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Comparative performance analysis of cogeneration gas turbine cycle for different blade cooling means

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ABSTRACT

The paper compares 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. Cogeneration cycle performance is evaluated using coolant flow requirements, plant specific work, fuel utilisation efficiency, power-to-heat-ratio, which are function of compressor pressure ratio and turbine inlet temperature, and process steam drum pressure. The maximum and minimum values of power-to-heat ratio are found with steam internal convection cooling and air internal convection cooling respectively whereas maximum and minimum values of fuel utilisation efficiency are found with steam internal convection cooling and closed loop steam cooling. The analysis is useful for power plant designers to select the optimum compressor pressure ratio, turbine inlet temperature, fuel utilisation efficiency, power-to-heat ratio, and appropriate cooling means for a specified value of plant specific work and process heating requirement.

1. Introduction

Combined heat and power (CHP) plants are fast becoming popular worldwide due to their capability of meeting both power and heating requirements. Growing number of CHP plants has made it inevitable to make them efficient, environment friendly and sustainable. One of the main targets in the development of combined heat and power (CHP) plants in the 21st century will be to decrease CO_2 emissions [1]. Reduction in CO_2 emissions may be achieved by replacement of fuels having lower C–H ratio fuels, such as coal by natural gas, which has resulted in a worldwide shift towards the use of natural gas in combined cycle and cogeneration plants. Natural gas offers the advantages of increase in overall efficiency of generation of electric energy and heat, due to effectiveness in utilisation of chemical energy of the fuel and reduced harmful impact of power plants on the environment through reduced emission of NO_x, particulate and GHG.

Japan and USA has developed gas turbines for combined cycles and co-generation application through AGT J-100 “Moonlight” project and ATS program respectively to achieve combined cycle efficiency greater than 60% [2] and lower GHG emission.

In a gas/steam combined power and heat plants, steam generated in HRSG is utilised for process heating. Cogeneration cycle refers to the specific arrangement when entire steam generated in HRSG is used for process work and there is no steam turbine for power generation.

Literature indicates significant work done in the area of cogeneration cycles by F.F. Huang and L. Wang [3], R.P. Allen and J.M. Kovacik [4], J.W. Baughan and R.A. Kerwin [5], I.G. Rice [6], R. Bhargava, A. Peretto [7], F.S. Basto, H.P Blanco [8], M. Bianchi, G.N. Montenegro, A. di Peretto [9], Andreas Poullikkas [10], B. Zaporowski, R. Szczerbowski [11], R. Yokoyama, K. Ito [12], T. Korakianitis et al. [13], S. Pelster et al. [14], etc.

In the present work the latest gas turbine MS9001 has been chosen as the topping cycle and combined with a dual pressure (2P) steam cycle as the bottoming cycle. The parametric analysis of the effect of blade cooling means on gas turbine based cogeneration cycle has been presented.

2. Cogeneration cycle configuration

Fig. 1 shows schematic of the proposed gas turbine cogeneration cycle. The exhaust of gas turbine is used to generate steam in the heat recovery steam generator (HRSG). The HRSG consists of a coolant steam generator (CSG) and a process steam genera-

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Nomenclature

A	area	m^2
c_p	specific heat	$kJ\,kg^{-1}\,K^{-1}$
F_{sa}	correction factor to account actual blade surface	
gt	gas turbine	
h	specific enthalpy	$kJ\,kg^{-1}$
$h_{f,\text{sat}}$	enthalpy of saturated liquid	$kJ\,kg^{-1}$
$\bar{h}$	convective heat transfer coefficient	$W\,m^{-2}\,K^{-1}$
ΔH_r	lower heating value	$kJ\,kg^{-1}\,K^{-1}$
$\dot{m}$	mass flow rate	$kg\,s^{-1}$
M	Mach number	
r_p	cycle pressure ratio	
R	gas constant	$kJ\,kg^{-1}\,K^{-1}$
R_c	$\frac{(T_{g,i}-T_b)\cdot c_{p,g}}{\varepsilon(T_b-T_{c,i})\cdot c_{p,c}}$	
$(R_{c,\text{film}})$	$\frac{(T_{g,i}-T_b)\cdot c_{p,g}\cdot(1-\eta_{\text{iso,film}})}{\varepsilon\cdot(T_b-T_{c,i})\cdot c_{p,c}}$	
$(R_{c,\text{trans}})$	$\frac{(T_{g,i}-T_b)\cdot c_{p,g}\cdot(1-(\eta_{\text{iso,trans}}))}{\varepsilon\cdot(T_b-T_{c,i})\cdot c_{p,c}}$	
p	pressure	bar
q	heat added	W
s	entropy	$kJ\,kg^{-1}\,K^{-1}$
sfc	specific fuel consumption	
t	pitch of blade	m
T	temperature	K
TIT	turbine inlet temperature = combustor exit temperature	K
W	specific work	$kJ\,kg^{-1}$
Y	a constant	

Greek symbols

υ	specific volume	$m^3\,kg^{-1}$
ρ	density	$kg\,m^{-3}$
λ	ratio	
γ	ratio of specific heat at constant pressure and constant volume	
η	efficiency	%
ε	effectiveness = $\frac{Tc.e-Tc,i}{Tb-Tc,i}$	%

Subscripts

a	air, ambient
b	blade
c	compressor, coolant
comb	combustor
cond	condensate
d/a	deaerator
e	exit
eco	economiser
f	fuel
film	film cooling
g	gas
gen	alternator
gt	gas turbine
hrsg	heat recovery steam generator
i	inlet, stage of compressor
in	inlet
j	coolant bleed points
iso	isothermal
net	difference between two values
p	pressure, pump
plant	combined-power-heat plant
pp	pinch point
pt	polytropic
s	steam
sat	saturated
sc	steam coolant
st	steam cycle
sh	superheated
trans	transpiration
w	water

ACRONYM

2P	dual pressure HRSG/bottoming cycle configuration
AFC	air film cooling
AICC	air internal convection cooling
ATC	air transpiration cooling
BD	blow down
BFP	boiler feed pump
C	compressor
CC	combustion chamber
CLSC	closed loop steam cooling
CSD	coolant steam drum
GT	gas turbine
PSD	process-heating steam drum
PSG	process-steam generator
SICC	steam internal convection cooling
SFC	steam film cooling
STC	Steam transpiration cooling

tor (PSG). The CSG is needed to generate coolant steam only when the gas turbine blades are to be cooled by steam; otherwise the gas turbine exhaust gases are directly fed to the PSG. Hot gas path components of the gas turbine need cooling. The different cooling means and methods analysed are shown in Fig. 2. In open loop cooling, the coolant after cooling the blades mixes with the hot gas, while in closed loop cooling, the exiting coolant pre-heats fuel (natural gas) before it burns in the combustor. The cooling methods are of internal convection, film or transpiration types. The cooling medium is air or steam.

