Critical relative humidity of evaporative cooling in CCPP based on thermodynamic and economic analyses

Hao Lyu¹, Jianhong Chen¹, * and Huanyan Huang²

¹College of Energy Engineering, Zhejiang University, Hangzhou, China.
²China Energy Engineering Group Zhejiang Electric Power Design Institute CO., LTD. Hangzhou, China.

*Corresponding author e-mail: 21627035@zju.edu.cn

Abstract. A calculation and analysis to determine the feasibility of inlet air evaporative cooling systems for combined cycle power plant (CCPP) is carried out. According to the psychrometric chart, the effects of ambient temperature and relative humidity (RH) on the temperature drop of the inlet air across an evaporative cooling system are analyzed. The power differences between CCPP with and without an inlet air evaporative cooling system in different environmental conditions are investigated. Based on the thermodynamic and economic analyses, a new concept, the critical RH, is proposed to evaluate the feasibility and economics of CCPP with inlet air evaporative cooling system considering different ambient conditions. If ambient RH is lower than the critical value, an inlet air evaporative cooling system is suitable and economical for regions considered. Sensitivity analysis of the factors affecting the critical RH is also carried out. The critical RH varies with climatic condition, electricity price, fuel price and expected payback period (PBT).

1. Introduction
Gas turbine and CCPP are widely applied in power generation with the advantages of high power-to-weight ratio, short duration of construction, quick-start, strong peak load regulation performance, low greenhouse gas emissions and so on. Despite these advantages, a certain disadvantage of gas turbine is that the power output and efficiency both decrease as ambient temperature increases [1-4]. According to statistics, as the compressor inlet air temperature rises 1 °C, the power output of gas turbine declines about 0.5-0.9% [5, 6]. So the power output of gas turbine and CCPP falls considerably in summer. However, the electricity power demand on these days rises dramatically because of the operation of air conditioners and cooling systems. Strictly speaking, the performance profile of gas turbine and the electric power demand profile go in opposite directions.

Performance losses of gas turbine and CCPP because of atmospheric conditions can be prevented by cooling inlet air before entering the compressor. Since this reduces the power consumption of compressor and increases the air density. Various inlet air cooling technologies are used to enhance the peaking capacity of gas turbines and CCPP, such as evaporative cooling, compression chiller, absorption chiller and thermal energy storage. The power output and efficiency of gas turbine and CCPP with and without an inlet air cooling system are analyzed [7-9]. Mohapatra [10] compared compression cooling with evaporative cooling integrated to a gas turbine power plant. Khaliq and
Choudhary [11] provided an exergy analysis of gas turbine with absorption chiller. The analysis indicated that the exergy losses in different components of gas turbine varied with the compressor inlet air temperature and ambient RH.

Compared with other inlet air cooling technology, evaporative cooling is the most economical system in hot and dry areas because of the ease of installation and its low initial investment cost [12,13]. De Lucia et al. [14] proposed that evaporative cooling could improve the power output by 2-4% according to atmospheric condition. Chaker et al. [15] concluded the potential cooling hours of direct evaporative cooling in 122 different locations.

Most of the previous studies concerned about the effects of evaporative cooling system on performance of gas turbine and CCPP. However, in which environmental conditions the evaporative cooling system are suitable and economical haven’t been carried out. Therefore, a new concept, critical RH, is proposed to evaluate the feasibility and economics of evaporative cooling system for CCPP according to real atmospheric conditions. For a given period, the economic benefit of the evaporative cooling system may be exactly equal to total investment cost in some condition. The RH of this condition is the critical value. If ambient RH is lower than critical value, the evaporative cooling system for CCPP is suitable and economical for regions considered. Otherwise, the evaporative cooling system is not economical or inlet air needs to be dehumidified before entering the evaporative cooling system. In addition, the sensitivity of critical RH for ambient temperature, electricity price and fuel price is carried out. With the concept and criteria proposed in our study, gas turbine operators can easily utilize it to determine the feasibility of the evaporative cooling system for their own gas turbine or CCPP.

