Regenerative Gas Turbine Power Plant: Performance & Evaluation

Momin Elhadi Abdalla
University of Khartoum, Sudan, mominhadi@yahoo.com

Siddharth Pannir
GenH Inc

Amina HusseinAhmed
GenH Inc

Aya Mahjob
GenH Inc

Salah Ahmed Abdalla
GenH Inc

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Abdalla, Momin Elhadi*[1]; Siddharth Pannir [2]; Hussein Ahmed, Amina[3]; Mahjob, Aya[4]; Abdalla, Salah Ahmed[5]

Organization(s): 1,3,4: University of Khartoum, Chemical Engineering Department, Khartoum, Sudan
2: GenH, Charlestown, Massachusetts, United States
5: University of Khartoum, Energy Research Centre, Khartoum, Sudan

[1] University of Khartoum, Chemical Engineering Department,Khartoum, Sudan, mominhadi@yahoo.com
[2]GenH Inc, Charlestown, Massachusetts, United States, Siddharth.pannir@genh.co

ABSTRACT

In this work, comprehensive operational and conceptual design basics of the Regenerative Gas Turbine were studied and applied to the Khartoum North Thermal Power Station, Sudan, which has a total power of 187MW. The analysis and results of this work were executed using the Engineering Equation Solver.The results show that, the increasing the effectiveness of the regenerative cycle increased the thermal efficiency. However, there is a turning point of compressor inlet temperature, after which the further increase of temperature and regenerator effectiveness will lead to decline in the thermal efficiency of the cycle. At lower regeneration and moderate regenerator effectiveness, the increase in compression ratio leads to an increase in thermal efficiency of the cycle. At the highest values of regeneration effectiveness, the increase in compression ratios reduced the thermal efficiency of the cycle. The results revealed that regeneration is more effective at lower pressure ratios, ambient temperatures, and low minimum (compressor) to maximum (combustor) temperature ratios. An increase in regeneration effectiveness decreases the specific fuel consumption for lower and moderate compression ratios. At higher compression ratios, increasing regenerator effectiveness leads to an increase in the specific fuel consumption (SFC) of the cycle. At low and moderate compressor inlet temperature, increasing the regenerator effectiveness decreases fuel demand in the combustor, which reflects in decreasing the heat rate to the combustor especially at higher regenerative effectiveness (e=95%). As the effectiveness varies between 10-75%, the compressor inlet temperature varies from 200K to 350K and the regenerator exhaust temperature exhibited different profiles according to the conditions of inlet temperature. It was found that power curve declines smoothly due to the increase in irreversibility of regeneration cycle and remains high at higher turbine inlet temperatures. Compressor inlet temperatures between 100-330K increase the regeneration effectiveness varying between 10-95%, resulting a in different profile of the combustor inlet temperature. The mass flow rate of the fuel in the combustor decreases with increasing regeneration effectiveness at lower compressor inlet temperatures. At higher inlet temperatures, the fuel flow rate will gradually increase with the regeneration effectiveness due increasing irreversibilities of the regenerator. For a compression ratio of 15, the fuel mass flow rate reaches the lowest value of (6.30 kg/sec) at the lowest ambient temperature of 200 K and a regenerative effectiveness of 95%. The increase of the lower heating value (LHV) leads to a gradual increase in the thermal efficiency of the regenerative gas turbine (RGT), due to increasing cycle power and combustor capacity. The results concluded that the regeneration effectiveness is higher at low and moderate compressor inlet temperatures and compression ratios, through which, avoiding the regenerator's irreversibility is possible.

1. INTRODUCTION

In light of the detrimental effects that fossil fuels have on changing the climate system, multiple attempts have been made to find ways for a safe and environmentally benign thermal power source. This has led to extensive research on lowering the gas emissions as well as increasing power plant efficiency based on complex [3] [24] gas turbine (GT) setups. Reducing long-term greenhouse gas (GHG) emissions of the energy industry [25] is one of the toughest challenges of the energy transition. These emissions result from the combustion of fossil fuels for energy purposes and from other process emissions.

Continuous population growth and increasing economic activity inSudan, has led to an increased need to demand a large build power supplier of the GT Plants [16]. These plants generate maximum output at summer ambient temperature ratings. The use of GT to generate electricity has become an attractive endeavor due to the comparatively low initial capital cost as well as its stability of supply under varying circumstances. Another outstanding feature of this equipment is its capability of quick starting using a wide variety of Fuels from natural gas to residual oil, biomass, and powdered coal[1][12][22][23]. Additionally,GT equipment benefit from better construction materials and the use of adequate blade cooling systems [15][17] to counter the inlet gas temperature which can often exceed 1200^{o}C [1][5][8][9]. As a result of this, the overall thermal efficiency of a GT plant can be about 35%, which is almost the same as that of a conventional steam power plant. Moreover, the GT normally characterized with its low weight per unit power, is used to drive aviation systems on all kinds on aircrafts. It is also being increasingly used in land vehicles like buses and trucks and to drive locomotives and marine ships. In oil and gas industries, the GT is widely employed to drive auxiliaries like compressors, blowers, and pumps [1] [11].

Researchers have conducted research into different methods to increase thermal efficiency of regenerative GTcycles [13][19], one of which is the reheating process used to increase thermal efficiency of gas and steam turbine cycles. Similarly, regeneration is utilized to increase thermal efficiencies of both the simple GT and steam turbine cycles. Another important procedure to increase the thermal efficiency of the power plant cycle is the combined cycle, which consists of a GT and a steam turbine cycles [1] [2] [6].

