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Thermodynamic analysis of advance technology combined cycle power plant emploing water/steam closed loop cooling in gas turbines

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ABSTRACT

Combined cycle performance can be improved by improving the performance of gas turbine based topping cycle. Amongst various options the effective cooling of gas turbine blade for realizing the higher turbine inlet temperatures is one of the attractive options. The present paper deals with the thermodynamic analysis of advanced technology gas/steam combined cycle based power plant having closed loop cooling in gas turbine stages employing water/steam as coolants. Results have been obtained for the cycle pressure ratios varying between 12 to 20 and turbine inlet temperatures (TITs) varying in 1400 K to 1700 K for studying the influence of water and steam coolants upon the overall cycle performance.

1. Introduction

The performance of combined cycle actually depends on the performance of gas turbine based topping cycle and steam turbine based bottoming cycle operating synergistically. Realization of higher turbine inlet temperature and the design of efficient turbine stage cooling for it are one of the major thrust areas for improving performance of combined cycle plants.

It is seen that as TIT increases the specific work output of gas turbines improves and also improves the performance of steam turbines in the gas/steam combined cycle arrangement through an important link i.e. HRSG whose performance also depends on the TIT aspects. In order to run the turbine within the metallurgical constraints some type of cooling [1,17,18] such as internal steam cooling, film steam cooling, internal air cooling, film air cooling, closed loop water cooling and/or open loop water cooling etc. are used in order to avoid reduction of operating life due to combination of oxidation, creep, and thermal stresses.

In this paper an advanced gas/steam combined cycle configuration having closed loop cooling (internal cooling) of gas turbine using water/steam as coolants is considered for thermodynamic analysis. The influence of internal water and steam cooling on the performance of gas / steam combined cycle is being studied. Internal cooling model for water and steam cooling has been used to evaluate the gas turbine cooling requirement.the effect of internal cooling upon the potential work of the topping cycle and the bottoming cycle are discussed here.

2. Configuration

Advanced technology gas/steam combined cycle configuration having a simple gas turbine in topping cycle and reheat steam turbine in bottoming cycle as shown in figure 1 has been con-Sid. ered [4,6]. Turbine stages are internally cooled employing water/steam as coolant. The requirement of water coolant is met from condensate leaving the condenser and steam coolant re-

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quirement is met from the steam generated in heat recovery steam generator (HRSG). Water coolant specifications are derived by condenser water characteristics, while for steam these are assumed as those of HRSG steam specifications. Steam generation in HRSG occurs at three pressure & temperature levels. In the case of steam cooled gas turbine the steam is extracted from the exit of high pressure steam turbine to cool turbine stages through internal channels [1,5]. The steam coolant leaving gas turbine is subsequently mixed with the IP drum steam at same pressure before reheated for further expansion in the intermediate pressure steam turbine. After cooling gas turbine stage, the water coolant leaving gas turbine is mixed in the deaerator. Fuel is preheated using the heat available with steam extracted from intermediate pressure steam generated in HRSG, thus reducing the fuel requirement.

3. Thermodynamic Analysis

Thermodynamic analysis of the configuration involved is done using the mathematical models for air, gas & steam properties and various components involved in cycle. Mathematical modeling used for this analysis is given ahead [2,3,7].

Air, gas and steam properties model: Thermodynamic analysis requires the gas, air and steam properties at every salient stage. In the present model the specific heat is considered to be function of temperature alone, neglecting the influence of pressure as the pressure is not very high in the cycle. Specific heat is considered as function of temperature given below:

$$c_p(T) = a + bT + cT^2 + dT^3 + - - - - - - - \text{The}$$

values of coefficients a, b, c and d differ for each fluid.

For the specific compositions by mass of air and natural gas as given below the specific heat polynomials are detailed herein, [9]:

Air:
$77.44\%N_2 + 20.76\%O_2 + 0.92\%Ar + 0.85\%H_2O + 0.03\% CO_2$

Natural Gas: $0.94\ CH_4 + 0.03\ C_2H_6 + 0.03\ N_2$
Specific heat polynomials, [10] considered are as under;

$c_{p,Air}(T) = [28.11 + 0.1967 \times 10^{-2}\ (T) + 0.4802 \times 10^{-5}\ (T)^2 - 1.966 \times 10^{-9}\ (T)^3]/29$, kJ/kg K

$c_{p,CO2}(T) = 0.506 + [0.136 \times 10^{-2} \times T] - [0.0796 \times 10^{-5} \times T^2] + [0.169 \times 10^{-9} \times T^3]$, kJ/kg K

$c_{p,H2O}(T) = 1.79 + [0.0107 \times 10^{-2} \times T] + [0.0586 \times 10^{-5} \times T^2] - [0.199 \times 10^{-9} \times T^3]$, kJ/kg K

$c_{p,N2}(T) = 1.032 - [0.0056 \times 10^{-2} \times T] + [0.0288 \times 10^{-5} \times T^2] - [0.103 \times 10^{-9} \times T^3]$, kJ/kg K

$c_{p,C.P.} = [[(0.99 \times c_{p,CO2}(T)) + (1.95 \times c_{p,H2O}(T)) + (7.4184 \times c_{p,N2}(T))]/10.3584]$, kJ/kg K

The enthalpy of gas is expressed as,

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

Ambient states are considered at 298.16K, 1.013 bar for air.

