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A COMPARATIVE STUDY OF USING SIMPLE AND EJECTOR-ABSORPTION REFRIGERATION FOR INLET AIR COOLING OF SIMPLE AND REGENERATIVE GAS TURBINE

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ABSTRACT
The exhaust gases of gas turbine power plant carry a significant amount of thermal energy that is usually expelled to the atmosphere; this causes a reduction in net work and efficiency of gas turbine. On the other hand, the generated power and efficiency of gas turbine plants depend largely on the temperature of the inlet air, So that they both increase as the inlet air temperature decreases. The mentioned two problems can be solved by installing an absorption refrigeration cycle (ARC) at gas turbine inlet, working with thermal energy of exhaust gases. In this research, effect of inlet air cooling on gas turbine performance is studied. The work shows that, the net work and the efficiency will increase by 6-10% and 1-5% respectively for every 10°C decrease of inlet temperature. Since, coefficient of performance (COP) of ARC is low, with high pressure ratios in simple gas turbine (SGT) and with low pressure ratios in regenerative gas turbine (RGT), thermal energy of exhaust gases can not supply all the needed thermal energy for refrigeration cycle. The results show that, when an ejector is included in refrigeration cycle, the need for external energy source required for refrigeration cycle is reduced.

KEY WORDS
Gas Turbine, Inlet Air Cooling, Absorption Refrigeration, Ejector, Regenerative Gas Turbine

1. Introduction

Gas turbines are known to have a number of attractive features, principally: low capital cost, compact size, short delivery, high flexibility, fast starting and loading, lower manpower operating needs, not needing to water sources and better environmental performance, compared with other electricity producing devices specially the steam turbine. However, it suffers from relatively lower efficiency and strong influence of climate conditions specially temperature on its behavior. Also thermal energy of exhaust gases is delivered to and wasted in the environment. This low grade thermal energy can be put to beneficial use in a heat exchanger of RGT and/or generator of absorption refrigeration cycle to increase the power and efficiency of gas turbine plants.

In recent years, several researches have been carried out to increase performance of gas turbine plants by using an ARC for inlet air cooling [1-7]. All of these works show an increase of power and efficiency of gas turbine by reduction of inlet air temperature. These articles have investigated the effect of inlet air cooling, but in addition to this, we have studied the amount of thermal energy of exhaust gases and their ability to satisfy the required thermal energy of refrigeration cycle in different conditions of both cycles.

It's shown that, with some pressure ratio (r_c) of gas turbine, because of low COP of ARC, energy of exhaust gases can not provide all needed energy of refrigeration system to have a low and constant inlet temperature. Using ejector in absorption refrigeration system brings about the advantages of absorption and ejector refrigeration systems and provides high COP. So that, the range of r_c with which the thermal energy of exhaust gas of both SGT and RGT is enough for refrigeration cycle, is extended. A number of models have been suggested to study the effect of ejector in COP of refrigeration cycle [8-14]. We have used Wen Sun's model [8] to simulate ejector and absorption cycle.

2. Assumptions

Following assumptions are made for gas turbine and refrigeration system in this work:
1. polytropic efficiency of compressor and turbine are 0.9 and 0.85 respectively.
2. Methane (CH_4) with low heat value of 50010kJ/kg is used as fuel and its pressure and temperature are the same as that of combustion chamber inlet.
3. Combustion efficiency and its pressure drop is 0.98 and 5% respectively and Pressure drop of compressor inlet and turbine outlet is 1kPa.
4. Heat exchanger efficiency in RGT is 0.85
5. Mechanical efficiency is 0.98
6. ISO standard conditions for inlet air is T=15°C, P=100kPa, φ=60%
7. Minimum inlet temperature to prevent icing at the compressor inlet is 12°C and minimum stack temperature is 100°C

8. Assumed condition for EARC is: condenser temperature (T_{cond})=45°C, absorber temperature (T_{abs})=45°C, evaporator temperature (T_{eva})=8°C, generator temperature (T_{gen})=95°C, heat exchanger efficiency in refrigeration cycle (η_{ex})=90%, generator pressure (P_{gen})=10KPa, ejector area ratio $(A_r) = A_j / A_t = 10$, nozzle efficiency (η_n) and diffuser efficiency (η_d) are 90%.

