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Analysis of a Regenerative Gas Turbine Cycle for Performance Evaluation

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ABSTRACT
The effect of a regenerative heat exchanger in a gas turbine is analyzed using a regenerative Brayton cycle model, where all fluid friction losses in the compressor is quantified by an isentropic efficiency term and all global irreversibilities in the heat exchanger are taken into account by means of an effective efficiency. This analysis, which generalizes that reported by Gordon and Huleihil for a simple, non-regenerative Brayton cycle, provides a theoretical tool for the selection of optimal operating conditions in a regenerative gas turbine for optimum value of compressor efficiency. Regenerative gas turbine engine cycle is presented that yields higher cycle efficiencies than simple cycle operating under the same conditions. The power output, efficiency and specific fuel consumption are simulated with respect to operating conditions. The analytical formulae about the relation to determine the thermal efficiency are derived taking into account the effected operating conditions (ambient temperature, compression ratio, regenerator effectiveness, compressor efficiency and turbine inlet temperature). Model calculations for a wide range of parameters are presented, as are comparisons with simple gas turbine cycle. The power output and thermal efficiency are found to be increasing with the regenerative effectiveness, and the compressor efficiency. The efficiency increased with increase the compression ratio to 15, then efficiency decreased with increased compression ratio, but in simple cycle the thermal efficiency always increases with increased in compression ratio. The increased in ambient temperature caused decreased thermal efficiency, but the increased in turbine inlet temperature increase thermal efficiency.
Keywords: Gas turbine cycle, Regeneration, compressor efficiency, TIT, OPR, thermal efficiency, regenerative effectiveness **.**

1. Introduction
Today, gas turbines are one of the most widely-used power generating technologies. Gas turbines are a type of internal combustion (IC) engine in which burning of an air-fuel mixture produces hot gases that spin a turbine to produce power. It is the production of hot gas during fuel combustion, not the fuel itself that gives the gas turbines the name. Combustion occurs continuously in gas turbines, as opposed to reciprocating IC engines, in which combustion occurs intermittently. So for understanding the history of the gas turbine, one would have to read several different papers and select material written by personnel from the aviation, and land-based sectors. At that point, one can “fill in the gaps”. What follows therefore are two different accounts of the gas turbine’s development. Neither of them is wrong. The first of these presents an aircraft engine development perspective [1].

In the original 19th-century Brayton engine, ambient air is drawn into a piston compressor, where it is compressed; ideally an isentropic process. The compressed air then runs through a mixing chamber where fuel is added, an isobaric process. The heated (by compression), pressurized air and fuel mixture is then ignited in an expansion cylinder and energy is released, causing the heated air and combustion products to expand through a piston/cylinder; another ideally isentropic process. Some of the work extracted by the piston/cylinder is used to drive the compressor through a crankshaft arrangement.

2. Materials and Methods:
2.1 Analysis of the Ideal Cycle
The Air Standard cycle analysis is used here to review analytical techniques and to provide quantitative insights into the performance of an ideal-cycle engine. Air Standard cycle analysis treats the working fluid as a calorically perfect gas, that is, a perfect gas with constant specific heats evaluated at room temperature. In Air Standard cycle analysis the heat capacities used are those for air. In the present work the heat capacities for air and gas are taken as per their chemical constituents. Regeneration involves the installation of a heat exchanger (recuperator) through which the turbine exhaust gases (point 4 in Fig.1) pass. The compressed air (point 2 in Fig.1) is then heated in the exhaust gas heat exchanger, before the flow enters the combustor (Fig.1).

