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Thermodynamic analysis of gas – steam combined cycle with carbon dioxide (CO_2) emissions saving

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Abstract
In this paper, cogeneration or combined heat and power (CHP) cycle has been analyzed in order to improve the efficiency of the gas – steam combined cycle and utilization of waste heat. The efficiency of the combined cycle is improved by decreasing the compressor inlet temperature (CIT) and increasing the turbine inlet temperature (TIT). It is observed that the cycle offers the advantage of making efficient use of the energy available in the fuel and in turn, eliminate some portion of pollution associated with the power generation. The study also reveals that if this cycle is being employed for cogeneration, there is a significant saving (11.60%) in the amount of Carbon dioxide (CO_2) emitted by the coal-fired thermal power plants.

Keywords: Carbon dioxide (CO_2) emissions saving, Cogeneration, Compressor inlet temperature, Efficiency, Gas – steam combined cycle, Model, Turbine inlet temperature.

1. Introduction

The surest way of reducing carbon emissions into the atmosphere is by reducing energy generation from fossil fuels [1], which can be achieved by incorporating the cogeneration technology. Cogeneration or Combined Heat and Power (CHP) can be defined as the sequential generation of two different forms of useful energy from a single primary energy source, typically mechanical energy and thermal energy. The two energies being utilized to generate electricity and process steam/hot water, respectively. The advantages generally reported from co-generating thermal and electrical energy rather than generating the same products in separate processes include: reduced energy consumption, reduced environmental emissions, and more economic, safe and reliable operation [2]. With rapid advances in the cooling techniques and blade materials, gas turbine combined cycle efficiency has improved significantly, thus, offering a good option for cogeneration. Also, the gas turbine is further recognized for its better environmental performance, manifested in the curbing of air pollution and reducing green house gases [3], the current major concern through out the world.

In a combined cycle power plant, the performance of the combined cycle depends on the individual performance of the topping and bottoming cycles. Gas turbine is seen to offer high specific work output if the turbine inlet temperature (TIT) could be increased. Thus, with increased TIT the performance of the heat recovery steam generator (HRSG) and consecutively the steam turbine improves thereby, offering improvement in combined cycle performance [4]. Also, the efficiency of the gas turbine cycle can be improved by the supply of cooled air i.e. by decreasing the compressor inlet temperature (CIT) with the help of refrigeration. Pioneering work in the area of efficiency enhancement of the gas – steam combined cycle has been done by Sanjay *et al* [5], Khodak and Ramakhova [6], Al-Fahed *et al* [7], Najjar [8], [9], Bolland [10].

Today, a lot of research is being done in order to reduce the levels of green house gases (GHGs – CO_2, CO, SO_2 and NO) in the atmosphere. The main source of CO_2 (direct GHG) is the combustion of fossil fuels such as coal, oil and gas in power plants. The CO_2 emitted as a product of combustion of coal (fossil fuel) is currently responsible for over 60% of the enhanced greenhouse effect [11]. Scientists project that excessive CO_2 emission into the atmosphere will increase the earth's surface temperature approximately 1.5 – 4 ^oC in the next 30 – 40 years. [12].

This paper demonstrates the improvements in the efficiency of the gas – steam combined cycle with the increase in the TIT and decrease in the CIT. In view of the measures being adopted worldwide to reduce the emissions of green house gases (GHGs), an attempt has been made in the paper to show the reduction in the carbon dioxide (CO_2) emissions achieved by way of cogeneration.

2. System configuration

A simple gas – steam combined cycle with vapor refrigeration system has been considered for the present study and analysis. The topping cycle is based on the Brayton cycle and the bottoming cycle on the Rankine cycle. For the gas turbine blade cooling, air bled from the compressor of the topping cycle is being employed. Figure 1 shows the schematic diagram of the cycle configuration and the corresponding T-s diagram is shown in Figure 2.