A parametric study was carried out changing one parameter at a time and keeping the others fixed. For ARC the assumptions are similar to these values, except that $P_{con} = P_{gen} = P_{sat\ T_{con}}$ and A_r, η_n and η_d are omitted.

3. Studied cycle

Fig. 1 shows RGT with Ejector- Absorption Refrigeration Cycle (EARC). When we study the effect of inlet air cooling in SGT, heat exchanger will be omitted; and when we want to use ARC, we omit the ejector and line 13 of steam. An ejector integrated in this way into the ARC increase the refrigerant flow rate from the evaporator and therefore raises the cooling capacity of the machine.

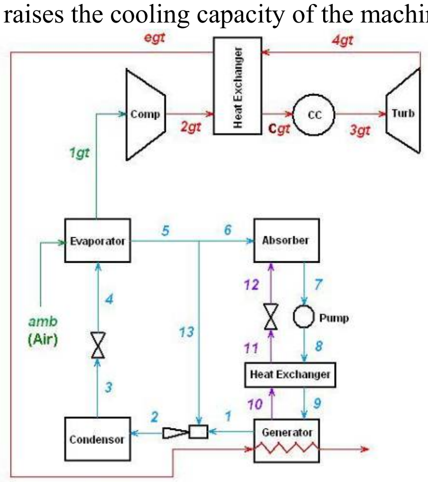


Figure 1. RGT with EARC

4. Effect of inlet air cooling on SGT and RGT performance

Specific work of SGT and RGT as a function of compressor inlet temperature (T_0) with different r_c and turbine inlet temperature (TIT) is shown in Fig. 2 and 3.

These schemes show that, as T_0 increases, specific work decreases. This variation is linear. The slop of this line increases with a rise in r_c and decreases slightly with rise in TIT. Fig. 2 and Fig. 3 show a comparison of specific work of SGT and RGT. In both figures, specific work increases with r_c to the maximum specific work that occurs in optimum r_c and then decreases. For each TIT the optimum r_c is given in Table. 1 that shows the maximum specific work and optimum r_c increases as TIT rises and this is expected from ideal cycle equations [21].

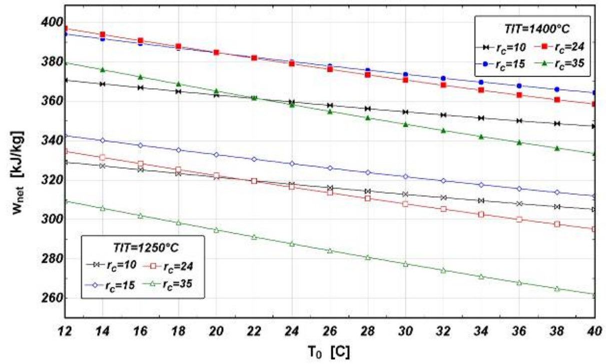


Figure 2. Effect of compressor inlet air temperature on SGT specific work in different r_c and TIT (φ_0=60%)

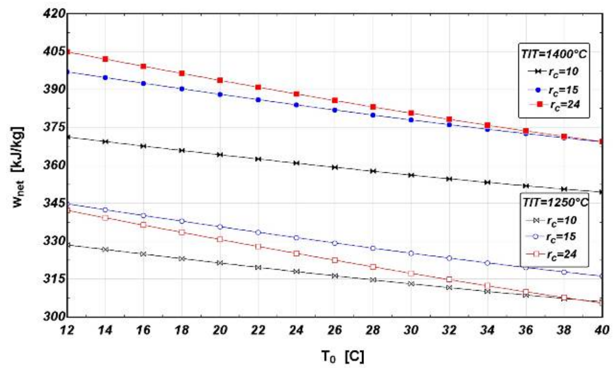


Figure 3. Effect of compressor inlet air temperature on RGT specific work in different r_c and TIT (φ_0=60%)

Table 1
The effect of TIT on gas turbine maximum specific work obtained with an optimum amount of r_c (φ_0=60%, T_0=15°C)