This work aims to reinforce the understanding of a regenerative GT as a thermal process utilizing the regeneratingenergy of the exhaust gases departing the turbine unit as well as applying similar design parameters to theKhartoum North Thermal Power Station (GT, 187 MW) [16] in Sudan. The code of the performance model for the regenerative GTwas developed on theEES.The effects of the operating parameters were analyzed on the RGT power plant. These operating parameters include compression ratio, ambient temperature, turbine inlet temperature, and regenerator effectiveness. This study pushes for establishing a qualified operational and conceptual design procedure for the regenerative GTunit. The work also presents a preliminary strategy toidentify the performance and evaluation criterion of the regenerative GT power plant utilizing the effect of various operating conditions.

2. MODELING OF COMPONENTS

Most of the properties of air and combustion gas products were predicted by the variation of specific heat and thermodynamic functions.

The GT power plants consist of four components including the compressor, combustion chamber (CC), turbine, and regenerator. The combined cycle arrangement considered in Fig.1 is a clear presentation on how to utilize the hot turbine exhaust gas. Fresh atmospheric air is filtered and drawn continuously into the compressor, and thenthe energy is added by the combustion of the fuel in the combustion chamber unit. The products of combustion are expanded through the turbine [7]and consequently produce electrical work while the rest of the exhaust gases aredischarged into theregenerator. The counter current *regenerator* allows the air exiting the compressor to be *preheated* before entering the combustor [4], thereby reducing the amount of the fuel that must be supplied to the combustor itself.

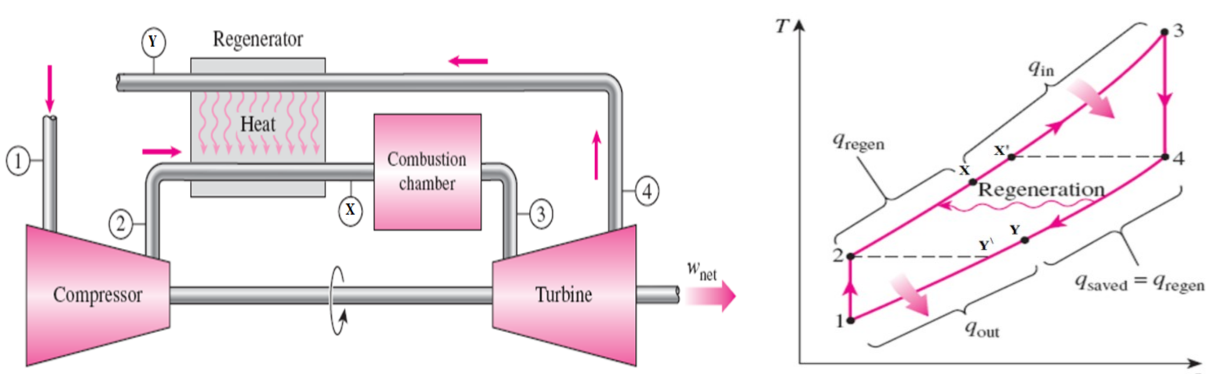

Fig.1: The regenerative GT cycle, T-S diagram.

The intake pressure at the compressor inlet was modeled with the following equation [13]:

$$P_1 = P_{atm} - \Delta P_{intake} \tag{1}$$

Where the intake pressure drop (ΔP_{intake}) was taken to be 0.005 bar, and the intake temperature was modeled as the ambient temperature. Theprocess on the temperature-entropy diagram is represented in Fig.1.The compressor compression ratio (r_P) can be defined as [2]:

$$r_P = \frac{P_2}{P_1} \tag{2}$$

where $P_1$and $P_2$are compressor inlet and outlet air pressure, respectively. Accordingly, the isentropic outlet temperature leaving the compressor is modeled by the equation[1][14][18]:

$$\frac{T_1}{T_{2s}} = \frac{P_2}{P_1}^{\frac{\gamma_a - 1}{\gamma_a}} \tag{3}$$

The specific heat ratio for air $\gamma_a$was taken as 1.4 and was predicted at $\gamma_g = 1.3$ for the gas.The isentropic efficiency of the compressor and turbine was taken to be in the range of 85% to 90%. The isentropic compressor efficiencyis expressed by the equation [4] [21]:

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{4}$$

Where, T_1 and T_2 are the compressor inlet and outlet air temperatures respectively and T_{2s} is the compressor isentropic outlettemperature. Thespecific work required to run the compressor work (W_C) is modeled with the following equation [21]:

$$W_c = m_a C_{P_a} \; T_2 - T_1 \; = m_a C_{P_a} T_1 \; \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \tag{5}$$

With the specific heat of air taken as $C_{P_{air}} = 1.005 \; \frac{kJ}{kgK}$, which can be substituted into Equations (6) and (7) for the range of [21]:

If ($T_1 \leq 800K$)

$$C_{P_{air}} = 1018.9 - 0.1378 \times T_1 + 1.9843 \times 10^{-4} \times T_1^2 + 4.2399 \times 10^{-7} \times T_1^3 - 3.7632 \times 10^{-10} \times T_1^4 \tag{6}$$

If ($T_1 > 800K$)

$$C_{P_{air}} = 7.9865 \times 10^2 - 0.5339 \times T_1 - 2.2882 \times 10^{-4} \times T_1^2 + 3.7421 \times 10^{-8} \times T_1^3 \tag{7}$$

T
he specific heat of the flue gas (C_{pg}) is given by Naradasuetal.(2007) [21]:

$$C_{P_g} = 1.8083 - 2.3127 \times 10^{-3} \times T + 4.045 \times 10^{-6} \times T^2 - 1.7363 \times 10^{-9} \times T^3 \tag{8}$$