Compressor : Axial flow type compressor is considered here. Ideally compression should be adiabatic while in reality it is not possible to realize it. The inefficiency in adiabatic compression process is accounted through polytropic efficiency which is assumed as 0.92 here.

Considering the polytropic efficiency the temperature and pressure variation during compression can be related as below ;

$$T_2/T_1 = (p_2/p_1)^{(n-1)/n}$$

$$(n-1)/n = (\gamma-1)/ \gamma\ \eta_{p,c}$$

and $(\gamma-1)/ \gamma = \bar{R} / \mu\ c_{p,a}$

Mathematical model based on energy balance and mass balance is given below.
Mass balance ; $m_1 = m_c + m_2$
Energy balance;

$$W_c = m_2 \int_{T1}^{T2} c_{p,a}(T)dT + m_c \int_{T1}^{Tc} c_{p,a}(T)dT$$

Combustion Chamber: Natural gas is used as fuel in combustion chamber. The losses arising in the combustion chamber due to incomplete combustion and pressure losses are taken into account by introducing the combustion efficiency and pressure drop as fraction of combustor inlet pressure i.e. 0.5 % considered here.

The mass and energy balance yields, Mass balance; $m_2 + m_f = m_3$ Energy balance;
$m_2 c_{p,a} T_2 + m_f CV. \eta_{comb} = m_3.c_{p,c.p.}T_3$

where, CV= 48990 kJ/kg = Lower heating value of Natural Gas

Fuel saving ratio (FSR) defined as the ratio of fuel saved per unit power output in combined cycle configuration to the fuel requirement per unit power output in gas turbine can be estimated as below; FSR $=1-(m_{f, cc}/m_{f, GT})$

Where, $m_{f, cc}= (m_f /$ Net cycle work out put) and $m_{f, GT}= (m_f /$ Net gas turbine work out put).

Gas Turbine: Steam or water is used as coolant for turbine blade cooling. Gas turbine blades cooled by internal convection are treated as heat exchangers operating at uniform temperature and the coolant exit temperature is expressed as a function of heat exchanger effectiveness. For the present analysis the turbine inlet temperature is considered between 1400 K to 1700 K. Polytropic efficiency of 0.90 is considered to account for non-idealities in expansion process. First law analysis yields following mass and energy balance.

Mass balance ; $m_4 = m_3$ Energy balance ;

$$W_T=m_3 \int_{T3}^{T4} c_{p,C.P.} (T).dT$$

Coolant mass required for closed loop cooling,[2],

$$m_g St_o A_w c_{p,g} (T_g - T_b)$$

$$m_c = \varepsilon A_g c_{p,c} (T_b - T_{c1})$$

where, $\varepsilon = (T_{c2} - T_{c1}) / (T_b - T_{c1})$

For calculations the blade temperature is considered as 1123 K, ratio of wall to gas surface area, $A_w /A_g =4.5$ (3-6), $\varepsilon=0.5$, $St_o=0.005$, $T_b=1123$ K

Heat recovery steam generator: Heat recovery steam generator (HRSG) is a kind of heat exchanger where gas turbine exhaust heat is utilized for steam generation. Depending upon requirement the steam is generated at different pressures and temperatures. HRSG produces low pressure (lp), intermediate pressure (ip) and high pressure (hp) steam as per requirement. Mass and energy balance on HRSG yields;

Mass balance; $m_s = m_{s,lp}+m_{s,ip}+m_{s,hp}$ Energy balance;

$$m_{s,lp}=(\varepsilon_{HRSG} .Q)/[\{(h_{3s}-h_{17s})/\rho_2 \}+\{(h_{6s}-h_{16s})/\rho_1 \}+(h_{9s}-h_{15s})]$$

HRSG effectiveness is considered as 0.90. Heat available in gas turbine exhaust,

$$Q=m_g . \int_{T4}^{T5} c_{p,C.P.} (T).dT$$

$\rho_1 ==0.72 =$ Ratio of mass of lp steam to mass of ip steam

$\rho_2 = 0.052 =$ Ratio of mass of lp steam to mass of hp steam

In HRSG the steam generation pressure and temperature considered are as given below. hp/ ip/ lp steam generation pressures = 165/ 23.8/ 2.2 bar hp/ ip/ lp steam generation temperatures = 565/ 565/ 277 ºC

Steam bled for fuel preheating is 50% of steam generated in ip drum in HRSG. Stack gas pressure and temperature in HSRG is 1.1 bar and 383.16 K.

Steam Turbine: Steam turbine used here has three stages namely high pressure (hp), intermediate pressure (ip) and low pressure (lp) stages. Steam generated in HRSG is sent for expansion in these stages. Inefficiency in expansion process is accounted by isentropic efficiency assumed to be uniform for all stages. Isentropic efficiency is considered as 0.90.

Energy balance on steam turbine estimates the theoretical work output;

$W_{ST,th} = m_{3s} (h_{3s}-h_{18s})+m_{6s}(h_{6s}-h_{11s})+m_{9s}(h_{9s}-h_{11s})-m_{bs}(h_{10s}-h_{11s})$

Steam turbine work output, $W_{ST} = W_{ST,th} \cdot_{isen,ST}$ Steam bleeding is undertaken at 2.2 bar pressure. Steam mass bled for D/A is 20 % of steam entering lp steam turbine.

Condensate Extraction Pump: The enthalpy at exit of condensate extraction pump and work requirement of condensate extraction pump can be obtained using following equations. Condenser pressure is 0.04064 bar.

