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Parametric Analysis of Mixed Gas-Steam Cycle for Various Blades Cooling Means

Shubham Tiwari[1], Laxmi Kant Sahu[2], Ankita Sharma[3]

[1]Department of Mechanical Engineering, Technocrats Institute of Technology Bhopal, MP, India; [2]Department of Mechanical Engineering, Institute of Technology, Korba, Chhattisgarh-495683, India; [3]Department of Mechanical Engineering, Technocrats Institute of Technology Bhopal, MP, India

Abstract

The present work deals with the parametric analysis of mixed cycle based gas turbine plants 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. With the help of input data, results have been plotted and analysed in terms of dependent and independent parameters. Total Energy approach is proposed to use all the heat energy in a power system at the different temperature levels at which it becomes available to produce work.

Key words: Simple injected gas turbine (SIGT), Air internal convection cooling (AICC), Air film cooling (AFC), Air transpiration cooling (ATC)

1. Introduction

Energy has been universally recognized as one of the most important inputs for economic growth and human development. These energies are utilized in many applications such as in gas turbine based thermal power plants. A large number of configurations of gas turbine based thermal power plants are possible. A simple and mixed gas turbine cycle is represented in figure 1.1 & 1.2.

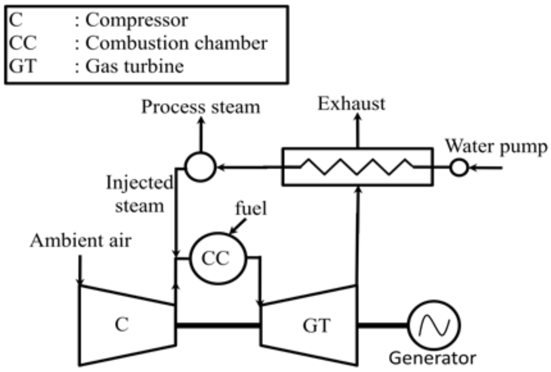

Figure 1.1 Schematic diagram of a Steam Injected Gas Turbine Cycle

Fifi and Antar (2016) have carried out the common cogeneration application in steam injected gas turbine. In steam injected gas turbine the heat of exhaust gases is used to produce the steam, and this steam injects to combustion chamber. The generated steam not only will be directed to the combustion chamber but also will be used to cool the blade turbine by using a closed loop. Athari et.al (2016) have carried out the results of energy and exergy analyses

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of two biomass integrated steam injection cycles and combined power cycles. Fog cooling, steam injection and adding steam turbine cycles to gas turbine cycles can enhance the performance of power generation systems.

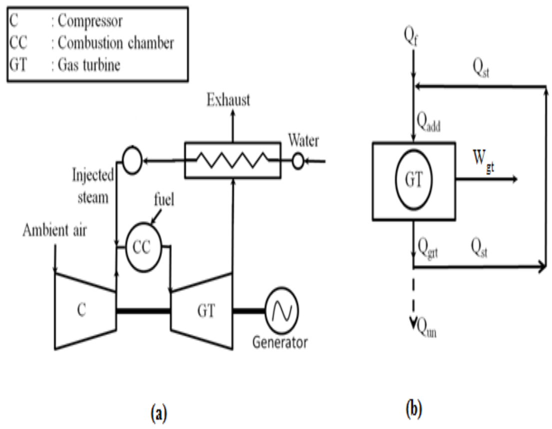

Figure 1.2 (a) Schematic diagram of a mixed cycle power plant (b) Energy flow diagram of mixed cycle power plant

Even with its lower heat value, biomass can be substituted for fossil fuels. Shukla and Singh (2016) Present paper deals with the study for performance evaluation of steam-injected gas turbine (STIG) based power plant with inlet evaporative cooling. It investigates 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 (2016) proposed combined scheme for applying a vapour compression refrigeration system to cool the condenser of a steam plant. In this scheme the maximum coefficient of heat transfer between the steam and refrigerant is achieved, and the condenser becomes compact. In addition, this combined scheme allows controlling the condenser temperature irrespective of the ambient conditions. El-Shazly et al. (2016) experienced 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. Mosafa et al. (2008) have demonstrated that in recent decade, more attention has been paid to reheat gas turbine cycles because of their high exhaust gas temperature. Using heat energy of exhaust gases in a steam generating system is one scheme that has been suggested. In this research, heat recovery steam generator (HRSG) is analyzed as an appliance for transferring heat between exhaust gases and water and design parameters were optimized.

