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Steady State Thermal Analysis of Gas Turbine Power Plant Cycles at Higher Temperatures

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Increasing the cost of fuels, growing concern of global warming and increasing demand of electrical energy, a compact, less polluting and efficient gas turbine power plant with regenerator and a suitable Waste Heat Recovery Steam Generator (WHRSG) presents a very attractive concept. Developments in metallurgical sciences, protective coatings and cooling methods have provided the options to design the components and a higher Turbine Inlet Temperature (TIT) may be achievable in near future. A design and analysis methodology for thermodynamic evaluation of the gas turbine cycles has been developed, validated and applied for the analysis. The analysis reveals that a higher TIT (1900K) is possible to achieve higher performance (70% first law efficiency and 55% second law efficiency) with lower fuel consumptions and pressure ratio (20).

Keywords: Cogeneration, Turbine inlet Temperature, Specific fuel consumption, Power-to-heat ratio, Process heat, pinch point.

Introduction

In the present times of fast depleting conventional fuel resources, growing concerns for environment damage and a sharp-increasing demand for electricity and process heat, the need for more efficient and less polluting cogeneration systems with regenerator and a suitable waste heat recovery steam generator (WHRSG) presents a very attractive concept. Moran[1] has established an analysis methodology for the regenerative gas turbine cogeneration system. This simple system integrated with regenerator and a waste heat recovery steam generator (WHRSG) is attached to utilize the waste heat. Y. Tamarin[2] has worked on super alloy and ceramic coatings and Hiroshi Harada[3] have studied on the improved high temperature unique materials proposed in Japan and being evaluated. Khaliq and Kaushik[4] have evaluated the thermodynamic performance of combustion gas turbine cogeneration system with reheat and studied the efficiency of system with WHRSG. Doek *et al*.[5] worked out the exergy analysis of gas turbine cogeneration system and calculated performance of gas turbine cogeneration systems as well as the exergy destruction in each component in the system when it is operated at part and full load conditions. Butcher and Reddy[6] analysed the second law efficiency of waste heat recovery and studied the effect of pinch point temperature on the efficiency of HRSG. Korakianitis[7] performed analysis of combined cogeneration power plant with various power and efficiency enhancement techniques. A conceptual gas turbine based cogeneration cycle with compressor inlet air cooling and evaporative after-cooling of the discharge has been proposed by Khaliq and Chaudhary[8]. Yokoyana *et al*.[9] worked on optimal design of gas turbine cogeneration plants in consideration of discreetness of equipment capabilities. Dellenback[10] studied a reassessment of an alternative regenerative cycle that can produce efficiency higher than conventional only for limited set of conditions. Kumar and Krishna[11,12] described the characteristics of gas turbine power plant adopting the air-cooling at intake with alternative regenerative configuration and performed the second law analysis. Kumar and Kachhwaha[13] analysed a regenerative cogeneration plant for first & second law efficiencies. The recently developed cogeneration system features the introduction of CH_4 – O_2 combustion system and the 1700°C

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(1973 K) – class high – temperature gas turbine will be available in near future for industrial applications[14].

A parametric study has been performed for wide range of variables such as compressor air inlet temperature, pressure ratio, turbine inlet temperature, pinch point temperature etc. The effect of these input parameters on first and second law efficiencies, specific fuel consumption, and the process steam quantity of the cycles has been analysed.

Power Cycle (Cogeneration Systems) Description

A standard reheat cycle (RC) cogeneration system consists of a closed Brayton cycle with $CH_4 - O_2$ combustion and a HRSG utilizing the waste heat of the cycle. The cycle is comprised of an air compressor, regenerator, combustor and a turbine. Air after compression in the compressor enters the regenerator where its temperature is raised by the exhaust gases from turbine. After passing through regenerator, it enters into combustion chamber where its temperature is further raised up to TIT by the combustion of fuel. The gases then expand in the turbine and produce the work output (alternator to generate electricity). The heat carried by the exhaust gases is recovered in the regenerator and heat recovery steam generator (HRSG) to increase the temperature of air and generate process steam respectively. In simple cycle (SC) cogeneration system regenerator is absent.

