EFFECT OF AMBIENT TEMPERATURE ON A COMBINED REGENERATIVE AND REHEAT GAS TURBINE CYCLE USING MATLAB

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Abstract- In recent development of gas turbine cycles, new software for calculating thermal efficiency and power output of a combined reheat-regenerative cycle are applied.There are one compressor and one turbine used in regeneration gas turbine cycle and one compressor and two turbines are used in reheat cycle. In the combined cycle, one compressor and two turbines namely, HP turbine and LP turbine are used in this work. The temperature after reheating assumes to be reaching at the same temperature of HP turbine inlet.A full numerical model for the engine is built. This model takes into account the variations in specific heat and the effects of turbine cooling flow. Also, the model considers the efficiencies of all components, effectiveness of heat exchangers and the pressure drop in relevant components. The thermodynamic analysis has been performed using MATLAB software. The parameters taken in the well defined range for overall pressure ratio, turbine inlet temperatures and ambient temperature. The cycle performs the analysis for various regenerative effectiveness and various turbine inlet temperatures. It is found that the mass flow ratedecreases on increasing the regenerative effectiveness while increases on increasing the turbine inlet temperature. Also found that the heat required in the burner decreases for higher regenerative effectiveness.

Keywords- Gas Turbine, Regenerative Effectiveness, Reheat cycle, Thermal Efficiency.

1. INTRODUCTION

Gas turbines can bedivided into important categories. There are industrial gas turbines and there are jet engine gas turbines. Both types of gas turbines have a short but interesting background. To define the compressor and the turbine, you need to use aerodynamics. The reheat process is replaced by processes of heating the expanded gases while passing through different turbine stator blades. Small amount of combusted gases is utilized to flow inside such blades for heating and mixing with the expanded gases [1]. The performance of actual cycles being used in gas turbines is discussed. A general overview of combined-cycle plants is provided [2]. The gas turbine engine is compact, has a light weight and can operate withmodification of the cycle. This makes gas turbine engines suitable in increasing the overall efficiency and power output. The economics of power generation using gas turbine engines depends on the fuel cost, running efficiency, maintenance cost, and first cost, in that order [3]. The increase in efficiency and net specific work will be preferred over a reduction in turbine inlet temperature and compressor pressure ratio [4]. A comparison is made between single-spool engine and two-spool engine with free power turbine. Also, the performance of the engine with inter-stage turbine burner is investigated and compared with engine employing the nominal reheat concept. The engine employing interstage turbine burners produces superior improvements in both net work and efficiency over all other configurations. The effects of ignoring the variations on specific heat of gases and turbine cooling flow on engine performance are estimated. Ignoring the variation in specific heat can cause up to 30% difference in net specific work. The optimum locations of the intercooler and the reheat combustor are determined using the numerical model of the engine. The maximum net specific work is obtained if the reheat combustor is placed at 40% of the expansion section. On the other hand, to get maximum efficiency the reheat combustor has to be placed at nearly 10%-20% of the expansion section. The optimum location of the intercooler is almost at 50% of the compression section for both maximum net specific work and efficiency [5].It should be noted that a limit in TIT is imposed by the metallurgical constraints. Also, a practical limit is set by turbo machinery considerations on the compressor pressure ratio for fixed rotating speed. Many modifications on the simple gas turbine engine have been proposed in order to improve its performance within the allowable range of TIT and compressor pressure ratio. One example is the gas turbine engine employing intercooling, recuperation, and reheating [6, 7]. A regenerative gas turbine engine, with isothermal heat addition, working under the frame of Brayton cycle has been analyzed [8]. The temperature during the heat addition is kept constant using a converging duct, thus, it results in an increase in Mach number due to energy conservation. The results show that this engine when designed according to the maximum power density condition gives the best performance and exhibits highest cycle efficiencies. However, the limit on the exit Mach number from the isothermal combustor is a big challenge and restricts the improvement in performance. It was proposed that combustion to be continued purposely inside the turbine to increase the efficiency and specific power of the engine [9, 10]. Ground-based gas-turbine engines for power generation have been analyzed, with the results showing even better performance gains compared with conventional engines. The challenges arising for such modification was reviewed to assess them against the gains [11]. The mathematical model of the plant should be as accurate as possible to select the optimum design point. Some modelingerrors could direct towards a design that results in more total cost. This is particularly important for complicated configurations that are proposed to enhance the performance of the plant. In such complicated engines, there is significant change in specific heat from component to component and sometimes from inlet to outlet of the same component. The effect of the variations in specific heat should be considered at each state as function of temperature and fuel-to-air ratio especially when a parametric study is conducted to determine the optimum design point of the plant. A constant value of specific heat has been used for pure air (all components before the combustion chamber) and a higher value has been used for the gases [12]. A similar method has been followed; however, the specific heat is estimated as function of temperature using a seventh order polynomial [13]. Moreover, the cooling flow, which is necessary for the two turbines, not only adds complication to the power plant, but also changes its performance. The utilization of cooling air causes losses in the performance in both efficiency and net specific work. However, the cooling flow is normally ignored in the analysis [14], and that causes deviation in the optimum design point. In order to work at the highest TIT and get the realistic optimum performance, the effect of turbine cooling flow should be considered in the analysis. According to the author's knowledge, all these modifications have not been compared together using a full and accurate numerical model of the engine. Different modifications in the gas turbine engine have to be quantified at different TIT and compressor pressure ratio using full realistic model of the engine. This paper aims to quantify the improvements in performance due each possible modification on the gas turbine engine at different TIT and compressor pressure ratio within practical ranges using full realistic numerical model of the engine. Also, it assesses the effects of ignoring variations in specific heat and turbine cooling flow in the accuracy of engine modeling. This is particularly important to identify a realistic optimum operating point of the power plant. Also, it helps the designer to assess the percentage improvement due to any modification against the added complication and hence initial and maintenance costs. The paper starts with introducing a complete numerical model of the engine. The model takes into account the efficiencies of different components and pressure drop at intercooler, combustors and heat exchanger. Also, it accounts for the effectiveness of the intercooler and the heat exchanger. The specific heat is calculated at each stage and the turbine cooling flow is included in the analysis.A regenerative gas turbine engine, with isothermal heat addition, working under the frame of a Brayton cycle has been analyzed. With the purpose of having a more efficient small-sized gas turbine engine, the optimization has been carried out numerically using the maximum power (MP) and maximum power density (MPD) method [15].