2. Modelling and analysis

2.1. Description of the system investigated

Figure 1 shows the schematic diagram of CCPP with evaporative cooling system. Ambient air is cooled in the evaporative cooling system because the heat is absorbed by the latent heat of vaporization involved in phase changes. Then the cooled air is sent to compressor for compression. The CCPP is composed of a GE PG9351 (FA) gas turbine, a triple pressure reheat heat recover steam generator (HRSG) and HP, IP and LP steam turbine. The main parameters and performance indexes of this CCPP without inlet air cooling at ISO ambient base load condition are shown in Table 1.

![Figure 1. Schematic diagram of CCPP with evaporative cooling system](image-url)
Table 1. Parameters of CCPP (ISO ambient base load)

| Parameter (unit)            | Value       | Parameter (unit)            | Value       |
|----------------------------|-------------|----------------------------|-------------|
| Ambient air temperature(°C) | 15          | Natural gas input (MW)      | 686.53      |
| Ambient RH (%)              | 60          | GT power output (MW)        | 251.43      |
| Ambient air pressure (kPa,a)| 101.35      | ST power output (MW)        | 143.12      |
| Natural gas flow (kg/s)     | 14.10       | Gross power output (MW)     | 394.55      |
| Air flow (kg/s)             | 639.81      | Gross thermal efficiency (%)| 57.48       |
| Air/fuel mass ratio         | 45.38       | Gross Heat rate (kJ/kWh)    | 6262.80     |
| Natural gas-LHV (MJ/kg)     | 48.69       |                             |             |

2.2. Thermodynamic analysis

2.2.1. The evaporative cooling system. Figure 2 illustrates the inlet air cooling process on the psychrometric chart. The cooling process follows a constant enthalpy line. In the cooling process, demineralized water is injected into the evaporative cooling system to cool inlet air from ambient condition \((T_a, \varphi_a)\) to a lower temperature. Part of the injected water evaporates, and then it cools the inlet air due to the absorption of latent heat. In ideal condition, the compressor inlet air temperature \((T_{i,c})\), can reach wet bulb temperature as RH rises up to 100% (point ‘s’). However, the outlet temperature is higher than wet bulb temperature in reality. The bold line \((a-i)\) and \((a-s)\) represent real cooling process and ideal theoretical cooling process, respectively.

![Figure 2. The inlet air cooling process on the psychrometric diagram](image)

The temperature drop \((\Delta T)\), which is the temperature difference between the inlet and outlet of the evaporative cooling system and is given by

\[
\Delta T = T_{a,dry} - T_{i,dry}
\]  

(1)

The ambient air consists of dry air and water vapor. The humidity ratio \((d)\) is identified as the mass of water vapor contained in per unit dry air and can be obtained from

\[
d = 0.622 \frac{p_v}{p_a - p_v}
\]  

(2)

\[
p_v = \varphi p_a
\]  

(3)
Where \( p_a \) is the atmospheric pressure, \( p_v \) is the partial pressure of water vapor, \( p_s \) is the saturation pressure of water vapor, \( \varphi \) is the RH of air.

2.2.2. CCPP performance. The performance parameters, including power output and efficiency of gas turbine, steam turbine and CCPP are calculated with some governing equations given as follows:

The compressor work is given by

\[
W_c = m_{a,i} \cdot c_{p,c,i} \cdot \left( T_{e,c} - T_{i,c} \right)
\]

(4)

Where \( m_{a,i} \) is the mass flow rate of inlet air, \( c_{p,c,i} \) is the specific heat at constant pressure of air across compressor, \( T_{e,c} \) and \( T_{i,c} \) are the air temperatures of the exit and inlet of compressor, respectively.

The power output and efficiency of gas turbine are given by

\[
W_g = m_{g,i} \cdot \left( h_{g,i} - h_{g,e} \right) - W_c
\]

\[
\eta_g = \frac{W_g}{m_f \cdot \text{LHV}}
\]

(5)

(6)

Where \( m_{g,i} \) is the amount of burnt gas entering to the turbine, \( m_f \) is the amount of natural gas, LHV is the lower heating value of fuel, \( h_{g,i} \) and \( h_{g,e} \) are the enthalpies of burnt gas entering and exiting to the turbine, respectively.