Dong et al. (2009) have used conjugate calculation methodology to simulate the C3X gas turbine vanes cooled with leading edge films of "shower-head" type. The temperature gradients formed between the cooled metallic vane and the hot gas can improve the stability of the boundary layer flow because the gradients possess a self stable convective structure.

Sanjay et al. (2009) have 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. The HRSG has a steam drum generating steam to meet coolant requirement, and a second steam drum generates steam for process heating. Gas turbine stage cooling uses open loop cooling or closed loop cooling schemes. Internal convection cooling, film cooling and transpiration cooling techniques employing steam or air as coolants are considered for the performance evaluation of the cycle. Ting and Chang (2008) have 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. Numerical simulations have shown that injecting a small amount of water droplets into the cooling air improves film-cooling performance significantly. Sanjay et al. (2008) have presented a comparative study of the influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle power plant. Krishnan et al. (2008) have attempted to establish 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. LTHC in the presence of film cooling is explored using a simple analytical approach and provides a simple baseline prediction methodology and sets direction for optimization needs.

A vast quantity of works have done on these field but here it is restricted mainly to those power plants which are associated with recovery and utilisation of waste heat as a modus operandi to enhance the power plant performance. These papers also present a simple and effective blade cooling techniques to improve the various design parameters. The Paper is organized as following the introduction, Section 2 describe the proposed modelling of various configurations of blade cooling to improve the efficiency of gas turbine based power plants. 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. Gas Turbine Blade Cooling Models:

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.

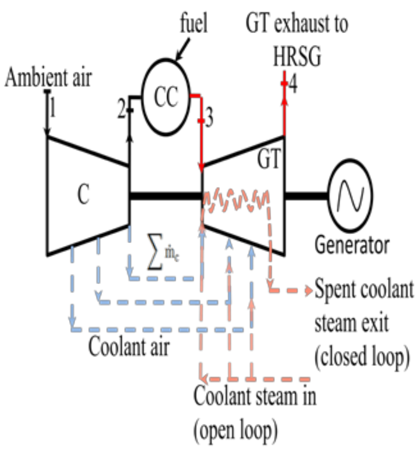

Figure 2.1 Schematic of a cooled gas turbine with open and closed loop blade cooling

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 (2002), Harlock et al. (2001), El-Masri (1988), etc. the model used for cooled turbine is the refined version of Louis et al. (2002) model and Harlock at al. (2001) 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.

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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.

2.1. 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

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

Using effectiveness from equation (2.2), we have

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

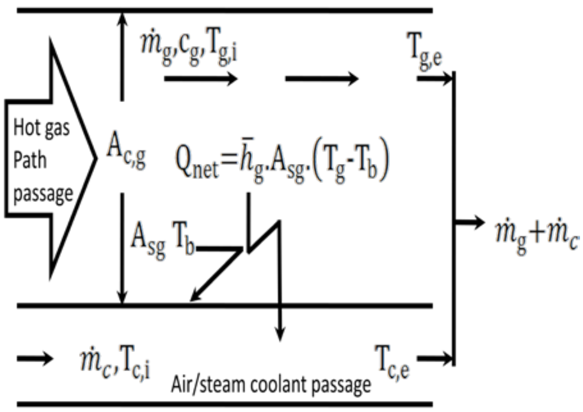

Figure 2.2 Model for open loop internal convection turbine blade cooling

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

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

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After rearranging, the equation (2.6) becomes

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

$$=\lambda.\left[\frac{C_{p,g}}{C_{p,c}}\right].\bar{S}t_{in}.\left[\frac{(T_{g,i}-T_b)}{\varepsilon.(T_b-T_{c,i})}\right] \qquad (2.7)$$

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

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

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

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

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

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

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

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 $c_{p,a}$ 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 exp