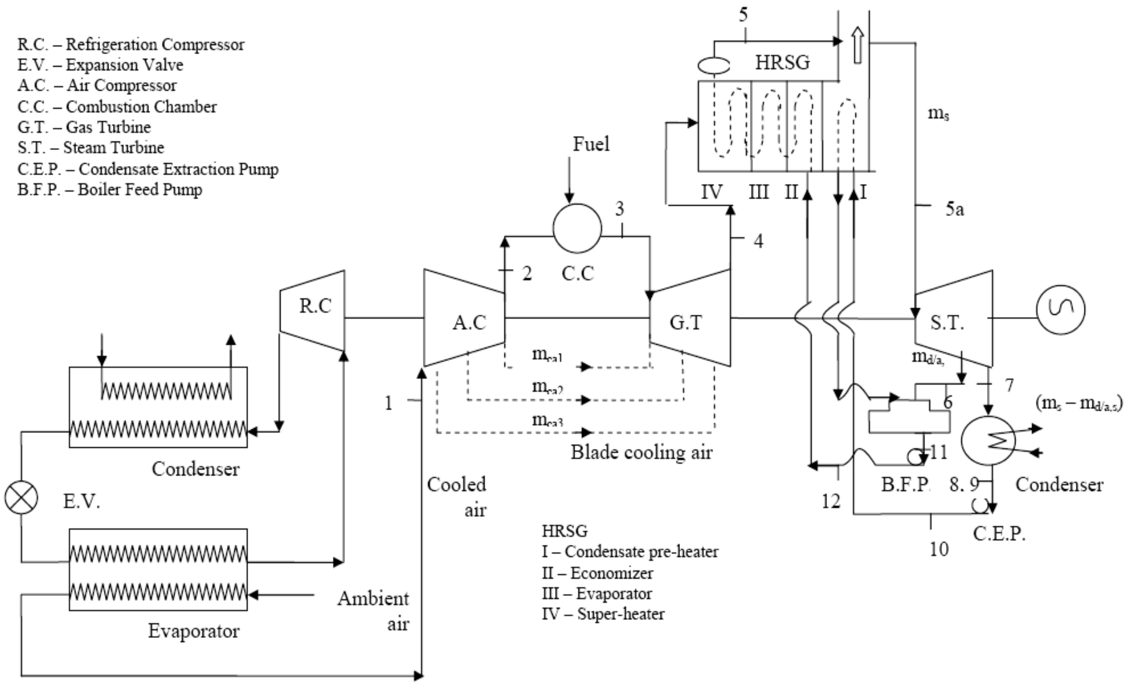

Figure1. Schematic diagram of basic gas – single pressure steam combined (BIP) cycle with vapor compression refrigeration cycle

3. Modelling and governing equations

For thermodynamic analysis of the described configuration, mathematical modelling has been done for each component of the cycle and based on these models the governing equations are written.

3.1 Inlet air refrigeration model

For the purpose of inlet air cooling, vapor compression refrigeration system is employed. Any change in the moisture content due to cooling of the air is ignored. Heat transfer rate are constrained to be finite in both evaporator and condenser. The v refrigeration system used is conceived as Carnot cycle but its actual COP is calculated by introducing the concept of refrigeration efficiency *(η_r)*. For accounting the inefficiencies of condenser and evaporator, the concept of effectiveness is introduced.

The optimized work of refrigeration is given by:

$$w_r = \frac{c_{pa}(T_o - T_1)\left[\varepsilon_e T_{hc} - \{T_o \varepsilon_e - (T_o - T_1)\}\right]}{\eta_r \{\varepsilon_e T_o - (T_o - T_1)\}} \tag{1}$$

where w_r is the work of refrigeration (kJ/kg of air), c_{pa} is the specific heat of air at constant pressure (kJ/kg K), T_o is the ambient temperature (K), T_1 is the compressor inlet temperature (K), T_{hc} is the temperature of refrigerant in the condenser (K), ε_e and ε_c is the effectiveness of heat transfer in the evaporator and condenser, respectively.

$$T_{hc} = \frac{b \pm \sqrt{(b^2 - 4ac)}}{2a} \tag{2}$$

where

$$a = X^2(T_o + T_1) \qquad X = \varepsilon_e(2 - \varepsilon_c)$$

$$b = (2X^2 + Y^2)T_o^2 + 2(X^2 - Y^2)T_1 T_o + Y^2 T_1^2 \qquad Y = \varepsilon_c(2 - \varepsilon_e)$$

$$c = (X^2 - Z)T_o^3 - ZT_1^2 + (X^2 + 2Z)T_1 T_o^2 \qquad Z = (2 - \varepsilon_c)\ (2 - \varepsilon_e)^2$$

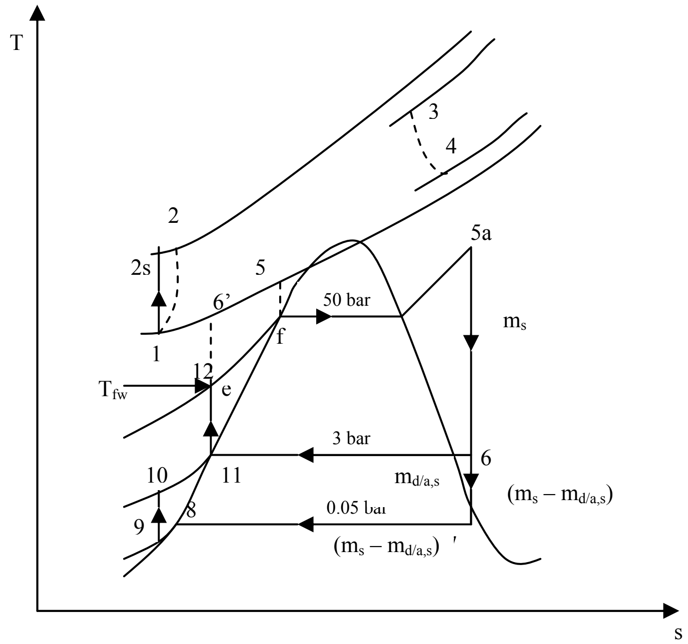

Figure 2. Representation of BIP cycle on T-s diagram

3.2 Compressor model

Compression process is adiabatic and the non-adiabatic compression caused by various losses is taken care of by assuming suitable polytropic efficiency. Air bled from the compressor for turbine blade cooling does not disturb the flow path of the compressor. The stage pressure is approximately equal and is given by:

$$r_{pstage} = \left(r_p\right)^{1/n} \tag{3}$$

where r_{pstage} stage pressure ratio of compressor, r_p pressure ratio of compressor and n is the number of stages.

