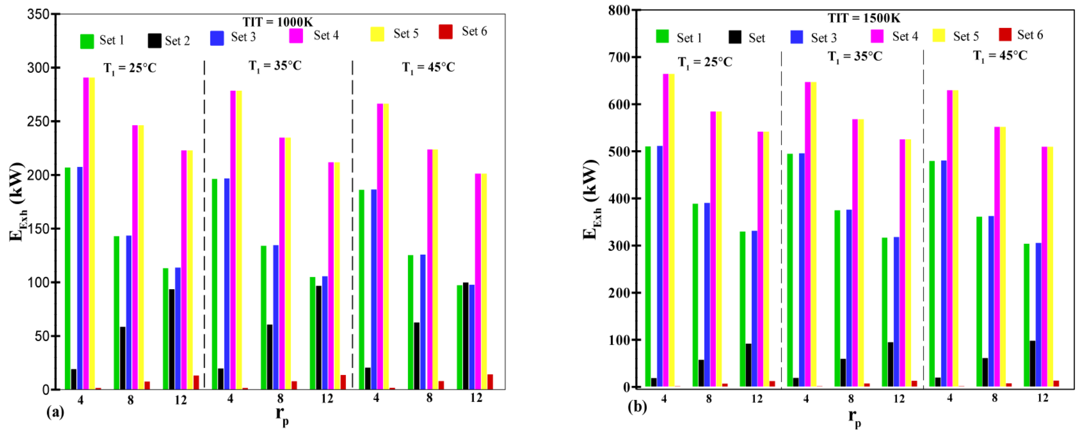

Figure 19. Variation in exergy loss by exhaust gases of selected sets (1 to 6) w.r.t pressure ratio and ambient temperature at (a) *TIT* 1000K & (b) 1500K.

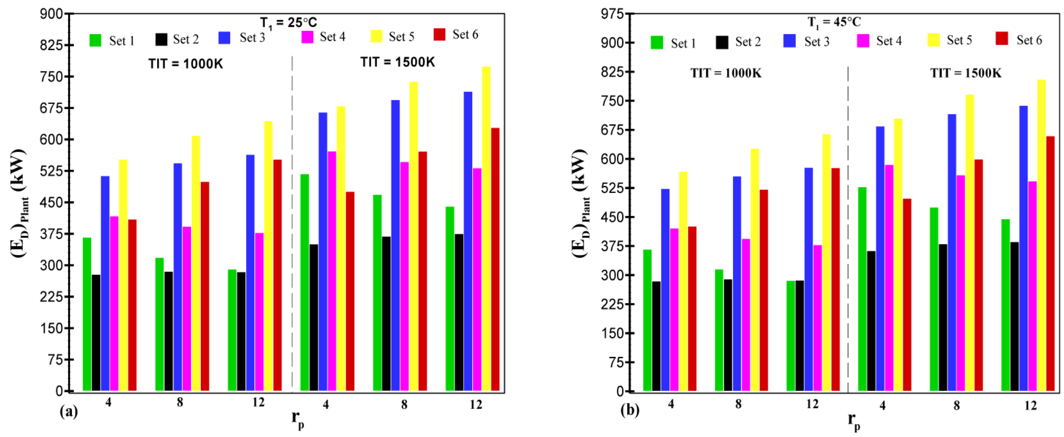

Figure 20. Variation in exergy destruction of plant of selected sets (1 to 6) w.r.t pressure ratio and ambient temperature at (a) TIT 1000K & (b) 1500K.

set1 to set 6 is plotted at different values of ambient temperature is shown is fig. 20 (a) and 20 (b) for TIT 1000K and 1500K respectively.

From these figures it is observed that set 5 and set 2 has maximum and minimum exergy destruction of the plant respectively for any value of ambient temperature and turbine inlet temperature (TIT). Exergy destruction of all Sets is least effected by pressure ratio and ambient temperature and maximum effected by TIT.

Second Law of efficiency Vs pressure ratio for set1 to set 6 is plotted for different values of ambient temperature is shown is fig. 21 for TIT 1000K and fig. 22 for TIT 1500K. It is noted that is almost same for set 1, set 3, set 4 and set 5 at r_p=4 for both TIT 1000K and 1500K for all values of ambient temperature. For all set under study set 6 have maximum Second Law of efficiency for all values of pressure ratio, ambient temperature and TIT. Second law of efficiency for set 2 at TIT 1000K decreases with increase of pressure ratio but at TIT 1500K second law of efficiency first increases then decreases after attending its peak value with increase of pressure ratio.

From the above discussion, it is found that as the net output and thermal efficiency of the plant increases the number of components of the plant increases, but as the number of components increases the exergy destruction of the plant also increases. The optimum performance of the plant is achieving when the net output & thermal efficiency reaches its maximum and exergy loss of exhaust gases & total exergy destruction is minimum. The variation of ratio of net output to exergy destruction of the plant Vs pressure ratio for different values of ambient temperature at TIT 1000K and 1500K is shown in fig. 23 (a) and 23 (b).

