Performance Evaluation of Gas-Steam Combined Cycle Having Transpiration Cooled Gas Turbine

Sanjay Kumar and Onkar Singh,
Department of Mechanical Engineering,
Harcourt Butler Technological Institute, Kanpur- 208002, U.P., INDIA,

ABSTRACT

In recent years improved gas turbine performance through developments in high temperature materials and blade cooling methods has made a positive impact on the combined cycle performance. Transpiration cooling technique has emerged as the most promising technique to improve the gas turbine cycle performance by allowing higher turbine inlet temperatures. This paper concentrates on improving the combined cycle performance by allowing higher turbine inlet temperatures (TIT) using transpiration cooling of gas turbine blades. A four-stage advanced gas turbine coupled with the dual pressure steam bottoming cycle is considered for the performance of combined cycle. Realistic input parameters used in current industrial practice have been considered for this study. The effects of variation of TIT on the performances of topping, bottoming and combined cycle are presented and discussed. For the combined cycle with transpiration cooled gas turbine an increase in TIT from 1600 K to 1800 K exhibits the combined cycle efficiency increase by 2.37 percent and the combined specific work increases by 185.42 kJ/kg. The results indicate that at a TIT of 1800 K the achievable efficiency of combined cycle with transpiration cooled gas turbine is 59.97 percent.

Keywords: combined cycle, temperature inlet temperatures, transpiration cooling, dual pressure HRSG, performance analysis

INTRODUCTION

During the last decades there has been continuous development of combined cycle power plants for getting their increased efficiency and low emissions. The gas-steam combined cycle uses the exhaust heat from the gas turbine cycle to increase the power plant output through steam cycle and boosts the overall cycle efficiency, substantially above that of the simple gas cycle or steam cycle in isolation. The foremost methods which help achieving better combined cycle performance are: increasing the inlet temperature of the gas turbine (TIT), inlet air-cooling, applying gas reheat, steam or water injection into the gas turbine, and diminishing the temperature differences in the heat recovery steam generator as well as lowering the condenser pressure.

Amongst various options the increase in turbine inlet temperature (TIT) and cycle pressure ratio has vast impact on the gas turbine cycle performance. The key technique for realizing higher TIT is to maintain the turbine blade metal temperature within the permissible metallurgical limit and this can be accomplished by employing an efficient cooling technique. The influence of various available cooling techniques e.g. convection, film and transpiration cooling techniques on gas turbine cycle performance and combined cycle performance have been investigated by researchers [1–3]. Studies show that transpiration cooling technique uses the coolant more effectively than the film and convection cooling techniques and is able to maintain the blade metal temperature within the permissible range while operating with high gas temperatures [4]. In transpiration cooling technique, the coolant is ejected out from a large number of cooling holes on the blade surface which entirely cover the blade surface by formation of a large number of small conjoint films of coolant discharging out from each hole over the complete blade surface. The efficacy of the film formation and its capability of isolating the blade from high temperature gas depend upon the interaction between the injected coolant and the mainstream gas. Work in this area has been done by Polezhaev, J. [4], Horlock et.al. [5] and Kumar, S. and Singh, O. [6].

In a gas-steam combined cycle the performance of bottoming cycle is improved by multi pressure steam generation in HRSG [7]. Bassily [8] found an optimum dual pressure combined cycle to be almost as efficient as the most efficient commercially available triple pressure reheat combined cycle. Chin and El-Masri [9] worked for optimization of a dual pressure bottoming cycle as a function of gas turbine exhaust temperature. In the present study the main objective is to evaluate the performance of combined cycle with modern dual pressure steam cycle and transpiration cooling of gas turbine blades. Based on study the cycle performance curves with the system parameters have been developed and discussed. Study shows that by using transpiration cooling higher TIT's can be employed which enhance the overall performance of combined cycle by improvement in gas cycle as well as in steam cycle by increasing superheat approach temperature differences in the heat recovery steam generator (HRSG).

NOTATION

c_p	constant pressure specific heat (kJ/kg-K)
F_{bd}	blowdown factor
h	specific enthalpy (kJ/kg),
LHV	lower heating value (kJ/kg)
Ma	mach number
$\dot{m}$	mass flow rate (kg/s)
Pr	Prandtl Number
p	pressure (Pa)
q	heat added per unit mass of working air (kJ/kg)
St	Stanton Number
T	temperature (K)
v	specific volume (m^3/kg)
W	work (kW)

Greek symbols

α	cooling hole inclination angle (degree)
ε	Effectiveness
γ	adiabatic index
η	efficiency (%)
λ	ratio of cooled surface area to hot gas flow cross sectional area