For the compression process,

$$T_2/T_1 = \left(p_2/p_1\right)^{(\gamma-1)/\gamma\eta_c} \tag{4}$$

where T_1 and T_2 are the compressor inlet and exit temperatures (K), p_1 and p_2 are the compressor inlet and exit pressures (Pa), γ is the specific heat ratio of air, η_c is the polytropic efficiency of compressor.

Mass balance gives:

$$\left(m_{comp}\right)_{in} = \left(m_{comp}\right)_{out} + \sum m_i \tag{5}$$

where $\left(m_{comp}\right)_{in}$ and $\left(m_{comp}\right)_{out}$ is the mass in and out of the compressor (kg) and $\sum m_i$ is the summation of cooling air mass (kg).
Energy balance gives:

$$w_{comp} = \left(m_{comp}\right)_{out}\left(h_{comp}\right)_{out} + \sum\left(m_i h_i\right) - \left(m_{comp}\right)_{in}\left(h_{comp}\right)_{in} \tag{6}$$

where w_{comp} is the compressor work (kJ), $\left(h_{comp}\right)_{in}$ and $\left(h_{comp}\right)_{out}$ is the enthalpy in and out of the compressor (kJ/kg), respectively.

3.3 Combustion chamber model
The losses inside the combustion chamber due to the incomplete combustion of fuel and friction losses are taken care of by introducing the concept of combustion efficiency and percentage pressure drop. Burning of fuel is assumed to be a simple heat transfer to the compresses air. The fuel-air ratio is estimated from the equation:

$$f \times \left(L.C.V.\right)_f = 1 \times c_{pg} \times (T_3 - T_o) - (1 - f) \times c_{pa} \times (T_2 - T_1) \tag{7}$$

where f is the fuel flow rate (kg/s), $(L.C.V)_f$ is the lower calorific value of fuel (kJ/kg), cpg is the specific heat of gas at constant pressure (kJ/kg K), T3 is the turbine inlet temperature (K).
From mass and energy balance:

$$\left(m_{cc,g}\right)_{out} = \left(m_{cc,a}\right)_{in} + m_f \tag{8}$$

where $\left(m_{cc,a}\right)_{in}$ and $\left(m_{cc,g}\right)_{out}$ is the mass of air in and mass of gas out of combustion chamber (kg), respectively, mf mass of fuel (kg).

$$\eta_{cc} m_f \left(L.C.V.\right)_f = \left(m_{cc,g}\right)_{out}\left(h_{cc,g}\right)_{out} - \left(m_{cc,a}\right)_{in}\left(h_{cc,a}\right)_{in} \tag{9}$$

where η_{cc} is the efficiency of combustion chamber, $\left(h_{cc,a}\right)_{in}$ and $\left(h_{cc,g}\right)_{out}$ is the enthalpy of air in and enthalpy of gas out of the combustion chamber (kJ/kg), respectively.

3.4 Gas turbine model
The turbine is treated as an expansion device whose walls continuously extract work rather than discreet extraction of work and losses due to the polytropic expansion in gas turbine are taken care by assuming suitable polytropic efficiency. The irreversible loss of stagnation pressure caused by mixing of cooling air and the hot gases has been assumed continuous throughout the turbine surface and is accounted by a factor fm which is less than unity.
From mass and energy balance:

$$\left(m_{gt,g}\right)_{out} = \left(m_{gt,g}\right)_{in} + \sum m_i \tag{10}$$

where $\left(m_{gt,g}\right)_{in}$ and $\left(m_{gt,g}\right)_{out}$ is the mass of gas in and out of gas turbine (kg), respectively.

$$w_{gt} = f_m \left[\left(m_{gt,g}\right)_{in}\left(h_{gt,g}\right)_{in} + \sum m_i h_i - \left(m_{gt,g}\right)_{out}\left(h_{gt,g}\right)_{out}\right]\eta_{mech} \tag{11}$$

where w_{gt} is the gas turbine work (kJ), fm is the work loss factor due to mixing of cooling air with hot gas in the gas turbine, $\left(h_{gt,g}\right)_{in}$ and $\left(h_{gt,g}\right)_{out}$ is the enthalpy of gas in and out of the gas turbine (kJ/kg), respectively and η_{mech} is the mechanical efficiency.