Thermodynamic modeling of cogeneration cycle components has done for performance prediction. Analysis has been done based upon the dependent parameters like coolant flow requirements, plant specific work, fuel utilisation efficiency (FUE), power-to-heat-ratio (PHR), as a function of independent parameters such as compressor pressure ratio ($r_{p,c}$), and turbine inlet temperature (TIT), and steam drum pressure.

3. Thermodynamic modeling and performance of cogeneration cycle

Thermodynamic modeling of the components of the cogeneration cycle using different means of cooling has been carried out and the governing equations to predict its performance are developed accordingly.

The cogeneration cycle performance parameters include $W_{\text{gt,net}}$, η_{plant}, FUE, PHR and W_{plant}, which are expressed as

$$W_{\text{gt,net}} = W_{\text{gt}} - \frac{W_c}{\eta_m} \tag{1}$$

$$W_{\text{plant}} = (W_{\text{gt,net}}) \cdot \eta_{\text{gen}} \tag{2}$$

$$\eta_{\text{plant}} = \frac{W_{\text{plant}}}{Q} = \frac{W_{\text{plant}}}{\dot{m}_f \cdot \Delta H_r} \tag{3}$$

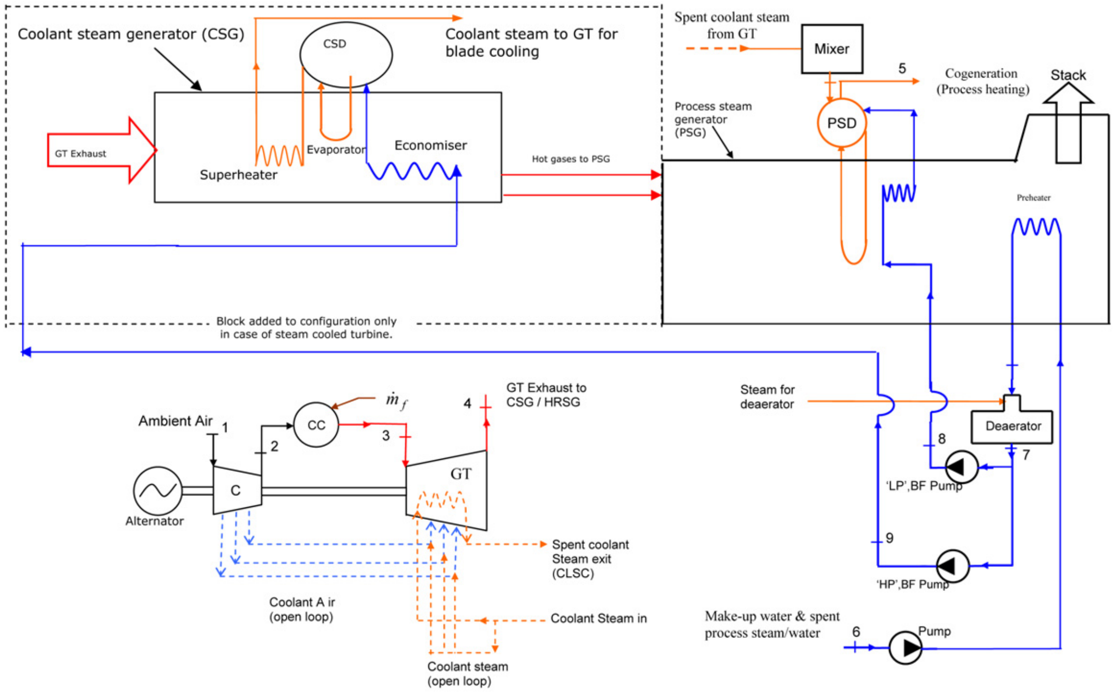


Fig. 1. Proposed schematic of MS9001 based cogeneration cycle configuration.

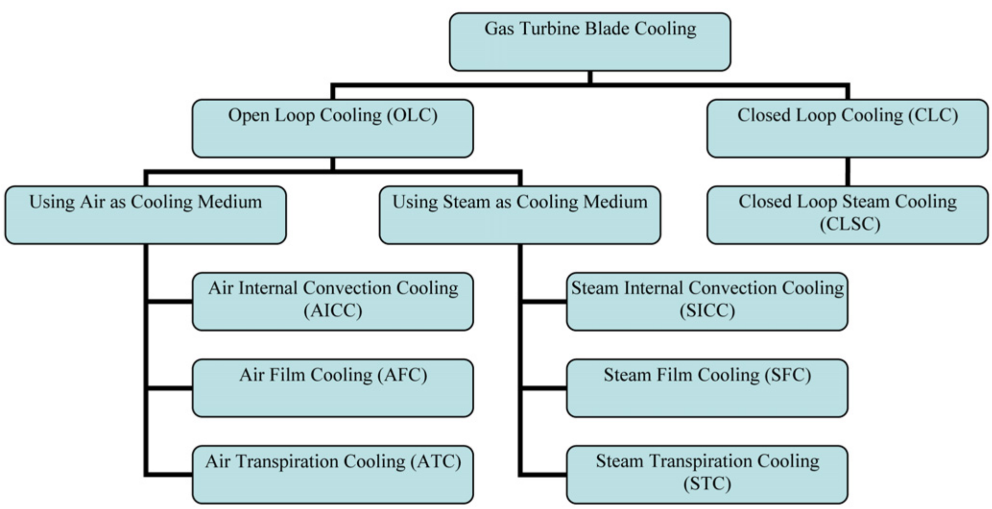


Fig. 2. Classification of blades cooling means analysed.

