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Parametric Analysis of Gas Turbine Cogeneration cycle for Various Blade Cooling Means

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Abstract

The present work deals with the first law thermodynamic analysis of cogeneration cycle employing different means of turbine blade cooling for various configurations. Based on modeling and governing equations a computer program has been constructed in C++ language, to do the first law thermodynamic analysis of eight configurations of cogeneration cycle. With the help of input data, results have been plotted and analyzed in terms of dependent and independent parameters.

Key words: Air film cooling, heat recovery steam generator (HRSG). Turbine inlet temperature (TIT), Specific fuel consumption (SFC)

1. Introduction

Energy is one of the major inputs for the economic development of any country and is vital to the sustenance of a modern economy. In the case of the developing countries, the energy sector assumes a critical importance in view of the ever increasing energy needs requiring huge investments to meet them. Co-generation refers to the simultaneous generation of power and heat. Thermal power plants in general, do not convert all of their available energy into electricity. In most power plants, a bit more than half is wasted as excess heat. By capturing the excess heat in the form of steam generated from the exhaust of gas turbine in heat recovery steam generator, Co-generation plants uses heat that would be wasted in a conventional power plant, potentially reaching an efficiency of up to 89%, compared with 55% for the best conventional plants. This means that less fuel needs to be consumed to produce the same amount of useful energy. The system is efficient and the cost of power production per kW is less. Also, less pollution is produced for a given economic benefit. A large number of configurations of cogeneration cycle plants are possible. Schematic and energy flow diagram of cogeneration cycle plant is represented in figure 1.1.

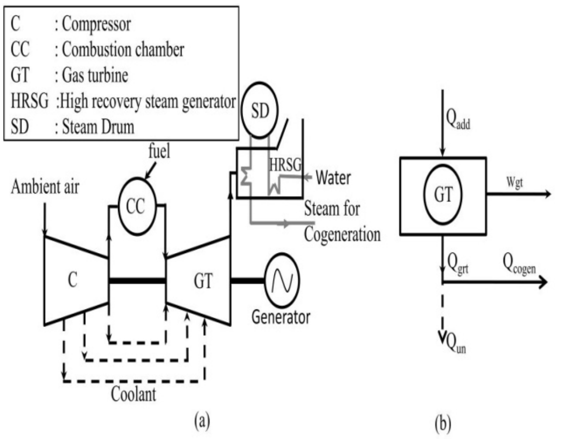

Figure 1.1(a) Schematic diagram of a cogeneration cycle plant (b) Energy flow diagram of cogeneration cycle power plant

Various researchers have contributed a lot in the field of Gas turbine, cogeneration plant and cooling means of the turbine. Shukla and Singh [1] investigated the combined effect of inlet evaporative cooling (IEC), steam injection (SI) and film cooling (FC) on the power augmentation of simple gas turbine cycle. Ahmed et al. [2] presents a proposal of a combined scheme for applying a vapour compression refrigeration system to cool the condenser of a steam plant. In this scheme the refrigerant (mostly liquid) leaving the throttle valve of the refrigeration system is circulated through the pipes of the shell and tube condenser, while the steam exhausting the plant turbine flows in the shell around the outer surfaces of the tubes. Mohamed et al. [3] experienced about the ambient temperatures rising during hot seasons have significant losses and impacts on both output power and efficiency of the gas turbine. When the ambient temperature increases, the air mass flow rate decreases, and hence leads to reduce the gas turbine produced power. Ambient air can be cooled by using either evaporative cooler or absorption chiller. Sanjay et al. [4] compared the thermodynamic performance of MS9001 gas turbine based cogeneration cycle having a two-pressure heat recovery steam generator (HRSG) for different blade cooling means. Dong et al. [5] have used the Conjugate calculation methodology to simulate the C3X gas turbine vanes cooled with leading edge films of “shower-head” type. Ting W. and Xianchang Li [6] successfully used Air film cooling to cool gas turbine hot sections for the last half century. A promising technology is proposed to enhance air film cooling with water mist injection. They obtained the adiabatic wall film cooling effectiveness and the heat transfer over a film -cooled surface that is made inclined at various angles with respect to a highly turbulent flow. Zhihong G. et al. [7] have measured the film cooling effectiveness on the surface of a high pressure turbine blade using the pressure sensitive paint (PSP) technique. Four rows of axial laid-back, fan-shaped cooling holes are distributed on the pressure side while two such rows are provided on the suction side. Krishnan V. et al [8] established an analytical study of low temperature hot corrosion (LTHC) in the context of high temperature turbines using coal gas or syngas with trace amount of sulphur in the fuel. Sanjay et al [9] have presented a comparative study of the influence of different means of turbine

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blade cooling on the thermodynamic performance of combined cycle power plant. Seven schemes involving air and steam as coolants under open and closed loop cooling techniques have been studied. Sanjay et al. [10] have presented parametric energy and exergy analysis of reheat gas–steam combined cycle using closed-loop-steam-cooling. Of the blade cooling techniques closed-loop-steam-cooling has been found to be superior to air-film cooling. The literature review on cogeneration cycle based power plants exhibits that an extensive research is needed in this direction to enhance the performance of these power plants. The present work deals with analysis of cogeneration cycles employing different means of turbine blade cooling for various configurations.

The Paper is organized as following the introduction, Section 2 describe the modelling of various configurations of cogeneration cycle plants by governing equations. Section 3 illustrates the results obtained from modelling of the various design parameters and also the results are analyzed and discussed. Finally conclusions are drawn in Section 4.