As shown in Figure 1, steam turbine is composed of HP, IP and LP stages. The steam from HRSG is firstly brought to HP steam turbine for expansion to do work and generate electricity. Then the steam returns back to the HRSG for reheating, and goes to IP steam turbine. Finally, the steam from IP and IP steam turbine is brought to LP steam turbine for further expansion. The power output and efficiency of steam turbine are expressed as

\[
W_s = \sum_{s} m_{s,i} \cdot \left( h_{s,i} - h_{s,e} \right)
\]

\[
\eta_s = \frac{W_s}{Q_0}
\]

(7)

(8)

Where \( Q_0 \) is the heat energy entering into HRSG, \( m_{s,i} \) is the amount of steam entering to each stage of the steam turbine, \( h_{s,i} \) and \( h_{s,e} \) are the enthalpies of steam entering and exiting to each stage of the steam turbine. Therefore, the power output and efficiency of CCPP are given by

\[
W_{\text{plant}} = W_g + W_s
\]

\[
\eta_{\text{plant}} = \frac{W_{\text{plant}} \cdot \eta_{\text{alt}}}{m_f \cdot \text{LHV}}
\]

(9)

(10)

Where \( \eta_{\text{alt}} \) is the efficiency of alternator.

2.3. Economic analysis

2.3.1. Electricity production enhancement. The evaporative cooling system causes an increase of the power output of CCPP and, consequently, electricity generation is improved. With the installation of
the evaporative cooling system, the power output of CCPP will change, including: 1) the power output of gas turbine increase since compressor inlet air temperature decrease ($\Delta W_{gt}$); 2) the power output of gas turbine and steam turbine increase since air density rises as $T_i,c$ declines ($\Delta W_d$); 3) the inlet air pressure drop rises as a result of the installation of the inlet air evaporative cooling system, which leads to a reduction of the power output of CCPP ($\Delta W_p$); 4) the energy consumption of auxiliary equipment of the evaporative cooling system increase ($W_a$). Therefore, the increase of the net power output of CCPP is:

$$\Delta W = \Delta W_{gt} + \Delta W_d - \Delta W_p - W_a$$  \hspace{1cm} (11)

The additional income of CCPP with the evaporative cooling system is

$$I_{add} = \Delta W \cdot \eta_{alt} \cdot p_e$$  \hspace{1cm} (12)

Where $p_e$ is the price of electricity.

2.3.2. Extra fuel consumption. With the application of the evaporative cooling system, the gross power output increases and, as a result, the fuel consumption increases, so fuel consumption increases. The extra fuel consumptions of CCPP with the inlet air evaporative cooling system ($\Delta F$, kg/s) are given by

$$\Delta F = (W_e \cdot \beta_e - W_0 \cdot \beta_0) / (3.6 \cdot LHV)$$  \hspace{1cm} (13)

Where $\beta$ is gross heat rate (kJ/kWh); the subscript e and 0 are the values of CCPP with and without the inlet air evaporative cooling system, respectively.

2.3.3. Initial investment cost. The initial investment cost of evaporative cooling system mainly depends on the maximum capacity of cooling load, and is given by

$$I = k_i \cdot Q_{CL,\text{max}}$$  \hspace{1cm} (14)

Where $k_i$ is the initial investment cost for per unit cooling load, $Q_{CL,\text{max}}$ is the maximum capacity of cooling load.

2.3.4. Critical RH. The annual net profit by installing the evaporative cooling system to CCPP consists of the following three parts: the additional income from electricity production enhancement, the increase of fuel consumption cost and annual operation and maintenance fees of evaporative cooling system.

$$S = (\Delta W \cdot \eta_{alt} \cdot p_e - \Delta F \cdot p_f) \cdot H - M$$  \hspace{1cm} (15)

Where $p_e$ is the price of electricity, $p_f$ is the price of natural gas, $H$ is operating hours, $M$ is annual maintenance and operation fees of the evaporative cooling system. So the payback period (PBT) is

$$PBT = \frac{I}{S}$$  \hspace{1cm} (16)

To gain a profit ($S>0$), the additional income should be greater than the sum of fuel consumption cost and operation and maintenance cost.

