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Second Law Analysis of Alternative Regenerative Gas Turbine Unit

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Abstract

The second law of thermodynamics has proved to be a very powerful tool in optimization of complex thermodynamic systems. This paper involves an exergy analysis of three configurations of gas turbine units, simple unit with no regenerator, conventional regenerative gas turbine unit and alternative regenerative gas turbine unit. Obviously, in conventional regenerative gas turbine cycle, the position of regenerator is placed immediately after the low pressure turbine. In the current study the position of regenerator is suggested to be immediately after the high pressure turbine. This may result in preheating the compressed air to a high temperature before admitting to the combustion chamber and also lowers the low pressure turbine exit temperature.

Results show that the second law efficiency of alternative regenerative gas turbine has been increased throughout the pressure ratio range (6-20) for fixed turbine inlet temperature (1400K) as well as throughout the turbine inlet temperature range (1100 -1750 K) for optimum pressure ratio nearly by (52% , 12%) and (23% , 11.6%) as compared to simple and conventional regenerative cycle respectively. The exergy loss due to irreversibility in combustion chamber, regenerative and exhaust gases was found to be decreased nearly by 60% and 39.4 respectively as compared to simple cycle as well as by32.5%, 22.5% and 13.9% respectively as compared to conventional regenerative cycle.

Keywords:- ***Gas turbine, Exergy, Regeneration, Irreversibility***

استخدام القانون الثاني للثرموداينمك لتحليل الطاقة المتوفرة في وحدة غازية ذات مبادل حراري ذو موقع بديل

الخلاصة

يتميز القانون الثاني للثرموديناميك بإمكانية استخدامه في إجراء المفاضلات بين أنظمة حرارية معقدة. يتضمن هذا البحث تحليل لثلاث أشكال من الوحدات الغازية وهي (البسيطة التي لا تحتوي على مبادل حراري وا لتقليدية التي تحتوي على مبادل حراري يوضع بعد التوربين ذو الضغط الواطيء ، وأخيرا الوحدة الغازية المقترحة في البحث الحالي والتي تتضمن تجهيز مبادل حراري ذو موقع بديل عن ذلك المتعارف عليه والذي يوضع عادة في مجرى الغازات العادمة الخارجة من التوربين الأخير (التوربين ذو الضغط الواطيء) والذي يستخدم لرفع درجة حرارة الهواء الخارج من الضاغطة قبل دخوله إلى غرفة الاحتراق . في هذا البحث تم اقتراح موقع المبادل الحراري بعد التوربين ذو الضغط العالي مباشرة ، وقبل التوربين ذو الضغط الواطيء، على اعتقاد إن الدورة الجديدة للوحدة الغازية تكون أكثر كفاءة واقل هدرا للطاقة المتوفرة.

بينت النتائج و اعتمادا على القانون الثاني ان كفاءة الوحدة الغازيةالمقترحة ازداد و لجميع قيم نسبة الانضغاط بثبوت درجة حرارة نواتج الاحتراق الداخلة الى التوربين (1400 K)، و كذلك لجميع درجات حرارة نواتج الاحتراق الداخلة الى التوربين (1100-1750 K) لافضل نسبة انضغاط تقريبا بحوالي (12%, 52%) و (11.6%, 23%) بالمقارنة مع الدورة البسيطة و التتقليدية على التوالي. اما بالنسبة للطاقة المهدورة نتيجةاللاانعكاسية في غرفة الاحتراق الداخلي ، المبادل الحراري و غازات العادم ، فقد وجد انها انخفضت تقريبا بحوالي (39.4%, 60%) على التوالي مقارنة بالدورة البسيطة، و تقريبا بحوالي (13.9%, 22.5%, 32.5%) على التوالي مقارنة بالدورة التقليدية.

Introduction

In gas turbine units, the temperature of the exhaust gases leaving the turbine is often considerably higher than the temperature of compressed air leaves compressor [1,2,3,4,5,6]. Therefore, the high pressure air leaving the compressor can be heated up by transferring heat to it from the hot gases leaving the turbine by a heat exchanger called regenerator before expelling to atmosphere. The thermal efficiency of gas turbine cycle increases as a result of regeneration since the portion of energy of exhaust gases that is normally rejected to the surrounding is now used to preheat the compressed air entering the combustion chamber this, in turn, decrease the mass of fuel required for the same turbine inlet temperature TIT [1,7,8].