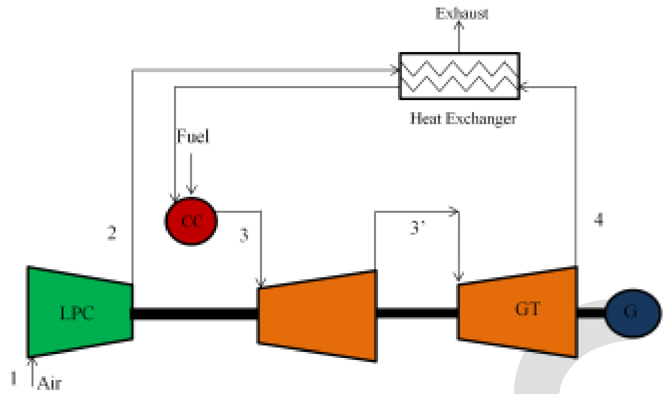

Fig.1 Schematic diagram of Regenerative gas turbine cycle

A gas turbine cycle is usually defined in terms of the compressor inlet pressure and temperature, p_i and T_i, the compressor pressure ratio, $r_p = p_e / p_i$, and the turbine inlet temperature, TIT, where the subscripts correspond to the inlet and exit.

2.2 Thermodynamic analysis of various components

Gas Model

The thermodynamic properties of air and products of combustion are calculated by considering variation of specific heat and with no dissociation. Table containing the values of the specific heats against temperature variation have been published in many references such as Chappel and Cockshutt [10]. The curve fitting the data is used to calculate specific heats, specific heat ratio, and enthalpy of air and fuel separately from the given values of temperature. Mixture property is then obtained from properties of the individual component and fuel air ratio (FAR). Following equations are used to calculate the specific heats of air and gas as a function of temperatures.

(1) If $T_a \le 800$

$$c_{pa} = (1.0189 \times 10^3) - (0.13784.T_a) + (1.9843 \times 10^{-4}.T_a^2) + (4.2399 \times 10^{-7}.T_a^3) - (3.7632 \times 10^{-10}.T_a^4) \quad (1)$$

(2) If $T_a > 800$

$$c_{pg} = 0.0086.c_{p,Ar} + 0.7154.c_{p,N_2} + 0.0648.c_{p,O_2} + 0.1346.c_{p,H_2O} + 0.0666.c_{p,CO_2}$$

In the above equations, T stands for gas or air temperature in deg K and $t = \frac{T}{100}$

Combustion Chamber

One of the goals of combustion chamber design is to minimize the pressure loss from the compressor to the turbine. Ideally, then, p3 = p2, as assumed by the Air Standard analysis. More realistically, a fixed value of the combustor fractional pressure loss, Δp_{cc}, (perhaps about 0.05 or 5%) may be used to account for burner losses:

$$\Delta p_{cc} = p_{b,i} - p_{b,e} \quad (2)$$

The rate of heat released by the combustion process may then be expressed as:

$$Q_a = \dot{m}_a (1 + FAR)\, c_{pg} (T_3 - T_2) \quad [kW] \quad (3)$$

where FAR is the mass fuel-air ratio

Energy balance:

$$\eta_b.\dot{m}_{f,cc}.LCV_f = \dot{m}_{g,e}.h_{g,e} - \dot{m}_{a,i}.h_{a,i} \quad (4)$$

The fuel to air ratio (FAR) is calculated as,

$$FAR = \frac{c_{pg}.T_e - c_{pa}.T_i}{\eta_b.LCV_f - c_{pg}.T_e} \quad (5)$$

In this equation T_e is the turbine inlet temperature, T_i is stagnation or total exit temperature of HP compressor, η_b is the combustion efficiency of the main combustion chamber, normally taken between 0.98 to 0.99 and LCV_f is the lower calorific value of the fuel taken as 42000 kJ/kgK assuming fuel as diesel. Values of specific heat of air and gases are to be calculated from the gas model.

Regenerator

A regenerator is modeled as a gas-to-gas counter flow heat exchanger (Fig 2). In this, heat is exchanged between compressed air coming out of the compressor and the hot gas exiting the gas turbine after expanding before entering the HRSG.

The advantage of using a recuperator is that, some heat is added to the compressed air in the recuperator itself before it enters into the combustion chamber, so the same turbine inlet temperature is achieved as that when no such recuperator is employed, with lower fuel consumption hence the efficiency of the plant increases. The following are the assumptions for modeling of a recuperator.

- A concept of effectiveness of the recuperator is introduced to account for its inefficiencies.
- There is a pressure drop in the streams passing through the recuperator, which is taken as percentage of inlet pressure.

