# INVESTIGATION ON THE OPTIMUM DESIGN OF HEAT EXCHANGERS IN A HYBRID CLOSED CIRCUIT COOLING TOWER

**M. M. A. Sarker**
Department of Mathematics, BUET, Dhaka-1000, Bangladesh.
Corresponding email: masarker@math.buet.ac.bd

***Abstract:*** *Investigation on the optimum design of a heat exchanger in a Hybrid Closed Circuit Cooling Tower having a rated capacity of 1RT is performed experimentally. The heat exchanger of dimension 0.4m×0.33m×0.572m has 15×7 bare type 15.88mm OD copper coils in staggered arrangement. The relevant design parameters were selected based on the typical East Asian meteorological constrains for the year-round smooth operation of the cooling tower. This study presents results related to the cooling capacity and the cooling efficiency with respect to wet bulb temperature and pressure drop with respect to air inlet velocity. Results are also presented in terms of number of transfer units (NTU). Cooling capacity was found to be close to the rated one for the wet mode but low in dry mode operation.*

***Keywords:*** *Hybrid closed circuit cooling tower, Cooling capacity, Wet mode, Dry mode*

## INTRODUCTION

Now-a-days, cooling towers are increasingly in use to remove heat from industrial processes, refrigeration and air-conditioning systems mainly because it is a relatively inexpensive and dependable heat rejection device. Cooling towers are categorized as crossflow when air flow is directed perpendicular to the water flow and termed as counterflow when the air flow is directed opposite to the water flow. Air flow first enters into an open area beneath the heat exchanger and is then drawn up vertically. In counter-flow cooling towers, water is sprayed into an air stream.

Simultaneous heat and mass-transport processes in a cooling tower lead to complicacy in design. Heat and mass are transferred and the water enthalpy decreases while that of air increases[1-3]. Depending on the heat transferring mode employed, cooling towers are called wet cooling towers when it operates on the principle of evaporation and known as dry cooling tower when operates by heat transmission through a surface that divides the cooling water from ambient air and rely mainly on convection heat transfer to reject heat. Khan et al.[4] claimed that evaporation is the predominant mode of heat transfer and they demonstrated that evaporation contributes about 62.5% of the total rate of heat transfer at the bottom of the tower and almost 90% at the top of the tower.

Numerical simulation results on counterflow wet cooling towers can be found from Gan et al.[5-7] and a number of mathematical models have been reported by Khan et al.[8], Soylemez[9] and Korenic[10]. Experimental studies[11-12] are also available on the closed wet cooling tower including a number of recent publications [2-3, 13]. Bentley et al.[14] developed an advanced wet/dry heat transfer surface for power plant cooling towers. In their experimental apparatus the wet surface area was five percent of the total heat transfer surface area. Their analytical model indicated that the yearly water consumption of a cooling tower with the new wet/dry surface would be less than half of that of a conventional wet cooling tower and fog plumes would be eliminated. Sarker et al.[15] investigated the thermal performance of a cross flow cooling tower experimentally and reported the effect of wet-bulb temperature for controlling the cooling water temperature at a desired level and an appropriate air flow rate to maintain the cooling performance. The available literature shows a lack of experimental data on the heat exchanger of the Hybrid Closed Circuit Cooling Tower (HCCCT).

HCCCT is a closed wet cooling tower which is capable of working both in wet and in dry mode. It works well in a plume and ice-free state maintaining lower noise level in dry mode during the mid-season and winter as long as the ambient temperature remains below 12-14°C. It operates smoothly in wet mode when the ambient temperature is high, water consumption is low and the process water can be cooled down to 4°C above the wet bulb temperature and can be packed in light and compact bundle with optimized circuitry. In order to increase the cooling