From the energy balance in the combustion chamber [1]:

$$m_a C_{P_a} T_x + m_f LHV + m_f C_{P_f} T_f = \; m_a + m_f \; C_{P_g} T_{IT} \tag{9}$$

Where $m_f$is the fuel mass flow rate in (kg/sec), $m_a$is the air mass flow rate (kg/sec), LHV is the fuel'slower heating value (the fuel used has a value of 48 MJ/kg),$T_{IT}$is the turbine inlet temperature, $C_{P_f}$is the specific heat of fuel, andT_f is the temperature of the fuel. The specific heat of the flue gas was modeled with $C_{P_g} = 1.07 \; kJ/kg.K$; efficiency was set at 95%, and a pressure drop of$\Delta P_{C,C} = 0.4785 \; bar$ in the combustor. Accordingly, the efficiency of the combustor was modeled as [1]:

$$\eta_{C,C} = \frac{m_g C_{P_g} T_{IT} - m_a C_{P_a} T_x}{m_f LHV} \tag{10}$$

The air fuel ratio at the combustor was modeled according to the following equation:

$$AFR = \frac{A}{F} = \frac{m_a}{m_f} \tag{11}$$

Where the total mass flow rate is given by:

$$m_g = m_a + m_f \tag{12}$$

The discharge gas of the turbine was predicted according to the equation:

$$\frac{T_3}{T_{4s}} = \quad \frac{P_3}{P_4}^{\frac{\gamma_g - 1}{\gamma_g}} \tag{13}$$

Where the actual outlet temperature leaving the turbine at the isentropic conditions was modeled according to:

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} \tag{14}$$

The regenerator effectiveness ε was modeled according to the equation [4]:

$$\varepsilon = \frac{T_x - T_2}{T_4 - T_2} \tag{15}$$

where T_x is the combustor inlet temperature.The shaft work produced from the turbine is determined by the equation [20]:

$$W_{RGT} = m_g C_{P_g} \; T_4 - TIT \; = m_g C_{P_g} \eta_t T_{IT} \; 1 - \frac{1}{r_p^{\frac{\gamma_a - 1}{\gamma_a}}} \tag{16}$$

The network from the GT unit was expressed by the equation:

$$W_{RGT,Net} = W_{RGT,Net} - W_C = m_g C_{P_g} \eta_t T_{IT} \; 1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}} \; - m_a C_{P_a} T_1 \; \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c} \tag{17}$$

The output power from theGT is expressed with the equation [1] [20]:

$$P_{RGT} = [W_{RGT,Net} - W_C] \times \eta_{Mech} \eta_{Gen} \tag{18}$$

The mechanical (η_{Mech}) and generator (η_{Gen}) efficiencies were taken to be 92% and 95% respectively.
The heat supplied was expressed with the equation [1]:

$$Q_{add} = m_g C_{P_g} T_{IT} - m_a C_{P_a} T_x \tag{19}$$

The heat supplied (per kg. air) to the combustor was modeled according to the equation [1]:

$$Q_{add} = \frac{m_f \times \eta_{C,C} \times LHV}{m_{air}} = \frac{\eta_{C,C} \times LHV}{AFR} \tag{20}$$

The GT efficiency was determined by the equation [1]:

$$\eta_{over,RGT} = \frac{W_{RGT,Net}}{Q_{add}} \tag{21}$$

Accordingly, the heat rate (HR) which is defined as the consumed heat to generate unit energy of electricity was determined by the equation [1] [10]:

$$HR = \frac{3600 * m_f * LHV}{\eta_{over,RGT}} \tag{22}$$

The specific fuel consumption (SFC) is determined by the equation [1]:

$$SFC = \frac{3600 * m_f}{\eta_{over,RGT}} \tag{23}$$

3. RESULTS AND DISCUSSIONS

The analysis and results of this work was executed using thermodynamic EES codes. The simulation results display the effectiveness of regeneration and other important parameters on the performance of the RGT.

As can be observed in Fig.2, the increaseof the compressor inlet temperature leads to a decrease in thermal efficiency of the cycle due to change in air density, increase in compressor work, and fuel demand.It was also observed that at constant compressor inlet temperature, the increases in effectiveness of the regenerative cycle increased the thermal efficiency (Fig.2) of the GTcycle. The results show that, there is a turning point of compressor inlet temperature (280K), through which the further increase of the temperature and the regenerator effectiveness will lead to declining thermal efficiency of the cycle. Regenerative effectiveness peaks and then declines due to friction, mechanical losses, and shifting of pressure drops during the heat exchange process between the regenerator and the combustor.

To a certain extent in RGT power plants, increasing the compression ratio results in an optimum thermal efficiency at varying regenerator effectiveness. As indicated in Fig.3, at lower and moderate regeneration effectiveness, increase in compression ratio leads to an increase in thermal efficiency of the cycle. However, at the highest values of regeneration effectiveness, increase in compression ratios will lead to a decline in thermal efficiency of the cycle i.e., for each degree of regeneration there is an optimum compression ratio for maximum RGT thermal efficiency. Generally, thermal efficiency reaches a maximum value at optimum compression ratio through which maximum real work occurs. Thereafter, work will decrease and increasing the compression ratio will reduce the thermal efficiency of the cycle, as shown in Fig.4.
With an air flowrate of 500 kg/sec, the thermal efficiency increases sharply (Fig.4), especially between compression

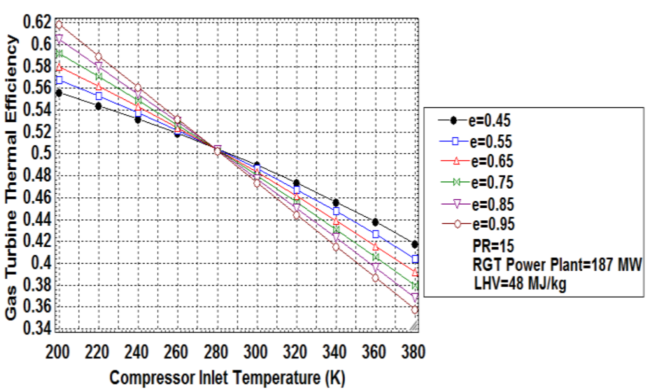

Fig.2: Thermal efficiency versus air (ambient) temperature for different regenerative effectiveness (e).