The critical RH is relevant to local environmental conditions, design parameters of CCPP, fuel price, electricity price and expected PBT. The calculation flow of the critical RH is shown in figure 3.
3. Results and discussion

In this study, the temperature drop of inlet air in different environmental conditions is obtained. The power outputs of CCPP with and without the evaporative cooling system are also compared. The critical RH of the evaporative cooling system for CCPP is investigated as well.

3.1. The temperature drop with the evaporative cooling system

Figure 4 illustrates the effect of environmental conditions on temperature drop \( \Delta T \) of air at the inlet and outlet of the evaporative cooling system. It can be observed that \( \Delta T \) rises continuously as ambient temperature rises. The higher of ambient temperature and the lower of RH, the difference between dry bulb temperature and wet bulb temperature is larger, which means the greater \( \Delta T \) can be achieved by evaporation cooling.

For a given ambient RH, \( \Delta T \) increases as ambient temperature rises. However, the increasing rate of \( \Delta T \) goes down as ambient RH rises. \( \Delta T \) even changes slightly as ambient RH reaches a turning point. For example, when RH is 60%, \( \Delta T \) increases slowly as ambient temperature rises. It is obviously that at high RH situation, the effectiveness of the evaporative cooling system is weak and \( \Delta T \) is not as large as that in arid conditions. This phenomenon indicates that the evaporative cooling system is not suitable in humid regions. For example, when ambient temperature is 35 °C and RH is 60%, \( \Delta T \) is merely 5.95 °C, however, as the RH decreases to 10%, \( \Delta T \) dramatically goes up to 18.58 °C, which is double larger than the former.

![Figure 4. Temperature drop (ΔT) at various ambient conditions](image-url)
3.2. **Comparison of power outputs of CCPP with and without the evaporative cooling system**

The power outputs of CCPP with and without the evaporative cooling system are compared as shown in figure 5. It is observed that power outputs of CCPP with and without the evaporative cooling system both decrease almost linearly as ambient temperature increases. However, the power output of CCPP without the evaporative cooling system decreases more quickly than that of CCPP with the evaporative cooling system. In the state of no evaporative cooling, when ambient RH is 60%, as ambient temperature increases from 25 °C to 30 °C, the power output decreases by 4.89%. For CCPP with the evaporative cooling system, the decrease is only 3.16%.

The results also show that the effectiveness of the evaporative cooling system for CCPP significantly depends on RH. For CCPP with the evaporative cooling system, the power output decreases more slowly in lower ambient RH. For example, as ambient temperature increases from 25 °C to 30 °C, the power output decreases 3.16% at 60% RH and 1.42% at 10% RH respectively.

3.3. **Effect of ambient conditions on power output of CCPP with the evaporative cooling system**

Figure 6 shows the variation of power output of CCPP with the evaporative cooling system in different ambient conditions. Although the temperature drop increases as ambient temperature rises, the power output of CCPP with the evaporative cooling system reduces. This is because for a higher ambient temperature, Ti,c is still high, though ambient air is cooled by the evaporative cooling system. As Ti,c rises, the mass flow of inlet air reduces and power consumption of compressor increases correspondingly. The increase in power consumption of compressor because of higher Ti,c leads to a reduction in CCPP power output. For a given ambient RH of 20%, as ambient temperature increases from 25 °C to 35 °C, the power output of CCPP with the evaporative cooling system reduces 15.61MW.

It is also observed that as ambient RH is at a low level, the power output of CCPP with the evaporative cooling system decreases slowly, however, as the ambient RH rises, the CCPP power output decreases more quickly. The outlet RH of the evaporative cooling system is assumed to be 95%, a lower ambient RH means more water evaporates and absorbs more latent heat. So the effect of the evaporative cooling system is improved and, consequently, temperature drop rises and Ti,c falls. At ambient RH of 80% and ambient temperature of 35 °C, the temperature drop is 2.36°C, while it goes up to 15.47°C at an ambient RH of 20%. This indicates less power is consumed by compressor, which results in the enhancement of CCPP power output.
Figure 6. Effect of ambient conditions on CCPP power output