The plant which having maximum value of this ratio have maximum performance. It is noted in general from figures that the performance of each set first increases and then decreases after achieving its peak value with the increase of

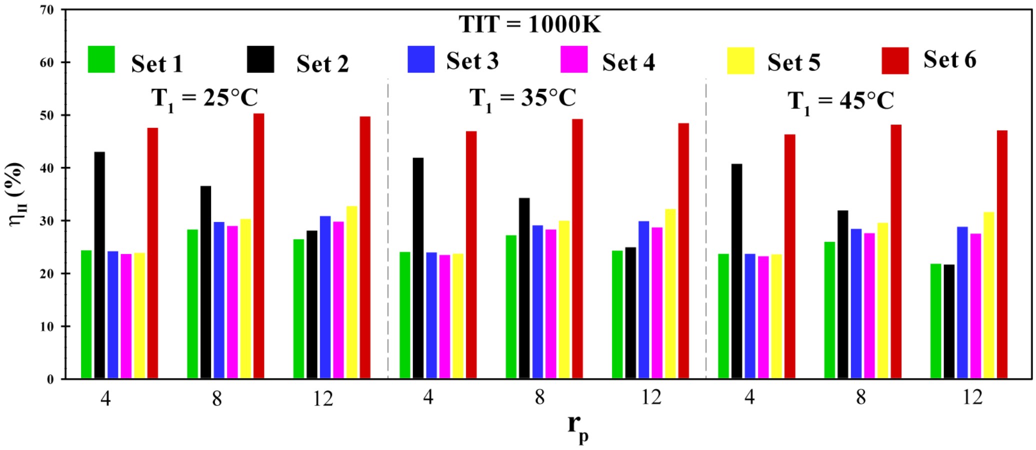

Figure 21. Variation in second law efficiency of selected Sets (1 to 6) w.r.t pressure ratio and ambient temperature at TIT 1000K.

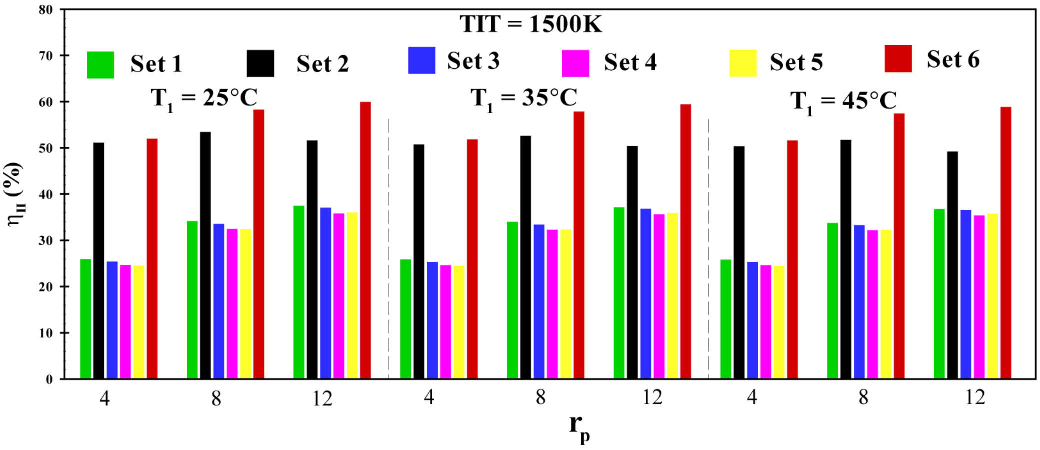

Figure 22. Variation in second law efficiency of selected sets (1 to 6) w.r.t pressure ratio and ambient temperature at *TIT* 1500K.

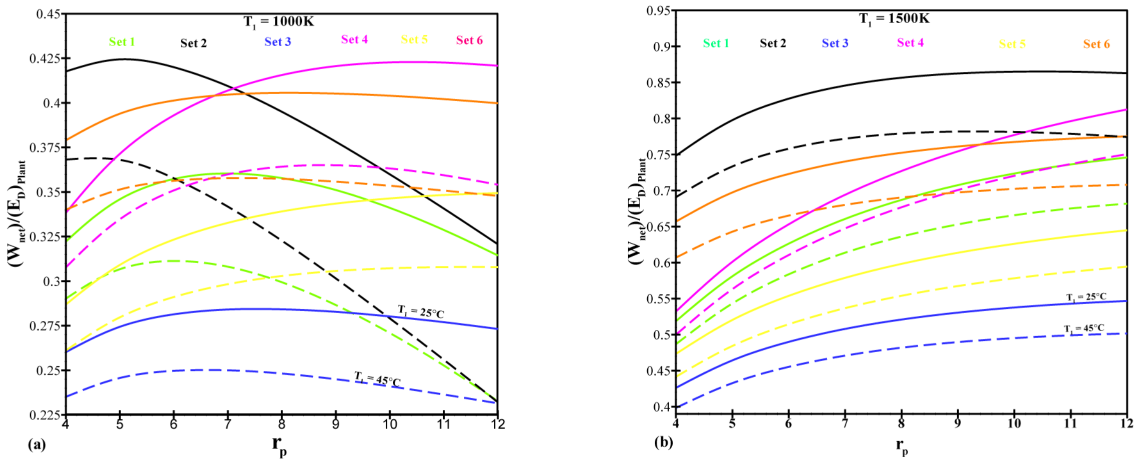

Figure 23. Variation of $(W_{net})/(E_D)_{Plant}$ of considered all sets w.r.t r_p for (a) TIT 1000K & (b) 1500K.