**Nomenclature**

| | |
|---|---|
| $A$ | Effective cross section area of cooling tower [m$^2$] |
| $a$ | Effective interfacial area per unit volume [m$^2$/m$^3$] |
| $a'$ | Heat transfer area per unit volume [m$^2$/m$^3$] |
| $C$ | Specific heat of the cooling water [kcal/kg.$^O$C] |
| $c$ | Specific heat of the spray water [kcal/kg.$^O$C] |
| $G$ | Flow rate of air [kg/h] |
| $h$ | Enthalpy [kJ/kg] |
| $i$ | Air enthalpy [kcal/kg.dry air] |
| $i^*$ | Saturated air enthalpy [kcal/kg.dry air] |
| $Ka$ | Volumetric overall heat transfer coefficient [kW/m$^3\Delta$i] |
| $k_{og}a$ | Overall mass transfer coefficients [kg/ m$^3$.hr.$\Delta$i] |
| $L$ | Flow rate of spray water [kg/h] |
| $Q$ | Cooling capacity [kcal/h] |
| $Re$ | Reynolds number [-] |
| $S$ | Surface area of tube [m$^2$] |
| $T$ | Temperature of cooling water [$^O$C] |
| $t$ | Temperature of the spray water [$^O$C] |
| $U$ | Overall heat transfer coefficient [kW/m$^2$K] |
| $W$ | Flow rate of cooling water [kg/h] |
| $Z$ | Effective height of cooling tower [m] |

***Subscripts***

| | |
|---|---|
| $i$ | Inlet |
| $o$ | Outlet |
| $g$ | Air |
| $I$ | Inner |
| $O$ | Outer |
| $L$ | Spray water |
| $W$ | Cooling water |

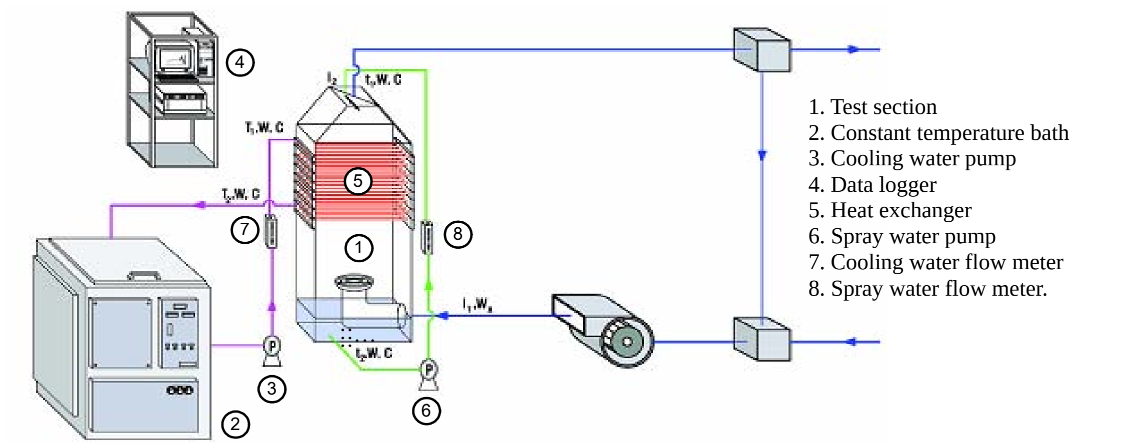


Figure 1: Schematic of the experimental apparatus.

Figure 2: Internal view of the heat exchanger.

**Table 1: Experimental conditions**

| | | | |
|---|---|---|---|
| Cooling water | Mass flowrate | [kg/h] | 420 - 1150 |
| | Inlet temperature | [$^{O}$C] | 37 |
| Spray water | Mass flowrate | [kg/h] | 1000 - 2000 |
| Air | Velocity | [m/s] | 2.0 - 4.0 |
| | Wet-Bulb Temp. | [$^{O}$C] | 16 - 32 |

capacity and to decrease the pressure drop, the heat exchanger plays a very important role. The objective of this study is to provide experimental results regarding the design parameters of the heat exchanger based on the optimum performance of a HCCCT.

**EXPERIMENTAL APPARATUS**

A schematic diagram of the experimental apparatus is shown in Fig. 1. Figure 2 gives a photograph of the heat exchanger. The heat exchanger has 15 rows and 7 columns of copper made bare coils set in staggered arrangement inside a casing of dimension 0.4m × 0.33m × 0.572m. The casing is made of acrylic sheet of 10 mm thickness so that the flow of water inside the tower can be clearly viewed from outside. The apparatus also includes constant temperature bath used for circulating cooling water at a desired temperature. A damper and a heater (not shown in the schematic) are used to control the temperature and the humidity of air. H300M type humidity and temperature sensor capable of measurement with an error of less than ± 0.5% is used for recording the temperature and the humidity of the air.

**EXPERIMENTAL PROCEDURE**

The cooling water is maintained at the desired level of temperature using the constant temperature bath and is recirculated by the pump through the coils of the heat exchanger. It exchanges heat with the downward falling spray water and upward flowing air. The temperature of the cooling water is recorded in every 2 seconds by a data logger and saved in a computer.