The enthalpy of air entering the combustion chamber is given by energy balance equation

$$\epsilon_{rc}=\frac{(T_{rc,g})_i-(T_{rc,g})_e}{(T_{rc,g})_i-(T_{rc,a})_i} \tag{6}$$

and

$$\dot{m}_{rc,g}.\ c_{p,g}.\ \varepsilon_{rc}.\ \left\{(T_{rc,g})_{in}-(T_{rc,g})_{out}\right\}=\dot{m}_{rc,a}.\ c_{p,a}.\ \left\{(T_{rc,a})_{out}-(T_{rc,a})_{in}\right\} \tag{7}$$

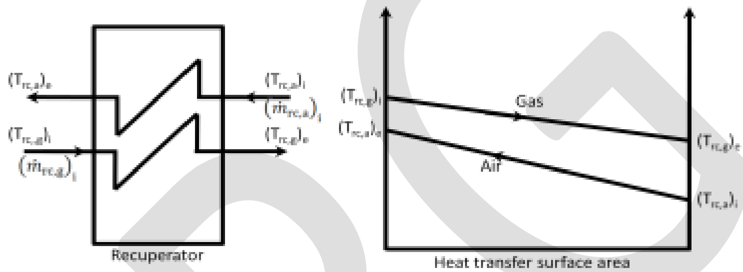

Fig.2 Schematic and temperature- heat transfer surface area for a counter-flow surface type recuperator

Figure 2 shows a gas turbine with a counter flow heat exchanger that extracts heat from the turbine exhaust gas to preheat the compressor discharge air to T_5 ahead of the combustor. As a result, the temperature rise in the combustor is reduced toT_4. T_5 a reduction reflected in a direct decrease in fuel consumed.

Note that the compressor and turbine inlet and exit states can be the same as for a simple cycle. In this case the compressor, turbine, and net work as well as the work ratio are unchanged by incorporating a heat exchanger.

The effectiveness of the heat exchanger, or regenerator, is a measure of how well it uses the available temperature potential to raise the temperature of the compressor discharge air. Specifically, it is the actual rate of heat transferred to the air divided by the maximum possible heat transfer rate that would exist if the heat exchanger had infinite heat transfer surface area.

The actual heat transfer rate to the air is $c_p(T_5 - T'_2)$, and the maximum possible rate is $c_p(T'_4 - T'_2)$ Thus the regenerator effectiveness can be written as

$$\varepsilon_{rc}=\frac{T_5-T'_2}{T'_4-T'_2} \tag{8}$$

It is seen that the combustor inlet temperature varies from T2 to T4 as the regenerator effectiveness varies from 0 to 1. The regenerator effectiveness increases as its heat transfer area increases. Increased heat transfer area allows the cold fluid to absorb more heat from the hot fluid and therefore leave the exchanger with a higher T_5.

On the other hand, increased heat transfer area implies increased pressure losses on both air and gas sides of the heat exchanger, which in turn reduces the turbine pressure ratio and therefore the turbine work. Thus, increased regenerator effectiveness implies a tradeoff, not only with pressure losses but with increased heat exchanger size and complexity and, therefore, increased cost.