2. Mathematical modeling of cooling blades of gas turbine

Unlike steam turbine blading, gas turbine blading need cooling. The objective of the blade cooling is to keep the metal temperature at the safe level, to ensure, a long creep life, low oxidation rates, and low thermal stresses. The universal method of blade cooling is by air bled from compressor or by other cooling fluid flowing through the internal pressure in the blades. The cooling techniques may be internal convection, film or transpiration cooling. The cooling medium may be air or steam. The cooling loop may be open or closed. Thus cooling means for gas turbine blading may be categorized mainly in two categories. Open loop cooling is further subdivided as Air internal convection cooling (AICC), Air film cooling (AFC), Air transpiration cooling (ATC), Steam internal convection cooling (SICC), Steam film cooling (SFC), Steam transpiration cooling (STC). Open and closed loop cooling of gas turbine is shown in Figure 2.1. In open loop cooling, the cooling fluid mixes with the working fluid (i.e. the combustion products), while in closed loop, the cooling fluid does not mix with working fluid. The different cooling techniques will require different amounts of coolant flows.

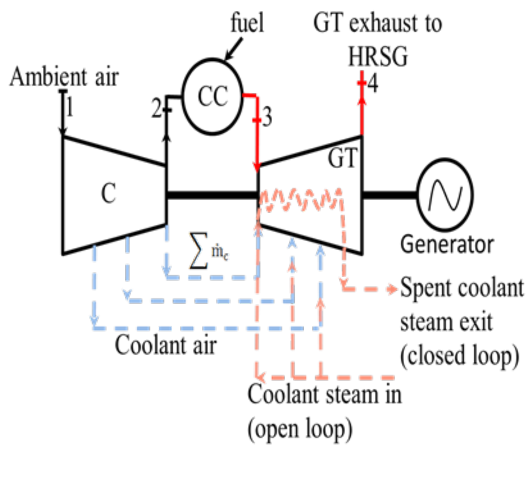

Figure 2.1 Air cooled gas turbine with open & closed loop blade cooling

Sources of losses in open loop cooling are much more than that is in closed loop cooling. In open loop cooling, the mixing of the spent cooling fluid with the main gas stream causes total pressure loss and reduction in gas enthalpy. There is a loss in turbine output due to reduction of mass flow rate of gas entering to the turbine due to bled air from compressor. Further, there is a cooling loss, which results in lower gas turbine exhaust temperature and also pumping loss, while closed loop cooling only suffers from cooling loss and there is no pressure loss due to mixing of two streams.

In the present study open loop air cooling techniques are used as cooling medium. Several approaches for the determination of open loop coolant flow are given in the literature such as Louis et al [11], Harlock et al. [12], El-Masri [13], etc. the model used for cooled turbine is the refined version of Louis et al. [11] model and Harlock at al. [12] depicted in Figures 2.2, 2.3, 2.4. The following assumptions are made for the development of model:

I. Gas turbine blades cooled by internal convection are treated as heat exchangers operating at constant temperature and the coolant exit temperature is expressed as a function of heat exchanger effectiveness.

II. A concept of isothermal effectiveness (η_{iso}) is introduced for film or transpiration cooling to account the reduced heat transfer rate from hot gas to blades. The isothermal effectiveness due to transpiration cooling $(\eta_{iso,\ trans})$ is higher than that due to film cooling. For internal convection cooling $\eta_{iso} = 0$.

III. A factor, $F_{sa} = 1.05$ is used to convert pitch line blade surface area to actual blade surface area.

Open loop cooling:

(i) Internal Convection Cooling, (ICC): A simple model for internal convection cooling of blades is shown in Figure 2.2. For an internally cooled turbine configuration, the ration of coolant to main gas flow $(\dot{m}_c/\dot{m}_g)$ is proportional to the difference of enthalpy, which drives the heat transfer to the blades to the ability of the coolant to absorb heat, which is also termed as cooling factor R_c. Thus,

$$\frac{\dot{m}_c}{\dot{m}_g} = \frac{\text{Heat transfer to blades}}{\text{Ability of coolant to absorb heat}} \propto \frac{h_{g,i}-h_b}{h_{c,e}-h_{c,i}} \propto R_c \quad (2.1)$$

The concept of heat exchanger effectiveness (ε) is introduced to account the exit temperature of coolant

$$\varepsilon = \frac{T_{c,e}-T_{c,i}}{T_b-T_{c,i}} \quad (2.2)$$

Thus, the cooling factor 'R_c' is expressed as

$$R_c = \frac{(T_{g,i}-T_b).c_{p,g}}{\varepsilon.(T_b-T_{c,i}).c_{p,c}} \quad (2.3)$$

From the equation (2.1, 2.2, 2.3) we conclude that cooling factor and cooling requirement are inversely proportional to the blade heat exchanger effectiveness. A simple heat balance for a typically internally convective cooled blade row is given by

$$Q_{net}=\dot{m}_c.c_{p,c}.(T_{c,e}-T_{c,i})=\dot{m}_g.c_{p,g}.(T_{g,i}-T_{g,e})=\bar{h}_g.A_{s,g}.(T_{g,i}-T_b) \quad (2.4)$$