In the wet-closed mode operation, the spray water passes around the coil bank and gains heat from the cooling water circulating inside. Finally the water is collected and returned to the water tank at the lower part. Mass transfer occurs when the spray water flows downward due to gravity. Make-up water is added to the recirculating spray water and the addition of water is controlled utilizing a ball tab. The spray water flow rate is controlled using the control valve and the bypass valve. After conducting the experiment for a while, the temperature of the spray water and the cooling water become stable and ready to be recorded.

Air is forced by the fan to flow in the upward direction countering the flow of the spray water. Air exchanges heat with cooling water as well as with the spray water in wet mode operation and finally comes out of the tower and is released to the environment through the duct in dry mode. In wet mode, air is recirculated after controlling the temperature and the humidity at the inlet of the heat exchanger with the help of the heating coil and the damper. Air velocity is maintained at a desired level by controlling the fan.

The humidity, the temperature and the volumetric flow rate of air and cooling water are maintained at a desired level. When steady state is attained, temperature of the cooling water at the inlet and the outlet, air velocity at the inlet and the pressure at the top and the bottom of the heat exchanger are collected at every sensor position in the various part of the tower and saved in a computer. The experimental conditions are shown in Table 1.

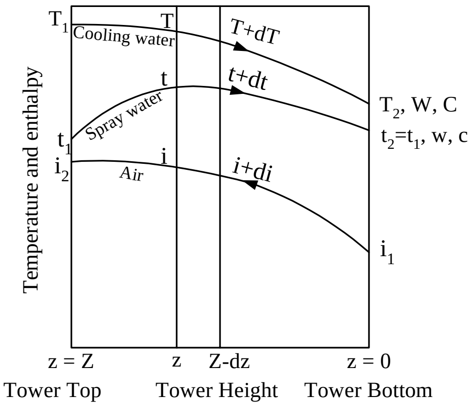


Figure 3: Temperature and enthalpy distribution.

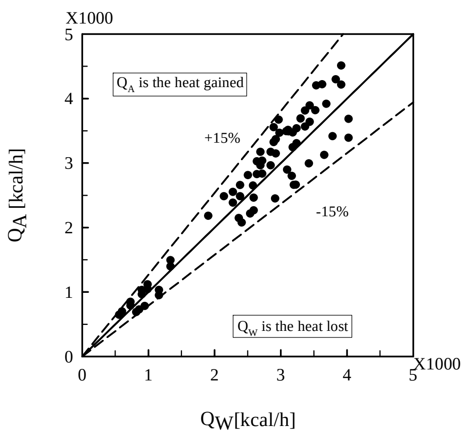


Figure 4: Heat balance of the experimental apparatus.

**THEORETICAL BACKGROUND**

The transfer of heat from water to air happens in two ways, namely by the evaporative heat transfer which depends on the latent heat of vaporization and also by the sensible heat transfer. Therefore, in calculating the heat transfer from water to air flow inside and outside the coils in a tower, the enthalpy relations are widely used. Figure 3 shows the temperature and the enthalpy distributions as a function of the distance of the tower from the top for every fluid inside the HCCCT. Tezuka et al.[17] has shown that the rate of heat loss by the cooling water with respect to an elementary area *dz* inside the cooler is given by

$$WC.dT = -U.a'.\left(T-t\right).S.dz \tag{1}$$

Again the rate of heat loss by the spray water with respect to the elementary area *dz* can be given by

$$wc.dt = -K_{og}a.\left(i^{*}-i\right).S.dz + U.a'.\left(T-t\right).S.dz \tag{2}$$

The heat gained by the air can be given by

$$W_A.di = -K_{og}a.\left(i^{*}-i\right).S.dz \tag{3}$$

**Number of Transfer Units**: Number of Transfer Unit (NTU) method is based on the principle that the actual heat transfer occurs between the fluids is a fraction of the maximum heat transfer possible. The maximum heat transfer possible can be theoretically achieved with a counterflow heat exchanger of infinite surface area[16-18]. NTU gives a measure of the exchangers' ability to change the temperature of the fluid that changes temperature most easily and is given by

