HEFAT2008
6th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
30 June to 2 July 2008
Pretoria, South Africa
Paper number: MN1

IMPROVING COMBINED CYCLE POWER PLANT PERFORMANCE IN ARID REGIONS

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ABSTRACT

In arid regions, where cooling water supplies are limited and highly regulated, combined cycle power plants (CCPPs) are turning to air-cooled condensers instead of water-cooled condensers. However, performance of air-cooled condensers can decline as ambient temperatures increase and result in loss of steam turbine power output. At the same time, as ambient temperature rises, net output of the gas turbine also can decline due to increased power consumption by the compressor and reduced power output by the gas turbine. In this paper, it is proposed to remedy these problems by pre-cooling the inlet air to the air-cooled condenser as well as the compressor, using a low-temperature thermal energy storage (TES) system. The TES is maintained around 5ºC by an absorption refrigeration system driven by the waste heat in the stack gases. A thermodynamic analysis of a 500-MW CCPP incorporating the above concept is presented.

INTRODUCTION

Two factors that can impair the performance of combined cycle power plants (CCPPs) in arid regions are addressed in this paper. The first one has direct impact on the steam turbine performance while the other has direct impact on gas turbine performance.

Impact on Steam turbine performance

Condensers in CCPPs reject heat by evaporative cooling in water-cooled recycling cooling towers, where about 2% of the recirculating cooling water is lost to evaporation and drift; and about 0.4% is continuously discharged to avoid build-up of impurities. Makeup water has to be added continuously to replenish that lost to evaporation, drift, and blow-down, amounting to about 1,363 L per MW-hr of power generated. In a 500-MW plant, for example, this demand totals to about 15 x 10^6 L/day, enough to provide the potable water demand of a population of 40,000.

NOMENCLATURE

C_p	[kJ/kg-K]	Specific heat at constant pressure
h	[kJ/kg]	Specific enthalpy
K	[kJ/hr-K]	Heat loss coefficient
m	[kg/hr]	Mass flow rate
Q	[kJ/hr]	Heat transfer rate
T	[K]	Temperature
V	[m^3]	Volume
W	[kJ/hr]	Work transfer rate
Special characters		
ΔT	[K]	Rise in temperature
ρ	[kg/m^3]	Density
Subscripts		
a1		Air flow through air compressor
a2		Air flow through air-cooled condenser
amb		Ambient
r		Refrigerant
s		Steam
w		Water
AB		Absorber
AC		Air compressor
CC		Combustion chamber
CO		Condenser
EV		Evaporator
GE		Generator
GT		Gas turbine
HE		Heat exchanger
ST		Steam turbine
TES		Thermal energy storage

Shortages of adequate water sources in arid regions and increasing regulatory restrictions are therefore forcing utilities to seek alternate technologies for heat rejection (Lees, 1995; Streng, 1998). In the US for example, the Environmental Protection Agency has recently proposed that power plants that require more than 7.6x10^6 L/day of water for cooling (equivalent plant capacity > 250 MW) must consider alternate technologies to determine the Best Available Technology (BET) for rejecting the waste heat.

Dry-cooling by air-cooled condensers (ACCs) is emerging as an alternative to the wet-cooling technology, where the waste heat is rejected to air, directly or indirectly. In addition to conserving limited water resources, ACCs are not plagued by many of the drawbacks associated with wet-cooling such as plume formation, brine disposal and legionella health risks. Dry-cooled plants also offer flexibility in plant siting (Conradie & Kroger, 1996); they can be located closer to load centers rather than to cooling water sources, saving transmission losses and improving reliability and security.

However, since ACCs operate with the ambient dry bulb temperature as the theoretical minimum attainable temperature, their efficiency can drop by about 10% when ambient temperatures rise (Gupta & Gorton 1973; Leung 1973).

Impact on gas turbine performance

When ambient temperature increases, density of the inlet air and hence the mass flow rate through the turbine decreases resulting in a drop in power output. At the same time, the compressor power increases due to the divergence of the constant pressure lines in the T-s diagram (Ameri et al., 2007). If the firing temperature is kept constant, the lower pressure of the compressor outlet results in further reduction of the turbine power output. Typically, the net power decreases by 5-9% of the ISO-rated power for every 10ºC increase above the 15ºC ISO standard (Zaki et al., 2007).