Thermodynamic Formulation

The mathematical formulation of the present analysis is based on Moran[1] with the following assumptions:

a. The cogeneration system operates at steady state.
b. Ideal gas mixture principles apply for the air and the combustion products. The dry air composition is 77.48%N_2, 20.59%O_2, 0.03%CO_2 and 1.90%H_2O (g).
c. The fuel is a generic hydrocarbon of composition (C_aH_b). Fuel (natural gas) is taken as methane and modeled as an ideal gas. The fuel is provided to the combustion chamber at the required pressure by throttling from a high-pressure source.
d. The performance of turbine and compressor are characterized by constant polytrophic efficiencies.
e. The regenerator efficiencies based on gas temperatures is assumed constant.
f. The combustion in the combustion chamber is complete and N_2 is treated as inert.
g. Heat transfer from the combustion chamber is 2% of the fuel Lower Heating Value (LHV) of the fuel. All other components operate without heat loss.

A computer program was developed to simulate the cogeneration power system, in which the control volume analysis of each component was performed using mass, energy and exergy balances for determining the thermodynamic properties at every key position of the cycle.

Exergy balance equations for a particular component (k) is expressed as [1]

$$E_{q,k} - W_{cv,k} + \Sigma E_{i,k} - \Sigma E_{o,k} - \Sigma E_{D,k} = 0 \quad \ldots (1)$$

where E_D denotes the rate of exergy destruction and E_q denotes the associated exergy transfer rate due to heat transfer. If the effects of kinetic and potential energy are ignored, the total exergy rate E_k of a component consisting of physical exergy and chemical exergy.

The physical exergy and chemical exergy of air, fuel-to-air ratio (λ), fuel and water can be determined as given in reference[1]. The chemical exergy of the gas mixture is obtained by summing the overall composition of air, that includes N_2, O_2, H_2O (g) and other gases[1].

For complete combustion of methane the chemical equation is given by

$$\lambda\, CH_4 + [X'_{N2}N_2 + X'_{O2}O_2 + X'_{CO2}\,CO_2 + X'_{H2O}H_2O]$$
$$\rightarrow\ 1+\lambda]\ \ X''_{N2}N_2 + X''_{O2}O_2 + X''_{CO2}\,CO_2 + X''_{H2O}H_2O] \quad \ldots (2)$$

The components of the combustion products (flue gas) are analysed for mole fractions (X″) for nitrogen, oxygen, carbon and hydrogen[6].Air enters the compressor at ambient conditions. This initial temperature and density of air dictate the amount of work required for compression, the fuel that can be burnt and the fuel required achieving a specified turbine inlet temperature. As a result, the net power output, the efficiency, the exhaust gas flow rate and temperature at the turbine exit (consequently the recoverable heat) are functions of the ambient conditions. The turbine outlet conditions can be obtained which reflects the mixture entropy change in the turbine.

The compressor exit air is preheated in the regenerator with effectiveness (ε) before it enters the combustor. The energy balance over the regenerator is

$h_3 - h_2 = (1+\lambda)(h_5 - h_6)$ … (3)

HRSG utilizes the waste heat of the flue gas from regenerator to produce steam. Water enters at ambient temperature (T_8) and pressure p_{sat} and converted into saturated steam. The properties of water and steam are calculated with the help of equations[1].

Performance Parameters

The relevant parameters required for the combined first and second law analysis of gas turbine cogeneration system may be considered as follows

First – law Efficiency (η_I)

The ratio of all the useful energy extracted from the system (electricity and process heat in present case) to the energy of fuel input is known as first-law efficiency.