Subscripts

aw	adiabatic wall
bfp	boiler feed pump
c	coolant, cooling
cc	combined cycle
cep	condensate extraction pump
ci	coolant in
comb	Combustor
cond	Condensate
dea	Deareator
ex	Exit
f	Fuel

fw	feed-water
g	Gas
gt	gas turbine
hp	high-pressure
hrsg	heat recovery steam generator
in	inlet
lp	low-pressure
mech	mechanical
p	polytropic
pp	pinch point
s	steam
sat	saturated
sh	super heater
st	steam turbine
stack	stack
sub	sub-cooled
trb	turbine

Acronyms

BFP	boiler feed pump
CEP	condensate extraction pump
Eco	economizer
Evp	evaporator
HP	high pressure
HPD	high pressure drum
HPST	high pressure steam turbine
HRSG	heat recovery steam generator
LP	low pressure
LPD	low pressure drum
LPST	low pressure steam turbine
SH	super heater
SFC	specific fuel consumption
TIT	turbine inlet temperature

COMBINED CYCLE CONFIGURATION

The schematic diagram of dual pressure combined cycle under consideration is shown in Figure 1. It is composed of a simple gas turbine cycle as topping cycle, involving transpiration cooled gas turbine blades and a steam cycle as bottoming cycle. The waste heat of gas turbine exhaust is recovered in a dual pressure heat recovery steam generator.

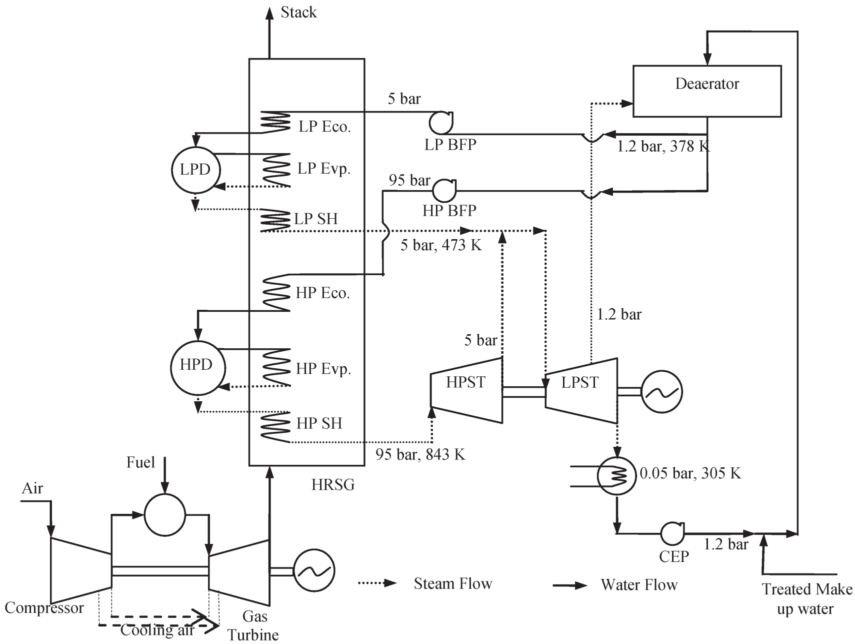

Figure 1. Schematic of dual pressure combined cycle configuration

GAS TURBINE CYCLE WITH TURBINE BLADE COOLING

Atmospheric air is compressed in a compressor to a higher pressure. The fuel is injected into the combustor to mix with the compressed air and is burnt at constant pressure. The hot gas from the combustor is allowed to expand through a gas turbine to achieve work. For cooling of gas turbine blades a fraction of the compressed air is bled from the compressor. This air bled from compressor is not available for the combustion process and also there is a loss of its pressure when flowing through the narrow cooling channels, which causes loss of performance in the gas turbine. The compression and expansion process with fluid friction have been considered polytropic. The specific heat of air in the compressor and that of working fluid in the gas turbine has been considered to vary with temperature. For the considered advanced four-stage gas turbine, the cycle efficiency of the cooled gas turbine is calculated by using mass and energy balance as described in [6].

As per cooling requirement a part of compressed air is bled from the compressor for cooling of gas turbine blades. In case of transpiration cooling the compressed air introduced into coolant injection holes on the blade surface mixes with expanding gases. Since the main flow and coolant flow differ in temperature, pressure and chemical composition, the mixing process is very obscure. Hence assuming adiabatic mixing of coolant and main flow the mixture properties i.e. temperature, specific heat, enthalpy etc are obtained as function of temperature and composition of mixed gas as described in [10].

