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Energy and Exergy Analysis of Reverse Brayton Refrigerator for Gas Turbine Power Boosting

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1. Introduction

The output of Gas turbine (GT) power plants operating in the arid and semiarid zones is affected by weather conditions where the warm air at the compressor intake decreases the air density and hence reduces the net output power far below the ISO standard (15 ^{o}C and 60% relative humidity). The power degradation reaches an average of 7% for an increase in temperature by only 10°C above the 15 ^{o}C ISO standard. Furthermore; in hot summer days the plants are overloaded due to the increase in demand at peak periods, to meet the extensive use of air-conditioning and refrigeration equipment. The current techniques to cool the air at the compressor intake may be classified into two categories; direct methods employing evaporative cooling and indirect methods, where two loops refrigeration machines are used. Erickson (2003) reviewed the relative merits, advantages and disadvantages of the two approaches; Cortes and Willems (2003), and Darmadhkari and Andrepont (2004) examined the current inlet air cooling technology and its economic impact on the energy market.

In direct cooling methods water is sprayed at the compressor inlet bell mouth either through flexuous media (cellulose fiber) or fogging (droplets size in the order of 20 micron) into the air stream, Ameri *et al*. (2004). All spray cooling systems lower the intake temperature close to the ambient wet bulb temperature; therefore the use of the spray cooling is inefficient in coastal areas with high air humidity. Ameri *et al*. (2004) reported 13% power improvement for air relative humidity below 15% and dry bulb temperature between 31°C and 39°C. In addition to the effect of the ambient air humidity, the successful use of the direct method depends on the spray nozzles characteristics, Meher-Homji *et al* (2002) and droplets size, Bettocchi *et al*. (1995) and Meher-Homji and Mee, (1999). In evaporative cooling there is, to some extent, water droplets carry over problem, addressed by Tillman *et al* (2005), which is hazardous for compressor blades. Therefore, evaporative cooling methods are of limited use in humid coastal areas. Alhazmy *et al* (2006) studied two types of direct cooling methods: direct mechanical refrigeration and evaporative water spray cooler, for hot and humid weather. They calculated the performance improvement for ranges of ambient temperature and relative humidity, and their results indicated that the direct mechanical refrigeration increased the daily power output by 6.77% versus 2.5% for spray water cooling.

Indirect cooling by mechanical refrigeration methods can reduce the air temperature to any desirable value, even below the 15ºC, regardless of the ambient humidity. There are two common approaches for air chilling: a) use of refrigeration units via chilled water coils, b) use of exhaust heat-powered absorption machines. As reported by Elliot (2001), application of mechanical air-cooling increases the net power on the expense of the thermal efficiency, 6% power boosting for 10ºC drop in the inlet air temperature. Using absorption machines was examined for inlet air-cooling of cogeneration plants, Ondrays *et al* (1991), while Kakaras *et al.* (2004) presented a simulation model for NH_3 waste heat driven absorption machine air cooler. A drawback of the mechanical chilling is the risk of ice formation either as ice crystals in the air or as solidified layer on surfaces, such as the bell mouth or inlet guide vanes, Stewart and Patrick, (2000).

Several studies have compared the evaporative and mechanical cooling methods; with better performance for mechanical cooling. Mercer (2002) stated that evaporative cooling has increased the GT power by 10–15%, while the improvement for refrigeration chillers has reached 25%. Alhazmy and Najjar (2004) concluded that the power boosting varied between 1-7% for spray cooling but reached10-18% for indirect air cooling. In a recent study by Alhazmy *et al* (2006), they introduced two generic dimensionless terms (power gain ratio (*PGR*) and thermal efficiency change factor (*TEC*)) for assessment of intake air cooling systems. They presented the results in general dimensionless working charts covering a wide range of working conditions. Zadpoor and Golshan (2006) discussed the effect of using desiccant-based evaporative cooling on gas turbine power output. They have developed a computer program to simulate the GT cycle and the NOx emission and showed that the power output could be increased by 2.1%. In another development Erickson (2003 & 2005) suggested combination of the methods combining waste driven absorption cooling with water injection into the combustion air for power boosting; the concept was termed the "*power fogger cycle*". A novel approach has been presented by Zaki *et al* (2007), where a reverse Joule-Brayton air cycle was used to reduce the air temperature at the compressor inlet. Their coupled cycle showed a range of parameters, where both the power and thermal efficiency can be simultaneously improved.

As revealed in the above account abundant studies are concerned with the first law analysis of air intake cooling but those focuses on the second law analysis are limited. The basics of the second law analysis have been established and employed on variety of thermal systems by number of researchers. Bejan (1987, 1997), Bejan et al 1996, Rosen and Dincer (2003 a, and 2003 b) Ranasinghe, *et al* (1987), Zaragut *et.al* (1988), Kotas *et al* (1991) and Jassim (2003a, 2003b, 2004, 2005 and 2006), Khir *et al* (2007) have dealt extensively with various aspects of heat transfer processes. Chen *et al* (1997) analyzed the performance of a regenerative closed Brayton power cycle then extended the method to a Brayton refrigeration cycle, Chen *et al* (1999). The analysis considered all the irreversibilities associated with heat transfer processes. The exergy analysis of Brayton refrigeration cycle has been considered by Chen and Su (2005) to set a condition for the maximum exergetic efficiency while Tyagi *et al* (2006) presented parametric study where the internal and external irreversibilities were considered. The maximum ecological function, which was defined as the power output minus the power loss was determined for Brayton cycle by Huang *et al* (2000). Their exergy analysis was based on an ecological optimization criterion and was carried out for an irreversible Brayton cycle with external heat source.

The objective of the present analysis is to investigate the potential of boosting the power output of gas turbine plants operating in hot humid ambiance. The previously proposed coupled Brayton and reverse Brayton refrigeration cycles is analyzed employing both the energy and exergy analysis. Both the thermal and exergetic efficiencies are determined and the exergy destruction terms are evaluated.

2. Energy analysis

2.1 Brayton power cycle

Details of the first law energy analysis have been presented in a previous study by Zaki *et al* (2007) but the basic equations are briefly given here. Figure 1 shows the components of a coupled gas turbine cycle with Brayton refrigeration cycle. The power cycle is represented by states 1-2-3-4 and the reverse Brayton refrigeration cycle is represented by states 1-6-7-8-1. Portion of the compressed air $\alpha \dot{m}_1$ at pressure P_6 is extracted from the mainstream and cooled in a heat exchanger to T_7 then expands to the atmospheric pressure and T_8 as seen in Fig.1. The ambient intake air stream at T_o mixes with the cold stream at T_8 before entering the compressor.

Figure 2 shows the combined cycle on the T-s diagram; states o-$\overline{2}$-3-4 represent the power cycle without cooling, while the power cycle with air inlet cooling is presented by states 1-2-3-4-1. States 1-6-7-8 present the reverse Brayton refrigeration cycle.

In the mixing chamber ambient air at $\dot{m}_o$, T_o and ω_o enters the chamber and mixes with the cold air stream having mass flow rate of $\alpha \dot{m}_1$ at T_8. Air leaves the chamber at T_1, which depends on the ambient air conditions and the extraction ratio α.

The mass and energy balance for the mixing chamber gives the compressor inlet temperature as;

$$T_1 = \frac{(1-a)c_{po}T_o + a\,c_{p8}T_8}{c_{p1}} \tag{1}$$

Air leaves the compressor at $P_6 = x\,P_1$ flows through the Brayton refrigerator and the rest at $P_2 = r\,P_1$ as the working fluid for the power cycle, x is defined as the extraction pressure ratio and r is the pressure ratio.