1.1 The Regenerative Cycle

The regenerative cycle is becoming prominent in these days of tight fuel reserves and high fuel costs. The amount of fuel needed can be reduced by the use of a regenerator in which the hot turbine exhaust gas is used to preheat the air between the compressor and the combustion chamber. The regenerator increases the temperature of the air entering the burner, thus reducing the fuel-to-air ratio and increasing the thermal efficiency.

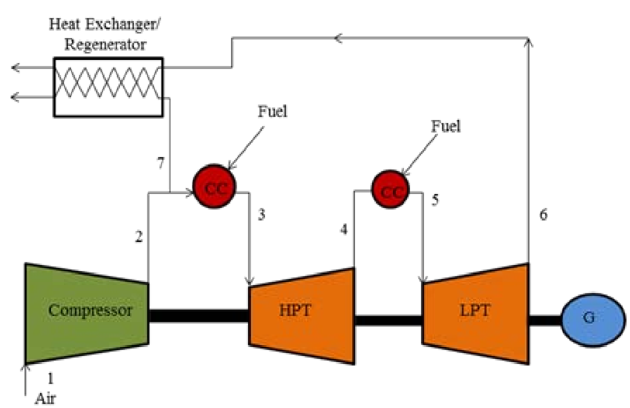

Fig.1 Schematic of combined regenerative and reheat cycle

1.2Reheat Cycle

In the reheat gas turbine cycle, there are two turbines used namely; high pressure turbine (HPT) and low pressure turbine (LPT). The gases first expand in the high pressure turbine and then in low pressure turbine. The temperature after expansion in the HPT becomes low and then further head added in the second combustor to increase the temperature before expansion in the LPT. Generally the temperature after expansion in the HPT has to be increases upto the same temperature of HPT inlet temperature. The pressure of expansion in the turbine has been takenas the root mean square value of the overall pressure for optimum results.