Second-law efficiency (η_{II})

According to the second law of thermodynamics, since exergy is more valuable than energy it is useful to consider both output and input in terms of exergy. The amount of exergy supplied in the product to the amount of exergy associated with the fuel is a relatively accurate measure of thermodynamic performance of a system.

Power-to-heat Ratio (R_{ph})

The cost effectiveness of any cogeneration system is directly related to the amount of power, it can produce for a given amount of process heat added. Hence, another parameter used to assess the thermodynamic performance of a cogeneration system is power-to-heat ratio (R_{ph}). In both the first law efficiency and power-to-heat ratio, power and heat are treated equal. This reflects that first law of thermodynamics is only concerned with quantity, not energy quality. Thus η_I and R_{PH} are also known as first law efficiencies.

Specific Fuel Consumption

Specific fuel consumption (sfc) is defined as the ratio of the mass flow rate of fuel of a gas turbine engine to its output power (kW), in specified units.

Pinch Point Temperature (T_{PP})

HRSG's are popular today because of the efficiency gains that are accomplished with the re-use of waste heat from turbine exhaust. Pinch point is used in sizing the heat transfer surface area of the HRSG and defined as the difference between the saturation temperature and the HRSG exit temperature. It is desirable to make the pinch point as small as possible without significant increase in the cost of the HRSG.

Waste Heat (Exhaust Temperature)

The enthalpy of the exhaust gases from HRSG to atmosphere can be calculated with the help of equation[1].

Parametric Study

The important operating parameters in the present cogeneration system are compressor pressure ratio (r_p), turbine inlet temperature (TIT), ambient temperature and relative humidity (φ), pinch point temperature (T_{pp}) and saturation steam pressure (p_s). The effect of variation of these parameters on first law efficiency (η_I) and power-to-heat ratio (R_{PH}) is obtained by the energy balance approach or the first law analysis of the cycle. However, the exergy destruction or thermodynamic losses of each component and second law efficiency (η_{II}) of the system has also been investigated under the exergy balance approach or second-law analysis. To study the effect of these operating variables on the performance parameter of the system, operating under different conditions, the following common characteristics and their corresponding ranges are discussed below:

i. The compressor pressure ratio (r_p) has been varied for a range of 5 to 50 [1].
i. Due to metallurgical advancements, the value of Turbine Inlet Temperature (TIT) is continuously increasing. In the present simulation work, the range selected for TIT is 1300 to 1900 K, which covers the existing, as well as advanced gas turbine to be developed in near future. The maximum value of 1900 K is based on the reference[14], which describes the conceptual design and the development of the blade for a 1700° C (1973 K) class high temperature gas turbine in a closed cycle power generation system.
i. Ambient temperature and pressure of air is taken as 298.15 K and 1.013 bar respectively[1].
i. The system component efficiencies[1] are given below
i. Compressor isentropic efficiency $\eta_c = 87\%$.
i. The turbine isentropic efficiency $\eta_r = 89\%$.
i. Efficiency of combustion chamber and reheater (η_{CC}) = 95%.
i. Generator efficiency (η_g) = 97%.

i. The pinch point temperature (T_{pp}) at HRSG is taken as 25°C and varied for a range of 0 to 50°C. The temperature of condensate return from process is taken 25°C (open circuit). Process steam pressure is taken between 10 - 20 bars.
i. Methane (CH_4) gas is taken as fuel having the lower heating value of 802361 kJ/Kmol and molecular weight of methane is equal to 16.043 Kg/Kmol.
i. System Component Losses

Air side pressure drop in regenerator of total inlet pressure = 2%

Gas side pressure loss in regenerator = 2%

Pressure loss in combustion chamber = 4%

Gas side pressure loss in HRSG = 5%

Effect of Pressure Ratio

The variation of first law efficiency and second law efficiency with pressure ratio for regeneration cycle (RC) and simple cycle (SC) at TIT 1900K is shown Fig. 1. For RC, initially the first and second law efficiencies increase with increasing pressure ratio and after reaching a maximum value it starts decreasing with further increase in pressure ratio, whereas for SC efficiencies increase continuously with increase in pressure ratio. The efficiencies are higher for the regeneration cycle as the exhaust from the turbine is utilized to heat the compressed air coming out from compressor exit. The difference between the efficiencies of RC and SC decreases with increase in pressure ratio. The first law efficiency decreases with pressure ratio after reaching a maximum value is due to the fact that at this stage, the exit air temperature of the compressor is higher than the exhaust temperature of the turbine.