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

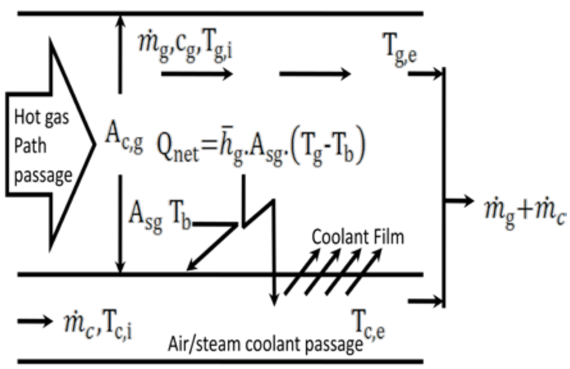

Figure 2.3 Model for open loop film cooling of turbine blade

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. 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

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

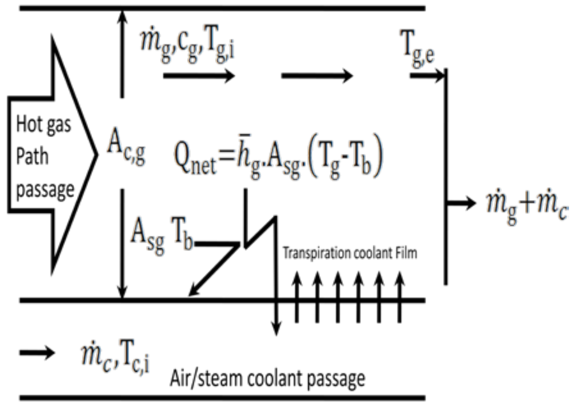

Figure 2.4 Model for transpiration cooling of turbine blade

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

The thermodynamic analysis of the selected configurations of mixed cycle power plants has been carried out on the basis of results obtained from the program developed, using the modelling of the different components of the cycle and governing equations from 2.1 to 2.15. The software has been validated with the published work of Bolland et al. (1991) for steam injected cycle, Yadav et al. (2003) for HAI cycle and Chiesa et al. (1995) for HAT cycle. Methodical swot of performance of configurations is enabled by separating this chapter in to numerous

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sections and sub-section combining the configurations into groups. The effects of cooling means on the performance have been studied for all the configurations.

3.1 Assumptions and Input Parameters

For basic cycle configuration, the range of selected compressor pressure ratio varies from 12 to 30. As per El-Masri (1988) it is desirable to go for higher TIT with the advancement of blade material technology available, thus the range of TIT was chosen to vary from 1400K to 2000K 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 843K due to metallurgical conditions in HRSG as per current state-of-art technology. It is essential that for better utilisation 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 i.e. 353K. The assumptions and input parameters are tabulated in Table 3.1

Table 3.1 Input Data used for Analysis

COMPONENT	**PARAMETER**	**UNIT**
Atmospheric conditions	p_o = 1.01325	bar
	T_o = 288	K
Inlet section	Pressure loss (Δp_{loss}) = 1.0 of entry pressure	%
Gas Properties	Cp = f(T) Enthalpy h = $\int c_p(T)dT$	kJ/kg-K kJ/kg
Compressor	i. Polytropic efficiency (η_{pc}) = 92.0 ii. Mechanical efficiency (η_m) = 98.0 iii. Air inlet temperature = 288 iv. Inlet pressure loss = 0.5% of entry pr.	% % K bar
Combustor	i. Combustor efficiency (η_{cc}) = 98.0 ii. Excess air = 200 (So air is 300% of the stoichiometric value) iii. Pressure loss (Δp_{cc}) = 2.0 of entry pressure	% % %

	iv. $(LHV)_f$ = 42000	kJ/kg
Gas turbine	i. Polytropic efficiency (η_{pt})=92.0 ii. Exhaust pressure= 1.08 iii. Turbine Blade Temprature=1122 iv. Intercooler pressure Loss=2.0 v. Intercooler effectiveness= 92 vi. Recuperator pressure Loss=2.0 vii. Recuperator effectiveness= 70	% bar K bar % bar %
HRSG	i. Steam pressure=70 ii. Steam temperature 843 iii. Steam reheat temperature=843(Max)	bar K K
Alternator	Alternator efficiency=98.	%
	5	

3.2 Gas Properties Evaluation as a Function of Temperature.

For evaluating properties of gas specific heat of air and combustion products are considered as a function of temperature and pressure. In real situations the specific 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 air to fuel ratio, which is governed by compressor pressure ratio and turbine inlet temperature.