The traditional way for placing the regenerator in gas turbine unit is too placed immediately after the low pressure turbine, which is called in current study "conventional regenerative gas turbine". Cohen, et al [8] has suggested using the use of regenerator in a traditional manner using the exhaust gases leaving the low pressure turbine as a heat source. This recovered heat from exhaust gases is used to preheat the compressed air before entering the combustion chamber. Several other methods including combine cycle and cogeneration are suggested by Bathe [9] and Khartchenko [10] to improve the gas turbine performance. Dellenback [4] has pointed out that the regenerator location after the low pressure turbine is inefficient thus suggesting the location of it after the high pressure turbine will improve the thermal efficiency. The author claimed that the effect is due to increase in amount of heat supplied to the compressed air beyond what is possible in a conventional regeneration. However, no exergy analysis of gas turbine components was made which is necessary to identify the irreversibility of the system[4].

In thermodynamics, the concern is not only for the quantity of energy but also for quality of energy[11,12]. The first law efficiency does not take into account the quality of energy. The current work is aimed to analyze the gas turbine efficiency with alternative regenerator not only from first law point of view but also from second law. The second law analysis is useful to identify the components having maximum irreversibility thus enables proper selection of the process for maintaining high quality of energy[13]. In the present study the position of regenerator is suggested to be kept immediately after the high pressure turbine which is called "alternative regenerative gas turbine". This thought to results in preheating the compressed air to considerably high temperature before admitting to the combustion chamber which may results in improving system efficiency due to the increase in average temperature of heat supplied and eventually reducing the low turbine exit temperature[2,4].

Gas Turbine Cycle Description

An alternative regeneration gas turbine unit with high and low pressure turbines is shown in figure (1). The high pressure turbine is used to drive the compressor, while the low

pressure turbine is develop the net work output. In this regeneration gas turbine cycle, air enters the axial flow compressor, compressed to high pressure according to the compressor pressure ratio. After compression, compressed air enters the regenerative wheel, heated up by the hot gases leaving the high pressure turbine before entering the combustion chamber. Clearly this process results in reducing the amount of fuel required to achieve the allowable temperature of high pressure turbine.

The hot gases enters the high pressure turbine, expands to a limit that the output of the turbine is adequate to drive the compressor. Therefore, hot gases leaving the high pressure turbine pass through the regenerator wheel and are directed to go further expansion in the low pressure turbine to produce the net power output, which functions of temperature drop in this turbine.

Mathematical Model Formulation

Figure (2), shows the T-S diagram for the suggested alternative regeneration gas turbine cycle. The actual and ideal processes are represented in dashed and full line respectively. In the axial flow compressor, air is compressed from point 1 to point 2. The compressor work can be obtained as follows[1,2,4]:

$$W_c = m_a (h_2 - h_1) \tag{1}$$

Or in term of temperature

$$W_c = m_a Cp_a (T_2 - T_1) \tag{2}$$

However, the power required to drive the compressor is equal to the power developed by the high pressure turbine, i.e.

$$W_c = W_{hpt}$$

Or can be written as follows:

$$m_a Cp_a (T_2 - T_1) = m_g Cp_g(T_3 - T_4) \tag{3}$$

Compressed air leaving the compressor, gained heat in the regeneration then enters the combustion chamber. The heat supplied in the combustion chamber is a result of burning of fuel and is equal to the heat absorbed by air. Hence;

$$m_g Cp_g T_3 - m_a Cp_a T_7 = m_f * LCV \tag{4}$$

In the regenerator, the heat gained by relatively cold air leaving the compressor is approximately equal to the heat lost by hot gases taking in consideration the effectiveness of the regenerator. Hence;

$$m_a Cp_a (T_7 - T_2) = m_g Cp_g(T_4 - T_5) (1-q_L) \tag{5}$$

The net power produced by the gas turbine unit is surely equal only to the power produced by the low pressure turbine. Hence;