The temperature of the air leaving the compressor at states 6 and 2 can be estimated assuming irreversible compression processes between states 1-2 and 1-6 and introducing isentropic compressor efficiency so that;

$$T_6 = T_1 + \frac{T_1}{\eta_{cx}}\left(x^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{2}$$

and

$$T_2 = T_1 + \frac{T_1}{\eta_{cr}}\left(r^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{3}$$

where, η_c is the compressor isentropic efficiency that can be estimated at any pressure z from Korakianitis and Wilson, (1994)

$$\eta_{cz} = 1 - \left(0.04 + \frac{z-1}{150} \right) , \text{ where } z \text{ is either } x \text{ or } r \tag{4}$$

The compression power between states 1 and 2 with extraction at state 6, separating the effects of the dry air and water vapor can be written as, Zaki *et al* (2007):

$$\dot{W}_c = \dot{m}_a \left[c_{pa} \frac{T_1}{\eta_{cr}} \left(r^{\frac{\gamma-1}{\gamma}} - 1 \right) + \omega_1 \left(h_{v2} - h_{v1} \right) + \left(\frac{a}{1-a} \right) \left(c_{pa} \frac{T_1}{\eta_{cx}} \left(x^{\frac{\gamma-1}{\gamma}} - 1 \right) + \omega_1 \left(h_{v6} - h_{v1} \right) \right) \right] \tag{5}$$

where h_{vn} is the enthalpy of the saturated water vapor at the indicated state n. Equation 5 is a general expression for the compressor work for wet air; for dry air conditions $\omega_1 = 0$ and for stand alone GT $\alpha = 0$.

Heat balance about the combustion chamber gives the heat rate supplied by fuel combustion (net calorific value *NCV*) as:

$$\dot{Q}_{cc} = \dot{m}_f NCV = \left(\dot{m}_a + \dot{m}_f \right) c_{pg} T_3 - \dot{m}_a c_{pa} T_2 + \dot{m}_v \left(h_{v3} - h_{v2} \right) \tag{6}$$

where h_{v2} and h_{v3} are the enthalpies of water vapor at the combustion chamber inlet and exit states respectively and f is the fuel to air ratio $f = \dot{m}_f / \dot{m}_a$ (related to the dry air flow rate). The total gases mass flow rate at the turbine inlet $\dot{m}_t = \dot{m}_a (1 + \omega_1 + f)$.

Substituting for T_2 from equation 3 gives the cycle heat input as:

$$\dot{Q}_{cc} = \dot{m}_a T_1 \left[(1+f) c_{pg} \frac{T_3}{T_1} - c_{pa} \left(\frac{r^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{cr}} + 1 \right) + \frac{\omega_1}{T_1} \left(h_{v3} - h_{v2} \right) \right] \tag{7}$$

The power produced by the turbine due to expansion of gases between states 3 and 4, Fig.1, is

$$\dot{W}_t = \dot{m}_a \left(1 + \omega_1 + f \right) c_{pg} \eta_t T_3 \left(1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}} \right) \tag{8}$$

The turbine isentropic efficiency may be estimated using the practical relation recommended by Korakianitis and Wilson (1994) as:

$$\eta_t = 1 - \left(0.03 + \frac{r-1}{180} \right) \tag{9}$$

Since, the gas turbine is almost constant volume machine at a specific rotating speed, the inlet air volumetric flow rate, $\dot{V}_a$ is fixed regardless of the intake air conditions. Equation 7 can be written in terms of the volumetric flow rate at the compressor inlet state by replacing

$\dot{m}_a$ by $\rho_a \dot{V}_a$. In the present analysis the moist air density ρ_a is assumed function of T_1 and ω_1. The Engineering Equation Solver (*EES*) software (Klein and Alvarado, 2004) has been used to calculate the wet air properties.

2.2 Brayton refrigeration cycle analysis

It is noted that the extraction pressure ratio x is the main parameter that determines the cold air temperature T_8 achievable by the air refrigeration cycle. The hot compressed air at P_6 and T_6 rejects its heat through a heat exchanger to cooling water. Similar to standard compressor intercoolers, state 7 will have the same pressure as P_6 for ideal case while T_8 depends on the cooling process. Irreversible expansion process between 7 and 8 with expansion efficiency η_{exp} yields,

$$T_8 = T_7 - T_7 \eta_{\exp} \left[1 - \left(\frac{1}{x} \right)^{\frac{\gamma-1}{\gamma}} \right] \tag{10}$$

The power output of the expander, $\dot{W}_{exp}$ is

$$\dot{W}_{exp} = a\,\dot{m}_1 (h_7 - h_8) \tag{11}$$

The heat release $\dot{Q}_{out}$ between states 6 and 7 is computed by:

$$\dot{Q}_{out} = \alpha\,\dot{m}_1 (h_7 - h_6) \tag{12}$$

Since many of the desalination plants in the Gulf area use dual purpose combined GT units for water and power production, it is suggested here to utilize the rejected heat ($\dot{Q}_{out}$, Fig. 1) for brine heating or any industrial process that requires low grade heat. In general, the lower limit for T_7 is determined by the ambient temperature.

Fig. 1. A schematic of a gas turbine with an air cooler cycle

The cold stream flow rate $a\dot{m}_1$ proceeds to the mixing chamber to cool down the air at the compressor intake. The mixture temperature T_1 depends on the mass flow rate and the temperature of each stream as seen in Eq. 6.

Fig. 2. *T-s* diagram for the proposed cycle

3. Exergy analysis

The exergy of a system is the maximum work obtainable as the system comes to equilibrium with the surrounding. The first law analysis (section 2) did not give information on heat availability at different temperatures. It is the second law which asserts that from engineering perspective. Therefore, exergy method is a technique based on both the first and second law of thermodynamics providing the locations where the exergy destruction and irreversibility are most. Ways and means can then be explored to reduce these exergy destructions to the practical minimum values.

The exergy destruction is a measure of thermodynamic imperfection of a process and is expressed in terms of lost work potential. In general, the expression for exergy destruction as articulated by Kotas (1995) is;

$$\dot{I} = T_o \left[\left(\dot{S}_{out} - \dot{S}_{in} \right) - \sum_{i=1}^{n} \frac{\dot{Q}_i}{T_i} \right] = T_o \dot{\Pi} \geq 0 \tag{13}$$

The exergy balance for an open system is,

$$\dot{E}_{in} + \dot{E}^Q = \dot{E}_{out} + \dot{W} + \dot{I} \tag{14}$$

where $\dot{E}^Q$ is the exergy associated with a given heat transfer rate, is termed as thermal exergy flow (kW). Quantities of lost work or rate of exergy destruction due to irreversibility can be estimated for each component of the coupled cycles as follows.

