EFFECT OF AMBIENT TEMPERATURE ON THE PERFORMANCE OF A COMBINED CYCLE POWER PLANT

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ABSTRACT
The aim of the present paper is to examine the effect of ambient temperature on the performance of a combined cycle power plant. For this work, the combined cycle plant chosen is NTPC (National Thermal Power Corporation) Dadri, India where a gas unit of 817 MW is installed. The effect of ambient temperature on combined cycle efficiency, gas turbine cycle efficiency, exergy destruction in different components, exergy loss via exhaust and air fuel ratio at lower and higher turbine inlet temperature are reported. The results show that the net decrease in combined cycle efficiency is 0.04% and the variation in exergy destruction of different plant components is up to 0.35% for every °C rise in ambient temperature.

Keywords: ambient temperature; gas turbine; steam turbine; combined cycle; exergy destruction.

L'EFFET DE LA TEMPÉRATURE AMBIANTE SUR LA PERFORMANCE D'UNE CENTRALE À CYCLES COMBINÉS

RÉSUMÉ
Le but de cet article est d'examiner l'effet de la température ambiante sur la performance d'une centrale à cycles combinés. À cette fin, la centrale choisie est le NTPC (National Thermal Power Corporation) Dadi, Inde, où une unité au gaz de 817 MW est installée. L'effet de la température ambiante sur l'efficacité des cycles combinés, l'efficacité du cycle de la turbine au gaz, de la destruction de l'exergie dans différents éléments de la centrale, de la perte de l'exergie via l'échappement, et du rapport du mélange air-carburant, ainsi que la température d'entrée plus élevée, est aussi transmise. Les résultats montrent que la réduction nette de l'efficacité des cycles combinés est de 0.04%, et la variation de la destruction de l'exergie dans les différents éléments de la centrale s'élève jusqu'à 0.35% pour chaque hausse de °C dans la température ambiante.

Mots-clés : température ambiante ; turbine au gaz ; turbine à la vapeur ; cycle combiné ; destruction de l'exergie.

A/F	air-fuel ratio
C_p	specific heat; kJ/kg k
f	fuel flow rate; kg/s
h	specific enthalpy; kJ/kg
HRSG	heat recovery steam generator
I	exergy; kw
M	mass flow rate of steam per cycle
m	mass flow rate; kg/s
P	pressure; bar
Q	energy; kw
s	specific entropy; kJ/kg
T	temperature; °C
TIT	turbine inlet temperature
W	work output; kw
Greek symbols	
η	efficiency
Subscripts	
0	ambient conditions
a	air
C	compressor
CC	combustion chamber
Comb	combined cycle
Exh	exhaust gas
f	fuel
g	gas
GT	gas turbine
GTC	gas turbine cycle
s	steam
ST	steam turbine

1. INTRODUCTION

The location of power plant plays a major role on its performance. The atmospheric air which enters the compressor gets hotter after compression and goes to combustion chamber. The effect of ambient temperature has been reported by several authors and more recently by De Sa and Zubaidy [1] that for every K rise in ambient temperature above ISO condition the gas turbine loses 0.1% in terms of thermal efficiency and 1.47 MW of its gross (useful) power output. While in the same year, Ibrahim et al. [2] reported that when ambient temperature increases from 273 to 333 K, the total power output increases about 7% for all configurations except the regenerative gas turbine. The overall thermal efficiency of the combined cycle obtained a maximum value with regenerative gas turbine configuration of about 62.8% at ambient temperature 273 K and the minimum value of the overall thermal efficiency was about 53% for intercooler gas turbine configuration at ambient temperature 333K. The gas turbine combined cycle inlet air cooling was studied by Yang et al. [3] and the effect of relative humidity and ambient temperature on gas turbine combined cycle inlet air cooling (GTCCIAC) efficiency ratio was shown. When relative humidity (RH) remains the same, GTCCIAC efficiency ratio increases with the rise of ambient air temperature (t_a). It is worth pointing out that GTCCIAC efficiency decreases in the case of $t_a < 15$°C due to the inlet air temperature characteristics of GTCC.