Enthalpy at exit of pump,

$h_{13s}=h_{12s}+v_{12s}(p_{13s}-p_{12s})$

Work requirement of condensate extraction pump (cep),

$W_{cep}= m_{12s} (h_{13s}-h_{12s})$

Deaerator : Energy balance equation for deaerator yields,

$m_s .h_{14s} = m_{bs} .h_{10s} + m_{13s} h_{13s}$

Feed Water Pump: Energy balance yields the work requirement of feed water pump (fwp) as under,

$W_{fwp} = m_s .(h_{15s} - h_{14s})$,where $h_{15s} = h_{14s} + v_{14s} . (p_{15s} - p_{14s})$

Using the work interaction at different components and fuel used various efficiency values can be obtained as following ;

Topping cycle efficiency = $(W_{GT} - W_C)/(m_f \times CV)$

Bottoming cycle efficiency = $(W_{ST} - W_{fwp} - W_{cep})/(Q)$

Combined cycle efficiency

$= \{(W_{GT} - W_C) + (W_{ST} - W_{fwp} - W_{cep})\}/(m_f . CV)$

4. Results And Discussion

Based on the thermodynamic analysis of combined cycle configuration using closed loop water/steam cooled gas turbine for varying cycle pressure ratio from 12 to 20 and turbine inlet temperature from 1400 to 1700K.

Fig 2 and 3 depict the variations of topping cycle and bottoming cycle efficiency with cycle pressure ratio at different TIT for water / steam cooled option at fixed turbine blade temperature of 1123K. Variations show that topping cycle efficiency is at maximum in both the cases of water & steam cooled gas turbine employed at higher cycle pressure ratio and higher TIT values. This may be attributed to the fact that at higher TIT and cycle pressure ratio the increase in net work available is more than the increment in the heat input thus showing overall gain in efficiency of topping cycle. Bottoming cycle efficiency also increases with increase in cycle pressure ratio and TIT in case of water cooled gas turbine. But in case of steam cooled gas turbine bottoming cycle efficiency is higher at lower TIT as less coolant is required and more steam is available for work in steam turbine. The bottoming cycle efficiency is more in water cooled gas turbine than steam cooled gas turbine for the same cycle pressure ratio and TIT because net work out put in steam turbine with water cooled gas turbine is more than steam cooled gas turbine as the steam partly

utilized for gas turbine cooling reduces the steam turbine work out put.

The variation in bottoming cycle efficiency at cycle pressure ratios varying from 12 to 18 at TITs 1400K, 1500K, 1600K and 1700K are shown in fig 3.

The influence of maximum gas turbine blade temperature varying from 850^0C (1123K) to 1150 ^{0}C (1423K) upon combined cycle efficiency and combined cycle specific work output for the varying TIT and cycle pressure ratio are presented in figures 4 to 7 for both water & steam cooled gas turbine. The higher TIT with respect to maximum gas turbine blade temperature is always desirable for getting higher efficiency and work output. Hence the maximum gas turbine blade temperature is an influential parameter and as the maximum gas turbine blade temperature increases the cooling requirement diminishes so the loss of work due to gas turbine blade/stage cooling is reduced. Increase in TIT is accompanied by the increase in turbine work output and at the same time stringent cooling requirements. Coolant mass requirement depends upon permissible maximum blade temperature and this coolant quantity required is less with higher permissible blade temperature.

Fig8 indicates variation of specific work output contributions of topping & bottoming cycle with TIT at different pressure ratios for closed loop water / steam cooled gas turbine having maximum permissible gas turbine blade temperature as 1123K. It is shown that there is not much variation in the topping and bottoming cycle work output contributions when either water or steam cooling is employed. However, the cycle specific work output is seen to be slightly more when steam cooling is used as compared to water cooling in gas turbine at cycle pressure ratio 12 for TIT range 1400 to 1700 K.

Fig 9 is the selection diagram for the combined cycle configuration considered employing water/steam cooling. This diagram can be used for getting cycle efficiency and cycle specific work output for cycle pressure ratios varying between 12 to 20 and turbine inlet temperatures in the range of 1400 K to 1700 K for the maximum gas turbine blade temperature of 1123K. The optimum cycle pressure ratio and TIT can be found out from this diagram for the combined cycle. In

case of water cooled gas turbine the maximum cycle efficiency of 57.18 % is obtained at the cycle pressure ratio of 20 and TIT of 1700 K while cycle specific work output is maximum with value of 1117.68 kJ/kg air at the cycle pressure ratio of 12 and TIT of 1700 K.