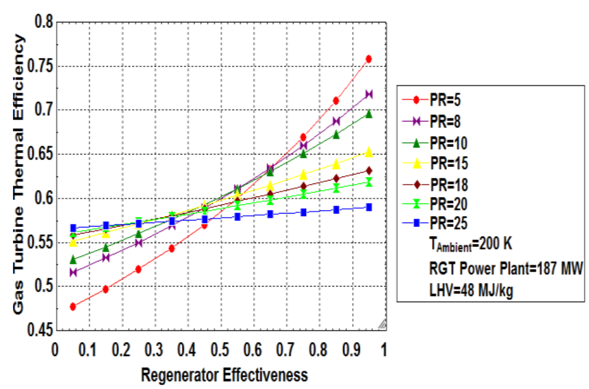

Fig.3: Variation of regenerator effectiveness with GT thermal efficiency at different compression ratios.

ratios of 4-10 where the efficiency isbetween 25-36%. Regeneration is more effective at lower pressure ratios, ambient temperaturesandlow minimum (compressor) to maximum (combustor) temperature ratios(10-17%).
The effect of different compression ratios, from 5 to 30, on specific fuel consumption (SFC) and ambient temperatures, between 200K and 300K,isshown in Fig.5. At constant compression ratio the increase of ambient temperature will lead to increased specific fuel consumption due to change in air mass and compressor work. The results indicate that the decrease in ambient temperature leads to decrease in specific fuel consumption for the whole cycle's conditions. As is evident from Fig.5, the decrease in ambient temperature led to the lowest specific fuel consumption (0.132 kg/kW.h), at optimum lowest compression ratio (rp=8) at 200K.Thereafter, the increase in compression ratio will lead to increase in fuel demand in the cycle at higher ambient temperatures.

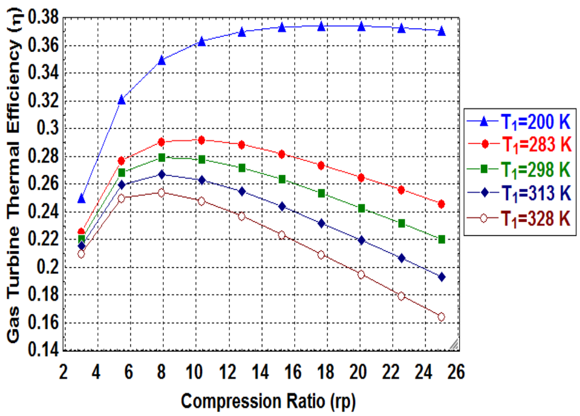

Fig.4: Thermal efficiency versus compression ratio for different inlet air temperatures.

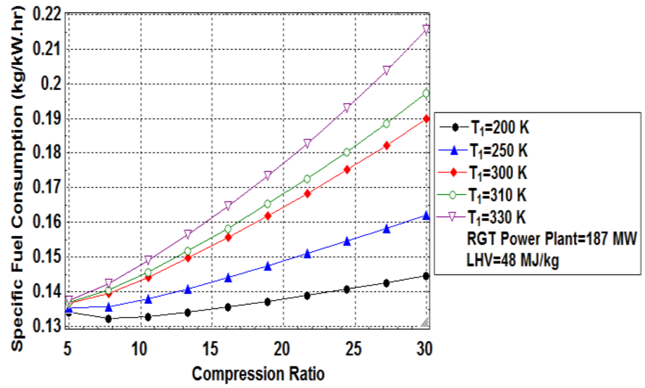

Fig.5: Effect of the compression ratio on the SFC for different ambient temperatures.

Fig.6, plots variationin compression ratio, ranging from4 through 30, with specific fuel consumption at different regeneration effectiveness,rangingfrom45% through 95%, at a constant inlet air temperature (T=200 K), and regenerative power of 187 MW.The results indicate that increasing the compression ratio leads to an increase in SFC. The general trends exhibit an increase in regeneration effectiveness which will decrease the specific fuel consumption for a lower and moderate compression ratio. At higher compression ratios the increase of regenerator effectiveness led to increase the SFC of the cycle.

Consequently, Fig.7 represents the effect of compression ratio values,rangingfrom 3 to 25, and regeneration effectiveness,varying between 45-95%,on thermal efficiency of the cycle. The addition of regenerationhas increased the thermal efficiency at constant ambient temperature, for low and moderate compression ratio's conditions. As evident in Fig.7, for the lowest ambient temperature (T=250 K), there is an optimum compression ratio (r_p=5) for which the maximum thermal efficiency (η=39%) varied with the maximum regeneration effectiveness (e=95%).