Together, the above factors can lead to significant loss of net power output during hot summer days. This problem is further deteriorate compounded by the fact that demand for power is highest also at the same time. Currently available or proposed remedies for mitigating the negative impacts of the above factors on the performance of the ACC and the gas turbine are summarized below.

Minimizing impact on ACC

Methods that have been used or proposed for minimizing the impact of high ambient temperature on ACCs include: 1) raising the steam turbine back pressure to increase the steam condensing temperature; 2) increasing the air flow through the ACC; 3) spraying water into the inlet of ACCs; 4) pre-cooling the air inflow to the ACC.

In the first method, raising the back pressure of the steam turbine increases the temperature of the exhaust steam to maintain the temperature differential in the cooling tower as ambient temperature rises. But, raising the back pressure reduces the enthalpy drop in the steam turbine and hence its power output.

In the second method, additional energy is utilized to provide higher air flow and the fans have to be oversized. In the third method, evaporative cooling of the inlet air is achieved by spraying water into the air inlet of ACCs at a rate of about 24 to 28 L/min of water per MW. As such, the cooling tower no longer qualifies as a “true” air-cooled tower (Larinoff, 1973).

In the fourth approach, a chilled-water thermal energy storage (TES) system is used to pre-cool the inflow air to the ACC when the ambient temperature increases above the design air inlet temperature. The temperature of the TES system is maintained by an absorption refrigeration system (ARS) driven by waste heat from the stack gases of the CCPP (Gadhamshetty et al., 2006).

Minimizing impact on gas turbine

Methods to minimize the impact of high ambient temperature on gas turbine output have focused on pre-cooling the inlet air. Some of the pre-cooling methods that have been adapted include evaporative cooling by water spraying (with or without media) or fogging at the inlet; and, chilling the air intake. The evaporative cooling methods can lower the inlet temperature to the ambient wet bulb temperature.

This technique has been shown to improve net power output of turbines by about 0.5 to 0.9% for every degree ºC drop in the compressor air inlet temperature. For example, Ameri et al (2007) have reported 9 to 14.5% increase in power output by evaporative cooling at three sites ranging in power output from 8 to 11 MW. This approach typically uses 0.007 kg water per kg of air, which has to be mineralized and pressurized for use in spray and fogging systems. Raw water can be used in the case of media evaporative systems.

In contrast, non-evaporative cooling methods using refrigeration systems can lower the air inlet temperature to any value and do not consume any water. Both mechanical and absorption refrigeration systems have been proposed with chilled-water cooling coils or with thermal energy storage systems (Zurigate et al., 2006).

Another innovative system that has been recently proposed where, a fraction of the air is extracted from the compressor at an intermediate pressure and cooled and expanded to obtain a cool air stream. This stream is then mixed with the ambient air inflow to effectively reduce the compressor inlet temperature. Simulation of this system has shown that inlet temperatures can be reduced below ISO standards and power output can be increased by 19% with slight loss in thermal efficiency (Zaki et al. 2007).

Several recent reports have summarized the merits and demerits of the available methods to minimize the adverse impacts of high ambient temperatures (Zurigate et al., 2006; Ameri et al., 2007; Zaki et al., 2007; Kumar et al., 2007).

PROPOSED SYSTEM

In the proposed system, a sensible heat, low temperature thermal energy storage (TES) system is used to pre-cool the ambient air supply to the air compressor (AC) and the air-cooled condenser (ACC) in a pre-cooler (PC). The TES system is maintained at 5ºC by a LiBr-H_2O absorption refrigeration system (ARS). The heat input to the generator of the ARS is provided by the stack gases via a heat exchanger (HE) while TES is maintained at the specified temperature by the evaporator of the ARS. A chilled water pump (CWP) circulates water between the TES and the PC, programmed to turn on when the ambient temperature exceeds the design inlet temperature to the ACC or the compressor. Thus, the propsoed system does not use any water, and requires only minimal extra energy for the transfer pumps. A schematic of this system is shown in Figure 1 and the details of the ARS are shown in Figure 2.