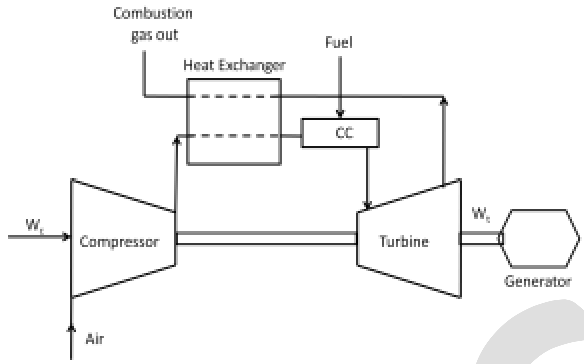

Fig. 3 Schematic of a Regenerative gas turbine

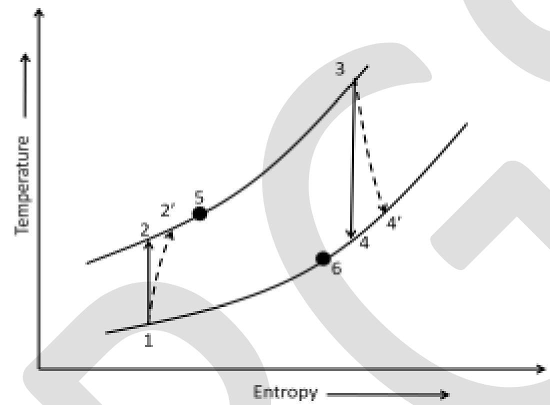

Fig. 4 T-s representation of Regenerative gas turbine cycle

The exhaust gas temperature at the exit of the heat exchanger may be determined by applying the steady-flow energy equation to the regenerator. Assuming that the heat exchanger is adiabatic and that the mass flow of fuel is negligible compared with the air flow and noting that no shaft work is involved.

2.3 Gas Turbine Analysis with Regeneration

Fig. 4 shows the T-S diagram for regenerative gas turbine cycle. The actual processes and ideal processes are represented in dashed line and full line respectively. The compressor efficiency (η_c), the turbine efficiency (η_t) and effectiveness of regenerator (heat exchanger) are considered in this study. These parameters in terms of temperature are defined as in (1) [8]:

$$\eta_c = \frac{T_2^{'} - T_1}{T_2 - T_1} \qquad (9)$$

$$\eta_t = \frac{T_4 - T_5}{T_4 - T_5^{'}} \qquad (10)$$

Regenerative effectiveness is given by

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \qquad (11)$$

The work required to run the compressor is expressed as in (2):

$$W_c = c_{pa} T_1 \left[\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \right] \qquad (12)$$

The work developed by turbine is then rewritten as in (5):

$$W_t = c_{pg} T_4 \eta_t \left[1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \right] \qquad (13)$$

where turbine inlet temperature (TIT) = T4 .The net work is expressed as in (6)

$$W_{net} = W_t - W_c \qquad (14)$$

i.e.

$$W_{net} = c_{pg} T_4 \eta_t \left[1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}}\right] - c_{pa} T_1 \left[\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right] \quad (15)$$

In the combustion chamber, the heat supplied by the fuel is equal to the heat absorbed by air, Hence,

$$Q_{ad} = c_{pg} \left[T_4 - T_1(1-\varepsilon) \times \left(1 + \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right) - \varepsilon \times T_4 \times \left[1 - \eta_t \left(1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}}\right)\right]\right] \quad (16)$$

Power output is given by:

$$P = \dot{m}_a \times W_{net} \quad (17)$$

Fuel to air ratio is given by

$$FAR = \frac{C.V.}{Q_{ad}} \quad (18)$$

And Specific Fuel consumption

$$SFC = \frac{C.V.}{AFR \times W_{net}} \quad (19)$$

Thermal Efficiency is given by

$$\eta_{th} = \frac{W_{net}}{Q_{ad}} \quad (20)$$

Work Ratio:

The work ratio is given by the following equation:

$$WR = \frac{W_{net}}{W_t} \quad (21)$$

Specific Fuel Consumption:

If the mass of fuel consumed is given in kg/s and the net work developed is in kW then the specific fuel consumption will be calculated in kg/kWh.

$$SFC = \frac{3600 \times m_f}{W_{net}} \;\; (Kg/kWh) \quad (22)$$

Net Power

Net power available is calculated by the following equation.

$$\text{Net Power} = m_a \times w_{net} \times \eta_{gen} \quad (23)$$

If mass of air flow is given in kg/s and net power is given in kJ/kg then the net power will be calculated in kW.

3. Results and Discussion

In the following paragraphs the results of the present work have been discussed with the help of graphs. The results are based on the software developed in C++ and afterwards graphs have plotted with the help of menu driven software "Origin 50". The graphs are plotted for various parameters for different compressor, turbine efficiency, turbine inlet temperature, overall pressure ratios and regenerative effectiveness. The results have been given the thermal efficiency, compressor work, specific fuel consumption and power developed etc. of the gas turbine cycle. The various compressor efficiencies have been considered for the various output values.