Using effectiveness from equation (2.2), we get

$$Q_{net}=\dot{m}_c.c_{p,c}.\varepsilon(T_b-T_{c,i})=\dot{m}_g.c_{p,g}.(T_{g,i}-T_{g,e})=\bar{h}_g.A_{s,g}.(T_{g,i}-T_b) \quad (2.5)$$

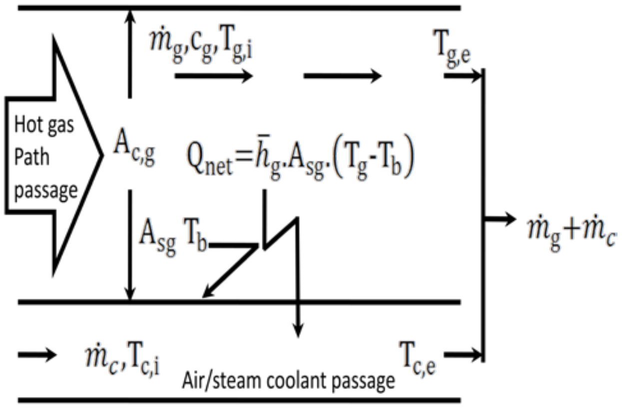

Figure 2.2 Model for open loop internal convection turbine blade cooling

There exists reasonable a constant ratio relationship between the exposed area of heat transfer ($A_{s,g}$) to the cross-section area of the main hot gas flow (A_g) for a set of similar gas turbine, i.e.

$$A_{s,g}=\lambda.A_g=(\lambda.\dot{m}_g)/(\rho_g.C_g)$$

Thus, by replacing $A_{s,g}$ equation (2.5) becomes

$$\dot{m}_c.c_p.\varepsilon(T_b-T_{c,i})=\lambda\left[\bar{h}_g/(\rho_g.C_g)\right].\dot{m}_g.c_{p,g}.(T_{g,i}-T_b) \quad (2.6)$$

After rearranging, the equation (2.6) becomes

$$\frac{m_c}{m_g}=\lambda.\left[\frac{c_{p,g}}{c_{p,c}}\right].\left[\frac{\bar{h}_g}{c_{p,g}.\rho_g.C_g}\right].\left[\frac{(T_{g,i}-T_b)}{\varepsilon.(T_b-T_{c,i})}\right]=\lambda.\left[\frac{C_{p,g}}{C_{p,c}}\right].\bar{S}t_{in}.\left[\frac{(T_{g,i}-T_b)}{\varepsilon.(T_b-T_{c,i})}\right] \quad (2.7)$$

Where $\bar{S}t_{in}=\frac{\bar{h}_g}{c_{p,g}.\rho_g.C_g}$ is the mean Stanton number based on the condition at cascade inlet. For a row in which the blade length is 'H', the blade chord is 'c' and the blade pitch is 't', the blade perimeter is 'S_g' the flow discharge angle is 'α', the ratio 'λ' is given by

$$\lambda=\frac{A_{sg}}{A_g}=\frac{2H.c}{H.t\cos\alpha}=\frac{2c}{t\cos\alpha}=\frac{S_g.F_{sa}}{t\cos\alpha} \quad (2.8)$$

Where, F_{sa} is correction factor to account for actual blade surface area. From equations (2.6), (2.7) and (2.8), we have

$$\frac{\dot{m}_c}{\dot{m}_g}=\bar{S}t_m.\left(\frac{S_g}{t\cos\alpha}\right).F_{sa}.\left(\frac{c_{p,g}}{c_{p,c}}\right).\left(\frac{T_{g,i}-T_b}{\varepsilon.(T_b-T_{c,i})}\right) \quad (2.9)$$

or

$$\frac{\dot{m}_c}{\dot{m}_g}=[\bar{S}t_m].\left[\frac{S_g}{t\cos\alpha}.F_{sa}\right].[R_c] \quad (2.10)$$

Equation (2.10) shows that the cooling requirement in a blade row depends upon average Stanton number ($\bar{S}_t$), turbine blade geometry $\left(\frac{S_g.\ F_{sa}}{t\cos\alpha}\right)$ And cooling factor (R_c).

In general $\bar{S}t_m=0.005$, $\frac{S_g}{t\cos\alpha}=3.0$ and if $F_{sa}=1.04$, so equation (2.10) takes the form as

$$\frac{\dot{m}_c}{\dot{m}_g}=0.0156R_c \quad (2.11)$$

Equations (2.10) and (2.11) form the basic to calculate the cooling requirements for all types of cooling means and only the expressions and values of R_c will change. In the present work internal convection cooling include air and steam as the cooling medium. So in calculating the cooling requirement of air and steam, the value of specific heat of coolant ($c_{p,c}$) will be taken accordingly in the expression of R_c equation (2.3) i.e. for air $c_{p,c}$ will be $c_{p,a}$ and for steam $c_{p,c}$ will be $c_{p,s}$.