More precisely, it was shown by Polyzakis et al. [4] that as the ambient temperature increases, the air density falls. Hence, for a given TIT, the mass flow through the engine is reduced. As a consequence, the engine output power is lower. While the ambient temperature rises, the net power generated in the combined-cycle thermal-plant decreases in spite of the use of the maximum supplementary-firing. Arrieta and Lora [5] registered that with a gas temperature of 675°C after the supplementary firing, the net electric-power varies in the range from 640 to 540 MW when the ambient temperature varies between 0 and 35°C. While the ambient temperature rises, the combined cycle thermal plant's heat-rate increases (that is, the efficiency decreases), in spite of the use of the minimum supplementary-firing temperature. The heat rate is even greater when the maximum supplementary-firing temperature is used. Sanaye and Tahani [6] discussed the effects of evaporative cooling on gas turbine performance. Evaporative cooling process occurs in both compressor inlet duct (inlet fogging) and inside the compressor (wet compression). By predicting the reduction in compressor discharge in air temperature, the modeling results were compared with the corresponding results reported in literature and an acceptable difference percent point was found in this comparison. Sue and Chuange [7] showed that the location of the power station plays an important role on its performance. The power output of a gas turbine increases as the inlet air temperature decreases.

Although sufficient literature is available on this topic, none author has reported in a simple and systematic manner about exergy destruction in different components of simple combined cycle power plant specially situated in Indian climatic conditions. Tiwari et al. [8] have carried out a detailed analysis of NTPC Dadri station India. The effect of ambient temperature is discussed in this paper.

2. SYSTEM DESCRIPTION

The schematic diagram of combined cycle is shown in Fig. 1. The gas turbine (Siemens AG, Germany, V 94.2, 131.3 MW, 2 No.) is shown as a topping plant, which forms a high temperature loop, whereas the steam turbine (BHEL India, 146.5 MW, two cylinder condensing reaction, 2×22 no of stages HP) forms the low temperature loop. The connecting link between the two cycles is the heat recovery steam generator (BHEL Trichy, Vertical forced Circulation) working on the exhaust of the gas turbine. A gas turbine cycle consists of an air compressor (Siemens KWU, Type Multistage axial flow, number of stages is 16), a combustion chamber (Vertical Silo Type) and a gas turbine. The turbine's exhaust gas goes to a heat-recovery steam-generator to generate superheated steam. That steam is used in a standard steam power cycle, which consists of a steam turbine, a condenser (BHEL, Rectangular, SCD2-1200) and a feed pump (BHEL Hyderabad, Vertical Mixed Flow). Both the gas and steam turbines drive electric generators (G1 & G2) [8].

The atmospheric air enters into compressor at state point '1' and after compression exit from state point '2'. The pressurized hot air enters into combustion chamber where combustion of fuel (natural gas) takes place and gases are generated. These high pressure high temperature (900–1400°C) gases enter into gas turbine at state point '3' and after doing useful work leave from state point '4'. The exit gas (500–600°C) from the gas turbine has still some energy for doing useful work, which is recovered in HRSG. The superheated steam (60 bar) generated in HRSG enters into the steam turbine at state point 'a' for further expansion. The exit steam from state point 'b' is condensed into a condenser up to a pressure of 0.88 bar of state point 'c'. Finally, the feed pump circulates the feed water to HRSG from state points 'c' to 'd'. In this way, the combined cycle executes and produces useful work. The exergy destruction (I) due to irreversibility in flow of different components is shown in Fig. 1 and exergy loss via exhaust is also taken into account.

3. THERMODYNAMIC ANALYSIS

The temperature–entropy (T–s) diagram of Brayton–Rankine combined power cycle shown in Fig. 2 is based on the schematic diagram shown in Fig. 1. The cyclic process 1-2-3-4-1 forms the Brayton cycle. The process 1-2 is adiabatic compression of air in compressor and 1-2s is isentropic compression. The process 2-3 is heat addition in combustion chamber with pressure drop from P2 to P3. Process 3-4 is the adiabatic