3.5 Cooling model
For the cooling of the turbine blades, the transpiration cooling has been considered. Each stage rotor and stator is cooled by bleed air from the compressor at appropriate bleed points. Bleed points are selected at slightly greater pressure (between 10 – 15%) than the entry point pressure of the compressor in order to overcome the friction and make the entry of cooling air possible in the turbine. Air is supplied to wall cavity from which it transpires uniformly across the porous wall surface and decreases the surface temperature to allowable level and mixes with the main gas flow. Figure 3 shows an element of

expansion path with cooling air. A concept of blade cooling effectiveness of cooling channel (εb), is assumed to take care of actual possible heat transfer as the cooling air temperature cannot attain the blade temperature even in transpiration cooling.

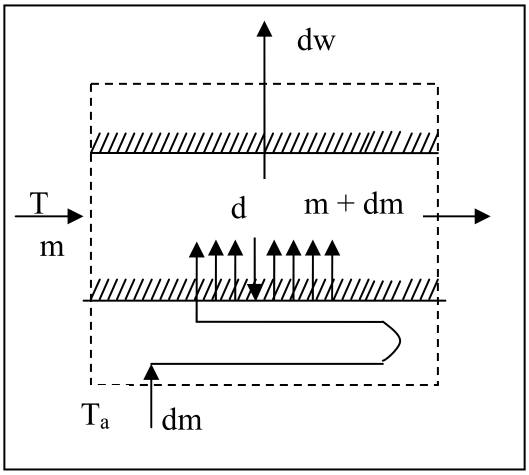

Figure 3. Element of expansion path with cooling air

Cooling mass of bled air is calculated for every rotor and stator individually starting from the last rotor by numerically integrating the following equation:

$$\frac{dm}{m} = \frac{c_{pg}{}^{2} St_o (A_w / A_g)(T - T_b)dT}{\left[\{\varepsilon_b c_{pa}(T_b - T_a) + v.c_{pg}(T - T_b)\}\gamma_a R_a M_o{}^{2} T_o + St_o (A_w / A_g) c_{pg} c_{pa} (T - T_b)(T - T_a)\right]} \quad (12)$$

where dm is the cooling air flow rate supplied at Ta (kg/s), St_o is the Stanton number, A_w is the Wall surface area (m2) and A_g is the gas flow path cross-sectional area (m2), T_b is the allowable blade temperature (K), T_a cooling air temperature (K), R_a is the gas constant of air (kJ/kg K), M_o is the Mach number, ε_b is the heat transfer effectiveness of the blade.

3.6 Heat recovery steam generator (HRSG) model

HRSG is the key component in which the remaining energy of exhaust from the gas turbine is utilized by raising high pressure high temperature steam through a number of heat transfer stages inside the HRSG. The tubes inside the HRSG are arranged in counter flow direction of the gas turbine exhaust gases. Figure 4 shows the temperature distribution in a single pressure HRSG.

The mass of the steam generated from the exhaust of gas turbine is estimated from the energy balance inside the HRSG as:

$$\varepsilon_{HRSG} m_g (h_g - h_{stk}) = m_s q_{HRSG} \quad (13)$$

where ε_{HRSG} is the heat transfer in the HRSG, mg is the mass of gas (kg), hg is the enthalpy of gas at the inlet to the HRSG (kJ/kg), hstk is the enthalpy of gas at the stack conditions (kJ/kg), ms is the mass of steam generated in the HRSG (kg), q_{HRSG} is the heat transfer in the HRSG (kJ/K).

3.7 Steam turbine model

The high pressure and high temperature steam obtained from the HRSG expands to the condenser conditions in the steam turbine. The expansion process through the turbine is treated as adiabatic. The skin friction coefficient and other internal losses are accounted by introducing the concept of isentropic efficiency.

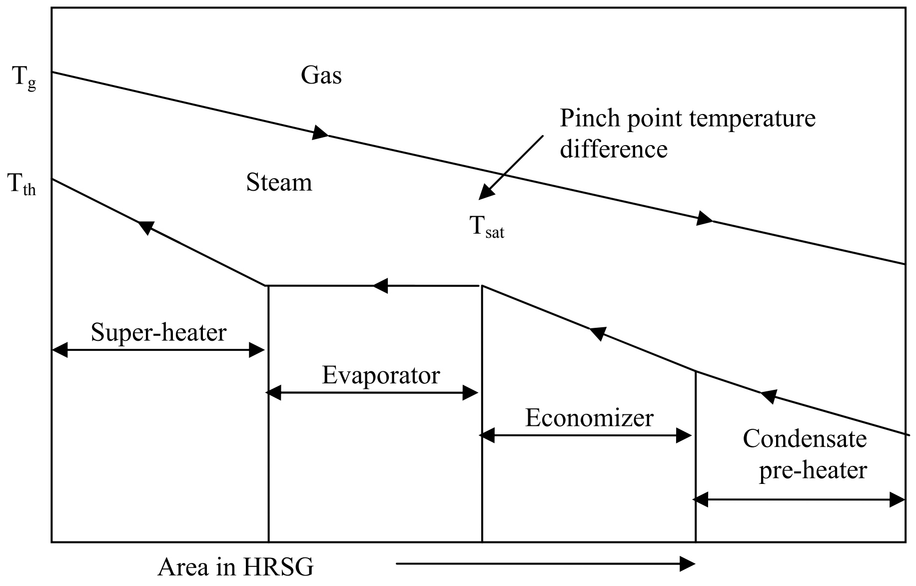

Figure 4. Temperature profile inside single pressure HRSG

Work of steam turbine is obtained in conventional manner by multiplying the mass of steam with enthalpy change during the expansion process.