1.3Reheat and Regenerative Cycle

There are one compressor and one turbine used in regeneration gas turbine cycle. In the reheat or recuperation cycle, one compressor and two turbines is used in the present work. Fig 1 and fig 2 represents the combined effect of reheating and regeneration of a gas turbine cycle.

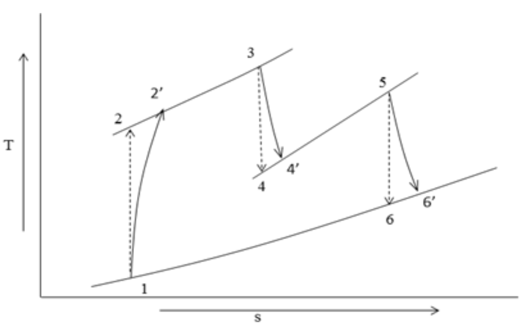

Fig. 2 T-s representation of combined regenerative and reheat cycle

2. MATERIALS AND METHODS

2.1Analysis of the Ideal Cycle

The two isentropic processes represent the compression (Compressor) and the expansion (Turbine Expander) processes in the gas turbine. A simplified application of the first law of thermodynamics to the air-standard Brayton cycle (assuming no changes in kinetic and potential energy) has the following relationships:

Work of compressor:

$$W_c = m_a(h_2 - h_1) \qquad (1)$$

Work of high pressure turbine:

$$W_{hpt} = (m_a + m_f)(h_3 - h_4) \qquad (2)$$

Work of low pressure turbine:

$$W_{lpt} = (m_g + m_{f2})(h_5 - h_6) \qquad (3)$$

Heat added to combustor 1 and combustor 2 given as follows:

$$Q_{2,3} = m_f \times LHV_{fuel} = (m_a + m_f)h_3 - m_a h_2 \qquad (4)$$

$$Q_{4,5} = m_f \times LHV_{fuel} = (m_a + m_f)h_5 - m_f h_4 \qquad (5)$$

Thus, the overall thermal cycle efficiency is:

$$\eta_{th} = W_{net}/(Q_{2'3} + Q_{4'5}) \qquad (6)$$

Actual compressor work and the actual turbine work are given by:

$$W_{ca} = \frac{m_a(h_{2'} - h_1)}{\eta_c} \qquad (7)$$

$$W_{ta} = (m_a + m_f)(h_3 - h_{4'})\eta_t \qquad (8)$$

The actual fuel required to raise the temperature from 2' to 3 is:

$$m_f = \frac{h_3 - h_{2'}}{(LHV)\eta_b} \qquad (9)$$

Thus, the overall adiabatic thermal cycle efficiency can be calculated by:

$$\eta_c = \frac{W_{act}}{(LHV)\eta_b} \qquad (10)$$

2.2Combustion chamber

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

$$Q_{add} = m_a(1+FAR)C_{pg}(T_3 - T_{2'}) \qquad (11)$$

Where, FAR is the mass fuel-air ratio and calculated by equation:

$$FAR = \frac{m'_{f,b}}{m'_a} = \frac{C_{pg}\cdot T_e - C_{pa}\cdot T_i}{\eta_b \cdot LCV_f - C_{pg}\cdot T_e} \qquad (12)$$

2.3 Turbine

The efficiency of gas turbine is taken care by considering polytropic efficiency of turbine.

$$W_t = \eta_t C_{pg}(T_3 - T_{4'}) \qquad (13)$$

The turbine power output is then $m_a(1+FAR)W_t$

The cycle thermal efficiency is the ratio of the network to the heat supplied to the heater:

$$\eta_t = \frac{W_{net}}{Q_{add}} = \frac{C_{pg}(T_3 - T_{4'}) - C_{pa}(T_{2'} - T_1)}{C_{pg}(TIT) - C_{pa}(T_{2'})} \qquad (14)$$