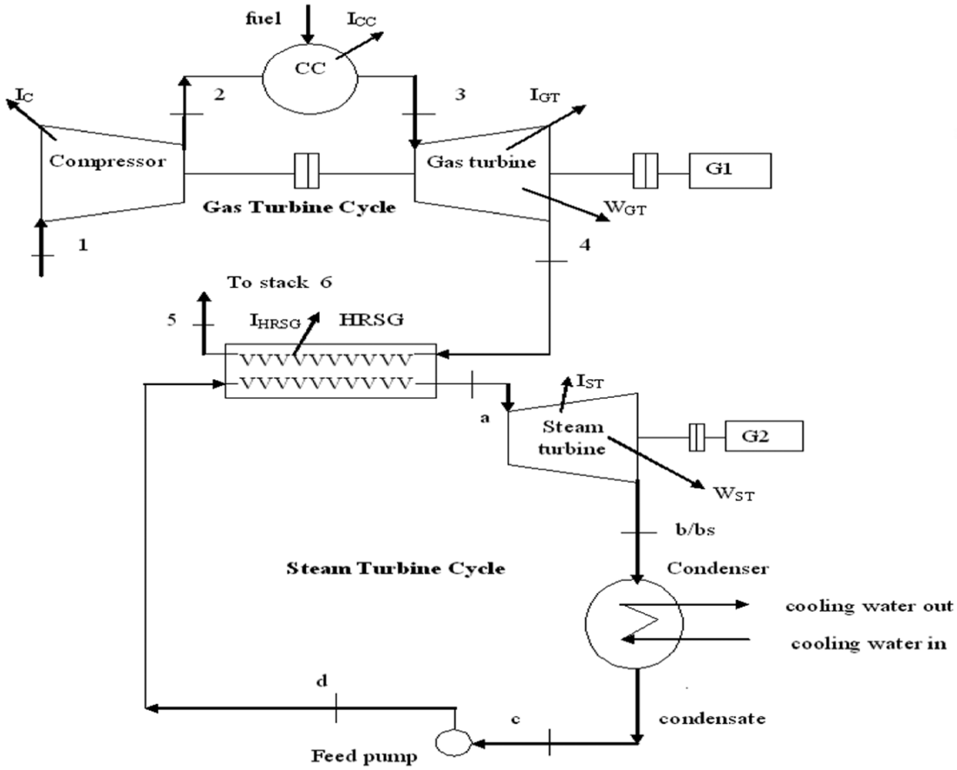

Fig. 1. Schematic diagram of combined gas/steam power cycle.

expansion in gas turbine. Process 4-5 is exhaust gas interaction in HRSG with pressure drop from P4 to P1. Point '5' shows the gas temperature leaving the evaporator section of HRSG and point '6' shows the stack gas temperature.

The cyclic process 'a-b-c-d-a' forms the Rankine cycle. The process a-b is the adiabatic expansion of steam in steam turbine and a-bs is isentropic expansion. The process b-c is condensation of exhaust steam in condenser and c-d is isentropic compression of water in feed pump. The point 'e' is the feed water temperature to the HRSG and point 'f' is the saturation temperature corresponding to steam pressure in the evaporator section.

If the flow rate of combustion gas is 1 kg/s and that of fuel is f kg/s, then the flow of air is $(1-f)$ kg/s and the flow of fuel is given by

$$f = \frac{(c_{pg}T_3 - c_{pa}T_2)}{(cv - c_{pa}T_2)} \tag{1}$$

$$\text{air-fuel ratio} = \frac{1-f}{f} \tag{2}$$

By energy balance, the mass flow rate of steam is

$$M_s = \frac{c_{pg}(T_4 - T_5)}{(h_a - h_f)} \tag{3}$$

Let the total heat transfer in the HRSG yield the stack temperature T_6. Then we obtain