Similarly when steam efficiency of 57.85 % is obtained at cycle pressure ratio of 20 and TIT of 1700K. Also the maximum combined cycle specific work output of 1133.71 kJ/kg air is obtained at the cycle pressure ratio of 12 and TIT of 1700K. In the present combined cycle arrangement the fuel heating is employed which results in reduced fuel requirement. Figure 10 shows the

variation of fuel saving ratio (FSR) with turbine inlet temperature at cycle pressure ratio of 20. It shows that the fuel saving ratio increases as turbine inlet temperature increases in both combined cycle configurations employing water and steam cooled gas turbine separately. However, in case of combined cycle having water cooled gas turbine the fuel saving ratio is always higher than combined cycle having steam cooled gas turbine. This may be attributed to the fact that the ratio of fuel consumed per unit power output in combined cycle is lesser in water cooled gas turbine than steam cooled gas turbine cycle for all TIT values.
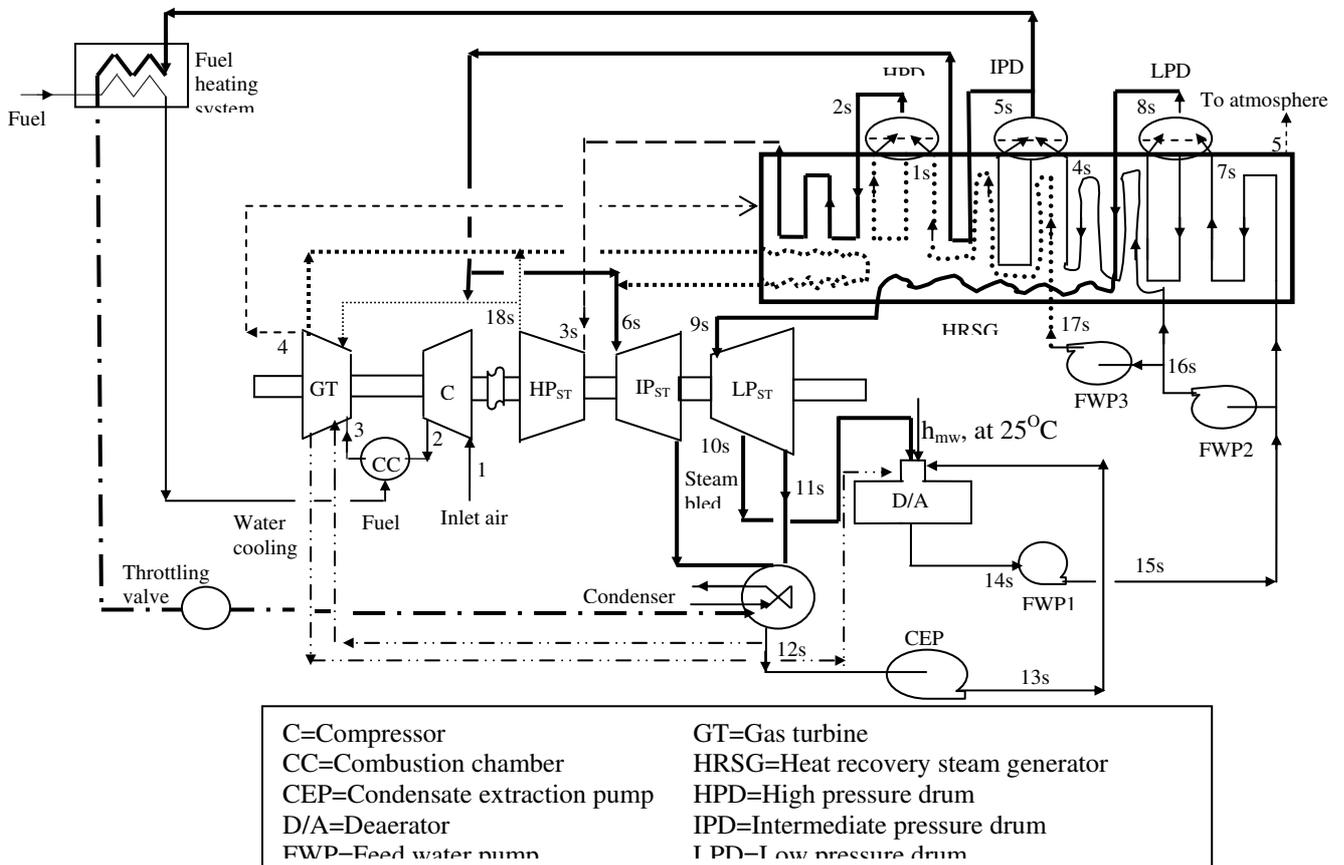

Fig. 1: Advance Technology gas / steam combined cycle having triple pressure steam generation in HRSG with closed loop cooling
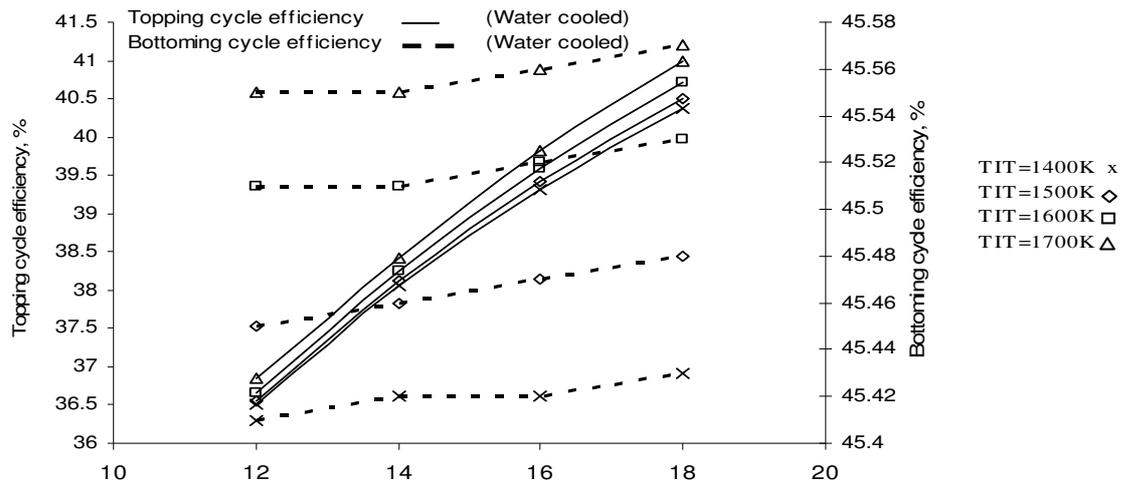
C=Compressor
CC=Combustion chamber
CEP=Condensate extraction pump
D/A=Deaerator
FWP=Feed water pump

