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Energy and Exergy Analysis of Simple and Regenerative Gas Turbines Inlet Air Cooling Using Absorption Refrigeration

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Abstract: This study deals with energy and exergy analysis of simple and regenerative gas turbines with inlet air cooling using lithium bromide-water absorption refrigeration cycle. A parametric study has been carried out and effects of important factors like compressor inlet air temperature, pressure ratio and turbine inlet temperature on the performance of the cycles were studied. The results show that for each 10°C decrease of inlet air temperature, net output power increases around 6-12% and the first and second law efficiencies increase around 2-7%. It is shown that the amount of this increase is higher when the pressure ratio is high and turbine inlet temperature is low. Also exergy destruction of all components has been calculated and the results indicate that the combustor has the largest contribution on exergy destruction and the exergy destruction of absorption refrigeration cycle is very low as compared to that of the gas turbine one. It is also found that the energy content of exhaust gases in both simple and regenerative gas turbines is more than enough to run the refrigeration cycle. The results of required cooling load can be helpful in estimating the cost of the necessary absorption refrigeration unit.

Key words: Gas turbine, inlet air temperature, pressure ratio, turbine inlet temperature, irreversibility, power enhancement, efficiency enhancement

INTRODUCTION

The thermodynamic processes of a simple gas turbine cycle can be approximately modeled as brayton cycle. Efficiency of this cycle is low because the back work ratio and exhaust temperature are high. Recovering the energy content of exhaust gases can improve the cycle performance. Different techniques have been employed for this purpose. Some of these techniques are: regeneration (Pilavachi, 2000), steam injection (Nishida *et al*., 2005), humid air turbine (Lazzaretto and Segato, 2002), chemical recuperation (Alves and Nebra, 2004), combined cycle (Marrero *et al*., 2002) and inlet air cooling (Kakaras *et al*., 2004). The last technique improves cycle performance for two reasons. First; decreasing inlet air temperature will decrease compressor specific work and therefore the gas turbine specific net work will increase. Second; as the temperature of the air decreases the air density and consequently the air mass flow rate increases which in turn causes an increase in output power. Depending on the gas turbine type, for each 10°C increase of inlet air temperature, the output power will decrease by around 6-10% (Kakaras *et al*., 2004). There are different methods of compressor inlet air cooling and each method has some advantages and some limitations (Wang and Chiou, 2004). Absorption refrigeration has been paid more attention as it can utilize low grade energy of the exhaust gas instead of electrical energy to provide the necessary cooling. The first law analysis of compressor inlet air cooling using lithium bromide-water absorption refrigeration, has been done recently by Kakaras *et al*. (2004), Salvi and Pierpaoli (2002), Bassily (2004), Amery and Hejazi (2004) and Boonnasa *et al*. (2006). Kakaras *et al*. (2004), Dawoud *et al*. (2005) and Hosseini *et al*. (2007) have reported the effect of inlet air cooling using other methods such as evaporative cooling, compression refrigeration and aqua-ammonia absorption refrigeration. Also, some have analyzed the effect of ambient temperature on the performance of gas turbine cycles (Erdem and Sevilgen, 2006). All these studies show that decreasing inlet air temperature results in an increase of the output power and efficiency of the gas turbine. However, in most of these works the analysis had been done based on the first law of thermodynamics and there is a lack of data in this issue from the point of view of second law. The present study deals with the effect of compressor inlet air cooling on the performance of the simple and regenerative gas turbines using lithium bromide-water absorption refrigeration and attention has been focused on both the first and second law of thermodynamics. A parametric study has been carried out and effects of important factors like

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compressor inlet air temperature, pressure ratio and turbine inlet temperature on the performance of the cycles were studied through a home made computer program.

MATERIALS AND METHODS

This study has been conducted at the University of Tabriz, Iran to analyze the effect of simple and regenerative gas turbines inlet air cooling using absorption refrigeration, from the point of view of the first and second law of thermodynamics. This research project was conducted from 1/1/2007 to 1/1/2008.

Regenerative gas turbine cycle with absorption inlet air cooling is shown in Fig. 1. In order to analyze the effect of inlet air cooling on the simple gas turbine cycle, the regenerator is omitted. Ambient air goes through the evaporator before entering the compressor and thus its temperature decreases. The generator receives energy from exhaust gases coming out of regenerator (or turbine). The remaining energy in exhaust gases can be utilized for any purposes. A water flow of 25°C is used as a cooling medium for the absorber and condenser. Other assumptions for the gas turbine and refrigeration cycles are shown in Table 1.