$$w_{st} = \sum \left[m_s \left\{ (h_{st})_{in} - (h_{st})_{out} \right\} \right] \eta_{mech} \tag{14}$$

where w_{st} is the steam turbine work (kJ), $(h_{st})_{in}$ and $(h_{st})_{out}$ enthalpy in and out of steam turbine (kJ/K), respectively and η_{mech} is the mechanical efficiency.

To remove any dissolved oxygen in the feed-water, a small portion of steam is bled from the steam turbine and supplied to the deaerator. The mass and energy balance equations in the deareator are:

$$m_{d/a,s} + m_{10} = m_s \tag{15}$$

where $m_{d/a,s}$ is the mass of steam bled (kg), m_s is the total mass of steam generated (kg).

$$m_{d/a,s} h_6 + m_{10} h_{10} = m_s h_{11} \tag{16}$$

3.8 Condenser model

In condenser, the steam after expansion condenses by giving its heat of condensation to the circulating cooling medium, which in the present work is taken to be water at atmospheric conditions. The effect of pressure loss of steam in the condenser on the enthalpy of condensate is taken care by introducing the concept of under-cooling of the condensate. The inefficiencies of the condenser are taken care by the term effectiveness of the condenser.

The cooling water flow requirement is estimated from the energy balance of the condenser as:

$$m_{cond,s} \varepsilon_{cond} \left[(h_{cond,s})_{in} - (h_{cond,s})_{out} \right] = m_w c_{pw} (T_{w,out} - T_o) \tag{17}$$

where $m_{cond,s}$ is the mass of steam entering the condenser (kg), ε_{cond} is the heat transfer effectiveness of the condenser, $(h_{cond,s})_{in}$ and $(h_{cond,s})_{out}$ is the enthalpy in and out of the condenser (kJ/K), respectively, m_w is the mass of cooling water (kg), c_{pw} is the specific heat of water at constant pressure (kJ/kg K), $T_{w,out}$ is the temperature of water at the outlet of the condenser (K).

3.9 Pump model

The total pump work is divided into two parts – the condensate extraction pump work and the HRSG feed pump work. The inefficiency of pump due various losses is taken care by overall efficiency of the pump. Also, being an incompressible fluid the specific volume of water is taken as constant and its value is taken at the inlet conditions of the pump. The condensate from the condenser is extracted by the condensate extraction pump and is raised to the steam bled pressure for mixing in the deaerator. The corresponding work is given by:

$$w_{p1} = \frac{(m_s - m_{d/a,s}) v_{f9} (p_{d/a} - p_{cond})}{\eta_p} \tag{18}$$

where w_{p1} is the condensate extraction pump work (kJ/kg), v_{f9} is the specific volume of steam at the inlet to the condensate extraction pump (m3/kg), $p_{d/a}$ is the deaerator pressure (Pa), p_{cond} is the condenser pressure (Pa) and η_p is the pump efficiency.

For the HRSG feed pump, the corresponding work is given by:

$$w_{p2} = \frac{m_s v_{f11} (p_{5a} - p_{d/a})}{\eta_p} \tag{19}$$

where w_{p2} is the HRSG feed pump work (kJ/kg), v_{f11} is the specific volume of steam at the inlet to the HRSG feed pump (m3/kg), p_{5a} is the steam inlet pressure (Pa) and η_p is the pump efficiency.

3.10 Carbon dioxide (CO_2) emissions saving

To calculate the carbon dioxide (CO2) emissions saving, the first step is to calculate the mass of coal that needs to be burnt in the coal-fired thermal power plant to produce the same amount of steam under the same operating conditions as is being produced by utilizing the heat of the exhaust gases of the gas – steam combined cycle power plant. This mass of coal ($m_{f,c}$) can be obtained from the equation given below as:

$$\eta_{boiler} = \frac{m_s \left(h_{\sup} - h_{f1}\right)}{m_{f,c} \times (G.C.V)_{f,c}} \tag{20}$$

where η_{boiler} is the boiler efficiency, $h_{\sup}$ is the enthalpy of superheated steam (kJ/K), h_{f1} is the enthalpy of feed-water and $(G.C.V)_{f,c}$ is the gross calorific value of coal (kJ/kg).