$$NTU = \frac{\left(U.a'.S.Z\right)}{WC} \tag{4}$$

Three new variables are defined as

$$R_1 = \frac{wc}{WC} \tag{5}$$

$$R_2 = \frac{K_{og}a}{U.a'} \tag{6}$$

$$R_3 = \frac{WC}{W_A} \tag{7}$$

Then Eqs (1-3) can be rewritten with the help of Eqs (5-7) in the following form

$$dT = -\left(T-t\right).dNTU \tag{8}$$

$$dt = \begin{bmatrix} -\left(\dfrac{R_2}{R_1}\right)\left(i^{*}-i\right) + \\ \dfrac{1}{R_1}\left(T-t\right) \end{bmatrix}.dNTU \tag{9}$$

$$di = -R_2.R_3\left(i^{*}-i\right).dNTU \tag{10}$$

If the values of $T_1$, $i_1$, $R_1$, $R_2$, $R_3$ are known, then from the above three equations the relation between the cooling water temperature at the outlet, $T_2$ and NTU can be derived by utilizing the relation of $t$ and $i$ and the condition $t_1 = t_2$. More details are available in Maiya[16] and Tezuka et al[17]. According to Tezuka et al., cooling efficiency of the cooling tower can be given by

$$E_A = \frac{T_1 - T_2}{T_1 - T_2^{*}} \tag{11}$$

where $T_1$ is the cooling water temperature at the inlet of the tower, $T_2$ is the cooling water temperature at the outlet and $T_2$* is the temperature of the saturated air corresponding to the enthalpy of the air at the inlet.

**RESULTS AND DISCUSSION**

The experiment was done with the standard design condition of cooling water mass flow rate of 780 kg/h, spray water mass flow rate of 1800 kg/h and air inlet-velocity of 2.75 m/s.

**Heat balance in the experimental apparatus:** The heat exchange rate of the HCCCT was estimated by establishing heat balance between the cooling water inside the coil of the heat exchanger and the air flowing outside the coil. To this end, following two equations are used.

$$Q_A = W.c_p\left(T_{Wi} - T_{Wo}\right) \tag{12}$$

$$Q_W = G.h_{a,o} - G.h_{a,i} \tag{13}$$

Here, $Q_W$ is the rate of heat loss and is shown in the horizontal axis of Fig. 4. $Q_A$ is the rate of heat gain and is shown in the vertical axis. The heat balance data those fall within $\pm 15\%$ were used.