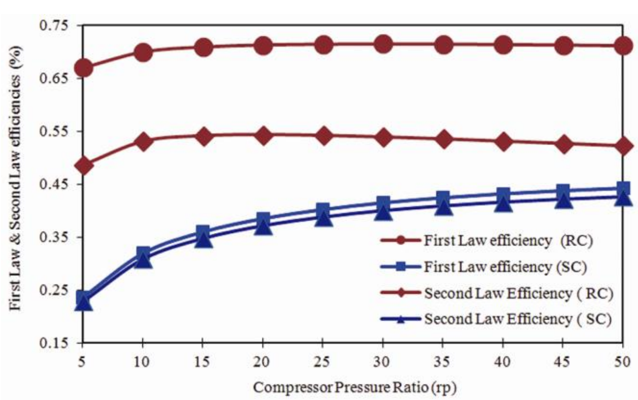

Fig. 1—Effect of Compressor Pressure Ratio (r_p) on First law and Second Law Efficiency of regeneration cycle (RC) and simple cycle (SC) [TIT=1900 K, T_0=298 K]

Thus, there is a reverse heat transfer from hot air to exhaust gases. The cooling of air in regenerator causes decreasing efficiency. From the analysis, it can be concluded that for TIT = 1900 K the optimum pressure ratio (r_p) is 20 and corresponding first law and second law efficiencies (η_I) are 71.2% and 54.3% respectively. For pressure ratio (r_p) value 20 and above the first law efficiency (η_I) is always more than 70%.Beyond the optimum pressure ratio (r_p) the second law efficiency (η_{II}) decreases with pressure ratio (r_p) relatively faster as compared to first law efficiency (η_I). Since the quality of fuel (i.e., the exergy associated with the heat addition) is more than the heating value or energy of the fuel because the exergy of fuel would increase while bringing it from the ambient pressure to combustion pressure at ambient temperature. Hence, exergy associated with the heat addition will be equal to exergy associated with the heating value of fuel plus exergy increase, i.e., mechanical exergy due to increase of pressure of fuel from the ambient to combustion state. Therefore, the second-law efficiency of the cycle is lower than the first-law efficiency. First and second law efficiencies increase with TIT for a corresponding value of pressure ratio. Increase in pressure ratio also increases the turbine work and the net work output of the cycle. Hence, the gas temperature at the exit of regenerator increases. Consequently, the energy available to generate the process heat increases. Hence, as pressure ratio (r_p) increases above 15, the power-to-heat ratio decreases, because at much higher-pressure ratio the process heat increases significantly.

Effect of Turbine Inlet Temperature

The variation in first law and second law efficiencies, power-to-heat ratio and specific fuel consumption with the change in turbine inlet temperature (TIT) for regeneration cycle and simple cycle are shown in Figures 2 and 3, respectively. Increase in TIT enhances the heat transfer rate by absorbing greater heat from turbine exhaust in the regenerator, and hence the temperature of air inlet to the combustion chamber would increase, which in turn would increase the mean temperature of heat addition that leads to reduction of the magnitude of heat addition in the cycle. Therefore, cycle efficiencies of RC and SC increase with an increase in TIT. However, for RC the power-to-heat ratio initially increases