GT=Gas turbine
HRSG=Heat recovery steam generator
HPD=High pressure drum
IPD=Intermediate pressure drum
LPD=Low pressure drum

Fig. 2: Variation of topping cycle & bottoming cycle efficiency with cycle pressure ratio for closed loop water cooled gas turbinein combined cycle having triple pressure steam generation
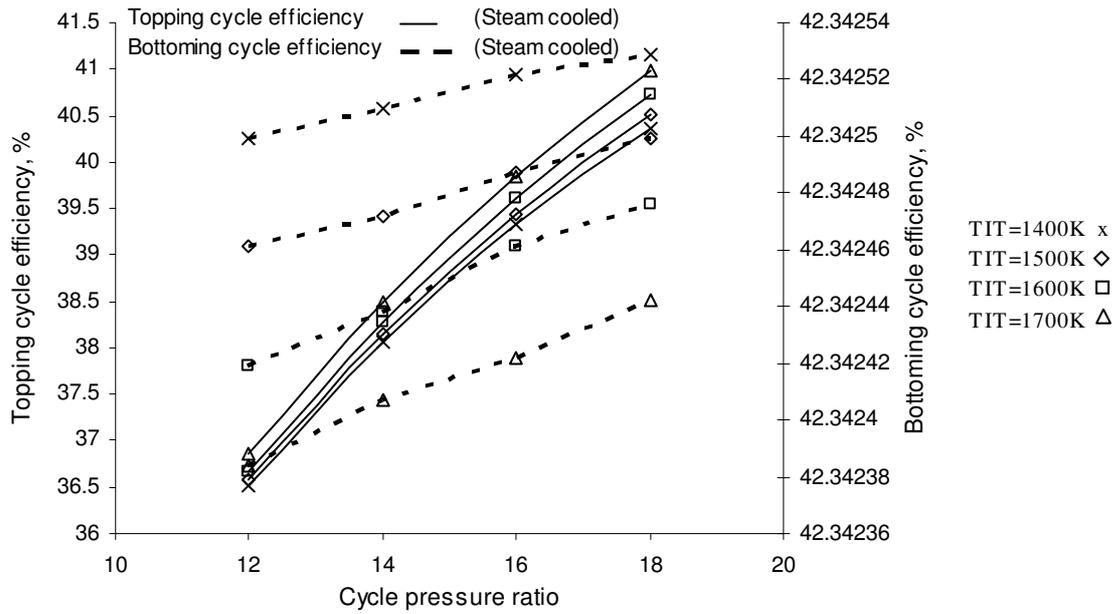