3.1 Mixing chamber

The rate of exergy destruction in the mixing chamber includes different forms of exergy; physical due to changes in the properties during mixing and that due to chemical reactions if any. The *physical exergy* of a mixture of N components can be evaluated from Jassim, *et al* 2004 as,

$$\left(\tilde{\varepsilon}_{ph}\right)_M = \sum_{j=1}^{N} y_j\, \tilde{\varepsilon}_j^{\Delta T} + \tilde{R}\, T_o\, \ell n\!\left(\frac{P}{P_o}\right) \tag{15a}$$

where P is the total pressure of the mixture. The equation may be written in another form, using tabulated values of properties ($\tilde{c}_p^h$ and $\tilde{c}_p^s$) as;

$$\left(\tilde{\varepsilon}_{ph}\right)_M = \sum_{j=1}^{N} y_j\left[\tilde{c}_p^h\left(T - T_o\right) - T_o\,\tilde{c}_p^s\, \ell n\!\left(\frac{T}{T_o}\right)\right]_j + \tilde{R}\, T_o\, \ell n\!\left(\frac{P}{P_o}\right) \quad \text{(kJ/kmol)} \tag{15b}$$

The mean physical exergies for the mixing streams are calculated using equation 15b. The values of mean molar isobaric exergy capacities for enthalpy $\left(\tilde{c}_{p,j}^h\right)$ and entropy $\left(\tilde{c}_{p,j}^s\right)$ have been evaluated from (Kotas, 1995, Table D3).

The chemical exergy of a mixture $\tilde{\varepsilon}_{oM}$ due to changes in number of moles is given by the following expression (Kotas, 1995)

$$\tilde{\varepsilon}_{oM} = \sum_j y_j\, \tilde{\varepsilon}_{oj} + \tilde{R}\, T_o \sum_j y_j\, \ell n\, x_j \quad \text{(kJ/kmol)} \tag{16}$$

where y_j is the mole fraction of component j in a mixture

$\tilde{\varepsilon}_o$ is the molar chemical exergy (*kJ/kmol*) evaluated from (Kotas, 1995, Table D3).

The irreversibility involved in mixing hot and cold air streams as shown in Fig. 1 assuming adiabatic mixing process ($\dot{Q}_o$ = 0) is,

$$\dot{I}_{mc} = \dot{E}_8 + \dot{E}_o - \dot{E}_1 \tag{17a}$$

or

$$\dot{I}_{mc} = a\dot{m}_1\,\varepsilon_8 + \dot{m}_o\,\varepsilon_o - \dot{m}_1\,\varepsilon_1 \tag{17b}$$

In general, the specific exergy component for each of the terms in Eq. 17b is,

$$\varepsilon = \varepsilon_o + \varepsilon_{ph} \tag{17c}$$

The values of ε_{ph} and ε_o can be calculated from Eqs. (15b) and (16), respectively. Hence, equations 17b and 17c give the total rate of exergy losses in the mixing chamber as,

$$\dot{I}_{mc} = a\dot{m}_1\left(\varepsilon_{o,8} + \varepsilon_{ph,8}\right)\!\big/\tilde{m}_8 + \dot{m}_o\left(\varepsilon_{o,o} + \varepsilon_{ph,o}\right)\!\big/\tilde{m}_o - \dot{m}_1\left(\varepsilon_{o,1} + \varepsilon_{ph,1}\right)\!\big/\tilde{m}_1 \tag{17d}$$

$$\dot{I}_{mc} = \dot{m}_a \left(1 + \omega_1\right) \left[\left(\frac{a}{1-a}\right) \frac{\left(\varepsilon_{o,8} + \varepsilon_{ph,8}\right)}{\tilde{m}_8} + \frac{\left(\varepsilon_{o,o} + \varepsilon_{ph,o}\right)}{\tilde{m}_o} - \left(\frac{1}{1-a}\right) \frac{\left(\varepsilon_{o,1} + \varepsilon_{ph,1}\right)}{\tilde{m}_1} \right]$$

3.2 Compressor

The rate of exergy destruction in the compressor (see Figs 1 and 2) for the compression process 1-2, is,

$$\dot{I}_c = \dot{m}_o \, T_o \left(s_2 - s_1\right) + a\left(\frac{\dot{m}_o}{1-a}\right) T_o \left(s_6 - s_1\right) \tag{18}$$

Substituting the entropy change for gases, Cengel and Bolos, 2005 gives

$$\dot{I}_c = \dot{m}_a \left(1 + \omega_1\right) T_o \left[c_{pa} \, \ell n\left(\frac{T_2}{T_1}\right) - R_a \, \ell n\left(\frac{P_2}{P_1}\right) + \left(\frac{a}{1-a}\right)\left(c_{pa} \, \ell n\left(\frac{T_6}{T_1}\right) - R_a \, \ell n\left(\frac{P_6}{P_1}\right)\right) \right] \tag{19}$$

The mixing chamber irreversibility is added to the compressor irreversibility as it is caused by air mixing before entering the compressor. Therefore the compressor effective power is,

$$\dot{W}_{eff,c} = \dot{W}_c + \dot{I}_c + \dot{I}_{mc} \tag{20}$$

3.3 Combustion chamber

The combustion process irreversibility is often accomplished by heat transfer, fluid friction, mixing and chemical reaction. In principle it is difficult to evaluate an absolute value for the contribution of each process to the total irreversibility. However, the process of combustion can be examined by assuming that it takes place under adiabatic conditions and neglect irreversibilities due to friction and mixing. The important term of fuel combustion is the chemical exergy ε_o which has been extensively dealt with by Kotas, 1995. The exergy due to chemical reaction of a combustible component (k) is function of other parameters as;

$$\varepsilon_o = f(\Delta h, \Delta s, T_o, X_k) \tag{21}$$

The subscript k refers to the component's mass (mole) fraction of products composition.

The chemical exergy ε_o may be expressed as percentage of the net calorific value (*NCV*) as,

$$\varepsilon_o = \varphi \, NCV \tag{22}$$

The value of φ is function of the fuel components mass to Carbon ratio, (*O/C*) for Oxygen, (*H/C*) for hydrogen and (*N/C*) for Nitrogen. For liquid fuels additional ratio for the sulfur content is included (*S/C*), and φ is in the order of 1.04 to 1.08. Typical values of φ for some industrial fuels are given in Kotas 1995. Therefore, the rate of exergy loss $T_o \Delta S_o$ in a combustion reaction is;

$$T_o \, \Delta S_o = \dot{m}_f \times NCV \left(\varphi - 1\right) = \dot{m}_a \times f \times NCV \left(\varphi - 1\right) \tag{23}$$

The exergy destruction rate during the combustion process is the entropy change between states 2 and 3 due to chemical combustion reaction is;

$$\dot{I}_{cc} = T_o \left[(S_P)_3 - (S_R)_2 \right] \tag{24a}$$

where $(S_R)_2 = (S_a)_2 + (S_f)_2$ and the subscripts P, R, a, and f represent *products*, *reactants*, *air* and *fuel*, respectively (Nag, 2002). Then

$$\dot{I}_{cc} = T_o \left[(S_p)_3 - \left[(S_a)_2 + (S_f)_2 \right] \right] \tag{24b}$$

Introduce the entropy difference at the reference state ΔS_o as

$$\Delta S_o = (S_P)_o - \left[(S_f)_o + (S_a)_o \right] \tag{24c}$$

The exergy destruction rate becomes

$$\dot{I}_{cc} = T_o \left\{ \left[(S_P)_3 - (S_P)_o \right] - \left[(S_a)_2 - (S_a)_o \right] + \Delta S_o \right\} \tag{24d}$$