2.4 Regeneration Effect

The fuel requirement can be reduced by the use of a regenerator inwhich the hot turbine exhaust gas preheats the air between the compressorand the combustion chamber. In an ideal case theflow through the regenerator is at constant pressure. The regenerator effectiveness is given by the following relationship:

$$\eta_{reg} = \frac{T_3 - T_{2'}}{T_5 - T_2} \qquad (15)$$

Thus, the overall efficiency for this system's cycle can be written as:

$$\eta_{Rcyc} = \frac{(T_4 - T_5) - (T_2 - T_1)}{(T_4 - T_3)} \qquad (16)$$

Increasing the effectiveness of a regenerator calls for more heat transfer surface area, which increases the cost, the pressure drop, and the space requirements of the unit.

$$\eta_{rc} = \frac{T_{2'} - T_1}{T_2 - T_1} \qquad (17)$$

Power output is given by:

$$P = m_a \times W_{net} \qquad (18)$$

Air to fuel ratio is given by:

$$AFR = \frac{LCV_f}{Q_{add}} \qquad (19)$$

Specific fuel consumption:

$$SFR = \frac{3600}{AFR \cdot W_{net}} \qquad (20)$$

Fuel to air ratio is given by:

$$AFR = \frac{1}{AFR} \qquad (21)$$

Thermal efficiency is given by:

$$\eta_{th} = \frac{W_{net}}{Q_{add}} \qquad (22)$$

3. RESULTS AND DISCUSSION

This paper presents the results of the combined cycle in the form of graphs for various parameters. The software developed in MATLAB 10 for the thermodynamic calculations of the cycle and then graphs are plotted in the same software.

Table 1 shows the input values of the cycle for the analysis of the present work.

Table 1 Input data for the analysis

Input values for the calculation of the cycle		
S. No.	**Symbols**	**Values**
1	η_t	0.92
2	η_b	0.98
3	η_c	0.92
4	P_1	1.01325 bar
5	γ_g	1.34
6	C_v	42000 kJ/kgK
7	m_a	1 kg
8	η_{reg}	0.95
9	C_{pg}	1.127 kJ/kgK
10	γ_c	0.2857
11	C_{pa}	1.005kJ/kgK
12	TIT	1000-1500K
13	OPR	10-40
14	T_a	280-330 K

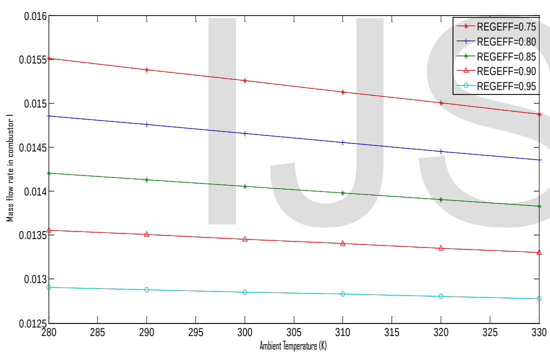

Fig. 3 Variation of mass flow rate in combustor 1 vs Ambient Temperature

Figure 3 represents the variation of rate of mass flow in combustor 1 with ambient temperature for different regenerative effectiveness. Mass flow rate is decreases at higher regenerative effectiveness and also decreases on increasing the ambient temperature for the same regenerative effectiveness. This is due to the fact that the temperature of the air supplied to the combustor has been increases on increasing the ambient temperature.

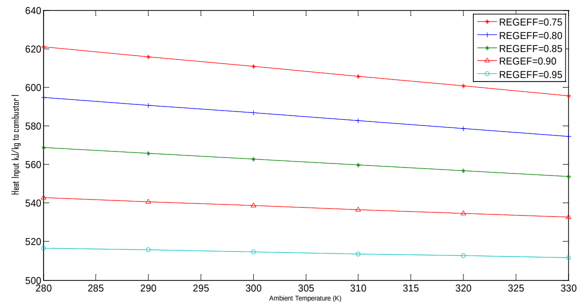

Fig.4 Variation of Heat supply in combustor 1 vs Ambient Temperature

Figure 4 exhibits the variation of heat supply in combustor 1 with ambient temperature for different regenerative effectiveness. Heat supply to combustor is decreases at on increasing the regenerative effectiveness. It has also been found that the heat supply decreases on increasing the ambient temperature for the same regenerative effectiveness. This is due to the fact that the temperature of the input air has been increased on increasing the ambient temperature.