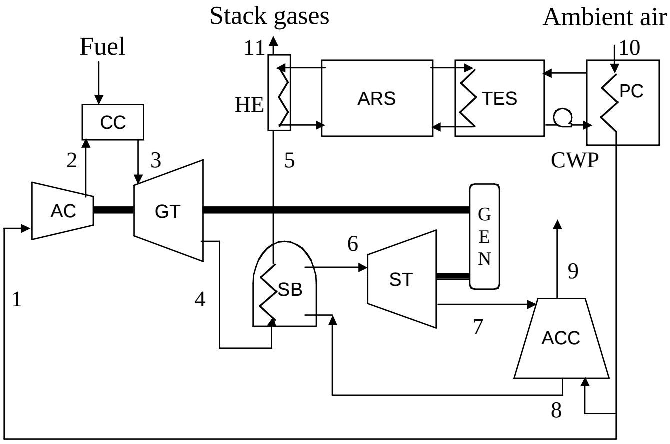

Figure 1. Schematic of the proposed system

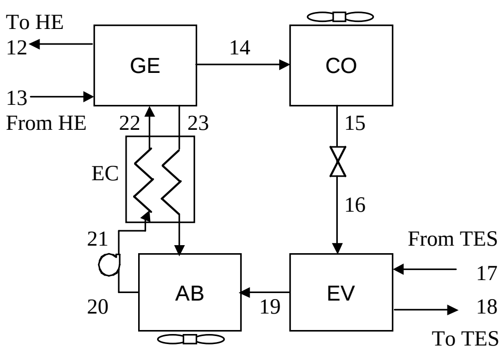

Figure 2. Details of the absorption refrigeration system

MODELING OF THE PROPOSED SYSTEM

The TES system is designed to maintain constant air temperature at the inlet to the compressor and to the air-cooled condenser of the CCPP. As such, the CCPP and the ARS are assumed to operate under steady state conditions while the temperature of the TES tank will vary in response to the ambient temperature. Hence, steady state mass and energy balances are used in the development of the mathematical model for CCPP and ARS.

Considering the CCPP, the following energy balance equations can be written for the air compressor (AC), combustion chamber (CC), gas turbine (GT), and the steam turbine (ST):

$$W_{AC} = m_{a1}(h_2 - h_1) \tag{1}$$

$$Q_{CC} = m_{a1}(h_3 - h_2) \tag{2}$$

$$W_{GT} = m_{a1}(h_4 - h_3) \tag{3}$$

$$W_{ST} = m_s(h_6 - h_7) \tag{4}$$

$$W_{net} = W_{GT} + W_{ST} - W_{AC} \tag{5}$$

where W and Q are the work and heat transfers; h is the specific enthalpy; and m is the mass flow rate. Similarly, considering the ARS, the following energy balance equations can be written for the generator (GE), condenser (CO),evaporator (EV), and the absorber (AB):

$$Q_{GE} = m_{r,14}h_{14} + m_{r,23}h_{23} - m_{r,22}h_{22} \tag{6}$$

$$Q_{CO} = m_{r,14}(h_{14} - h_{15}) \tag{7}$$

$$Q_{EV} = m_{r,14}(h_{19} - h_{16}) \tag{8}$$

$$Q_{AB} = m_{14}h_{19} + m_{24}h_{24} - m_{22}h_{20} \tag{9}$$

Considering the steam boiler SB,

$$m_{a1}(h_4 - h_5) = m_s(h_6 - h_8) \tag{10}$$

Considering the heat transferred from the stack gases in SHE to the generator of the ARS,

$$Q_{GE} = m_{a1}(h_5 - h_{11}) \tag{11}$$

Considering the air-cooled condenser, ACC,

$$m_s(h_7 - h_8) = m_{a2}(h_9 - h_1) \tag{12}$$

Considering the pre-cooler, PC,

$$m_w C_{p,w}\Delta T = m_{a1}C_{p,a}\left(T^*_{a1} - T_{10}\right) + m_{a2}C_{p,a}\left(T^*_{a2} - T_{10}\right) \tag{13}$$

where m_w is the mass of water recirculating through the PC and ΔT is its rise in temperature; T* values are the design inlet temperatures to the AC (= 27ºC) and the ACC (= 20ºC) respectively, and $T_{10} = T_{amb}$.