TIT [C]	r_c	Max w_{net} [kJ/kg]
1100	13.08	289.5
1250	17.5	339.2
1400	20.1	394.8

Efficiency of SGT and RGT as a function of T_0 with different r_c and TIT is shown in Fig. 4 and Fig. 5 respectively. From these schemes, it is clear that efficiency of SGT and RGT decreases with an increase in inlet temperature and the amount of decrease increases with an increase of r_c. Efficiency of SGT increases with r_c to the maximum amount that occurs in optimum r_c and then decreases. For each TIT the optimum r_c is given in Table. 2. The efficiency of RGT decreases with an increase of r_c, because, when r_c increases, temperature of gases in outlet of turbine decrease and temperature of air in outlet of compressor increases, so, the recovered thermal energy in heat exchanger falls until zero corresponding to the r_c that at this point efficiency of SGT and RGT equals. These pressure ratios are presented in Table. 3 for each TIT. For higher values of r_c the heat exchanger would cool the air leaving the compressor and so reduce the efficiency.

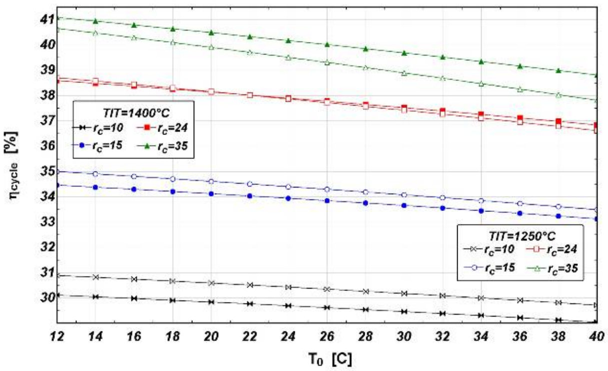


Figure 4. Effect of compressor inlet air temperature on SGT efficiency in different r_c and TIT (φ_0=60%)

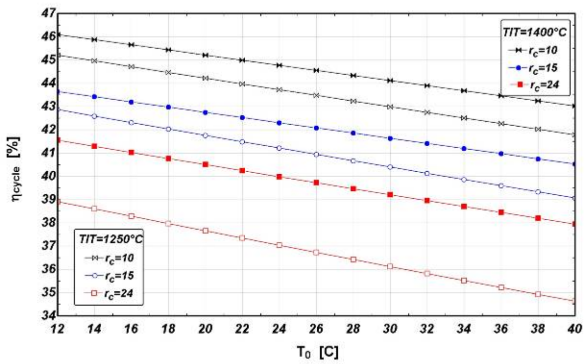


Figure 5. Effect of compressor inlet air temperature on RGT efficiency in different r_c and TIT (φ_0=60%)

Table 2

Effect of TIT on SGT maximum efficiency and its r_c

(φ_0=60%, T_0=15°C)

TIT [C]	r_c	Max η_{Cycle} [%]
1100	35.98	39.42
1250	47.6	40.94
1400	62.6	42.57

Table 3

Maximum r_c for RGT with different TITs.

TIT [C]	1100	1250	1400
r_c	19	24	29

5. Cooling capacities obtainable from the SGT and RGT exhaust gases in ARC and EARC

In previous sections, it was shown that a decrease of compressor inlet air temperature causes an increase of efficiency and specific work of gas turbine. In this section we will consider, the required cooling capacity (Q_{eva}) to have a specified temperature and humidity in compressor inlet air, the required thermal energy in generator (Q_{gen}) to achieve the cooling capacity and the thermal energy available in exhaust gases ($Q_{gen,available}$) of SGT and RGT.

In Fig. 6, Q_{eva} is shown as a function of ambient temperature (T_{amb}) at the compressor inlet conditions.

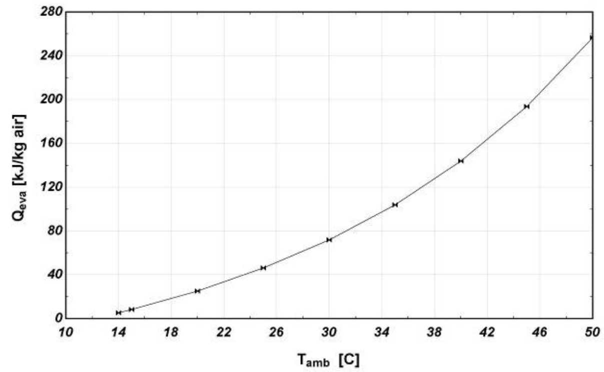


Figure 6. Required cooling capacity for inlet air cooling as a function of ambient temperature (T_0=12°C, φ_{amb}=60%, φ_0=60%)