$$W_{net} = m_g\, Cp_g(T_5 - T_6) \qquad (6)$$

Therefore, the thermal efficiency of the cycle based on the first law of thermodynamic can be obtained as follows:

$$\eta_I = \frac{m_g\, Cp_g\, (T_5 - T_6)}{m_f * LCV} \qquad (7)$$

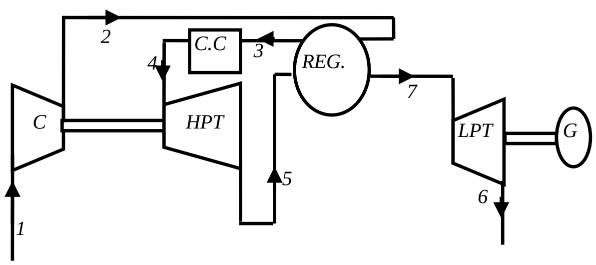


Figure (1): Schematic diagram of alternative regeneration gas turbine unit

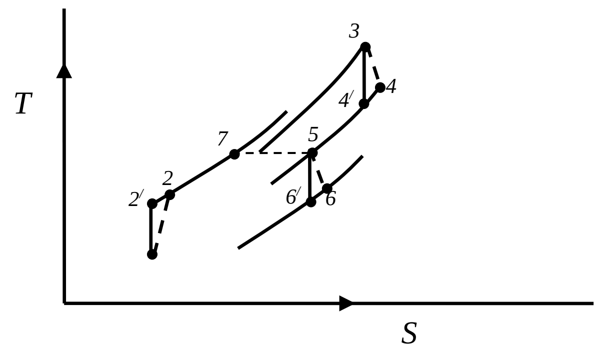


Figure (2) : T-S diagram for alternative regeneration gas turbine unit

Cycle Analysis Based on The 2nd law

Applying the second law of thermodynamic for each individual component of the gas turbine and as follows:

Compressor

The compressor irreversibility (I_c) can be obtained as follows[4]:

$$I_c = m_a\, T_o\, (S_2 - S_1) \qquad (8)$$

Where $$S_2 - S_1 = Cp_a \ln(\frac{T_2}{T_1}) - R_a \ln(\frac{P_2}{P_1}) \qquad (9)$$

The second law efficiency of the compressor can be written as follows;

$$\eta_{II_C} = \frac{m_a Cp_a\, (T_2 - T_1) - m_a\, T_o\, (S_2 - S_1)}{m_a Cp_a\, (T_2 - T_1)} \qquad (10)$$

High pressure turbine

The high pressure turbine irreversibility can be obtained as follows

$$I_{hpt} = m_g\, T_o\, (S_4 - S_3) \tag{11}$$

Where $$S_4 - S_3 = Cp_g \ln\left(\frac{T_4}{T_3}\right) - R_a \ln\left(\frac{P_4}{P_3}\right) \tag{12}$$

The second law efficiency is given as:

$$\eta_{\Pi_{hpt}} = \frac{W_{hpt}}{m_g Cp_g (T_3 - T_4) - m_g T_o (S_4 - S_3)} \tag{13}$$

Low pressure turbine

In the low pressure turbine, its irreversibility (I_{Lpt}) can be calculated as follows:

$$I_{Lpt} = m_g\, T_o\, (S_6 - S_5) \tag{14}$$

Where $$S_6 - S_5 = Cp_g \ln\left(\frac{T_6}{T_5}\right) - R_a \ln\left(\frac{P_6}{P_5}\right) \tag{15}$$

The second law efficiency can be written as follows:

$$\eta_{\Pi_{hpt}} = \frac{W_{Lpt}}{m_g Cp_g (T_5 - T_6) - m_g T_o (S_5 - S_6)} \tag{16}$$

Regenerator

The amount heat of loss in the regenerator can be assessed as follows:

$$Q_{Lreg} = m_g\, Cp_g\, (T_4 - T_5)\, q_L \tag{17}$$

Irreversibility of the regenerator I_{reg} can be obtained using the following equation:

$$I_{reg} = m_g\, Cp_g\, (T_4 - T_5)\, q_L + T_o\, [\, m_a\, (S_7 - S_2) - m_g\, (S_4 - S_5)] \tag{18}$$