$$T_6 - T_4 = \frac{M_s(h_a - h_e)}{c_{pg}} \tag{4}$$

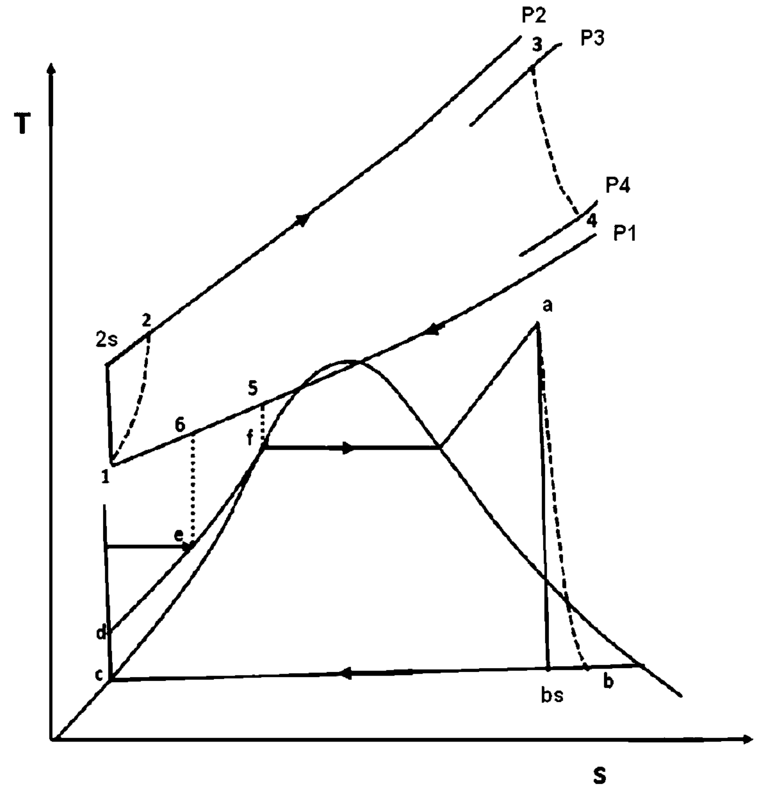

Fig. 2. Temperature–entropy (T–s) diagram of Brayton–Rankine combined power cycle [8].

The work output of steam turbine cycle is given by

$$W_{ST} = m_s(h_a - h_{bs})\eta_{st} \tag{5}$$

The mass flow rate of gas in the cycle is

$$m_g = \frac{m_s}{M_s} \tag{6}$$

The mass flow rate of air in compressor is

$$m_a = (1-f)m_g \tag{7}$$

So the work output of gas turbine cycle is

$$W_{GT} = m_g c_{pg}(T_3 - T_4) - m_a c_{pa}(T_2 - T_0) \tag{8}$$

The mass flow rate of fuel in combustion chamber is

$$m_f = f \cdot m_g \tag{9}$$

And the combined cycle work is

$$W_{Comb} = W_{ST} + W_{GT} \tag{10}$$

The first law efficiency of combined cycle is

$$\eta_{1,Comb} = \frac{W_{Comb}}{m_f \cdot cv} \tag{11}$$

The efficiency of gas turbine cycle is

$$\eta_{GTC} = \frac{W_{GT}}{m_f \cdot cv} \tag{12}$$

The exergy destruction due to irreversibility in different components is the following:

Compressor:

$$I_C = m_a T_0 (s_2 - s_1) \tag{13}$$

Combustion chamber:

$$I_{CC} = T_0 \left[\left(m_g C_{pg} \ln \frac{T_3}{T_0} - m_g R_g \ln \frac{P_3}{P_0} \right) - \left(m_a C_{pa} \ln \frac{T_2}{T_0} - m_a R_a \ln \frac{P_2}{P_0} \right) + \Delta S_0 \right] \tag{14}$$

Gas turbine:

$$I_{GT} = m_g T_0 (S_4 - S_3) \tag{15}$$

HRSG:

$$I_{HRSG} = T_0 [m_s (s_a - s_e) + m_g (s_6 - s_4)] \tag{16}$$

Steam turbine:

$$I_{ST} = m_s (s_b - s_a) T_0 \tag{17}$$

Exhaust loss: The exergy loss via exhaust is given by

$$I_{Exh} = \int_{T_6}^{T_0} \left(1 - \frac{T_0}{T} \right) dQ \tag{18}$$

The Carnot efficiency is

$$\eta_{Carnot} = \left(1 - \frac{T_0}{T_3} \right) \tag{19}$$

The energy input to the combustion chamber is

$$Q = m_f \cdot cv \tag{20}$$

The exergy input to the combustion chamber is

$$I = Q \cdot \eta_{Carnot} \tag{21}$$

Entropy change is

$$\Delta S_0 = I - Q \tag{22}$$

The second law efficiency of the combined cycle is

$$\eta_{2,Comb} = \frac{W_{GT} + W_{ST}}{I} \tag{23}$$

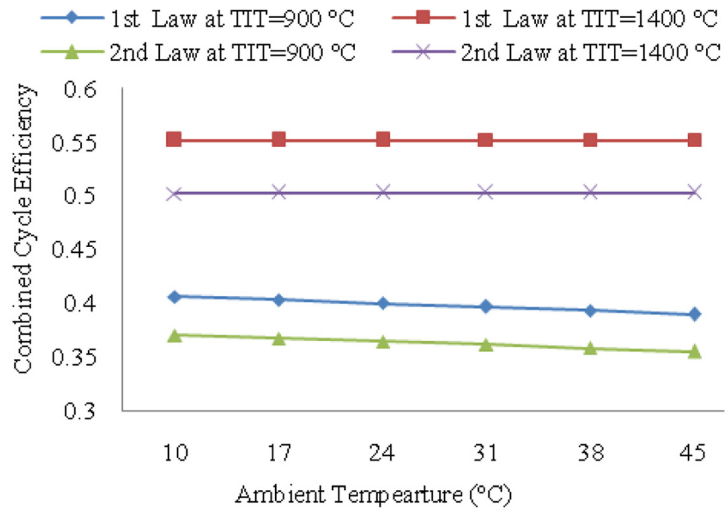

Fig. 3. Variations in combined cycle efficiency.

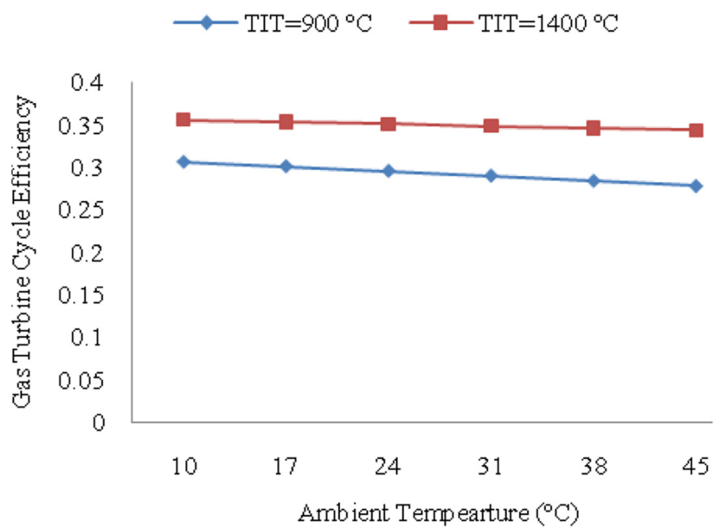

Fig. 4. Variations in gas turbine cycle efficiency.

4. RESULTS AND DISCUSSION

Based upon the methodology developed and the equations derived here, the effect of ambient temperature on the performance of combined cycle and exergy destruction in different components of gas/steam combined power cycle is shown graphically.

In Fig. 3, the variation in the first and second law efficiency of the combined cycle with variations in ambient temperature is shown. It is clear from the graph that as the ambient temperature increases from 10 to 45°C, the first and second law efficiency decreases because more fuel consumption is required in the combustion chamber with the increase in ambient temperature. At a turbine inlet temperature of 900°C, the variation is slightly greater than the turbine inlet temperature of 1400°C. There is little influence at 1400°C TIT on the first and second law efficiency because the higher TIT is dominating the influence of ambient temperature. The net decrease in efficiency is found to be 0.04% for every °C rise in ambient temperature for both first and second law. The maximum first law efficiency reaches up to 55%.