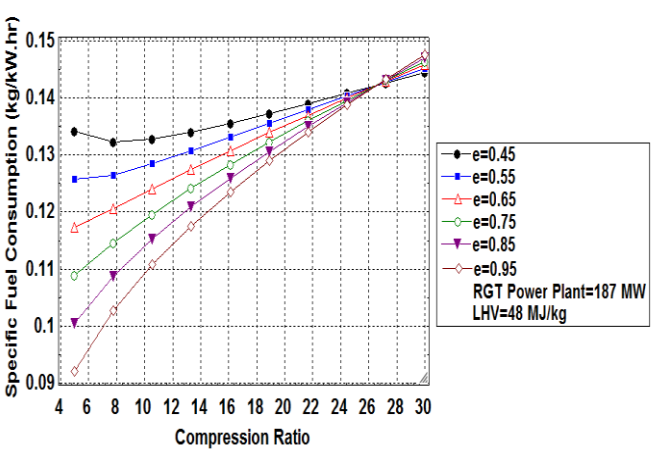

Fig.6: Influence of compression ratio on specific fuel consumption at different regeneration effectiveness(e.)

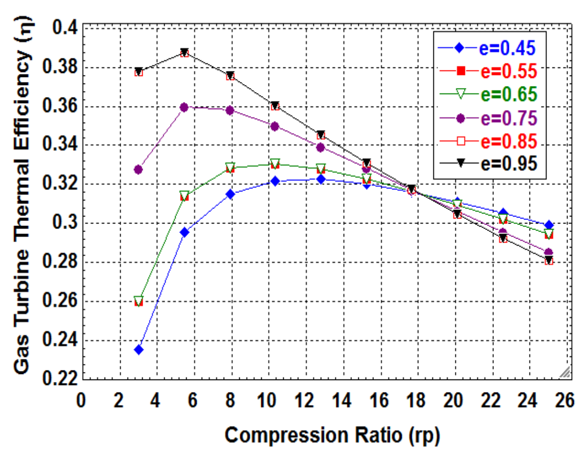

Fig.7: Effect of compression ratio and regeneration effectiveness (e) on thermal efficiency.

Fig.8, displays the influence of compressor inlet temperatures between 200K to 340K, on the combustor heat rate at different regeneration effectiveness varying from 45-95%.At constant overall compression ratios of (PR=15) and overall cycle power of 187 MW, the increase of compressor inlet temperature leads to increase the combustor heat rate. At low and moderate compressor inlet temperature, the addition of the regenerator effectiveness decreases fueldemand in the combustor, which reflects inthe decreasing heat rate to the combustor. At higher compressor inlet temperature, the demand of fuel by combustor will increase and thus increasing theregenerator effectiveness will lead to an increase in the combustor heat rate.

At constant overall cycle power of 187 MWand regeneration effectiveness of 95%, Fig.9 plots the variation of different compression ratio, varying between, 5-40, with the compressor work at different ambient temperatures ranging from 200-300K. Increasing the compression ratio led toan increase in the compressor's work and the highest values of work were reached at higher ambient temperatures.

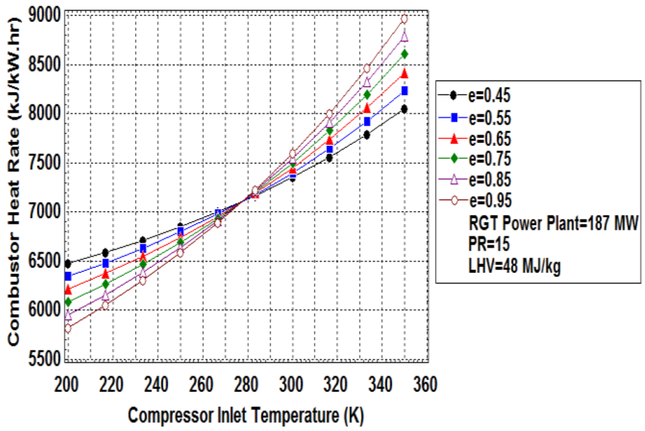

Fig.8: Variation of compressor inlet temperature with the combustor heat rate including different regeneration effectiveness (e).

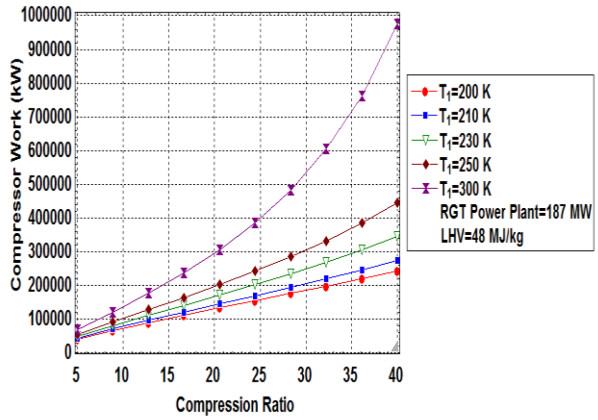

Fig.9: Variation of compression ratios with compressor work at different ambient temperatures.

The relationship betweenregenerator effectiveness, varying from5-95%, andcombustor fuel mass flow rate at different compression ratios (PR=5-25), is plotted in Fig.10. As can be observed from Fig.10, at lower and moderate regenerator effectiveness, anincrease in compressionratios decreases the combustor fuel mass flow rate of the cycle. At higher regenerator effectiveness, the increase of the compression ratio leads to increase in fuel demand due to irreversibilities at the regenerator and the combustor.

In Fig.11, the relationship of regenerator effectiveness to the regenerator exhaust temperature at varying compressor inlet temperatures is plotted. As the effectiveness varies between 10-75%,the compressor inlet temperature varies from 200K to 350K, the regenerator exhaust temperature exhibited different profiles according to the conditions of the inlet temperature. The regenerator exhaust temperature revealed decreasing value at lower compressor temperatures and increasing values at higher compressor inlet temperatures due to the increase in the irreversibilities at the compressor and regenerator.