Fig. 3: Variation of topping cycle & bottoming cycle efficiency with cycle pressure ratio for closed loop steam cooled gas turbine in combined cycle with triple pressure steam generation.
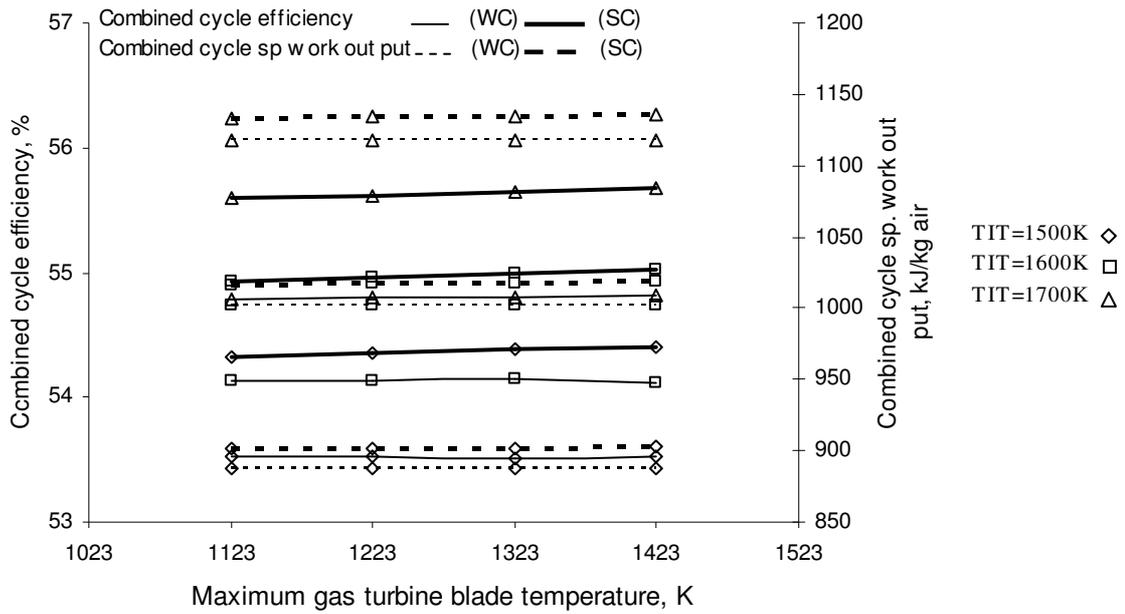
Fig. 4: Variation of combined cycle efficiency & cycle specific work output with maximum gas turbine blade temperature having water steam cooled gas turbine at cycle pressure ratio 12.
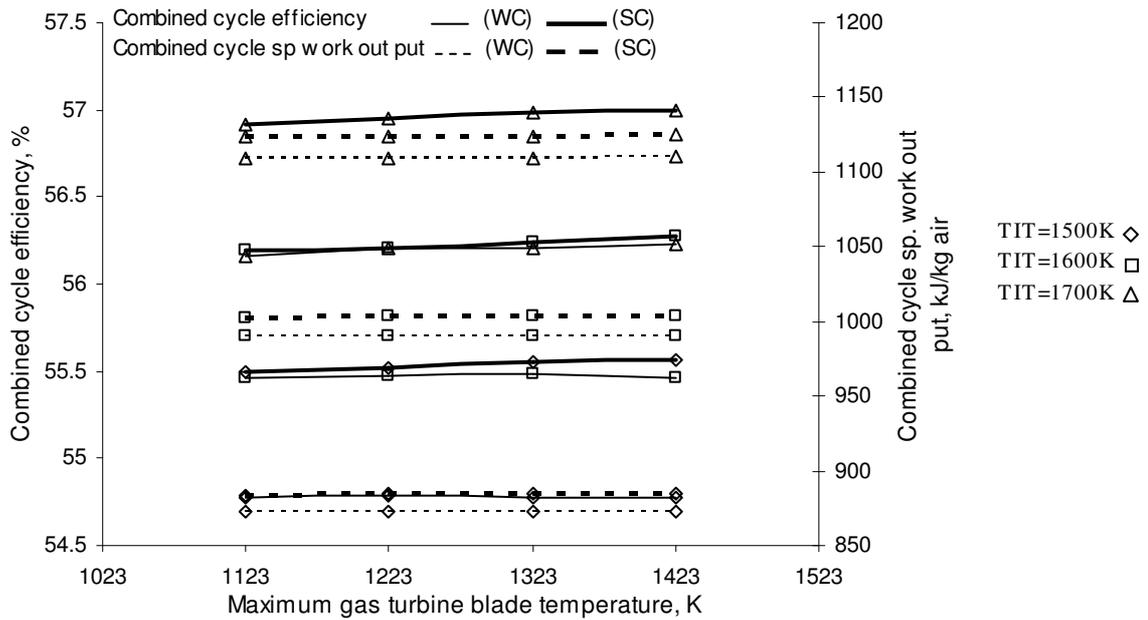

Fig. 5: Variation of combined cycle efficiency & cycle specific work output with maximum gas turbine blade temperature having water steam cooled gas turbine at cycle pressure ratio 16.
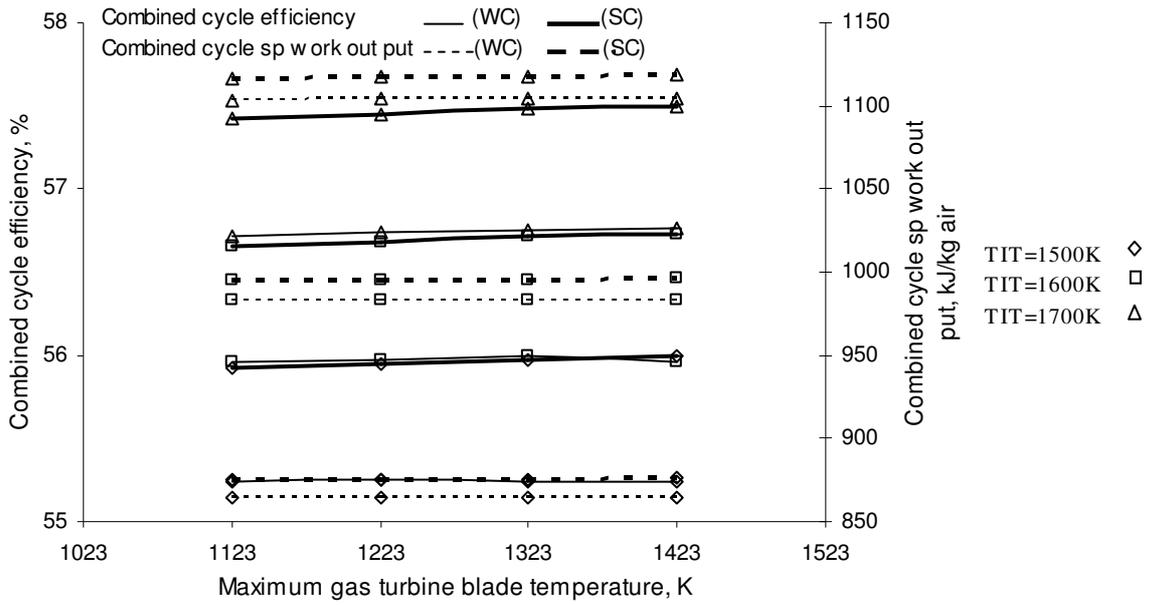
Fig. 6: Variation of combined cycle efficiency & cycle specific work output with maximum gas turbine blade temperature having water steam cooled gas turbine at cycle pressure ratio 18.
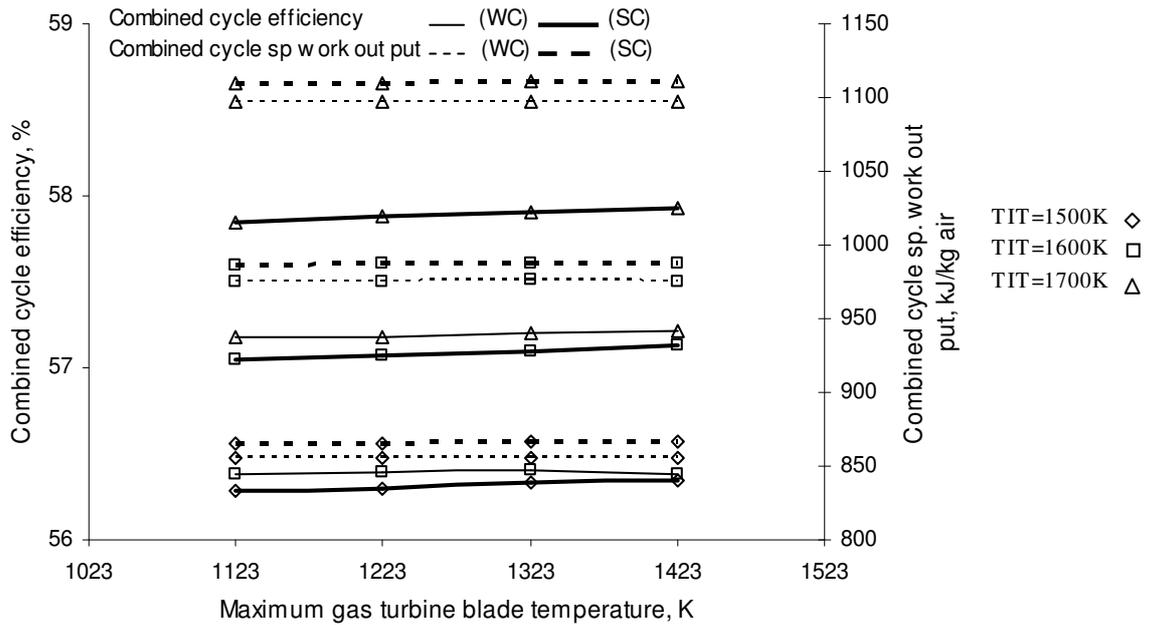