Fuel utilisation efficiency (FUE) of a plant is the ratio of all useful energy (power and process heat) extracted from the system to the input fuel energy given by

$$\mathrm{FUE} = (W_{\mathrm{plant}} + Q_p)/\dot{m}_f \cdot (LCV)_f \tag{4}$$

where, Q_p = heat energy available in HRSG for process heating and given by

$$Q_p = m_{s,\mathrm{psd}} * (h_{s,\mathrm{out,psd}} - h_{w,6}) \tag{5}$$

It is desirable to enhance the value of FUE.

The power-to-heat-ratio (PHR) is another important performance parameter for the cogeneration plant

$$\mathrm{PHR} = W_{\mathrm{plant}}/Q_p \tag{6}$$

The value of power-to-heat-ratio depends upon the process heat demand. Its value may be high or low, and the other parameters of cogeneration plants are selected accordingly.

3.1. Gas properties

In the present gas model, it is assumed that specific heat of gas varies with temperature only in the form of polynomials given as:

$$c_p(T) = a + bT + cT^2 + dT^3 + \cdots \tag{7}$$

where, a, b, c and d are coefficient of polynomials, values of which are taken from the work of Touloukian and Tadash [15]. Enthalpy of gas is given as:

$$h = \int_{T_a}^{T} c_p(T)\,dT \tag{8}$$

The ambient air entering the compressor is assumed to be at 1 bar, 288 K, 50% relative humidity, enthalpy assigned zero value. The composition by weight of natural gas as fuel is taken to be CH_4 = 86.21%, C_2H_6 = 7.20%, CO_2 = 5.56%, N_2 = 1.03%.

3.2. Compressor

Axial flow type compressor is used in the cycle. The thermodynamic losses in compressor are incorporated in the model by introducing the concept of polytropic efficiency. Temperature and pressure variation in the compressor is related by the expression

$$\frac{dT}{T} = \left[\frac{R}{\eta_{\mathrm{pt},c} c_{p,a}}\right]\frac{dp}{p} \tag{9}$$

Using mass and energy balance across control volume of compressor, the compressor work is calculated as follows:

$$\dot{m}_{c,i} = \dot{m}_{c.e} + \sum \dot{m}_{c,j} \tag{10}$$

$$W_c = \dot{m}_e h_e + \sum \dot{m}_{c,j} h_j \tag{11}$$

In case of closed loop cooling due to absence of air-bleeding from compressor, $\sum m_{c,j} = 0$.

Eqs. (7) to (11) are used for estimating the compressor work.

3.3. Combustor

Combustor model includes the consideration of losses in combustor and estimation of fuel requirement for achieving specified temperature at its exit. The combustor losses considered are due to incomplete fuel combustion and pressure loss. Loss due to incomplete fuel combustion is taken care by the use of the concept of combustion efficiency, while pressure loss is accounted by taking percentage pressure drop of the combustor inlet pressure. Fuel requirement for attaining the desired TIT is obtained from the mass and energy balances across the combustor. The mass balance of the combustor is given as

$$\dot{m}_e = \dot{m}_i + \dot{m}_f \tag{12}$$

and energy balance of combustor is given as

$$\dot{m}_f \cdot \Delta H_r \cdot \eta_{\mathrm{comb}} = \dot{m}_e \cdot h_e - \dot{m}_i \cdot h_i \tag{13}$$

3.4. Cooled gas turbine

Large number of models for estimating gas turbine blade coolant requirement and related studies have been reported in the literature [16–21]. Blade cooling has been an area challenging the development of gas turbines since efforts started to increase the efficiency of the cycle by increasing TIT. Technical report submitted by Ainley [16] and subsequent work of Goldstein et al. [17] were some of the related works done in this field. Ainley analysed the design of internal-convection air-cooled blades and reported the cooling effectiveness, wherein the proposed equations gave quite good approximations of the temperature distribution based on his full solutions. He found that the blade temperature is by no means constant; the maximum blade temperature occurs at the end of the blade.

Goldstein et al. have discussed the effectiveness of film cooling of blades. These [16,17] works have formed the basis of the subsequent works of Torbidoni et al. [18], Torbidoni et al. [19], and Ghigliazza et al. [20]. Torbidoni et al. in their work have proposed a blade cooling model and evaluated the performance of cooled gas turbines [18]. Their work concludes that only marginal gain in performance can be observed when TIT is increased much further, benefits of increase in TIT would only come about with improvement in blade material technology and improved heat transfer methods to restrict coolant flow. Torbidoni et al. [19] in their work have developed a semi-empirical equation and further developed the cooling model to give more accurate coolant flow prediction.

The above works available in the literature [16–21] have formed the basis of the present work to take steam as coolant in all cooling methods namely internal convection, film, and transpiration and closed loop steam cooling. The cooled gas turbine model used in the cogeneration cycle here is the refined version of Louis et al. [21] and Horlock et al. [22] cooling models and is detailed in author's earlier work [23].

Figs. 3(a) and (b) represent the internal convection cooling model based cooling means namely AICC, SICC, and CLSC. The gas turbine blades are treated as heat exchangers operating at uniform temperature and the coolant exit temperature given as a function of heat exchanger effectiveness, ε.

Figs. 3(c) and (d) represent the film and transpiration cooling model based cooling means namely AFC, ATC, SFC, and STC. To account for the reduced heat transfer rate from hot gas to blades, the term isothermal effectiveness [21] is considered. It is defined as the ratio of difference between heat transfer fluxes with and without film/transpiration cooling to an isothermal wall. In the case of film cooling, the coolant exits from the leading edge of blade and a film is formed over the blade surface, which reduces the heat transfer from the hot gas to the blade surface. Thus the cooling is due to the joint action of internal convection and film cooling. While in transpiration cooling, numerous small holes are made on the surfaces of the blades forming a porous wall through which coolant comes out and forms an effective thick coolant film on the surface resulting in reduction in heat transfer from hot gases to blades. In this case also, cooling is due to joint action of internal convection and cooling film that has transpired.