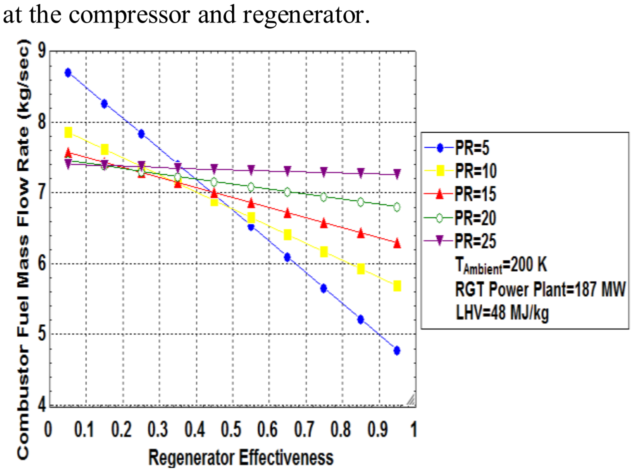

Fig.10: Variation of regenerator effectiveness with fuel mass flow rate in the combustor at different compression ratios.

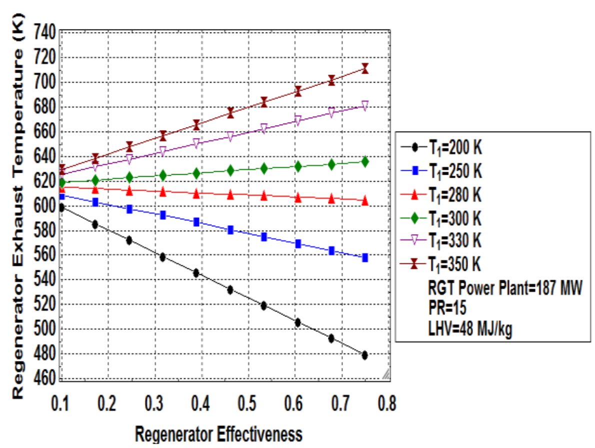

Fig.11: Effect of regenerator effectiveness on the regenerator exhaust temperature at different compressor inlet temperatures.

Fig.12,shows that increasing the turbine inlet temperature (T_{IT}) with a low regenerator effectiveness will result in an increased regenerator exhaust temperature due to gradual increase in cycle power and turbine outlet temperature. Although regenerator heat exchanger factor "effectiveness" promotes higher turbine inlet temperatures rather than the exhaust temperatures, the irreversibilities friction, mechanical losses, and fluctuation of average mean temperature will lead to depreciating efficiency.

Fig.13, plots the effect of turbine inlet temperature, between 1000-1800K, on RGT thermal efficiency for regenerationeffectiveness, ranging from45-95%. As can be noted from Fig.13, the thermal efficiency of the cycle increasesgraduallywith increasingturbine inlet temperature, as there is a further increase of the cycle's power. Thermal efficiency remains high at higher regeneration effectiveness.

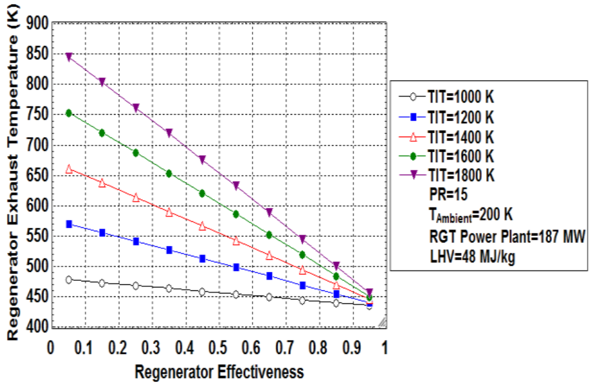

Fig.12: Effect of regenerator effectiveness(e) on regenerator exhaust temperature at different turbine inlet temperatures.

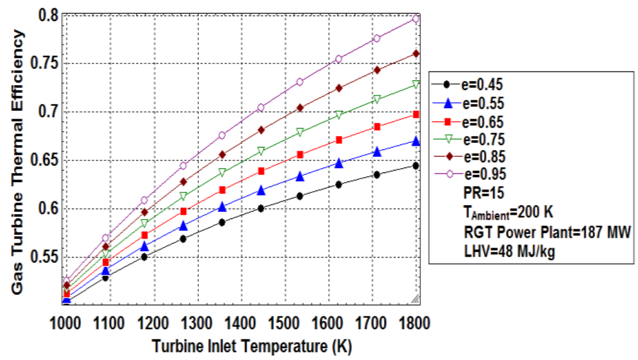

Fig.13: Effect of turbine inlet temperatures (TIT) on RGT thermal efficiency at different regenerator effectiveness.

With compression ratio held constant at 10,air flow rate at 500 kg/sec, and compressor inlet temperature of 200K,Fig.14presents the effect of regenerator effectiveness on the GT power at different turbine inlet temperatures. The power curve smoothly declines due to increase inthe regenerator's irreversibility. The power remains high at higher turbine inlet temperatures.

At compressor inlet temperaturesbetween 100-330K and regeneration effectivenessbetween 10-95%,there aredifferent profilesof the combustor inlet temperature as shown in Fig.15. Increasing the combustor inlet temperature sharply reduces the amount of specific fuel consumption, particularly at lower ambient temperatures.

The combustor inlet temperature increases value at lower compressor temperatures, while decreasing at higher compressor inlet temperatures from the increase in irreversibilities at the regenerator and combustor. At the highest regeneration effectiveness, the combustor inlet temperature stabilizes.