Fig. 7: Variation of combined cycle efficiency & cycle specific work output with maximum gas turbine blade temperature having water steam cooled gas turbine at cycle pressure ratio 20.
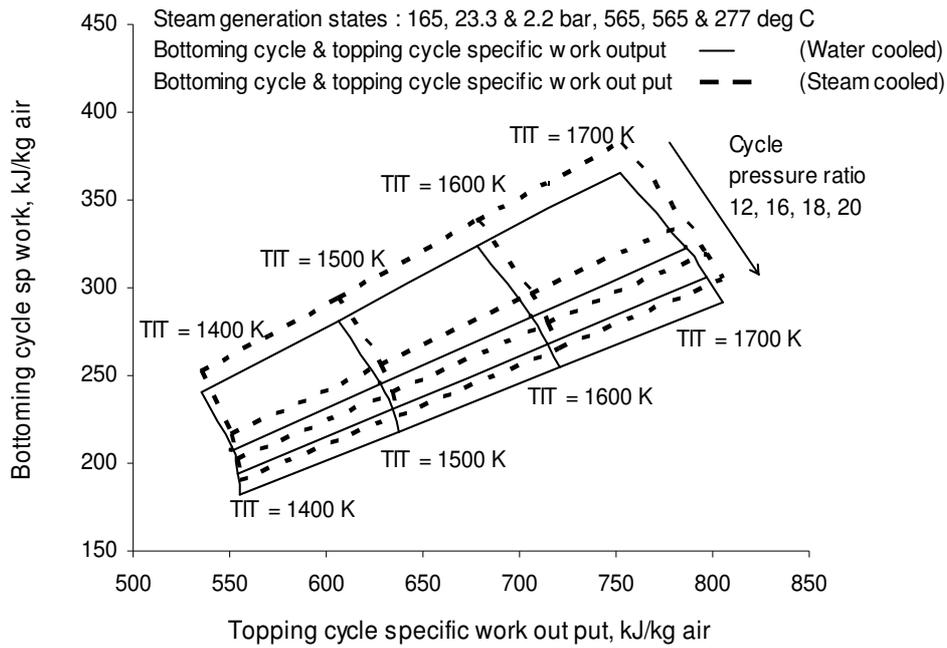
Fig. 8: Variation of bottoming cycle and topping cycle specific work output with TIT and cycle pressure ratio having water/ steam cooled gas turbine.
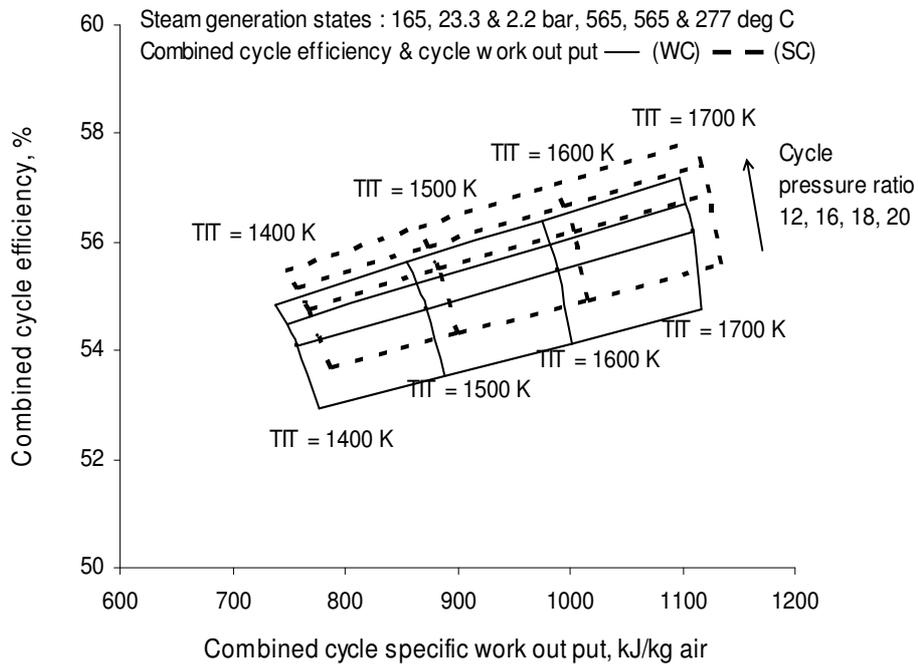