(i) Film cooling (FC): in film cooling, the coolant is injected from the leading edge of the blade and forms a film over the blades which reduce the heat transfer from gas to blades. A simple model for film cooling is shown in Fig. 2.3. A concept of isothermal effectiveness for film cooling $(\eta_{iso})_{film}$ is introduces and the resulting cooling factor is expressed by

$$(R_c)_{film} = \frac{(T_{g,i}-T_b).\, c_{p,g}.(1-\eta_{iso})_{film}}{\varepsilon.\,(T_b-T_{c,i}).\, c_{p,c}} \qquad (2.12)$$

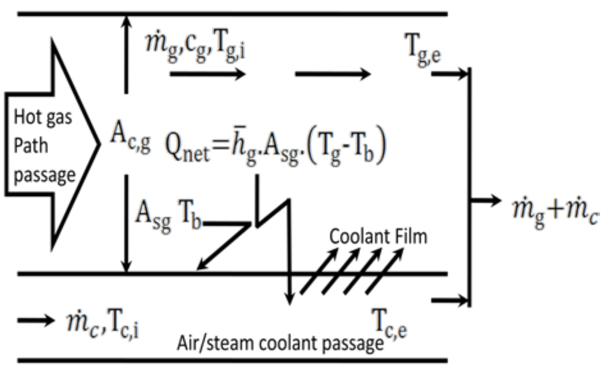

Figure 2.3 Model of open loop film cooling turbine blade

The value of $(\eta_{iso})_{film}$ is taken as 0.4. Thus the cooling requirement for film cooling is expressed as

$$\frac{\dot{m}_c}{\dot{m}_g} = \bar{S}t_i.\left[\frac{S_g}{t\cos\alpha}.\ F_{sa}\right].\,[R_c]_{film} = 0.0156[R_c]_{film} \qquad (2.13)$$

For air and steam film cooling, the value of $c_{p,c}$ in equation (2.12) will be taken according to the coolant used.

(iii)

Transpiration Cooling (TC): In transpiration cooling, coolant passes through numerous, very small channels/holes in the blade wall i.e., the wall is made up of porous material. A simple model for transpiration cooling is shown in Fig 2.4.

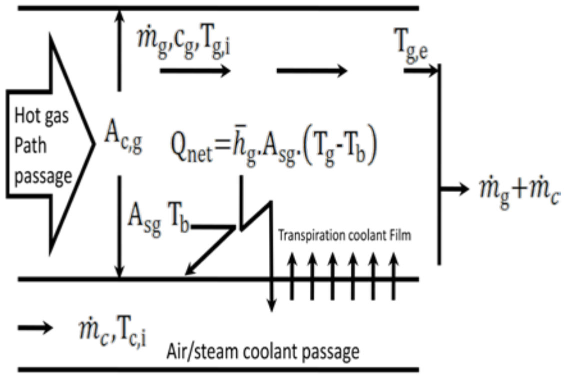

Figure 2.4 Model for transpiration cooling of turbine blade

The heat transfer co-efficient within the wall channels is so high that the coolant achieves the wall temperature before emerging from the wall. It assumed that the coolant flow velocity is uniformly distributed and normal to the wall. A new thick transpired boundary layer is formed and serves to provide thermal protection for the blade from hot gas. A concept of isothermal effectiveness for transpiration air-cooling $(\eta_{iso})_{trans}$ is introduced whose value is taken as 0.5 and the resulting cooling factor is given by

$$(R_c)_{trans}=\frac{(T_{g,i}-T_b).\ c_{p,g}.\ \left(1-(\eta_{iso})_{trans}\right)}{\varepsilon(T_b-T_{c,i}).\ c_{p,c}} \qquad (2.14)$$

For air and steam transpiration cooling the value of $c_{p,c}$ will be taken according to the coolant used. Thus the coolant requirement for transpiration cooling is requirement for transpiration cooling is

$$\frac{\dot{m}_c}{\dot{m}_g}=\bar{S}t_{in}.\ \left[\frac{S_g}{t\ \cos\alpha}.\ F_{sa}\right].\ [R_c]_{trans}=0.0156[R_c]_{trans} \qquad (2.15)$$

3. Results and Discussion

With the help of assumptions and input data listed in subsequent articles results have been obtained and plotted using origin graphic package for dependent versus independent variables. Eight selected configurations for performance have been studied by constructing a computer code “NabhCogen” in C++ language on the basis of modeling and governing equations. For the systematic study of performance of configurations this chapter is divided in to several sections and sub-section combining the configurations into groups. The effects of cooling means on the performance have been studied for selected configurations. In the study of cogeneration plants the required steam pressure and temperature varies widely according to use (customer requirements).

3.1 Assumptions and Input Parameters

Based on above modeling and governing equations, the parametric analysis has been carried out by using the input data which conform to the current state-of-art technology with common practical data of various manufacturers and research papers. For basic cycle configuration, the range of compressor pressure ratio selected varies from 12 to 30. From the available literature [4] it is desired to go for higher TIT with the advancement of blade material technology available, so the range of TIT varies from 1400K to 2000K while TIT for regeneration cycle is taken as 2200K with allowable blade temperature of 1122K. The selection of inlet steam temperature is governed by the turbine exhaust temperature and selected approach temperature. The maximum steam temperature is limited to 570^{o}C due to metallurgical conditions in HRSG as per current state-of-art technology. It is essential that for better utilization of waste heat stack temperature should be as low as possible. But for controlling the maintenance and life of stack the selection of stack temperature is limited by dew point temperature. In this study stack temperature is taken as 80^{o}C.The assumption and input parameters are tabulated in 4.1.