Coolant flow requirement in case of internal convection cooling, film cooling and transpiration cooling is estimated from respective cooling models. The expression [23] for coolant mass flow rate for various blade-cooling methods are rewritten:

Air internal convection cooling:

$$\left(\frac{\dot{m}_c}{\dot{m}_g}\right)_{\mathrm{AICC/SICC}} = 0.0156[R_c]_{\mathrm{AICC/SICC}} \tag{14}$$

where

$$[R_c]_{\mathrm{AICC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g}}{\varepsilon(T_b - T_{c,i}) \cdot c_{p,c}} \tag{14a}$$

$$[R_c]_{\mathrm{SICC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g}}{\varepsilon(T_b - T_{c,i}) \cdot c_{p,c}} \tag{14b}$$

$c_{p,c}$, $T_{c,i}$ are for coolant air or steam as the case may be.

Film cooling:

$$\left(\frac{\dot{m}_c}{\dot{m}_g}\right)_{\mathrm{AFC/SFC}} = 0.0156[R_{c,\mathrm{film}}]_{\mathrm{AFC/SFC}} \tag{15}$$

where

$$[R_{c,\mathrm{film}}]_{\mathrm{AFC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g} \cdot (1 - \eta_{\mathrm{iso,film}})}{\varepsilon \cdot (T_b - T_{c,i}) \cdot c_{p,c}} \tag{15a}$$

$$[R_{c,\mathrm{film}}]_{\mathrm{SFC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g} \cdot (1 - \eta_{\mathrm{iso,film}})}{\varepsilon \cdot (T_b - T_{c,i}) \cdot c_{p,c}}$$

$$\eta_{\mathrm{iso,film}} = 0.4 \tag{15b}$$

and $c_{p,c}$, $T_{c,i}$ are for coolant air or steam as the case may be.

Transpiration cooling:

$$\left(\frac{\dot{m}_c}{\dot{m}_g}\right)_{\mathrm{ATC,STC}} = 0.0156[R_{c,\mathrm{trans}}]_{\mathrm{ATC,STC}} \tag{16}$$

where

$$[R_{c,\mathrm{trans}}]_{\mathrm{ATC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g} \cdot (1 - \eta_{\mathrm{iso,trans}})}{\varepsilon \cdot (T_b - T_{c,i}) \cdot c_{p,c}} \tag{16a}$$

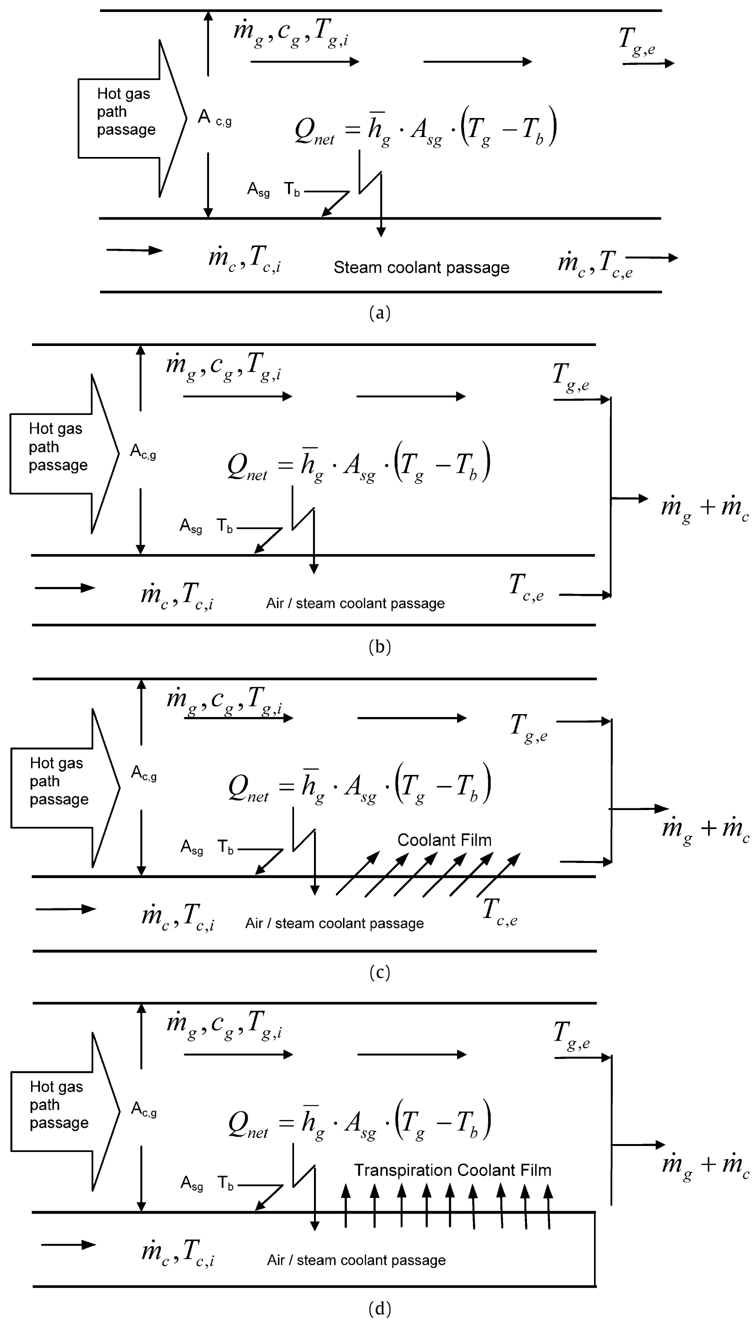


Fig. 3. Different blade cooling models in case of (a) Closed loop steam cooling, (b) Internal convection open loop air/steam, (c) Film air/steam cooling, (d) Transpiration air/steam cooling.

$$[R_{c,\text{trans}}]_{\text{SFC}} = \frac{(T_{g,i} - T_b) \cdot c_{p,g} \cdot (1 - \eta_{\text{iso,trans}})}{\varepsilon \cdot (T_b - T_{c,i}) \cdot c_{p,c}}$$

$$\eta_{\text{iso,trans}} = 0.5 \tag{16b}$$

and $c_{p,c}$, $T_{c,i}$ are for coolant air or steam as the case may be.