Where

$$S_7 - S_2 = Cp_a \ln\left(\frac{T_7}{T_2}\right) - R_a \ln\left(\frac{P_7}{P_2}\right) \tag{19}$$

$$S_4 - S_5 = Cp_g \ln\left(\frac{T_4}{T_5}\right) - R_a \ln\left(\frac{P_4}{P_5}\right) \tag{20}$$

The second law efficiency can be worked out as follows:

$$\eta_{\Pi_{hpt}} = \frac{m_a Cp_a (T_7 - T_2) - m_a T_o (S_7 - S_2)}{m_g Cp_g (T_4 - T_5) - m_g T_o (S_4 - S_5)} \qquad \textbf{(21)}$$

Combustion chamber

The combustion chamber irreversibility (I_{cc}) can be obtained as follows:

$$I_{cc} = T_o [(S_P)_3 - (S_R)_7] \qquad \textbf{(22)}$$

Where

$$(S_R)_7 = (S_a)_7 + (S_f)_o \qquad \textbf{(23)}$$

Hence $$I_{cc} = T_o [\{(S_P)_3 - (S_P)_o \} - \{(S_a)_7 - (S_a)_o \} + \Delta S_o] \qquad \textbf{(24)}$$

Or it can be written as follows:

$$I_{cc} = T_o [m_g Cp_g \ln(\frac{T_3}{T_o}) - m_g R_g \ln(\frac{P_3}{P_o})] - T_o [m_g Cp_g \ln(\frac{T_7}{T_o}) - m_a R_a \ln(\frac{P_7}{P_o})] + T_o \Delta S_o \qquad \textbf{(25)}$$

Where

$T_o (\Delta S_o)$:is the rate of exergy loss in the combustion

and $$T_o (\Delta S_o) = \Delta G_o - \Delta H_o \qquad \textbf{(26)}$$

also $$\Delta H_o = m_f * LCV \qquad \textbf{(27)}$$

However the ratio of $\frac{\Delta G_o}{\Delta H_o}$ can be obtained using the following formula[2,4] :

$$\frac{\Delta G_o}{\Delta H_o} = 1.041 + 0.1728 * \alpha \qquad \textbf{(28)}$$

Where α is the ratio of mass of hydrogen to the mass of carbon.
Finally the rate of exergy loss due to exhaust gases can be obtained as follows:

$$I_{exh} = \int_{T_{6'}}^{T_4} (1 - \frac{T_o}{T}) dq = m_g Cp_g [(T_{6'} - T_o) - T_o \ln(\frac{T_{6'}}{T_o})] \qquad \textbf{(29)}$$

The overall second law efficiency can be formed finally as follows:

$$\eta_{\Pi_{overall}} = \frac{m_g Cp_g (T_5 - T_6)}{m_g Cp_g (T_5 - T_6) + I_c + I_{hpt} + I_{Lpt} + I_{cc} + I_{reg} + I_{exh}} \qquad \textbf{(30)}$$

Gas Turbine System Comparison

There are three types of gas turbine units are adopted in order to make a comparison: namely, simple, conventional and alternative regenerative configuration. Clearly, the simple gas turbine is that with no regenerator as shown in figure (3) and its T-S diagram is given in figure (4). The conventional gas turbine is that where the regenerator is placed after the low pressure turbine as shown in figure (5) and its T-S diagram given in figure (6). The alternative gas turbine configuration is that where the regenerator is placed after the high pressure turbine as shown in figure (1) and its T-S diagram is given if figure (2).

The inlet temperature and pressure are assumed to be 25 ºC and 1.01 bar respectively, the maximum high turbine inlet temperature is set to be 1400 K, pressure drop due combustion chamber and regenerator is assumed to be 0.15 bar. The isentropic efficiencies of turbines and compressor are 0.89 and 0.86 respectively while the effectiveness of regenerator is taken as a maximum value of 0.85. The property of air is exhaust gas are assumed to be temperature dependant and are assessed throughout the cycle.