Table 4.1 Input Data used for Analysis

Parameter	Symbol	Unit
GasProperties	Cp=f(T)	KJ/KgK
	Enthalpy h=fc$_p$(T)dT	KJ/Kg
Compressor	Polytropic efficiency ($_{pc}$)= 92.0	%
	Mechanical efficiency (m)=98.0	%
	Air inlet temperature =288	K
	Inlet plenum loss=0.5% of entry pr.	bar
Gas turbine	Polytropic effifciency($_{pt}$)=92.0	%
	Exhaust pressure= 1.08	bar
	Turbine Blade Temprature=1122	K
	Intercooler pressure Loss=2.0	bar
	Intercooler effectiveness=92	%
	Recuperator pressure Loss=2.0	bar
	Recuperator effectiveness=70	%
HRSG	Steam pressure=70	bar
	Steam temperature 843	K
	Steam reheat temperature=843(Max)	K
Alternator	Alternator efficiency=98.5	%

4.2 Gas Properties Evaluation as a Function of Temperature.

Unlike steam tables, ready-made gas properties table are not available for finding out specific heat of air combustion products as a function of temperature and pressure. In real situations the specifics heats of air and products of combustion are a function of temperature at moderate pressure. Further, the percentage of excess air in combustion products is a function of A/F ratio, which is governed by compression pressure ratio and turbine inlet temperature.

4.3 Effect of different cooling means for various TIT and r_p on plant specific work, plant efficiency, specific plant specific fuel consumption and m_c/m_g for BGT configuration

- **Variation on plant efficiency and plant specific work**

Figure 4.1 to 4.7 are plotted for different cooling means. From the graphs it is observed that for AICC, AFC, ATC, SICC, SFC and STC plant efficiency increases with increase in TIT and r_p. This is because of the fact that increment in TIT increases the turbine work meanwhile compressor work remains same for any value of TIT which results in the increment in plant specific work. As the higher value of TIT demands slight increase in plant specific fuel consumption the value of plant efficiency increases. For higher value of r_p the outlet temperature of the compressed air increases which in turn lowers the amount of plant specific fuel required for generation of same value of TIT. Although the slight increase in compressor work is observed for higher r_p but the high value of r_p also increases the turbine work which in turn compensate the loss. Owing to these facts the plant efficiency increases.

It is observed from the results that in all the cases of cooling means, at any TIT the plant efficiency increase with r_p. At any TIT except above r_p =27, the plant specific work first increases and then decreases with increase in r_p. There exist an optimum r_p at any TIT with reference to plant specific work and this is also a function of cooling means. The value of optimum $r_{p,}$ at any TIT with reference to the maximum plant efficiency is due to the combined effect of many factors. With increasing r_p and TIT, the compressor work input, the plant specific fuel consumption and coolant air requirements increase, the gas cycle work increases but restricted by the increasing pumping, cooling and mixing losses.

- **Variation of plant specific work for varying compressor pressure ratio**

Figure 4.8 depicts the variation of plant specific work with compressor pressure ratio, for different means of cooling for BGT configuration at TIT=1700 K and T_b=1122 K. It is observed that there is significant increase in plant specific work with change in compressor pressure ratio for all cooling means. This is attributed to the fact that the increasing compressor work input is not able to offset the advantage gained by increasing turbine output and lesser plant specific fuel consumption requirement, while the coolant requirement are more or less unaffected with increasing r_p upto r_p=30. The highest and lowest plant specific works are exhibited by SICC and AICC cooling means respectively. This is because of the fact that in open loop steam cooling, SICC, the gas turbine acts as steam injected gas turbine in gas turbine itself which expands and yields additional work due to higher steam coolant required as compared to STC and SFC.

- **Variation of plant efficiency for varying compressor pressure ratio**

The effect of rp on plant efficiency is depicted in Fig. 5.9 for various means of cooling at TIT=1700 K and Tb=1122 K for BGT system. It is obvious from the results that at TIT=1700 K the gas cycle efficiency first increases slowly upto rp=27 and afterwards rapidly in case all cooling means. This is attributed to the fact explained earlier.

- **Variation of specific plant specific fuel consumption for varying compressor pressure ratio**

Fig 4.10 shows the variation of 'sfc' with various rp for constant TIT = 1700K and Tb = 1122K using different cooling means. The results show that lowest 'sfc' is possible by selecting higher r_p. This is because of the fact that at higher r_p inlet temperature of air at combustor increases which in turn reduces the amount of enthalpy required at combustor for same TIT and thus the specific plant specific fuel

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consumption decreases. Higher and lower specific fuel consumption are observed in case CLSC and AICC respectively.

- **Variation of mass of coolant to gas $(\dot{m}_c/\dot{m}_g)$ for varying compressor pressure ratio**

Fig, 4.11 depicts a comparative study of mass of coolant to gas $(\dot{m}_c/\dot{m}_g)$required for varying r_p at fixed value of TIT=1700K and T_b=1122K for various cooling means. The higher $(\dot{m}_c/\dot{m}_g)$ is exhibited by all air cooling means as compared to steam cooling means. It can be seen that variation in cooling flow rates with change in r_p for all cooling means is not appreciable, rather monotonous.