Since the temperature of the TES system is dependent on the ambient temperature, which, in turn, is an arbitrary function of time, an unsteady state heat balance equation has to be written for the TES system:

$$V_{TES}\rho_w C_{p,w}\frac{dT_{TES}}{dt} = -Q_{EV} + m_w C_{p,w}\Delta T + K\left(T_{amb} - T_{TES}\right) \tag{14}$$

where V is the volume of TES, ρ is the density of water, C_p its specific heat, ΔT is the rise in temperature of the recycling water; K is an overall heat loss coefficient, and T is the temperature.

The above model is applied in this paper to evaluate the feasibility of the ARS-driven thermal energy storage system in pre-cooling the ambient air for a 500-MW CCPP in the arid Southwest region of the U.S. The specifications of this plant are listed in Appendix I(a). The following three cases are compared using typical annual ambient temperature data:

- Case I: pre-cooling the air supply to ACC of steam cycle
- Case II: pre-cooling the air supply to AC of gas cycle
- Case III: pre-cooling the air supply to both the ACC of steam cycle and AC of gas cycle

The equations are solved using Engineering Equation Solver (F-Chart Software) and Extend (Imagine That Inc.).

RESULTS

The performance parameters used to compare the three systems include the TES tank volume, temperature profiles of the air at the AC and ACC inlet, and the power loss averted. The temperatures and mass flow rates at the various state points are listed in Appendix I (b). The COP of the ARS is 0.67.

Power loss in gas turbine plat

The decline of gas turbine plant net output with ambient temperature for this site is shown in Figure 3. As can be seen from this plot, gas turbine output decreases with increasing ambient temperature at a rate of about 0.6 MW/ºK. The proposed pre-cooling system is able to avert this power loss even up to ambient temperatures of 32ºC.

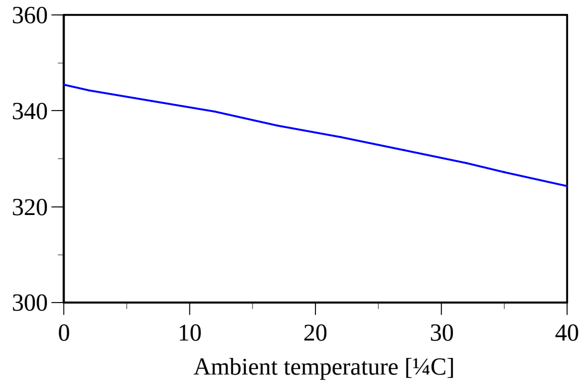

Figure 3. Gas turbine net output vs. ambient temperature.

Details of the analysis of Case I have been presented previously (Gadhamshetty et al., 2006). This analysis showed that a tank volume of 4,500 m^3 will be required to maintain the air temperature at the inlet to the ACC at the design value of 20ºC throughout the year. Simulations under ambient air temperatures up to 40ºC indicated that the proposed system is capable of maintaining the rated net power output of the plant with minimal fluctuations.

A major concern with the proposed TES system is the large volume of the storage tank, which is a function of plant capacity as well as the design inlet air temperature to ACC, T*. To evaluate the relationship between the three, simulations were done for power plant capacities ranging from 100 to 1,000 MW and T* ranging from 20 to 24ºC. Figure 4 shows the variation of the TES volume as a function of net plant output of CCPPs for selected values of T*. As expected, the volume of the TES increases with an increase in net plant output for any value of T* and the volume of the TES decreases with an increase in T* at a given CCPP output. For example, the volume requirement for the 500-MW CCPP can be reduced