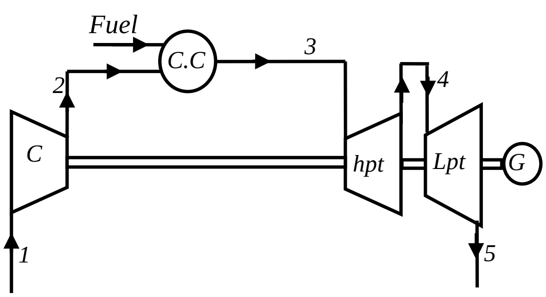


Figure (3) : Simple gas turbine unit

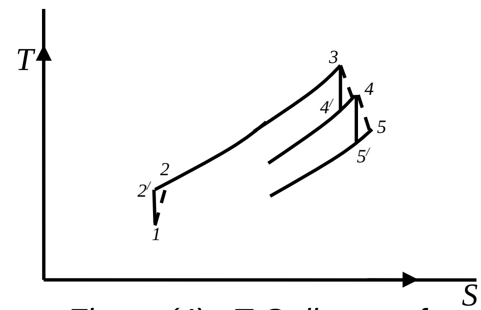


Figure (4) : T-S diagram for simple gas turbine unit

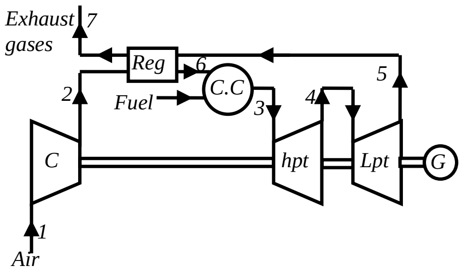


Figure (5): Gas turbine unit with conventional regenerator

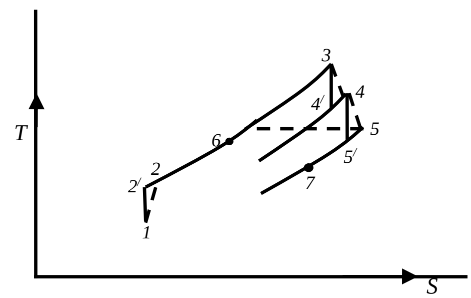


Figure (6) : T-S diagram for conventional regeneration gas turbine unit

Results and Discussions

For the assumed values of pressure ratio and turbine inlet temperature, the pressures and temperatures at different points in the cycles are evaluated. Thereafter, the overall second law efficiency of the three gas turbine units including simple, conventional an alternative regenerative gas turbine unit along with the irreversibilities of individual components is evaluated.

The results obtained are pictured in graphical forms as given in figure 7 to 14. figure 7, shows the relationship between the pressure ratio and second law efficiency at turbine inlet temperature of 1400 K for the three gas turbine units. It is quite clear that the proposed alternative regenerator cycle gives better efficiency when compared to simple and conventional regenerator system. Furthermore, for maximum efficiency the optimum pressure ratio for the simple system was found to be much higher than the conventional and alternative regenerator systems. It reveals the fact that adopting regeneration technique higher efficiency can be obtained at a lower optimum ratio. However, for any given pressure ratio the alternative regenerator system has higher second law efficiency than that for conventional and simple one.

Figure (8) shows the relationship between the second law efficiencies and turbine inlet temperature at optimum pressure ratio. It can be seen that the second law efficiency increases with increasing turbine inlet temperature. However, the efficiency of the alternative regeneration system is the highest at any given permissible temperature.

Figure (9) indicates the relationship between the pressure ratio and irreversibility losses in the combustion chamber. It is quite obvious that the irreversibility in the alternative regeneration system is far less than that of conventional and simple one. Figure (10) shows the relationship between the pressure ratio and exhaust gases irreversibilities of the systems. It can be seen that the irreversibility of exhaust gases of alternative regeneration system is quite low than those of conventional and simple system. This is because the exhaust gases temperature is least in alternative system as shown in figure (11). Figure(12) indicates the variation of regenerator irreversibilities with pressure ratio for conventional and alternative regeneration system only since no regeneration is applied in simple system. It can be seen that the irreversibility of alternative system is far less than that of conventional system and in particular for low pressure low. However, when both systems operate at optimum nearly pressure ratios the irreversibilities are approach each other and values are considerably reduced.