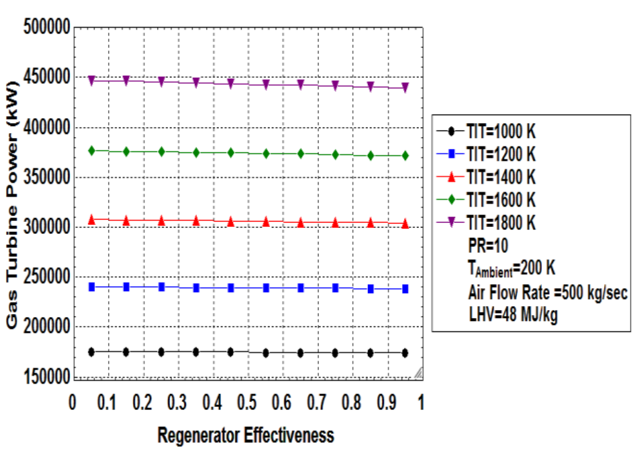

Fig.14: Effect of regenerator effectiveness on RGT thermal efficiency at different turbine inlet temperatures (TIT).

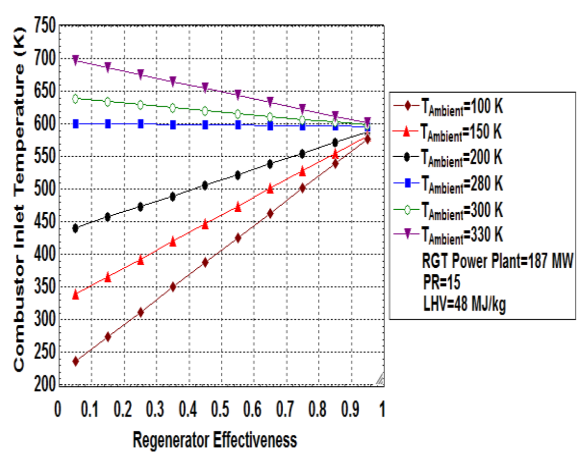

Fig.15: Effect of regenerator effectiveness on the combustor inlet temperature at different compressor inlet temperatures.

Fig.16, plots the variation of air mass flow rate, between 200-500 kg/s,toGT power at different compressor inlet temperatures rangingfrom 200-330K. The increase in mass flow rate of air directly increasesthe power of the plant, reaching a maximum value at the lowest ambient temperatures. The flow rate of the air is the major controlling parameter of increasing the power for the GTcycle. However,increasing the airflow rate will require more fuel inside the combustor, gradually increasing the specific fuel consumption of the cycle.
Fig.17, indicates the influence of regenerator effectiveness on the combustor's fuel mass flow rate at different compressor inlet temperatures varying between 200-350K.As the regenerator effectiveness varies from 5% to 95%,the compressor inlet temperature ranges from 200K to 350K, the mass flow rate of the fuel in the combustor exhibited different profiles according to the conditions of the inlet temperature, as shown in Fig.17.

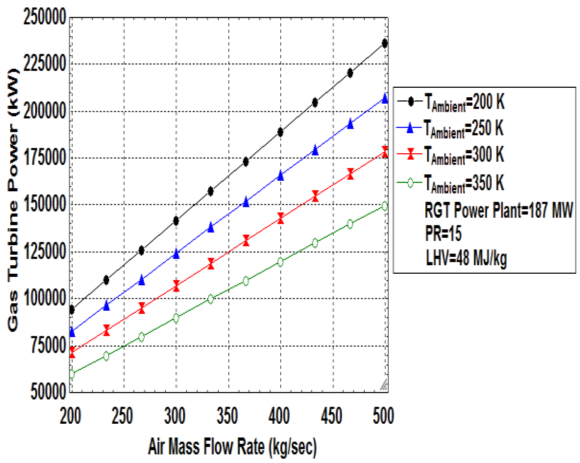

Fig.16: Variation of air mass flow rate with RGT power at different compressor inlet temperatures.

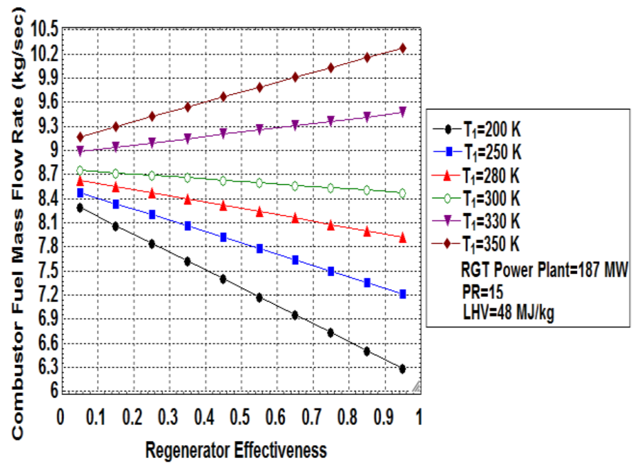

Fig.17: Influence of regenerator effectiveness on the fuel mass flow rate in the combustor at different compressor inlet temperatures.