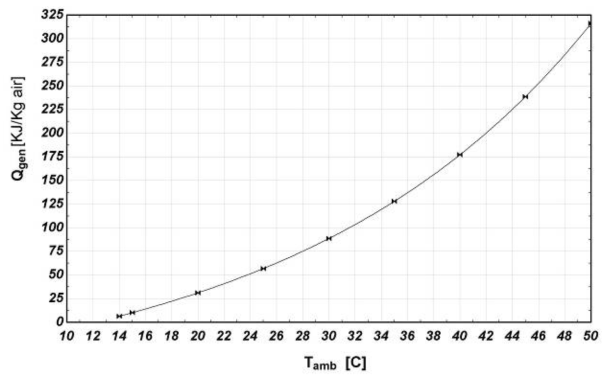


Figure 7. ARC required energy as a function of ambient temperature for providing specified inlet cooling (T_0=12°C, φ_{amb}=60%, φ_0=60%)

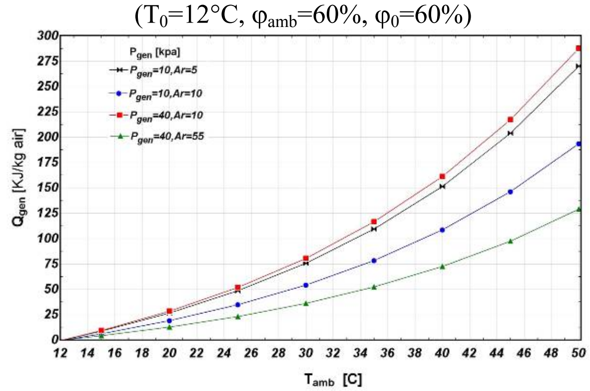


Figure 8. EARC required energy as a function of ambient temperature, P_{gen} and A_r for providing specified inlet cooling (T_0=12°C, φ_{amb}=60%, φ_0=60%)

It indicates that, an increase of T_{amb} causes an increase of Q_{eva} to achieve compressor inlet condition ($T_0 = 12°C, \varphi_0 = 60\%$). To achieve the cooling capacity, Q_{gen} of ARC and EARC is calculated as a function of T_{amb} and it is shown in Figs. 7-8 respectively. It is clear that when T_{amb} increases Q_{gen} increases too. Also, Fig. 8 shows increase of Q_{gen} with increase of P_{gen} and with decrease of A_r .

Figs. 9-12 show the variation of Q_{gen} with refrigeration system parameters for ARC and Figs. 13-16 show these variations for EARC. In general, Q_{gen} of EARC is lower than that of ARC because of ejector presence that increases COP of refrigeration cycle. In both ARC and EARC, Q_{gen} increases with increase of absorber, condenser, generator temperatures and decrease of evaporator temperature. Also, in EARC, Q_{gen} increases when P_{gen} increases and/or A_r decreases, but we will focus on the same parameters of EARC and ARC (absorber, condenser, generator and evaporator temperatures).

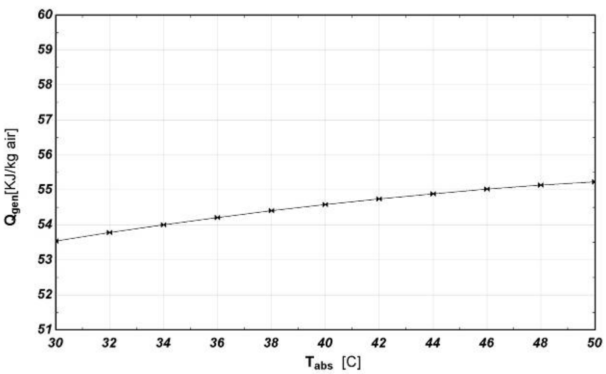


Figure 9. ARC required energy as a function of T_{abs} for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

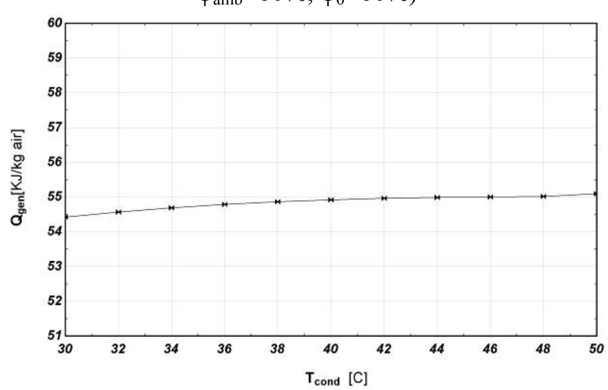


Figure 10. ARC required energy as a function of T_{cond} for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