For thermodynamic analysis each component of the system has been treated as a control volume at steady state and the principles of mass conservation, first and second law of thermodynamics are applied to the component. The mass balance for each component can be expressed as:

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \tag{1}$$

The first law of thermodynamics yields the energy balance of each component as follows:

$$\sum (\dot{m}h)_{in} - \sum (\dot{m}h)_{out} + \dot{Q}_{cv} - \dot{W}_{cv} = 0 \tag{2}$$

Exergy rate balance for each component of the system is expressed as (Kotas, 1995):

$$\dot{I} = \sum \dot{E}_{in} - \sum \dot{E}_{out} \tag{3}$$

where, $\Sigma\dot{E}_{in}$ and $\Sigma\dot{E}_{out}$ are the sum of all exergy transfers entering and coming out of the component. These exergy transfers can be associated with work, heat or mass transfer. If the effects of kinetic and potential energies are ignored, the specific exergy of a fluid stream will be defined as:

$$\varepsilon = \varepsilon_{ph} + \varepsilon_{ch} \tag{4}$$

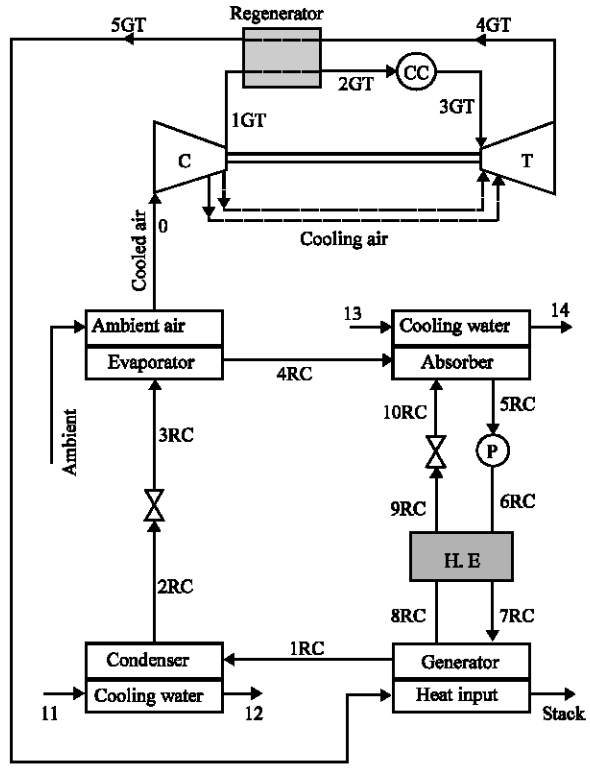

Fig. 1: Regenerative gas turbine cycle with absorption inlet air cooling

Table 1: Assumptions for the gas turbine and refrigeration cycles

Turbine polytropic efficiency	0.87
Compressor polytropic efficiency	0.86
Mechanical efficiency	0.98
Combustor efficiency	0.99
Regenerator effectiveness	0.75
Refrigeration cycle heat exchanger effectiveness	0.70
Pressure loss of combustor	5%
Pressure loss of regenerator	2%
Pressure loss of air in the evaporator	2%
Generator temperature	105°C
Absorber/Condenser temperature	47°C
Evaporator temperature	8°C
Compressor inlet air temperature/RH	10°C/60%
LHV of fuel	802361 kJ kmol^{-1}
Chemical exergy of fuel	836510 kJ kmol^{-1}

The first and second law efficiencies for the gas turbine cycle are defined as follows:

$$\eta_{cycle} = \frac{w_{net}}{f\ .LHV} \tag{5}$$

$$\eta_{II,cycle} = \frac{w_{net}}{f\ .\ \varepsilon_{fuel}} \tag{6}$$

where, w_{net} is the specific net work based on the unit mass flow of air entering the compressor, f is the mass basis fuel-air ratio and the LHV and ε_{fuel} are the lower heat value and specific exergy of fuel, respectively.

RESULTS AND DISCUSSION

Considering the assumptions shown in Table 1 and the equations of mass, energy and exergy balances, a parametric study was performed by means of the computer program using EES software and effect of inlet air cooling was studied for simple and regenerative gas turbines in different r_c s and TIT s.

Analyzing the effects of T_0 and r_c: Figure 2-5 show the effect of inlet air temperature and pressure ratio on the specific net work, first and second law efficiencies and irreversibility of SGT and RGT cycles at TIT = 1200°C.

From Fig. 2, it is found that for both SGT and RGT cycles, w_{net} is increased as T_0 decreases. However, the rate of this increase is more pronounced in high pressure ratios. The little difference between the values of specific net work of the SGT and RGT cycles is due to the difference between the amounts of turbine blade's cooling air fraction and fuel consumption in these cycles.

Figure 3 shows an almost linear variation of the first law efficiency with T_0 for both SGT and RGT cycles. Comparatively higher increase of efficiency with a decrease in T_0 is indicated when the pressure ratio is high. Also Fig. 3 reveals a more pronounced effect of T_0 for RGT cycle.

A comparison between Fig. 3 and 4 show a similar trend for first and second law efficiencies of SGT and RGT cycles with respect to T_0 and r_c.

Figure 5 shows a negligible effect of T_0 on irreversibility of SGT cycle. However, in this cycle, the irreversibility is decreased with increasing pressure ratio. Also Fig. 5 indicates higher values of irreversibility for RGT cycle when T_0 and r_c are high.