Figures (13) and (14) show the irreversibilities losses in low and high pressure turbines respectively. It can be observed that the irreversibility for both turbines is increase for the alternative system in comparison to conventional and simple systems. However, their values are considerably low in comparison to the value of irreversibilities of combustion chamber, exhaust gases and regenerator, so it has no significant effect on second law efficiency.

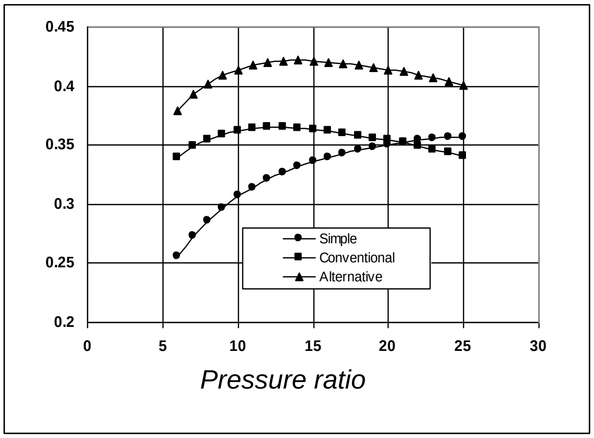
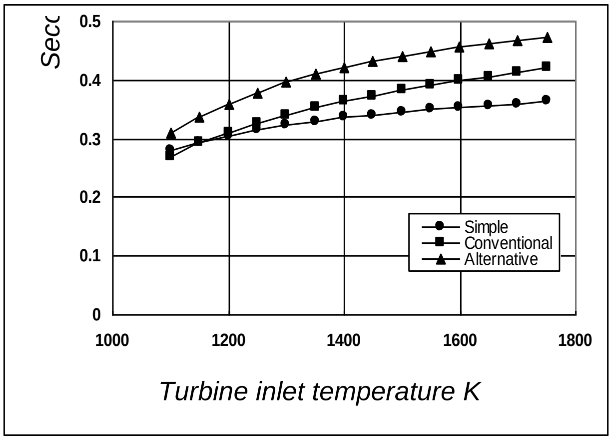
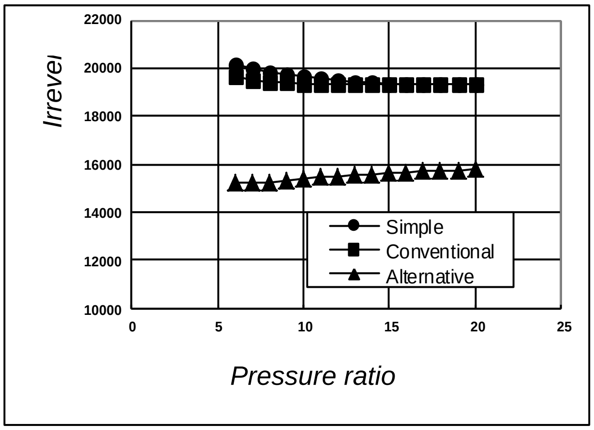


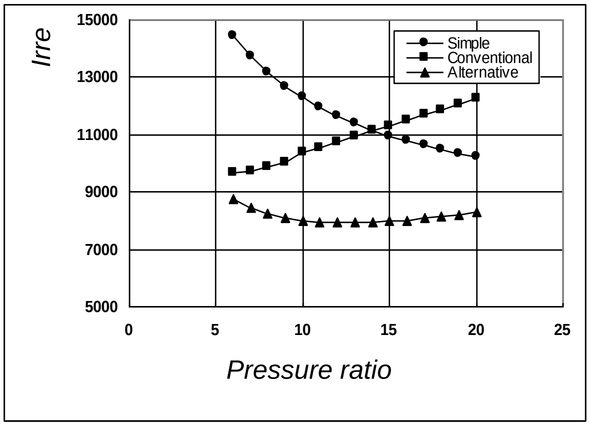
Figure (7): the relationship between the pressure ratio and second law efficiency