pressure ratio. It is also found that the performance of each cycle decreases with increase of ambient temperature and increases with increase of TIT. For TIT 1000K, the performance of set 2 and set 4 at r_p=7.1, set 1 and set 5 at r_p=9.6 and set 2 & set 5 at r_p=10.7 is same at T_1=25^{0}C whereas the performance of set 2 and set 4 at r_p=6.3, set 4 and set 6 at r_p=6.7, set 1 and set 5 at r_p=7.7, set 2 and set 5 at r_p=8.8 and set 1 set 2 and set 3 at r_p=12 is same when T_1=45^{0}C. For TIT 1500K, the performance of set 4 and set 6 at r_p=8.75 is same when T_1=45^{0}C.

CONCLUSIONS

In present study, six selected configurations have been investigated parametrically for energy and exergy. Thermodynamic assessments have been effectively achieved for the compressor pressure ratio from 4 to 14 and TIT from 1000K to 1500K. Based on the thermodynamic analyses, the following are the main conclusions listed below:

- From the above discussion it is noted that with the increase of ambient temperature, the work output and thermal efficiency decreases in all consider sets.
- For every 10^{0}C rise in ambient temperature, the maximum decrease in peak work output is noted in set 4 which is 6.2% followed by set 5 and set 6 which is 4.5% at *TIT* 1000K whereas the maximum decrease in peak work output is noted in set 1 and set 2 which is 3.6% followed by set 5 and set 6 at *TIT* 1500K.
- For every 10^{0}C rise in ambient temperature, the maximum decrease in peak thermal efficiency is noted in set 1 and set 2 which is almost 5% at *TIT* 1000K whereas the maximum decrease in peak thermal efficiency is noted in set 2 only which is 1.9% followed by other sets at *TIT* 1500K. The minimum decrease in peak thermal efficiency is noted in set 5 for both *TIT* 1000K and 1500K.
- For every 10^{0}C rise in ambient temperature, the maximum and minimum rise in lowest value of SFC is observed in set 3 which is 5.3% and set 5 which is 3.2% respectively at *TIT* 1000K but at *TIT* 1500K, the maximum rise in lowest value of SFC is observed in set 2 which is 1.2%.
- The maximum output decreases in set 2 whereas increases in set 3, set 4, set 5, and set 6 under the conditions when set 1 achieve its peak value of output.
- The thermal efficiency of set 2, set 3, set 4, set 5 and set 6 is more as compared to set 1 at *TIT* 1000K whereas the thermal efficiency of set 2, set 3, and set 6 is more and set 4, set 5 is less as compared to set 1 at *TIT* 1500K under the conditions of set 1 to attend its peak thermal efficiency.
- The SFC of set 2, set 3, set 4, set 5 and set 6 is less as compared to set 1 at *TIT* 1000K whereas the SFC of set 2 and set 6 is less and set 3, set 4, set 5 is more as compared to set 1 at *TIT* 1500K under the conditions where set 1 attend its least value of SFC.
- The exergy loss by exhaust gases of set 4 and set 5 increased by 107% whereas set 6 is decreased by 85.5% as compared to set 1 under the conditions when the exergy loss by exhaust gases of set 1 is minimum at *TIT* 1000K.
- The total exergy destruction of set 3, set 4, set 5, and set 6 is increased by 102%, 32.6%, 133.3%, and 102% respectively as compared to set 1 under the conditions when the total exergy destruction of set1 is minimum at *TIT* 1000K.

From the above, it is concluded that although the net output and thermal efficiency of set 6 is maximum but, the total exergy destruction is also maximum as compared to other consider sets because the total number of components in set 6 is highest as compared to other consider sets. This increase in total number of components in set 6 results in higher initial investment and maximum maintenance costs. In view of this, it is found that set 2 is found to be best in terms of the ratio of net output to exergy destruction of the plant as well as initial, operational, and maintenance cost.

NOMENCLATURE

P	Work output, W
Q	Rate of heat supplied, W
$\dot{m}$	Mass flow rate, kg/s
η_{th}	Thermal efficiency
ε	Heat Exchanger Effectiveness
r_p	Pressure Ratio
γ	Specific Heat Ratio
T	Temperature, K
TIT	Turbine Inlet Temperature, K
HE	Heat Exchanger
HPC	High Pressure Compressor
LPC	Low Pressure Compressor
HPT	High Pressure Turbine
LPT	Low Pressure Turbine
SFC	Specific Fuel Consumption, Kg/kW.hr
E	Exergy, W
LCV	Lower Calorific Value of Fuel

Subscripts

c	Compressor
t	Turbine
net	Net
a	Air
f	Fuel
g	Gas

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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