3.3 Effect of different cooling means for various TIT and r_p on plant specific work, plant efficiency, plant specific fuel consumption and $\dot{m}_c/\dot{m}_g$ for SIBGT configuration

- **Variation of plant efficiency and plant specific work with various TIT and r_p**

Figures 3.1 to 3.3 are plotted for different cooling means. At any TIT except above r_p =27, the plant specific work first increases and then decreases with increase in r_p. This is because of the facts that increase in TIT increases the turbine work meanwhile compressor work remains same for any value of r_p which results in the increase in plant specific work. As the higher value of TIT demands slight increase in plant 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 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 compensates the loss. Owing to these facts the plant efficiency increases. It is observed that there is discontinuity after r_p greater than 27. It may be due to the increased air temperature at the exit of the compressor requiring gradually decreasing fuel flow rate to get given TIT. There may be requirement of cooling rather than heating in the combustor.

- **Variation of coolant requirement with SFC for various compressor pressure ratio vs TIT**

Figures 3.4 to 3.6 show the variation of coolant requirement with specific fuel consumption (SFC) for varying compressor pressure ratio and TIT for different open air cooling means. It is evident from the plots that with increase in TIT at a given r_p there is increase in SFC as well as coolant requirement. Increase in coolant requirement is obvious as higher value of TIT requires more coolant because of constraints in material properties. However, increase in coolant requirement is also observed in case of pressure ratio increment as this is also one of the methods for increasing the turbine temperature. Higher specific fuel consumption is observed for higher values of r_p because of extra compressor work required to compress the air.

- **Variation of plant efficiency with various compressor pressure ratio**

Figure 3.7 depicts the variation of plant efficiency with compressor pressure ratio, for different means of cooling for SIBGT configuration at TIT=1700 K and T_b=1122 K. It is observed that there is significant increase in plant efficiency with increase 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 hence 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 efficiency is exhibited by AICC and ATC cooling means respectively. In ATC, coolant passes through numerous channels/holes in the blade wall which in turn lowers the temperature of expanding gas and thus the efficiency.

- **Variation of plant specific work with various compressor pressure ratio**

The effect of r_p on plant specific work is depicted in Figure 3.8 for various means of cooling at TIT=1700 K and Tb=1122 K for

SIBGT system. It is obvious from the results that the gas cycle specific work first increases slowly upto r_p=27 and afterwards rapidly in case all cooling means. This behaviour is because of the fact that the increasing compressor work input is not able to offset the advantage gained by increasing turbine output and lesser fuel requirement, while the coolant requirement are more or less unaffected with increasing r_p. Obviously, the cooling schemes needing less cooling requirement such as ATC offers higher value of plant specific work for all value of r_p at any TIT.

- **Variation of specific fuel consumption with various compressor pressure ratio**

Figure 3.9 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 rp. This is because of the fact that at higher rp inlet temperature of air in the combustor increases which in turn reduces the amount of heat required at combustor for same TIT and thus the specific plant specific fuel consumptions decreases. Higher and lower specific plant specific fuel consumption are observed in case ATC and AICC respectively although difference is minute.

- **Variation of mass flow rate of coolant to gas with various compressor pressure ratio**

Figure 3.10 depicts a comparative study of mass flow rate 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 ratio $(\dot{m}_c/\dot{m}_g)$ is exhibited by AICC as followed by AFC and then ATC. It can be seen that flow rate decreases for r_p 12 to 24 and remains more or less constant thereafter for a given method of cooling. This is because of the fact that compressor pressure ratio marginally effects the air coolant temperature.