Figure(8): the relationship between the turbine inlet temperature and second law efficiency



Figure(9): the relationship between the pressure ratio and the combustion chamber irreversibility



Figure(10): the relationship between the pressure ratio and the exhaust gases irreversibility

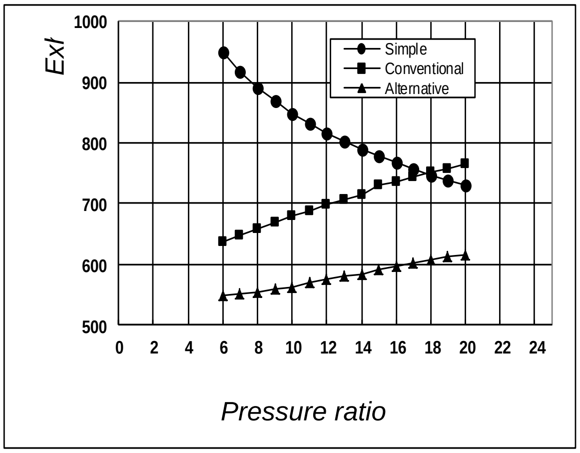
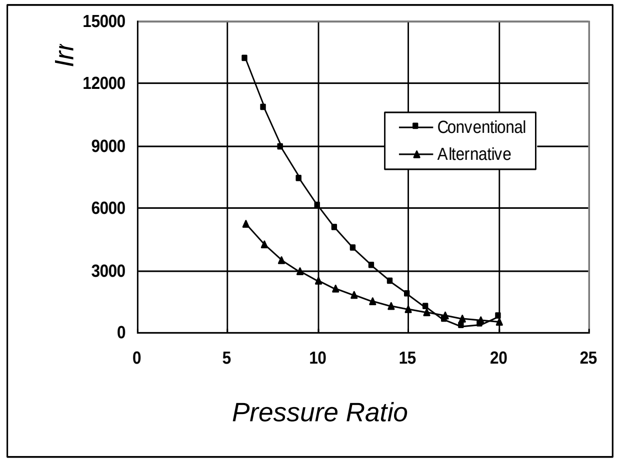
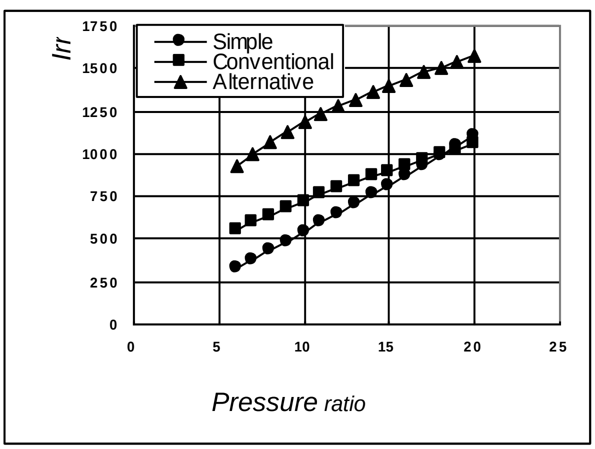


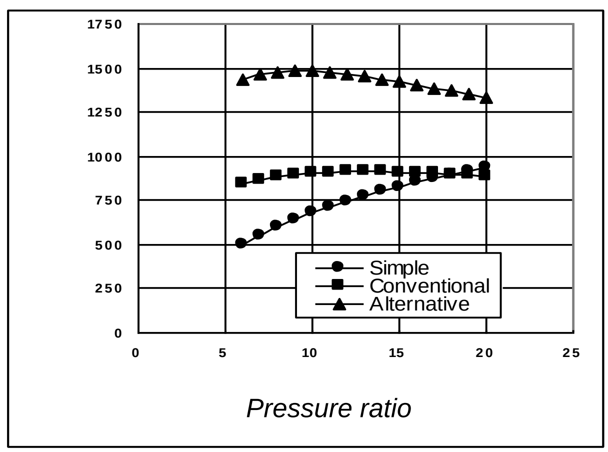
Figure (11): the relationship between the pressure ratio and exhaust temperature for low pressure turbine