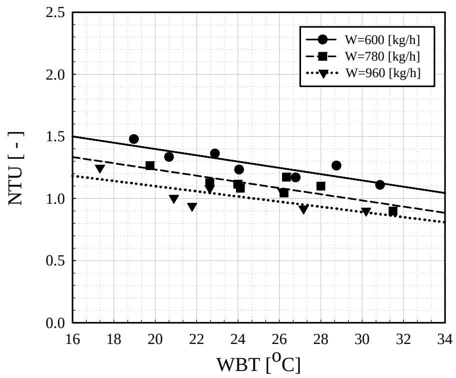


Figure 5: NTU with respect to WBT

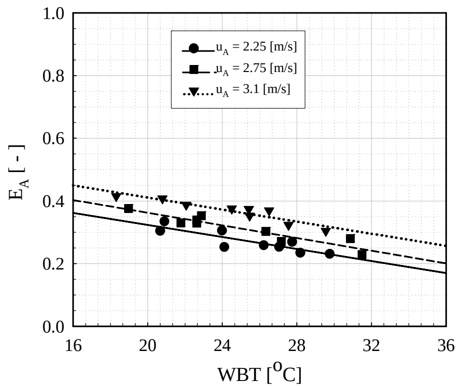


Figure 6: Cooling efficiency vs. WBT

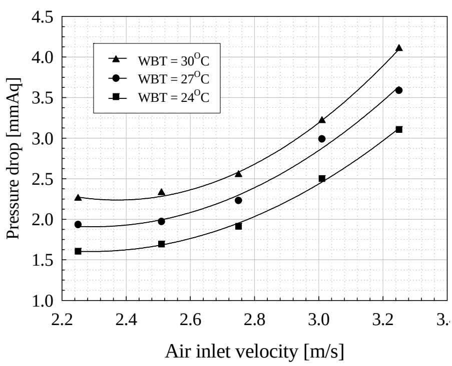


Figure 7: Pressure drop in HCCCT for wet mode

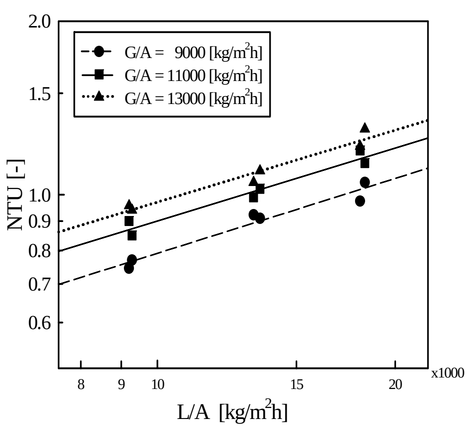


Figure 8: NTU vs. spray water mass flux

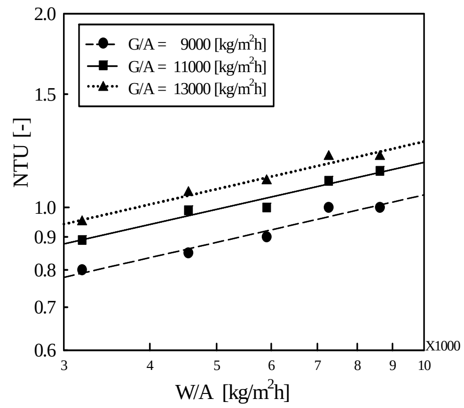


Figure 9: NTU vs. cooling water mass flux

**The NTU in wet mode operation:** The Number of Transfer Unit (NTU) as a function of the wet bulb temperature is shown in Fig. 5. NTU, which is a measure of the heat exchanger surface area, is defined in Eq. (4). The figure shows a decreasing trend of NTU as the WBT increases. This is due to the fact that overall heat transfer coefficient decreases as WBT increases, resulting in a decrease in the difference of the inlet and the outlet temperatures of the cooling water. NTU is seen to be higher with the cooling water having lower flow rate, as expected.

**Cooling efficiency by WBT:** In Fig. 6, the thermal efficiency of the HCCCT for different air flowrate has been shown against WBT. From this figure, it can be seen that the thermal efficiency decreases as WBT increases which is brought about by the temperature fall at the outlet of the heat exchanger. The thermal efficiency of the HCCCT becomes higher with increasing air flow rate. At the design condition, efficiency is 0.3358 at an air inlet velocity of 2.75 m/s.

**The pressure drop with variable airflow rate:** The pressure drop inside the heat exchanger of the HCCCT with respect to WBT having variable air flow rate is shown in Fig. 7. Pressure drop is seen to increase with the increase of both the air flow rate and the WBT. For the standard design condition, i.e., at an air flow of 2.75 m/s and at WBT = 27°C, static pressure drop found was to be 2.2 $mmH_2O$ [1 $mmH_2O$= 9.8 Pa]. Pressure drop is augmented due to spray water.

**NTU with respect to spray water and cooling water mass fluxes:** Figures 8 and 9 show the NTU profile corresponding to the spray water and the cooling water mass fluxes having variable air mass fluxes. NTU is seen to increase almost linearly with the corresponding increase in the mass fluxes of both the spray water and the cooling water. This is simply because if the mass flow rates of spray and cooling water increases, the heat transfer coefficients increases both inside and outside of the pipe and thus brings about an increase in NTU for each case.

**The cooling capacity in wet mode against WBT:** The cooling capacity in wet mode operation of the HCCCT at various WBT having different cooling water flow rate is shown in Fig 10. The heat exchanger's common feature is that the temperature range decreases with both the increases of the WBT and the cooling water flow rate and vice versa and is seen from this figure as well. It is evident from Fig. 10 that cooling capacity increases with the increase of the cooling water flow rate but decreases with the increase of WBT as well as by the decrease of the cooling water flow rate. The capacity at a WBT of 27°C was about 3000 kcal/h,