- **Variation of plant efficiency with various TIT**

Figure 3.11 shows the effect of turbine inlet temperature (TIT) on plant efficiency. It is observed that plant efficiency increase with the increases in TIT at constant r_p for cooling means. Higher value of plant efficiency may be observed with AICC while lower value is for ATC and AFC. Further it is observed for AICC that there is slight decrement of plant efficiency. It decreases up to TIT=1600K then a steep increment is observed. This is due to extra amount of fuel needed to heat injected steam upto TIT which is not compensated by the additional amount of work done by gas turbine owing to combined effect of amount of injected steam and higher value of specific heat of mixture of gas and steam.

- **Variation of plant specific work with various TIT**

Figure 3.12 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 behavior 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 AICC while lower value is for ATC.

- **Variation of plant specific fuel consumption with various TIT**

Figure 3.13 shows the effect of varying turbine inlet temperature (TIT) on plant specific fuel consumption. It is observed that plant specific fuel consumption increases with the increase in TIT at constant rp for all cooling means. It is because of the fact that increase in fuel consumption does not yield corresponding work output, due to increasing cooling requirement, pumping, cooling and mixing losses in turbine expansion path. Since the coolant requirement is higher in the case of ATC at higher TIT highest specific plant specific fuel consumption is seen with ATC and lowest specific plant specific fuel consumption is seen with AICC.

- **Variation of mass of coolant to gas $(\dot{m}_c/\dot{m}_g)$ with various 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 ATC. This is because of the fact that the transpiration cooling provides better heat transfer due to large number of holes in blades through which coolant passes and forms a film of coolant serving as thermal barrier for hot gas. 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 AFC and ATC are quite close to each other because the values of specific heat of air, entering to the gas turbine blades and heat exchanger (blades) effectiveness are same in both the cases. Fig. 3.14 shows that if TIT is increased beyond 1700 K (temperature level in modern turbines), ATC cooling is the best coolant option followed by AFC. 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.

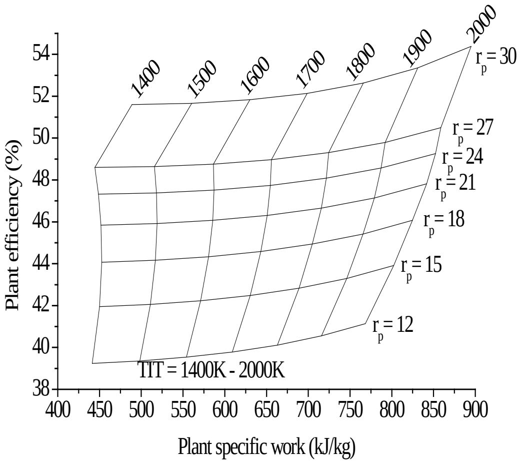

Fig. 3.1 Variation of plant efficiency with plant specific work for SIBGT using AFC

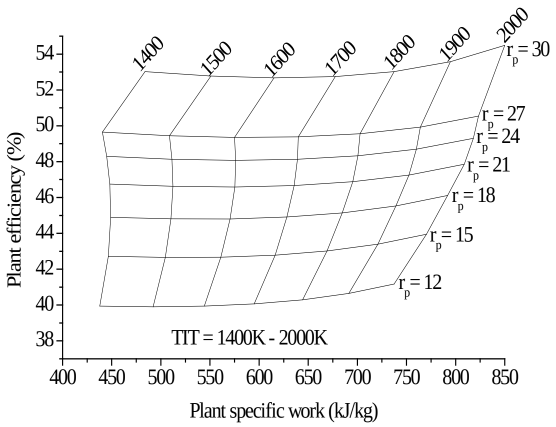

Fig. 3.2 Variation of plant efficiency with plant specific work for SIBGT using AICC

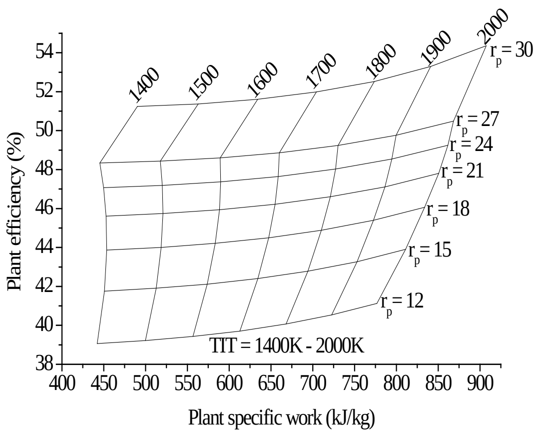

Fig. 3.3 Variation of plant efficiency with plant specific work for SIBGT using ATC