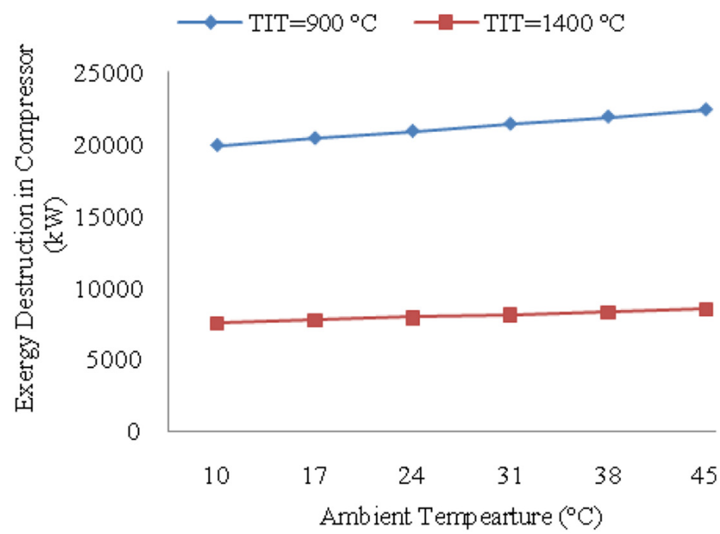

Fig. 5. Exergy destruction in the compressor.

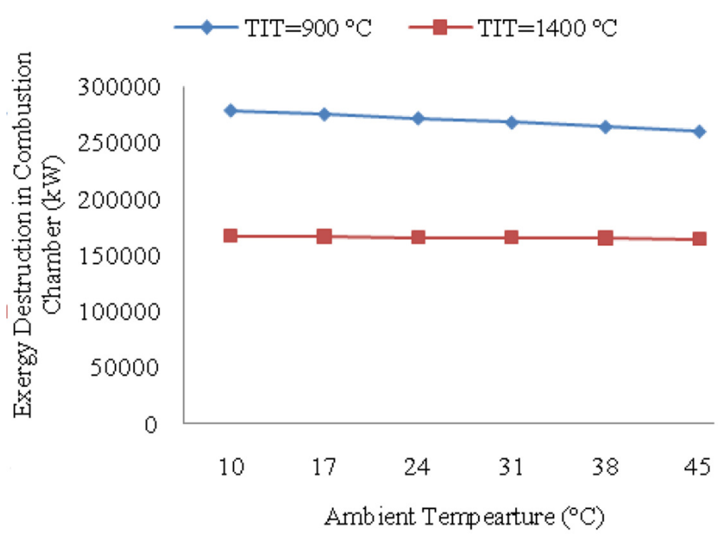

Fig. 6. Exergy destruction in the combustion chamber.

In Fig. 4, the effect of ambient temperature on the efficiency of gas turbine cycle is shown. The efficiency of gas turbine cycle decreases with increase in ambient temperature. The reason is that the net power output of gas turbine cycle decreases because of increase in compressor work. The mass flow rate of gases is also reduced.

At low turbine inlet temperature, the efficiency decreases more rapidly than higher turbine inlet temperature. The net decrease in efficiency is 0.07% at lower TIT and 0.03% at higher TIT for every °C rise in ambient temperature. The maximum efficiency of gas turbine cycle is found to be 35%. In Fig. 5, the effect of ambient temperature on exergy destruction in the compressor is shown at low and higher TIT. The exergy destruction increases with increase in ambient temperature. The reason is that the temperature of air leaving the compressor increases with the increase in ambient temperature so the irreversibility increases. The net increase in exergy destruction is 0.35% for every °C rise in ambient temperature at lower and higher TIT.

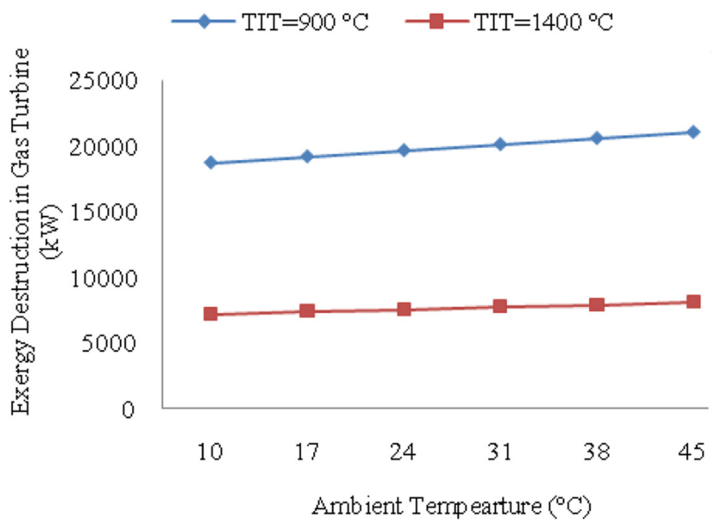

Fig. 7. Exergy destruction in the gas turbine.