Next step is to calculate the amount of CO2 emission (Q_{CO_2}) associated with the mass of coal burnt. This is given by:

$$Q_{CO_2} = C \times m_{f,c} \times \eta_{c,f} \tag{21}$$

where C is the carbon fraction of the fuel (coal) and it can be obtained from the molecular formula of coal C135H96O9NS and $\eta_{c,f}$ is the combustion efficiency of the fuel. In the similar manner, the CO2 emission associated with natural gas can be obtained. In the case of natural gas to calculate the carbon fraction of the fuel, components other than methane are assumed to be negligible in the gas composition.

4. Results and discussion

Thermodynamic analysis of the combined cycle configuration under consideration has been carried out using the modelling and governing equations developed in previous part. The input data used for the analysis is given in Table 1. Performance curves have been plotted using the results obtained.

To study the effect of inlet air cooling, the range of Tci is selected from 250 – 300 K. The upper range (i.e. 300 K) represents the case of compression without inlet air cooling i.e. air at ambient temperature is compressed.

Figure 5(a) shows the effect of CIT on the blade coolant requirement and the fuel requirement. The mass of cooling air required decreases continuously with decreasing Tci. This is due to the fact that the temperature of cooling air bled from the compressor is lesser as compared to the un-cooled inlet air case. Also, the mass of fuel required increases with decrease in Tci. This is due to the fact that the compressor outlet temperature also decreases, thus requiring more amount of heat.

Figure 5(b) shows the effect of CIT on the work of refrigeration and the compressor work. It can be noted from the figure that the work of refrigeration increases exponentially with decrease in Tci. As expected the work is zero for Tci = 300 K when no inlet air cooling is employed i.e. air at ambient conditions is supplied to the compressor. The compressor specific work decreases with decrease in Tci This is due to the fact that the when the temperature of air is lowered, its density increases which leads to lesser volume of air required for same mass handled.

Figure 5(c) shows the effect of CIT on the plant specific work and plant efficiency. The results show that the plant specific power and plant efficiency both increase continuously with decrease in Tci. The increase in plant specific power with lower Tci is mainly attributed to the decreased compressor work. Here, one thing should be noted that although decreasing Tci upto 275 K or 270 K will yield enhanced specific power yet, there is a danger of ice formation in the compressor suction line. For this reason Tci is taken as 280 K for further analysis.

Table 1. Input data

Atmospheric Air Conditions:	**HRSG and Bottoming Cycle:**
Pressure (p_o) = 1.013 bar	Deaerator pressure = 3 bar
Temperature (T_o) = 300 K	Condenser pressure = 0.05 bar
R_a = 0.2864 kJ/kg K	Steam pressure and temperature for single pressure steam combined cycle = 50 bar, 425 ^oC
γ_a = 1.4	Effectiveness of HRSG (ε_{hrsg}) = 0.95
Refrigeration System:	Isentrpoic efficency of steam turbine (η_{st}) = 88%
Effectiveness of condenser (ε_c) = 0.88	Maximum pinch point temperature difference = 10 ^oC
Effectiveness of evaporator (ε_e) = 0.91	Throttling valve and piping loss in bottoming cycle = 5% of entry pressure
Pressure loss in the evaporator = 2% of entry pressure	Under-cooling in condenser and deaerator of bottoming cycle = 5 ^oC
Refrigeration efficiency (η_r) = 91%	C_p water = 4.2 kJ/kg K
Compressor:	Allowable stack temperature = 170 ^oC
Polytropic efficiency of compressor (η_{pc}) = 89%	Pump efficiency (η_p) = 75%
Allowable stage pressure ratio = 3.15	
Combustion Chamber:	**For CO_2 Emissions Saving:**
Combustion efficiency (η_{cc}) = 99%	Boiler efficiency (η_{boiler})= 82%
Fuel L.C.V. = 45000 kJ/kg	Feed water temperature = 170 ^oC
Pressure loss in combustion chamber = 2% of entry pressure	Gross Calorific Value of coal = 14840 kJ/kg
Gas constant for fuel in gaseous state, R_g = 0.2836 kJ/kg K	Carbon fraction of coal = 0.849948
γ_g = 1.333	Carbon fraction of natural gas = 0.75
	Combustion efficiency of coal = 26%
	Combustion efficiency of natural gas = 80%
Gas Turbine:	
Mach Number (M_o) = 0.6	
Stanton Number (St_o) = 0.005	
Polytropic efficiency of gas turbine (η_{pgt}) = 89%	
A_w/A_g = 4	
Exhaust pressure of gas turbine = 1.08 bar	
Heat transfer effectiveness of blade cooling (ε_b) = 0.95	
Work loss factor due to mixing of cooling air and hot gas (f_m) = 0.995	