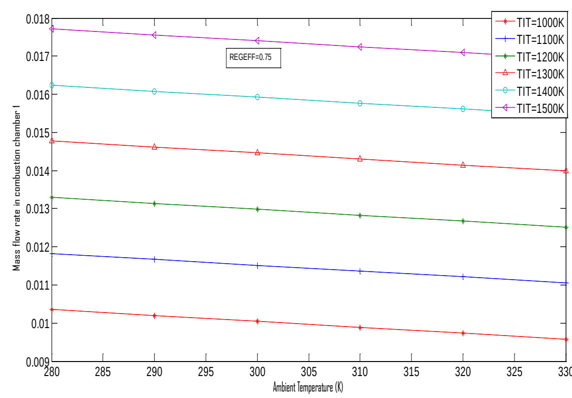

Fig.5 Variation of Mass flow rate in combustor 1 vs Ambient Temperature

Figure 5 shows the variation of mass flow rate in combustor 1 with different ambient temperatures. On increasing the ambient temperature, the mass flow rate decreases for a given value of TIT. On the other hand

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the mass flow rate increases on increasing the turbine inlet temperature for a given value of ambient temperature.

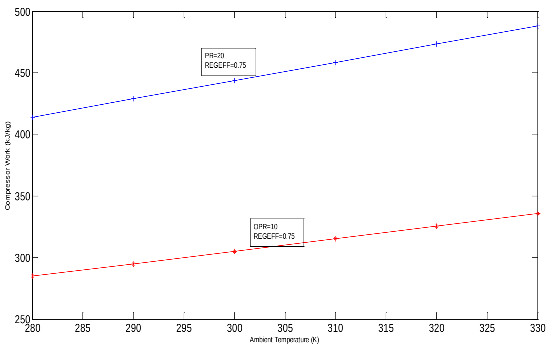

Fig. 6 Variation of Compressor Work vs Ambient Temperature

Figure 6 represents the variation of compressor work with ambient temperature for two different values of overall pressure ratios. At higher OPR, the compressor work is more than the one at lower OPR provided regenerative effectiveness does not change.

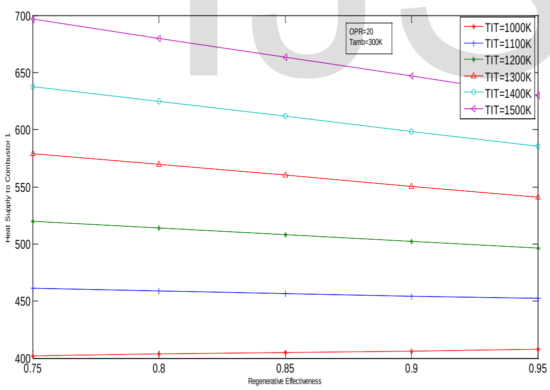

Fig. 7 Variation of Heat supply to combustor 1 vs Regenerative effectiveness

Figure 7 represents the variation of heat supply to combustor 1 with regenerative effectiveness for different values of TIT. The heat supply decreases on increasing the regenerative effectiveness for a given value of TIT. At higher TIT, the heat supply is more as compared to lower TIT.