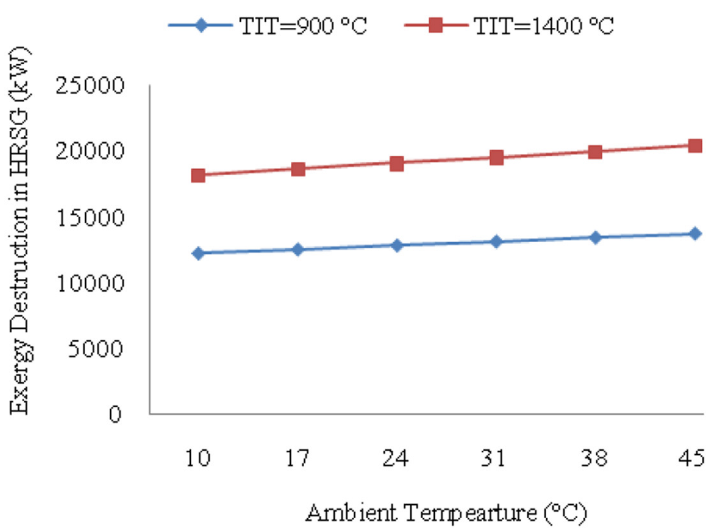

Fig. 8. Exergy destruction in the HRSG.

In Fig. 6, the effect of ambient temperature on exergy destruction in combustion chamber is shown. The exergy destruction reduces with the rise in ambient temperature.

The high pressure and high temperature air leaving from compressor become hotter with the ambient temperture entering into the combustion chamber while gases leaving from combustion chamber are also at higher temperture thereby reducing the irreversibility. At higher and lower TIT, the exergy destruction reduces by 0.34 and 0.21% respectively for every °C rise in ambient temperature.

In Fig. 7, the exergy destruction variation in gas turbine is shown. For any particular TIT, the exergy destruction increases with ambient temperature. The reason is that the exhaust gas temperature increses with ambient temperature resulting in higher irreversibility. So the exergy destruction is 0.32% increasing for every °C rise in ambient temperature at both lower and higher TIT.

In Fig. 8, the variation of exergy destruction in the HRSG is shown. The exergy destruction increases with ambient temperature because of increase in exhaust gas temperature from gas turbine resulting in higher

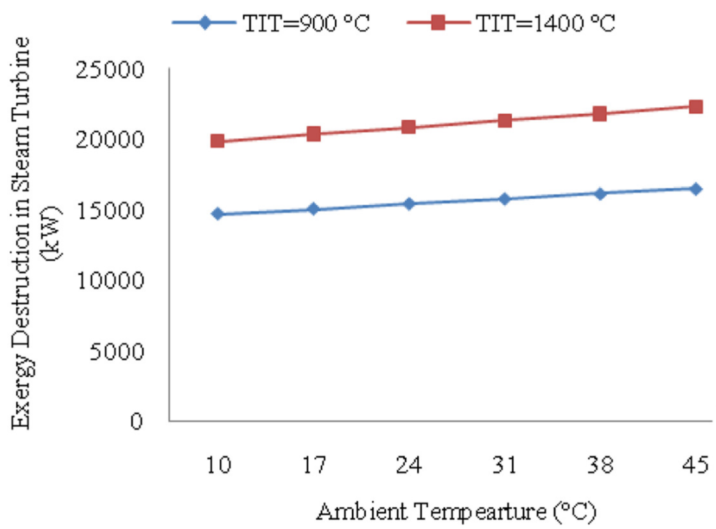

Fig. 9. Exergy destruction in the steam turbine.