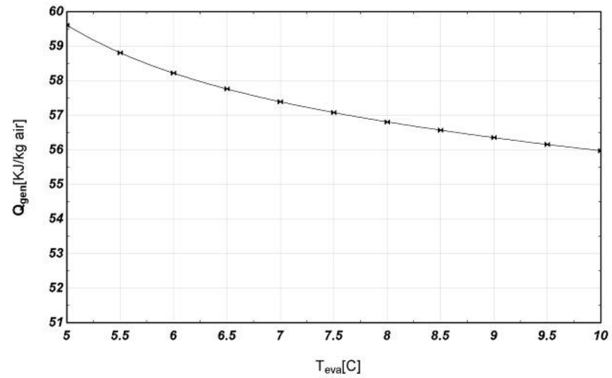


Figure 11. ARC required energy as a function of T_{eva} for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

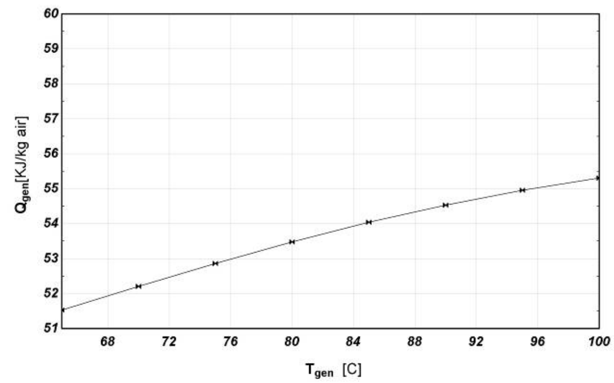
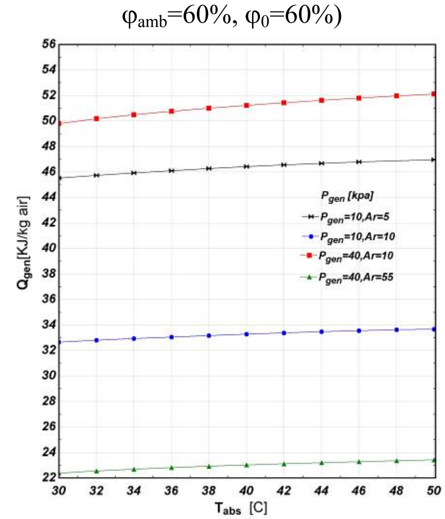


Figure 12. ARC required energy as a function of T_{gen} for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)



Figured 13. EARC required energy as a function of T_{abs} , P_{gen} and A_r for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

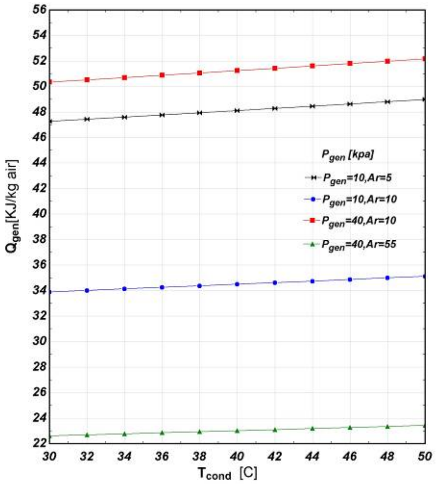


Figure 14. EARC required energy as a function of T_{cond}, P_{gen} and A_r for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

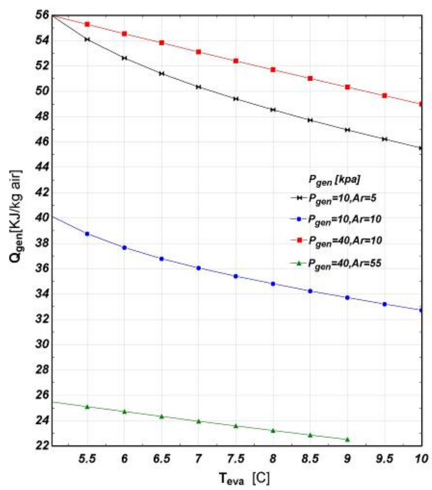


Figure 15. EARC required energy as a function of T_{eva}, P_{gen} and A_r for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

