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 a 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 Boonmasa 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
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:

\[ \sum m_i - \sum m_{out} = 0 \]  

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

\[ \sum (m h)_{in} - \sum (m h)_{out} + Q_{in} - W_{net} = 0 \]  

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

\[ 1 = \sum E_{in} - \sum E_{out} \]  

where, \( \Sigma E_{in} \) and \( \Sigma E_{out} \) 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:

\[ e = e_{th} + e_{ch} \]  

**Table 1:** Assumptions for the gas turbine and refrigeration cycles

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Assumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine polytropic efficiency</td>
<td>0.87</td>
</tr>
<tr>
<td>Compressor polytropic efficiency</td>
<td>0.86</td>
</tr>
<tr>
<td>Mechanical efficiency</td>
<td>0.98</td>
</tr>
<tr>
<td>Combustor efficiency</td>
<td>0.99</td>
</tr>
<tr>
<td>Regenerator effectiveness</td>
<td>0.75</td>
</tr>
<tr>
<td>Refrigeration cycle heat exchanger effectiveness</td>
<td>0.70</td>
</tr>
<tr>
<td>Pressure loss of combustor</td>
<td>5%</td>
</tr>
<tr>
<td>Pressure loss of regenerator</td>
<td>2%</td>
</tr>
<tr>
<td>Pressure loss of air in the evaporator</td>
<td>2%</td>
</tr>
<tr>
<td>Generator temperature</td>
<td>105°C</td>
</tr>
<tr>
<td>Absorber/Condenser temperature</td>
<td>47°C</td>
</tr>
<tr>
<td>Evaporator temperature</td>
<td>8°C</td>
</tr>
<tr>
<td>Compressor inlet air temperature/RH</td>
<td>10°C/60%</td>
</tr>
<tr>
<td>LHV of fuel</td>
<td>802361 kJ/kmol</td>
</tr>
<tr>
<td>Chemical energy of fuel</td>
<td>85651.0 kJ/kmol</td>
</tr>
</tbody>
</table>

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

\[ \eta_{cy} = \frac{W_{net}}{W_{LHV}} \]  

\[ \eta_{E,nets} = \frac{W_{net}}{e_{fuel}} \]  

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 $\epsilon_{\text{adj}}$ 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 $r_c$ and $\text{TIT}$s.

**Analyzing the effects of $T_o$ 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 $\text{TIT} = 1200^\circ$C.

From Fig. 2, it is found that for both SGT and RGT cycles, $w_n$ is increased as $T_o$ 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 $T_o$ for both SGT and RGT cycles. Comparatively higher increase of efficiency with a decrease in $T_o$ is indicated when the pressure ratio is high. Also Fig. 3 reveals a more pronounced effect of $T_o$ 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 $T_o$ and $r_c$.

Figure 5 shows a negligible effect of $T_o$ 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 $T_o$ and $r_c$ are high.

**Analyzing the effects of $T_o$ and $\text{TIT}$:** 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 $T_o$ on $w_n$, $\eta_{\text{cycle}}$, and $\eta_{\text{fuel}}$ at low $\text{TIT}$s.

Figure 7 and 8 indicate that increasing $\text{TIT}$ 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 $\text{TIT}$ increases. However, for RGT cycle increasing $\text{TIT}$ will 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 $\text{TIT}$ increases. This is due to the increase of irreversibility in combustor and turbine when $\text{TIT}$ is increased. Also Fig. 9 shows that the value of irreversibility of RGT is lower than that for SGT, particularly at high $\text{TIT}$s. This is because of this fact.
that the irreversibility in the combustor of RGT cycle is lower than the one in SGT.

**Comparing the effects of $T_o$ 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 $\eta_{cycle}$. As it can be seen lowering $T_o$ 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 $\eta_{cycle}$ with $T_o$ and/or TIT are almost similar. However, $T_o$ is more effective on $\eta_{cycle}$ while TIT has strong influence on $w_{net}$. At practical pressure ratios (around 14), $\eta_{cycle}$ 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 $T_o$ 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 $\eta_{cycle}$ 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 $\eta_{cycle}$ of the SGT at TIT = 1200°C

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

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 $\eta_{cycle}$ 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 $T_o$, $w_{net}$ and $\eta_{\text{cycle}}$ of the RGT at $TIT = 1200^\circ C$

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

Like the case of SGT cycle (Fig. 13, 14), at different conditions of $T_1$ and $TIT$ there are some pressure ratios with which $w_{net}$ and $\eta_{\text{cycle}}$ 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 bring 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 \( \eta_{cycle} \). Comparing the results for SGT and RGT cycles, reveals a similar variation of \( r_1 \) with \( T_0 \) for maximum \( w_{net} \) and \( \eta_{cycle} \). However, the pressure ratio for maximum \( \eta_{cycle} \) 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, \( r_1 = 12 \) 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 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 \( r_1 \) 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 \( w_{net} \) and \( \eta_{cycle} \) of both SGT and RGT cycles due to inlet air cooling, is higher in high \( r_1 \) 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**

- E : Exergy (kW)
- f : Mass basis fuel-air ratio
- GT : Gas Turbine
- h : Enthalpy (kJ kg⁻¹)
- i : Irreversibility (kJ kg⁻¹)
- f : Irreversibility (kW)
- LHV : Lower Heat Value of fuel (kJ kg⁻¹)
- m : Mass flow rate (kg sec⁻¹)
- 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)
- W : Work (kW)
- e : Specific exergy (kJ kg⁻¹)
- η : Efficiency
SUBSCRIPTS

amb : Ambient
c : Compressor
ev : Control volume
ch : Chemical
gen : Generator of refrigeration cycle
in : Inlet stream
out : Outlet stream
ph : Physical
ρ : Density (kg m⁻³)
C : Compressor inlet
I : First law
II : Second law

REFERENCES