Analyzing the effects of T_0 and TIT: With a pressure ratio of r_c = 12, the effects of compressor inlet air temperature and turbine inlet temperature on the specific net work, first and second law efficiencies and irreversibility of SGT and RGT cycles are shown in Fig. 6-9.

For both SGT and RGT cycles, Fig. 6-8 show comparatively higher influence of T_0 on w_{net}, $\eta_{I,cycle}$ and $\eta_{II,cycle}$ at low TIT s.

Figure 7 and 8 indicate that increasing TIT results in a decrease of the first and second law efficiencies for SGT cycle and it is due to the increase of fuel consumption as TIT increases. However, for RGT cycle increasing TIT will

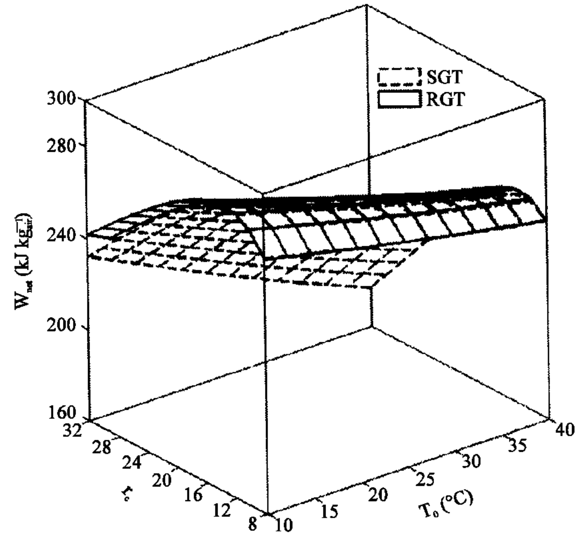

Fig. 2: Effect of T_0 and r_c on specific net work of SGT and RGT cycles at TIT = 1200°C

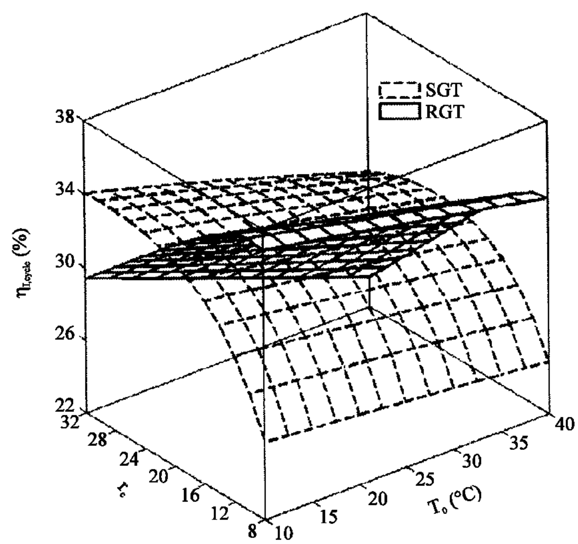

Fig. 3: Effect of T_0 and r_c on first law efficiency of SGT and RGT cycles at TIT = 1200°C

increase the first and second law efficiencies. We should note that these results are with r_c = 12 and at r_c s above 16 an opposite result has been obtained, but it is not shown here.

It can be seen from Fig. 9 that the irreversibility of SGT and RGT cycles will increase as TIT increases. This is due to the increase of irreversibility in combustor and turbine when TIT is increased. Also Fig. 9 shows that the value of irreversibility of RGT is lower than that for SGT, particularly at high TIT s. This is because of this fact

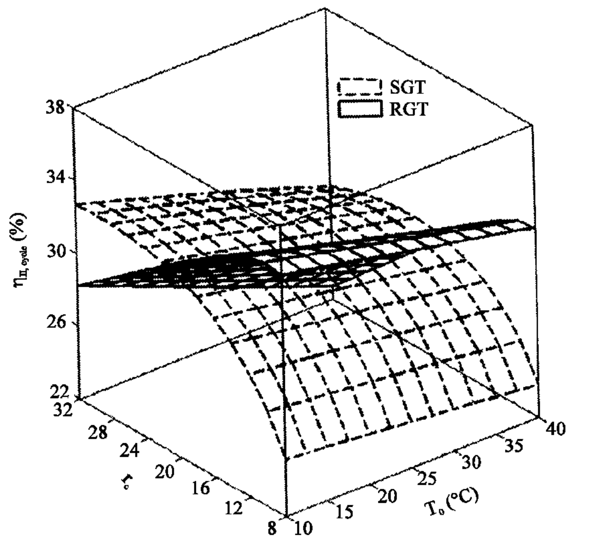

Fig. 4: Effect of T_0 and r_c on second law efficiency of SGT and RGT cycles at TIT = 1200°C