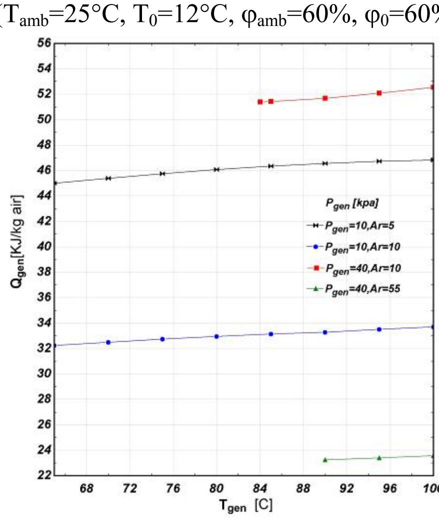


Figure 16. EARC required energy as a function of T_{gen}, P_{gen} and A_r for providing specified inlet cooling (T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%)

In order to study the amount of required energy that can be provided from exhaust gases, available energy in exhaust gases of SGT and RGT is presented in Fig. 17 and Fig. 18 with different TITs.

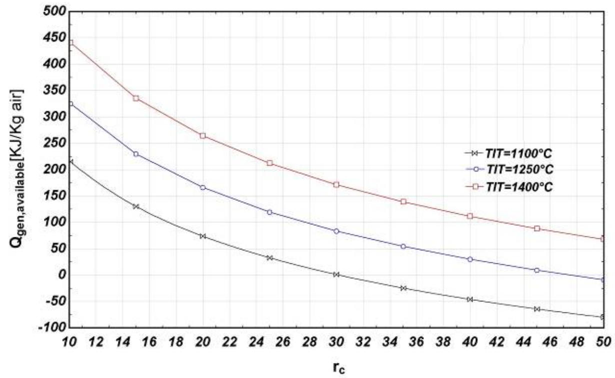


Figure 17. Thermal energy of SGT exhaust gases as a function of r_c and TIT (T_0=12°C, φ_0=60%)

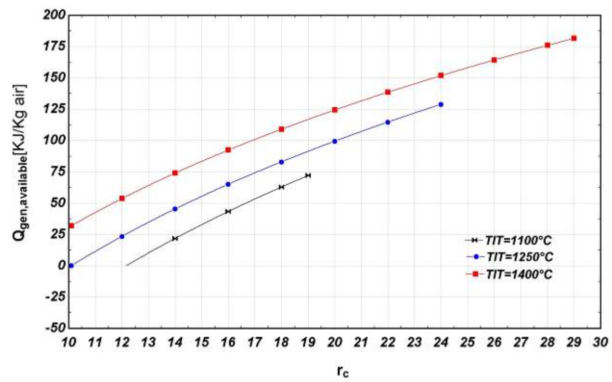


Figure 18. Thermal energy of RGT exhaust gases as a function of r_c and TIT (T_0=12°C, φ_0=60%)

The effective factors in the amount of $Q_{gen,available}$ are r_c and TIT. From the Figures it is clear that, for SGT, $Q_{gen,available}$ decreases with increase of r_c as the turbine outlet temperature falls. But in RGT the trend is inverse because of decreased recovered thermal energy in heat exchanger. In both SGT and RGT $Q_{gen,available}$ increases with an increase of TIT.

Now, we characterize the conditions with which Q_{gen} can be supplied from $Q_{gen,available}$. The tables 4-6 are obtained from a comparison of Figs. 7-16 with Figs. 17-18. Table. 4 shows the Variation of r_c with ambient temperature with which Q_{gen} equals to $Q_{gen,available}$ for all studied cycles. This table indicates that, for SGT, in all r_c lower than that mentioned in the table, Q_{gen} can be provided from $Q_{gen,available}$ completely, but for RGT with r_c higher than mentioned in table, up to the value given in Table. 3 for each TIT, Q_{gen} can be supplied from $Q_{gen,available}$. The same discussion can be made for Tables 5 and 6, which show the effect of TIT and different parameters of refrigeration cycle on the r_c with which Q_{gen} equals to $Q_{gen,available}$.