- **Variation of plant specific work for varying TIT**

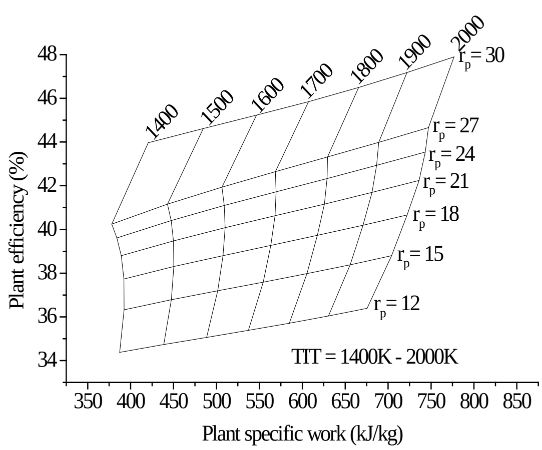
Fig. 4.12 shows the effect of varying turbine inlet temperature (TIT) on plant specific work. It is observed that plant specific work increases linearly with the increase in TIT for all cooling means. This behaviour is because of the fact that increment in TIT increases the turbine work meanwhile compressor work remains same at constant rp, which results in the increment in plant specific work. Higher value of plant specific work is observed for SICC while lower value is for AICC.

- **Variation of plant efficiency for varying TIT**

Fig. 4.13 shows the effect of varying turbine inlet temperature (TIT) on plant efficiency. It is observed that plant efficiency increases with the increase in TIT at constant rp for all cooling means. For higher value of TIT gas turbine output is higher and to achieve higher TIT although plant specific fuel consumption at combustor is higher but compressor work requirement is constant for constant rp so it increases the plant efficiency. For AICC higher value of plant efficiency may be observed while lower value is for CLSC.

- **Variation of specific plant specific fuel consumption for varying TIT**

Fig. 4.14 shows the effect of varying turbine inlet temperature (TIT) on plant specific plant specific fuel consumption. It is observed that specific plant specific fuel consumption decreases with the increase in TIT at constant rp for all cooling means. It is because of the fact that specific plant specific fuel consumption is inversely proportional to plant efficiency. As in fig.



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4.13 plant efficiency increases with the TIT thus the specific plant specific fuel consumption decreases with increase in TIT. Highest specific plant specific fuel consumption is seen with CLSC and lowest specific plant specific fuel consumption is seen with AICC.

- **Variation of mass of coolant to gas $(\dot{m}_c/\dot{m}_g)$ for varying TIT**

The coolant requirement increases with increase in TIT for all types of cooling considered. The minimum coolant requirement is found to be in the case of STC. The maximum cooling requirement is found in the case of AICC and it increases faster with increase in TIT as compared to other cooling means. The results of CLSC and SICC overlap each other because the values of specific heat of steam, entering to the gas turbine blades and heat exchanger (blades) effectiveness are same in both the cases. Fig. 4.15 shows that if TIT is increased beyond 1700 K (temperature level in modern turbines), steam cooling is the best coolant option whereas for air-cooling ATC, followed by AFC are the next options. ATC offers reduced heat transfer due to transpired coolant film completely shrouding the blade surface. Similarly, AFC also provides coolant film over blade surface, which acts as thermal barrier for the hot gas.

Figure 4.1 Basic gas turbine with convection cooling.

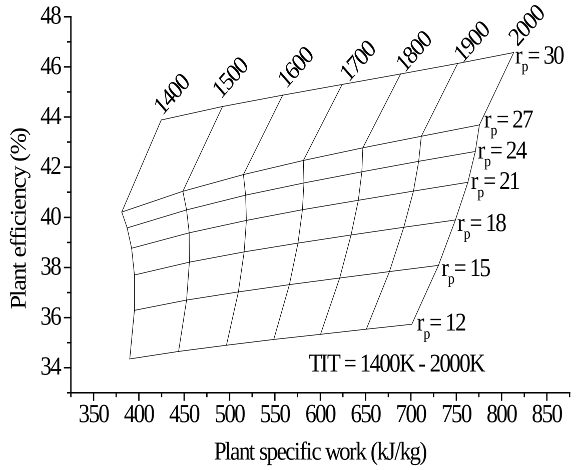

Figure 4.2 Basic gas turbine withfilm cooling.

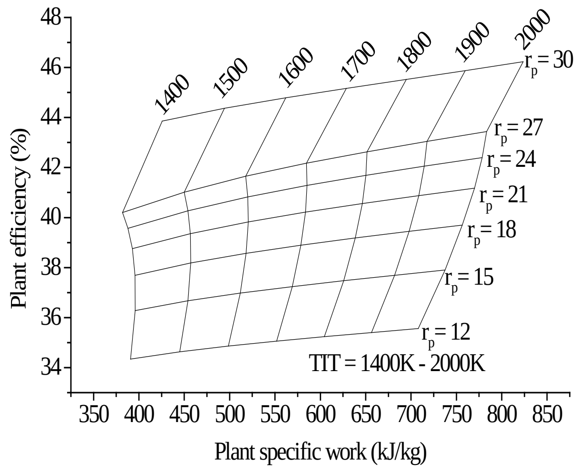

Figure 4.3 Basic gas turbine with transpiration cooling.

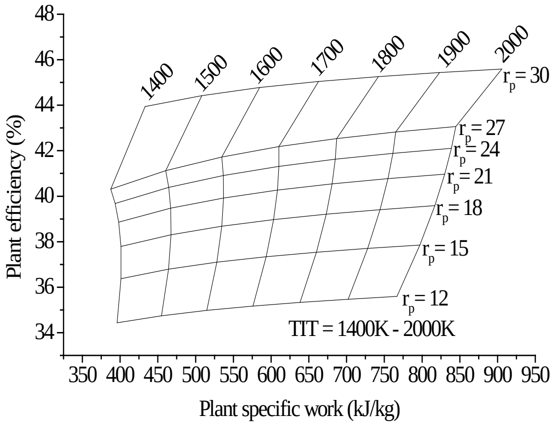

Figure 4.4 Basic gas turbine with open loop steam convection cooling.