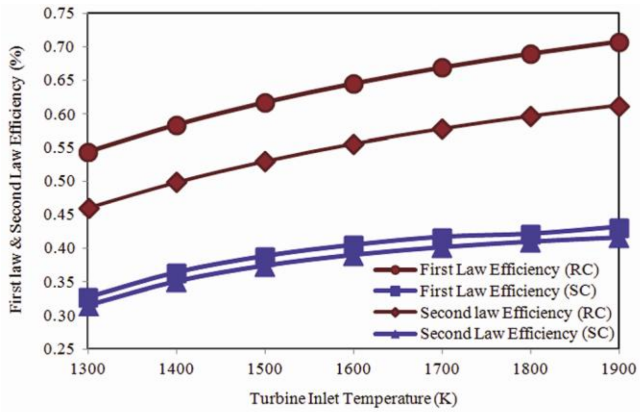

Fig. 2—Effect of Turbine Inlet Temperature (K) on First Law efficiency, Second Law Efficiency of Regeneration cycle (RC) at r_p = 20 and Simple Cycle (SC) at r_p = 40

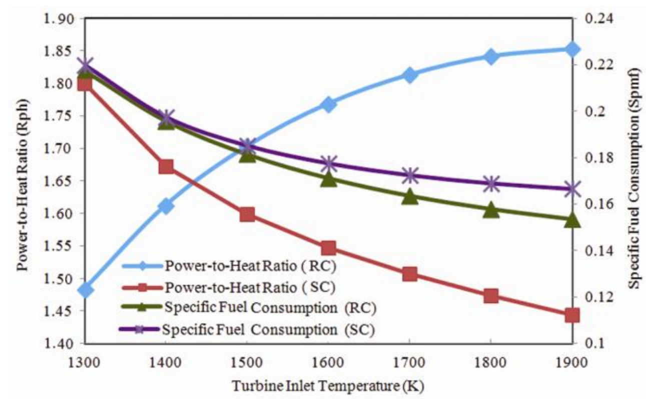

Fig. 3—Effect of Turbine Inlet Temperature (K) on Power-to-heat Ratio (R_{ph}) and Specific Fuel Consumption (sfc) of Regeneration cycle (RC) and Simple Cycle (SC)

Table 1—Comparisons of performance parameters at TIT 1700 K and 1900 K

Nature of power cycle	First Law Efficiency (%)		Second Law Efficiency (%)		Power – to-heat Ratio		Specific Fuel Consumption (Kg/kwh)	
	TIT 1700 K	TIT 1900 K	TIT 1700 K	TIT 1900 K	TIT 1700 K	TIT 1900 K	TIT 1700 K	TIT 1900 K
Cogeneration cycle with regenerator (RC)	66.9	70.33	57.82	61.3	1.81	1.85	0.1635	0.1534
Simple cycle (SC)	41.7	43.2	40.2	41.6	1.5	1.44	0.1725	0.1666

appreciably with increase in TIT because the increase in generation of process heat is relatively lower than the electric power output. At higher TIT, the rate of increase of power –to-heat ratio is low. In case of SC, power to heat ratio decreases due to significant increase in turbine exhaust temperature. The use of regenerator reduces specific fuel consumption which increases power-to-heat ratio (R_{ph}) of the regenerative cycle.

Conclusions

The analysis reveals that a higher TIT up to 1900 K is possible to achieve higher performance of power cycles with lower fuel consumptions. The output performances of the cycles are shown in Table 1. Considering the recent developments in metallurgy, applying advances blade cooling methods, a higher TIT up to 1900 K may be achievable in near future and consequently higher efficiencies (70% first law efficiency and 55 % second law efficiency at pressure ratio 20) can be easily achieved. The reduction in inlet temperature of air significantly improves the second law efficiency of the system. The analysis reveals that the components of the cycles are satisfied with the varying input parameters. The studies may be continued by exploring the metallurgical developments. The present study can be further extended for exergy analysis and techno-economic analysis. The design methodology and the program developed for the present analysis can be utilized for the analysis of the power cycles for several varying parameters with the biofuels and further other developments.

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