Substitute for the entropy change for reactants and products gives;

$$\dot{I}_{cc} = \dot{m}_a T_o \left\{ \begin{array}{l} (1 + f + \omega_1) \left[c_{pg} \, \ell n \left(\dfrac{T_3}{T_o} \right) - R_g \, \ell n \left(\dfrac{P_3}{P_o} \right) \right] - \\ (1 + \omega_1) \left[c_{pa} \, \ell n \left(\dfrac{T_2}{T_o} \right) - R_a \, \ell n \left(\dfrac{P_2}{P_o} \right) \right] \end{array} \right\} + T_o \Delta S_o \tag{24e}$$

where $R_g = \dfrac{c_{pg}(\gamma - 1)}{\gamma}$

The total energy input $\dot{Q}_{eff,cc}$ to the combustion chamber is then

$$\dot{Q}_{eff,cc} = \dot{Q}_{cc} + \dot{I}_{cc} \tag{25}$$

3.4 Gas turbine

The rate of exergy destruction in the gas turbine is,

$$\dot{I}_t = \dot{m}_a (1 + f + \omega_1) T_o (s_4 - s_3)$$

$$= \dot{m}_a (1 + f + \omega_1) T_o \left[c_{pg} \, \ell n \left(\dfrac{T_4}{T_3} \right) - R_g \, \ell n \left(\dfrac{P_4}{P_3} \right) \right] \tag{26}$$

The effective power output of the turbine can be expressed in terms of exergy destruction rate as;

$$\dot{W}_{eff,t} = \dot{W}_t - \dot{I}_t \tag{27}$$

3.5 Heat exchanger

During the air cooling process 6-7 (Fig. 1), the rate of exergy destruction in the heat exchanger is,

$$\dot{I}_{HX} = a\dot{m}_1 T_o \left(s_6 - s_7 \right) + \frac{T_o}{T_6} \dot{Q}_{out} \tag{28}$$

where $\dot{Q}_{out}$ is the rate of heat rejection from the heat exchanger at T_6, $= \alpha\dot{m}_1 \left(h_6 - h_7 \right)$.

The net energy output in the *HX* is then;

$$\dot{Q}_{eff,HX} = \dot{Q}_{out} + \dot{I}_{HX} \tag{29}$$

3.6 Expander

The rate of exergy destruction in the expander (process 7-8) is similar to that in the turbine and can be expressed as,

$$\dot{I}_{exp} = a\dot{m}_1 T_o \left(s_8 - s_7 \right)$$

$$= a\left(\frac{\dot{m}_a}{1-a} \right) \left(1 + \omega_1 \right) T_o \left[c_{pa} \ell n \left(\frac{T_8}{T_7} \right) - R_a \ell n \left(\frac{P_8}{P_7} \right) \right] \tag{30}$$

The effective power output of the expander is then

$$\dot{W}_{eff,exp} = \dot{W}_{exp} - \dot{I}_{exp} \tag{31}$$

To that point the exergy destruction have been evaluated for all the system components and the total rate of exergy losses in the plant, Fig. 1 is;

$$\sum \dot{I} = + \dot{I}_{mc} + \dot{I}_c + \dot{I}_{cc} + \dot{I}_t + \dot{I}_{HX} + \dot{I}_{exp} \tag{32}$$

4. Performance of the integrated system

For the proposed cycle the net power output and heat input, can be easily calculated using equations 5, 8 and 11. If the expander power $\dot{W}_{exp}$ is recovered, then the net output of the cycle may be expressed as:

$$\dot{W}_{net,withcooling} = \dot{W}_t - \left(\dot{W}_{comp} - \dot{W}_{exp} \right) \tag{33}$$

An advantage of the present cycle is the availability of useful energy $\dot{Q}_{out}$ that may be utilized for any process. Let us define a term for the useful heat input as the net heat utilized to produce the shaft power, therefore, equations 7 and 12 give,

$$\dot{Q}_{useful} = \dot{Q}_{cc} - \dot{Q}_{out} \qquad (34)$$

An integral useful thermal efficiency $\left(\eta_{th,u}\right)$ may be defined as:

$$\eta_{th,u} = \frac{\dot{W}_{net}}{\dot{Q}_{useful}} \qquad (35)$$

If the GT is in operation without the cooling cycle then $\dot{Q}_{out} \to 0$ and $\dot{W}_{exp} \to 0$, equation 35 leads to the conventional thermal cycle efficiency expression (without cooling $\eta_{th,nocooling}$).

The net power for the GT unit without cooling is obtained by introducing $\alpha = 0$ and $\omega_1 = \omega_o$ in Equation 5 to get $\dot{W}_{c,no\ cooling}$. Similarly the turbine power without cooling $\dot{W}_{t,no\ cooling}$ is obtained by subsisting $\omega_1 = \omega_o$, $T_1 = T_o$ and f is calculated from Eq. 8 using $T_{\bar{2}}$ instead of T_2 Therefore,

$$\dot{W}_{net,no\ cooling} = \left|\dot{W}_t - \dot{W}_c\right|_{no\ cooling} \qquad (36)$$

Adopting the terminology proposed by Alhazmy *et al* (2006) and employed in our previous work Zaki, *et al* (2007) to evaluate the thermal efficiency augmentation; a thermal efficiency change term based on the useful input energy TEC_u is defined as:

$$TEC_u = \frac{\eta_{th,u} - \eta_{th,no\ cooling}}{\eta_{th,no\ cooling}} \times 100\% \qquad (37)$$

The subscript *u* refers to useful indicating that the GT plant is serving adjacent industrial process so that the conventional thermal efficiency definition is not applicable for this condition. For the present parametric analysis let us focus on a simple GT plant just coupled to a Brayton refrigerator without use of the cooling energy; the term $\dot{Q}_{out}$ in Eqs. 34 is eliminated and the performance of the coupled cycles may be determined from a power gain ratio (*PGR*) and thermal efficiency change (*TEC*) terms as:

$$PGR = \frac{\dot{W}_{net,with\ cooling} - \dot{W}_{net,no\ cooling}}{\dot{W}_{net,no\ cooling}} \times 100\% \qquad (38)$$

$$TEC = \frac{\eta_{th,with\ cooling} - \eta_{th,no\ cooling}}{\eta_{th,no\ cooling}} \times 100\% \qquad (39)$$

The *PGR* is a generic term that takes into account all the parameters of the gas turbine and the coupled cooling system. For a stand-alone gas turbine under specific climatic conditions; *PGR* = 0. If Brayton refrigerator is used, the *PGR* increases with the reduction of the intake temperature. However, the *PGR* gives the percentage enhancement in power generation; the *TEC* of a coupled system is an important parameter to describe the fuel utilization effectiveness.

From the second law analysis the performance of a system may be articulated by the exergetic efficiency. For the proposed system define the exergetic efficiency η_{ex} as a ratio of total exergy output $\left(\dot{E}_{out}\right)$ to input $\left(\dot{E}_{in}\right)$ rates as;

$$\eta_{ex} = \frac{\dot{E}_{out}}{\dot{E}_{in}}$$
(40)

Exergy balance for the system shown in Figure (1) can be expressed in the following form,

$$\dot{E}_{out} = \dot{W}_{eff,t} + \dot{W}_{eff,\exp} - \dot{W}_{eff,c}$$
(41)

The total rate of exergy input is

$$\dot{E}_{in} = \dot{Q}_{eff,cc} - \dot{Q}_{eff,HX}$$
(42)

Expressions for the effective power terms $\dot{W}_{eff,c}$, $\dot{W}_{eff,t}$, and $\dot{W}_{eff,\exp}$ are given by equations 20, 27 and 31 respectively, while $\dot{Q}_{eff,cc}$ and $\dot{Q}_{eff,HX}$ are determined from equations 25 and 29.