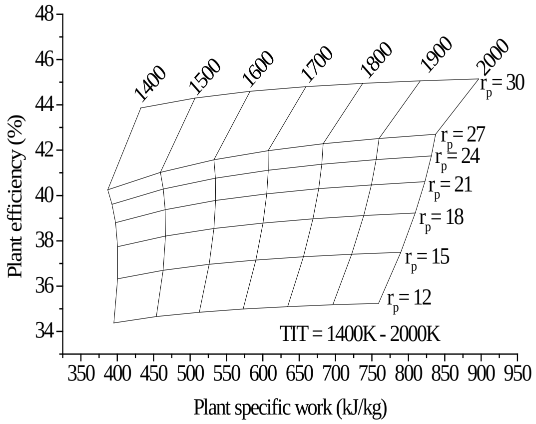

Figure 4.5 Basic gas turbine with open loop steam film cooling.

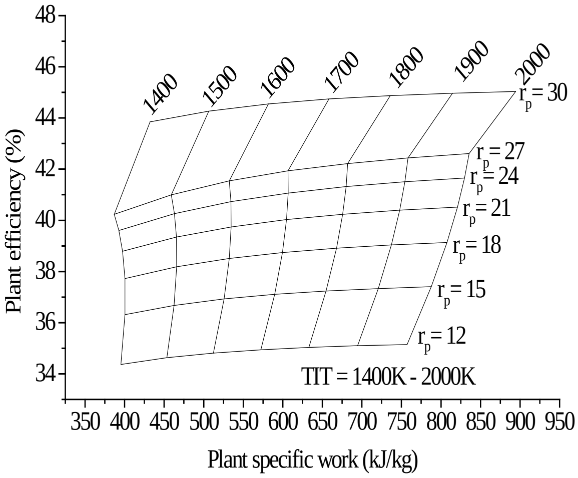

Figure 4.6 Basic gas turbine with open loop steam transpiration cooling.

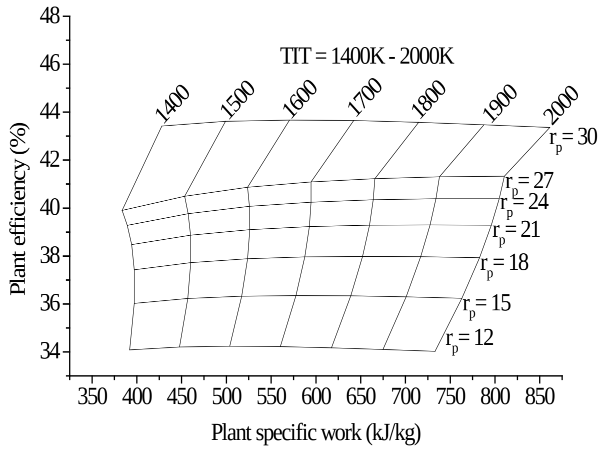

Figure 4.7 Basic gas turbine with closed loop steam convection cooling.

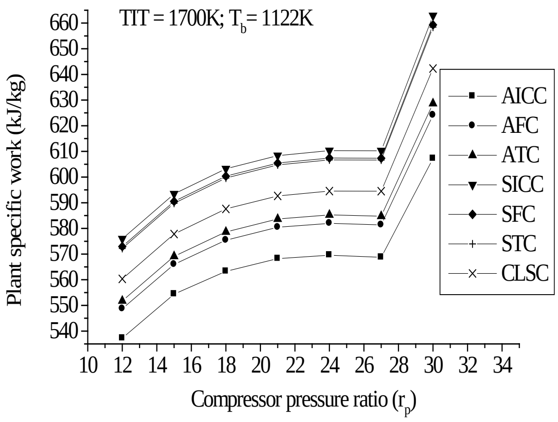

Figure 4.8 Compressor pressure ratio vs. Plant specific work with different cooling means for basic gas turbine.

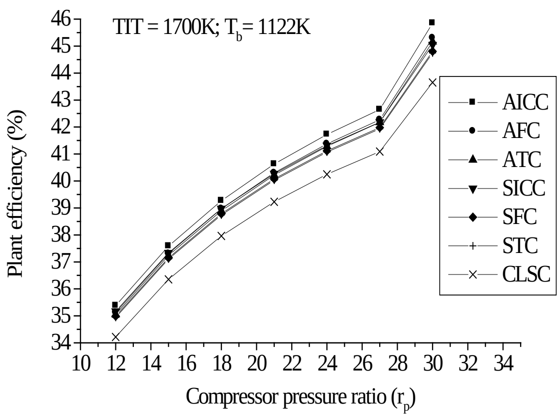

Figure 4.9 Compressor pressure ratios vs. Plant efficiency with different cooling means for basic gas turbine.

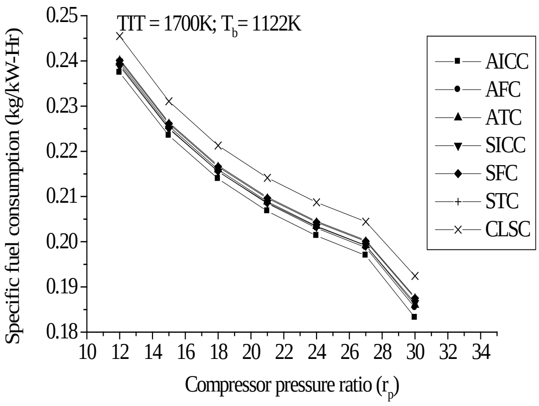

Figure 4.10 Compressor pressure ratio vs. sfc with different cooling means for basic gas turbine.