The total pressure losses in mixing of coolant and mainstream, is calculated as [5].

$$\frac{\Delta p}{p_{trb,in}} = -\frac{\dot{m}_c}{\dot{m}_g}\gamma\frac{Ma_g^2}{2}.\left\{1+\frac{T_c}{T_g}-2\left(\frac{T_c}{T_g}\right)^{1/2}\cos\alpha\right\} \tag{1}$$

The coolant requirement for transpiration cooling of gas turbine blades is given by [10].

$$\frac{\dot{m}_c}{\dot{m}_g} = \lambda.St_g \ln\left[\frac{c_{pg}}{c_{pc}}.\frac{(T_g - T_{bo}) - \varepsilon_{aw}\left[T_g - \{T_{ci} + \eta_c(T_{bo} - T_{ci})\}\right]}{\eta_c(T_{bo} - T_{ci})} + 1\right] \tag{2}$$

STEAM TURBINE CYCLE

Steam cycle with dual pressure HRSG configuration is adopted in the considered combined cycle. The temperature-entropy diagram for the steam cycle is shown in Figure 2. The steam expanded up to condenser pressure in LP steam turbine is condensed to saturated water state in condenser and is pumped to deareator through condensate extraction pump. In deaerator the air removal process takes place using the steam bled from the low pressure turbine resulting in saturated feed water at deaerator pressure. For optimum heat recovery from the exhaust, the deaerator pressure is obtained by considering the deaerator temperature ratio given as

Table 1. Gas turbine cycle parameters

Compressor polytropic efficiency ($\eta_{p,cmp}$) = 90 %
Compressor inlet temperature =15°C
Compressor inlet pressure = 1.01325 bar
Combustor efficiency (η_{comb}) = 98.5 %
Pressure loss in combustion chamber = 3 % of entry pressure
Exhaust pressure = 1.12 bar
Gas Turbine polytropic efficiency ($\eta_{p,trb}$) = 90 %
Mechanical efficiency (η_{mech}) = 99 %
Transpiration cooling efficiency (η_c) = 0.75 [5]
Adiabatic wall film effectiveness (ε_{aw}) = 0.4 [5]
Turbine blade temperature = 830°C
Prandtl no. Pr_g = 0.7
Reynolds no. $Re_g = 1.6x10^6$
Natural gas composition (% vol): CH_4 = 90.00; C_2H_6 = 4.50; N_2 = 4.00; CO_2 = 1.50;
Air composition (by % volume): O_2 = 21; N_2 = 79
LHV of natural gas = 44769 kJ/kg
Mach number of mainstream flow (Ma_g) = 0.95
Stoichiometric air-fuel ratio = 15:1
Excess air for combustion = 300%

$$\theta_{dea} = \frac{T_{fw,dea,ex} - T_{cond}}{T_{fw,sat,hp} - T_{cond}} \tag{3}$$

The hot water from the deaerator is pumped to the economizer section of HRSG through LP and HP boiler feed pumps respectively and is further heated to the saturation temperatures corresponding to its pressure. The saturated water is converted into steam in the LP and HP evaporators and further the steam is superheated in the corresponding superheaters. The superheated steam from HP superheater is expanded in high pressure steam turbine and is mixed with the superheated steam coming from LP superheater. The enthalpy of mixed steam is obtained considering the adiabatic mixing process.

The mixed steam is then expanded in low pressure steam turbine and appropriate part of steam is bled for deaeration process at required pressure. The mass of steam bled for deaeration is calculated as

$$m_{s,dea} \cdot (h_{s,dea,in} - h_{cond,cep,ex}) = m_s \cdot (h_{fw,dea,ex} - h_{cond,cep,ex}) \tag{4}$$

The steam generated in dual pressure HRSG is calculated by satisfying the two pinch points and the minimum value of stack temperature and quality of steam at the exhaust of LP steam turbine based on the

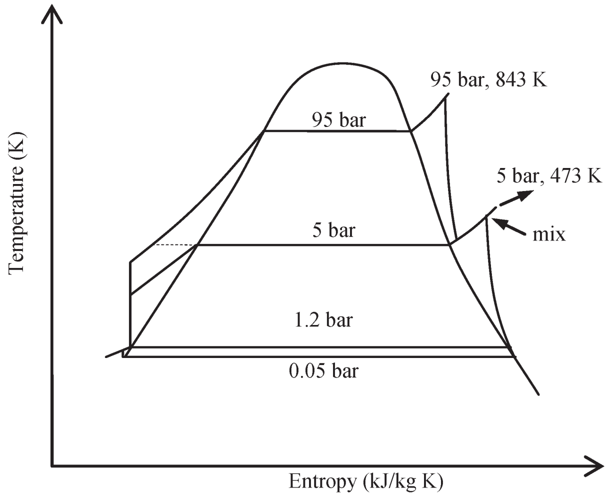

Figure 2. Temperature-entropy diagram for the steam cycle

optimization method proposed by [9]. Approach point in economizer is assumed to be zero. The steam cycle analysis data is given in Table 2.