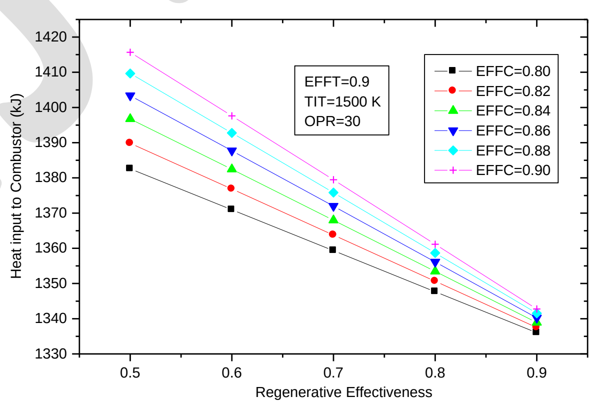

Fig. 5 Variation of Heat input to combustor with Regenerative effectiveness

Figure 5 represents the heat input to combustor with different regenerative effectiveness for different compressor polytropic efficiency. On increasing the compressor efficiency, heat input to combustor has been increases. On the other hand, increase in the regenerative effectiveness results in the decrease in the heat input to combustor of the gas turbine cycle

Figure 6 represents the compressor work with different regenerative effectiveness for different compressor polytropic efficiency. On increasing the compressor efficiency, compressor work has been decreases. On the other hand, increase in the regenerative effectiveness results in the increase in the compressor work of the cycle.

Figure 7 represents the thermal efficiency with different regenerative effectiveness for different compressor polytropic efficiency. On increasing the compressor efficiency, thermal efficiency has been increases. On the other hand, increase in the regenerative effectiveness also results in the increase in the thermal efficiency of the gas turbine cycle.

Figure 8 represents the specific fuel consumption with different regenerative effectiveness for different compressor polytropic efficiency. On increasing the compressor efficiency, specific fuel consumption has been decreases. On the other hand, increase in the regenerative effectiveness also results in the decrease in the specific fuel consumption of the gas turbine cycle.