The irreversibilities of heat exchangers $\left(\dot{I}_{HX}\right)$, expanders $\left(\dot{I}_{\exp}\right)$ and mixing chamber $\left(\dot{I}_{mc}\right)$ are all equal to zero when the GT operates on a stand alone mode.

Similar to the criteria adopted for the energy analysis let us define dimensionless terms as the e*xergetic power gain ratio* (*PGR_ex*) and *exergetic thermal efficiency change* (*TEC_ex*) as:

$$PGR_{ex} = \frac{\left(\dot{E}_{out}\right)_{cooling} - \left(\dot{E}_{out}\right)_{nocooling}}{\left(\dot{E}_{out}\right)_{nocooling}} \times 100\%$$
(43)

$$TEC_{ex} = \frac{\eta_{ex,cooling} - \eta_{ex,nocooling}}{\eta_{ex,nocooling}} \times 100\%$$
(44)

5. Results and discussion

In order to evaluate the integration of Brayton refrigerator for intake air-cooling on the GT performance, a computer program has been developed using EES (Klein and Alvarado 2004) software. The computation procedure was first verified for the benchmark case of simple open cycle with dry air as the working fluid, for which $\alpha = 0$, $\omega_o = 0$ $f = 0$ and assuming isentropic compression and expansion processes, Eqs. 5, 7, 8 and 37 leads to the expression for the thermal efficiency of the air standard Brayton cycle.

$$\eta_{th} = 1 - \frac{1}{r^{(\gamma-1)/\gamma}}$$
(45)

Similarly from the exergy point of view and for the same ideal conditions, all exergy destruction terms (Equations 17d, 24e, 26, 28, 30) and the exergetic efficiency (equation 44) leads to the same expression, Eq. 45.

For the present analysis, the ambient air at Jeddah, Saudi Arabia (Latitude 22.30o N and Longitude 39.15o E) a typical city with over 40 GT plants operating under sever weather conditions was selected (ambient temperature of T_o = 45oC and φ_o = 43.3%). The air temperature at the expander inlet T_7 is determined by the requirements of the heating

process; the coolant flow rate can be controlled to obtain different outlet temperature suitable for the industrial process. For 10K terminal temperature difference the value of T_7 can be fixed to 55°C. Table I shows the range of parameters for the present analysis.

In the previous study by the authors Zaki *et al* (2007), the effect of the extraction pressure ratio (x) and the ratio between the mass rates (α) has been investigated. It has been established that for a fixed extraction pressure (P_6), increasing (α) increases the chilling effect and hence T_1 decreases as seen in Fig. 4. It is possible to adjust the values of α and x to operate the GT close to the ISO standard (288.15 K the dotted line in Figure 4). It is seen that a reasonable range of operation is bounded by $2 < x < 4$ and $0.2 < \alpha < 0.3$ for which the intake temperature drops from 318 K to values between 310 and 288 K. Extending the range to reach temperatures below 273 K is constrained by frost formation at the compressor mouth and expander outlet.

Parameter	Range
Ambient air Max. ambient air temperature, T_a Relative humidity, RH_o Volumetric air flow rate, $\dot{V}_a$ Net Calorific Value, NCV Reference ambient temperature, T_o	318.15 K 43.4 % 1 $m^3\ s^{-1}$ 42500 $kJ\ kg^{-1}$ 298.15 K
Gas Turbine Pressure ratio P_2/P_1 Turbine inlet temperature T_3 Specific heat ratio of gas, γ Specific heat of gas, c_{pg}	 10 1373.15 K 1.333 1.147 kJ kg^{-1} K^{-1}
Air Compressor Standard environment temp., T_o Extraction pressure ratio P_x/P_1 Extracted mass ratio, a Specific heat ratio of air, γ	 298.15 K $2 \rightarrow 9$ $0.1 \rightarrow 0.5$ 1.4
Heat exchanger ΔT	 10 K
Fuel Fuel oils, ϕ	 1.08

Table 1. Range of parameters for the present analysis

Figure 5 shows the *PGR* and *TEC* variations within pressure extraction ratio up to 4 and $0 \leq \alpha \leq 0.5$. It is seen that the power gain increases linearly with α, which means enhanced chilling effect due to the large air mass rate ($\dot{m}_a \, \alpha$) passing through the refrigeration cycle.

As seen in the figure the power is boosted up while the thermal efficiency (Eq. 39) decreases, at x = 3 and 4, due to the increase in fuel consumption rate. The drop in the *TEC* is quite large for high α and x values. For moderate values as α = 0.2 and extraction pressure of 4 bar, the power increases by 9.11%, while the thermal efficiency drops by only 1.34%. This result indicates that passing 20% of the intake air at 4 bars pressure thorough a Reverse Brayton refrigerator increases the power by 9.11% with only drop in thermal efficiency of 1.34%. Therefore, selection of the operation parameters is a trade off between power and efficiency, boosting the power on penalty of the thermal efficiency. It is of interest to determine the parameters at which the power is boosted while the thermal efficiency of the GT stays the same as that for the condition of a stand alone operation. This means that the *TEC* (Eq. 39) approaches zero. Solving Eq. 39 shows that within a narrow range of extraction pressure (x = 2.6 to 2.8) as shown in Figure 3, the plant thermal efficiency is slightly altered from the stand alone efficiency and *TEC* is independent on the extracted mass ratio α. Though, the efficiency does not change appreciably the power gain strongly depends on α.
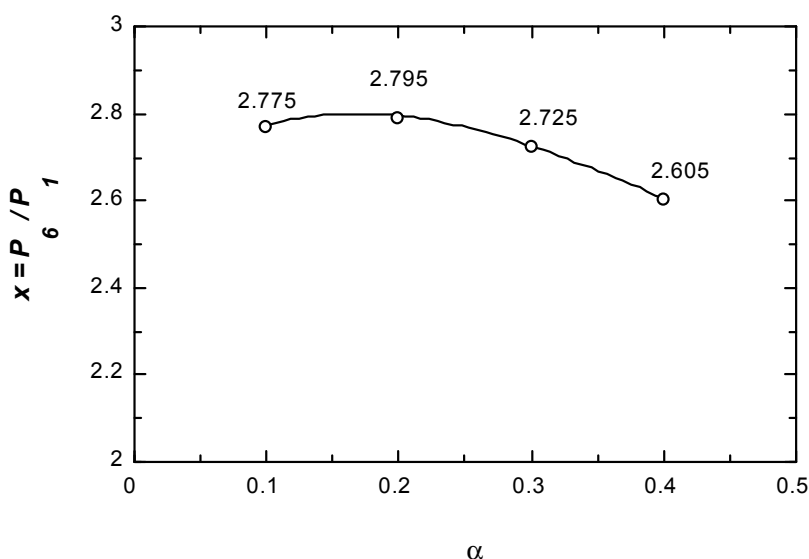
Fig. 3. Solution of Eq. 39, *TEC* = 0 gives the conditions for constant thermal efficiency