Figure(12): the relationship between the pressure ratio and the regenerator irreversibility



Figure(13): the relationship between the pressure ratio and the high pressure turbine irreversibility



Figure(14): the relationship between the pressure ratio and the low pressure turbine irreversibility

References

1- *Yunus A. Cengel, Michael A. Boles, Thermodynamics an Engineering Approach, McGraw-Hill Com. 1989.*

2- *Ashock Kumar,S S Kachhwaha and R S Mishra, Thermodynamic Analysis of a Regenerative Gas turbine Cogeneration Plant, Journal of Scientific and Industrial Research, Vol.69, pp 225-231 , March 2010.*

3- *Alaandro Franco, Claudio Casarosa, On Some Perspective for Increasing the Efficiency of Combined Cycle Power Plants, Applied Thermal Engineering,Vol.22, pp1501-1518, 2002.*

4- *P.A. Dellenback, Improved Gas Turbine Efficiency Through Alternative Regenerator Configuration', Transaction of ASME. Journal of Engineering for Gas Turbines and Power, vol.124, July 2002, p 441-446.*

5- *Taniquchi H., Miyamae, S., et al., Power Generation Analysis for High Temperature Gas Turbine in Thermodynamic Process, Journal of Propulsion and Power. Vol.16, pp.557-561, 2000.*

6- *De Lucia M., Bronconi, R., and Carneval, E., Performance and Economic Enhancement of Cogeneration Gas Turbine Through Compressor Inlet Air Cooling, ASME Journal of Engineering for Gas Turbine and Power, Vol. 116, pp.360-365, 1994.*

7- *Jack D. Muttingly, Element of Gas Turbine Propulsion, McGraw-Hill, Inc, 1996.*

8- *H. Cohen. G F Rogers and H. H. Saravanamuttoo., Gas Turbine Theory , Fourth edition , Longman group, Harlew, England, 1996.*

9- *W.W. Bathe, Fundamental of Gas Turbines, Second Edition, John Wiley and Sons, New York, 1996.*

10- *N.V. Khartchenko, Advanced Energy Systems, Taylor and Francis, Washington DC, 1998.*

11- *A. Khaliq, K.Choudhary, Combined First and Second Law Analysis of Gas Turbine Cogeneration System with Inlet Cooling and Evaporative After cooling of the Compressor Discharge , Journal of Engineering for Gas Turbine and Power, Vol. 19 , pp1004-1011, Oct. 2007.*

12- Roberto Carapellucci, A Unified Approach to Assess Performance of Different Techniques for Recovering Exhaust Heat From Gas Turbines, Energy Conversion and Management Vol.50, pp1218-1226, 2009.

13- X. Q. Kang, R.Z . Wang, X.H.Huang, Energy Optimization Mode for A CCHP System with Available Gas Turbines, Applied Thermal Engineering,Vol.5, pp377-391, 2005.

Nomenclature:

Cp	Specific heat	(kJ/kg.K)
G	Gibbs function	(kJ/kg)
H	Enthalpy	(kJ)
h	Specific enthalpy	(kJ/kg)
I	Irreversibility	(kJ/kg)
LCV	Lower calorific value	(kJ/kg)
m	Mass	(kg)
P	Pressure	(bar)
Q_{Lreg}	Heat loss in the regenerator	kJ
ql	Specific heat loss in the regenerator	(kJ/kg)
R	Specific gas constant	(kJ/kg.K)
r_p	Pressure ratio	
S	Entropy	(kJ/kg.K)
T	Temperature	(K)
TIT	Turbine inlet temperature	(K)
W	Work	(kJ)
ε	Effectiveness	
η	Efficiency	
γ	Specific heat ratio	
Δ	Difference	

Subscripts:

a	Air
c	compressor
CC	Combustion chamber
exh	Exhaust gases
f	Fuel
g	gas
hpt	High pressure turbine
Lpt	Low pressure turbine
o	atmospheric
p	products
reg	regenerator
1,2,3..	Thermodynamic states