The total mass of steam generated at two pressure levels is given as

$$\dot{m}_s = \dot{m}_{s,hp} + \dot{m}_{s,lp} \tag{5}$$

The fraction of energy used to heat steam at or above $T_{sat,lp}$,

$$A = \frac{q_{st} - \{m_{s,hp}(h_{fw,satlp,hp} - h_{fw,sub,hp}) + m_{s,lp}(h_{fw,sat,lp} - h_{fw,sub,lp})\}}{q_{st}} \tag{6}$$

where $h_{fw,satlp,hp}$ is enthalpy of feed water at LP saturation temperature but at HP pressure

$$q_{st} = m_{s,hp}(h_{s,sh,hp} - h_{fw,sub,hp}) + m_{s,lp}(h_{s,sh,lp} - h_{fw,sub,lp}) + blowdown \tag{7}$$

and

$$blowdown = F_{bd} \cdot \{m_{s,lp}(h_{fw,sat,lp} - h_{fw,sub,lp}) + m_{s,hp}(h_{fw,sat,hp} - h_{fw,sub,hp})\} \tag{8}$$

The fraction of energy used to heat steam at or above $T_{sat,hp}$,

$$C = \frac{m_{s,hp}(h_{s,sh,hp} - h_{fw,sat,hp})}{q_{st}} \tag{9}$$

The fraction of heat on gas side above $T_{g,pp,lp}$ must be equal to A i.e.

$$A = \frac{h_{g,hrsg,in} - h_{g,pp,lp}}{h_{g,hrsg,in} - h_{g,stack}} \tag{10}$$

Similarly the fraction of heat on gas side above $T_{g,pp,hp}$ must be equal to C i.e.

$$C = \frac{h_{g,hrsg,in} - h_{g,pp,hp}}{h_{g,hrsg,in} - h_{g,stack}} \tag{11}$$

where $h_{g,pp,lp}$ and $h_{g,pp,hp}$ are enthalpy of gas at temperatures $T_{g,pp,lp}$ and $T_{g,pp,hp}$, each pair (*h and T*) corresponding to LP pinch point and HP pinch point, respectively. The pinch points are related with gas temperatures $T_{g,pp,lp}$ & $T_{g,pp,hp}$ and feed water saturation temperature as

$$\Delta T_{pp,lp} = T_{g,pp,lp} - T_{fw,sat,lp} \tag{12}$$

$$\Delta T_{pp,hp} = T_{g,pp,hp} - T_{fw,sat,hp} \tag{13}$$

Maximum steam is generated when pinch point temperature differences $\Delta T_{pp,lp}$ and $\Delta T_{pp,hp}$ are minimum. To optimize the steam cycle performance initially $\dot{m}_s$ =1 is assumed and certain value is assigned to $\dot{m}_{s,lp}$. This gives the value of $\dot{m}_{s,hp}$ and subsequently that of the fraction 'A' from equation (6). For the given LP pinch point temperature difference equation (10) gives the value of $h_{g,stack}$ and hence that of $T_{g,stack}$. Using equations (9), (11) and (13) the HP pinch point temperature difference is found. This iteration process continues until the values of $\Delta T_{pp,hp}$ and $T_{g,stack}$ are satisfied. The mass of steam generated per unit of gas mass flow is given as

$$\frac{m_s}{m_g} = \frac{(h_{g,hrsg,in} - h_{g,stack}) \cdot \varepsilon_{hrsg}}{q_{st}} \tag{14}$$

Work of expansion through steam turbine is given by mass and energy balance as

$$W_{st} = m_{s,st}(h_{s,st,in} - h_{s,st,ex})$$

where

$$(h_{s,st,in} - h_{s,st,ex}) = \eta_{st,ise}(h_{s,st,in} - h_{s,st,ex,ise}) \tag{15}$$

Net steam cycle output is

$$W_{net,st} = W_{st} \cdot \eta_{st,mech} - \frac{W_{bfp}}{\eta_{bfp}} \tag{16}$$

where W_{bfp} the work done on boiler feed water pumps is given by

$$W_{bfp} = v_{fw,bfp,in}(h_{fw,bfp,ex} - h_{fw,bfp,in})$$

Net work output of combined cycle is

$$W_{cc} = W_{net,gt} + W_{net,st} \tag{17}$$

Efficiency of combined cycle is given by

$$\eta_{cc} = \frac{W_{cc}}{\dot{m}_f . LHV . \eta_{comb}} \tag{18}$$

ANALYSIS OF RESULTS

Performance evaluation of combined cycle with two-pressure HRSG and transpiration cooled gas turbine blades has been carried out for input data given in Table 1 & 2. In the considered four-stage gas

Table 2. Steam cycle parameters

Min. LP pinch point temperature difference = 5 K
Min. HP pinch point temperature difference = 5 K
HP steam pressure = 95 bar
HP superheat temperature = 843 K
LP steam pressure = 5 bar
LP superheat temperature = 473 K
HP steam turbine isentropic efficiency = 88 %
LP steam turbine isentropic efficiency = 92 %
Mechanical efficiency of steam turbines = 98.5 %
Condenser pressure = 0.05 bar
Overall efficiency of the boiler feed pump = 80 %
Blow-down loss factor = 0.02
HRSG effectiveness = 0.90
Minimum allowable stack temperature = 363 K
Pressure drop of gas in HRSG = 3 % of inlet pressure
Min. dryness fraction of steam at the outlet of steam turbines = 0.88
Pressure drop in HRSG, deaerator and condenser is neglected.
Heat loss in HRSG, turbines, condenser, and deaerator is neglected
Deaerator temperature ratio, $\theta_{dea} = \dfrac{T_{fw,dea,ex} - T_{cond}}{T_{fw,sat,hp} - T_{cond}} = 0.27$