The change of the *PGR* within a range of pressure ratios is presented in Fig 6 a, indicating increase in *PGR* up to certain α followed by a descending trend. For a constant value of α the power gain increases with the extraction pressure reaching a peak value then decreases. As seen in Fig. 6 b. the increase in x or P_6 while T_7 is constant means that the outlet temperature from the expander moves from T_8 to $T_{\bar{8}}$, which means lower intake temperature ($T_{\bar{8}} < T_8$) with tendency to increase the *PGR*. On the other hand increase in x results high net compressor work ($\dot{W}_{comp} - \dot{W}_{exp}$) in Equation 33. High net compressor work

reduces the nominator of Eq. 38 and decreases the *PGR*. Therefore, the *PGR* ascending descending trend in Figure 6 a. is expected.
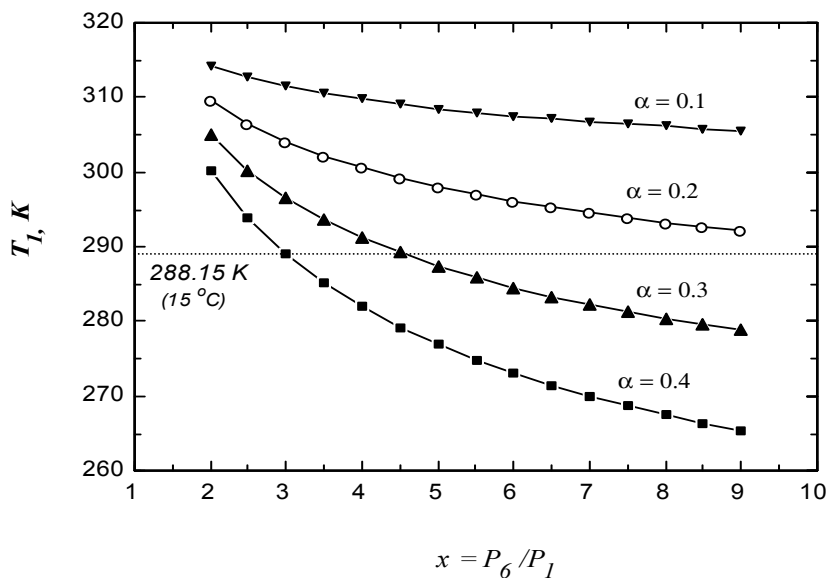

Fig. 4. Effect of extraction pressure x and mass ratio α on the gas turbine intake air temperature.
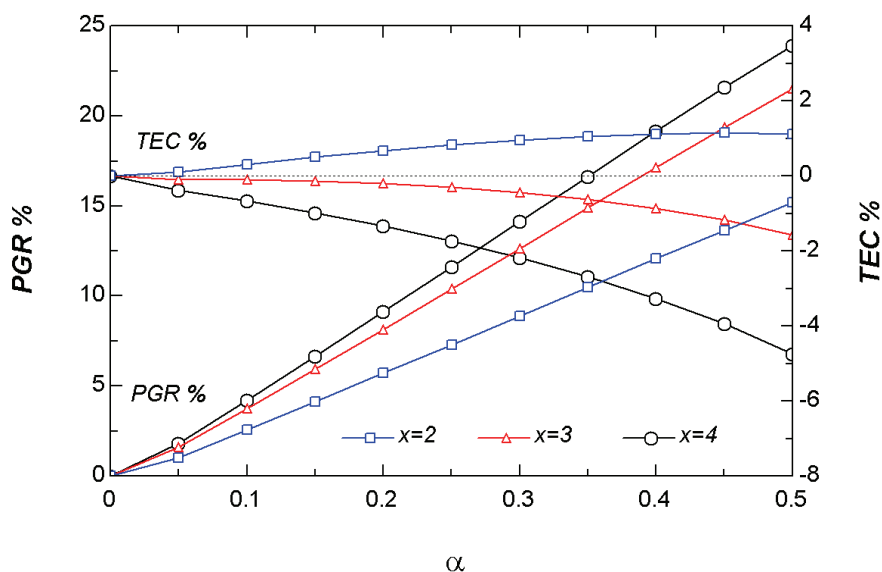

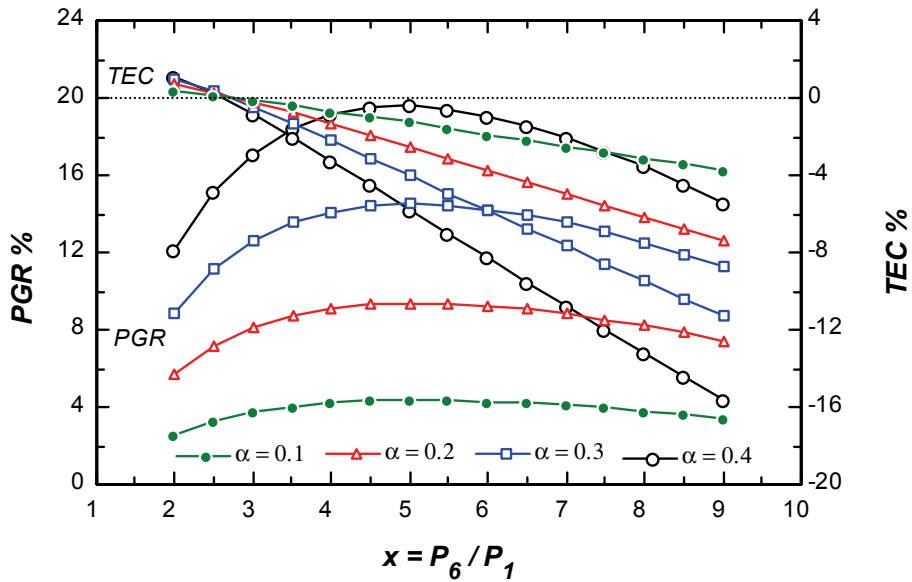Fig. 5. Power enhancement (Eq.38) and thermal efficiency change (Eq. 39) for GT plant, $r = 10$

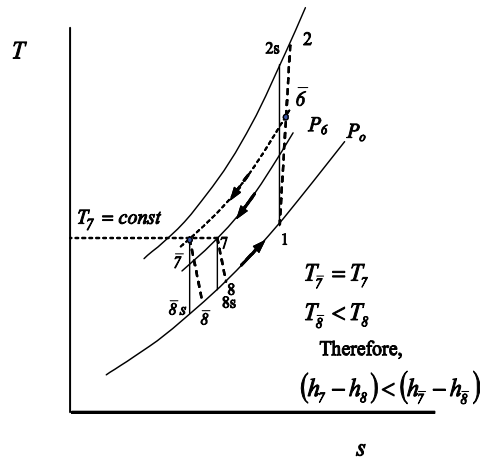Fig. 6a. Power gain ratio and thermal efficiency change factors for a gas turbine cooled by air Brayton refrigerator

Fig. 6b. *T-s* diagram to illustrate the effects of increasing the extraction pressure. Increasing the extraction pressure P_6 reduces the compressor intake temperature and increases the difference ($\dot{W}_{comp} - \dot{W}_{exp}$).