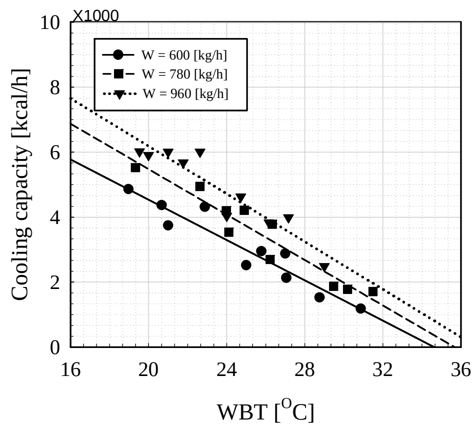


Figure 10: Cooling capacity in wet mode

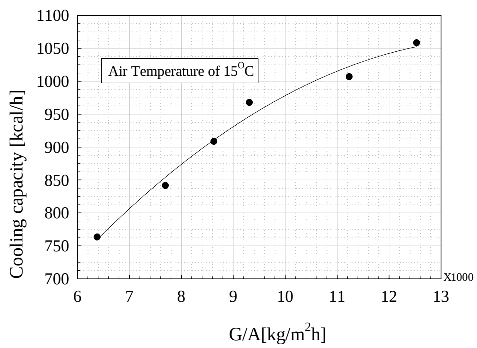


Figure 11: Cooling capacity by air mass flux

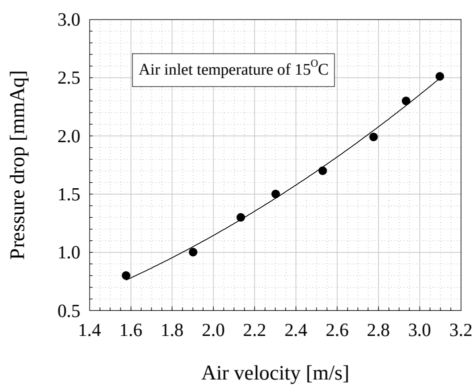


Figure 12: Pressure drop of air in dry mode.

which is 23% lower than the rated one. The lower cooling water mass flow rate of 780 kg/h could be a reason for this.

**The dry mode operation in winter season:** In all cooling towers reported so far, water is sprayed even in winter to get the desired cooling effect, but in that case, heating coil is installed in the spray water collecting tank to prevent ice formation. Therefore, the cost for keeping the provision of installing heating coil as well as the cost incurred by the energy consumption increases. The most economic operation is one of the primary objectives of this study, so it is expected that, the HCCCT will remain in operation even in winter utilizing the cold air only so that the operating cost can be minimized. Under these circumstances, performance of the HCCCT during the dry mode operation is important as well.

**Cooling capacity by variable air mass flux:** Cooling capacity of the HCCCT in dry mode with respect to a variable air mass flow rate per unit area has been shown in Fig. 11. The cooling capacity is found to increase with the increasing air flow rate. At the design condition, the cooling capacity is found to be around 1064.25 kcal/h. The capacity is a bit low especially when the design target is set to be 1RT. [1RT = 3900 kcal/h]. The reason for this poor capacity is the absence of the spray water so that the HCCCT was operated only with cold air. Even though the cooling capacity is poor, the cost incurred is lower.

**Pressure drop in dry mode operation:** Since no spray water was used in dry mode operation of the HCCCT, pressure drop is expected to be lower than in wet mode operation. Pressure drop with respect to air inlet velocity is shown in Fig. 12 and is seen to increase almost linearly with the increase of the air velocity and at an air velocity of 2.75 m/s. The pressure drop is found to be around 1.9 $mmH_2O$. The lower pressure drop means that the dry mode operation of the HCCCT will consume less energy and thus can reduce the operational cost considerably.

## CONCLUSION

The heat exchangers have been studied experimentally to identify the optimum design parameters at the level of the best performance of a HCCCT having a rated capacity of 1RT. In the standard designed condition, i.e., at cooling water having mass flow rate of 780 kg/h and at a WBT of 27°C, the thermal efficiency E was found to be 0.3358 for the air flow rate of 2.75 m/s. At an inlet air velocity of 2.75 m/s and WBT = 27°C, static pressure drop is measured to be 2.2 $mmH_2O$. Experimental study reveals that, for most cases in wet mode operation, a wet-bulb temperature of 27°C with a cooling water mass flow rate of 780 kg/h having an air velocity of about 2.75 m/s produced reasonable results especially with respect to the typical East-Asian meteorological constrains. The cooling capacity in wet mode operation was about 0.8RT which was within 23% of the rated capacity. In dry mode operation in the cooling capacity was about 0.3RT, it is evident that this poor capacity is mainly due to the absence of spray water and the HCCCT can be utilized in dry mode when the cooling requirement is lower and the economic operation is preferred. Results obtained from this study are supposed to provide experimental data which could be referred for the optimum design of the heat exchanger in a HCCCT.

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