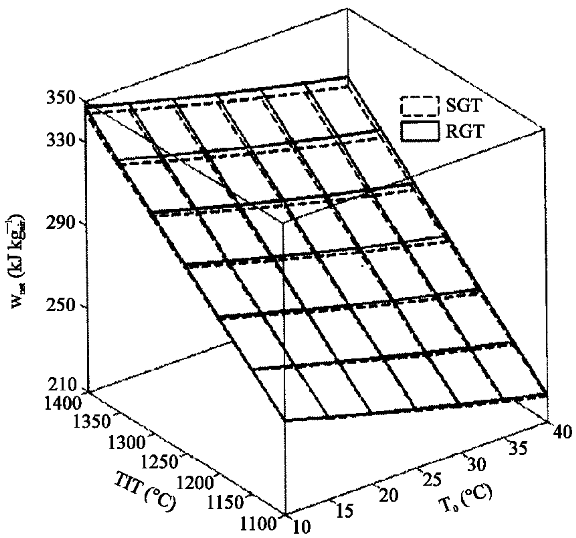

Fig. 6: Effect of T_0 and TIT on specific net work of SGT and RGT cycles at r_c = 12

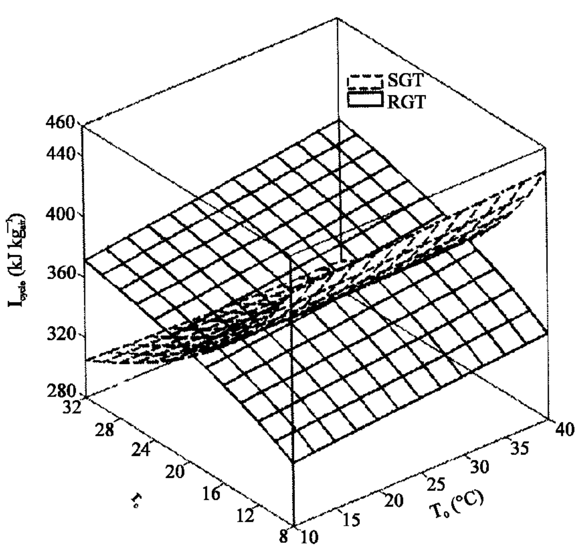

Fig. 5: Effect of T_0 and r_c on irreversibility of SGT and RGT cycles at TIT = 1200°C

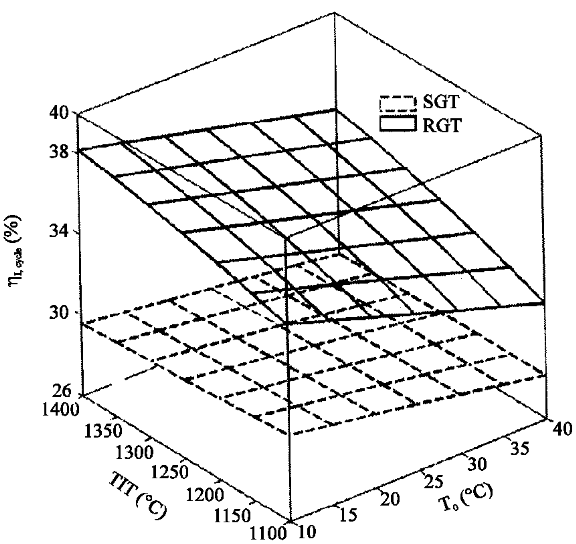

Fig. 7: Effect of T_0 and TIT on first law efficiency of SGT and RGT cycles at r_c = 12

that the irreversibility in the combustor of RGT cycle is lower than the one in SGT.

Comparing the effects of T_0 and TIT: Interrelation between compressor inlet air temperature, specific net work and first law efficiency of SGT is shown in Fig. 10 and 11. Also Fig. 12 indicates the interrelation between TIT, w_{net} and $\eta_{I,cycle}$. As it can be seen lowering T_0 or increasing TIT increases both specific net work and first law efficiency.

An interesting comparison can be made between Fig. 11 and 12. As they shown the variations of w_{net} and $\eta_{I,cycle}$ with T_0 and/or TIT are almost similar. However, T_0 is more effective on $\eta_{I,cycle}$ while TIT has strong influence on w_{net}. At practical pressure ratios (around 14), $\eta_{I,cycle}$ does not have a significant variation with increasing TIT from 1100 to 1400°C while there is an increase of around 7.5% on the efficiency as T_0 decreases from 40 to 10°C with TIT = 1200°C. From these figures it can also be concluded that an increase

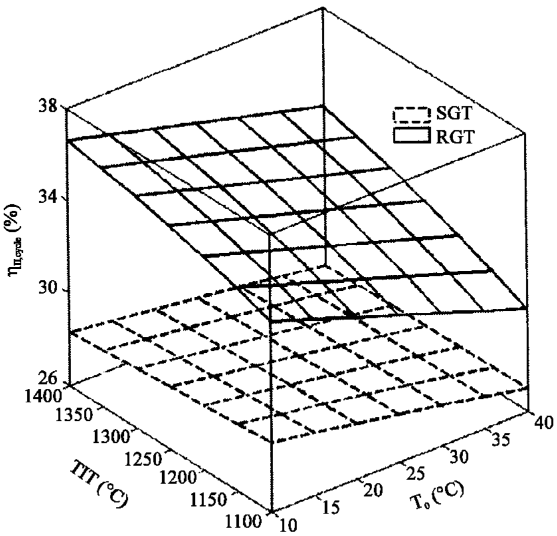

Fig. 8: Effect of T_0 and TIT on second law efficiency of SGT and RGT cycles at $r_c = 12$