Figure 7. Effect of ambient temperature and 3.4. Effect of ambient conditions on CCPP power increase with the evaporative cooling system

Figure 7 shows the power increase (ΔW) of CCPP after the application of the evaporative cooling system in different ambient conditions. The result illustrates the advantage of CCPP with the evaporative cooling system over that without inlet air cooling. For a given RH, as ambient temperature rises, the additional power generation due to the installation of the evaporative cooling system increases. At a higher ambient temperature and lower RH, the difference between wet bulb temperature and dry bulb temperature is higher, so a greater temperature drop can be achieved and as a result, less power will be consumed by compressor. Therefore, the difference between power output of CCPP with and without evaporative cooling system is greater, namely power output increase is greater. When RH is very low, ΔW increase sharply as ambient temperature rises. While RH rises near saturation, the increase tendency becomes slow. When ambient temperature is 35 °C, the power increase is 9.45MW at ambient RH of 80%, while it rises dramatically to 52.56MW at ambient RH of 20%.

3.4. The critical RH varies with ambient temperature, electric price and PBT
The critical RH is a value depending on economic analysis of PBT to determine the feasibility and economics of the evaporative cooling system for CCPP according to different ambient conditions. When the ambient RH is lower than the critical value, the actual PBT of CCPP unit with inlet air evaporative cooling system installation is shorter than the expected PBT. On the contrary, when the ambient RH is higher than the critical value, the actual PBT is longer than expected PBT. The critical ambient RH can be used to evaluate the feasibility of implementation of the evaporative cooling system for CCPP.
The variation of critical RH with different annual average temperature and PBT is shown in figure 8. The annual average temperature is the average temperature of 5 months (May-September) in a year. It is obviously that the shorter of PBT, the critical RH is lower for a given annual average temperature. This is because in a lower RH, the difference between the wet bulb and dry bulb temperatures is greater, which means a larger temperature drop is possible. The larger temperature drop leads to more power increase and electricity generated. As the additional generated electricity increase, the net income for the year rises and the PBT becomes shorter. It is also observed that as the payback become longer, the critical RH for different annual temperature gradually tends to be close. This is because as the RH increases to a high level, the temperature drop decreases and tends to the same.

Figure 8. The critical values of RH vary with annual average temperature and PBT

The sensitivity of critical RH varies with electricity price and PBT is shown in figure 9. It illustrates that the shorter of PBT, the critical RH is lower for a given electricity price. The lower of RH, the temperature drop and power output increase is greater as explained earlier. For the same temperature drop and additional generated electricity, the higher of electricity price, the more of net income is gained. Therefore, as the electricity price increases, the critical RH for the same PBT is higher. For example, for a given PBT of 2.5 years, the critical RH is 42% for electricity price of 0.08 $/kWh and 61% for electricity price of 0.09 $/kWh, respectively.

Figure 9. The critical values of RH vary with electricity price and PBT

4. Conclusions
Based on the thermodynamic and economic analyses and calculation of critical RH, the following conclusions have been drawn:

(1) The temperature drop (ΔT) increases as ambient RH declines. At ambient temperature of 35 °C and RH of 60%, ΔT is merely 5.95 °C, while ΔT is 18.58 °C at 10% RH.
(2) As ambient temperature rises, the power outputs of CCPP with and without evaporative cooling system both decrease almost linearly. In the state of no cooling, when ambient RH is 60%, as ambient temperature increases from 25 °C to 30 °C, the power output decreases by 4.89%. For CCPP with the evaporative cooling system, the decrease is only 3.16%.

(3) The application of the evaporative cooling system improves the CCPP power output. The power increase is greater at higher ambient temperature and lower ambient RH.

(4) The calculation and analysis of critical RH in different ambient conditions are carried out. It is observed that the shorter of expected PBT, the critical RH is lower for a given annual average temperature.

(5) The sensitivity of critical RH varies with electricity price and expected PBT is analyzed. For a given expected PBT of 2.5 years, the critical RH is 42% for electricity price of 0.08 $/kWh and 61% for electricity price of 0.09 $/kWh, respectively.

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