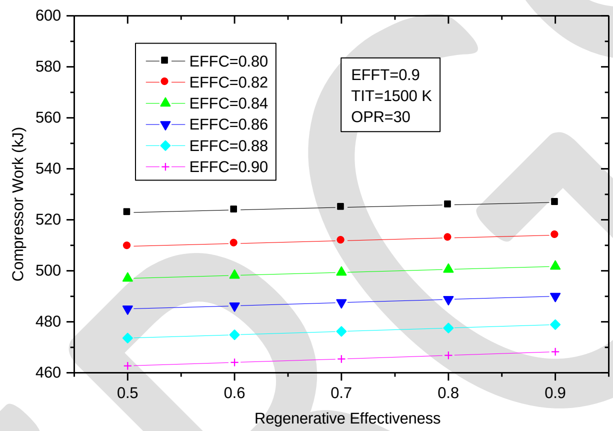

Fig. 6 Variation of Compressor work with Regenerative effectiveness

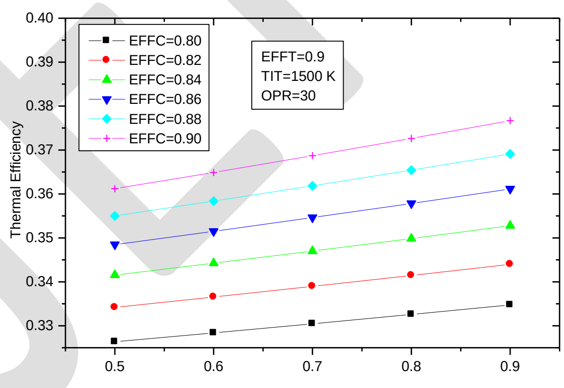

Fig. 7 Variation of Thermal efficiency with Regenerative effectiveness

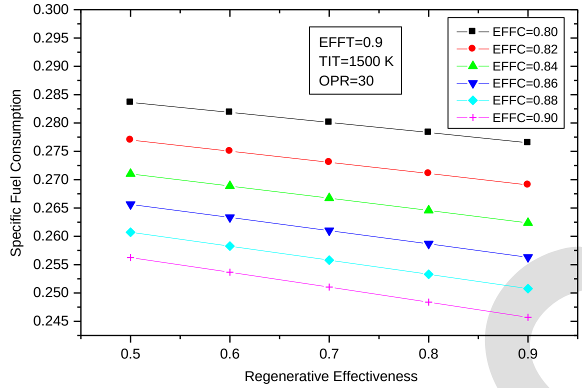

Fig. 8 Variation of Specific fuel consumption with Regenerative effectiveness

Figure 9 represents the heat supply to the combustor with different regenerative effectiveness for different compressor ratio. On increasing the compressor ratio, heat supply to the combustor has been increases. On the other hand, increase in the regenerative effectiveness results in decrease in the heat supply to the combustor of the cycle.

Figure 10 represents the thermal efficiency with different regenerative effectiveness for different compression ratio or overall pressure ratio (OPR). On increasing the OPR, thermal efficiency decreases. On the other hand, increase in the regenerative effectiveness results in increase in the thermal efficiency of the gas turbine cycle. At low OPR, thermal efficiency increases on increasing in the regenerative effectiveness but for higher values of OPR, the rate of increase in the thermal efficiency is very slow. Therefore, the optimum value of thermal efficiency has an optimum value of overall pressure ratio.

Figure 11 represents the heat input to the combustor with different turbine inlet temperature (TIT) for different compressor efficiency. On increasing the TIT, heat input to the combustor increases.

Figure 12 represents the compressor work with different compressor efficiency for different turbine inlet temperature (TIT). On increasing the turbine inlet temperature, compressor work increases. On the other hand, increase in the compressor efficiency, results in decrease in the compressor work of the gas turbine cycle.

Figure 13 represents the thermal efficiency with different compressor efficiency for different turbine inlet temperature (TIT). On increasing the turbine inlet temperature, thermal efficiency increases. On the other hand, increase in the compressor efficiency, results in increase in the thermal efficiency of the gas turbine cycle. Figure 14 represents the specific fuel consumption with different compressor efficiency for different turbine inlet temperature (TIT). On increasing the turbine inlet temperature, specific fuel consumption increases. On the other hand, increase in the compressor efficiency, results in decrease in the specific fuel consumption of the gas turbine cycle.