Closed loop steam cooling:

$$\left(\frac{\dot{m}_c}{\dot{m}_g}\right)_{\text{CLSC}} = 0.0156[R_c]_{\text{CLSC}} \tag{17}$$

Quantum of gas turbine output is estimated based on the assumptions [23] discussed ahead.

Each row of the turbine is treated as an expander whose walls continuously extract work. Expansion process in turbine is polytropic and the polytropic efficiency takes care of losses in the expansion process. Temperature at any point in expansion path in turbine is determined by

$$\frac{dT_g}{T_g} = \left[\left\{\frac{p + dp}{p}\right\}^{R \cdot \eta_{\text{pt}}/c_{p,g}} - 1\right] \tag{18}$$

In case of open loop cooling the mixing of coolant with expanding gases causes a drop in stagnation temperature of expanding gases, which will be absent in case of closed loop cooling. Mixing of coolant with expanding gases also causes a stagnation pressure drop, which has been considered in the model (Fig. 4). The loss of irreversibility due to mixing of coolant with hot gas in open loop cooling for a row given by Shapiro [24] is expressed as:

$$\frac{dp_{\text{loss}}}{p_0} = -\frac{\dot{m}_c}{\dot{m}_g} \cdot \gamma \cdot M^2 \cdot Y \tag{19}$$

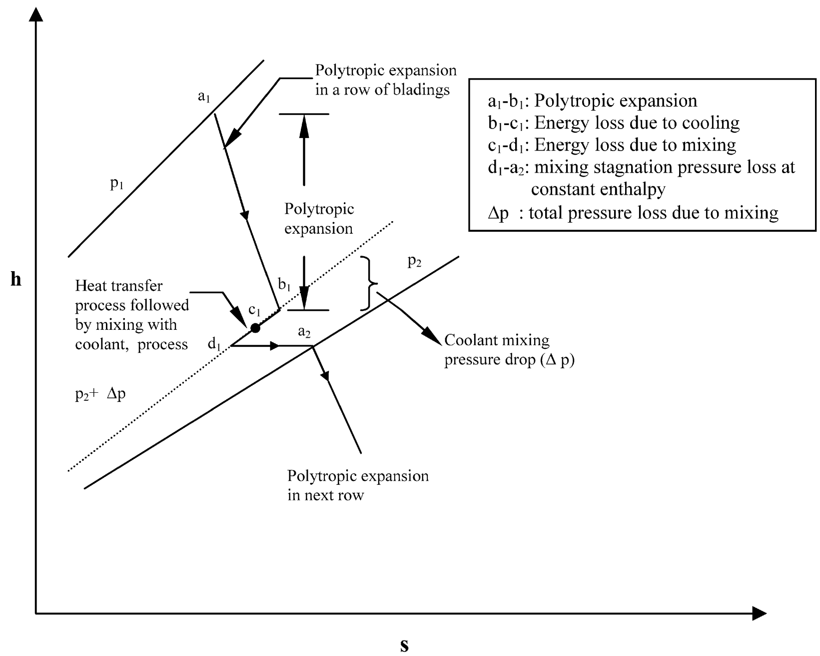


Fig. 4. Expansion path in cooled turbine row.

where, the average value of γ, M and Y is taken to be 1.4, 0.5 and 0.3 respectively [21,24].

Gas turbine work output is given by Eq. (20), under:

$$W_{\text{gt}} = \sum^{\text{row}} \dot{m}_{g,i} \cdot (h_{g,a_1} - h_{g,b_1})_{\text{cooled}} + \sum^{\text{row}} \dot{m}_{g,i} \cdot (h_{g,i} - h_{g,e})_{\text{uncooled}} \tag{20}$$

In case of closed loop cooling, the gas turbine work is given as:

$$W_{\text{gt}} = \dot{m}_{g,i} \cdot (h_{g,i} - h_{g,e}) + \sum \dot{m}_c \cdot (h_{c,i} - h_{c,e}) \tag{21}$$

where, 'i' and 'e' denote the inlet and exit condition respectively of a row or whole turbine as the case may be.

3.5. Heat recovery steam generator

Heat recovery steam generator is the integral part of cogeneration cycle in view of process heating requirements. Steam generated in the heat recovery steam generator meets the process heating requirements. Apart from meeting the process-heating requirements the steam generated can also be used as coolant for gas turbine stage cooling. In the cogeneration cycle an unfired HRSG has been considered. Depending upon the steam requirement, the HRSG is considered to be generating steam at dual pressure or single pressure. In an unfired HRSG (Fig. 1), gas turbine exhaust is used for generating coolant steam in coolant steam generator (CSG) for steam-cooled turbine, and process steam in the process steam generator (PSG). The selection of coolant steam pressure is based on the compressor pressure ratio. Ref. [25] outlines the detailed procedure of analysis of steam cycle. The selection of pressure and temperature of process heating steam is based on process-heating requirement (Table 1) [26–30].

3.6. Deaerator

Thermodynamic modeling of deaerator is done neglecting undercooling of feedwater at the exit of deaerator. The amount of steam extracted for deaeration is given by the mass and energy balance equation written under:

Table 1
Input data for analysis [20–24].