turbine with compressor pressure ratio of 23:1, for the first stage blades transpiration cooling and for subsequent stages film/convection cooling has been used. For cooling of gas turbine blades compressed air bled from compressor is used as coolant. As TIT is increased the coolant requirement also increases in order to maintain the turbine blade temperature within the allowable limits and is depicted in Figure 3. As TIT increases from 1600 K to 1700 K the coolant requirement increases by 39 percent.

Figure 4 shows the effect of varying TIT on percentage heat recovery in HRSG. It is evident that as TIT increases, temperature of exhaust from gas turbine increases and also its enthalpy increases. Since the maximum steam generation temperature in HRSG is fixed hence the difference between the exhaust gas temperature and steam generation temperature increases with increasing TIT. This increase in temperature difference augments the heat recovery in hp section of HRSG. For an increase of TIT from 1600 K to 1700 K the exhaust gas temperature increases from 846 K to 907 K and heat recovery percentage in hp section increases by 6.4 percent.

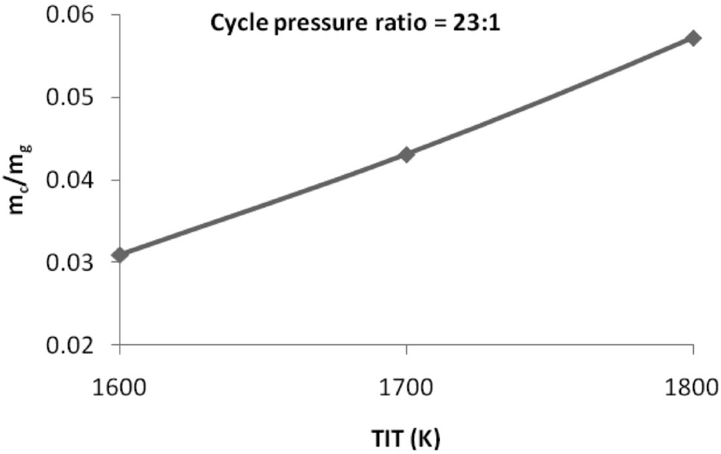

Figure 3. Effect of TIT on coolant to gas mass flow ratio

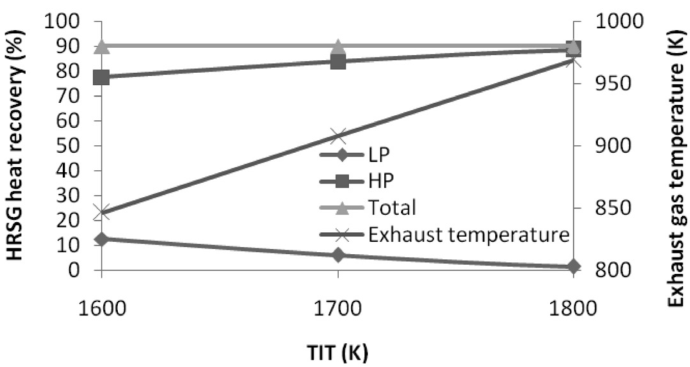

Figure 4. Effect of TIT on percentage heat recovery in HRSG and gas turbine exhaust temperature

Figure 5 shows the temperature heat transfer diagram of dual pressure HRSG for a TIT of 1600 K. For fixed values of HP and LP steam pressures, temperatures and LP pinch point, the heat recovery in HP section is greater than that in HP section for minimum values of HP pinch point and stack temperature. Further Figure 6 depicts the variation of HP pinch point temperature and stack temperature with TIT. With increase in TIT the stack temperature reduces due to increased utilization of heat available for steam generation. Stack temperature as well as HP pinch point temperature difference decreases with increase in TIT. Therefore

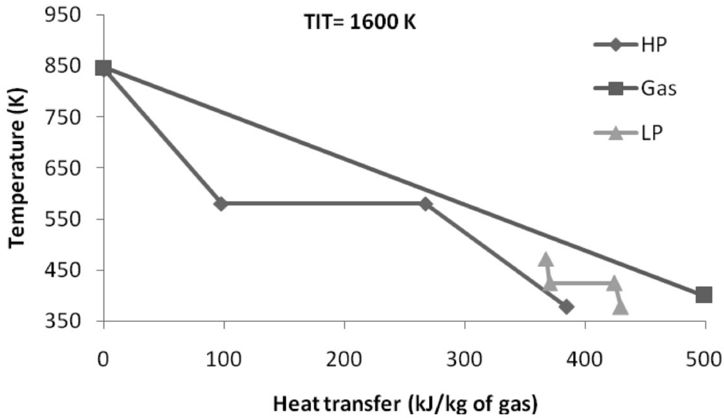

Figure 5. Temperature-heat transfer diagram of dual pressure HRSG for a TIT of 1600 K