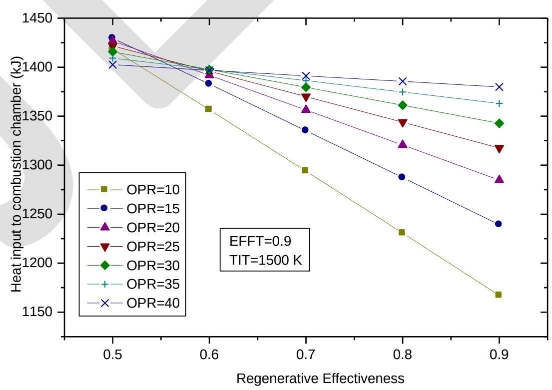

Fig.9 Variation of Heat input to combustor with Regenerative effectiveness

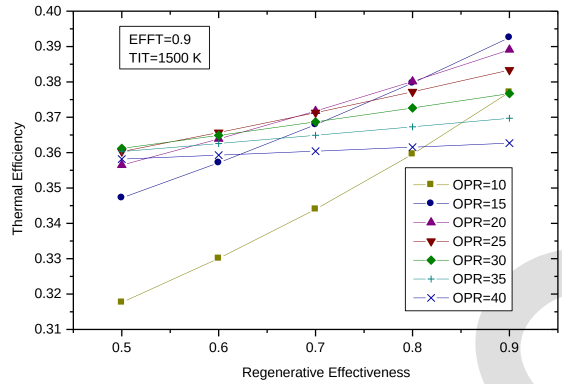

Fig. 10 Variation of Thermal Efficiency with Regenerative effectiveness

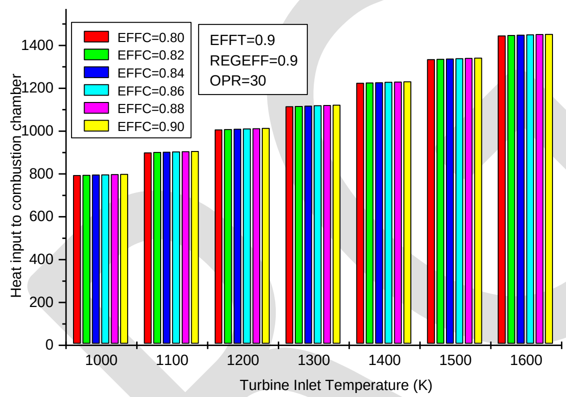

Fig.11 Variation of Heat input to combustor with TIT

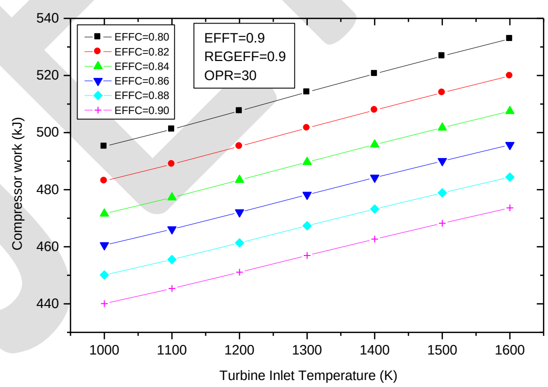

Fig. 12 Variation of Compressor work with Regenerative effectiveness

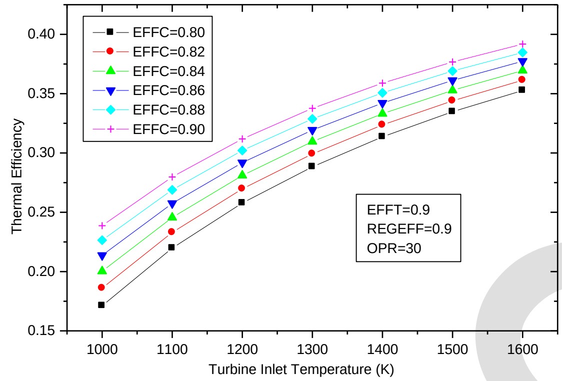

Fig. 13 Variation of Thermal efficiency with Turbine inlet temperature

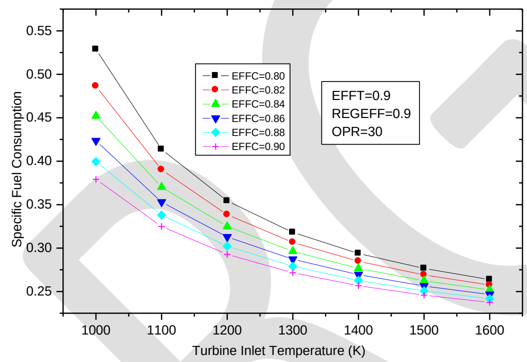

Fig. 14 Variation of Specific fuel consumption with Turbine inlet temperature

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4. Conclusion

The regenerative gas turbine power plant has been analyzed for various parameters. The most important parameter which has been covered in this work is the various polytropic efficiencies of compressor. The gas turbine regeneration package is designed to increase the efficiency of gas turbines with exhaust gas recirculation. Besides the gas turbine itself, four basic components are used to make up the efficiency package: the first is an exhaust gas cooler, the second is an exhaust gas cooling fan, the third is a recycle exhaust gas regenerator and finally there is a compressed ambient air regenerator. Regenerative gas turbine engine cycle is presented that yields higher cycle efficiencies than simple cycle operating under the same conditions. The power output, efficiency and specific fuel consumption are simulated with respect to operating conditions. The analytical formulae about the relation to determine the thermal efficiency are derived taking into account the effected operation conditions (ambient temperature, compression ratio, regenerator effectiveness, compressor efficiency, and turbine inlet temperature). Model calculations for a wide range of parameters are presented, as are comparisons with variable turbine and compressor efficiencies of gas turbine cycle. The power output and thermal efficiency are found to be increasing with the regenerative effectiveness, and the compressor and turbine efficiencies. The efficiency increased with increase the compression ratio to 15, then efficiency decreased with increased compression ratio, but in simple cycle the thermal efficiency always increase with increased in compression ratio. The increased in ambient temperature caused decreased thermal efficiency, but the increased in turbine inlet temperature increase thermal efficiency.