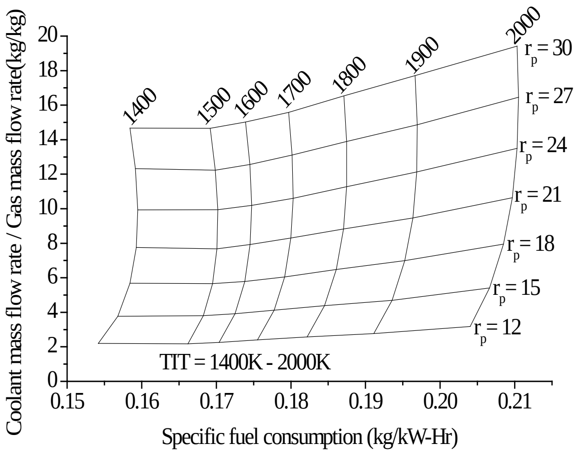

Fig. 3.4 Variation of coolant requirement with SFC for SIBGT using AICC

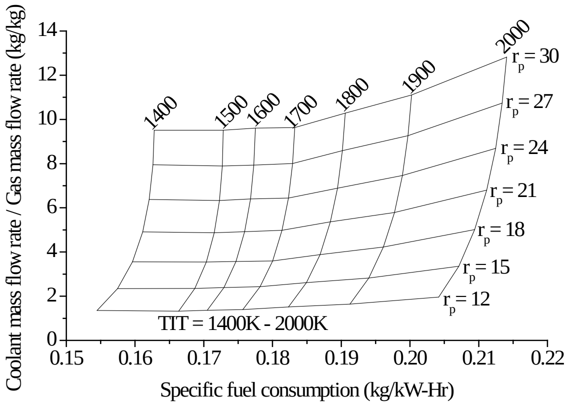

Fig. 3.5 Variation of coolant requirement with SFC for SIBGT using AFC

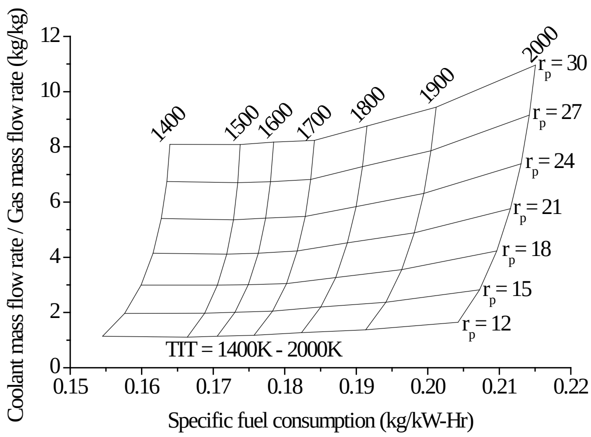

Fig. 3.6 Variation of coolant requirement with SFC for SIBGT using ATC

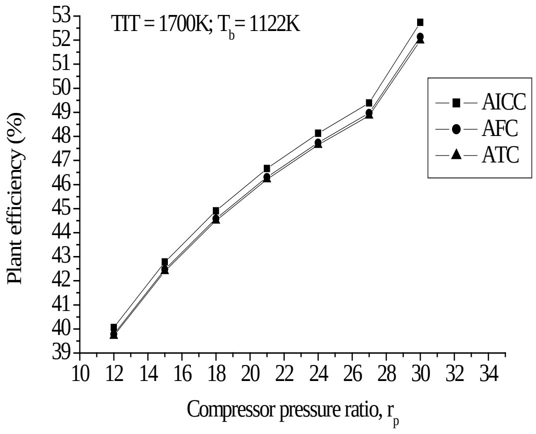

Fig. 3.7 Variation of plant efficiency with r_p for SIBGT using AICC, AFC and ATC