Parameter		Symbol	Unit
Gas properties:		$c_p = f(T)$	kJ/kg K
		Enthalpy $h = \int c_p(T)\,dT$	kJ/kg
Compressor	i.	Polytropic efficiency (η_{pc}) = 92.0	%
	ii.	Mechanical efficiency (η_m) = 98.5	%
Combustor	i.	Combustor efficiency (η_{comb}) = 99.5	%
	ii.	Pressure loss (p_{loss}) = 2.0% of entry pressure	%
	iii.	Lower heating value (ΔH_r) = 42.0	MJ/kg
Gas turbine	i.	Polytropic efficiency (η_{pt}) = 92.0	%
	ii.	Exhaust pressure = 1.08	bar
	iii.	Exhaust hood temperature loss = 4	K
	iv.	Turbine blade temperature = 1123	K
HRSG	i.	Effectiveness = 98.0	%
	ii.	Pressure loss = 10% of entry pressure	%
	iii.	Stack (minimum temperature) = 363.0	K
	iv.	Process steam / 'lp' pressure = 6.0	bar
	v.	Gas/steam approach temperature difference = 20.0	K
	vi.	Pinch-point temperature difference = 10.0	K
	vii.	Pressure of blade coolant steam = 35	bar
Alternator		Alternator efficiency = 98.5	%

$$\dot{m}_{s,d/a,i} \cdot h_{s,d/ai} + (\dot{m}_s - \dot{m}_{s,d/a,i}) h_{w,6} = \dot{m}_s \cdot h_{w,d/a,e} \tag{22}$$

where, $\dot{m}_s$ is in liquid phase.

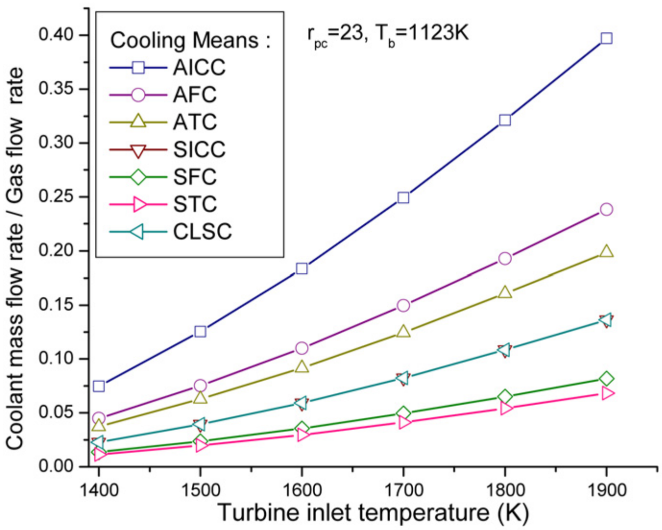
3.7. Feed pump

The mass and energy balance gives the feed-pump work requirement as under:

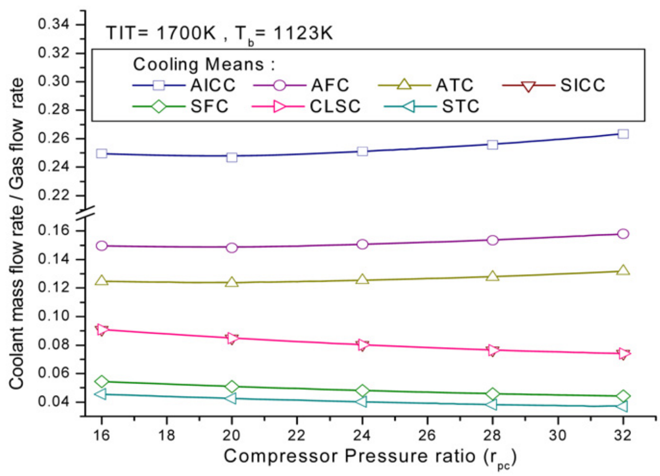
$$W_p = \sum \upsilon_{w,i} \cdot (p_e - p_i) \tag{23}$$

4. Results and discussion

The effect of the operating parameters on cycle performance for different means of cooling with air and steam has been presented and discussed.



(a)



(b)

Fig. 5. Effect of cycle parameters on coolant flow rate. (a) Effect of TIT, (b) Effect of r_{pc}.

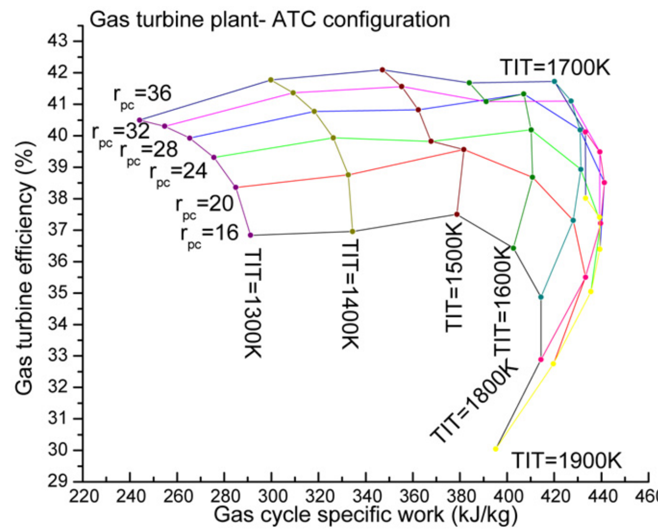


Fig. 6. Gas turbine efficiency versus gas cycle specific work for different $r_{p,c}$ and TIT in case of gas turbine–air transpiration cooling configuration.

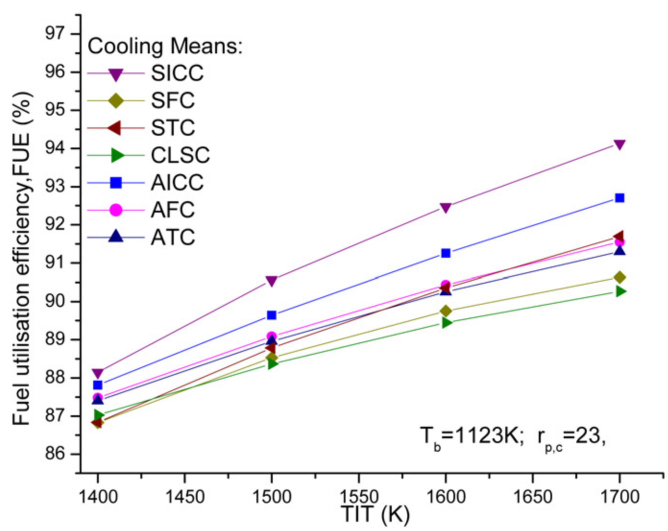


Fig. 7. TIT versus fuel utilisation efficiency for various blade cooling means.