REFERENCES:

[1] “The History of Aircraft Gas Turbine Development in the United States”, St. Peter, J., Published IGTI, ASME, 1999.
[2] Sanjay et al. (2008), “Influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle”, Applied Thermal Engineering 28, 2315–2326

[3] Sanjay et al., “Comparative performance analysis of cogeneration gas turbine cycle for different blade cooling means”, International Journal of Thermal Sciences 48 (2009) 1432–1440
[4] Ashley De S and Sarim Al Zubaidy, “Gas turbine performance at varying ambient temperature”, Applied Thermal Engineering 31 (2011) 2735e2739
[5] J.W. Baughn, R.A. Kerwin, A comparison of the predicted and measured thermodynamic performance of a gas turbine cogeneration system, ASME Journal of Engineering for Gas Turbine and Power 109 (1987) 32–38.
[6] I.G. Rice, Thermodynamic evaluation of gas turbine cogeneration cycles: Part 1. Heat balance method analysis, ASME Journal of Engineering for Gas Turbine and Power 109 (1987)
[7] R. Bhargava, A. Peretto, A unique approach for thermo-economic optimization of an intercooled, reheated and recuperated gas turbine for cogeneration application, ASME Journal of Engineering for Gas Turbine and Power 124 (2001) 881–891
[8] F.S. Basto, H.P. Blanco, Cogeneration system simulation and control to meet simultaneous power, heat and cooling demands, ASME Journal of Engineering for Gas Turbine and Power 127 (2005) 404–409.
[9] M. Bianchi, G.N. Montenegro, A. di Peretto, Cogenerative below ambient gas turbine performance with variable thermal power, ASME Journal of Engineering for Gas Turbine and Power 127 (2005) 592–598.
[10] A. Poullikkas, An overview of current and future sustainable gas turbine technologies,
Renewable and Sustainable Energy Reviews 9 (2005) 409–443.
[11] Torbidini, L. and A. Massardo, Analytical Blade Row Cooling Model for Innovative Gas Turbine Cycle Evaluations Supported by Semi-Empirical Air- Cooled Blade Data. Journal of Engineering for Gas Turbines and Power, 2004 126: p. 498-506.
[12] Vittal, S., P. Hajela, and A. Joshi, Review of Approaches to gas turbine life management, in 10th AIAA/ISSMO Multidisciplinary Analysis and Optimization. 2004, AIAA: Albany, NY.
[13] Zifeng Yang and Hui Hu, “An experimental investigation on the trailing edge cooling of turbine blades”,Propulsion and Power Research 2012;1(1):36–47
[14] Cun-liang Liu et al., “Film cooling performance of converging slot-hole rows on a gas turbine blade”, International Journal of Heat and Mass Transfer 53 (2010) 5232–5241
[15] Mahmood Farzaneh-Gord and Mahdi Deymi-Dashtebayaz, “Effect of various inlet air cooling methods on gas turbine performance”, Energy 36 (2011) 1196-1205
[16] J. H. Horlock (2003), “Advanced Gas Turbine Cycles”, F. R. Eng., F.R.S. ELSEVIER SCIENCE Ltd The Boulevard, Langford Lane Kidlington, Oxford OX5l GB, UK
[17] Thamir K. Ibrahim et al., “Improvement of gas turbine performance based on inlet air cooling systems: A technical review”, International Journal of Physical Sciences Vol. 6(4), pp. 620-627, 18 February, 2011
[18] Ashok D. Rao and David J. Francuz, “An evaluation of advanced combined cycles”, Applied Energy 102 (2013) 1178–118