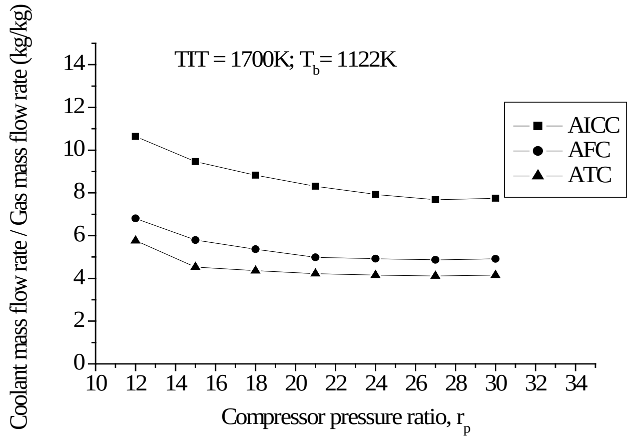

Fig. 3.10 Variation of coolant requirement with r_p for SIBGT using AICC, AFC and ATC

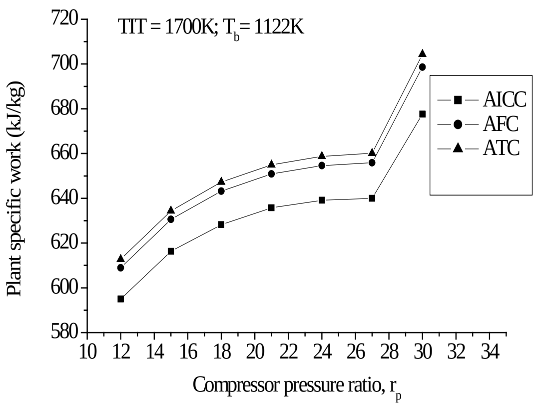

Fig. 3.8 Variation of plant specific work with r_p for SIBGT using AICC, AFC and ATC

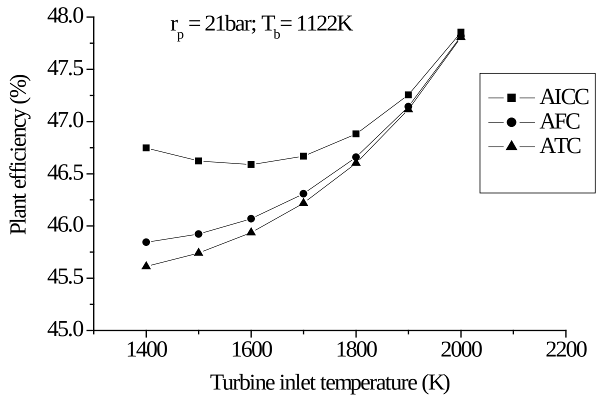

Fig. 3.11 Variation of plant efficiency with TIT for SIBGT using AICC, AFC and ATC

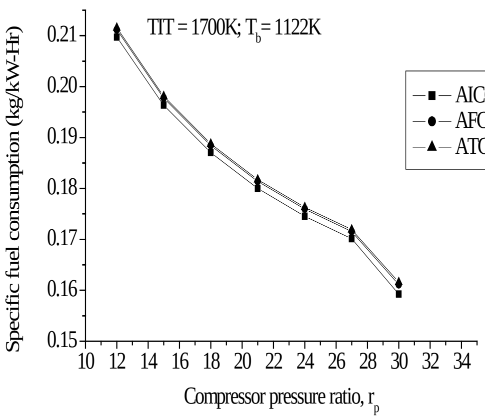

Fig. 3.9 Variation of specific fuel consumption with r_p for SIBGT using AICC, AFC and ATC

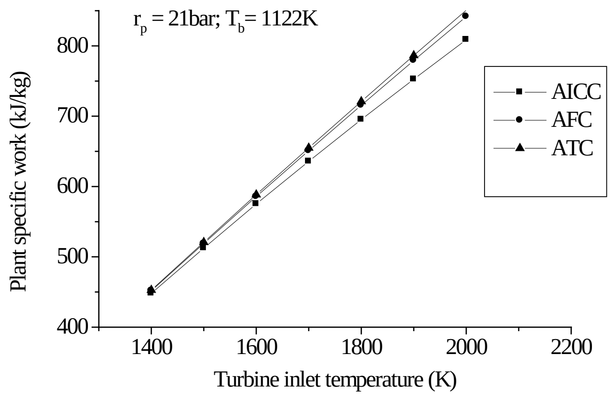

Fig. 3.12 Variation of plant specific work with TIT for SIBGT using AICC, AFC and ATC