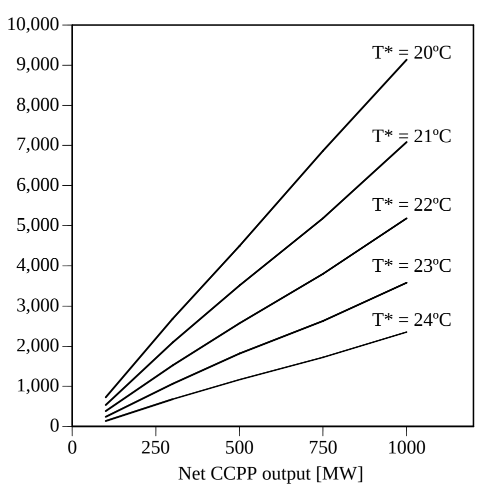

Figure 4. Sensitivity of TES volume to design temperature

22% from 4,500 m^3 to 3,500 m^3 if an increase of just 1ºC in the air inlet temperature to the ACC can be tolerated.

In Case II, the air supply to the air compressor is pre-cooled to maintain its temperature below 27ºC. This is achieved by turning on the pump to circulate the chilled water from the TES to the pre-cooler whenever the ambient temperature exceeded 27ºC. The simulated temperature profiles at the AC inlet and in the TES are compared against the ambient temperature over a year in Figure 5.

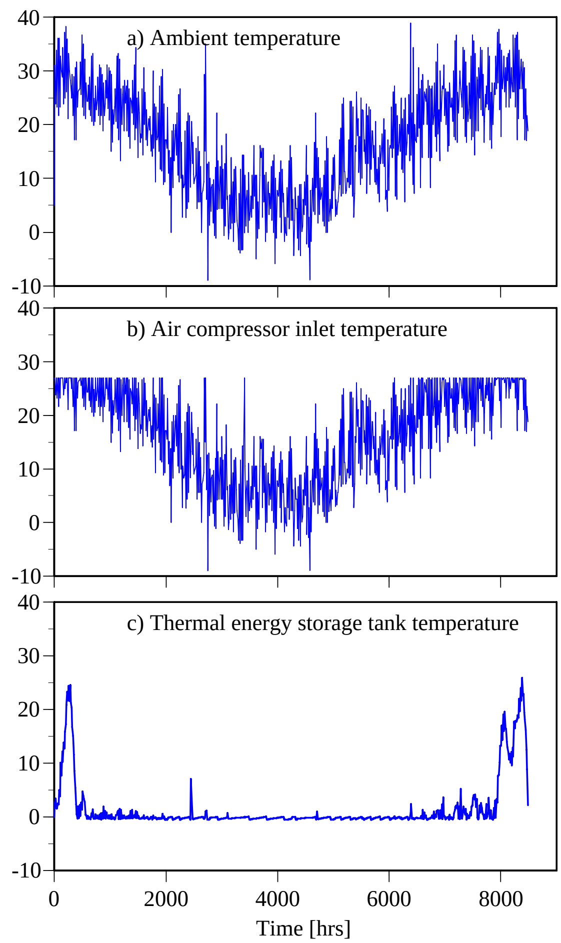

Figure 5. Temperature profiles over 1 year for System II

In this case, the net power output of the gas turbine plant was maintained at the design value of 330 MW over the year; the evaporator duty of the ARS was 4 MW and the volume of the TES tank was 3.4 m^3. The air flow rate handled by the pre-cooler was 1,252 kg/s. At an ambient temperature of 32ºC, for example, the power loss averted was 2.6 MW.

In Case III, the total of the air supply to the air compressor and that to the ACC is pre-cooled to maintain its temperature below 27ºC. This is achieved by turning on the pump to circulate the chilled water whenever the ambient temperature exceeded 27ºC. The simulated temperature profiles at the AC inlet and in the TES are compared against the ambient temperature over a year in Figure 6.