4.1. Effect of cooling means on coolant flow with TIT and $r_{p,c}$

4.1.1. Variation with TIT

Fig. 5(a) shows the effect of TIT on blade coolant requirement for various cooling means for a compressor pressure ratio of 23. 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. 5(a) 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. The values of coolant flow requirements calculated using Eqs. (14), (15), (16), and (17), compare well with those of Horlock et al. [22] with variation in the range of 2.5 to 3.5% only.

4.1.2. Variation with $r_{p,c}$

Fig. 5(b) shows the requirements of coolant flow rates for different means of blade cooling with variation in compressor pressure ratio at fixed value of TIT of 1700 K. It can be seen in this figure that variation in cooling flow rates with change in $r_{p,c}$ for all cooling means is not appreciable, rather monotonous.

4.2. Effect of variation of $r_{p,c}$ and TIT on gas turbine performance

Fig. 6 shows the design monogram of gas turbine power plant using ATC means. The behaviour of parameters such as plant efficiency and specific work for various $r_{p,c}$ and TIT suggests that the performance is limited by cooling penalties beyond TIT of 1600 K. Further results suggest the best-suited configuration for cogeneration plant.

4.3. Effect of cooling means on FUE with variation in TIT and $r_{p,c}$

Effect of TIT on fuel utilisation efficiency of the gas turbine based cogeneration plant for various means of cooling is shown in Fig. 7 for $r_{p,c} = 23$ and $T_b = 1123$ K. The results show that the value of FUE increases with TIT for all types of cooling. The

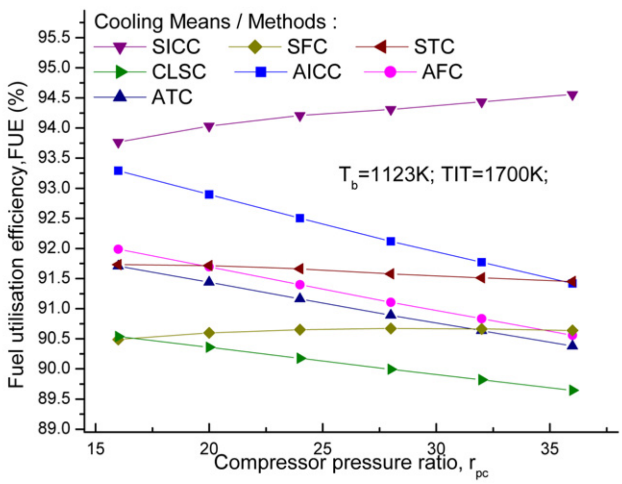


Fig. 8. Effect of $r_{p,c}$ on fuel utilisation efficiency for various blade cooling means.

highest value of FUE is exhibited by SICC scheme, while the lowest value of FUE is offered by closed loop steam cooling (CLSC). This is attributed to the fact that SICC offers higher gas turbine specific work as compared to all the air-cooling means for all the values of TIT considered. The heating value in HRSG also gets increased due to addition of coolant steam in gas turbine (exhaust). The fuel requirement is the same in all the steam cooling means, so FUE has higher value in the case of SICC. The next to the highest value of FUE is exhibited by AICC. This interesting behaviour may be explained by the fact that in AICC means, the fuel requirement increases at a lower rate with increase in TIT as compared to other cooling means due to larger increase in amount of coolant bled from compressor. Though, the gas turbine specific work is lesser for AICC as compared to other cooling means, the thermal energy remains the same for all air-cooling means, resulting in a significantly higher value of FUE. The results suggest that if the value of FUE is the only criteria for selection, then among air-cooling means, AICC is the best choice for co-generation, as it is desirable to have the maximum possible value of FUE.

Fig. 8 depicts the effect of $r_{p,c}$ on FUE for various means of cooling for gas turbine based cogeneration plant at TIT = 1700 K and $T_b = 1123$ K. In all air-cooling means, at all values of TIT, with the increase of $r_{p,c}$, the value of FUE decreases. This is because of the fact that at a certain value of TIT, the gas cycle exhaust temperature decreases significantly with the increase of $r_{p,c}$ and, the gas cycle specific work increases marginally, resulting in less thermal energy available in HRSG for process work. Though, there is a decrease in fuel requirement with increase in $r_{p,c}$, the combined effect decreases the value of FUE.

Open and closed loop steam cooling means behave differently with FUE. In open loop steam cooling means, the value of FUE increases, while in closed loop steam cooling it decreases with the increase of $r_{p,c}$. This is attributed to the fact that the gas cycle specific work increases at higher rate in open loop steam cooling compared to air-cooling. As a result combined effect increases the value of FUE with increase in the value of $r_{p,c}$ in closed loop steam cooling.

4.4. Effect of cooling means on PHR with variation $r_{p,c}$ and TIT

The effect of variation of $r_{p,c}$ at TIT = 1700 K and $T_b = 1123$ K on PHR for gas turbine based cogeneration plant is shown in Fig. 9. For all cooling means PHR increases with $r_{p,c}$ at a given value of TIT. This is because of the fact that with increase of $r_{p,c}$, the heat energy available in HRSG for process heating reduces and the ratio of power-to-heat increases.