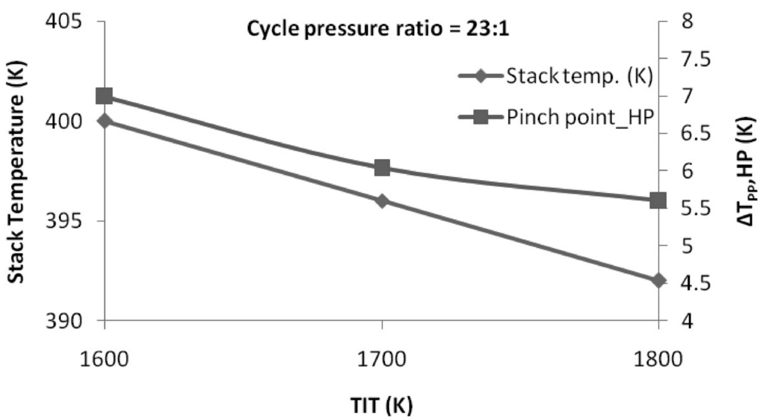

Figure 6. Effect of TIT on stack temperature and 'hp' pinch point temperature difference

the heat recovery in HP section increases and that in LP section decreases with increasing TIT in such a proportion that the total heat recovery percentage being constant at 90 percent. An increase in HP heat recovery and a reduction in LP heat recovery with increasing TIT results into increased HP steam generation rate and reduced LP steam generation rate respectively as shown in Figure 7. For an increase of TIT from 1600 K to 1700 K the HP steam generation per unit of gas flow increases from 0.13

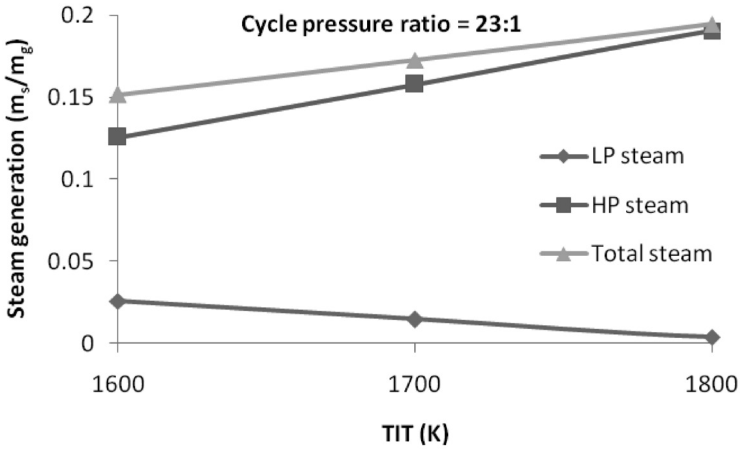

Figure 7. Effect of TIT on steam generation in HRSG

to 0.16 and the LP steam generation per unit of gas flow decreases from 0.026 to 0.015. Similarly for an increase of TIT from 1700 K to 1800 K the HP steam generation per unit of gas flow increases from 0.16 to 0.19 and the LP steam generation per unit of gas flow decreases from 0.015 to 0.004. With increase in TIT the increase in HP steam generation is greater than the decrease in LP steam generation rate and therefore total steam generation rate increases with increasing TIT.

Figure 8 depicts the variation of gas cycle efficiency, steam cycle efficiency and combined cycle efficiency with TIT. With increase of TIT, each of the topping, bottoming as well as combined cycle efficiency increases. For an increase of TIT from 1600 K to 1700 K the topping cycle i.e. gas turbine cycle efficiency increases by 0.32 percent and for an increase of TIT from 1700 K to 1800 K it increases by 0.20 percent. In higher TIT range of 1700 K to 1800 K the improvement in gas cycle efficiency is slower than that for the TIT range of 1600 K to 1700 K. A reason for this is the occurrence of dilution losses due to increased coolant requirement in first and second stage gas turbine blades where open loop cooling techniques i.e. transpiration cooling and film cooling respectively have been used. The dilution losses adversely affect the gas turbine work output. Another reason is the requirement of cooling of third and fourth stage turbine blades where closed loop cooling technique i.e. convection cooling has been used in which the coolant bled from the compressor does not participate in expansion process. The efficiency of bottoming