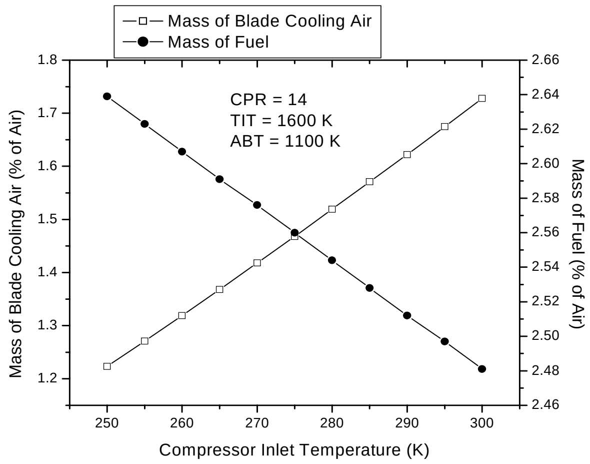

Figure 5(a). Variation of mass of blade cooling air requirement and mass of fuel with CIT

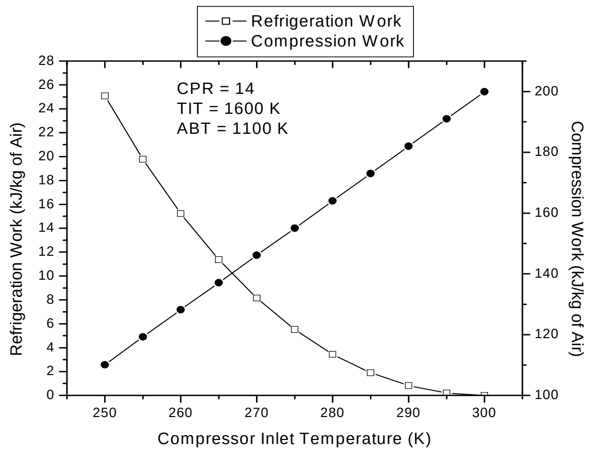

Figure 5(b). Variation of refrigeration work and compression work with CIT

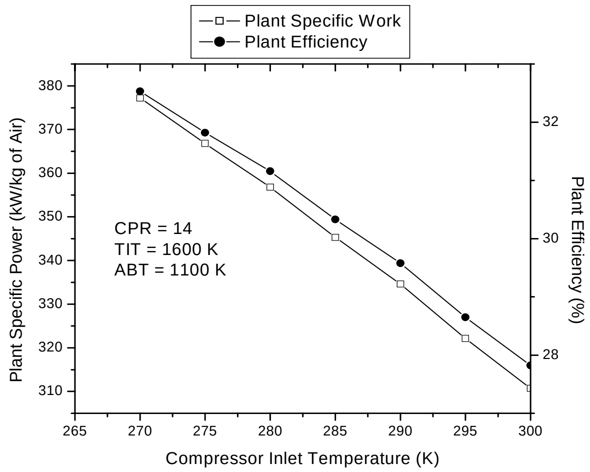

Figure 5(c). Variation of plant specific work and plant efficiency with CIT

Figure 6(a) shows the effect of TIT on the blade coolant requirement. From the figure it can be seen that the cooling air requirement keeps on increasing with increase in T_{ti}, which is very obvious and as expected.

Figure 6(b) shows the effect of TIT on the various specific works involved in the cycle. It is apparent that the work of compression is varying with T_{ti}, since the cooling air mass is being extracted from different compressor stages. Due to increased blade cooling air, the specific work of compressor decreases with increases T_{ti}. From the figure it can be seen that increase in the T_{ti} results in subsequent increase in the gas turbine specific work. As expected, the bottoming cycle specific work increases with increase in T_{ti} because of the increases gas turbine exhaust temperature which eventually increases the energy available in HRSG. The plant specific work increases with increase in T_{ti} because of the increased gas turbine specific work and increased bottoming cycle specific work with increase in T_{ti}. But, here it should be noted that at still higher T_{ti}, the increase in the blade cooling air requirement may offset the increased plant specific work so obtained. Figure 6(c) shows the effect of TIT on the various efficiencies involved in the cycle.