To illustrate the effect of the heat rate ($\dot{Q}_{out}$) during the cooling process (process 6-7, Fig. 1) on the overall system performance, the useful thermal efficiency change (*TEC_u*, Eq. 37) is plotted with the control variables α and x in Figure 7. It is seen that for all operation conditions *TEC_u* is positive, which means that the energy utilized for an industrial process

has been deducted from the heat released by fuel. Though the result was expected, the values here can not be compared to the stand alone plant or even with the case where the cooling water energy is wasted. The figure shows also the *PGR* variation, which is the same whether the energy is utilized or wasted as indicated by Equations 35 and 38.
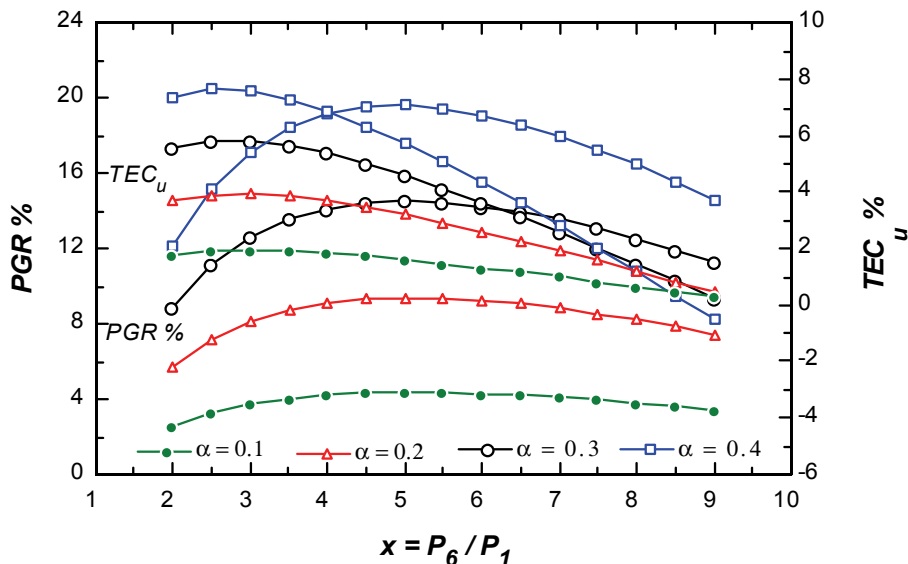
Fig. 7. The useful thermal efficiency change as function of α and x

The results of the energy analysis showed clearly the advantages of the proposed cooling system and the expected magnitude of performance enhancement. However, the practical range of operation parameters was determined the contribution of each of the system components to the irreversibility is only determined through equations 17d, 19, 24e, 26, 28 and 30. Table 2 shows numerical values for the exergy rate of the different components at $x = 3.5$ and variable α.

a	$\dot{I}_c$	$\dot{I}_{cc}$	$\dot{I}_t$	$\dot{I}_{HX}$	$\dot{I}_{exp}$	$\dot{I}_{mc}$	$\dot{I}_{total}$
0.00	21.4	360.8	22.7	0	0	0	0
0.05	21.76	369.9	22.97	17.39	0.3103	0.6484	433
0.10	22.18	381.3	23.3	35.68	0.6644	1.377	464.5
0.15	22.65	393.2	23.64	54.95	1.07	2.133	497.6
0.20	23.2	405.4	24.00	75.28	1.539	2.917	532.4
0.25	23.83	418.2	24.36	96.77	2.082	3.733	568.9
0.30	24.56	431.4	24.74	119.5	2.718	4.581	607.5
0.35	25.42	445.1	25.12	143.8	3.468	5.464	648.3
0.40	26.44	459.3	25.52	169.6	4.361	6.386	691.6
0.45	27.66	474.1	25.94	197.3	5.437	7.349	737.8
0.50	29.14	489.5	26.36	227.1	6.753	8.356	787.2

Table 2. Cycle components irreversibility values (kW) for $x =3.5$

It is seen that the major contribution comes from the combustion chamber where the irreversibility is the highest and presents 62 to 85% of the total irreversibility. The heat exchanger comes next with nearly 5 % to 46% of the magnitude of the combustion chamber. Both the compressor and turbine are of nearly equal magnitude. Therefore, optimization of the system should focus on the irreversibilities in the combustion chamber and heat exchanger. The mixing chamber irreversibility is found to be very small compared to the other components as seen in Table 2 and can be ignored in future studies.
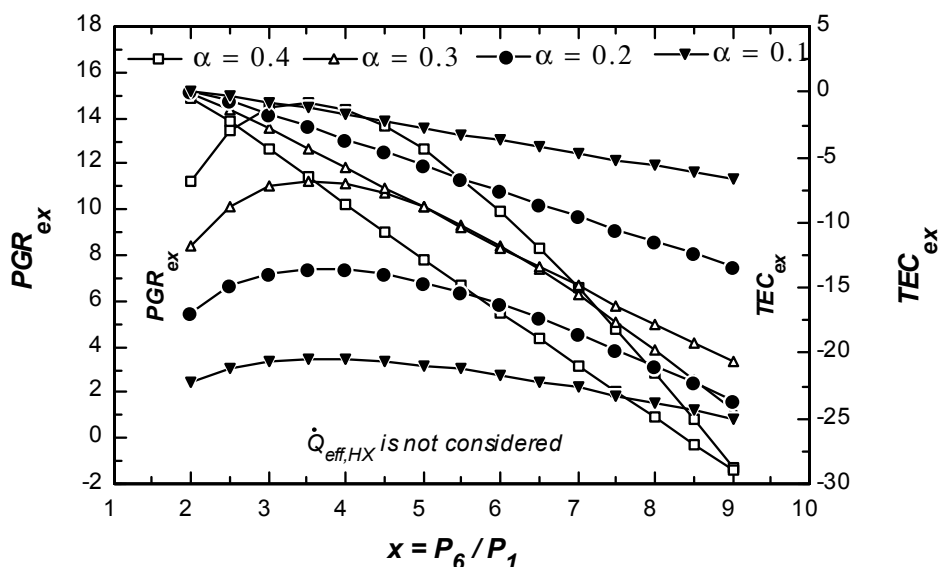
Fig. 8. Dependence of exergetic power gain ratio and exergetic thermal efficiency on the extracted pressure ratio without taking $\dot{Q}_{eff,HX}$ into account

The variations of the exergetic *PGR* and exergetic *TEC* (equations 43 and 44) are shown in Figure 8 for the same range of parameters as in Fig. 6 a. Clearly this figure shows that the irreversibilities give lower values for the power gain and efficiency as compared to the energy analysis. The difference, at the maximum *PGR*, conditions is in the order of 20 to 25% as seen in Table 3. For example, the maximum *PGR* is reduced from 9.37 in case of first law of analysis to 7.436 when the second law was considered.

α	$(PGR)_{max}\%$ Fig. 6 a	$(PGR_{ex})_{max}\%$ Fig. 8	Reduction %
0.1	4.336	3.448	20.478
0.2	9.375	7.436	20.683
0.3	14.5	11.25	22.414
0.4	19.58	14.66	25.128

Table 3. Effect of irreversibility on power gain ratio

The energy destruction terms in Table II show dependency on α at constant pressure ratio x. The difference between the gain in power as predicted by the energy and exergy analysis

is affected by both the amount of air that circulate in the refrigeration loop and the coupling pressure ratio x. Figure 9 indicates clearly that the difference increases at high extraction ratio (more air is bleed from the main compressor) and high extraction pressure (large portion of the energy is wasted in the heat exchanger). Within the previously selected range of parameters $2 < x < 4$ and $0.2 < \alpha < 0.3$ the maximum difference is in the order of 5%.
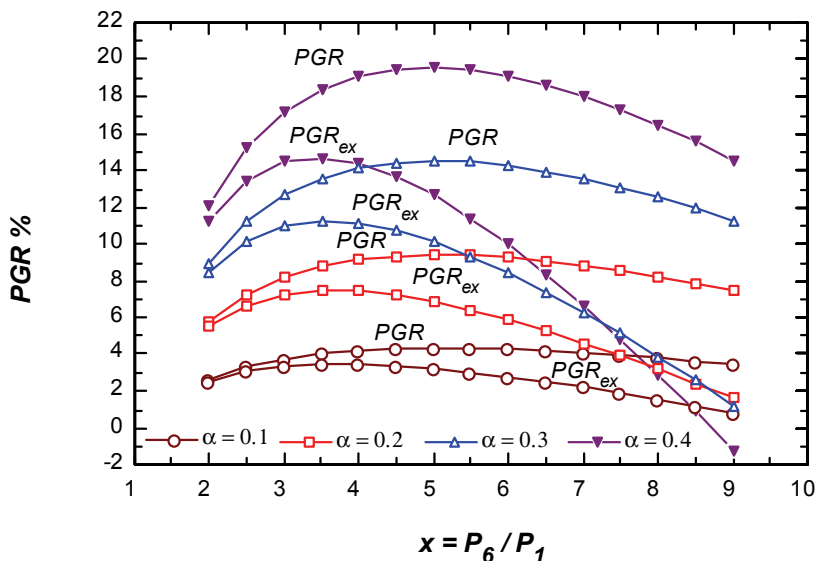