Fig.18, shows that the fuel lower heating value (LHV) has great influence on the cycle's efficiency. The increase in LHV leads to a gradual increase in the thermal efficiency of the RGT, because of an increased in cycle power and the combustor capacity. At higher LHV of 50 MJ/kg, inlet temperature of 200 K, and power output of 187MW, the regenerative effectiveness increases the RGT thermal efficiency gradually, reaching a lower value of 59.40% at 45% regenerator effectiveness and a higher value of 65.40% at 95% regenerator effectiveness.The results show that the regeneration effectiveness is more effective at low inlet temperatures through which the regenerator's irreversibility can be avoided.
The mass flow rate of the fuel in the combustor decreases with increasing regeneration effectiveness at lower compressor inlet temperatures as shown in Fig. 19. However, at ~318K, the regenerator effectiveness does not affect the relationship between combustor fuel mass flow rate and compressor inlet temperature. Following this point, higher compressor inlet temperature and regenerator effectiveness increase the fuel flow rate, from increasingirreversibilities in the combustor and regenerator. For a regenerative power of 187MW, and compression

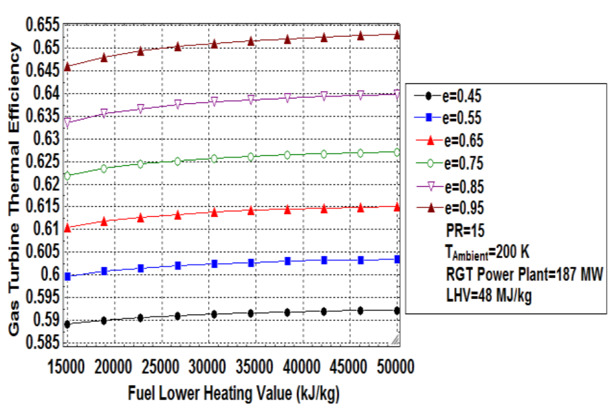

Fig.18: Variation of fuel lower heating value with RGT thermal efficiency at different regenerator effectiveness.

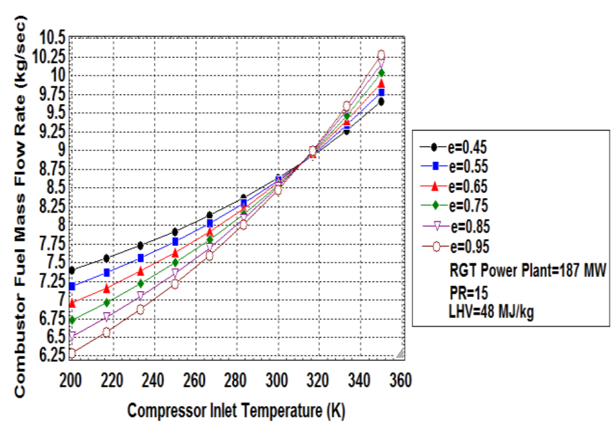

Fig.19: Influence of compressor inlet temperature on the fuel mass flow rate in the combustor at different regenerator effectiveness.

ratio of 15, the fuel mass flow rate reaches the lowest value of (6.30 kg/sec) at the lowest ambient temperature of 200 K and a regenerative effectiveness of 95%.The fuel mass flow rate reaches the highest value of (10.25 kg/sec) at the highest ambient temperature of 350 K and a regenerative effectiveness of 95%.

CONCLUSIONS

This work discussed the performance and evaluations of the RGT power plants including the effect of the regeneration. The results show various reasons and justifications for using the regenerative unit including different aspects of the fuel demand and thermal efficiency. A rigorous parametric study was introduced and executed for each unit of the plant's cycle. In addition, the work delivered various results and investigations with different variables such as compressor parameters and regeneration effects on the output power and the thermal efficiency of the RGT power plant applied to the Khartoum North Thermal Power Station (GT,187 MW).The variation in operating conditions (regenerative effectiveness, compression ratio, turbine inlet and exhaust temperature, combustor inlet temperature, combustor fuel mass flow rate,fuel'scaloric value, and ambient temperature) on the performance of GT (thermal efficiency, compressor work, power, specific fuel consumption, heat rate) were successfullyinvestigated. The parametric study revealed thatthe regenerative effectiveness, compression ratio, inlet air temperaturehad a significant effect on the thermal efficiency and power output of a RGTpower plant. The major suggestions to enhance the thermal efficiency of the regenerative cycle is the development of multistage turbine expansions with reheat units to increase the turbine inlet temperature beside multistage compressions with intercoolingunits which demands lower compressor inlet temperatures and fuel consumptions.

NOMENCLATURE

Symbols

T	Temperature	(K)	T_x	Combustor Inlet Temperature	(K)
S	Entropy	(kJ/kg.K)	$\Delta P_{C,C}$	Combustor Pressure Drop	(bar)
P	Pressure	(kPa)	$\eta_{C,C}$	Combustor Efficiency	-
r_P	Compression Ratio	-	ε	Regenerator Effectiveness	-
γ	Specific Heat Ratio	-	$\dot{W}_{GT}$	Turbine Shaft Work	(MW)
η_C	Isentropic Compressor Efficiency	-	η_T	Turbine Efficiency	-
T_S	Compressor Isentropic Temperature	(K)	P_{GT}	GT Power	(MW)
$\dot{W}_C$	Specific Compressor Work	(MW)	$\dot{Q}_{add}$	Heat Supplied	(kW)
$\dot{m}_a$	Air Mass	(kg.air)	C_{P_a}	Heat Capacity of Air	(kJ/kg.K)
$\dot{m}_f$	Fuel Mass	(kg.fuel)	C_{P_f}	Heat Capacity of Fuel	(kJ/kg.K)
$\dot{m}_g$	Gas Mass	(kg.gas)	C_{Pg}	Heat Capacity of Flue Gas	(kJ/kg.K)

Subscripts

PR	Pressure ratio
RGT	Regenerative Gas Turbine
Mech	Mechanical
Gen	Generator
AFR	Air Fuel Ratio
GHG	Greenhouse Gas
EES	Engineering Equation Solver
CC	Combustion Chamber
ATM	Atmospheric
LHV	Caloric Value
TIT	Turbine Inlet Temperature
HR	Heat Rate

SFC Specific Fuel Consumption

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