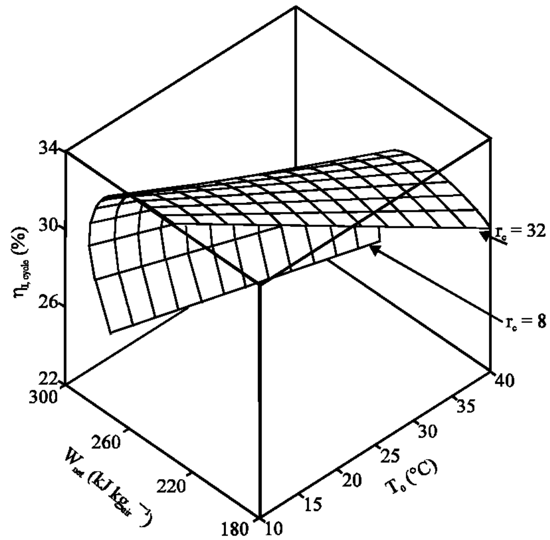

Fig. 10: Interrelation between T_0, w_{net} and $\eta_{I,cycle}$ of the SGT at TIT = 1200°C

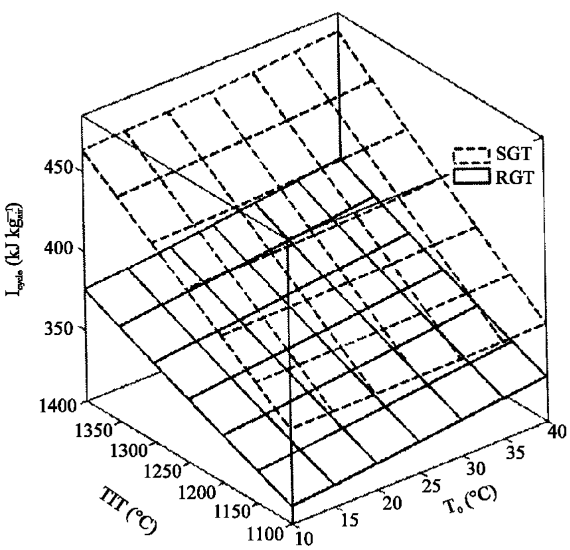

Fig. 9: Effect of T_0 and TIT on irreversibility of SGT and RGT cycles at $r_c = 12$

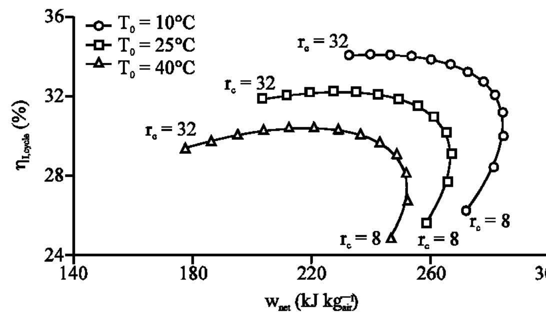

Fig. 11: Interrelation between T_0, w_{net} and $\eta_{I,cycle}$ of the SGT at TIT = 1200°C

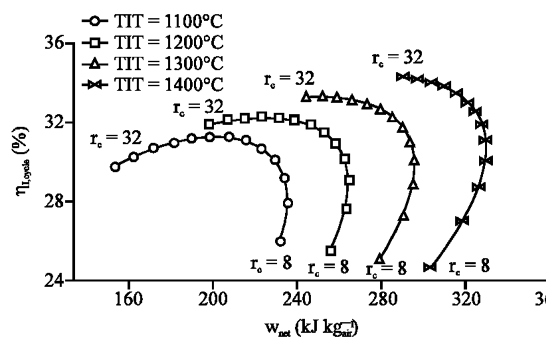

Fig. 12: Interrelation between TIT, w_{net} and $\eta_{I,cycle}$ of the SGT at T_0 = 25°C

of 100°C in TIT has almost the same effect on w_{net} as a decrease of T_0 by 30°C.

Figure 13 and 14 show the variation of pressure ratio with respect to compressor inlet air temperature which brings about the maximum value of specific net work and first law efficiency of SGT. It is evident that the pressure ratio which brings about the maximum value of specific net work and first law efficiency of SGT cycle increases as T_0 decreases and/or TIT increases. Also Fig. 13 and 14 show that the pressure ratio for maximum $\eta_{I,cycle}$ is greater than the one for w_{net}.