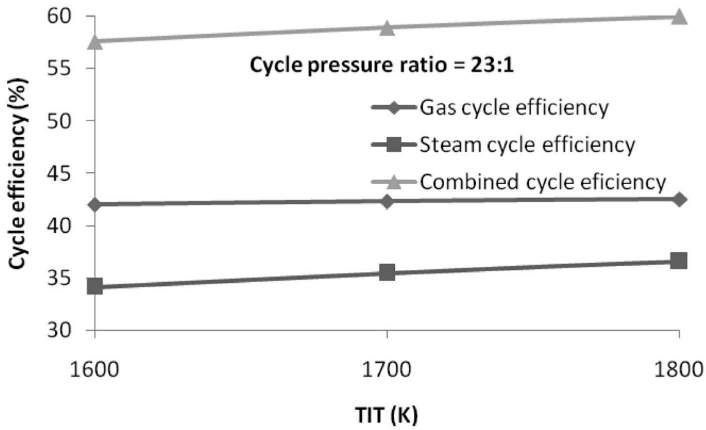

Figure 8. Effect of variation of TIT on cycle efficiency

cycle increases by 1.30 percent for increase in TIT from 1600 K to 1700 K and for further increase in TIT upto 1800 K the efficiency increases by 1.13 percent. This occurs because the exhaust gas temperature and hence its enthalpy increases with increasing TIT resulting into increased HP heat recovery and increased HP steam generation as depicted in Figs. 4 and 6 respectively. Due to improvement in both the topping cycle and bottoming cycle efficiency the overall efficiency of combined cycle also improves with increase in TIT. For an increase in TIT from 1600 K to 1700 K the combined cycle efficiency increases by 1.30 percent and for further increase in TIT from 1700 K to 1800 K the combined cycle efficiency increases by 1.07 percent. For an increase in TIT from 1600 K to 1800 K the combined cycle efficiency increases by 2.37 percent.

Figure 9 shows the variation of net specific work of topping cycle, bottoming cycle and combined cycle with TIT. The net specific work of topping cycle is higher than that of bottoming cycle at each TIT. At a TIT of 1600 K, the net specific work for the topping cycle is 457.62 kJ/kg and for the bottoming cycle is 169.53 kJ/kg, resulting into 627.15 kJ/kg of specific work for combined cycle. The net specific work increases continually with the increase in TIT for each of the topping cycle, bottoming cycle as well as combined cycle. For an increase of TIT from 1600 K to 1700 K the topping cycle specific work increases by 60.12 kJ/kg and for an increase of TIT from 1700 K to 1800 K it increases by 58.71 kJ/kg. Similarly for an increase of TIT from 1600 K to 1700 K the bottoming cycle specific work output increases by 32.84 kJ/kg and for an increase of

TIT from 1700 K to 1800 K it increases by 33.75. Due to improvement in specific work output of both the topping cycle and bottoming cycle the net specific work output of combined cycle also improves with increase in TIT and for an increase in TIT from 1600 K to 1800 K the combined specific work increases by 185.42 kJ/kg.

Figure 10 shows the variation of specific fuel consumption of topping cycle and combined cycle with TIT. As expected the specific fuel consumption of both the gas turbine cycle as well as of combined cycle reduces with TIT and the specific fuel consumption of combined cycle is lower than that of gas turbine cycle. For an increase of TIT from 1600 K to 1800 K the specific fuel consumption of combined cycle decreases by 1.24 percent.