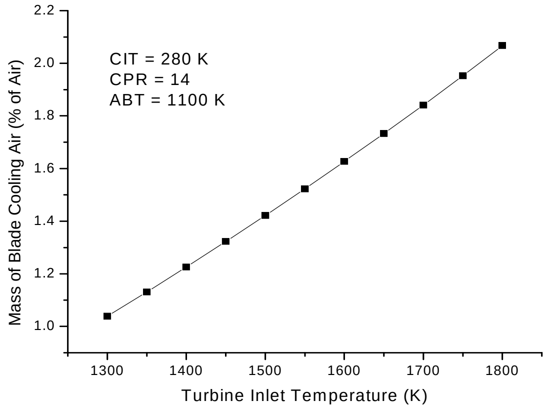

Figure 6(a). Variation of mass of blade cooling air requirement with TIT

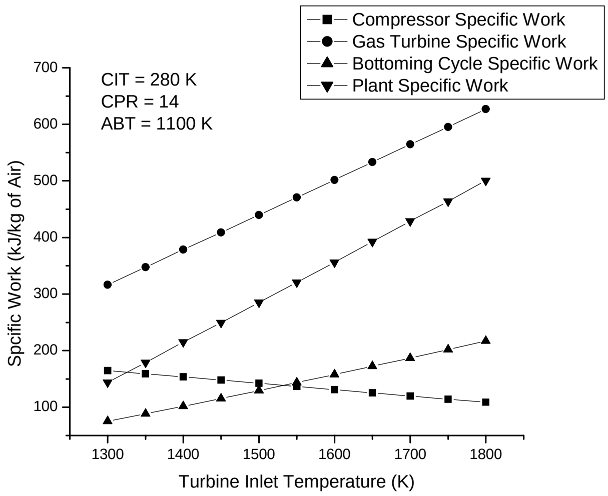

Figure 6(b). Variation of specific works with TIT

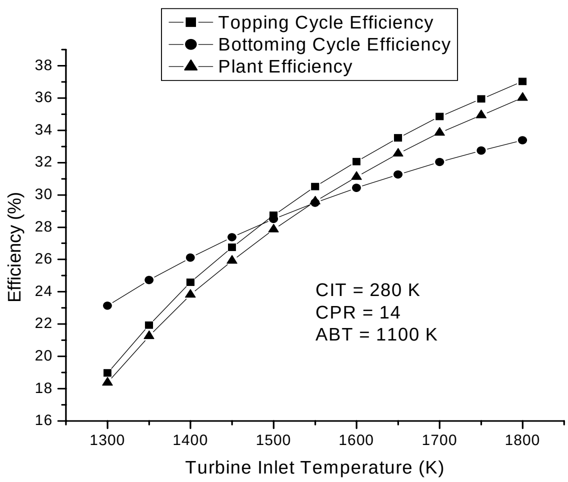

Figure 6(c). Variation of efficiencies with TIT

Figure 7 shows the carbon dioxide (CO2) emissions from the proposed cycle and the coal-fired thermal power plant. For CIT = 280 K, TIT = 1600 K and CPR = 14, the mass of steam utilized by the steam turbine is 4.23 kg/kWh. To produce the same amount of steam at the same generation pressure and temperature (i.e. pressure = 50 bar and temperature = 425 oC), the mass of coal that needs to be burned in a coal-fired thermal power plant is calculated to be 0.8848 kg/kWh which leads to 0.1955 kg/kWh of CO2 emission in the atmosphere. Also, the amount the CO2 emission associated with the fuel (natural gas) burnt in the gas turbine of the gas – steam combined cycle plant to produce the same amount of steam under same operating conditions comes out to be 0.10272 kg/kWh. Here, it can be noted that in the case of cogeneration, the amount of CO2 emission is reduced by 0.09278 kg/kWh when compared to the coal-fired thermal power plant. The above reduction in the CO2 emission is because of the waste heat utilization in the steam cycle instead of the burning of coal to generate steam. It has been found that 0.8 – 0.9 kg/kWh CO2 is emitted in Indian power plants. So, with the cogeneration the net saving in the CO2 emission comes out to be 11.60% per kWh.

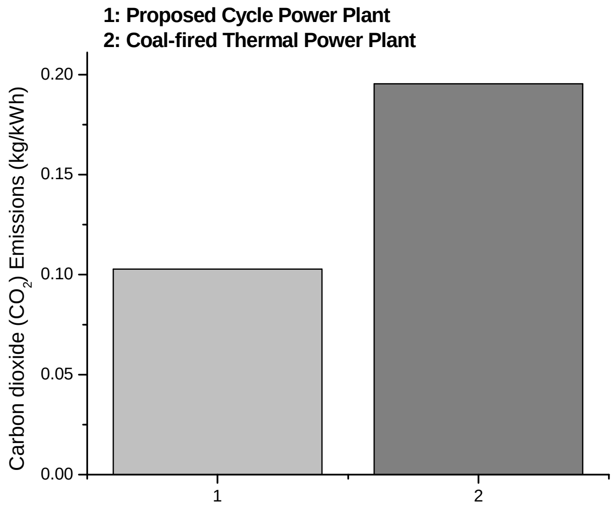

Figure 7. Carbon dioxide (CO_2) Emissions from present cycle and coal-fired thermal power plant

5. Conclusion

Based on the results obtained, it can be concluded that the performance of the gas – steam combined cycle can be improved by lowering the compressor inlet temperature and/or increasing the turbine inlet temperature. With compressor inlet air cooling by 10 degrees from 300 K the plant specific power is found to increase by 7.69% and the plant efficiency improves by 6.33%. An increase of 100 degrees in the turbine inlet temperature results in a sharp increase in the blade cooling air requirement. For an increase of 50 degrees from 1600 K in TIT, the increase in the plant specific work and the plant efficiency is 10.22% and 4.66%, respectively. This increase in TIT is limited by the blade cooling technique involved and the material of the turbine blade. With the proposed cycle being employed for cogeneration, there is a significant saving in the amount of CO_2 emitted by the coal-fired thermal power plants around 11.60% per kWh of the power generated by burning of coal. Thus, this combined cycle can be looked upon as a good option for reduction in the level of green house gases (GHGs) in the atmosphere.

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