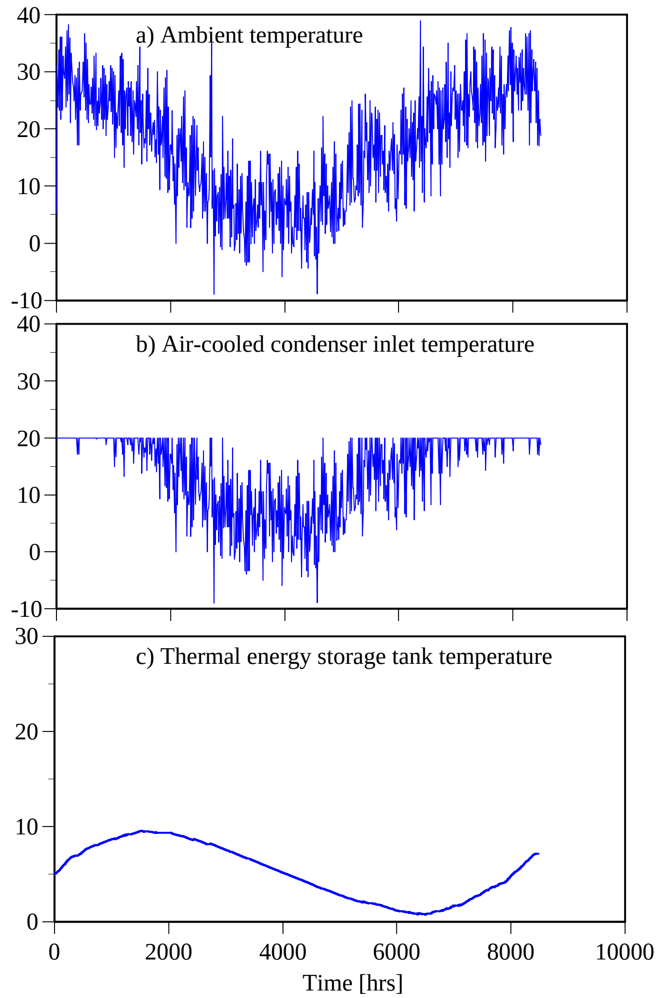

Figure 6. Temperature profiles over 1 year for System III

In this case also, the net power output of the gas turbine plant was maintained at the design value of 330 MW over the year; the evaporator duty of the ARS was 46.2 MW and the volume of the TES tank was 4,500 m^3. The air flow rate handled by the pre-cooler was 16,023 kg/s. At an ambient temperature of 32ºC, for example, the power loss averted was 7.2 MW.

Findings from the analysis of the three cases are summarized in Table I.

Table I. Summary of findings for the three cases

	Case I	Case II	Case III
Air flow rate [kg/s]	14,771	1,252	16,023
Evaporator duty [MW]	46.2	4.0	46.2
Volume of TES tank [m^3]	4,500	3.4	4,500
Power loss eliminated [MW]	4.2	2.6	7.2

CONCLUSIONS

The proposed pre-cooling at the inlets of the compressor and the air-cooled condenser may be a feasible approach for easing the power loss associated with the operation of the air compressor and ACC at high ambient temperatures. The proposed system(s) can yield power savings ranging from 4 to 8% of the rated power in combined cycle power plants. Gain in net power production and conservation of water resources may offset the capital costs associated with the TES tank and the absorption refrigeration system. Replacement of the sensible heat TES with a latent heat TES may further reduce the TES volume requirements.

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APPENDIX I:

a) Plant specifications

Combined Cycle Power Plant:

Net power output	MW	500
Air compressor inlet temperature	K	300
Pressure ratio	-	14
Compressor isentropic efficiency	%	82
Gas turbine isentropic efficiency	%	86
Gas turbine inlet temperature	K	1,400
Pump isentropic efficiency	%	95
Steam turbine inlet temperature	K	823
Steam turbine inlet pressure	kPa	8,000
Steam turbine isentropic efficiency	%	86
Condenser pressure	kPa	20
Stack temperature	K	460

Absorption Refrigeration System:

Evaporator temperature	K	278
Absorber temperature	K	311
Condenser temperature	K	313
Generator temperature	K	363
Economizer efficiency	%	70
Generator pressure	kPa	
Evaporator pressure	kPa	

b) Temperature and mass flow rates

State Point	Temp [K]	Mass rate [kg/s]
1	300	1252
2	699	1252
3	140	1252
4	821	1252
5	460	1252
6	803	175
7	333	145
8	433	175
10	Tamb	3070
11	395	1252
12	363	1252
13	460	1252
14	N.A	24
15	318	24
16	318	24
17	Ttes	14371
18	278	na
19	278	24
20	318	453
21	318	453
22	353	453
23	373	429