Figure 15 to 17 show the interrelation between compressor inlet air temperature, turbine inlet temperature, specific network and first law efficiency of RGT cycle. A comparison between Fig. 16 and 17 show that the

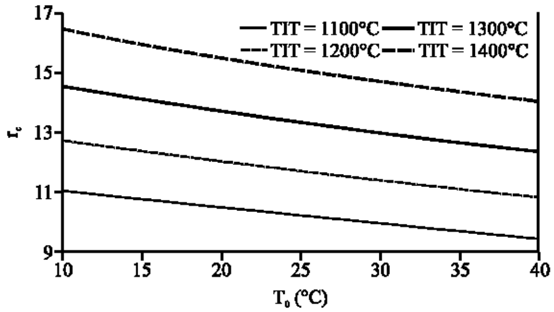

Fig. 13: Pressure ratios which brings about the maximum value of specific net work of SGT

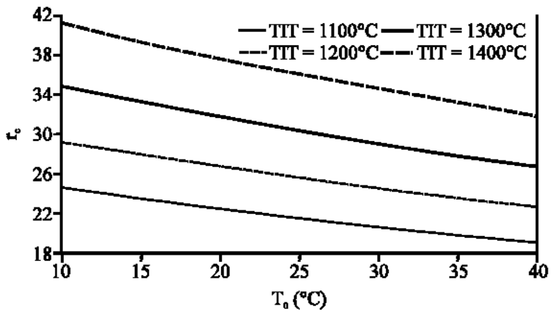

Fig. 14: Pressure ratios which brings about the maximum value of first law efficiency of SGT

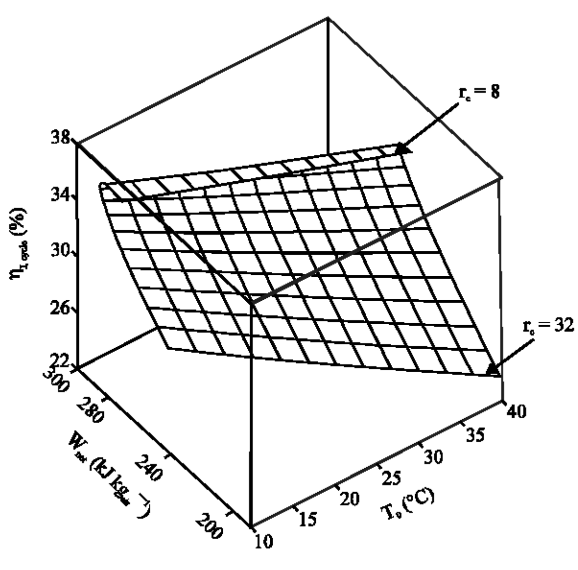

Fig. 15: Interrelation between T_0, w_{net} and $\eta_{I,cycle}$ of the RGT at TIT = 1200°C

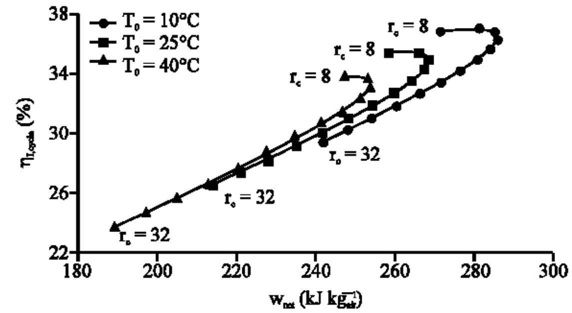

Fig. 16: Interrelation between T_0, w_{net} and $\eta_{I,cycle}$ of the RGT at TIT = 1200°C

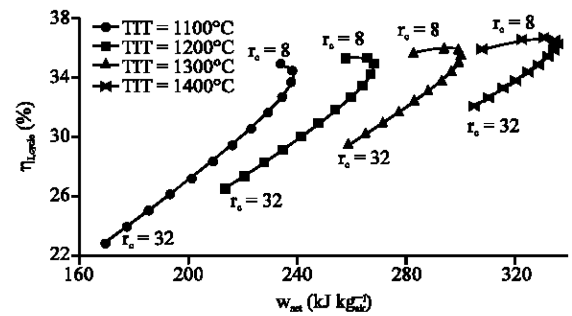

Fig. 17: Interrelation between TIT, w_{net} and $\eta_{I,cycle}$ of the RGT at T_0 = 25°C

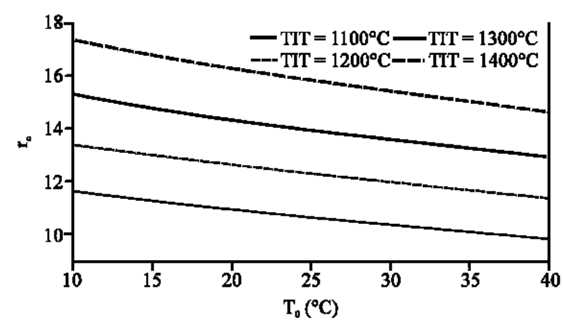

Fig. 18: Pressure ratios which brings about the maximum value of specific net work of RGT

variation of T_0 and/or TIT has a similar effect on w_{net} and $\eta_{I,cycle}$ of RGT; so that decreasing T_0 or increasing TIT results in an increase of w_{net} and $\eta_{I,cycle}$. This result is similar to the corresponding one for SGT cycle. Another implication of these figures is the negligible effect of TIT on $\eta_{I,cycle}$ at practical pressure ratios (around r_c = 14), while decreasing T_0 by 30°C can increase the efficiency quite considerably (around 11%).