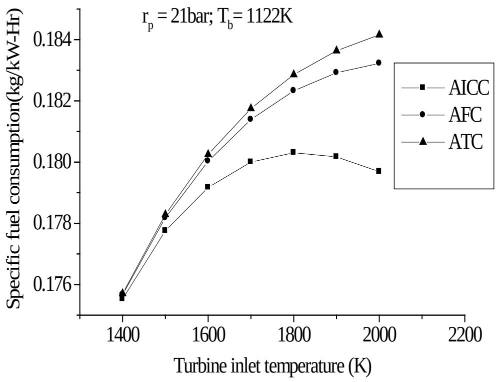

Fig. 3.13 Variation of SFC with TIT for SIBGT using AICC, AFC and ATC

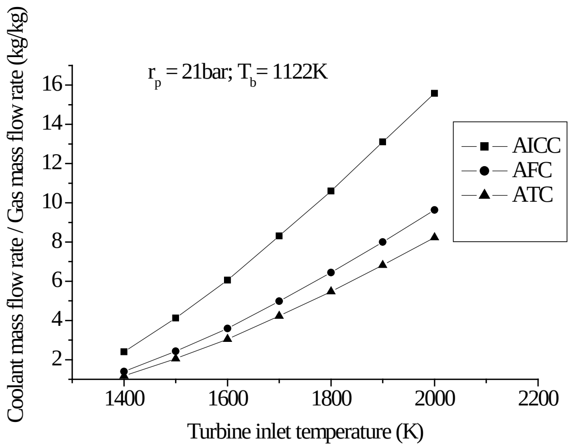

Fig. 3.14 Variation of coolant requirement with TIT for SIBGT using AICC, AFC and ATC

4. Conclusion

From theoretical, numerical and simulation analysis it is found that mixed cycle offers high plant efficiency and plant specific work compared to basic gas turbine cycle. There is an appreciable enhancement in the mixed cycle specific work (12 to 16%) and efficiency (2 to 4%) over the basic gas turbine cycle but at the complexity of plant. The stack temperature in steam injected cycle is higher signifying the less utilization of waste heat energy, however a separate provision of steam generation in HRSG for cogeneration may serve the purpose. The effect of r_p on cooling flow requirement is negligible for all three cooling means. The cooling flow requirement increases with the increase in turbine inlet temperature (TIT). AICC and ATC cooling method is recommended if the plant efficiency and plant specific work is main consideration respectively.

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AUTHORS

Shubham Tiwari received the Bachelors of Engineering degree in Mechanical Engineering from Annamalai University, Tamilnadu, India in 2013 and Masters of Technology in Heat Power and thermal Engineering from Technocrats Institute of Technology, Bhopal, India in 2016. He had worked in Institute of Technology, Korba (C.G) as an Assistant Professor in Mechanical Engineering department.

Laxmi Kant Sahu received the Bachelors of Engineering degree in Mechanical Engineering from Govt. Engineering College, Bilaspur, Chhattisgarh Swami Vivekananda Technical University, Bhilai, Chhattisgarh, India in 2010 and Masters of Technology in Machine Design from Guru Ghasidas Vishwavidyalaya (a central university), Bilaspur, Chhattisgarh, India in 2014. He had worked in Institute of Technology, Korba (C.G) as an Assistant Professor in Mechanical Engineering department.

Ankita Sharma received the Bachelors of Engineering degree in Mechanical Engineering from Institute of Technology, Korba, Chhattisgarh Swami Vivekananda Technical University, Bhilai, Chhattisgarh, India in 2014 and Masters of Technology in Heat Power and thermal Engineering from Technocrats Institute of Technology, Bhopal, India in 2016. He had worked in Govt. Polytechnic College, Korba (C.G) as an Assistant Professor in Mechanical Engineering department.