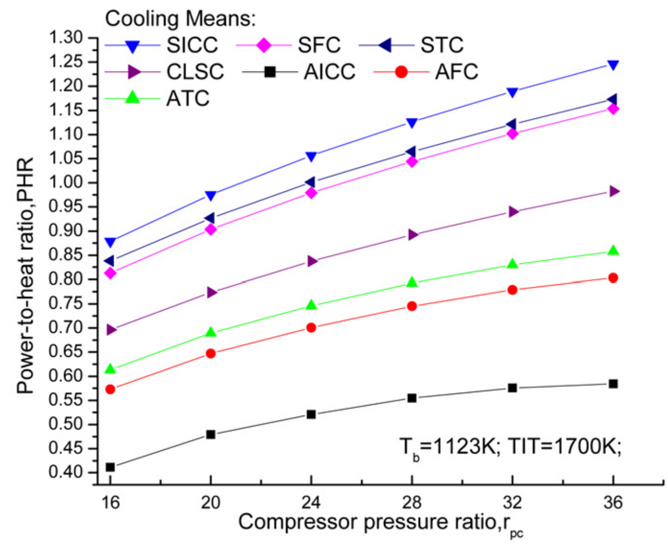


Fig. 9. Effect of $r_{p,c}$ on power-to-heat ratio for various means of blade cooling.

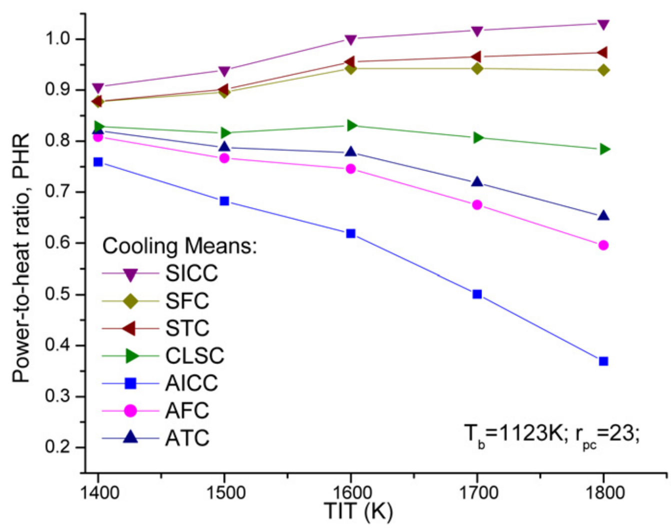


Fig. 10. Effect of TIT on power-to-heat ratio for various means of blade cooling.

The value of power-to-heat ratio of a cogeneration plant, an important parameter may vary depending upon the requirements. Fig. 10 demonstrates the effect of cooling means on PHR at $r_{p,c} = 23$ for the plant. The behaviour of PHR curves for steam cooling is different than those of air-cooling means. At a certain value of $r_{p,c}$, the value of PHR decreases with increase in the value of TIT for air-cooling means, while in open loop steam cooling, it increases with increase in the value of TIT and for that of CLSC it exhibits a minor decrease. The minimum and maximum values of PHR are exhibited by AICC and SICC respectively.

4.5. Effect of variation of HRSG steam drum pressure on FUE and PHR

Fig. 11 depicts the effect of process steam pressure variation on FUE and PHR for cogeneration plant using AFC and ATC. For studying the effect of steam drum pressure/process steam pressure on cycle performance, the cogeneration gas turbine is considered coupled to a single pressure HRSG generating process steam for process heating.

The value of FUE decreases while value of PHR increases with increase of process steam pressure for both AFC and ATC. This is because of the fact that with increase of process steam pressure generated in HRSG, the heat utilised for steam generation

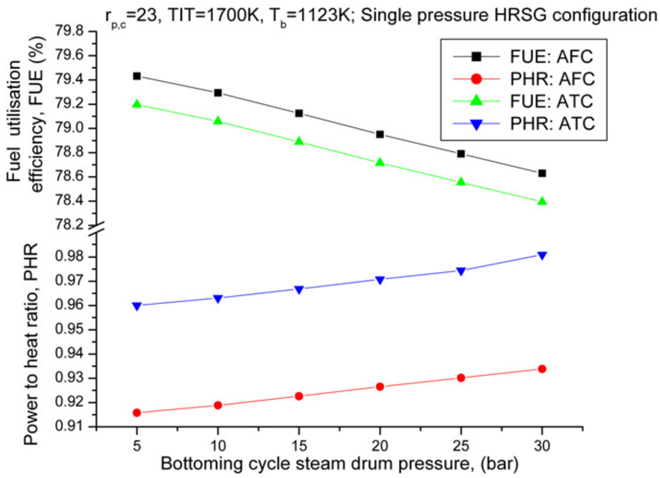


Fig. 11. Effect of steam drum pressure on PHR and FUE in case of single pressure HRSG for air film cooling and air transpiration cooling.

decreases resulting in decrease in the value of FUE and increase in that of PHR.

5. Conclusions

1. The gas turbine based cogeneration cycle having steam-internal convection cooling scheme offers the highest value of fuel utilisation efficiency followed by air internal convection cooling.
2. Fuel utilisation efficiency of the plant at a given value of compressor pressure ratio increases with increase in value of turbine inlet temperature for all cooling means.
3. At a given value of turbine inlet temperature, fuel utilisation efficiency decreases with increase in the value of compressor pressure ratio, for all air cooling and closed loop steam cooling, but in the case of open loop steam cooling, it increases with increase in value of compressor pressure ratio.
4. At a given value of compressor pressure ratio, power-to-heat ratio of a cogeneration plant increases with increase in value of turbine inlet temperature in the case of open loop steam cooling, while it decreases with increase in value of turbine inlet temperature for air cooling means.
5. The minimum and maximum values of power-to-heat ratio are found with air internal convection cooling and steam internal convection cooling respectively.
6. For all cooling means, the value of power-to-heat ratio of gas turbine based cogeneration plant increases with compressor pressure ratio at a given value of turbine inlet temperature.
7. In the case of air film cooling the value of fuel utilisation efficiency is higher than that for air transpiration cooling and it decreases with increase of process steam pressure.
8. The value of power-to-heat ratio in case of air transpiration cooling is higher as compared to air film cooling and it increases with the increase in the value of process steam pressure.
9. The analysis is useful for power plant designers to select the optimum compressor pressure ratio, turbine inlet temperature, fuel utilisation efficiency, power-to-heat ratio, and appropriate cooling means for a specified value of plant specific work and process heating requirement.

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