Like the case of SGT cycle (Fig. 13, 14), at different conditions of T_0 and TIT there are some pressure ratios with which w_{net} and $\eta_{I,cycle}$ is maximized and the value of these pressure ratios can be obtained from Fig. 16 and 17. These results which is shown in Fig. 18 and 19 indicate

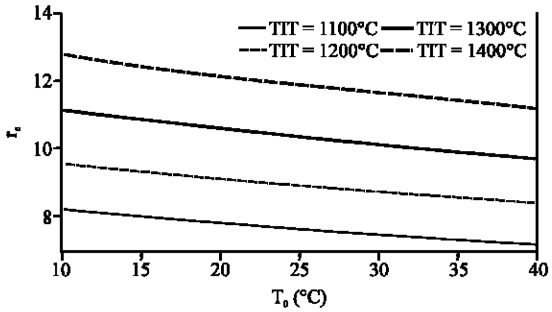

Fig. 19: Pressure ratios which brings about the maximum value of first law efficiency of RGT

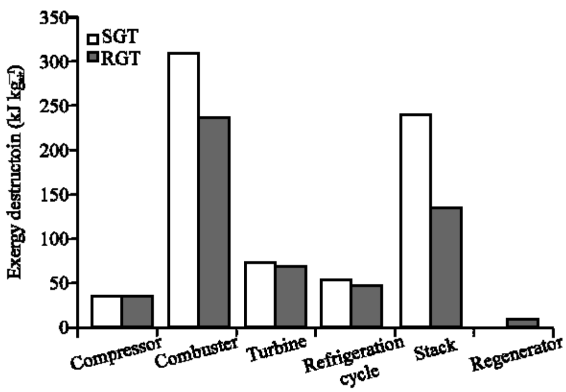

Fig. 20: Exergy destruction at the various components of the SGT and RGT cycles with inlet cooling

that decreasing T_0 causes higher pressure ratios to maximize w_{net} and $\eta_{I,cycle}$. Comparing the results for SGT and RGT cycles, reveals a similar variation of r_c with T_0 for maximum w_{net} and $\eta_{I,cycle}$. However, the pressure ratio for maximum $\eta_{I,cycle}$ in SGT is significantly greater than the one in RGT.

Exergy destruction: Exergy destruction at the various components of the SGT and RGT cycles with inlet air cooling is shown in Fig. 20. For Fig. 20 TIT = 1200°C, r_c = 12 and T_{amb} = 25°C.

As it shows the combustor has the largest exergy destruction and the irreversibility in the absorption refrigeration cycle is very low as compared to that of the gas turbine. Also, stack loss in RGT cycle is considerably lower than that in SGT cycle.

Effect of cooling on inlet air density: In all the results regarding the variation of w_{net} which previously discussed, one important fact has to be considered. As volume flow rate of the inlet air to the gas turbine is constant, decreasing air temperature will increase the

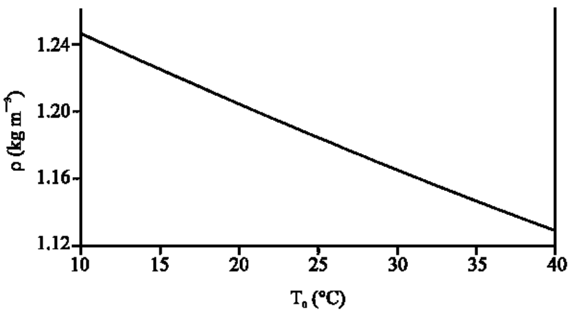

Fig. 21: Variation of air density with temperature

Table 2: The percentage enhancement of the SGT for each 10°C decrease of T_0

TIT (°C)	r_c	Enhancement of w_{net} (%)	Enhancement of $\dot{W}_{net}$ (%)	Enhancement of $\eta_{I,cycle}$ (%)
1100	8	3.7	7.3	2.0
1100	12	4.8	8.4	2.5
1100	18	6.6	10.2	3.4
1100	24	8.8	12.4	4.6
1400	8	2.8	6.2	1.7
1400	12	3.3	6.8	1.9
1400	18	4.2	7.7	2.3
1400	24	5.1	8.7	2.6

Table 3: The percentage enhancement of the RGT for each 10°C decrease of T_0

TIT (°C)	r_c	Enhancement of w_{net} (%)	Enhancement of $\dot{W}_{net}$ (%)	Enhancement of $\eta_{I,cycle}$ (%)
1100	8	3.6	7.1	3.2
1100	12	4.6	8.2	4.1
1100	18	6.2	9.8	5.5
1100	24	8.1	11.7	7.2
1400	8	2.6	6.1	2.4
1400	12	3.1	6.7	2.8
1400	18	3.9	7.5	3.4
1400	24	4.7	8.2	4.1

density and consequently the output power of the gas turbine. This increase is shown in Fig. 21. Therefore, the reason for output power increase is two fold: (1) increase in specific net work because of a decrease in compressor specific work and (2) an increase of mass flow rate.