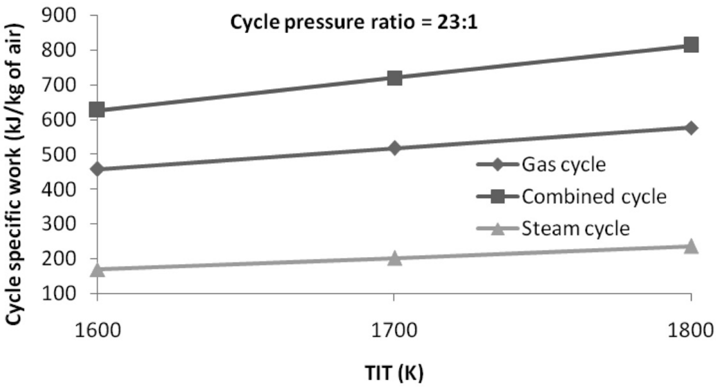

Figure 9. Effect of TIT on net specific work output

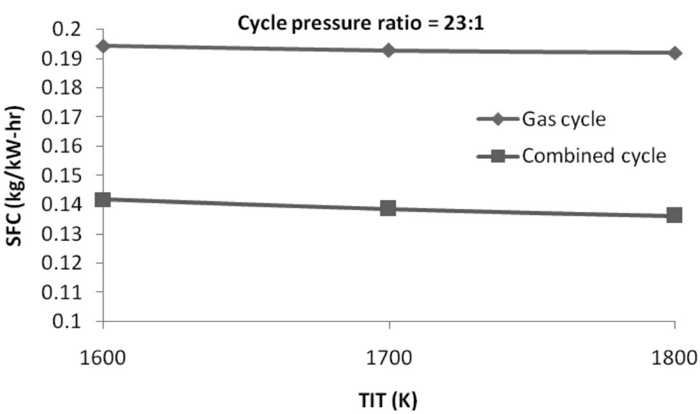

Figure 10. Variation of specific fuel consumption with TIT

CONCLUSION

A general model of gas-steam combined cycle with transpiration cooled turbine blades has been developed. The computer code developed on the basis of modeling has been used to compare the transpiration cooled combined cycle performance for varying TIT. Coolant requirements, steam generation rate in HRSG, heat recovery in HRSG, cycle efficiencies, cycle specific work output and specific fuel consumption of the combined cycle are compared for TIT at a fixed pressure ratio of 23:1. The study shows that the coolant required for cooling gas turbine blades increases with increasing TIT. For a fixed maximum steam generation temperature in HRSG the difference between the exhaust gas temperature and steam generation temperature increases with increasing TIT which augments the heat recovery in HRSG. For an increase of TIT from 1600 K to 1700 K the exhaust gas temperature increases from 846 K to 907 K and heat recovery percentage in hp section increases by 6.4 percent. Due to increased heat recovery in HRSG the total steam generation rate increases with increasing TIT. All the above effects improve the efficiency and specific work of each of the topping, bottoming as well as combined cycle with increase of TIT. For an increase in TIT from 1600 K to 1800 K the combined cycle efficiency increases by 2.37 percent and the combined specific work increases by 185.42 kJ/kg. It is concluded that by using transpiration cooling higher TIT's can be used which enhances the overall performance of combined cycle by improvement in gas cycle as well as in steam cycle by diminishing the temperature differences in the heat recovery steam generator and by lowering the stack temperature.

References

1. Sanjay, Singh, O., Prasad, B.N., "Influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle," Elsevier, *Applied Thermal Engineering*, Volume 28, 2008, pp. 2315-2326.

2. Sanjay, Singh, O., Prasad, B.N., "Thermodynamic modeling and simulation of advanced combined cycle for performance enhancement," *J. Power and Energy Proc. IMechE*, Vol. 222 Part A:, 2008, pp. 541-555.

3. Sanjay Kumar and Singh, O., "Thermodynamic evaluation of different gas turbine blade cooling techniques," IEEE Xplore Conference Proceedings-Second International Conference on Thermal Issues in Emerging Technologies, 2008. ThETA '08'; 237-244 (Digital Object Identifier 10.1109/THETA.2008.5167172).

4. Polezhaev, J., "The transpiration cooling for blades of high temperatures gas turbine," *Pergamon, Enegy Convers. Mgmt.*, Volume 38, 1997, pp. 1123-1133.

5. Horlock, J.H., Watson, D.T., Jones T.V., "Limitations on gas turbine performance imposed by large turbine cooling flows," *ASME Journal of Engineering For Gas Turbines and Power*, Volume 123, 2001, pp. 487-494.

6. Kumar, S. and Singh, O., "Thermodynamic performance evaluation of gas turbine cycle with transpiration cooling of blades using air vis-à-vis steam," Proc. IMechE, Part A: *J. Power and Energy*, 224 (A8), 2009, 1039-1047. DOI 10.1243/09576509JPE964

7. Srinivas, T., "Thermodynamic modeling and optimization of multi-pressure heat recovery steam generator in combined power cycle," *Journal of Scientific and Industrial Research*, Volume 67, 2008, pp. 827-834.

8. Bassily, A.M., "Modelling and numerical optimization of dual- and triple-pressure combined cycles," Proc. IMechE, Part A: *J. Power and Energy,* 218, 2004, 97-109.

9. Chin, W.W. and El-Masri, M.A., "Exergy analysis of combined cycles: part 2 – Analysis and optimization of two-pressure steam bottoming cycles," *ASME Journal of Engineering For Gas Turbines and Power*, Volume 109, 1987, pp. 237-243.

10. Kumar, S. and Singh, O., "Performance evaluation of transpiration cooled gas turbine for different coolants and permissible blade temperatures considering the effect of radiation," Proc. IMechE, Part A: *J. Power and Energy*, 225, 1156-1165. DOI 10.1177/0957650911404305.

ABOUT THE AUTHORS

Sanjay Kumar, is working as Teacher Fellow at Harcourt Butler Technological Institute, Kanpur, India. He has been graduated from BIET, Jhansi and postgraduated from HBTI, Kanpur. He has been awarded the Fellowship by Gautam Buddh Technical University, Lucknow, India and has been teaching for last five years in the Department of Mechanical Engg. at HBTI, Kanpur. He is a member of The Institution of Engineers (India). His area of interest is thermal power engineering and design. E-mail: sanjaykumarkanpur@gmail.com.

Onkar Singh, Ph.D., is Professor and Head of the Mechanical Engineering Department at Harcourt Butler Technological Institute, Kanpur, India. He has been graduated from HBTI, Kanpur and has completed his postgraduation and Ph.D. from MNNIT, Allahabad. Dr. Singh is a fellow of The Institution of Engineers (India). Dr. Singh specializes in thermal power engineering. E-mail: onkpar@rediffmail.com.