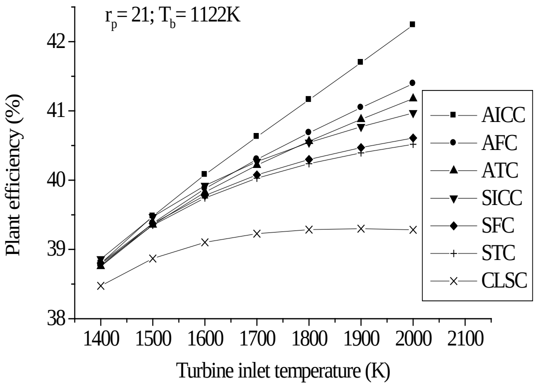

Figure 4.13 TIT vs. plant efficiency with different cooling means for basic gas turbine.

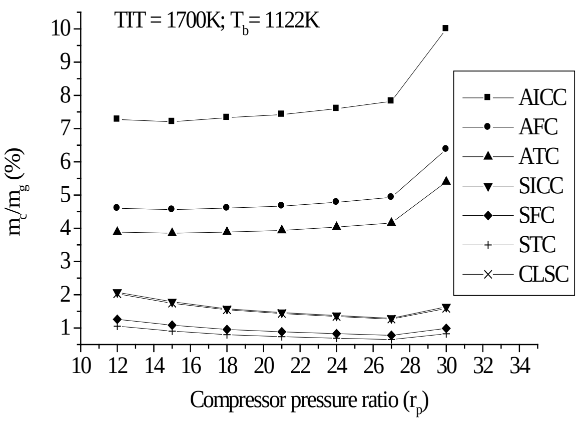

Figure 4.11 Compressor pressure ratio vs m_c/m_g with different cooling means for basic gas turbine.

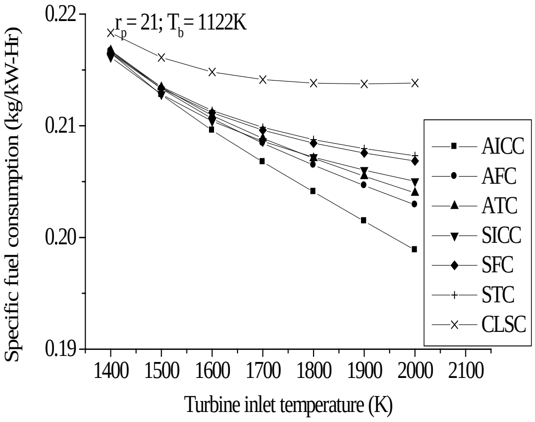

Figure 4.14 TIT vs. SFC with different cooling means for basic gas turbine.

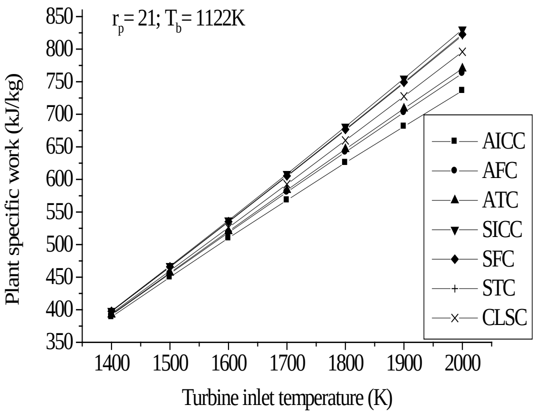

Figure 4.12 TIT vs. plant specific work with different cooling means for basic gas turbine.

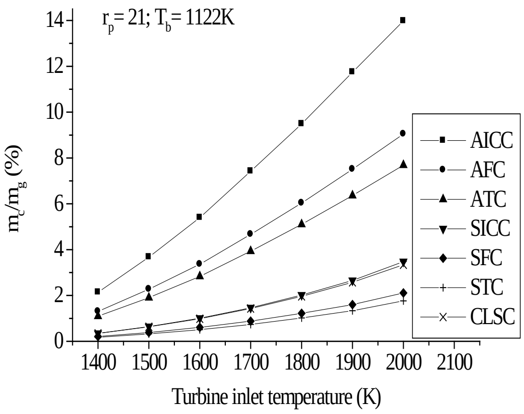

Figure 4.15 TIT vs. m_c/m_g with different cooling means for basic gas Turbine.

4. Conclusion

Based on mathematical modeling of various elements of the cogeneration cycle and governing equations, using seven cooling means of gas turbine blade cooling has been developed in C++ language to predict the thermodynamic performance of eight configurations of cogeneration plants. For all seven cooling means, the coolant requirement increases linearly with TIT as expected. The effect of r_p on cooling flow requirement is negligible for all cooling means. The behavior of basic gas turbine (BGT) plant specific work with r_p at higher and lower range of TIT is different. For higher range of TIT it increases with r_p while for lower range of TIT it first increases and then decreases for all seven cooling means. BGT plant efficiency increases linearly with increase in TIT at any r_p for all seven cooling means. For basic gas turbine plant efficiency is higher with AICC and minimum with CLSC while plant specific work is higher with SICC and minimum with AICC at constant TIT = 1700K.

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