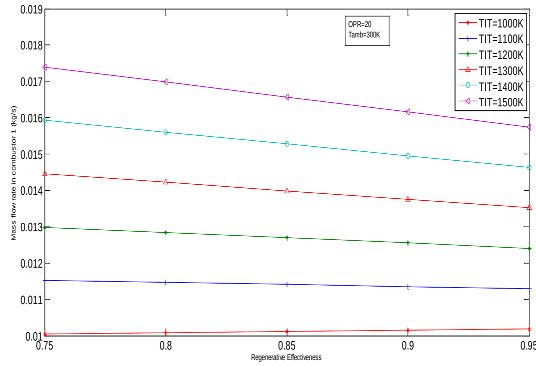

Fig. 8 Variation of Mass flow rate in combustor 1 vs Regenerative effectiveness

Figure 8 represents the variation of mass flow rate in combustor 1 with regenerative effectiveness for different TIT. The mass flow rate decreases on increasing the ambient temperature for a given value of TIT.

For a regenerator assumed to have an effectiveness varies from 75% to 95%, the efficiency of the regenerative cycle is higher than its counterpart in the simple cycle. The work output per kg of air is about the same or slightly less than that experienced with the simple cycle. Increasing the pressure ratio and the turbine firing temperature increases the Brayton cycle efficiency. The increase in the pressure ratio increases the overall efficiency at a given firing temperature; however, increasing the pressure ratio beyond a certain value at any given firing temperature can actually result in lowering the overall cycle efficiency. It should also be noted that the very high-pressure ratios tend to reduce the operating range of the turbine compressor. This causes the turbine compressor to be much more intolerant to dirt build up in the inlet air filter and on the compressor blades and creates large drops in cycle efficiency and performance. In some cases, it can lead to compressor surge, which in turn can lead to a flameout, or even serious damage and failure of the compressor blades and the radial and thrust bearings of the gas turbine.

4. CONCLUSION

From the preceding paragraphs the mass flow rate and the heat supplied decreases on increasing the ambient temperature for a given value of regenerative effectiveness. Though, a decrease in these parameters is less for higher value of regenerative effectiveness (0.95). Mass flow rate also decreases with increase in ambient temperature for a particular value of TIT. Work required by the compressor increases on increasing the ambient temperature for a given value of regenerative effectiveness though this increase in work is prominent for higher overall pressure ratio. The heat supplied to the combustor 1 decreases on increasing the regenerative effectiveness that reflects the higher efficiency of the engine for higher regenerative effectiveness. This decrease in heat supply remains constant for low turbine inlet temperature. The same result was observed for the mass flow rate.

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Nomenclature:-

1,2,3……. = State points
FAR = Mass fuel-air ratio
REGEFF = Regenerative Effectiveness
TIT = Turbine Inlet Temperature
OPR = Overall Pressure Ratio
AFR = Air fuel ratio
SFC = Specific fuel consumption
η_t = Turbine Efficiency
η_b = Burner Efficiency
η_c = Compressor Efficiency
γ_g = Specific heat ratio
m_a = Mass of air (kg)
T_a = Ambient temperature (K)
C_{pa} = Specific heat of air (kJ/kg-K)
C_{pg} = Specific heat of gas (kJ/kg-K)
p = Pressure (N/m^2)
W_c = Work of compressor (kJ/kg)
h = Enthalpy (kJ/kg)
W_{hpt} = Work of high pressure turbine (kJ/kg)
m_f = Mass of fuel (kg)
W_{lpt} =Work of low pressure turbine (kJ/kg)
m_g = Mass flow rate of gas (kg)
m_{f2} = Mass of fuel in combustor 2 (kg)
W_{cyc} =Total work output (kJ/kg)
$Q_{2,3}$ = Heat added to combustor 1

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LHV_{fuel} = Lower calorific vlue of fuel (kJ/kg)

$Q_{4,5}$ = Heat added to combustor 2

η_{th} = Overall thermal cycle efficiency

W_{net} = Net work output (kJ/kg)

W_{ca} = Actual compressor work (kJ/kg)

W_{ta} = Actual turbine work (kJ/kg)

Q_{add} = Rate of heat released

m'_{fb} = Mass of fuel in burner (kg)

T_e = Exit temperature of burner(K)

T_i = Inlet temperature of burner (K)

W_t = Turbine work (kJ)

η_{reg} = Regenerator effictiveness

η_{Rcyc} = overall efficiency

P = Power output (kJ)