Enhancements of output power and efficiency: Using absorption refrigeration for inlet air cooling in SGT and RGT cycles the percentage enhancement of operating parameters for each 10°C decrease of inlet air temperature with different r_c s and two TIT s are presented in Table 2 and 3.

According to Table 2 and 3 as, it can be seen from the corresponding figures the enhancement in w_{net}, $\dot{W}_{net}$ and $\eta_{I,cycle}$ of both SGT and RGT cycles due to inlet air cooling, is higher in high r_c s and low TIT s. Also calculations show that the percentage enhancement of the first and second law efficiencies is identical.

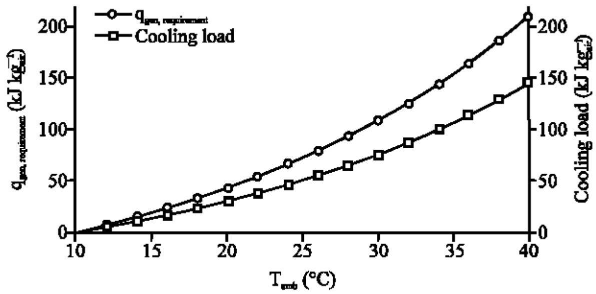

Fig. 22: Required thermal energy and cooling load to cool the ambient air to 10°C in compressor inlet

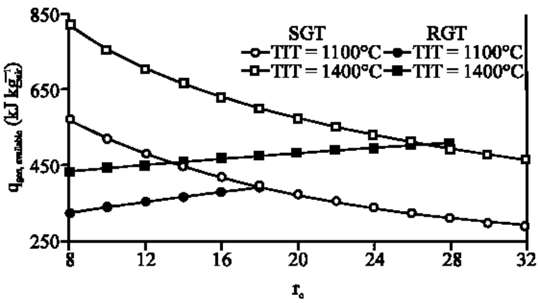

Fig. 23: Available thermal energy in exhaust gases

Available and required energy for refrigeration cycle: Here, the required thermal energy in the generator and cooling load in the evaporator of the refrigeration cycle for cooling the ambient air (Fig. 22) and the energy content of exhaust gases (Fig. 23) are computed and it is concluded that the amount of energy available in exhaust gases is always more than enough to run the refrigeration cycle. Also, considering the required cooling load which is shown in Fig. 22, the installation cost of the absorption chiller can be estimated.

With the pressure ratios higher than 18 when TIT = 1100°C and 28 when TIT = 1400°C the regenerator is useless, because utilizing regenerator will decrease the cycle efficiency.

CONCLUSIONS

These results have been obtained for SGT and RGT cycles with absorption inlet air cooling:

- For each 10°C decrease of inlet air temperature, the output power increases around 6-12% and the first and second law efficiencies increase around 2-7%. The former finding is in good agreement with those reported by Amery and Hejazi (2004) and Dawoud *et al.* (2005)
- A better enhancement in output power and efficiency was obtained at higher pressure ratios and lower turbine inlet temperatures
- The percentage enhancement of the first and second law efficiencies is identical
- The efficiency enhancement of the RGT cycle is higher than that for SGT one, but corresponding the output power a reverse result was obtained

Also the results show that:

- The largest irreversibility occurs in the combustion chamber and the irreversibility in the absorption refrigeration cycle is very low as compared to that of the gas turbine
- Variation of the first and second law efficiencies of the SGT and RGT cycles with respect to the parameters such as compressor inlet air temperature, pressure ratio and turbine inlet temperature is similar and a maximum first law efficiency corresponds to a maximum second law efficiency
- At low pressure ratios in SGT cycle and high pressure ratios in RGT cycle just a part of the available energy in exhaust gases is consumed in refrigeration cycle and the rest can be used for some other purposes such as steam injection, cogeneration and etc.

NOMENCLATURE

$\dot{E}$: Exergy (kW)
f : Mass basis fuel-air ratio
GT : Gas Turbine
h : Enthalpy (kJ kg^{-1})
i : Irreversibility (kJ kg^{-1})
$\dot{I}$: Irreversibility (kW)
LHV : Lower Heat Value of fuel (kJ kg^{-1})
$\dot{m}$: Mass flow rate (kg sec^{-1})
$\dot{Q}$: Heat transfer (kW)
r : Pressure ratio
RC : Refrigeration Cycle
RGT : Regenerative Gas Turbine
RH : Relative Humidity of inlet air
SGT : Simple Gas Turbine
T : Temperature (°C)
TIT : Turbine Inlet Temperature (°C)
w : Specific work (kJ kg^{-1})
$\dot{W}$: Work (kW)
ε : Specific exergy (kJ kg^{-1})
η : Efficiency

SUBSCRIPTS

amb	:	Ambient
c	:	Compressor
cv	:	Control volume
ch	:	Chemical
gen	:	Generator of refrigeration cycle
in	:	Inlet stream
out	:	Outlet stream
ph	:	Physical
ρ	:	Density (kg m^{-3})
0	:	Compressor inlet
I	:	First law
II	:	Second law

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