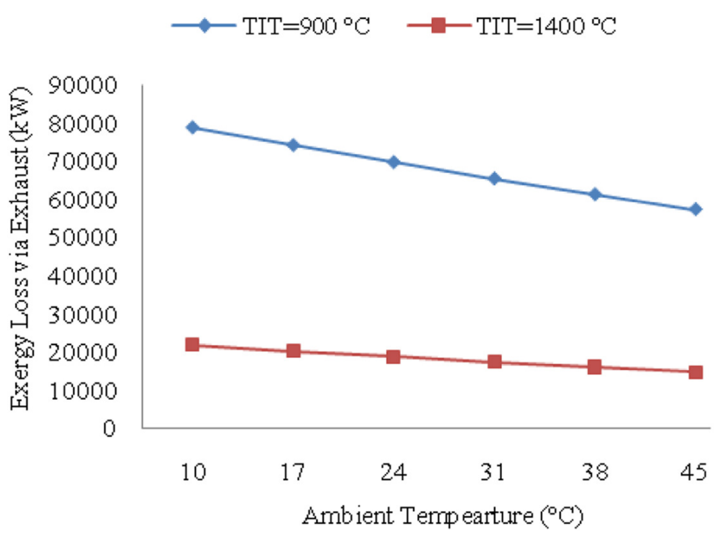

Fig. 10. Exergy loss via the exhaust.

irreversibility. 0.35% exergy destruction is found to increase for every °C rise in ambient temperature at lower and higher TIT.

In Fig. 9, the exergy destruction variation in steam turbine is shown. It is very difficult to predict the effect of ambient temperature on the exergy destruction of steam turbine. It is clear from the graph that as the ambient temperature increases the exergy destruction also increases. The exergy destruction is found to be 0.35 and 0.32% increase for every °C rise in ambient temperature at lower and higher TIT respectively.

In Fig. 10, the variation of exergy loss via the exhaust is given. The exhaust gases are leaving at higher temperature while the ambient temperature is also increasing in reducing the irreversibility. There is a 0.92% fall in exergy loss calculated at lower TIT and 1.14% at higher TIT for every °C rise in ambient temperature.

In Fig. 11, the effect of ambient temperature on air fuel ratio is shown. When hot air (with increase in ambient temperature) enters inside the combustion chamber the requirement of fuel flow rate increses so

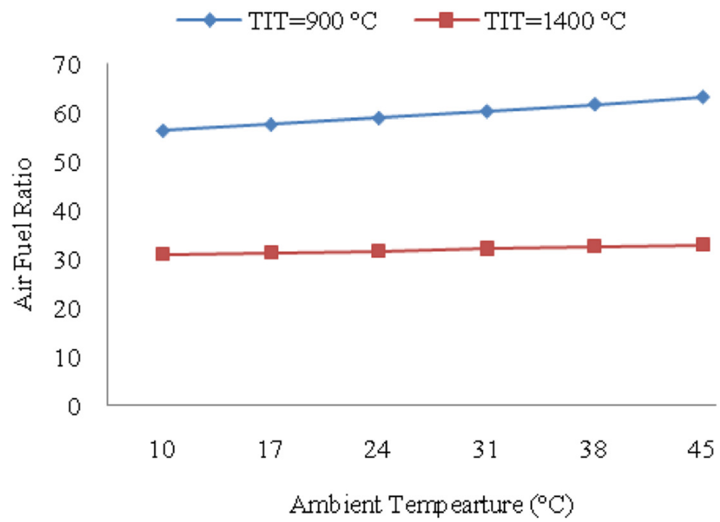

Fig. 11. Variations in the air-fuel ratio.

more air is drawn into the combustion chamber thereby incresing the A/F ratio. There is 2.3% of A/F ratio increasing at lower TIT and 0.17% A/F at higher TIT for every °C rise in ambient temperature.

5. CONCLUSIONS

On the basis of the above analysis, the following conclusions can be made by varying the ambient temperature:

1. The combined cycle loses its efficiency by about 0.04% for every °C rise in ambient temperature.
2. The gas turbine cycle efficiency decreases by 0.03 to 0.07% for every °C rise in ambient temperature.
3. The exergy destruction in combustion chamber decreases with an increase in ambient temperature from 0.21 to 0.34% for every °C rise in ambient temperature.
4. The exergy destruction in compressor, gas turbine, HRSG and steam turbine increases with an increase in ambient temperature from 0.32 to 0.35% for every °C rise in ambient temperature.
5. The ambient temperature also affects the exergy loss via exhaust by 0.92 to 1.14% for every °C rise in ambient temperature.
6. The air-fuel ratio increases with increase in ambient temperature.

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