It is shown that, by using ejector in refrigeration cycle, the range of r_c that, the thermal energy of exhaust gases is enough for refrigeration cycle is extended in both SGT and RGT.

6. Conclusion

In this research firstly the effect of inlet air cooling on performance of gas turbine is studied. In both SGT and RGT cycles a reduction of inlet temperature showed an increase of power and efficiency especially with high r_c. The maximum power and efficiency of SGT and the r_c corresponding to these maximum amounts increased with a decrease of inlet temperature. These values are presented in Tables 7 and 8.

Table 4

Variation of Q_{gen} and $Q_{gen,available}$ equivalence r_c with ambient temperature (TIT=1250°C)

	ARC			EARC		
T_{amb}[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
25	56.81	34.52	15.14	34.81	38.9	13
35	128	23.98	23.85	78.42	30.8	17.5
40	177.2	19	31.9	108.6	26.4	21.2

Table 5

Variation of Q_{gen} and $Q_{gen,available}$ equivalence r_c with TIT(T_{amb}=25°C)

	ARC			EARC		
TIT[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
1100	56.81	21.9	17.35	34.81	24.7	15.18
1250	56.81	34.52	15.4	34.81	38.9	13
1400	56.81	56.91	12.29	34.81	37.5	10.35

Table 6

Variation of Q_{gen} and $Q_{gen,available}$ equivalence r_c with parameters of refrigeration cycle(T_{amb}=25°C, T_0=12°C, φ_{amb}=60%, φ_0=60%, TIT=1250°C)

	ARC			EARC		
T_{gen}[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
75	52.86	35.27	14.73	32.74	39.35	12.75
85	54.04	35.04	14.85	33.15	39.3	12.85
95	56.81	34.54	15.14	34.81	38.9	13
T_{eva}[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
5	59.61	34	15.42	40.2	37.7	13.5
8	56.81	34.52	15.14	34.81	38.9	13
10	55.98	34.7	15.04	32.72	39.35	12.08
T_{abs}[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
30	54.2	35	14.85	33.07	39.3	12.85
45	56.84	34.52	15.4	34.81	38.9	13
50	63.34	33.35	15.8	46.92	36.4	14.15
T_{cond}[C]	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)	$Q_{gen}=Q_{gen,available}$ [kJ/kg air]	r_c (SGT)	r_c (RGT)
30	54.96	34.85	14.94	33.9	39.1	12.9
45	56.81	34.52	15.4	34.81	38.9	13
50	64.15	33.2	15.9	35.13	38.85	13.05

Table 7
Effect of compressor inlet air temperature on SGT maximum specific work and its r_c (TIT=1250°C)

T_0[C]	r_c	Max W_{net} [kJ/kg air]
12	16.4	343
15	16.11	339.2
20	15.65	333
40	14.12	312.2

Table 8
Effect of compressor inlet air temperature on SGT maximum efficiency and its r_c (TIT=1250°C)

T_0[C]	r_c	Max η_{cycle} [%]
12	49.05	41.32
15	47.6	40.94
20	45.34	40.32
40	38.02	37.86

These tables show that, the maximum specific work and efficiency of SGT increases 1.9% and 1.6% respectively with every 5°C decrease of inlet air temperature. It is clear that, the maximum specific work of RGT behaves like SGT and their amounts are the same approximately.

Also, it is shown that for SGT, with low r_c exhaust gases have enough thermal energy to supply required energy of ARC, but in RGT, with low r_c exhaust gases do not carry enough energy to provide all required energy of ARC. In SGT with high r_c exhaust gases do not provide the needed energy for ARC. Utilization of ejector decreases required energy of refrigeration system, so that, the range of r_c with which, the thermal energy of exhaust gases is enough for refrigeration cycle is extended for both SGT and RGT cycles.

Finally, the effect of refrigeration cycle parameters on the range of r_c with which Q_{gen} can be provided from $Q_{gen,available}$ completely is studied. It was shown that, Q_{gen} decreases with a decrease of absorber, condenser, generator temperatures and an increase of evaporator temperature in both ARC and EARC and Q_{gen} decreases with a decrease of P_{gen}. Therefore with these variations the mentioned range of r_c will be extended.

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