Fig. 9: Variation of combine cycle efficiency and combined cycle specific work output with TIT and cycle pressure ratio having water/ steam cooled gas turbine.
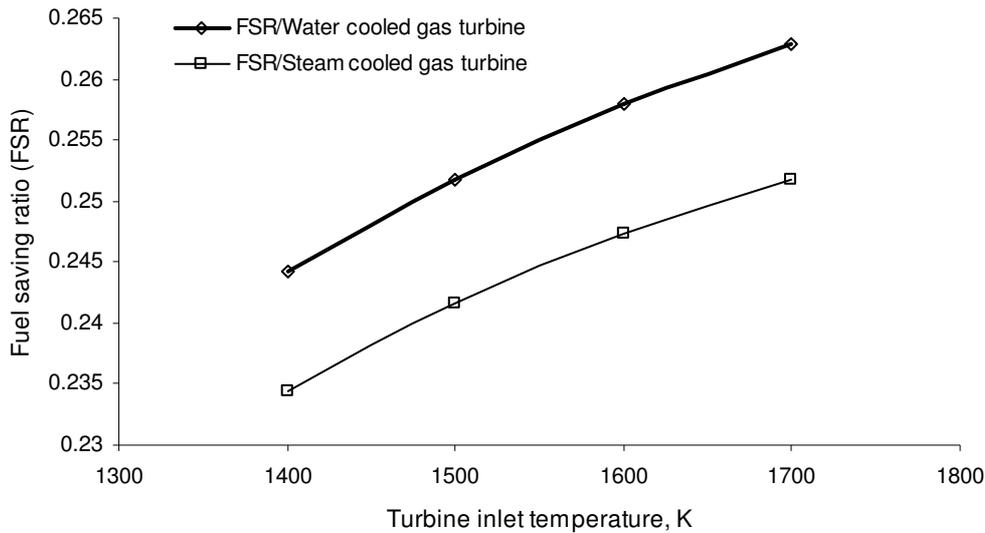
Fig. 10: Variation of fuel saving ratio with turbine inlet temperature at cycle pressure ratio 20 for water/ steam cooled gas turbine

5. Conclosion

The conclusions of the analysis of advanced technology gas/steam combined cycle configuration employing water/steam closed loop cooling in gas turbine are as follows:

- Selection diagram obtained can be used for predicting the cycle performance at any TIT and cycle pressure value within the cycle pressure ratio of 12 to 20 and TIT within 1400 K to 1700 K.

- Closed loop cooling of gas turbine stages employing steam is advantageous over the water as coolant.

- Maximum combined cycle efficiency of 57.93% and combined cycle specific work output of 1111.16 kJ/kg air can be obtained from combined cycle using steam cooled gas turbine stages at cycle pressure ratio of 20 and TIT of 1700 K when maximum gas turbine blade temperature is maintained at 1423 K.

- For the permissible blade temperature of 1423 K the maximum combined cycle efficiency is 57.22 % and combined cycle specific work output of 1097.89 kJ/kg air at the cycle pressure ratio of 20 and TIT of 1700 K in case of the water cooled gas turbine stages.

Nomenclature

c_p	: Specific heat at constant pressure
CV	Lower heating value of fuel
H	Specific enthalpy mMass flow rate
n	Polytropic index
: p	: Pressure
Q	Heat available in gas exhaust
$\bar{R}$	Universal gas constant
T	Temperature
v	: Specific volume
W	Work rate

Greek letters

η	Efficiency
γ	Ratio of specific heats
μ	: Molecular weight
ε	Effectiveness of cooling channel
ρ	Ratio of masses of steam

Subscripts

a	: air
b	: bleed / blade
c	: compressor / coolant
comb	: combustion
cc	: combined cycle
c1, c2	: coolant at inlet & exit
C.P	: combustion product
f	: fuel
g	: gas

isen   : isentropic
p   polytropic
S   steam
ST   steam turbine
th   : theoretical
1, 2, ..,   state points in fig 1
1s, 2s

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