Fig. 9. Energetic and exergetic power gain ratio for a gas turbine cooled by air Brayton refrigerator
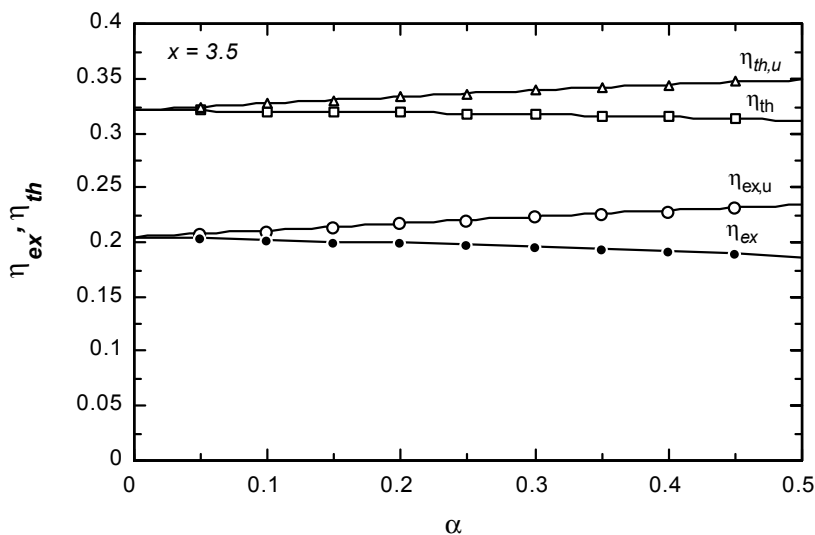
Fig. 10. Thermal efficiency, useful thermal efficiency and exergetic efficiencies dependency on α.

Figure 10 shows the magnitudes of the thermal efficiency, the useful thermal efficiency (equation 35) and the exergetic thermal efficiency (equation 40) for $0 \leq \alpha \leq 0.5$ and extraction pressure of 3.5 bars. It is clear that utilization of the cooling energy for any process enhances the use of the input fuel for which $\eta_{th,u} > \eta_{th}$ and $\eta_{ex,u} > \eta_{ex}$ as seen in the figure. In general, the thermal efficiency weights all thermal energy equally, whilst the exergetic efficiency acknowledges the usefulness of irreversibilities on its quality and quantity. Thus, exergetic efficiency is more suitable for determining the precise power gain ratio. The figure shows also that the useful efficiency whether energetic or exergetic is higher than the corresponding values when wasting the heat exchanger heat rejection $\dot{Q}_{out}$.

6. Conclusions

In this study a new approach for boosting the power of gas turbine power plants by cooling the intake air is analyzed by the energy and exergy methods. The gas turbine inlet temperature is reduced by mixing chilled air from a Brayton refrigeration cycle and the main intake air stream. The air intake temperature depends on two parameters, the cold air stream temperature from the reverse Brayton cycle and the ambient hot humid air conditions. The energy analysis of the coupled Brayton-reverse Brayton cycles showed that the intake air temperature could be reduced to the ISO standard (15ºC) and the gas turbine performance can be improved. This study demonstrated the usefulness of employing exergy analysis and the performance improvement was expressed in terms of generic dimensionless terms, *exergetic power gain ratio* (*PGR*$_{ex}$) and *exergetic thermal efficiency change* (*TEC*$_{ex}$) factor.

The performance improvement of a GT irreversible cycle of 10 pressure ratio operating in hot weather of 45ºC and 43.4% relative humidity was investigated for extraction pressures from 2 to 9 bars and cold to hot air mass rate ratio from 0.1 to 0.5. The results showed that the combustion chamber and the cooling heat exchanger are the main contributors to the exergy destruction terms; the combustion chamber irreversibility was the highest and presented 62 to 85% of the total irreversibility. The heat exchanger comes next with nearly 5 % to 46% of the combustion chamber. The irreversibility of mixing chamber was found to be small compared to other components and can be safely ignored. On basis of the energy analysis the GT power can be boosted up by 19.58 % of the site power, while the exergy analysis limits this value to only 14.66% due to exergy destruction in the components of the plant. The irreversibility can be reduced by optimal design of the combustion chamber, the heat exchanger and selecting optimum operational parameters of the coupled power and refrigeration units.

7. Nomenclature

c_{pa} = specific heat of air at constant pressure, *kJ kg*$^{-1}$ *K*$^{-1}$
c_{pg} = specific heat of gas at constant pressure, *kJ kg*$^{-1}$ *K*$^{-1}$
$\dot{E}$ = total exergy rate, *kW*
f = fuel to air ratio
$\dot{I}$ = rate of exergy destruction, *kW*
h = specific enthalpy, *kJ.kg*$^{-1}$

$\dot{m}$	= mass flow rate, *kg s⁻¹*
NCV	= net calorific value, *kJ kg⁻¹*
P	= pressure, *kPa*
PGR	= power gain ratio
PGR_{ex}	= exergetic power gain ratio
$\dot{Q}$	= heat rate, *kW*
$\dot{Q}_{out}$	= heat release rate from the condenser, *kW*
r	= pressure ratio = P_2/P_1
s	= specific entropy, *kJ kg⁻¹K⁻¹*
T	= temperature, *K*
TEC	= thermal efficiency change factor
TEC_u	= useful thermal efficiency change factor
TEC_{ex}	= exergetic thermal efficiency change
$\dot{W}$	= output power (according to subscript), *kW*
x	= extraction pressure ratio, P_6/P_1
y	= mole fraction of fuel components

Greek symbols

α	= fraction of air mass flowing through the cooler cycle
ε	= specific exergy (according to subscript)
ε_o	= exergy due to chemical reaction, *kJ mol⁻¹*
ω	= specific humidity, kg water vapor/kg dry air
ω_o	= ambiant specific humidity, kg water vapor/kg dry air
η	= efficiency (according to subscript)
γ	= specific heat ratio

Subscripts

o	= ambiant
a	= dry air
c	= compressor
cc	= combustion chamber
cy	= cycle
ex	= exergy
exp	= expander
g	= gas
HX	= heat exchanger
p	= products
pg	= product gas
mc	= mixing chamber
R	= reactants
t	= turbine
u	= useful
v	= water vapor

Superscripts

~	= mean molar value of a property

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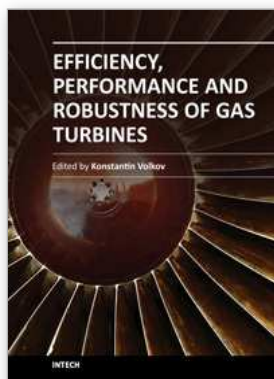
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